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Thermodynamic Study on the Suction Cooling Gas Turbine Cycle Combined with Absorption-Type Refrigerating Machine Using Wasted Heat*

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Gas turbine, generally, has comparatively low value of thermal efficiency and specific power, and the variations of these values are considerably large when the ambient temperature is varied.

This paper is intended to present a solution to this problem by adopting a new cycle concept. To increase the temperature ratio of the cycle is one of the most effective ways to improve thermal efficiency and specific power.

In this case, the low-temperature absorption-type refrigerating machine, which is operated by using the gas turbine wasted heat as a heating source to the generator, is introduced as one of the cooling devices to lower the intake air temperature to the compressor.

The primary goal of our study is to investigate the general thermodynamic characteristics of this combined cycle.

It has been made clear that the new cycle can achieve remarkably higher thermal efficiency and specific power than the gas turbine running alone. Furthermore, the cycle has a desirable merit of lessening the variation in these values, when the ambient temperature increases.

1. Introduction

To increase the temperature ratio of the cycle is one of the most effective ways for improving thermal efficiency and specific power of the gas turbine. On the other hand, there is a recent tendency to increase the thermal efficiency of the plant system as a whole by using the wasted heat of the gas turbine exhaust effectively. In this study, as a method for increasing the temperature ratio of the cycle, the low-temperature absorption-type refrigerating machine, which is operated by using the gas turbine wasted heat as a heating source to the generator, is introduced as one of the cooling devices in the suction cooling gas turbine cycles to lower the intake air temperature to the compressor and the analysis of this combined cycle has been performed thermodynamically for the design point performance⁽¹⁾.

The aim of this paper is to give the solutions for the demerits of the gas turbine, that is; it gives fairly low values of thermal efficiency and specific

power, also the variations of thermal efficiency and specific power are considerably large when the ambient temperature varies.

As was expected, the thermal efficiency and specific power are highly improved by this combined cycle and especially the cycle has a desirable merit of lessening the variation of these values, when the ambient temperature varies.

2. Composition of the suction cooling gas turbine cycle combined with absorption-type refrigerating machine using the wasted heat

Here, we consider the general mode of operation of the combined cycle. The cycle treated is the conventional regenerative gas turbine cycle (1/C/E) combined with low-temperature ammonia absorption-type refrigerating machine, and its complete combined arrangement of the cycle, in its simplest form, appears as shown diagrammatically in Fig. 1. Absorption-type refrigerating machine is usually operated by mainly using steam or hot water as a heating source to generator (G), but, in this cycle, the remaining high temperature turbine exhaust heat available after being partially recovered in regenerator (R), is fed to generator.

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Thus, an intake air passing through the suction cooler (W) to compressor (K) being cooled, its temperature (T_1) falls from the ambient temperature (T_0) to nearly the suction cooler inlet temperature of the brine (T_8').

3. Basic formula of the cycle

3.1 Symbols

The basic formulae are derived by referring to Fig. 1 and Fig. 2 considering the variation of the specific heat of the working fluid, when its temperature and compositions vary. The pressure losses and leakage losses are also taken into account.

η_K : compressor adiabatic efficiency

η_T : turbine adiabatic efficiency

η_R : effectiveness of regenerator

η_B : combustion efficiency

η_m : mechanical efficiency

η_w : temperature efficiency of the suction cooler

η_a : thermal efficiency

L : power output (PS)

L_s : specific power (PS/kg/sec)

Q : heat quantity (kcal)

q : heat quantity (kcal/kg)

ϕ : pressure ratio

ϕ_K : compressor pressure ratio

ϕ_T : turbine expansion ratio

ε : pressure loss rate

β : leakage loss rate

A : thermal equivalent of mechanical work (kcal/kg-m)

f : fuel air ratio

R : gas constant (kg-m/kg $^{\circ}$ K)

H_u : lower calorific value (kcal/kg)

θ : suction temperature decreasing ratio

τ : temperature ratio

c_p : constant pressure specific heat (kcal/kg $^{\circ}$ K)

c_v : constant volume specific heat (kcal/kg $^{\circ}$ K)

m : $(\kappa-1)/\kappa$

α : sharing ratio of the turbine exhaust heat

H_F : coefficient of performance of refrigerating cycle

D : circulating mass flow rate of refrigerant kg/sec

G : mass flow (kg/sec)

P : pressure (kg/m 2)

T : temperature ($^{\circ}$ K) [t : ($^{\circ}$ C)]

Subscripts

K : compressor

B : combustor

T : turbine

R : regenerator

E : evaporator

G : generator

W : suction cooler

A : isentropic process

a : air

b : brine

g : combustion gas

m : mean value

0 : ambient condition

1, 2,7: each state condition of gas turbine

1', 2',9': each state condition of refrigerating machine

3.2 Formulae of thermal efficiency and specific power

The compressor work input L_K is given by

Fig. 1 Schematic cycle arrangement of the suction cooling gas turbine cycle combined with absorption-type refrigerating machine using the wasted heat

