An Analysis of Gas-Steam Total Power Recovery Energy System

By

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1. INTRODUCTION

In recent years power recovery from the waste energy sources exhausted from various technological processes deserves more attention for achieving better energy conservation of the whole process, improving the environmental pollution and obtaining of extra power and electricity. The investigation and development of the power recovery technology and the related energy systems for such purpose become one of the practical aspect in thermal engineering science and energy industry. Generally speaking, waste energy power recovery system could be distinguished as high temperature, medium temperature and low temperature energy system according to the temperature range of the exhausted energy carrying working medium from the technological process. The high or medium temperature power recovery energy system are better developed and utilized as compared with the low temperature one due to their obvious energy conservation potential and econimcal profit.

Power recovery energy system usually employs a "process" of energy conversion thermodynamically when the exhausted working medium has a pressure potential, but has to follow a "cycle" of energy conversion when only temperature potential exists. The former kind of power recovery energy system usually consists of purification or particle separation system, power machinery train system including turbo expander, blower or motor/generator, special valves and control system as well as the waste heat boiler system for further utilizing the exhaust heat from the turbo expander. At present time the waste heat boiler usually generate low pressure steam for local technological use. In case some combustible gas is contained in the waste energy source, the supplementary or catalytic combustion system is added to further utilize the heat content and raise the initial temperature of the whole system. A "process" energy conversion is very effective from the thermodynamic point of view, because in this situation the expander efficiency, which is higher than any Carnot cycle for nowadays temperature of energy conversion, nearly represents the thermal or power recovery efficiency of the whole system, if the exhaust temperature is not too different from the surrounding temperature. In case waste boiler system is employed the total

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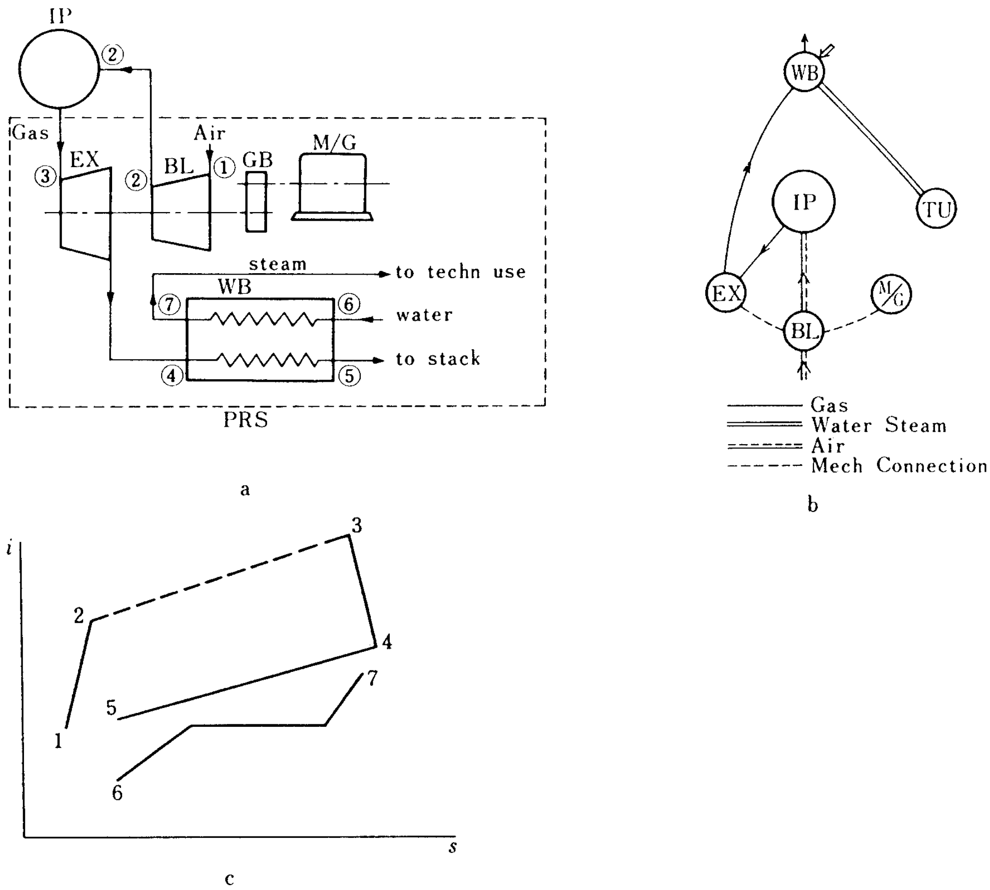

Fig. 1. Scheme of the conventional power recovery energy system (PRS).
EX=Expander, BL=Blower, GB=Gearbox, M/G=Motor/Generator, WB=Waste heat boiler, IP=Industrial process.

