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CHARACTERISTICS OF A CLOSED CYCLE GAS TURBINE WITH SUPER CRITICAL CO_2 AS WORKING FLUID

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ABSTRACT

Cycle characteristics of closed gas turbines using super critical carbon dioxide as a working fluid are investigated. It is found an anomalous behavior of physical properties of CO_2 at pseudo-critical point may limit heat exchange rate of a regenerative heat exchanger due to the presence of pinch point inside the regenerative heat exchanger. Taking such pinch problem into consideration, the cycle efficiency of Brayton cycle is assessed. Its value is found limited to 39% degraded by 8% compared with the case without the pinch present inside. As an alternative a part flow cycle is investigated and its characteristics have been identified. It is revealed that the part flow cycle is effective to recover heat transfer capability and may achieve the cycle thermal efficiency of 45% under maximum operating conditions of 20MPa and 800K. Optimal combination of turbine expansion ratio and a part flow ratio to give highest thermal efficiency is clarified. Parametric study is carried out. It is found that both adiabatic efficiency of turbo-machinery and temperature effectiveness of regenerative heat exchangers may not affect cycle efficiency significantly.

Key words: **carbon dioxide, super critical, part-flow cycle, microchannel, pressure loss, recuperator**

NOMENCLATURE

A	Heat transfer area	[m^2]
CEP	Compressor exit pressure	[Pa]
L	Length of heat exchanger	[m]
M	Number of heat elements	[-]
MC	Main compressor	
MCHE	Microchannel heat exchanger	
P	Pressure	[Pa]
Q	Heat load from entry to coordinate *x*	[W]
Q_0	Heat duty of a whole heat exchanger	[W]
q	Heat load of a heat element	[W]
RC	Recompressing compressor	
T	Temperature	[K]
TIT	Turbine inlet temperature	[K]
TIP	Turbine inlet pressure	[Pa]
ΔT	Local temperature difference between two fluids	[K]
ΔT_{LMTD}	Logarithmic mean temperature difference	[K]
$\Delta \overline{T}$	Generalized mean temperature difference	[K]
W	Mass flow rate of fluid	[kg/s]
x	Coordinate(axial position of a heat exchanger)	[m]

Greek

α	Heat transfer coefficient	[W/m^2K]
λ	Heat conductivity	[W/mK]
ρ	Fluid density	[kg/m^3]
η_{temp}	Temperature effectiveness	[-]
η	Cycle thermal efficiency	[-]
ψ	Part flow ratio	[-]
φ	Turbine expansion ratio	[-]

Superscripts

-	Average over an entire heat exchanger
1	Hot fluid
2	Cold fluid

Suffices

i	Inlet
o	Outlet
j	j-th heat element

1. INTRODUCTION

Cycles of Supercritical carbon dioxide (S-CO_2) gas turbines were studied in the past [1]. However, it has not been practiced yet, mainly because of lack in high performance heat exchangers to meet cycle requirement. Recently S-CO_2 cycle for fast gas-cooled reactors based on part flow cycle was studied[2]. However, further work still remains to be done on heat transfer limitation anticipated in regenerative heat exchangers (recuperators thereafter), and an optimal combination of part flow ratio and turbine expansion ratio. The present paper is the continuation of the initial studies presented in [3, 4]. The first paper [3] revealed that high performance recuperator is indispensable in order to realize S-

CO_2 gas turbines with the thermal efficiency over that of steam turbine. Coincidently, novel microchannel heat exchangers demonstrated high temperature effectiveness over 98% [4]. The cycle analysis of the reference [3] assumed implicitly that a given temperature effectiveness of the recuperator is always achievable. It has been newly found that this assumption is not always valid and under a certain operating condition may result in the reduction of heat exchange rate in the recuperator, which causes cycle thermal efficiency to be lowered. In the present paper cycle characteristics are assessed with heat transfer limitation taken into consideration. Moreover, the optimal combination of part flow ratio and turbine expansion ratio was sought. Furthermore, sensitivities of important cycle parameters were clarified.

2. HEAT TRANSFER LIMITATION

2.1 Variations of temperature profile in heat exchangers

For counter-flow heat exchangers, temperature profiles of two fluids are illustrated in Fig.1.

