621.165:536.717

Paper No. 169-9

Performance Analysis of the Steam Turbine Cycle combined with the Heat Pump*

By Takashi SONODA**, Takeshi SAITO*** and Hiroshi TANIGUCHI****

The performance of the steam turbine cycle combined with the heat pump is analyzed from the standpoint of both the 1st law and 2nd law of thermodynamics. This cycle is useful in order to reduce the thermal pollution locally caused by the waste heat from the steam condenser and to raise the total energy efficiency.

The following factors are considered here to analyze the performance of this cycle ; the internal efficiency of the steam turbine , the pressure of the steam condenser in the steam cycle, the kind of thermal media (NH_3 ,R11), the condensing temperature, the evaporating temperature , the outlet temperature of the cooling water from the evaporator and the internal efficiency of the compressor in the heat pump cycle.

1. Introduction

Recovering the waste heat from the steam power plant by the heat pump and using the heat for space heating or as other heat sources, whose temperature is raised to about 100°C, makes it possible to reduce the thermal pollution locally caused by the heat released by the steam condenser to surroundings. On the other hand, from the standpoint of energy utilization it is expected to raise the overall thermal efficiency, because the generated electrical power by the steam turbine and the recovered heat are both useful [1][2][3]. But the net electrical power generated becomes less by the value consumed by the heat pump compressor.

This paper presents an analysis of the performance of the steam turbine cycle combined with the heat pump from the standpoint of both the 1st law and 2nd law of thermodynamics [4][5]. In order to project this cycle and make the realization of such a plant possible, it is necessary to study the thermodynamic performance and solve many problems expected in the practical operation as compared with those of the conventional back pressure or bleeding steam turbine plants. Further as the capacity of a combined heat pump in this plant is supposed considerably larger than the one operated already, there are many problems to be solved in future , such as the improvement of the performance of the heat pump compressor [1][6]. Taking matters descrived above into consideration, the authors try to analyze the thermodynamic performance of this cycle which should be at first investigated . As the factors to affect the performance the following ones are considered ; the internal efficiency of the steam turbine , the pressure of the steam condenser in the steam cycle, the kind of the thermal media (NH_3 , R11), the condensing temperature , the evaporating temperature, the outlet temperature of the cooling water from the evaporator and the internal efficiency of the compressor in the heat pump cycle.

The heat whose temperature is raised by the heat pump is transferred to thermal medium of the heat utilization side , such as steam or hot water ,and used for any purposes.

In this paper it is proposed that one part of this heat is used for heating the boiler feed water. This may be regarded as the regenerative process from the standpoint of the thermodynamic cycle and the effect of this process on the performance is also studied.

The conversion efficiencies into the electrical power and the heat, and the sum of them , so-called thermal utilization factor which are derived from the 1st law of thermodynamics , and the exergy efficiency from exergy viewpoint are adopted as measures of the performance.

* Received 16th May, 1977

** Research Associate, Department of Mechanical Engineering, Faculty of Engineering, Hokkaido University, Sapporo.

*** Professor, School of Energy Engineering , Toyohashi University of Technology.

**** Professor, Department of Mechanical Engineering, Faculty of Engineering, Hokkaido University.

Nomenclature

- e : specific exergy kcal/kg
- E : exergy kcal
- G : weight flow rate kg/s
- h : specific enthalpy kcal/kg
- H : heat generated kcal/s
- p : pressure kg/cm^2
- P : electrical power generated kcal/s
- Q : heat amount kcal/s
- s : specific entropy kcal/kg°K
- S : entropy kcal/°K
- t : temperature °C
- T : temperature °K
- Δt : temperature difference deg
- W : power, or electrical power kcal/s
- η : conversion energy efficiency from the 1st law of thermodynamics
- ζ_{ex} : exergy efficiency from the 2nd law of thermodynamics