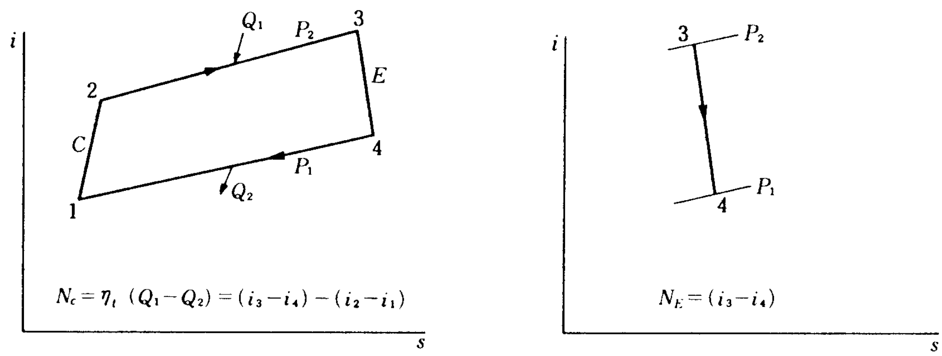

Fig. 2. Cycle energy conversion and process energy conversion.

power-plus-heat or energy utilization efficiency is high also (see Fig. 1 and 2).

One typical example in industrial application is that used in the fluidized catalytic cracker unit (FCCU) in petroleum industry. Flue gas from the regenerator of the unit with 2.5–3.5×10^5 Pa in pressure and 600–750°C in temperature could effectively generate power in such power recovery system. The main problem exists in the containment of solid catalytic particles in the flue gas, which will cause serious erosion and short operating life of power recovery turbo expander, since the transform of potential energy of particulate flue gas to mechanical one is in terms of high speed gas flow and deviations in the blade passages. However, by virtue of proper development of the separation system and the successful research and development of erosion withstanding turbo expander, the FCCU power recovery is now a practical mean of energy conservation in petroleum industry. The other remarkable example is the power recovery system for blast furnace top gas in metallurgical industry.

In the above example, the power-plus-heat or energy utilization efficiency of the energy system may achieve as high as 70% and the heat-electricity ratio is about 3. So still 30% of the heat energy entering into the system is exhausted to the surroundings and the heat-electricity ratio is too high and not controllable to meet heat electricity balance in the local plant. So question is then arised that could the power recovery efficiency be further raised and the heat-electricity ratio be decreased and controllable. One way of improving as discussed e.g., in Ref. 2 is to link or combine the power recovery energy system with other power machinery or system such as gas turbine engine system and in this case the exhaust from the turbo expander in the power recovery system could be lead to heat the air from the compressor of gas turbine before entering the combustion chamber and etc. Such improving means need certain amount of extra equipments and fuel and may not be good to industrial users from the mean and purpose of power "recovery".

In case where more electricity power is needed and no extra technological steam supply is asked for in certain period, a concept of TOTAL power recovery and energy system is suggested here incorperated with the illustrative example of FCCU power recovery. This not only could make more efficient power utilization of the waste energy contained in the exhausted flu gas i.e., produce the electricity wholly in high quality energy form, but also inproves the manoeuvrability of operation and the working conditions of the particulate flow turbo expander with no need of establishing extra equipments.

