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The Suction Air Cooling Gas Turbine with Vapor Compression Refrigerator*

(3rd Report, On the Preformance at the Partial Load)

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The authors analyze the performance at the partial load of the suction air cooling gas turbine cycle with a vapor compression refrigerator. The partial load is adjusted by changing the inlet gas temperature of the turbine, keeping the number of revolutions of the turbine and the flow rate of the inlet air of the air compressor in a constant state. The analytic results show that the gains in thermal efficiency and specific power at the partial load are in the same degree as at the design point, and that this cycle can suppress a variation of the performance caused by a change in atmospheric temperature at the partial load as well as at the design point, compared with conventional cycle.

1. Introduction

At present, the development of technologies concerning energy saving is an important social subject of research awaiting solution. As one of various measures for energy saving and at the same time taking into consideration the siting difficulties of a large thermal power plant, adoption of an on-site small comprehensive plant system, of which the main prime mover is gas turbine etc., has been proposed and in parallel with this investigations have been carried out from the standpoint of environmental protection and economy. In spite of its many advantages, the gas turbine considered one of the main prime movers for such a total energy system has only a thermal efficiency inferior to those of steam prime mover and reciprocating internal combustion engine and the variation of its performance is disadvantageously remarkable under a change in the atmospheric temperature. Thus the cycle of a gas turbine must be imporved to be used as a main prime mover in the above-mentioned system.

As one of methods for reducing such disadvantages of gas turbine, the authors have proposed a cycle of cooling the suction air with a vapor compression refrigerator. In the perceding 1st[1] and 2nd[2] reports, the general characteristics and the optimum point of the cycle were thermodynamically investigated, respectively, and it was demonstrated that in comparison with any conventional cycle without suction air cooling, this cycle can improve considerably both the thermal efficiency and the specific power and further mitigate any performance variation caused by a change in the atmospheric temperature. From the standpoint of planning and utilizing a cycle like this, on the other hand, it is also important to investigate, besides the performance at design point, the variable-load, that is, partial load performance.

The authors have here investigated the partial load performance of a suction air cooling gas turbine cycle with a vapor compression refrigerator and confirmed that according to this cycle, it is possible even under partial load to obtain gains of the same order as those in thermal efficiency and specific power at the design point in comparison with a usual uniaxial regenerating open cycle, it is possible to mitigate the performance variation caused by a change in the atmospheric temperature and there is no remarkable difference of the general tendency under partial load from that in a conventional cycle, where the suction air is not cooled. These results are described in the following.

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Nomenclature

η_C : adiabatic efficiency of air compressor
η_T : adiabatic efficiency of gas turbine
η_{CR} : adiabatic efficiency of refrigerant compressor
η_m : mechanical efficiency
η_B : combustion efficiency
η_{EX} : temperature-efficiency of regenerator
η_R : temperature-efficiency of cooler
η : thermal efficiency
L : power, kg m/s
L_s : specific power, PS/kg/s
Q : quantity of heat per unit time, kcal/s
q : quantity of heat per unit mass per unit time, kcal/kg/s

J : mechanical equivalent of heat, kg m/kcal
A : thermal equivalent of work, kcal/kg m
h : specific enthalpy, kcal/kg
s : specific entropy, kcal/kg °K
c_p : specific heat at constant pressure, kcal/kg °K
κ : ratio of specific heats at constant pressure and volume
m : molecular weight, kg/kmol
K : coefficient of overall heat transmission, kcal/m^2 h °C
α : heat transfer coefficient, kcal/m^2 h °C
ε : coefficient of performance
G : flow rate of working fluid at gas turbine side, kg/s
G_R : flow rate of refrigerant, kg/s
R_e : Reynolds number
P_r : Prandtl number
β : loss rate of working fulid
P : pressure, kg/cm^2 abs
ϕ : pressure ratio
ε_l : loss rate of pressure
T : temperature, °K
t : temperature, °C
$\mathfrak{R}$: universal gas constant, kg m/kmol °K
R_a : gas constant of air, kg m/kg °K
V : evaluation function for cycle Eq.(35)
I_f : factor of V
f : fuel-air ratio

Subscripts

C : air compressor
CC : combustion chamber
T : turbine
EX : regenerator
R : refrigerant
a : air
g : gas
CR : refrigerant compressor
E : evaporator
F : condenser
SA : with suction air cooling (suction air cooling cycle)
CON : without suction air cooling (conventional cycle)
D : design point
P : partial load
opt : optimum point
ad : adiabatic change
st : saturation state
Ref : reference value
$\overline{ab}$: mean value between a and b
$*$: ratio to the value at the design point

