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Studies on Carbon Dioxide Cycles for Power Generation*

(1st Report, Fundamental Condensation Cycles)

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Further improvements in thermal efficiency of steam and nuclear power plants are an important problem in the present energy crisis. This report presents the performance characteristics of two fundamental condensation cycles employing carbon dioxide as a working medium. The effects of cycle parameters on thermal efficiency in non-reheat and reheat condensation cycles and values of optimum reheat pressures are discussed. Furthermore, losses in each component in different CO₂ cycles by utilizing the exergy concept based on the second law of thermodynamics were analyzed and the effects of regenerating on irreversible losses were discussed. As the result, it was shown that at turbine inlet pressure of about 250 ata and turbine inlet temperatures higher than 650 °C CO₂ condensation cycles in double- or triple-reheating give a better performance than conventional steam power-plant cycles.

1. Introduction

Developments of alternative energy resources to fossile fuels and methods for economy of energy are the very important and urgent problems in the present energy crisis. One of the methods for economy of energy is to improve the thermal efficiency of thermal engines which convert the thermal energy of fossile fuels and nuclear fuels to mechanical and electrical energy. But usual steam power plants, internal-combustion engine plants, and gas turbine plants at the present time have been developed at a level where a prominent increase of the cycle efficiency can not be expected. Therefore, studies on a power plant using an alternative fluid to water as the working medium, with much higher efficiency, are necessary.

He and CO₂ have been proposed as the working fluids of a closed-cycle gas turbine system combined with gas-cooled reactors at the earlier World Power Conference. In the past the performance of cycles and designs of turbine and heat exchangers of CO₂ power plants have been investigated by Strub & Keller⁽¹⁾, Angelino⁽²⁾⁽³⁾, Watzel⁽⁴⁾, Pfost⁽⁵⁾, Smal'ko⁽⁶⁾ and Gokhshtein⁽⁷⁾⁽⁸⁾. Angelino has investigated the effect of maximum pressure and temperature on performance of several CO₂ condensation cycles. This may be the only report concerning the thermal efficiency of CO₂ cycles.

Therefore, there are remarkably few data available on cycle performance for planning the plants. Either, the effects of cycle parameters such as component efficiencies and pressure losses on the cycle efficiency and the performance of reheat cycles have not been made clear.

This report presents the effects of cycle parameters on performance of the non-reheat and the reheat condensation cycles employing CO₂ as the working fluid. Furthermore, an analysis of losses in the components by the exergy concept based on the second law of thermodynamics was conducted for CO₂ condensation cycles.

The distinctive advantages of CO₂ plants are as follows;

- (i) The efficiency of CO₂ cycle at a constant turbine inlet temperature is 7 ~ 9 % higher than that of gas turbine cycle using He as the working fluid. At turbine inlet temperatures above 650 °C the cycle efficiency also is higher than that of steam power cycle.
- (ii) The size of CO₂ turbine becomes reduced as compared with the steam power turbines owing to the smaller specific volume.
- (iii) The thermodynamic properties of CO₂ are comparatively stable and CO₂ is non-inflammable.

2. Configuration of non-reheat condensation cycles

A schematic diagram of the arrangement of two cycles and a T-S diagram are shown in Figs. 1(a), (b) and (c). Fig. 1 (a) shows a single-regenerative cycle and the process is ; points 5 ~ 1 : cooling and condensation by means of external medium (condenser W), 1 ~ 2 : liquid-phase compression (pump P), 2 ~ 6' : heat-exchange in regenerator R, 6' ~ 3 : heating by means of external heat sources (heater H), 3 ~ 4 : expansion in turbine T and 4 ~ 5 : heat exchange in R. The cycle is expressed as lines 1-2-6'-3-4-5-1

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on the T-S diagram in Fig. 1 (c).

(a) Single-regenerative cycle

(b) Double-regenerative cycle

(c) T - S diagram

H : Heater, R : Regenerator,
T : Turbine, C : Compressor,
W : Condenser, G : Generator

Fig. 1 Diagrammatic arrangement of CO₂ cycle and T-S diagram

Fig. 1 (b) shows a double-regenerative cycle presented by Angelino⁽³⁾. The exhaust gas from turbine T is cooled in a high-temperature and a low-temperature regenerator R2, R1, and flows into sections; a fractional flow rate α to condenser W and a fractional flow rate $(1 - \alpha)$ to compressor C. The liquid which has been condensed in condenser W is fed by pump P, and enters the low-temperature regenerator R1. Then, it changes into gas-phase and flows into the high-temperature regenerator R2, together with the fractional flow rate $(1 - \alpha)$ compressed in compressor C, and is heated to a prescribed temperature in heater H. The cycle is expressed as lines 1-2-7-6-3-4-8-5-1 in Fig. 1 (c). That is, the purpose of this cycle is to decrease the irreversible loss caused by the temperature difference in the regenerator R.