2. Description and Analysis of The Total Power Recovery Energy System

As shown in Fig. 3, the main idea of the suggested total power recovery energy system is to conduct the steam generated from the waste heat boiler to mix with the exhausted gas and form a gas-steam mixed working medium in the system. The steam pressure should be selected or regulated according to the pressure of the initial gas so as to guarantee the good mixing. Fig. 3a is the schematic representation of th energy system, 3b is the loop diagram of the different mediums and 3c is the process on i-s diagram. The steam mixes with the initial gas in the mixed chamber or pipe with a result that the gas-steam working medium will pass through the espander in a different temperature and also with decreased particle or containmination concentration. The exhaust temperature of the mixture from the expander will also differ from that of gas alone and thus influence the amount and temperature of the steam generated in waste heat boiler. Equilibrium operation condition is established finally. Thus all the

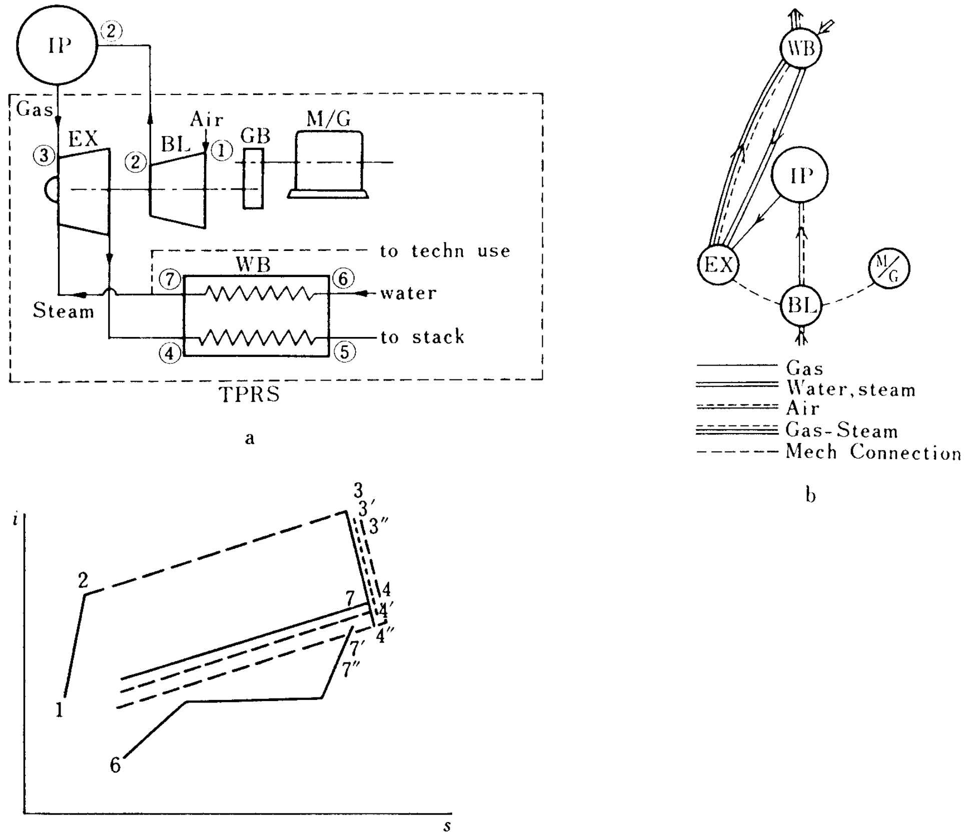

Fig. 3. Schme of the total power recovery energy system (TPRS).
EX=Expander, BL=Blower, GB=Gearbox, M/G=Motor/Generator, WB=Waste heat boiler, IP=Industrial process.

working medium goes through the expansion process and the potential energy is totally converted to power output minus only the loss in expander.

Let P_{go}, T_{go}, G_g represent the inlet initial gas pressure, temperature and mass flow rate respectively. Correspondingly P_{so}, T_{so}, G_s are those of steam generated and P_{mo}, T_{mo}, G_m represent the corresponding parameters after mixing. Assuming the gas and steam are perfect gas, the average thermodynamic properties of the mixture can be written as:

$$C_{pm}=\frac{C_{pg}G_g+C_{ps}C_s}{G_g+G_s}$$

$$k_m=\frac{k_G G_g+k_s G_s}{G_g+G_s}$$

$$i_m=C_{pm}(T_m-T_B)=\frac{G_g C_{pg}(T_{g0}-T_B)+G_s C_{ps}(T_{s0}-T_B)}{G_g+G_s}$$

where T_B is the common thermodynamic base temperature of calculation and according to the thermodynamic table of steam T_B=273 °K is used here. So the temperature of the mixed gas-steam is

$$T_{m0}=\frac{G_g C_{pg} T_{g0}+G_s C_{ps} T_{s0}}{G_g C_{pg}+G_s C_{ps}} \tag{1}$$