Fig. 2 Temperature-entropy diagram of the suction cooling cycle

$$L_K = \frac{G_0 T_0 \theta}{75 A \eta_K} [c_{pam}]_1^{2A} (\phi_K^{ARa/[c_{pam}]_1^{2A}} - 1), \dots (1)$$

and compressor outlet temperature T_2 is

$$T_2 = T_0 \theta \left\{ 1 + \frac{1}{\eta_K} \frac{[c_{pam}]_1^{2A}}{[c_{pam}]_1^2} (\phi_K^{ARa/[c_{pam}]_1^{2A}} - 1) \right\} \dots (2)$$

here $\theta = T_1/T_0$.

From the definition of the temperature efficiency of suction cooler, the compressor inlet temperature

T_1 becomes

$$T_1 = T_0 - \eta_{wa}(T_0 - T_8'), \dots (3)$$

and the brine temperature at the evaporator outlet T_8' is $T_8' = (T_E + \Delta T)$,

and also the brine temperature at the suction cooler outlet T_9' is given by

$$T_9' = T_8' + \eta_{wb}(T_0 - T_8'),$$

here, ΔT is the temperature difference between T_8' and T_E ($T_8' > T_E$).

Turbine output L_T is

$$L_T = G_0 \{ (1 - \beta_K) + f \} \frac{T_0 \theta \tau}{75 A} \eta_T [c_{pgm}]_4^3 (1 - \phi_T^{ARg/[c_{pgm}]_4^3}) \dots (4)$$

here, $\tau = T_3/T_1$.

Turbine outlet temperature T_4 is

$$T_4 = T_0 \theta \tau \left\{ 1 - \frac{[c_{pgm}]_4^3}{[c_{pgm}]_4^3} \eta_T (1 - \phi_T^{ARg/[c_{pgm}]_4^3}) \right\} \dots (5)$$

Now, the total added heat to the combustor Q_B is given by

$$Q_B = \frac{G_0 T_0 \theta}{\eta_B} \{ (1 - \beta_K) + f \} [c_{pgm}]_5^3 (\tau - \tau_5) = G_0 f H_u \dots (6)$$

and, from Eq. (6), the fuel air ratio f is given by

$$f = \{ (1 - \beta_K) [c_{pgm}]_5^3 (\tau - \tau_5) \} / \{ \eta_B H_u - [c_{pgm}]_5^3 (\tau - \tau_5) \} \dots (7)$$

here, $\tau_5 = T_5/T_1$.

From the definition of effectiveness of regenerator, air temperature at the regenerator outlet T_5 is given as

$$T_5 = T_0 \theta \left\{ \left[1 + \frac{1}{\eta_K} \frac{[c_{pam}]_1^{2A}}{[c_{pam}]_1^2} (\phi_K^{ARa/[c_{pam}]_1^{2A}} - 1) \right] \left\{ 1 - \eta_R \frac{\{ (1 - \beta_K) + f \} (1 - \beta_T) [c_{pgm}]_2^4}{(1 - \beta_K) [c_{pgm}]_2^5} \right\} \right. \right. \\ \left. \left. + \tau \left\{ 1 - \frac{[c_{pgm}]_4^3}{[c_{pgm}]_4^3} \eta_T (1 - \phi_T^{ARg/[c_{pgm}]_4^3}) \right\} \eta_R \frac{\{ (1 - \beta_K) + f \} (1 - \beta_T) [c_{pgm}]_2^4}{(1 - \beta_K) [c_{pgm}]_2^5} \right\} \right\} \dots (8)$$

The formulae for thermal efficiency η_a and specific power output L_s of the cycle are consequently given by using the formulae derived above,

$$\eta_a = \frac{\eta_m \eta_B \{ (1 - \beta_K) + f \} \tau \eta_T [c_{pgm}]_4^3 (1 - \phi_T^{ARg/[c_{pgm}]_4^3}) - (1/\eta_K) [c_{pam}]_1^{2A} (\phi_K^{ARa/[c_{pam}]_1^{2A}} - 1)}{\{ (1 - \beta_K) + f \} [c_{pgm}]_5^3 (\tau - \tau_5)} \dots (9)$$

$$L_s = \frac{T_0 \theta}{75 A} \eta_m \left\{ \{ (1 - \beta_K) + f \} \tau \eta_T [c_{pgm}]_4^3 (1 - \phi_T^{ARg/[c_{pgm}]_4^3}) - \frac{1}{\eta_K} [c_{pam}]_1^{2A} (\phi_K^{ARa/[c_{pam}]_1^{2A}} - 1) \right\} \dots (10)$$