Subscripts

- B : boiler, or fuel

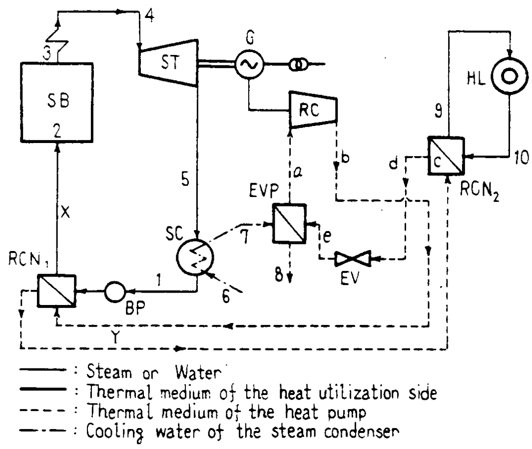
c : refrigeration cycle
H : heat generated
P : electrical power generated
R : thermal media, or heat pump cycle
s : steam, or isentropic
st : steam turbine
u : surroundings
w : cooling water
' : cycle points in Fig.2 and 3 in case that the temperature of the steam turbine condenser equals that of the heat pump evaporator
" : cycle points in Fig.2 in case that the power generated by the heat engine equals that required by the heat pump

For other scripts, refer to Fig.1.

2. The Steam Turbine Cycle combined with the Heat Pump

Figure 1 illustrates the steam turbine plant arrangement combined with the heat pump, where the heat pump recovers the waste heat transferred to the cooling water in the steam turbine condenser. In the case of combining the heat pump with the steam turbine plant operated already, this arrangement may be adopted as shown in Fig.1. But if the temperature of the cooling water entering the steam condenser is to be made to fall below the surrounding temperature by the heat pump evaporator (EVP) installed before the steam condenser, the condensing steam pressure becomes lower, resulting in an increase in the generated power by the steam turbine. Further, if the steam condenser is replaced by a heat pump evaporator, the expansion end pressure of the steam turbine can be selected below the saturation pressure corresponding to the surrounding temperature. The effect of these arrangements on the performance is discussed later in this paper . In the illustrated arrangement , the thermal medium leaving the heat pump compressor(RC) can heat the boiler feed water, and the two heat exchangers, RCN_1 , for the above mentioned purpose and RCN_2 for the heat generation, are connected in series here.

Figure 2 shows a Carnot cycle T-S diagram of the heat engine combined with the heat pump [2] . Here, the area AGHDA represents the waste heat amount from the heat engine and equals the area L'MN'I'L' = the area LMNIL which corresponds to the recovered heat amount by the heat pump. When the temprature of the low heat reservoir of the heat engine equals that of the evaporator of the heat pump , the driving power for the compressor is represented by the area KL'I'J'K equal to the area EADFE. In the back-pressure steam turbine plant where the generated electrical power and heat are the same as those of the combined plant dealt with here, the thermodynamic performance is the same since the generated electrical power and heat are represented by the areas BEFCB and EGHFE respectively. But some difference between the temperature of the low heat reservoir and that of the evaporator is practically required in the steam turbine cycle combined with the heat pump. In the case of the difference shown in Fig.2 the driving power for the heat pump compressor becomes more by the power corresponding to the area J'I'L'LIJJ'.



SB: steam boiler
ST: steam turbine
G: electrical generator
SC: steam condenser
BP: feed water pump of the boiler
RC: compressor of the heat pump
EVP: evaporator of the heat pump
EV: expansion valve
RCN_1: condenser of the heat pump, feed water heater of the boiler
RCN_2: condenser of the heat pump
HL: heat load

Fig.1 The Steam Turbine Plant Arrangement combined with the heat pump

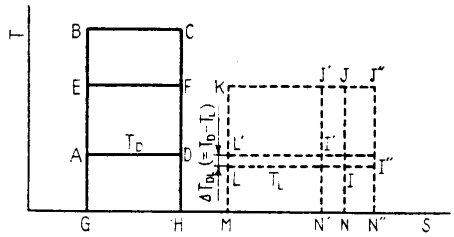


Fig.2 Carnot Cycle T-S Diagram of the Heat Engine combined with the Heat Pump

Therefore, the conversion efficiency into the heat is more than that of the back-pressure steam turbine plant. Thermal utilization factors of both plants are the same and equal to 100%, but the exergy efficiency (effectiveness) of the combined plant is less because of the exergy loss suffered in the heat exchange process under the finite temperature difference at the evaporator.