From the expansion process on $i-s$ diagram in Fig. 4, we have

$$ts=\frac{T^0_{m0}-T_{m2}}{T^0_{m0}-T'_{m2}}$$

and

$$P^0_{m0}=P_{m2}\left(\frac{T^0_{m0}}{T_{m2}}\right)^{k_m/(k_m-1)}$$

The steam generated is in a pressure higher than that of the initial gas and is injected into the mixed chamber under high speed. So after mixing, the total pressure of the mixture may be higher than that of initial gas i.e.,

$$P^0_{m0}=P^0_{g0}+\Delta P_m$$

If t_B denote the heat transfer temperature drop in boiler, then

$$T_{s0}=T'_{m2}-\Delta t_B$$

From the above relations the temperature of generated steam is found by

$$T_{s0}=T_m-\eta_{Ts} T_{m0}\left[1-\left(\frac{P_2}{P^0_{g0}+\Delta P_m}\right)^{(k_m-1)/k_m}\right]+\Delta t_B \tag{2}$$

Here we could take $\eta_{Ts}=\eta_T$. In the analysis we assume $P_{go}=P_{mo}$.
Again from heat equilibrium condition one get:

$$\eta_B(G_g+G_s)C_{pm}(T_{s0}+\Delta t_B-T_D)=G_s C_{ps1}(T_{s0}-T_p)+G_s[r+C_{ps2}(T_p-T_w)] \tag{3}$$

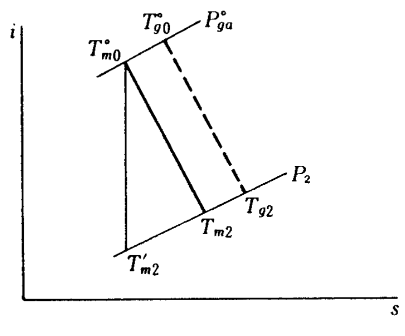

Fig. 4. Expansion process in turbo expander.

where η_B is the efficiency of waste heat boiler, T_D the temperature of exhaust from boiler, T_p the saturated temperature corresponding to steam pressure, C_{psl} the average specific heat of steam, C_{ps2} average specific heat from hot water temperature to saturated temperature and r is the latent heat of vaporization.

The above (1), (2) and (3) equations are enough for solving the three unknowns G_s, T_{mo} and T_{so}. Substituting equation (2) into (1) and equating with (3), the final steam temperature in stable or equilibrium condition is given by:

$$T_s^0=\frac{(DF+CI-EH-BJ)+\sqrt{(DF+CI-EH-BJ)^2-4(EF-CJ)(BI+HD)}}{2(EF-CJ)} \quad (4)$$

in which

$$A=\frac{1}{1-\eta_{TS}\left[1-\left(\frac{P_2}{P_{g0}^0+\Delta P_m}\right)^{(k_m-1)/k_m}\right]},$$

$$B=G_gC_{pg}(T_{g0}-A\Delta t_B), \qquad C=G_gC_{pg}A,$$

$$D=C_{ps}A\Delta t_B, \qquad E=C_{ps}(A-1),$$

$$F=\eta_BG_gC_{pg}, \qquad H=\eta_BG_gC_{pg}(T_D-\Delta t_B),$$

$$I=\eta_BC_{pg}(T_D-\Delta t_B)+(r+T_p\Delta C_{ps}-C_{ps2}T_w),$$

$$J=C_{ps1}-\eta_BC_{pg}$$

and

$$\Delta C_{ps}=C_{ps2}-C_{ps1}$$

Fig. 5 and Fig. 6 show the variation of calculated steam temperature, mixed inlet gas-steam temperature and steam flow rate with the original waste gas parameters T_{go} and P_{go} for a small FCCU power recovery system if it is changed to total power recovery energy system. The data used in the calculation is:

$$G_g=12.1\ \text{kg/s}, \qquad P_2=1.05\times10^5\ \text{Pa}, \qquad C_{pg}=1.214\ \text{J/kg °C},$$

$$C_{ps}=2.093\ \text{kJ/kg °C}, \qquad \Delta t_B=50\ \text{°C}, \qquad t_D=140\ \text{°C},$$

$$\eta_B=0.8, \qquad \eta_T=0.8$$

It can be seen from the figures that the steam mass flow rate and its temperature decrease with the increase of initial waste gas pressure, but increase with the increase of gas temperature.