3.3 Heating source temperature in generator

The heating source temperature at the generator inlet T_6 is readily obtained from the definition of effectiveness of regenerator, using the formulae (2) and (5).

$$T_6 = T_0 \theta \left\{ \left[1 + \frac{1}{\eta_K} \frac{[c_{pam}]_1^{2A}}{[c_{pam}]_1^2} (\phi_K^{ARa/[c_{pam}]_1^{2A}} - 1) \right] \eta_R \frac{[c_{pgm}]_2^4}{[c_{pgm}]_6^4} \right. \\ \left. + \tau \left\{ 1 - \frac{[c_{pgm}]_4^3}{[c_{pgm}]_4^3} \eta_T (1 - \phi_T^{ARg/[c_{pgm}]_4^3}) \right\} \left(1 - \eta_R \frac{[c_{pgm}]_2^4}{[c_{pgm}]_6^4} \right) \right\} \dots (11)$$

And, the heating source temperature at the generator outlet T_7 can be derived as follows.

Coefficient of performance in the absorption-type refrigerating cycle H_F , which is a significant performance parameter, would be commonly defined as

$$H_F = q_E/q_G \dots (12)$$

Where, q_E and q_G refer to the refrigeration capacity and the added heat to the generator for the unit mass flow rate of the refrigerant.

Total heat added to the generator Q_G for generating the required circulating mass flow of the refrigerant D becomes

$$Q_G = G_b [c_{pgm}]_7^6 (T_6 - T_7) = D q_G \dots (13)$$

and refrigeration capacity Q_E , obtained in this case corresponding to the circulating mass flow of the brine G_b , is

$$Q_E = G_b [c_{pbm}]_8^{9'} (T_9' - T_8') = D q_E \dots (14)$$

Thus, the temperature T_7 becomes, by substituting formulae (11), (12) and (14) into (13),

$$T_7 = T_0 \theta \left\{ \left[1 + \frac{1}{\eta_K} \frac{[c_{pam}]_1^{2A}}{[c_{pam}]_1^2} (\phi_K^{ARa/[c_{pam}]_1^{2A}} - 1) \eta_R \frac{[c_{pam}]_2^4}{[c_{pam}]_6^4} \right] + \tau \left\{ 1 - \frac{[c_{pam}]_4^3}{[c_{pam}]_4^3} \eta_T (1 - \phi_T^{ARa/[c_{pam}]_4^3} - 1) \right\} \left(1 - \eta_R \frac{[c_{pam}]_2^4}{[c_{pam}]_6^4} \right) - \left\{ \frac{G_b [c_{pam}]_8^{9'} (\tau_9' - \tau_8')}{G_b [c_{pam}]_7^6 H_F} \right\} \right\} \dots (15)$$

here, $\tau_8' = T_8'/T_1$, $\tau_9' = T_9'/T_1$.

It should be noted that the continuous operation of the absorption-type refrigerating plant requires the minimum heating source temperature t_{Gmin} in the generator for generating the required amount of refrigerant vaporization. The operating region of this cycle, therefore, should be defined with $t_7 \geq t_{Gmin}$ for the sake of economical operation and better performance of the refrigerating plant.

3.4 Sharing ratio of the turbine exhaust heat

It is desirable that the turbine exhaust heat is shared effectively by the regenerator in the gas turbine and the generator in the refrigerating machine for the continuous operation of the cycle. Now, the sharing ratio of the turbine exhaust heat, denoted by the symbol α , is defined as the ratio of the turbine exhaust heat recovered by the regenerator to the total turbine exhaust heat available, thus

$$\alpha = \frac{[c_{pam}]_6^4 \left\{ \tau \left\{ 1 - \frac{[c_{pam}]_4^3}{[c_{pam}]_4^3} \eta_T (1 - \phi_T^{ARa/[c_{pam}]_4^3} - 1) \right\} - \tau_8 \right\}}{[c_{pam}]_7^4 (\tau_4 - \tau_7)} \dots (16)$$

where, $\tau_4 = T_4/T_1$, $\tau_7 = T_7/T_1$.