If only the heat generating cycle [9] is supposed in which the power generated by the heat engine is totally consumed as the driving power of the heat pump compressor, the area BADCB equals the area KLI"J"K in Fig.2 and the heat generated by this cycle is represented by the area KMN"J"K. That is more than the fuel energy by the energy corresponding to the area INN"I"I.

T-S and p-h diagrams in the actual case are shown in Fig.3, in which the steam turbine is operated as the Rankine cycle and the heat pump as the dry vapor compression cycle.

In T-S diagram the hatched area indicates the exchanged heat for heating the boiler feed water in use of the generated heat by the heat pump side. Therefore the temperature of the generated heat in this

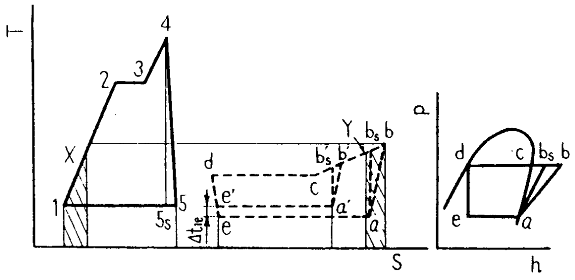


Fig.3 T-S and p-h Diagrams of the actual Steam Turbine Cycle combined with the Heat Pump

case is denoted by t_Y and that heat amount becomes less by the heat required for heating the boiler feed water.

3. Efficiencies of the Combined Cycle

In order to analyze the effect of the factors on the performance of the cycle shown in Fig.3 the supplied fuel energy, the generated electrical power and heat, and 1st law and 2nd law efficiencies as mentioned above are obtained here.

The steam turbine cycle;

supplied fuel energy to the boiler,

$$Q_{SB}=\{G_S(h_4-h_1)-Q_{Bw}\}/\eta_{SB} \quad \cdots\cdots (1)$$

where Q_{Bw} : the heat amount for heating the boiler feed water;

power generated by the steam turbine,

$$W_{ST}=G_S(h_4-h_5)=G_S\eta_t(h_4-h_{5S}) \quad \cdots\cdots (2)$$

waste heat from the steam condenser,

$$Q_{SC}=Q_{SB}-W_{ST}=G_S(h_5-h_1) \quad \cdots\cdots (3)$$

heat amount transferred to the cooling water,

$$Q_{SC}=G_w(h_7-h_6)=G_w(t_7-t_6) \quad \cdots\cdots (4)$$

accordingly, flow rate of the cooling water,

$$G_w=Q_{SC}/(t_7-t_6) \quad \cdots\cdots (5)$$

The heat pump cycle;

heat absorbed by the evaporator,

$$Q_{EVP}=G_R(h_a-h_e)=G_w(t_7-t_8)$$
$$=Q_{SC}(t_7-t_8)/(t_7-t_6) \quad \cdots\cdots (6)$$

where,

$$(t_7-t_8)/(t_7-t_6)=1-(t_8-t_6)/(t_7-t_6)$$
$$=1-\Delta t_{86}/\Delta t_{76}$$

and then

$$Q_{EVP}=(1-\Delta t_{86}/\Delta t_{76})Q_{SC} \quad \cdots\cdots (6)$$

flow rate of the thermal media,

$$G_R=Q_{EVP}/(h_a-h_e)$$
$$=(1-\Delta t_{86}/\Delta t_{76})Q_{SC}/(h_a-h_e) \quad \cdots\cdots (7)$$

driving power for the heat pump compressor,

$$W_{RC}=G_R(h_b-h_a)$$
$$=(1-\Delta t_{86}/\Delta t_{76})Q_{SC}(h_b-h_a)/(h_a-h_e) \cdots\cdots (8)$$

where, $\varepsilon_c=(h_a-h_e)/(h_b-h_a)$ is ususally defined as the coefficient of performance of the refrigeration cycle.