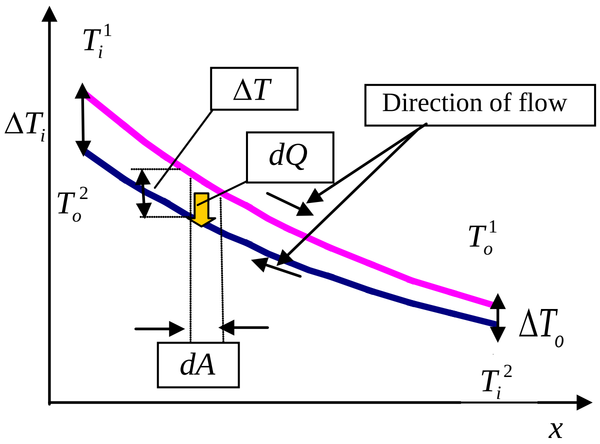

Fig.1 Variation of the fluid temperatures in a counter-flow heat exchanger

In the case of constant thermal properties, following explicit expressions are obtained [5]:
Heat load Q is expressed as function of x, a longitudinal distance

$$\frac{Q}{Q_0} = (e^{\frac{x}{L}ln\frac{\Delta T_o}{\Delta T_i}} - 1)/(\frac{\Delta T_o}{\Delta T_i} - 1) \quad (1)$$

and temperature difference between two fluids is expressed as a linear function of the heat load

$$\Delta T = \Delta T_i \left[\left(\frac{\Delta T_o}{\Delta T_i} - 1 \right) \frac{Q}{Q_0} + 1 \right] \quad (2)$$

Fig.2 shows temperature difference profile using Eq.(2) with $\Delta T_i = 28$, $\Delta T_o = 9$, $Q_0 = 4.6kW$ under constant physical property. Mean temperature difference by Logarithmic mean temperature difference method (LMTD method) gives 16.7K and pinch point appears at the entry of cold fluid. It should be noted that 100% of temperature effectiveness could be reached in an asymptotic sense. Here, temperature effectiveness on hot fluid basis is defined as

$$\eta_{temp} = (T_i^1 - T_o^1)/(T_i^1 - T_i^2) \quad (3)$$

and T_o^1 can approach T_i^2 in an asymptotical sense with heat transfer area increased.

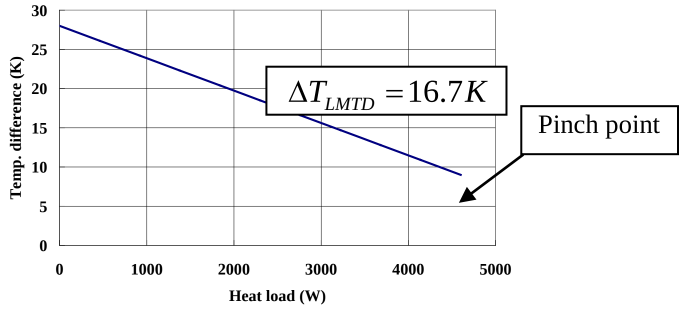

Fig.2 Temperature difference profile under constant physical property

2.2 Temperature profile calculation for distributed physical property

Given $Q_0, T_i^1, T_i^2, W_1, W_2, P_i^1$ and P_i^2, temperature profiles of both fluids and resultant temperature difference profile may be obtained as follows. Calculating specific enthalpies at inlets one can obtain specific enthalpy and temperature of cold side fluid (indicating by superscript 2) at outlet. Flow is counter current as shown in Fig.1. Next, heat duty Q_0 is expressed by a sum of M sets of constant heat element $q(= Q_0/M)$ each of which is composed of computational nodes sequentially numbered from inlet of hot fluid downward to outlet. Calculation starts from the first element. Enthalpies and temperatures of both fluids at the right node can be calculated from element-wise heat balance with known values of the first node T_i^1 and T_o^2. Thermal properties of each fluid were obtained by PROPATH [6] in this study. As the right node shares the left node of the second heat element, calculation proceeds in sequence until heat load Q reaches the heat duty Q_0 given. At each node temperature difference of two fluids is calculated. Thus average temperature difference $\Delta \overline{T}$ is obtained by

$$\frac{1}{\Delta \overline{T}} \equiv \frac{1}{Q_0} \int_0^{Q_0} \frac{dQ}{\Delta T(Q)} = \frac{1}{Q_0} \sum_{j=1}^{M} 2q/(\Delta T_j + \Delta T_{j+1}) \quad (4)$$