Nomenclature

- A : heat equivalent of work, kcal/kgm
 α : condensed flow rate fraction, (kg/s)/(kg/s)
 $\epsilon_B, \epsilon_{RF}, \epsilon_{RE}$: dimensionless fractional pressure losses
 η_{th} : thermal efficiency
 η_T, η_P, η_C : over-all turbine, pump and compressor efficiencies
 Δi : cold end (minimum) enthalpy difference, kcal/kg
 L_T, L_P, L_C : works in turbine, pump and compressor, (kgm/s)/(kg/s)
 Q_1, Q_2 : heat supplied and heat wasted (kcal/s)/(kg/s)
 Δt : cold end (minimum) temperature difference, °C

Subscripts

- 1, 2, 3, ... : refer to each point of cycles in Fig. 1 for i and t, or turbine inlet, condensation and reheat pressure for P
 adi : adiabatic

3. Comparison of thermal efficiency between the single- and the double-regenerative cycles

Thermal efficiency of the two cycles in Fig. 1 (a) and (b) can be expressed from heat-balance calculations as follows, For single-regenerative cycle ;

$$\eta_{th,1} = 1 - Q_2/Q_1 = 1 - \frac{AL_p + \Delta i}{AL_T + \Delta i} \quad \dots\dots\dots (1)$$

For the double-regenerative cycle ;

$$\eta_{th,2} = 1 - \alpha Q_2/Q_1 = 1 - \frac{AL_c + \Delta i_2}{AL_c + \Delta i_1} \frac{AL_p + \Delta i_1}{AL_T + \Delta i_2} \quad \dots\dots\dots (2)$$

where

$$\alpha = \frac{i_8 - i_5}{i_7 - i_2} = \frac{AL_c + \Delta i_2}{AL_c + \Delta i_1} \quad \dots\dots\dots (3)$$

From Eqs. (1) and (2) the thermal efficiency difference $\Delta \eta_{th}$ between the two cycles is derived as,

$$\Delta\eta_{th} = \eta_{th,2} - \eta_{th,1} = \frac{A(AL_p + \Delta i_1)(\Delta i_1 - \Delta i_2)(L_T - L_c)}{(AL_T + \Delta i_1)(AL_T + \Delta i_2)(AL_c + \Delta i_1)} \dots\dots\dots (4)$$

Substitution $\Delta i_1 - \Delta i_2 = (1-\alpha)(AL_c + \Delta i_1)$ into Eq. (4) gives

$$\Delta\eta_{th} = \frac{A(AL_p + \Delta i_1)(1-\alpha)(L_T - L_c)}{(AL_T + \Delta i_1)(AL_T + \Delta i_2)} \dots\dots\dots (5)$$

Therefore, the difference of thermal efficiency between the above two cycles is affected by the value of L_T (L_c) relative to that of L_C (L_T).

$$\eta_{th,2} \geq \eta_{th,1} \leftrightarrow L_T \geq L_c \dots\dots\dots (6)$$

Here, substitution of $L_T = \eta_T L_{T,adi}$ and $L_C = L_{C,adi} / \eta_C$ into Eq. (7) gives,

$$\eta_{th,2} \geq \eta_{th,1} \leftrightarrow \eta_T \eta_C \geq \left(\frac{L_c}{L_T} \right)_{adi} \dots\dots\dots (7)$$

Accordingly, it is clear that thermal efficiency of the double-regenerative cycle $\eta_{th,2}$ becomes higher than that of single-regenerative cycle $\eta_{th,1}$ when Eq. (7) holds. Also, the cycle efficiency is improved further by making η_C higher and Δt_2 lower.

4. Performance characteristics of the non-reheat cycles

The performance characteristics of the above cycles, that is, the effects of cycle parameters on thermal efficiency are calculated by utilizing the data on the thermal properties of CO_2 which are derived from Vukalovich & Altunin⁽⁹⁾. The assumptions are as follows; (i) $\eta_T = 90\%$, $\eta_C = \eta_p = 85\%$, (ii) $\Delta t = \Delta t_1 = 15^\circ C$, (iii) $P_2 = 50$ ata, (iv) the pressure drop is neglected. Also, the radiant heat loss is neglected in all calculations.