Then Eq. (8) yields

$$W_{RC}=(1-\Delta t_{86}/\Delta t_{76})Q_{SC}/\varepsilon_C \quad \cdots\cdots (9)$$

electrical power generated by the steam turbine combined with the heat pump,

$$P=W_{ST}-W_{RC}$$
$$=G_S(h_4-h_5)-(1-\Delta t_{86}/\Delta t_{76})Q_{SC}/\varepsilon_C \quad \cdots(10)$$

heat generated,

$$H=Q_{EVP}+W_{RC}-Q_{Bw}=G_R(h_b-h_d)-Q_{Bw}$$
$$=(1-\Delta t_{86}/\Delta t_{76})Q_{SC}\varepsilon_R/\varepsilon_C-Q_{Bw} \quad \cdots\cdots (11)$$

where, $\varepsilon_R=\varepsilon_C+1$

exergy of the heat generated,

$$E_H=G_R e_H=G_R\{(h_Y-h_d)-T_u(s_Y-s_d)\} \quad \cdots(12)$$

exergy of the supplied fuel energy,

$$E_{SB}=Q_{SB} \quad \cdots\cdots (13)$$

Efficiencies of the combined cycle; conversion efficiencies into the electrical power and the heat,

$$\eta_P=P/Q_{SB},\quad \eta_H=H/Q_{SB},\quad \eta_T=\eta_P+\eta_H \quad \cdots(14)$$

exergy efficiency (effectiveness),

$$\zeta_{ex}=(P+E_H)/E_{SB} \quad \cdots\cdots (15)$$

4. Discussion of Calculation Results

Under assumed conditions shown in Table 1, calculation is carried out in order to investigate the effect of factors on the performance efficiencies derived in the preceding section.

Table 1 Assumed Conditions

Inlet Steam Condition to the Steam Turbine	$p_4=150$ ata, $t_4=550$°C
Boiler Efficiency	$\eta_{SB}=1.0$
Internal Efficiency of the Steam Turbine	$\eta_{st}=0.85$, $1.0\sim0.5$
Steam Condenser	$p_5=0.043$ ata ($t_5=30$°C), 0.023 ata, 0.009 ata
Surrounding Temperature	$t_u=15$°C
Cooling Water	$t_6=15$°C, $t_7=25$°C, 5°C, $t_8=15$°C, 20°C ($\Delta t_{86}=0,5$°C)
Evaporating Temperature of the Heat Pump	$t_e=10$°C, -30°C$\sim$30°C ($\Delta t_{1e}=20$°C, $0\sim60$°C)
Internal Efficiency of the Heat Pump Compressor	$\eta_{RC}=0.8$, $1.0\sim0.6$
Condensing Temperature of the Heat Pump	$t_d=50$°C, $20\sim100$°C
Feed Water Heater of the Boiler (RCN_1)	$\Delta t_{bx}\geq5$°C, $\Delta t_{cz}\geq3$°C
	(Refer to Fig.6)

Figures 4 and 5 show the temperature distribution in the steam condenser and the heat pump evaporator.

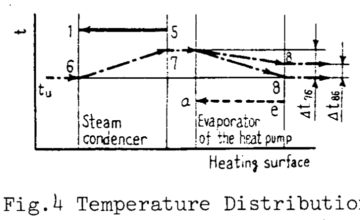


Fig.4 Temperature Distribution in the Steam Condenser and the Heat Pump Evaporator

The evaporator in Fig.4 is installed

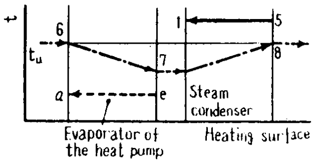
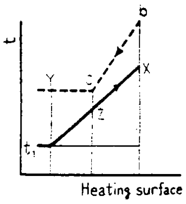


Fig.5 Temperature Distribution in the Steam Condenser and the Heat Pump Evaporator



z: point of the boiler feed water side at pinch point

Fig.6 Temperature Distribution in the Condenser RCN_1 of the Heat Pump

as in Fig.1 and the waste heat transferred to the cooling water is absorbed. In case of Fig.5, the evaporator is installed before the steam condenser and the cooling water at the surrounding temperature is cooled, lowering the steam condenser pressure (temperature) .