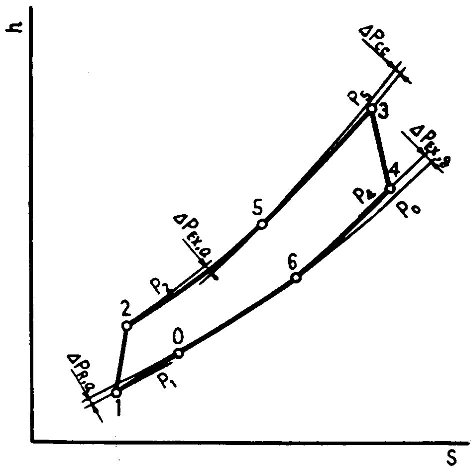

Fig.1 h-s diagram of the gas turbine cycle with suction air cooling

2. Constitution of the cycle

The authors here utilize a cycle arrangement, similar to that in the 1st report. This cycle is a combination of a uniaxially open gas turbine (1/C/E) with a vapor compression refrigerator (refrigerant R-22). Fig. 1 shows an h-s diagram of the gas turbine and Fig. 2 a P-h diagram of the cooling system. Suction air to the air compressor is cooled from the atmospheric temperature t_0 to t_1 by an evaporator based on the direct expansion system, that is, a suction air cooler. Load regulation will be carried out by changing the inlet gas temperature of the turbine t_3, the rotation frequency of turbine and the air flow rate at the inlet of the air compressor being kept constant. In Figs. 1 and 2, the number of each part refers to the following :

0 : the atmospheric condition
1 : inlet of the air compressor
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
1' : inlet of the expansion valve
2' : inlet of the suction air cooler
3st' : outlet of the suction air cooler
3' : inlet of the refrigerant compressor
4' : inlet of the condenser

3. Fundamental equations of the cycle

Fundamental equations are laid down under the following conditions of cycle formation :

(1) The specific heat at constant pressure and the ratio of specific heats of working air and combustion gas depend on the temperature and the composition[6].

(2) The rotation frequencies of turbine and compressor are constant.

(3) The inlet air flow rate and the adiabatic efficiency of the air compressor are constant.

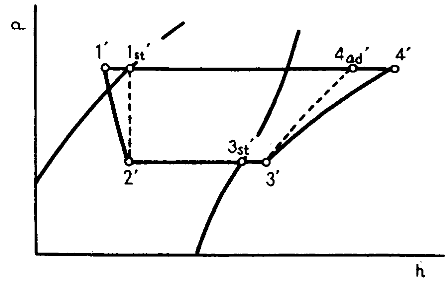

Fig.2 P-h diagram of the cooling system

(4) Various quantities concerning the turbine follow the ellipse law [4].

(5) The refrigerant compressor used is a turbo compressor, of which adiabatic efficiency depends on the compression ratio and refrigerant flow rate.

(6) The combustion and mechanical efficiencies are constant.

(7) The loss rate of pressure and flow rate in each section of the cycle is constant and heat loss of the system is only in exhaust gas.

(8) The heat-transmitting areas of the regenerator and the suction air cooler are constant.

(9) As refrigerant, R-22 [3] is used, it is superheated by 5°C at the start of its compression and subcooled in front of the expansion valve and the enthalpy at the starting point of evaporation is equal to the value at the wet saturation point on the condensation side.

(10) The temperature of refrigerant condensation depends on that of the atmosphere (of the cooling water).

3.1 Fundamental equations concerning design points

A representative pressure ratio ϕ on the gas turbine side is defined as follows :

$$\phi=P_2/P_0 \quad \cdots\cdots (1)$$

When the pressure loss of working fluid is included, the compression ratio of air compressor ϕ_C and the expansion ratio of turbine ϕ_T can be written as follows :

$$\phi_C=\frac{P_2}{P_1}=\frac{1}{(1-\varepsilon_{l,R,a})}\phi \quad \cdots\cdots (2)$$

$$\phi_T=\frac{P_3}{P_4}=\frac{(1-\varepsilon_{l,EX,a}-\varepsilon_{l,CC})}{(1+\varepsilon_{l,EX,g})}\phi \quad \cdots\cdots (3)$$

where ε_l is the loss rate of pressure, which is given, from the pressure loss ΔP shown in Fig. 1, by the following equations :

$$\left.\begin{array}{ll}\varepsilon_{l,R,a}=(P_0-P_1)/P_0=\Delta P_{R,a}/P_0, & \varepsilon_{l,EX,a}=(P_2-P_5)/P_2=\Delta P_{EX,a}/P_2 \\ \varepsilon_{l,CC}=(P_5-P_3)/P_2=\Delta P_{CC}/P_2, & \varepsilon_{l,EX,g}=(P_4-P_6)/P_0=\Delta P_{EX,g}/P_0\end{array}\right\} \quad \cdots\cdots (4)$$