4.1 The effects of turbine inlet pressure P_1 and temperature t_3

The effects of P_1 and t_3 on η_{th} are shown in Fig. 2. Two-dot-dash lines in the figure show the condition that turbine exhaust temperature t_4 is equal to regenerator exit temperature, t_5 in Fig. 1 (a) or t_8 in Fig. 1 (b). For $P_1 \leq 200$ ata in the single-regenerative cycle and $P_1 \leq 150$ ata in the double-regenerative cycle, η_{th} increases rapidly with P_1 and then becomes flat and takes a maximum value, which is shown by a circle, and then decreases very slowly. Thus the maximum value of η_{th} is little affected by turbine inlet pressure P_1 . For example, for $t_3 = 600^\circ C$ in the double-regenerative cycle maximum thermal efficiency $\eta_{th,max}$ is 49.6% at $P_{1,opt} = 320$ ata, and 49.3% at $P_1 = 250$ ata, that is, its difference, 0.3%, is small. Thus a higher thermal efficiency is attainable at about 250 ata in CO_2 condensation cycles.

On the other hand, the thermal efficiency increases remarkably with turbine inlet temperature t_3 . For example, as t_3 rises from $600^\circ C$ to $700^\circ C$ for $P_1 = 250$

Fig. 2 The effects of turbine inlet pressure P_1 and temperature t_3

ata, η_{th} is improved approximately 4.1% from 49.3% to 53.4% in the double-regenerative cycle and 3.4% from 42.4% to 45.8% in the single-regenerative cycle. This effect is explained by Eqs. (1) and (2). In Eqs. (1) and (2) the waste heat αQ_2 (Q_2) depends only on P_1 , not on t_3 . In the equation, $Q_1 = AL_T + \Delta i_2$ (Δi), then Δi_2 (Δi) is not affected by t_3 , but L_T increases with t_3 . Therefore, η_{th} is improved with an increasing t_3 . Improvement of η_{th} can be expressed, assuming that Q_1 changes to kQ_1 ($k > 1$) by raising t_3 , as follows;

$$\Delta\eta_{th} = \left(1 - \frac{\alpha Q_2}{kQ_1} \right) - \left(1 - \frac{\alpha Q_2}{Q_1} \right) = \frac{\alpha Q_2}{Q_1} \left(1 - \frac{1}{k} \right) \dots\dots\dots (8)$$

4.2 The effect of condensation pressure P_2

Condensation pressure P_2 is influenced by temperature of the cooling medium. As the critical temperature of CO_2 is as low as $31.05^\circ C$ (the critical pressure is 75.3 ata), low temperature of the cooling medium is needed to condense the CO_2 gas. When the temperature of the cooling medium is about $28^\circ C$, the condensation pressure P_2 is approximately 70 ata, and when a large volume of cooling water in winter season is available, P_2 can be lowered to 60 ata for $22^\circ C$ and 50 ata for $14^\circ C$. The effect of P_2 on the performance of the cycles is shown in Fig. 3 for $P_1 = 300$ ata, $t_3 = 600^\circ C$ and $P_1 = 400$ ata, $t_3 = 800^\circ C$. The thermal efficiency is improved by (0.6 ~ 1.0)% with a decrease of 5 ata in P_2 .

4.3 The effects of η_T , η_C and η_p

The effects of η_T and η_C on thermal efficiency are shown in Fig. 4 for $P_1 = 250$ ata and $t_3 = 800^\circ C$. It is seen that the thermal efficiency decreases by (0.3 ~ 0.4)% in the single-regenerative cycle and (0.3 ~ 0.4)% in the double-regenerative cycle for a decrease of 1% in η_T in the

range of η_T from 60 to 100 %. For a decrease of 1 % in η_C it decreases by 0.1 % in the range $60 < \eta_C < 90$ %. And the effect of η_T on η_{th} is larger than that of η_C .

As regards the effect of η_p , since the pump work is small owing to the liquid-phase compression, the decrease of η_{th} is as slight as 0.05 %, for the decrease of $\eta_p = 1$ %.