Figure 6 shows the temperature distribution of the heat exchanger (RCN_1) when one part of the heat generated by the heat pump condenser is used for heating the boiler feed water. Here, the values of properties of the working media in this cycle, steam, NH_3 and R11, such as enthalpy and entropy are obtained from the references (10)～(12) as functions of temperature and pressure. Numerical calculations are carried out by using the electronic digital computer.

4.1 Coefficients of Performance (COP) of NH and R11, and Their Outlet Temperatures from the Compressor

Figure 7 shows the COP ε_{RS} of the heat pump cycle and the outlet temperature t_{bS} from the compressor, in which the variable presented on the horizontal axis is the condensing temperature t_d and the parameter t_e is the evaporating temperature.

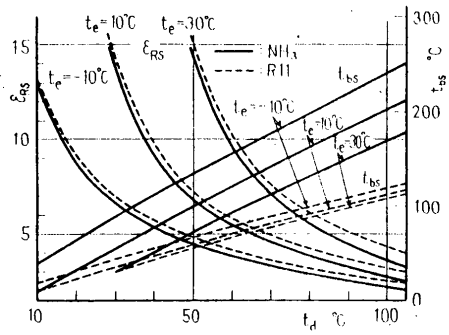


Fig.7 Effect of the Condensing Temperature t_d on the COP ε_{RS} of NH_3 , R11 and the Outlet Temperature t_{bS} of the Heat Pump Compressor

The COP of R11 is larger than that of NH_3 and the difference increases as the condensing temperature t_d becomes higher. On the other hand, the outlet temperature from the compressor of NH_3 is higher and more dependent on the evaporating temperature.

Under the same evaporating and condensing temperature, the more driving power for the compressor in case of NH_3 is required, but the temperature of the heat generated can be far higher than R11.

4.2 Effect of the Condensing Temperature of the Heat Pump

Figures 8 to 10 show the effect of the condensing temperature t_d on the efficiencies η_P, η_H, η_T from the 1st law and the exergy efficiency ζ_{ex} (effectiveness) from the 2nd law. In Figures the broken lines indicate the case that one part of the generated heat by the heat pump is used for heating the boiler feed water. The calculation result as shown in Fig.8 is obtained under the conditions that the thermal medium NH_3 is used and the parameter Δt_{86} which implies the ratio of the heat absorbed by the evaporator to the waste heat equals 0.

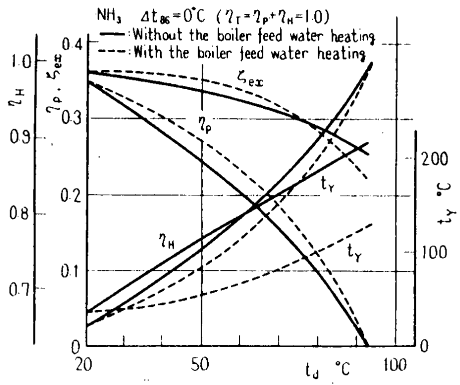


Fig.8 Effect of the Condensing Temperature t_d on the Efficiencies η_P, η_H and ζ_{ex} and the Inlet Temperature t_Y of the Thermal Medium to the Condenser RCN_2(NH_3, $\Delta t_{86}=0°C$)

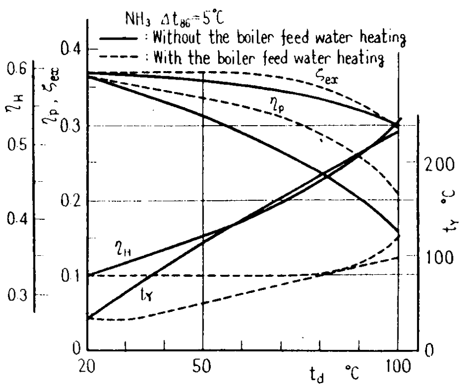


Fig.9 Effect of the Condensing Temperature t_d on the Efficiencies η_P, η_H and ζ_{ex} and the Inlet Temperature t_Y of the Thermal Medium to the Condenser RCN_2(NH_3, $\Delta t_{86}=5°C$)

The higher condensing temperature t_d is adopted resulting in the higher temperature of the generated heat by the heat pump cycle, the more driving power for the compressor is