4. General performance characteristics of the suction cooling gas turbine cycle combined with absorption-type refrigerating machine

First of all, the relation between the compressor inlet temperature T_1 and the ambient temperature T_0 for the evaporating temperature of the refrigerant T_E in the absorption-type refrigerating machine is obtained, and the general thermodynamic performance characteristics of the cycle, with the variation of the ambient temperature t_0 as 0°C , 15°C , 30°C and 45°C , are determined under the fixed turbine inlet temperature $t_3 = 750^\circ\text{C}$. The numerical calculation is performed by using the formulae obtained in Chapt. 3 and M.H. Edson's formula⁽²⁾ for the constant pressure specific heat which varies with absolute temperature T and the compositions of working medium for each value of the fuel air ratio. Assumptions for the calculation of the design point performance are listed

Table 1 Assumptions for the calculation of the design point performance

Compressor adiabatic efficiency	$\eta_K = 0.85$
Turbine adiabatic efficiency	$\eta_T = 0.85$
Combustion efficiency	$\eta_B = 0.98$
Effectiveness of regenerator	$\eta_R = 0 \sim 0.85$
Mechanical efficiency	$\eta_m = 1.00$
Temperature efficiency of the suction cooler	$\eta_{wa} = 0.85$ (air side) $\eta_{wb} = 0.81$ (brine side)
Ambient pressure	$P_0 = 10\,332.3 \text{ kg/m}^2$
Lower calorific value of fuel	$H_u = 10\,420 \text{ kcal/kg}$
Thermal equivalent of mechanical work	$A = 1/426.8 \text{ kcal/kg} \cdot \text{m}$
Universal gas constant	$R = 847.9 \text{ kcal/kmol} \cdot ^\circ\text{K}$
Pressure loss rate	$\epsilon_{Ba} = 3\%$ $\epsilon_{Bg} = 2\%$ $\epsilon_{wa} = 3\%$ $\epsilon_{wg} = 5\%$ $\epsilon_B = 2\%$
Leakage loss rate	$\beta_K = 1\%$ $\beta_T = 1\%$

in Table 1.

4.1 Performance of the absorption-type refrigerating machine and the compressor inlet temperature

(a) Determination of the parameters in the calculation of the absorption refrigerating cycle

The absorption-type refrigerating machine shows different performances in its practical uses and in the scheme of cycle arrangement. Also, even in the case of operation with the brine temperature kept constant at the evaporator outlet, the performance of the cycle varies greatly under the influence of correlation of the complex parameters such as the heating source temperature, cooling water temperature and its amount of supply, circulating mass flow rate of aqua-ammonia solution, amount of circulating refrigerant and the density difference between the strong solution and the weakened solution.

Therefore, in this study, the empirical data only for the essential factors, which effect a change mostly in the performance of the refrigerating cycle, are selectively used.

Thus, the analysis of the absorption-type refrigerating cycle⁽³⁾ has been performed for the most fundamental single stage ammonia absorption-type refrigerating machine with the heat exchanger HE, the rectifier S and the dephlegmator D. The refrigerating capacity of the cycle and the heat balance for each component per unit circulating mass flow of the refrigerant are obtained by using the i - ξ diagram (enthalpy-density diagram) of the aqua-ammonia solution.

In the numerical analysis, the minimum heating source temperature is $t_{Gmin} = (t_7 + 5)^\circ\text{C}$, because the temperature t_7 is limited by the evaporating temperature which is slightly lower than t_{Gmin} . And as the brine temperature at the evaporator outlet t_8' is higher than the evaporating temperature t_E , their

temperature difference is fixed as $\Delta t = 3^\circ\text{C}$. Cooling water supplies to the condenser C and the dephlegmometer D are in series and parallel to the absorber in this case.

Cooling water inlet temperature in every case is fixed at $t_w = 15^\circ\text{C}$. The other assumptions for the calculation of the cycle mentioned above are listed in Table 2.

(b) Relation between the evaporating temperature of the refrigerant and the coefficient of performance of the absorption-type refrigerating cycle

The value of t_E is commonly given by the required temperature t_g' , and the obtainable minimum evaporating temperature in the single stage absorption-type refrigerating machine depends largely upon the value of density difference $\Delta\xi$, the values of strong solution density ξ_r and weakened solution density ξ_a , cooling water inlet temperature and its amount of supply.

The value of density difference higher than $\Delta\xi = 8\%$ is used in the large refrigerating capacity refrigerating machine, so that the weakened solution temperature at the generator outlet should be high in order to get a low value of ξ_a for the given condensing and evaporating pressure. In this case, however, the higher temperature heating source is needed and consequently, the value of coefficient of performance of the cycle decreases. Figure 3 shows the relation between t_E and H_F with constant values of $\Delta\xi = 10\%$, $t_w = 15^\circ\text{C}$ and $t_{G \text{ min}} = 105^\circ\text{C}$. The empirical data on the practical cases are also plotted in the same figure for comparison.