Also, for some values of η_T and η_C , for example, for $\eta_C < 40$ % at $\eta_T = 80$ %, η_{th} in the double-regenerative cycle is lower than in the single-regenerative cycle, and the reason can be explained by the relationship of Eq. (6) mentioned in Chap. 3.

4.4 The effects of Δt_1 (Δt) and Δt_2

The effects of Δt_1 (Δt) and Δt_2 on performance of the cycles are shown in Fig. 5 for $P_1 = 300$ ata and $t_3 = 600$ °C. In the figure the constant total surface of the regenerators per specific outlet 1 kw in the double-regenerative cycle is shown by two-dot-dash lines, calculated by the following assumptions; (i) the type is a counter-flow concentric cylindrical heat-exchanger with 20 mm inner cylinder diameter, (ii) the mean velocity on both sides is 10 m/s, (iii) the heat transfer coefficients are given by $Nu = 0.023Re^{0.8}Pr^{0.4}$.

The over-all heat transfer coefficients are (800 ~ 1000) kcal/m²h°C in the low-temperature regenerator R1 and (600 ~ 800) kcal/m²h°C in the high-temperature R2. This figure shows that higher efficiency can be obtained by selecting a value of Δt_1 lower than that of Δt_2 for a constant regenerator-surface. The value of η_{th} decreases linearly when Δt_1 (Δt) or Δt_2 increases by (0.9 ~ 1.2) % with $\Delta t_1 = 10$ °C, and by (0.9 ~ 0.4) % with Δt_2 .

4.5 The effect of the pressure losses

The pressure drops in the components were neglected in the calculations described above. Then the effect of the pressure losses on performance of the cycles is shown in Fig. 6 for $P_1 = 300$ ata and $t_3 = 600$ °C. This was evaluated as follows. The pressure drops are ΔP_B , ΔP_{RF} and ΔP_{RE} , respectively in the heater H, the high- and the low-pressure section in regenerator R1,

Fig. 3 The effect of condensation pressure P_2

R2, and expressed as the dimensionless values of $\epsilon_B = (\Delta P_B/P_1) \times 100$ %, $\epsilon_{RF} = (\Delta P_{RF}/P_1) \times 100$ % and $\epsilon_{RE} = (\Delta P_{RE}/P_2) \times 100$ % obtained by dividing each pressure drop ΔP by P_1 and P_2 . It is shown that η_{th} decreases linearly with an increasing ϵ_B , ϵ_{RF} or ϵ_{RE} , and both $\eta_{th,1}$ and $\eta_{th,2}$ decrease by about 0.07 % for the rise of $\epsilon_B = 1$ %, and for the rise of ϵ_{RF} or $\epsilon_{RE} = 1$ % $\eta_{th,1}$ decreases by 0.3 % and $\eta_{th,2}$ by 0.5 %.

4.6 Specific output and volumetric flow rate

The specific output per turbine inlet flow rate 1 kg, the heat drop in compressor and pump, and value of α are shown in Fig. 7. The specific output increases with P_1 and t_3 , and it is 39.62 kcal/kg ($AL_T = 55.81$, $AL_C = 31.24$, $AL_p = 8.61$ and $\alpha = 0.665$) for $P_1 = 300$ ata and $t_3 = 600$ °C and about 1/8 as large as that of steam power plants whose condensation pressure is 0.05 ata. Therefore, the mass flow rate per constant specific output is 8 times as large as that of steam plants.

The volumetric flow rates at the turbine inlet and the turbine exit are shown against P_1 and t_3 in Fig. 8. It is seen from this figure that the volumetric flow rate in the CO₂ cycles per specific output 1 kw is 0.127 m³/h, which is 4 times

Fig. 4 The effects of η_T and η_C

Fig. 5 The effects of Δt_1 (Δt) and Δt_2

as large as that of H_2O , $60.36 \text{ m}^3/\text{h}$. Therefore, the leakage-loss at the turbine blade tip is smaller in the high-pressure stages and the blade length in the low-pressure stages is much smaller than those of H_2O . Therefore, CO_2 has the favorable properties for size and construction of the turbine.

5. Reheat cycles

Reheating is a useful means of improving thermal efficiency. Then, the effects of the cycle parameters on the performance of the CO_2 reheat cycles are presented. The calculations were conducted with the same assumptions as described in the previous Chapter.