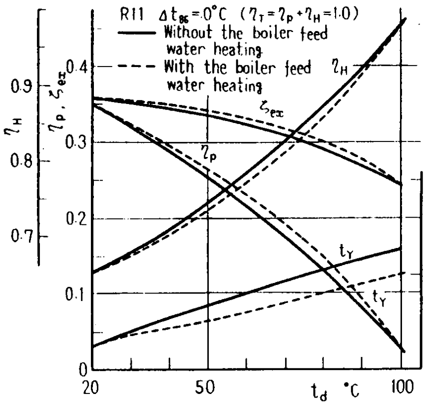


Fig.10 Effect of the Condensing Temperature t_d on the Efficiencies η_P, η_H and ζ_{ex}, and the Inlet Temperature t_Y of the Thermal Medium to the Condenser RCN_2 (R11, Δt_{86} = 0°C)

required and the conversion efficiency η_P into the electrical power decreases. This tendency becomes remarkable in the region of the higher condensing temperatures t_d . The exergy efficiency has a similar tendency but the decreasing rate is less than η_P. When the boiler feed water is heated by one part of the generated heat, it is shown that η_P is larger than that without heating by 2.5 % at t_d = 50°C. The effect of this regenerative process is larger in case of Δt_{86} =5°C as shown in Fig.9. Figure 10 shows the efficiencies, η_P , η_H and ζ_{ex} when the thermal medium R11 is used. The tendency is almost the same as that of NH_3, but the decreasing rate of η_P and ζ_{ex} and the effect of the regeneration by the boiler feed water heating are less than those of NH_3.

4.3 Effects of the Evaporating Temperature and the Internal Efficiency of of the Compressor in the Heat Pump Cycle

In Figures 11 to 12 the effects of the evaporating temperature, that is the temperature of the thermal medium in the heat pump absorbing the waste heat from the steam turbine plant are shown. Both the evaporating temperature t_e and the difference Δt_{1e} between the steam condensing temperature and the heat pump evaporating one are adopted as the variable of the horizontal axis in those Figures. When the waste heat transferred to the cooling water is absorbed by the evaporator as shown in Fig.4, the temperature difference Δt_{1e} of some magnitude is actually needed, related to the temperature of the cooling water. In this calculation, however, the value of Δt_{1e} is adopted up to 0°C since the steam turbine condenser can be replaced by the evaporator. With the lower evaporating temperature t_e , namely, the larger Δt_{1e} the efficiencies η_P and ζ_{ex} decreases the more remarkably. This tendency is the same for both thermal media, NH_3 and R11. Since the compressor outlet temperature of the heat pump becomes higher with a lower evaporating temperature t_e as in Fig.7, it

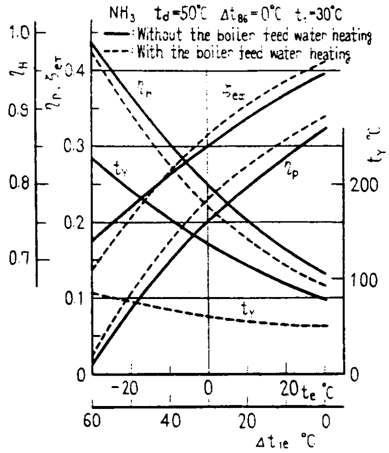


Fig.11 Effect of the Evaporating Temperature t_e on the Efficiencies η_P , η_H and ζ_{ex} , and the Inlet Temperature t_Y of the Thermal Medium to the Condenser RCN_2 (NH_3)

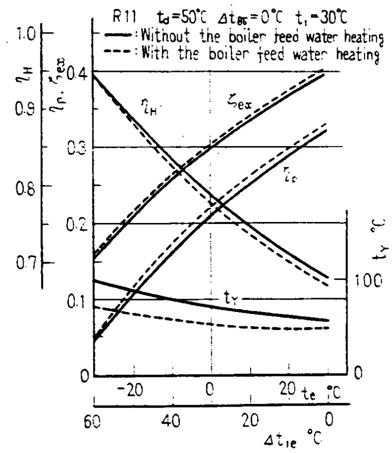


Fig.12 Effect of the Evaporating Temperature t_e on the Efficiencies η_P , η_H and ζ_{ex} , and the Inlet Temperature t_Y of the Thermal Medium to the Condenser RCN_2 (R11)

may be estimated that a better effect of heating the boiler feed water on the efficiencies appears. However, the effect on η_P of NH_3 becomes smaller with a lower t_e and the value of ζ_{ex} becomes smaller than that without the heating as shown in Fig.11.