The flow rates in each section on the gas turbine side G and of the refrigerant compressor G_{CR} are defined as follows :

$$\left.\begin{array}{l}G_C=G_a(1-\beta_C/2),\quad G_{EX,a}=G_a(1-\beta_C),\quad G_{CC}=G_a(1-\beta_C)(1+f) \\ G_T=G_a(1-\beta_C)(1-\beta_T/2)(1+f),\quad G_{EX,g}=G_a(1-\beta_C)(1-\beta_T)(1+f) \\ G_{CR}=G_R(1-\beta_{CR}/2)\end{array}\right\} \quad \cdots\cdots (5)$$

where G_a and G_R are the flow rates at the inlet of air and refrigerant compressors, β is the loss rate of flow given by the following equations and f the fuel-air ratio :

$$\beta_C/2=(G_a-G_C)/G_a,\quad \beta_T/2=(G_{CC}-G_T)/G_{CC},\quad \beta_{CR}/2=(G_R-G_{CR})/G_R \quad \cdots\cdots (6)$$

The temperature at each section on the gas turbine side can be determined by each of the following equations :

$$T_2=T_1\{1+(\phi_C{}^{M_{12}}-1)/\eta_C\} \quad \cdots\cdots (7)$$

$$T_4=T_3\{1-(1-1/\phi_T{}^{M_{34}})\eta_T\} \quad \cdots\cdots (8)$$

$$T_5=T_2+(T_4-T_2)\eta_{EX} \quad \cdots\cdots (9)$$

$$T_6=-\frac{G_{EX,a}c_{p\overline{25}}}{G_{EX,g}c_{p\overline{46}}}(T_5-T_2)+T_4 \quad \cdots\cdots (10)$$

where M is given by either of the following two equations, in which the suffix, for example, $\overline{12}$ means a mean value between the positions 1 and 2 :

$$M_{12}=AR_a/c_{p\overline{12}},\qquad M_{34}=A\mathfrak{R}/m_{\overline{34}}c_{p\overline{34}} \quad \cdots\cdots (11)$$

Based on these flow rates and temperatures, the power and the quantity of heat in each element of gas turbine can be obtained from the following equations :

Power of air compressor : L_C

$$L_C=JG_Cc_{p\overline{12}}(T_2-T_1)/\eta_m \quad \cdots\cdots (12)$$

Output of gas turbine : L_T

$$L_T=JG_Tc_{p\overline{34}}(T_3-T_4)\eta_m \quad \cdots\cdots (13)$$

Quantity of heat for heating : Q_{CC}

$$Q_{CC}=G_{CC}c_{p\overline{53}}(T_3-T_5)/\eta_B \quad \cdots\cdots (14)$$

Assume the following relations between the temperature of cooling water t_w and that of the atmosphere t_0 as well as between the temperature of refrigerant condensation t_F and that of cooling water and assume these relations hold even under partial load :

$$t_w=\frac{1}{3}t_0+5,\quad t_F=t_w+10 \quad \cdots\cdots (15)$$

The evaporation temperature T_E can be determined from the following equation :

$$T_E=T_0-(T_0-T_1)/\eta_R \quad \cdots\cdots (16)$$

Based on Fig. 2, the refrigerating capacity q_R and the coefficient of performance ε can be expressed as follows :

$$q_R=i_{3st}'-i_2'=i_{3st}'-i_{1st}'=(G_a/G_R)c_{p\overline{01}}(T_0-T_1) \quad \cdots\cdots (17)$$

$$\varepsilon=q_R/(i_4'-i_3')=q_R/\{(i_{4ad}'-i_3')/\eta_{CR}\} \quad \cdots\cdots (18)$$