5.1 The effects of reheat pressures P_3 , P_4 and reheat temperature t_R

5.1.1 Single reheating

The effect of reheat pressure P_3 on η_{th} is shown against reheat temperature t_R in Fig. 9 for $P_1 = 300 \text{ ata}$ and $t_3 = 600^\circ\text{C}$. η_{th} is improved from 49.5 % in the non-reheat cycle to a maximum value of 52.2 % at an optimum reheat pressure $P_{3, \text{nopt}}$, 175 ata, and the value of maximum thermal efficiency decreases by (0.3 ~ 0.6) % with a decrease of the reheat temperature t_R of 20°C . Optimum reheat pressure $P_{3, \text{nopt}}$ for maximum η_{th} is shown by a dotted line in the figure. It is about 123 ata for $t_R = 600^\circ\text{C}$, and 100 ata for 400°C , and decreases with a decrease of reheat temperature t_R .

5.1.2 Double reheating

The relation between η_{th} and reheat pressures P_3 , P_4 is shown in Fig. 10 for $P_1 = 300 \text{ ata}$ and $t_3 = t_R = 600^\circ\text{C}$. It is seen that a maximum value of η_{th} is 52.2

Fig. 6 The effect of pressure drop

% in single-reheating and η_{th} is improved by approximately 1 % in double-reheating. Also, optimum reheat pressures at the 1st and 2nd stages, $P_{3, \text{nopt}}$, $P_{4, \text{nopt}}$, are 165 ata and 90 ata, respectively.

The effect of reheat temperatures on performance of the double-reheating cycle is shown in Fig. 11 for $P_1 = 300 \text{ ata}$, $P_3 = 165 \text{ ata}$ and the same 1st-reheat temperature as t_3 . It is seen that the optimum 2nd-reheat pressure $P_{4, \text{nopt}}$ is about 90 ata when 2nd-reheat temperature is equal to turbine inlet temperature t_3 , and it decreases by (3 ~ 10) ata when the 2nd-reheat temperature decreases by 20°C . Also, η_{th} decreases by (0.2 ~ 0.4) %.

Fig. 7 Mechanical work and condensed flow rate fraction α

Fig. 8 Volumetric flow rate

5.2 The effects of turbine inlet pressure P_1 and temperature t_3

In single-reheating for $t_3 = t_R = 600^\circ\text{C}$, the relation between reheat pressure P_3 and η_{th} is shown against turbine inlet pressure P_1 in Fig. 12. The optimum reheat pressure $P_{3,opt}$ is given as dotted line. It is about 125 ata for $P_1 = 300$ ata and 140 ata for 400 ata, and increases with P_1 . Improvement in η_{th} of single-reheating relative to that of non-reheating is shown against P_1 and t_3 in Fig. 13. Here the reheat pressures P_3 are determined from the equation of $P_3 = \sqrt{P_1 P_2}$ (see Eq. (12)) and it is assumed that the reheat temperature t_R is equal to the turbine inlet temperature t_3 . The higher P_1 is and the lower t_3 is, the better is the improvement of η_{th} . Therefore, the optimum turbine inlet pressure $P_{1,opt}$ in the reheat cycles increases over that in the non-reheat cycles. For example, $P_{1,opt}$ is about 320 ata in non-reheating and 400 ata in single-reheating for $t_3 = 600^\circ\text{C}$. And $P_{1,opt}$ increases from about 390 ata in non-reheating to about 450 ata in single-reheating. But the effect of $P_{1,opt}$ on η_{th} is small, that is, the change of η_{th} is 0.2 % at the maximum by the deviation of 50 ata from $P_{1,opt}$.

5.3 The effect of condensation pressure P_2

The effect of P_2 on η_{th} is shown a-

Fig. 9 The effect of reheat pressure P_3 and reheat temperature t_R

Fig. 10 The effect of reheat pressures P_3 , P_4 (double-reheating)

gainst reheat pressure P_3 in Fig. 14 for $P_1 = 300$ ata and $t_3 = t_R = 600^\circ\text{C}$ in single-reheating. Optimum reheat pressure $P_{3,opt}$ increases with P_2 as shown in a dotted line. $P_{3,opt}$ increases from about 100 ata to about 150 ata as P_2 increases from 35 ata to 70 ata, and η_{th} decreases by (0.7 ~ 2) % as condensation pressure P_2 increases by 5 ata.

5.4 Selection of optimum reheat pressure

It was shown in the above sections that the higher P_1 , P_2 , and t_R were, the higher was the optimum reheat pressure. In this section, the optimum pressure is discussed.