Figure 13 indicates the effect of the compressor internal efficiency η_{RC} on the efficiencies η_P and ζ_{ex}. The reducing rates of η_P and ζ_{ex} become larger with a smaller η_{RC} . The effect of the kind of the thermal media on the efficiency ζ_{ex} does not appear and the values of η_{RC} for NH_3 and R11 are almost similar for smaller η_{RC}.

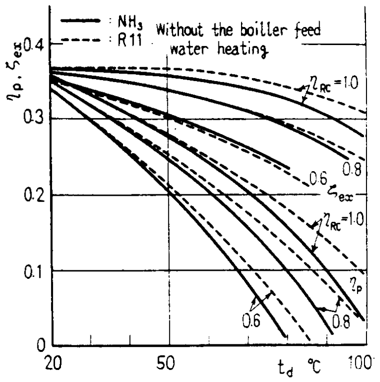


Fig.13 Effect of the Internal Efficiency η_{RC} of the Heat Pump Compressor on the Efficiencies η_P and ζ_{ex}

4.4 Effects of the Condensing Pressure (Temperature) and the Internal Efficiency of the Steam Turbine

The lower pressure of the steam turbine condenser results in an increasing power generated by the steam turbine and if the conditions of the heat pump side are kept invariable, it may be estimated that the more electrical power is generated the larger effect of the regenerative process becomes with the boiler feed water heating. But this is dependent on the arrangement in the cycle of the evaporator to absorb the waste heat from the steam turbine plant. Therefore, the pressure (temperature) of the steam turbine condenser cannot be selected quite independently. Here, three arrangements are considered. The first (denoted by I) is shown as Figs.1 and 4, the second (II) in Fig.5 and the third (III) is the case that the evaporator is used as the steam turbine condenser.

Figure 14 shows the variations of η_P and ζ_{ex} when the temperatures of the condensing

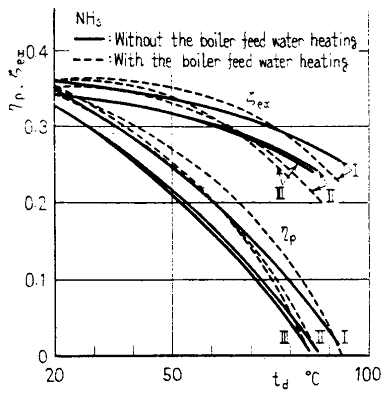


Fig.14 Effect of the Pressure (Temperature) of the Steam Turbine Condenser on the Efficiencies η_P and ζ_{ex}

steam equal 30°C, 20°C and 5°C for I, II and III respectively. Then the temperature differences Δt_{ie} are adopted to be 20°C, 20°C and 10°C. Accordingly, the evaporating temperatures t_e equal 10°C, 0°C and -5°C respectively.

The effect of heating the boiler feed water appears strongly in case of III and the difference becomes about 4% at t_d =50 °C. But the values of η_P and ζ_{ex} in case of I are larger than those of other arrangements , II and III . This implies that the effect of the evaporating temperature in the heat pump cycle appears stronger than that of the pressure (temperature) of the steam turbine condenser with the condensing temperature t_d kept constant.

Figure 15 shows the effect of the internal efficiency η_{st} of the steam turbine

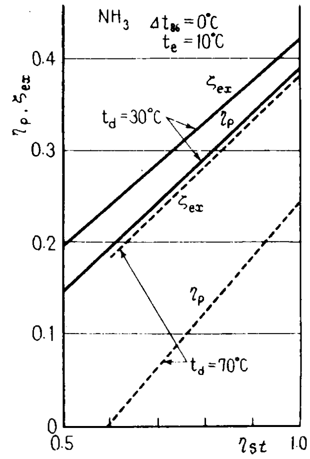


Fig.15 Effect of the Internal Efficiency η_{st} of the Steam Turbine on the Efficiencies η_P and ζ_{ex}

on the efficiencies η_P and ζ_{ex} . Under the conditions of the heat pump side kept invariable, the lower internal efficiency η_{st} which results in a decrease of the generated power by the steam turbine, the lower η_P and the more increase of the heat generated. As in this Figure the rate of the efficiency η_{st} lowered to 10 % makes the value of η_P smaller by about 6% at t_d =70°C. This depends on the condensing temperature t_d of the heat pump and the decreasing rate of η_P is larger with the higher t_d .