The power of refrigerant compressor L_{CR} is given by the following equation :

$$L_{CR}=JG_Rq_R(1/\varepsilon)/\eta_m \quad \cdots\cdots (19)$$

From the above-mentioned various equations, the specific power L_S and the thermal efficiency η of the suction air cooling cycle can be calculated by the following equations:

$$L_S=(L_T-L_C-L_{CR})/(75G_a) \quad \cdots\cdots (20)$$

$$\eta=A(L_T-L_C-L_{CR})/Q_{CC} \quad \cdots\cdots (21)$$

Determine the heat-transmitting area of heat exchanger. On the assumption that it is a counterflow-type regenerator, its heat-transmitting area A_{EX} is given by the following equation :

$$A_{EX}=\frac{G_{EX,a}c_{p\overline{25}}(T_5-T_2)}{K_{EX}\Delta t_{m,EX}} \quad \cdots\cdots (22)$$

where K and Δt_m are the coefficient of overall heat transmission and the logarithmic mean temperature difference, respectively, and given by the following equations :

$$K_{EX}=1/(1/\alpha_{EX,a}+1/\alpha_g) \quad \cdots\cdots (23)$$

$$\Delta t_{m,EX}=\{(T_4-T_5)-(T_6-T_2)\}/\log_e\{(T_4-T_5)/(T_6-T_2)\} \quad \cdots\cdots (24)$$

When the suction air cooler is also counterflow-type, its heat-transmitting area A_R, its coefficient of overall heat-transmitting area K_R and its logarithmic mean temperature difference $\Delta t_{m,R}$ can be expressed as follows :

$$A_R=\frac{G_ac_{p\overline{01}}(T_0-T_1)}{K_R\Delta t_{m,R}} \quad \cdots\cdots (25)$$

$$K_R=1/(1/\alpha_{R,a}+1/\alpha_R) \quad \cdots\cdots (26)$$

$$\Delta t_{m,R}=(T_0-T_1)/\log_e\{(T_0-T_E)/(T_1-T_E)\} \quad \cdots\cdots (27)$$

3.2 Characteristic equation under partial load

Based on the ellipse law, the expansion ratio ϕ_T and the adiabatic efficiency η_T of gas turbine can be written as follows :

$$\phi_{T,P}=\sqrt{T_{3,D}/\{T_{3,D}-T_{3,P}(1-1/\phi_{T,D}^2)\}} \quad \cdots\cdots(28)$$

$$\eta_{T,P}=\eta_{T,D}\{1-0.5(\sqrt{AL_{T,D}/AL_{T,P}}-1)^2\} \quad \cdots\cdots(29)$$

Assume that the adiabatic efficiency of refrigerant compressor η_{CR} follows the following equation, obtained by a functional approximation of the characteristic curve [5] :

$$\eta_{CR,P}=\eta_{CR,D}\frac{\{1-(R_R-1)^2\}}{0.4(3/R_R+2)}\left\{\frac{3}{2}(1/R_R+1)-R_N\right\}\sqrt{R_N} \quad \cdots\cdots(30)$$

where

$$R_R=G_{CR,P}/G_{CR,D}, \quad R_N=R_R^{0.5}/(\phi_{R,P}/\phi_{R,D})^{0.75} \quad \cdots\cdots(31)$$

In the above-mentioned equation (31), ϕ_R is the comprssion ratio of refrigerant compressor and can be expressed as follows :

$$\phi_R=P_4'/P_3' \quad \cdots\cdots(32)$$

Using McAdams' equation concerning forced convection and taking into consideration that the fuel-air ratio of working fluid is low, the authors assume that the heat transfer coefficients $\alpha_{EX,a}$, $\alpha_{EX,g}$ and $\alpha_{R,a}$ on the air and the gas sides of regenerator and on the air side of suction air cooler, respectively, are expressed by the following equations :

$$\left.\begin{aligned}\alpha_{EX,a,P}&=\alpha_{EX,a,D}(R_{e,P}/R_{e,D})^{0.8}(P_{r,P}/P_{r,D})^{0.4}\\&\fallingdotseq\alpha_{EX,a,D}\{(T_5+T_2)_P/(T_5+T_2)_D\}^{0.8}\\\alpha_{EX,g,P}&\fallingdotseq\alpha_{EX,g,D}\{(T_4+T_6)_P/(T_4+T_6)_D\}^{0.8}\\\alpha_{R,a,P}&\fallingdotseq\alpha_{R,a,D}\{(T_0+T_1)_P/(T_0+T_1)_D\}^{0.8}\end{aligned}\right\} \quad \cdots\cdots(33)$$

Assume that the relation [5] of the heat transfer coefficient α_R on the refrigerant side of suction air cooler with the heat load of refrigeration q_R is given by the following equation :

$$\alpha_{R,P}=\alpha_{R,D}(q_{R,P}/q_{R,D})^{1.122} \quad \cdots\cdots(34)$$

The influence on the behavior under partial load may be represented here with $\alpha_{R,a,P}$, because the scale factor due to frosting on suction air cooler depends remarkably on the time and operation system and the object of the present research consists in investigating the change in performance of the cooler under partial load.