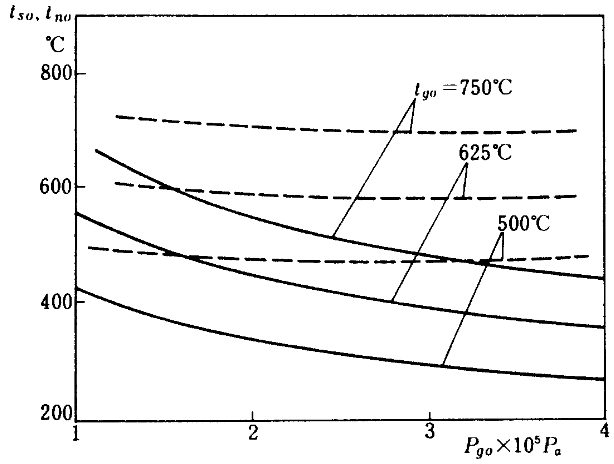

Fig. 5. Variation of $\genfrac{}{}{0pt}{}{t_{so}}{t_m}$, t_{mo} with $\genfrac{}{}{0pt}{}{P_{go}}{t_s}$ and t_{go}.

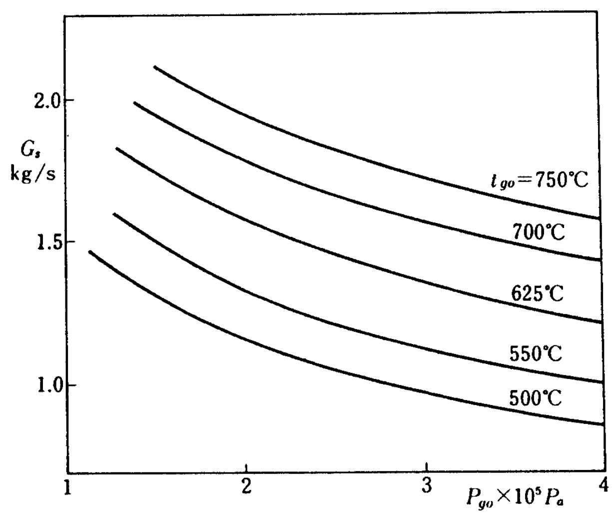

Fig. 6. Variation of G_3 with P_{go} and t_{go}.

2. Comparison With Conventional System

In order to explain the feasibility of the proposed system a comparison is made between this system and the conventional system of generating electricity by passing the steam through steam turbine and condensor i.e., a tandem gas-steam combined cycle which is schematically shown in Fig. 7.

The criteria for comparison is defined as ratio of relative increase of power output and is denoted by R,