,and

$$\Delta \overline{T} = (\sum_{j=1}^{M} 2/M(\Delta T_j + \Delta T_{j+1}))^{-1} \quad (5)$$

2.3 Thermal property of carbon dioxide near critical point and pinch point in heat exchanger

A supercritical state in the vicinity of the critical point of carbon dioxide (7.4MPa, 304K) suits the operating condition at inlet to compressor where the real gas effect is significant. As the value of compressibility coefficient of working fluid becomes so small there that the reduction of compressor work can be achieved [3]. Fig.3 shows phase diagram of carbon dioxide and its behavior of thermal properties expected in the recuperator. It is seen isobaric specific heat has a peak at pseudo critical point. Taking two typical pressures at pseudo critical state appearing in the cycle as an example, the effect of distributed properties was examined. Properties were calculated by PROPATH [6].

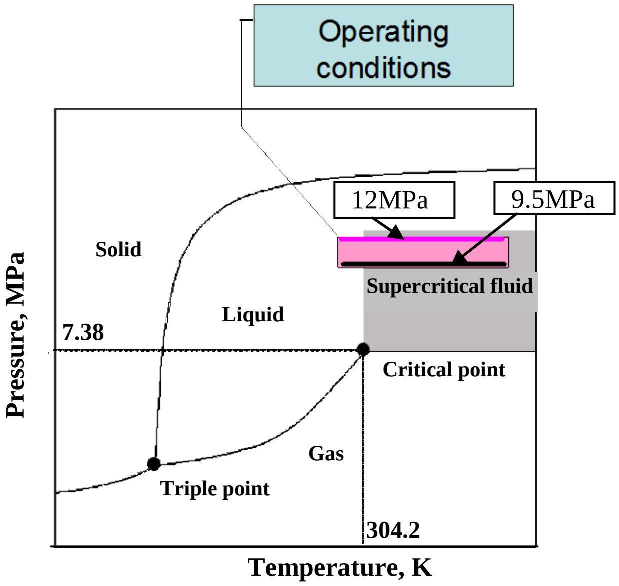

Fig. 3A CO_2 phase diagram

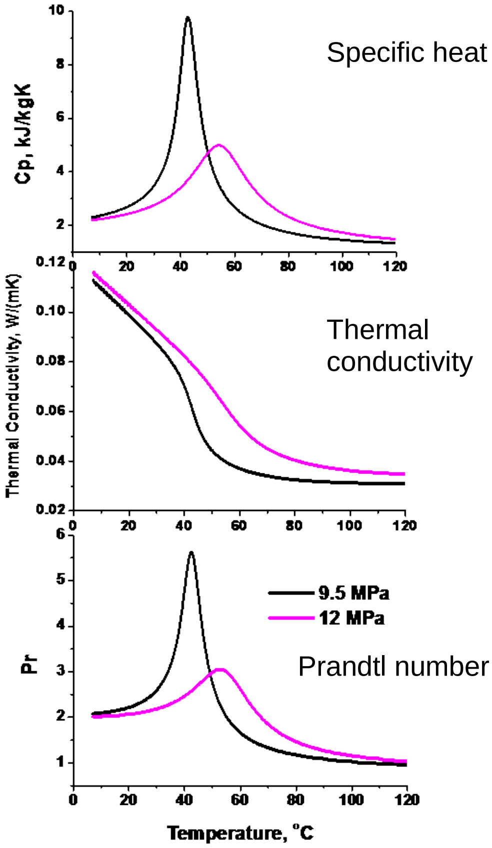

Fig.3B Thermal properties near critical point

Fig.3 Behavior of physical property of carbon dioxide in pseudo-critical state

Fig.4 shows temperature profiles and a variation of temperature difference expected in a regenerative heat exchanger of CO_2 cycle. Figs.4A and 4B show them for hot fluid pressures are 20MPa and 12MPa respectively. The cold fluid pressure is about 7.6MPa for both cases and thermodynamic state of the cold fluid is near the critical point where non-linear characteristics of physical properties exist. Due to this the temperature difference is not linear unlike Fig.2. The pinch point appears inside the heat exchanger in Fig.4A, which implies that heat transfer limitation exists regardless of the heat transfer area, which is quite different from the case of constant thermal property. This is what is called heat transfer limitation.