As can be seen in the figure, t_E falls with decreasing H_F and this tendency is reasonably similar to the practical case. This will be easily understood

Table 2 Assumptions for the calculation of the absorption-type refrigerating cycle

Ambient temperature	$t_0 = 0^\circ\text{C}, 15^\circ\text{C}, 30^\circ\text{C}, 45^\circ\text{C}$
Evaporating temperature of refrigerant	$t_E = 0^\circ\text{C}, -15^\circ\text{C}, -30^\circ\text{C}, -45^\circ\text{C}$
Temperature of refrigerant solution at the condenser outlet	$t_2' = 30^\circ\text{C}$
Temperature of strong solution at the absorber outlet	$t_4' = 30^\circ\text{C}$
Temperature of weakened solution at the generator outlet	$t_6' = 100^\circ\text{C}$
Density of refrigerant vapour at the dephlegmometer outlet	$\xi_1' = 99.5\%$
Density difference between strong solution and weakened solution	$\Delta\xi = 10\%$
Condensing pressure	$P_c = 3.0 \sim 12 \text{ ata}$
Evaporating pressure in the evaporator	$P_E = 0.5 \sim 3.0 \text{ ata}$
Temperature efficiency of the suction cooler	$\eta_{wa} = 0.85$ (air side) $\eta_{wb} = 0.81$ (brine side)
Temperature of strong solution at the heat exchanger outlet	$t_5' = 77 \sim 79^\circ\text{C}$
Temperature of weakened solution at the heat exchanger outlet	$t_7' = 47 \sim 51^\circ\text{C}$
Brine (CaCl 29.9%) mass flow rate	$G_b = 0.38 \text{ kg/sec}$

from the fact that the refrigerating capacity decreases when t_E is lowered keeping other conditions constant in the given refrigerating machine.

(c) Relation among the evaporating temperature, refrigerating capacity and heat added to the generator.

The relation between the refrigerating capacity Q_E and the added heat to the generator Q_G under the same conditions as in Fig. 3, taking t_E and D as variables, is shown in Fig. 4. Now, when the cooling water conditions are fixed, Q_G , which is required for operating the absorption refrigerating cycle to the given value of t_E under certain condition

Fig. 3 Relation of the evaporating temperature and the coefficient of performance for the single stage absorption-type refrigerating machine

Fig. 4 Performance curve for the single stage absorption-type refrigerating machine

in the cycle, can be determined. For instance, from Fig. 4, $Q_G=545$ kcal/sec is required for getting $t_E=-30^\circ\text{C}$ in the case of $D=1.0$ kg/sec and the refrigerating capacity can be obtained as $Q_E=244$ kcal/sec.

Therefore, the value of the coefficient of performance becomes $H_F=545/244=0.45$. But the actual value of H_F becomes lower and drops to approximately 80~85%, considering the heat losses in the absorber, rectifier and system as a whole. Thus, in this case the value is assumed to be 83%, and therefore $H_F=0.37$. On the other hand, in the case of operation of the cycle under the fixed condition of t_E , that is, under condition of the constant value of t_s' , it would be seen that the amount of D should be controlled by the controlling method of throttling in order to match Q_E to the variation of the load.

In Fig. 5, the relation of the required magnitudes of D and t_E for the unit mass flow of intake air to the gas turbine cycle is shown for t_0 , as a parameter, with the value of temperature efficiency of the suction cooler $\eta_{wa}=0.85$.

From the figure, the required D for cooling the unit mass flow of intake air to the gas turbine can be obtained when the refrigerating load varies with t_0 .

(d) Relationship between evaporating temperature and compressor inlet air temperature

The variation of the suction temperature t_1 with t_E for t_0 , as a variable, is shown in Fig. 6 under the same conditions as in Fig. 5.

As can be seen in Fig. 6, the relationship of t_E and t_1 varies linearly with t_0 , and the variation of t_1 due to t_E is large, but the variation of t_1 is

Fig. 5 Relation of the circulating mass flow rate of the refrigerant for the unit mass flow of an intake air to the gas turbine cycle and the evaporating temperature

slight when t_E is constant.

4.2 Sharing of the turbine exhaust heat

The remaining turbine exhaust heat available, which was partially recovered in the regenerator, is used as a heating source to the generator.

In this case, it is desirable that the turbine exhaust heat is shared suitably by the components for the continuous operation of the cycle, taking account of the high thermal efficiency of the whole system and all time generation of the required amount of the refrigerant vapour from the generator for the refrigerating load demanded.

Figure 7 shows the variation of the sharing ratio with the effectiveness of regenerator as a variable under the conditions of $t_E=-30^\circ\text{C}$, $t_0=15^\circ\text{C}$ and $t_3=750^\circ\text{C}$.

From the figure, it is obvious that α decreases as ϕ increases, and the proportion of the exhaust

Fig. 6 Relation of the compressor inlet air temperature and the evaporating temperature

Fig. 7 Variation of the shearing ratio of the turbine exhaust heat

heat share to the generator increases; the reason for this lies in the fact that the turbine exhaust gas temperature falls as the turbine expansion ratio increases. And therefore the exhaust heat available which can be exchanged in the regenerator decreases.