$$R=\frac{N_g+N_s}{N_g} \tag{5}$$

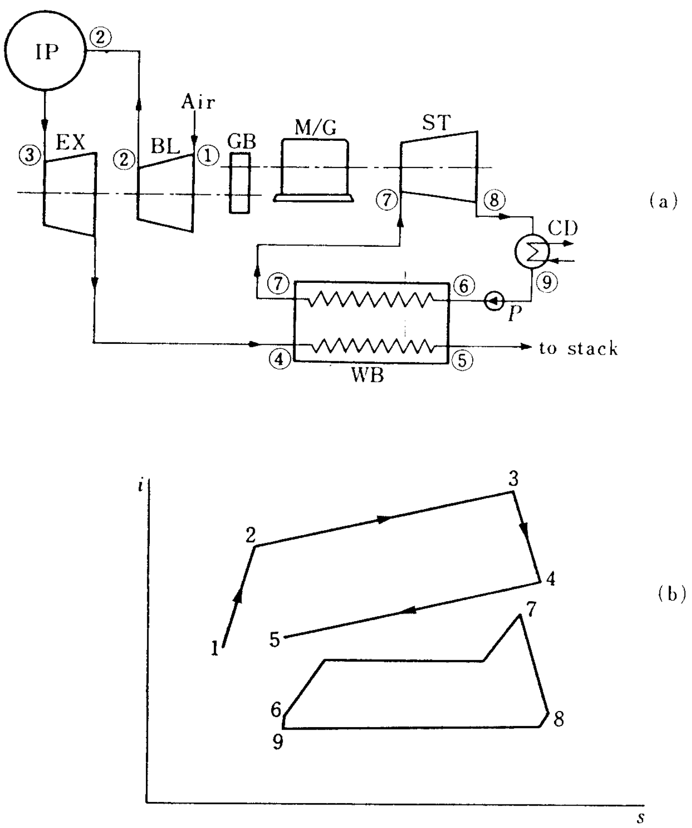

Fig. 7. Scheme of the conventional conbined steam power recovery energy system.
EX=Expander, BL=Blower, GB=Gearbox, M/G=Motor/Generator, WB=Waste heat boiler, IP=Industrial process ST=Steam turbine, CD=Condenser, P=Pump.

1) For the suggested total energy system, after mixing with steam the effective power output is

$$N_{(g+s)}=(G_g+G_0)C_{pm}T_{m0}\left[1-\left(\frac{P_2}{P^0_{g0}+\Delta P}\right)^{(k_m-1)/k_m}\right]\eta_{TS}-N_p \tag{6}$$

where N_p is the power required for driving pump and can be roughly estimated by

$$N_p=G_s(P_{0s}-P_2)/427\gamma_m\eta_p$$

Now let $\eta_T=\eta_{TS}$, the power increase ratio will be

$$R_G=\frac{N_{(g+s)}}{N_g}=\left(1+\frac{G_sC_{ps}}{G_gC_{pg}}\right)\frac{T_{m0}}{T_{g0}}-\frac{G(P_{0s}-P_2)}{\gamma_m\eta_pG_gC_{pg}T_{g0}\left[1-\left(\frac{P_2}{P^0_{g0}}\right)^{(k_g-1)/k_g}\right]} \tag{7}$$

2) For conventional combined steam cycle system, the power output of steam turbine can be estimated by

$$N_s = Q_s \eta_{sc} \tag{8}$$

where Q_s is the total heat energy input to the waste heat boiler for steam generation,

$$Q_s = G_g C_{pg}(T_{g2} - T_D) = G_g C_{pg} T_{g0} \left\{ 1 - \eta_T \left[1 - \left(\frac{P_2}{P^0_{g0}} \right)^{(k_g - 1)/k_g} \right] - \frac{T_D}{T_{g0}} \right\} \tag{9}$$

η_{sc} is the thermal efficiency of the steam power plant at the given initial steam parameters. The power increase ratio is then

$$R_s = \frac{N_g + N_s}{N_g} = 1 + \frac{\eta_{sc}}{\eta_T} \frac{1 - \eta_T \left(1 - \left(\frac{P_2}{P^0_{g0}} \right)^{(k_g - 1)/k_g} \right) - \frac{T_D}{T_{g0}}}{1 - \left(\frac{P_2}{P^0_{g0}} \right)^{(k_g - 1)/k_g}} \tag{10}$$

Finally we use R_N to denote the relative power increase effect of the total power recovery energy system

$$R_N = \frac{R_s}{R_G} = \frac{(G_g C_{pg} + G_s C_{ps}) T_{m0} \left[1 - \left(\frac{P_2}{P^0_{g0} + \Delta P} \right)^{(k_m - 1)/k_m} \right] \eta_{Ts} - \frac{G_s (P_{0s} - P_2)}{427 \gamma_m \eta_p}}{G_g C_{pg} T_{g0} \left\{ \left[1 - \left(\frac{P_2}{P^0_{g0}} \right)^{(k_g - 1)/k_g} \right] \eta_T + \left[1 - \eta_T \left(1 - \left(\frac{P_2}{P^0_{g0}} \right)^{(k_g - 1)/k_g} \right) - \frac{T_D}{T_{g0}} \right] \eta_{sc} \right\}} \tag{11}$$

Fig. 8 and Fig. 9 are the calculated results according to the above expressions for the same FCCU power recovery system. η_{sc} for low pressure steam is taken as 0.15 averagely in the calculation.