In this situation one cannot assume arbitrary temperature effectiveness at the regenerative heat exchanger but must examine pinch temperature difference is positive in every solution. Thus, in the present analysis a solution was obtained under the condition that the pinch point is kept positive.

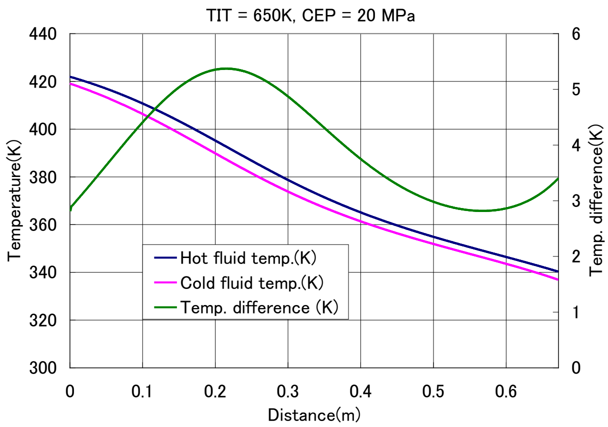

Fig.4A Temp. profile at TIT=850K,CEP=20MPa

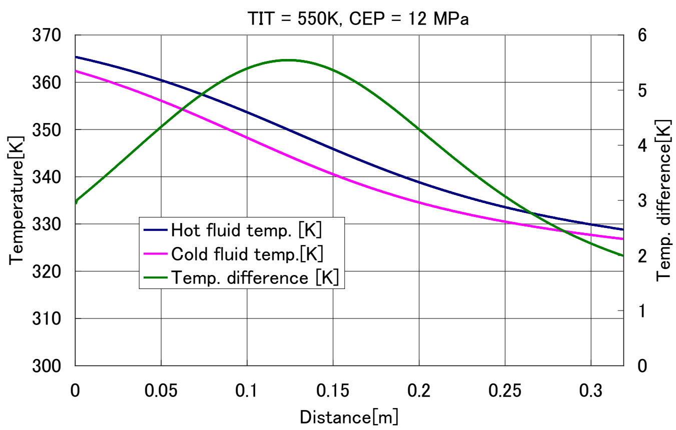

Fig.4B Temp. profile at TIT=550K, CEP=12MPa

Fig.4 Temperature profile in heat exchanger using S-CO_2 as a heating medium

3. CYCLE MODIFICATION

3.1 Part-flow cycle

Fig.6 shows a part flow cycle. The difference from the conventional closed Brayton cycle lies in the addition of a second compressor. Flow is split at position 9 and a bypass flow enters the second compressor to return to a recuperator. Part flow ratio ψ is defined as a ratio of the rate of flow entering main compressor to a total flow. In the special case that ψ is unity, the part flow cycle coincides with the Brayton cycle. In this configuration cycle thermal efficiency is expected to increase because of enhanced heat recovery from turbine exhaust gas. The division of a recuperator into two viz. RHX1 and RHX2 may moderate heat transfer limitation and increase cycle thermal efficiency because it would be confined in a second recuperator RHX2 in which hot fluid experiences pseudo-critical state.