4.3 The heating source temperature and the operating region

In order to operate the absorption-type refrigerating machine, the minimum heating source temperature is necessary for generating the required amount of the refrigerant vapour. If the heating source temperature is low and the temperature of aqua-ammonia solution (weakened solution) at the generator outlet does not reach a certain degree, the density difference of the cycle is reduced and it causes an increase in the circulating mass flow of the solution and a decrease of H_F . Thus the performance of the absorption-type refrigerating machine falls.

The value of $t_{G \min}$ is to any extent open to choice, and generally the higher the value of $t_{G \min}$, the greater will be H_F . Therefore it will be possible to operate the refrigerating machine more economically. However, the value of $t_{G \min}$ is determined in practice by the properties of the aqua-ammonia solution, and is, moreover, limited by the temperature of the heating source medium, such as the steam, hot water or exhaust gas.

In the ammonia absorption-type refrigerating machine, generally, the refrigerant density is low, because it is almost impossible to vapourize the ammonia completely as a refrigerant from water, since the difference in the boiling points between water and ammonia is small in the refrigerant vapour produced in the generator.

Therefore, as indicated previously, $t_{G \min}$ is assumed to be 5°C higher than that of the vapourizing temperature of aqua-ammonia solution in the

generator, and the value is set as $t_{G \min} = 105^\circ\text{C}$, considering the better performance of the refrigerating machine.

The cycle can therefore be operated within the region of $t_7 \geq t_{G \min}$, which represents the operating limit of this cycle.

Figure 8 shows the variation of t_7 with ϕ for η_R , as variable, under the conditions of $t_0 = 15^\circ\text{C}$ and $t_E = -30^\circ\text{C}$.

And the temperature of $t_{G \min}$ is denoted in the same figure by the chain-double-dotted line and then the cycle is operated within the region of $t_7 \geq t_{G \min}$.

From the figure, the temperature of t_7 is seen to fall with an increasing value of η_R , due to the increasing amount of the exhaust heat recovery in the regenerator.

The variation of t_7 with ϕ is shown in Fig. 9 and Fig. 10. From these figures, the temperature t_7 is seen to fall with the lower value of t_E when t_0 is constant and with the higher value of t_0 when t_E is constant. That is, the lower the temperature t_7 becomes, the lower the value of t_E and the higher the

Fig. 8 Effect of the effectiveness on the heating source temperature and the operating region

Fig. 9 Variation of the heating source temperature for the evaporating temperature

Fig. 10 Variation of the heating source temperature for the ambient temperature

value of t_0 become for the constant value of η_R , and then the operating region of the cycle is restricted to within narrow limits.

Moreover, in order to extend the operating region of the cycle, as a matter of consideration, a mode of operation of the cycle is introduced to feed the turbine exhaust heat directly to the generator by using the by-pass valve V_3 (see Fig. 1).

4.4 Effect of the effectiveness of the regenerator on thermal efficiency

The results of analysis on η_a of the cycle with ϕ , taking η_R as variable, using the formula (9), are shown in Fig. 11, in the case of $t_0=15^\circ\text{C}$, $t_E=-30^\circ\text{C}$. In the figure, the maximum thermal efficiency $\eta_{a\max}$ is plotted by the thick chain-dotted line. The chain-double-dotted line and the fine chain-dotted line refer respectively to the operating region and outside of the operating region of the cycle.

From the figure, it is obvious that the effectiveness η_R influences thermal efficiency largely as in the case of the conventional regenerative cycle, and the higher value of η_R gives the higher thermal efficiency.

The optimum pressure ratio ϕ_{opt} exists for $\eta_{a\max}$ and the value of ϕ_{opt} , which is higher than that of the conventional regenerative cycle, moves to the lower region of the pressure ratio ϕ .

4.5 Effects of the evaporating temperature on thermal efficiency

Figure 12 shows the effects of t_E with ϕ , taking t_E as a variable, under the conditions of $t_0=15^\circ\text{C}$, and $\eta_R=0.85$.

Inspection of the figure reveals that the evaporating temperature gives definite influence on the thermal efficiency and η_a increases with a decrease of the value of t_E . The optimum pressure ratio ϕ_{opt} exists for $\eta_{a\max}$ and it moves, in this case, to the

higher region of the pressure ratio.

The operating region of the cycle, when t_0 is constant, moves widely for the variation of t_E and the region is restricted to within narrow limits with a decrease of the value of t_E . Generally, the thermal efficiency increases with a decrease in t_E , and there is a favorable tendency that the curves of thermal efficiency lie flatly over a wide range of ϕ , but the value of $\eta_{a\max}$ within the operating region is not always high when t_E is low. For instance, inspecting the figure, when $t_E=-30^\circ\text{C}$, the value shows $\eta_{a\max}=35.4\%$ for $\phi_{\text{opt}}=4.6$, but when $t_E=-45^\circ\text{C}$, $\eta_a=34.6\%$ for $\phi=9.0$.