Fig. 11 The effect of 2nd-reheat pressure P_4 and 2nd-reheat temperature

Fig. 12 The effect of turbine inlet pressure P_1 (single-reheating)

In Eqs. (1) and (2), α , Q_2 , L_C and L_p become constant when the cycle parameters except t_3 , the reheat pressure, t_R and η_T are assumed to be constant. And Q_1 is expressed from heat-balance calculations as follows;

$$Q_1 = AL_T + \alpha Q_2 - (1-\alpha)AL_C - AL_p \dots (9)$$

where $\alpha = 1$ in the single-regenerative cycle. Accordingly Q_1 , that is, η_{th} is dependent only on L_T . Then the optimum reheat pressure is equal to the reheat pressure for the maximum L_T .

Assuming that the working fluid is a perfect gas, the total turbine work is given by

$$AL_{T,total} = c_p \sum_{n=1} \eta_{T,n} T_n \{1 - (P_n/P_{n+1})^{-m}\} \dots (10)$$

where T is inlet temperature in each stage, $m = (\kappa - 1)/\kappa$, κ is the specific heat ratio and the subscript n refers to each reheat stage. Then, the optimum reheat pressure is given by a relation $(\partial AL_{T,total} / \partial P_n) = 0$,

$$P_{n,opt} = \sqrt[m]{\frac{\eta_{T,n}}{\eta_{T,n-1}} \frac{T_n}{T_{n-1}}} \sqrt{P_{n-1} P_{n+1}} \dots (11)$$

Fig. 13 Improvement of thermal efficiency η_{th} by single-reheating

Fig. 14 The effect of condensation pressure P_2

Here, when η_T and T_n are assumed to be equal in each stage, the following equations are obtained⁽¹⁰⁾.

For single reheating,

$$P_{3,\eta opt} = \sqrt{P_1 P_2}$$

For double reheating,

$$P_{3,\eta opt} = \sqrt[3]{P_1^2 P_2}, \quad P_{4,\eta opt} = \sqrt[3]{P_1 P_2^2} \dots (12)$$

The values of $P_{3,\eta opt}$ derived from Eq. (12) are shown in Fig. 15 in comparison with the calculated values presented in the previous section. It is seen from the figure that the maximum deviation of the values by Eq. (12) from the actual values is about 10 ata for $P_1 = (400 \sim 600)$ ata. But the effect of the deviations of reheat pressure P_3 from $P_{3,\eta opt}$ causes only a change of 0.1 % in η_{th} at the maximum. Therefore, Eq. (12) can be used within a reasonable error.

5.5 The effect of the number of reheats

The effect of the number of reheat stages on η_{th} is shown in Fig. 16 for the condition of the same reheat temperature

Fig. 15 The optimum reheat pressure $P_{3,\eta opt}$

Fig. 16 The effect of number of reheat on η_{th}

t_R as t_3 for $P_1 = 300$ and 600 ata. Here, the reheat pressures are determined from Eq. (12). The higher P_1 is and the lower t_3 is, the greater is the rate of the improvement in η_{th} . The improvement in η_{th} is about (2 ~ 3) % in single-reheating, (0.5 ~ 1) % in double-reheating and (0.3 ~ 0.5) % in triple-reheating for $P_1 = 300$ ata. Thus, the performance gains in reheats diminish rapidly with an increasing number of reheats.

Fig. 17 Comparison of thermal efficiency of CO₂ cycles with that of steam and perfect-gas cycles

6. Comparison of thermal efficiency η_{th} with that of steam and perfect-gas turbine cycle

A comparison of the performance of the CO₂ cycles with that of the steam and gas-turbine cycles is shown in Fig. 17. It is seen from the figure that the CO₂ condensation cycles give a better efficiency than the reheat steam cycle at the turbine inlet temperatures above 650 °C, and the higher t_3 is, the larger becomes the superiority. Namely, the performance of the CO₂ cycles is improved at a rate of (3 ~ 5) % per a 100 °C rise in t_3 in the same way as with the perfect gas-turbine cycles. On the other hand, the improvement of the steam cycles is about half that of the CO₂ cycles.