The temperatures of air and of gas at the outlet of regenerator are taken to be T_5 and T_6 satisfying Eq.(22) for T_2 and T_4 under partial load, respectively, and the evaporation temperature of refrigerant to be t_E satisfying Eq.(25) for T_0 and T_1 under partial load. If the values of ϕ_T, η_T, η_{CR}, T_5 and T_6 obtained from the above-mentioned are substituted into the previously derived equations for design point on the assumption that the turbine inlet temperature is $T_{3,P}$ then various quatities under partial load can be determined. By substituting $T_1=T_0$ and $\varepsilon_{lR,a}=0$ into the above-mentioned equations, it is possible to obtain the performance values of cycle without suction air cooling.

Although the above-mentioned equations were introduced with the phenomena simplified and approximated in order to moderate the influence of form and dimensional effect of the equipment, analysis was numerically carried out because of the existence of non-linear elements in pressure and temperature, and solutions were obtained when the relative errors between two consecutive iterations for T_2, T_4, f, η_T and η_{CR} and between respective values and those at the design points of A_R and A_{EX} for T_5, T_6 and T_E were less than 10^{-6}.

4. Calculation results and discussion

4.1 Establishment of design point

Table 1 shows various values used for determining the design points. The pressure ratios and the air temperatures at the outlet of suction air cooler, which give the maximum values of thermal efficiency and specific power, are, as shown up to the preceding reports, different in general. For this reason, the following evaluation function V is here defined and the pressure ratio ϕ and the outlet temperature of suction air cooler t_1 for the maximum value of V are taken as design values :

$$V=\left[\left(\frac{\eta-\eta_{\mathrm{Ref}}}{\eta_{\mathrm{Ref}}}\right)+\left(\frac{L_S-L_{S,\mathrm{Ref}}}{L_{S,\mathrm{Ref}}}\right)\Big/I_f\right]\Big/(1+1/I_f) \quad \cdots\cdots(35)$$

4.1.1 Without suction air cooling

The maxima of thermal efficiency and specific power are 32.21% at $\phi=3.6$ and 148.0 PS/kg/s at $\phi=6.0$, respectively. With these values written as $\eta_{\mathrm{Ref},D}$ and $L_{S,\mathrm{Ref},D}$, the thermal efficiency is considered to be more important than the specific power and the value of V to be obtained when $I_f=2$ is substituted into Eq.(35) is written as $V_{D,\mathrm{CON}}$, which is shown in Fig. 3. Based on this results, the design pressure ratio without suction air cooling is determined to be 4.4 .

Table 1 Set values and performance values at the design point

adiabatic efficiency of air compressor	$\eta_C=0.90$
adiabatic efficiency of refrigerant compressor	$\eta_{CR}=0.90$
adiabatic efficiency of turbine	$\eta_T=0.81$
mechanical efficiency	$\eta_m=0.99$
combustion efficiency	$\eta_B=0.99$
temperature-efficiency of regenerator	$\eta_{EX}=0.85$
temperature-efficiency of cooler	$\eta_R=0.85$
temperature at the inlet of the turbine °C	$t_3=750$
atmospheric temperature °C	$t_0=15$
loss rate of pressure	$\varepsilon_{l,R,a}=0.03$
	$\varepsilon_{l,EX,a}=0.03$
	$\varepsilon_{l,EX,g}=0.03$
	$\varepsilon_{l,CC}=0.02$
loss rate of working fluid	$\beta_C=0.03$
	$\beta_T=0.03$
	$\beta_{CR}=0.03$
component of fuel kg/kg	$c=0.85$
	$h=0.15$
heat transfer coefficient kcal/m²h°C	$\alpha_{EX,a}=10^2$
	$\alpha_{EX,g}=10^2$
	$\alpha_{R,a}=10^2$
	$\alpha_R=10^3$