In this case, the value of η_a is low in spite of the low value of t_E and $\eta_{a\max}$ is out of the operating region. It is suggested that there may exist a certain limited value of t_E in the cycle.

Fig. 12 Effect of the evaporating temperature on thermal efficiency

Fig. 11 Effect of the effectiveness on thermal efficiency

Fig. 13 Effect of ambient temperature on thermal efficiency

4.6 Effect of the ambient temperature on thermal efficiency

The variation of η_a with ϕ for t_0 as a variable is shown in Fig. 13, under the fixed conditions of $t_E = -30^\circ\text{C}$ and $\eta_R = 0.85$. The effect of t_0 on η_a is very slight, but η_a decreases with the higher value of t_0 and then the operating region of the cycle is restricted to within narrow limits, that is; when t_E is constant, there is a certain limit to improve η_a by the falling of t_0 .

4.7 Effect of the evaporating temperature on the specific power

In Fig. 14, the effect of t_E on the specific power with ϕ is shown under the conditions of $t_3 = 750^\circ\text{C}$ and $t_0 = 15^\circ\text{C}$, using the formula (10).

The specific power L_s , which is hardly affected by η_R , varies definitely by t_E and it increases with a fall of the value of t_E . And also, the value of the optimum pressure ratio ϕ_{opt} for the maximum specific power $L_{s,\text{max}}$ exists in the cycle and the value of ϕ_{opt} moves to the range of the higher value of ϕ when the value of t_E falls, as in the case of the thermal efficiency described previously.

The tendency of the movement of the operating region of the cycle is almost the same as that of the thermal efficiency described in the previous chapter.

Now, the cycle treated in this study has the particular nature that the specific weight of the working fluid of the cycle is increased as the temperature of t_1 falls remarkably. In the same figure, the power output L' with consideration of the increase of mass flow due to the specific weight

increased is denoted by fine lines. The value of L_s does not vary by the increase in the specific weight of the working fluid due to the falling of the temperature t_1 , but the power output L' can be definitely increased as shown in the figure and the tendency described above is remarkable for the lower value of t_E .

4.8 Effect of the ambient temperature on specific power

Figure 15 shows the relations between the specific power L_s and the power output L' considered with the increase of the specific weight, with ϕ for t_0 as a variable, under the constant value of $t_E = -30^\circ\text{C}$.

As can be seen in the figure, when t_E is constant, the variation of either L_s or L' due to t_0 is slight, and the latter, however, increases with the higher value of t_0 , because the temperature difference between t_0 and t_1 becomes large with a lowering t_0 and this causes further increase in the specific weight of the working fluid.

Figures 16 and 17 show the performance characteristics of the suction cooling cycle.

The relations of η_a and L_s , in the case of $t_E = -30^\circ\text{C}$, for t_0 and ϕ as variables are shown in Fig. 16, and the relations of these for t_E and ϕ , as variables, in the case of $t_0 = 30^\circ\text{C}$, are also shown in Fig. 17. The operating region and the outside of the operating region for the cycle are respectively shown, in the figures, by the chain-double-dotted line and the chain-dotted line.

The figures will clearly show the effects of t_0 and t_E on η_a and L_s , described previously in the chapters 4.5 ~ 4.8

Fig. 14 Effect of the evaporating temperature on the specific power and power output

Fig. 15 Effect of ambient temperature on the specific power and the power output

4.9 The optimum conditions in the suction cooling cycle

In the foregoing chapters, the effects of η_R , t_E and t_0 on η_a and L_s for the cycle were mainly discussed.

As the result of the discussion, it was found that the cycle had the operating region for the steady operation of the refrigerating machine and then the performance characteristics of the cycle within the operating region can be determined by the correlation between η_R , t_E and t_0 . By inspecting Fig. 18, the

Fig. 16 Performance of suction cooling cycle ($t_E = -30^\circ\text{C}$, $\eta_R = 0.85$)

Fig. 17 Performance of suction cooling cycle ($t_0 = 30^\circ\text{C}$, $\eta_R = 0.85$)

illustration mentioned above can be seen clearly and the optimum conditions for $\eta_{a\max}$ of the cycle will also be given in the figure.

In Fig. 18, the curves for $\eta_{a\max}$ with a varying value of η_R for the constant evaporating temperature t_E and the curves for $\eta_{a\max}$ with a varying value of t_E for the constant effectiveness of η_R are shown together, under the conditions of $t_3 = 750^\circ\text{C}$ and $t_0 = 30^\circ\text{C}$. The operating region is presented on the same figure by the chain-double-dotted line, and the cycle can be therefore operated below the line.