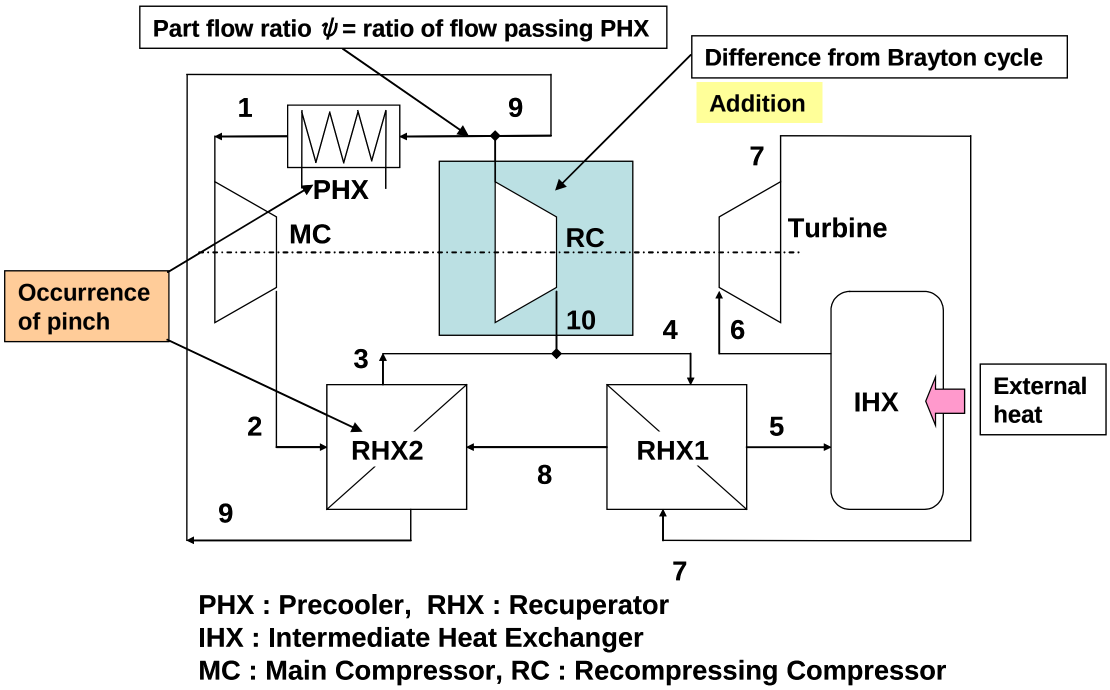

Fig.5 Part-flow cycle configuration

3.2 Calculation condition

Cycle calculation condition is listed in Table 1. All the external heat was assumed to be taken in the cycle. Turbine inlet pressure (TIP) is selected as about 20MPa because closed cycle generally has its maximal thermal efficiency in the range of turbine expansion ratio of around 2.5~3 while compressor inlet pressure (CIP) should be in the vicinity of critical pressure (7.38MPa) in order to take advantage of reduced work. Turbine inlet temperature (TIT) of 800K was selected where ordinary metal material of stainless steel can be used in view of initial cost reduction. It is important to note that part flow ratio is a free parameter in the present analysis, which leads to non-coincident temperatures appearing at node 3 and 10.

Based on experiments using newly developed micro channel heat exchanger (MCHE) [4], practicable values of thermal-hydraulic parameters were selected. Pressure drop ratio is defined as pressure drop to system pressure ratio. Fig.6 shows the configuration of assumed micro-channel heat exchanger (MCHE). Its hydraulic diameter is typically ~1mm. With this heat exchanger, high temperature effectiveness at the same time with small pressure drop can be realized.

Table 1 Calculation condition

Equipment	Pamrameter	Unit	Value
Compressor	Inlet temp of MC, CIT	K	305
	Adiabatic efficiency	-	0.9
	Exit pressure, CEP(nominal)	MPa	20
Turbine	Inlet temp., TIT(nominal)	K	800
	Adiabatic efficiency	-	0.93
	Expansion ratio, φ	-	to be optimized
Heat exchanger			
RHX	Temp. effectiveness(target)	%	98
	Press. drop ratio (hot fluid)	%	0.4
	Press. drop ratio (cold fluid)	%	1.2
IHX	Press. drop ratio	%	1.2
PHX	Press. drop ratio	%	1
	Part-flow ratio, ψ	-	to be optimized