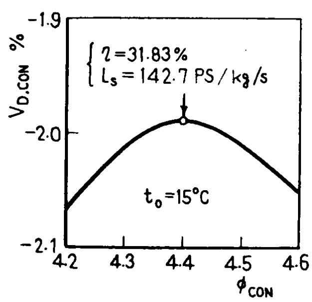

Fig.3 $V_{D,\mathrm{CON}}$ **near the design point**

4.1.2 With suction air cooling

With η_{Ref} and $L_{S,Ref}$ similar to those in the preceding section, the pressure ratio and the outlet tmeperature of suction air cooler can be determined at the design point, provided that taking into consideration the complication of the equipment due to suction air cooling, I_f in Eq.(35) is taken to be equal to 4 . The result is shown in Fig. 4, where the value of $V_{D,SA}$ is the highset at ϕ= 5.1 and t_1=−22°C so that these values are adopted as the pressure ratio and the outlet temperature of suction air cooler at the design point.

4.2 Partial load performance

Based on the above, the values eatablished as shown in Tables 1 and 2 are used to investigated the partial load performance. $\eta_{Ref,P}$ and $L_{S,Ref,P}$ are the values at the design point of a conventional cycle without suction air cooling in order to compare and investigate the performance of the suction air cooling cycle with those of the conventional as reference.

4.2.1 Change of performance values due to t_1 near the design point

The air temperature t_1 at the outlet of suction air cooler (the inlet of air compressor) may be easily affected by the temperature and humidity on the air side as well as the operating condition on the cooling side. If a little change in t_1 makes the performance values of the cycle remarkably shift, the actual operation can be disturbed in more cases. Figure 5 shows the influences of a change in t_1 on the performance values. As can be seen from these diagrams, η gradually decreases and V_P remarkably changes as t_1 becomes lower than the design temperature, −22°C, whereas η rises to some degree or becomes flat and V_P changes more slowly as t_1 exceeds the design temperature. In general, however, the variation width of V_P as the result of a change in t_1 is narrower within the range shown in Fig. 5 so that it is possible to expect that the operation is easy and the performance values near the design point can be obtained even at t_1 , higher by 2 to 3°C than the design temperature.

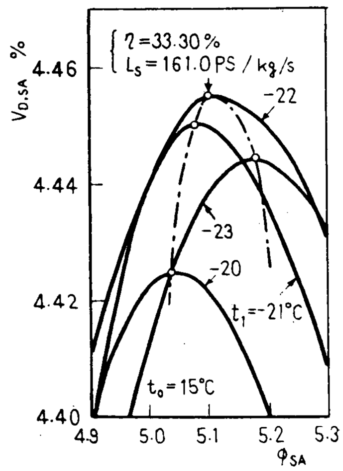

Fig.4 $V_{D,SA}$ near the design point

Table 2 Set values at the partial load

reference values for calculation of V_P	$\eta_{Ref,P}$=31.83% $L_{S,Ref,P}$=142.7 PS/kg/s
without suction air cooling pressure ratio factor of V_P	ϕ_{CON}=4.4 I_f=2
with suction air cooling pressure ratio outlet temperature of suction air cooler factor of V_P	ϕ_{SA}=5.1 $t_{1,SA}$=−22°C (at t_0=15°C) I_f=4

4.2.2 Optimum outlet temperature of suction air cooler

The optimum outlet temperature of suction air cooler $t_{1,opt}$ follows any change in the atmospheric temperature t_0. When t_0 changes, that is, there exists a value of $t_{1,opt}$, which gives the highest V_P in the same way as at the disign point also under partial load. This $t_{1,opt}$ is hardly affected by various parts, other than the cooling system. The result is shown in Fig. 6, where the difference between $t_{1,opt}$ and t_0 becomes larger with an increase of the atmospheric temperature.

4.2.3 Turbine inlet temperature

Figure 7 shows a relationship between the turbine inlet temperature t_3 and the load factor $L_{S,P}/L_{S,D}$. Regardless of suction air cooling, the tendencies as a whole coincide with each other. In general, however, t_3

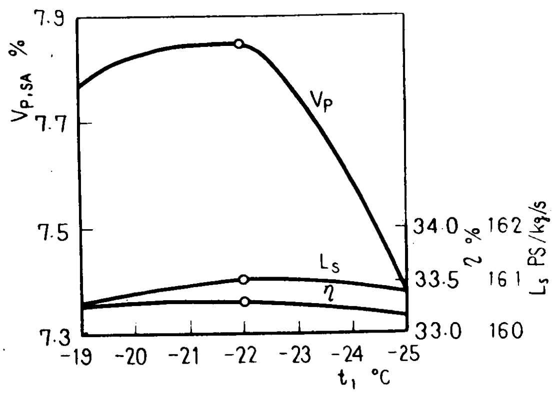

Fig.5 The influence of a change in t_1 on the values of the thermal efficiency, specific power and $V_{P,SA}$ near the design point