5. Conclusions

The following results are obtained by analyzing the effect of the factors on the thermodynamic efficiencies in the steam turbine cycle combined with the heat pump, which is intended to reduce the thermal pollution caused locally and to raise the energy utilization efficiency.

(1) With a higher temperature of the heat generated by this combined cycle, which is the condensing temperature of the heat pump, the conversion efficiency η_P into the electrical power becomes smaller and instead the conversion efficiency η_H into the heat becomes larger. On the other hand, the exergy efficiency (effectiveness) ζ_{ex} , based on the 2nd law of thermodynamics, also becomes smaller. NH_3 and R11 as the thermal media of the heat pump cycle are considered here and the decreasing rate of η_P in case of NH_3 is larger than that of R11 and those of ζ_{es} of both media are almost the same.

(2) With a lower evaporating temperature of the heat pump the efficiencies η_P and ζ_{es} decrease and its tendency is larger than with a higher condensing temperature as mentioned above.

Analysis of the effect of the compressor internal efficiency η_{RC} of the heat pump on the efficiencies η_P and ζ_{es} shows that they decrease considerably with a smaller η_{RC} .

(3) As the internal efficiency η_{st} of the steam turbine is smaller, both η_P and ζ_{es} decrease linearly and the decreasing rate is larger with a higher condensing temperature t_d of the heat pump. At t_d = 70°C in case of NH_3 η_P becomes smaller by about 6% with a smaller internal efficiency η_{st} by 10 %. Furthermore, the influence of the pressure in the steam turbine condenser on the efficiencies η_P and ζ_{es} is small in this combined cycle.

(4) When one part of the heat generated by the heat pump condenser is used for heating the boiler feed water, the efficiency η_P and the exergy efficiency ζ_{es} become larger and η_P attains larger by 2.5% at t_d = 50°C in case of NH_3. This effect of the regenerative process is larger in case of NH_3 than R11.

Acknowledgement

The authors wish to thank Mr.TANAKA, who is a technician of their laboratory and the members in the Heat Engine I Laboratory of Hokkaido University and also staffs of the Computing Center of Hokkaido University for their cooperation.

References

(1) Mori, Y. and Shioda, S., Energy Conversion Engineering (in Japanese), (1975), Kyoritsu Shuppan Co.

(2) Kaisersot,K.,BWK,Vol.27, No.5(1975), p.219.

(3) Saito, T., Text of the 17th Course, (in Japanese), Japan Soc. Mech. Engrs. (Hokkaido District), (1977), p.84.

(4) Sonoda, T., et al., Preprint of Japan Soc. Mech. Engrs., (in Japanese) , No. 762-2 (1976-9), p.84.

(5) Sonoda, T., et al., Priprint of Japan Soc. Mech. Engrs., (in Japanese) , No. 775-1 (1977-3), p.46.

(6) Engineering Technical Division, Prospect for Energy Conversion Techniques (in Japanese), (1975), p.2.

(7) Sonoda, T., et al., The Thermal and Nuclear Power (in Japanese), Vol.27, No. 2 (1975-2), p.20.

(8) Sonoda, T., et al., Bull. JSME, Vol. 19, No. 127 (1976-1), p. 45.

(9) Berg, C.A., Mech. Eng., Vol. 96, No. 5 (1974-5), p. 30.

(10) Japan Soc. Mech. Engrs., Steam Table (1968).

(11) Tanishita, I., Engineering Thermodynamics, (in Japanese), (1964), Shokabo Co.

(12) Japan Association of Refrigeration , Handbook of Refrigeration and Air Conditioning (in Japanese), 1976