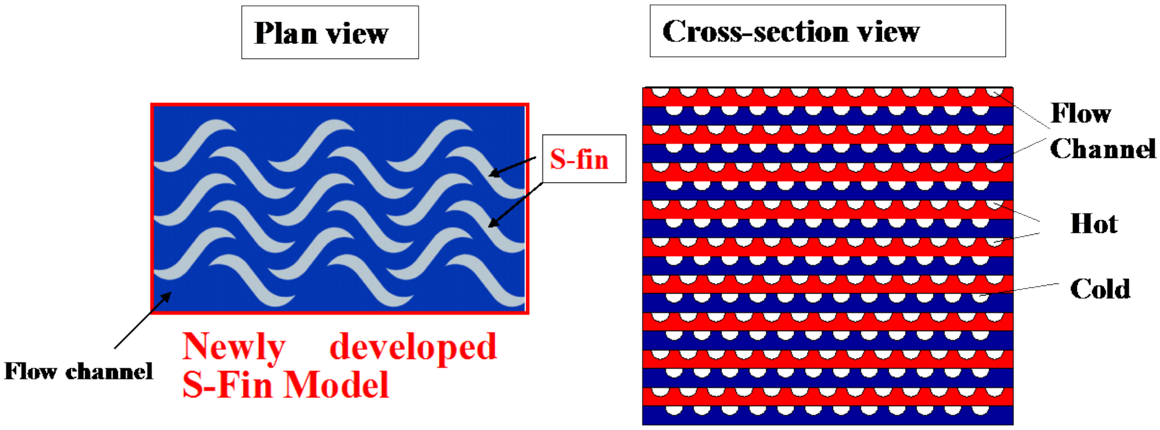

Fig. 6 S-fin micro-channel heat exchanger

4. RESULTS AND DISCUSSION

4.1 Thermal efficiency in an operable range

Calculation was carried out with part flow ratio ψ and turbine expansion ratio φ were varied as parameters as shown in Figs.7 and 8. Fig.7 shows that turbine expansion ratio φ can be selected in the range 2.2<φ<2.65 in which it is possible to avoid negative pinch as well as secure supercritical state at compressor inlet.

In the case φ=2.2 part flow cycle coincides with Brayton cycle because positive pinch is available under the condition ψ=1 alone. In the range φ>2.2 part flow operation is available. It is practicable in the range that $\psi_{min} < \psi < 1$. As ψ reduces from unity the pinch becomes reduced and cycle thermal efficiency is increased monotonously. The marginal line shows the position where $\psi = \psi_{min}$. As shown in Fig.8 maximum thermal efficiency in (ψ, φ) parametric space appears at the operation condition (0.68, 2.51) to yield 45% whilst Brayton cycle gives about 39%. T-s diagram of the operation to give maximum cycle thermal efficiency is shown in Fig.9. Thermodynamic state at compressor inlet is seen to be quite close to critical point. Where two streams join at node 3 and 10, same static pressure condition was applied in the calculation but not for temperatures. It should be noted that resultant temperature difference between two streams is about 8degC and acceptable free from thermal shock. On the other hand external heat in a temperature range lower than about 680K (400degC) cannot be directly recovered from the present system. In practice, however, this could be solved by the introduction of an economizer and is not a critical design issue. For example, in the case of combustion gas, flue gas from IHX could be used to warm intake air before combustion through the economizer. In the case of solar thermal energy or nuclear energy, coolant temperature can be easily controlled by changing flow rate of coolant to meet CO_2 temperature.