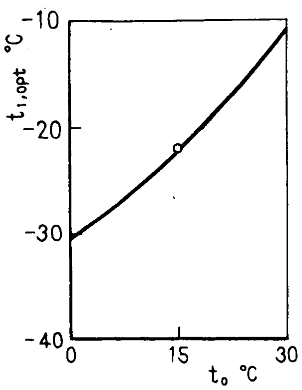

Fig.6 The optimum outlet temperature of suction air cooler

with suction air cooling is higher slightly than that without one under partial load and t_3 with suction air cooling does not so remarkably depend on the atmospheric temperature as that without one.

4.2.4 Thermal efficiency

Figure 8 shows the thermal efficiency as a function of the atmospheric temperature and the load factor. Regardless of suction air cooling, the partial load characteristics of the thermal efficiency have similar tendencies. The dependence of the thermal efficiency on the atmospheric temperature is moderated by suction air cooling and this effect of suction air cooling becomes more remarkable particularly with an increase of the atmospheric temperature. In Fig. 8, the thermal efficiencies are compared, regardless of suction air cooling, under the same power and the suction air cooling decreases the flow rate of working air by about 9.2%.

4.2.5 V_P

As mentioned above, suction air cooling makes different the meaning of specific power. Consequently, the authors here trially evaluate comprehensively the thermal efficiency and the specific power with V defined in Eq.(35). In case this evaluation function V is used as V_P under the established values in Table 2, it is possible to consider this function to be an index which indicates the coefficient of utilization and the performance values of a plant on the basis of the design point of a conventional cycle. Fig. 9 shows V_P on the partial load under the same flow rate of air. Since at the design point, t_3=750°C, the increases of $L_{S,P}/L_{S,CON,D}$ and V_P as a result of suction air cooling become more remarkable particularly with an increase of the atmospheric temperature and this tendency is maintained even under partial load, it appears that the usefulness of the cycle has been confirmed.