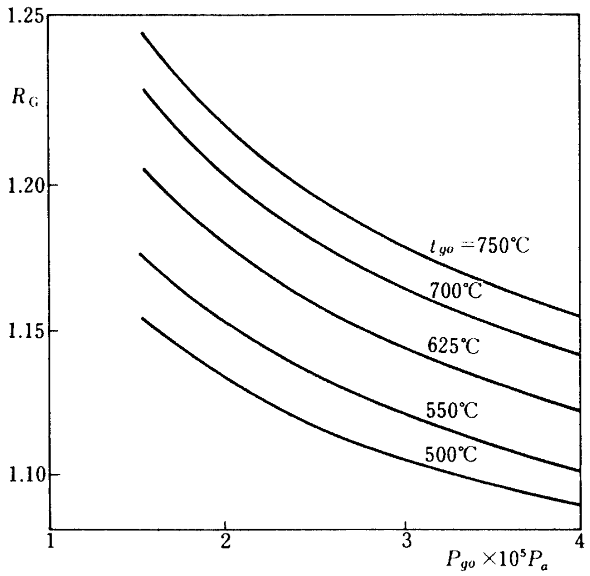

Fig. 8. Variation of relative power output increase R_G with P_{go} and t_{go} for TPRS.

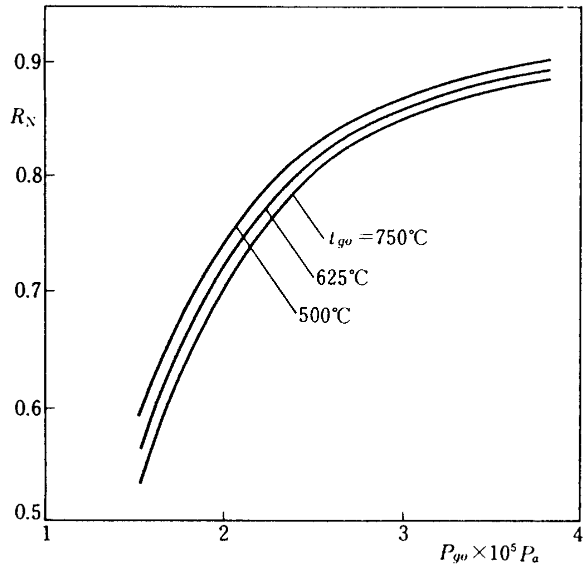

Fig. 9. Variation of ratio of relative power output increase R_N with P_{go} and t_{go}.

4. Results and Conclusion

1) The suggested total power recovery energy system can effectively increase the power output. The power increase ratio R_G increases with the increase of initial waste gas temperature T_{go}. For the calculated example of FCCU power recovery energy system, R_G achieves 1.16 at the design operation point. This greatly increases the power recovery efficiency of the unit.

2) It is more effective in increasing power output in comparison with the conventional combined steam cycle system (depending on magnitude of η_{sc}) under usual waste gas initial temperature as R_N is less than one and in the design point of the example R_N is nearly 0.75. Besides, the initial parameters of the mixed gas-steam could be regulated through steam throttling or by-pass vavles to improve manoeuvrability of operation and to match the variation of loading of FCCU.

3) After mixing, the concentration of the particle content lowers down, becuase of the increase of working medium flow rate. For the given example, the flow rate increases by 1.124, so this is equivalent to decrease of the concentration to 81% of original value. In cooperation with the decreasing of the initial temperature, this improves the operating condition of the expander, increases its life time and reliability.

4) If adding before expander an injector-compressing system, the gas-steam pressure could be further raised as usually the steam pressure is much higher than the gas. This will further increase R_G and power output and remain the steam in a pressure suitable simultaneously for technological use.

NOMENCLATURE
C_p=specific heat at constant pressure
G=mass flow rate
i=enthalpy
k=specific heat ratio
P=pressure
Q=amount of heat input to the boiler
r=latent heat of vaporization
t=temperature in °C
T=temperature in °K
η=efficiency
γ=density

Footnote
0=at inlet of expander
2=at exhaust of expander
b=boiler
g=initial waste gas
m=mixture of gas and steam
p=pump
s=steam
t=turbo expander
sd=steam cycle

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