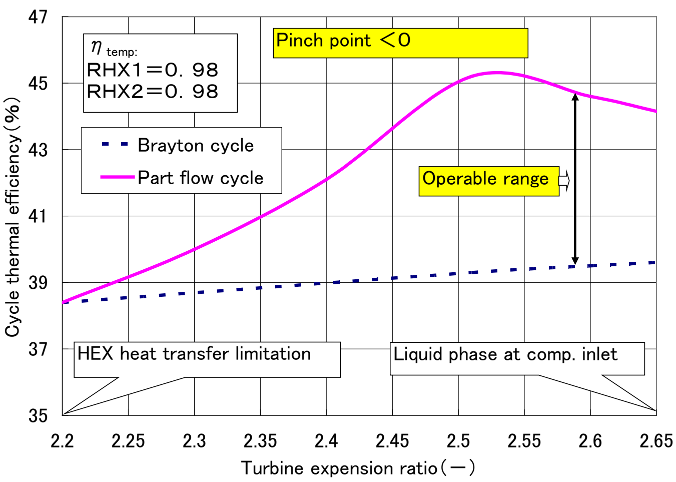

Fig.7 Cycle efficiency as function of turbine expansion ratio

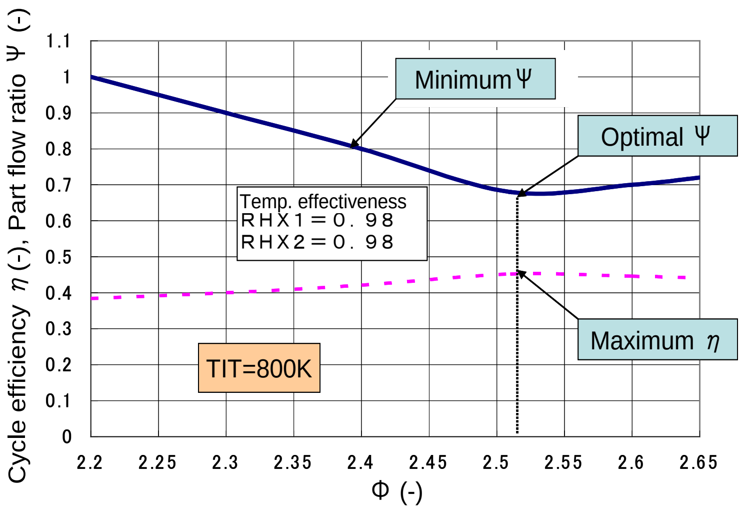

Fig.8 Optimal part flow ratio ψ

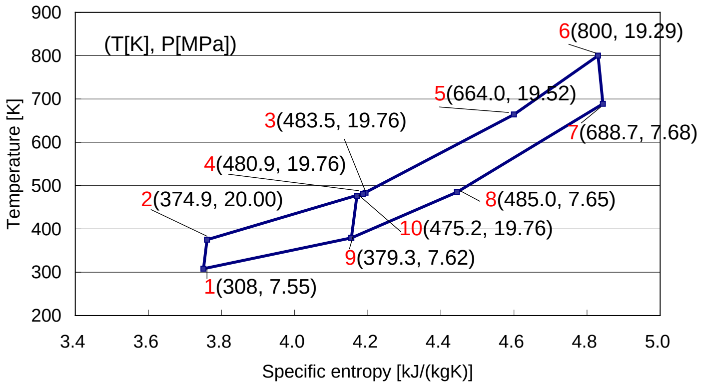

Fig.9 T-s diagram under operation condition optimized for part flow cycle

4.2 Comparison with Brayton cycle

Fig.10 shows thermal efficiency as function of specific power output. Highest system pressure and part flow ratio ψ were kept at 20MPa and 0.68 respectively. Given turbine inlet temperature TIT and turbine expansion ratio φ, a set of thermal efficiency and specific power output are obtained from the figure. Thermal efficiency of part flow cycle is always higher than Brayton cycle under the condition of the same TIT whilst a bit larger specific power output is obtained in Brayton cycle than in part flow cycle. Improvement of thermal efficiency due to part flow operation is secured by about 5% regardless of TIT. Turbine expansion effect on thermal efficiency is not significant at higher TITs. Figs. 11 and 12 show optimized values of φ and ψ respectively corresponding to each design point of part-flow cycle depicted in Fig.10. It is seen φ and ψ vary significantly with CEP and slightly with TIT

4.3 Sensitivity analyses

4.3.1 Temperature effectiveness

Temperature effectiveness of two recuperators was varied and largest thermal efficiency at each effectiveness was obtained by optimizing the value of φ. Newly developed S-fin microchannel heat exchanger as shown in Fig.6 was assumed as the heat exchangers. Heat exchanger volume means the sum of volumes of RHX1 and RHX2. For temperature effectiveness more than 95% the volume of RHX2 increases rapidly as a result of increase in length to cause higher pressure drop. This deteriorates thermal efficiency. Trade off of these effects seems to give rise to maximum in thermal efficiency appearing around 99% of temperature effectiveness. Moderate dependency was observed Even for 85% of temperature effectiveness thermal efficiency maintains about 40% of thermal efficiency.