As it is shown in the figures, $\eta_{a\max}$ increases as η_R increases when t_E is constant and also it increases as t_E falls when η_R is constant. Moreover, when both t_E and η_R are constant, $\eta_{a\max}$ decreases with the rise of t_0 , as already indicated.

The operating region of the cycle is restricted to within narrow limits with the lower value of t_E and the higher value to t_0 . Therefore, it will be seen that the value of η_R should be as high as possible and the selection of the value of t_E must consider the variation of t_0 in order to satisfy the optimum conditions for the cycle within the operating region, and also the optimum conditions for L_s , generally, do not coincide with that for η_a , but it can be said that the selection of t_E for the variation of t_0 is entirely the same as in the case of thermal efficiency.

5. Comparison of the cycle performances

The relations between η_a and L_s , taking t_0 and ϕ as variables, are shown in Fig. 19 and Fig. 20

Fig. 18 Relation of maximum thermal efficiency and the optimum pressure ratio

under conditions of $t_3=750^\circ\text{C}$ and $\eta_R=0.85$ for the fixed values of $t_E=-15^\circ\text{C}$ and $t_E=-30^\circ\text{C}$.

In the figures, the performance for the conventional regenerative cycle is depicted by the dashed lines and the fine lines plotted in Fig. 20 refer to the performance of the suction cooling cycle with consideration of the increased specific weight of working fluid due to the fall of the temperature t_1 .

According to the figures, the comparative study for the performances of the suction cooling cycle and the conventional regenerative cycle has revealed the following ;

(1) The thermal efficiency η_a of the suction cooling cycle can be improved remarkably when t_0 increases.

(2) The variation of η_a and L_s for the variation of t_0 is very slight and its tendency is remarkable with the higher value of t_0 .

(3) The power output L' with consideration of the increased specific weight of working fluid due to the falling of t_1 is very large.

6. Results

A suction cooling gas turbine cycle combined with the absorption-type refrigerating machine using the turbine wasted heat as a heating source to the generator in the low-temperature single stage ammonia absorption-type refrigerating machine is devised and besides the investigation has been performed on the fundamental and general thermodynamic characteristics of the cycle.

The conclusions are summarized as follows ;

(1) Depending on the mode of the combined

cycle of the gas turbine and the absorption-type refrigerating cycle, higher values of the specific power and the thermal efficiency can be obtained by lowering the intake air temperature to the compressor.

(2) These values of the specific power and the thermal efficiency increase definitely with the higher value of the ambient temperature.

(3) The variations of the specific power and the thermal efficiency due to the ambient temperature are very slight.

(4) The actual power output with consideration of the increased specific weight of working fluid due to the lowering of the intake air temperature to compressor is very large.

(5) This cycle has the operating region which is restricted by the heating source temperature to the generator and the region is generally restricted to within narrow limits with the lower value of the evaporating temperature of the refrigerant and the higher value of the ambient temperature.

(6) It will be desirable that the value of η_R is as high as possible and the selection of the value of the evaporating temperature consider the variation of the ambient temperature in order to satisfy the optimum conditions within the operating region of the cycle.

(7) The optimum pressure ratios for the maximum thermal efficiency and the maximum specific power exist and these values are higher than those of the conventional regenerative cycle.

As a result, it is made clear that the demerits of the gas turbine pointed out in the introduction can be remarkably improved, that is ; this cycle can

Fig. 19 Cycle performances

Fig. 20 Cycle performances ($t_E = -30^\circ\text{C}$)

achieve higher values of the thermal efficiency and the specific power than the gas turbine running alone. Furthermore, the cycle has a desirable merit of lessening the variation of these values, when the ambient temperature varies. In addition authors have no doubt that the investigation of this cycle will contribute greatly to the fields of the practical application of the gas turbine.

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Discussion

K. SUNOBE: In Section 4.3, it is found that the exhaust gas temperature at the generator outlet is assumed $t_7 \geq t_{Gmin}$ and thus the cycle can be operated within the region where this value of the temperature is higher than that of 105°C.

But, I think this temperature limitation should be restricted to the higher value of it for the anti-corrosion of the pipe lines in the generator.

Now, for instance, how will the operating region of the cycle move when the value of its temperature is higher than that of 150°C?

Authors' closure

Authors assumed the value of the exhaust gas temperature as 105°C, with consideration of a better

performance of an ammonia refrigerating machine with an increased density of the refrigerant and of an effective recovery of the turbine exhaust heat and of using fine quality fuel, such as gaseous fuel, kerosene, light oil fuel, etc.

However, it might be better to keep the higher value of its temperature 170°C rather than 150°C when the heavy oil fuel is burnt for avoiding the low temperature corrosion.

In this case, the operating limit of the cycle will be restricted to within fairly narrow limits, but it is possible to extend its region by varying the effectiveness of the regenerator using the by-pass valve V_3 shown in Fig. 1.