7. The analysis of losses in the component by exergy concept

While an analysis by heat-balance calculation based on the first law of thermodynamics gives many useful information on cycle performance, it does not give the information on the cause of each loss in a cycle. Then, a quantitative analysis of losses in each component in the CO₂ cycle by utilizing the concept of exergy⁽¹¹⁾ based on the second law of thermodynamics was made. In the analysis⁽¹²⁾ it is assumed that the exergy supplied to heater is equal to the supplied heat Q_1 . As the result, the waste heat αQ_2 becomes equal to the sum of exergy loss LW in each component and unused exergy E_X rejected to the condenser W. Namely,

$$\alpha Q_2 = \Sigma(LW) + E_X \quad (13)$$

$$\eta_{th,E} = 1 - \{\Sigma(LW) + E_X\} / E_S$$

$$= 1 - \alpha Q_2 / Q_1 = \eta_{th} \quad (14)$$

Table 1 Analytical results of exergy-losses

(per turbine inlet flow rate 1 kg)

$P_1=200$ ata
 $P_2=50$ ata
 $t_3=600$ °C
 $\eta_T=90\%$
 $\eta_C=\eta_P=85\%$
 Basic condition :
 50 ata
 14.3 °C

term	units	kcal/kg	%	kcal/kg	%	kcal/kg	%	kcal/kg	%	kcal/kg	%
Exergy supplied		97.33	100.00	67.12	100.00	70.02	100.00	58.33	100.00	100.38	100.00
Exergy-loss in heater		40.10	41.20	25.52	38.02	24.85	35.49	21.66	37.13	37.83	37.69
Exergy-loss in regenerator		14.60	15.00	2.70 } 5.21 2.51 }	4.02 } 7.76 3.74 }	3.49 } 6.00 2.51 }	4.98 } 8.56 3.58 }	0.19 } 4.03 1.69 }	0.32 } 6.91 2.90 }	17.06	17.00
Exergy-loss in compressor		—	—	0.84	1.25	0.84	1.20	0.37 } 1.09 0.72 }	0.64 } 1.83 1.24 }	1.46	1.45
Exergy-loss in turbine		2.03	2.08	2.03	3.02	1.90	2.72	2.03	3.47	1.91	1.90
Exergy-loss in pump		0.77	0.79	0.46	0.68	0.46	0.65	0.29	0.67	—	—
Unused exergy rejected to condenser (cooler)		0.53	0.54	0.31	0.46	0.31	0.45	0.27	0.46	5.82	5.80
Sum of exergy-losses and unused exergy		58.03	59.62	34.37	51.21	34.36	49.07	29.47	50.52	64.08	63.84
Useful work		39.30	40.38	32.75	48.79	35.66	50.93	28.86	49.48	36.30	36.16

where $\eta_{th,E}$ is exergy efficiency in the cycle. The results of the exergy-analysis are shown in Table 1 for five different CO₂ cycles. It can be seen from the table that the exergy-losses in the regenerators and the compressor in the double-regenerative cycle (b) decrease to 6.5 kcal/kg from the value of 14.6 kcal/kg in the single-regenerative cycle (a) because of a decrease of the temperature difference in the regenerators in the cycle (b) with the fractional flow. Thus, the effectiveness of the fractional flow can be explained quantitatively. Comparison of the reheat cycle (c) with the cycle (b) shows that the exergy-losses in the regenerators increase a little from 5.21 kcal/kg to 6.00 kcal/kg, but those in the heater decrease from 25.52 kcal/kg to 24.85 kcal/kg and also in the turbine from 2.03 kcal/kg to 1.90 kcal/kg. As the result, the thermal efficiency of the cycle (c) is improved to 50.93 % against the value of 48.79 % in the cycle (b). In a triple-regenerative cycle (d) the losses in the regenerators decrease further to 4.03 kcal/kg and also in the heater decrease to 21.66 kcal/kg. As the result, the thermal efficiency is improved better than that of the cycle (b). Also, in a single-intercooling and single-reheating gas-turbine cycle (e), the exergy-losses in the regenerator and the cooler are larger than those in the other condensed cycles.

8. Conclusions

Thermal efficiency of two fundamental non-reheat and reheat CO₂ condensation cycles is discussed. It is shown that at turbine inlet pressure of about 250 ata and turbine inlet temperatures higher than 650 °C the CO₂ condensation cycles in double- or triple-regenerating give a better performance than the ordinary steam power-plant cycles, for instance, with single-reheat and 5th-extraction stages. Also, an analysis of losses in each component in the CO₂ cycles by the exergy-concept is made, and the effects of regenerating and reheating on the exergy losses are discussed.

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