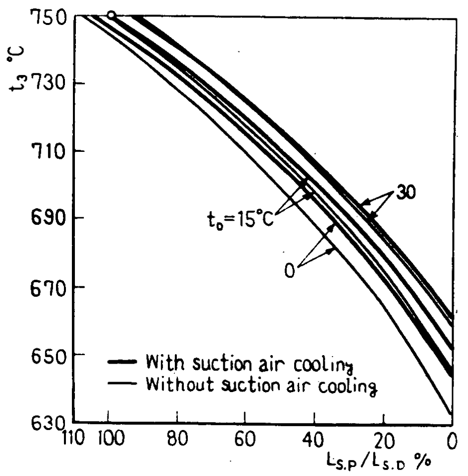

Fig.7 The turbine inlet temperature under partial load

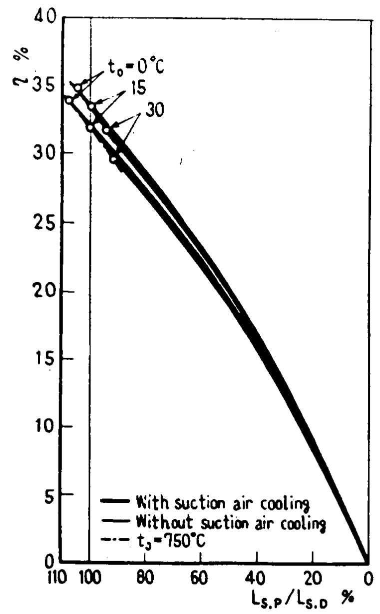

Fig.8 The thermal efficiency under partial load

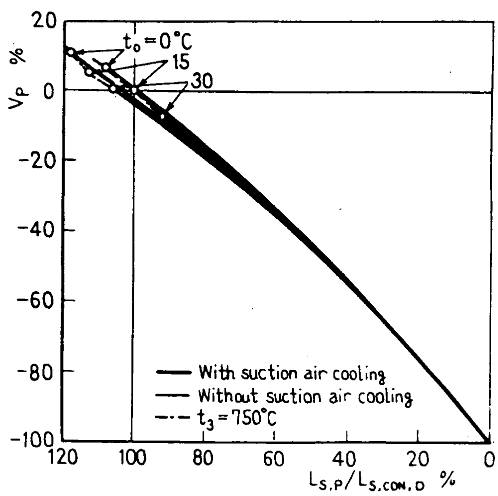

Fig.9 V_P under partial load

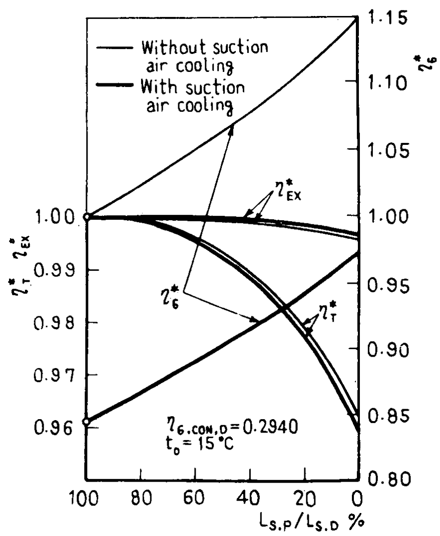

Fig.10 The characteristic values under partial load

4.2.6 Other characteristic values

Under partial load at the atmospheric temperature 15°C , Fig. 10 shows the temperature-efficiency of regenerator η_{EX} , the adiabatic efficiency of turbine η_T and an index η_6 , which represents the exhaust gas temperature T_6 and is defined by the following equation, in the form of a ratio ($\eta^*=\eta_P/\eta_D$) to the characteristic value at the design point :

$$\eta_6=(T_6-T_0)/(T_5-T_0) \quad \cdots\cdots(36)$$

Under partial load, η_{EX}^* does hardly change, whereas η_T^* does remarkably. However, either of them does depend only slightly on suction air cooling. On the other hand, η_6 changes remarkably under partial load and depends also considerably on suction air cooling.

In a similar manner of representation in Fig. 10, Fig. 11 shows the dependences, on the atmospheric temperature, of the temperature-efficiency of suction air cooler η_R , the adiabatic efficiency of refrigerant compressor η_{CR} , the compression ratio of the same ϕ_R^* and the refrigerant flow rate G_R^*. η_{EX}^* changes gradually, whereas the remaining characteristic values do remarkably. In particular, η_{CR}^* depends very remarlably on G_R^* and ϕ_R^*.

5. Conclusions

The authors investigated the performance of a suction air cooling gas turbine cycle with a vapor compression refrigerator and in succession to the investigations on the general characteristic performance, the 1st report and on the optimum point, the 2nd report, they carried out an analysis of partial load performance in this report. Based on this investigation carried out on the assumption that the load is regulated by adjusting the inlet gas temperature of gas turbine, its number of revolutions and the air flow rate at the inlet of air compressor being kept constant, the authors obtained the following conclusions besides those already described :

(1) The partial load characteristics of the thermal efficiency has a tendency apporximately similar to that without suction air cooling.

(2) Even under partial load, suction air cooling improves the efficiency in the same degree as that at the design point.

(3) Suction air cooling does not so remarkably modify the dependence of turbine inlet temperature on the load factor.

(4) Also under partial load, suction air cooling considerably moderates the dependence of performance values on the atmospheric temperature in the same manner as at the design point.

(5) Slight variation of temperature at the outlet of suction air cooler (the inlet of air compressor) does not so remarkably affect the performance of the cycle as a whole.

Since even under partial load the present cycle can improve the performance of a conventional and it is possible to reutilize the high temperature exhaust gas, it appears that this cycle can be effectively applied to various stationary prime movers such as main one in an on-site comprehensive plant etc.

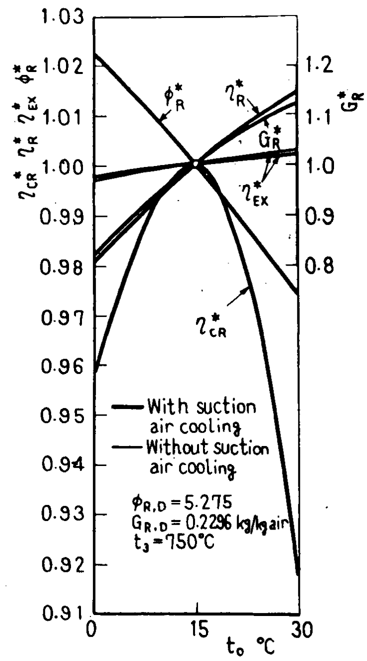

Fig.11 The dependences of the characteristic value on the atmospheric temperature t_0

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