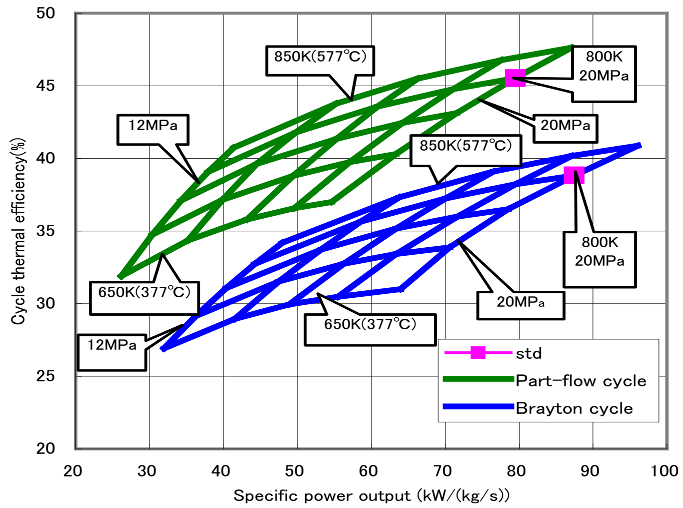

Fig.10 Thermal efficiency as function of specific output

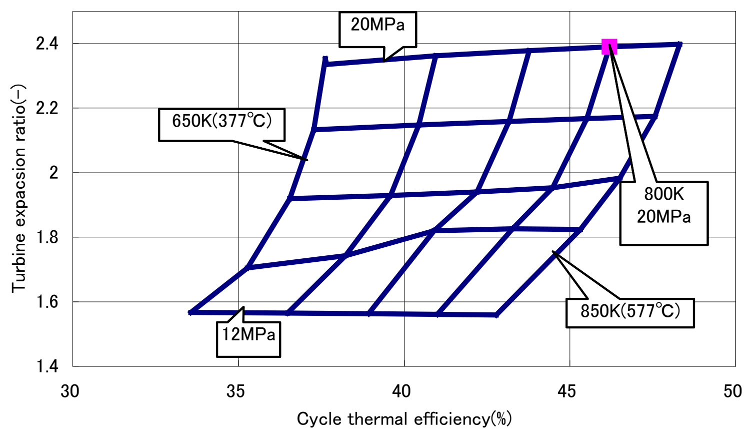

Fig.11 Optimized turbine expansion ratio

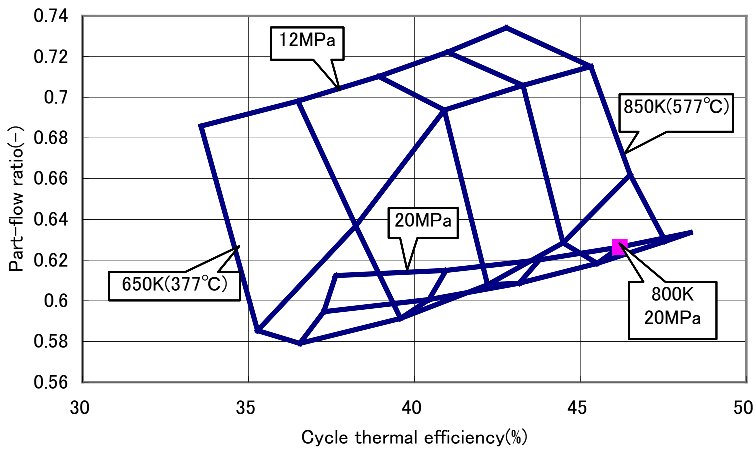

Fig.12 Optimized part-flow ratio

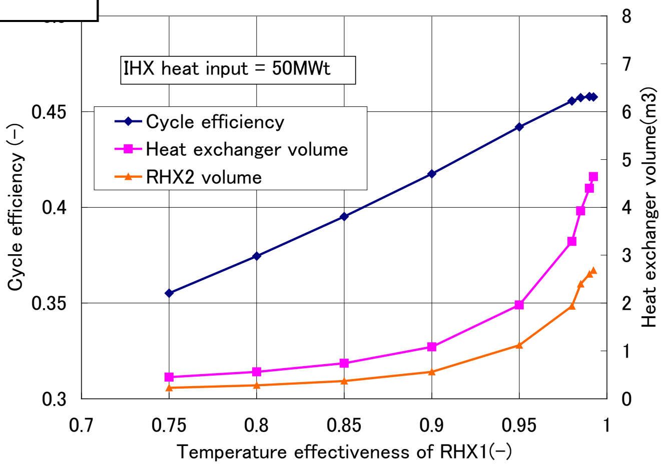

Fig.13 Effect of temperature effectiveness on thermal efficiency

4.3.2 Effect of adiabatic efficiency of compressor and turbine

Adiabatic efficiencies of both compressor and turbine were varied and their effect on thermal efficiency was identified. The result is shown in Fig.14. Much weaker effect than in open cycle was observed. This comes from it that efficient heat recovery at recuperator will enhance regenerative effect and suppress heat loss. Noting that the density of supercritical carbon dioxide is about 150 times larger than that of ambient air, energy density is quite high compared with steam rankine cycle or open Brayton cycle. Hence axial flow compressor or turbine seems hard to apply due to high loading on blades. In turn centrifugal compressor and/or radial turbine set is to be used despite of lower adiabatic efficiency. From Fig.14, however, cycle efficiency will not be affected so much. Thus it is concluded that the present S-CO_2 part flow cycle may fit small capacity typical in renewables.

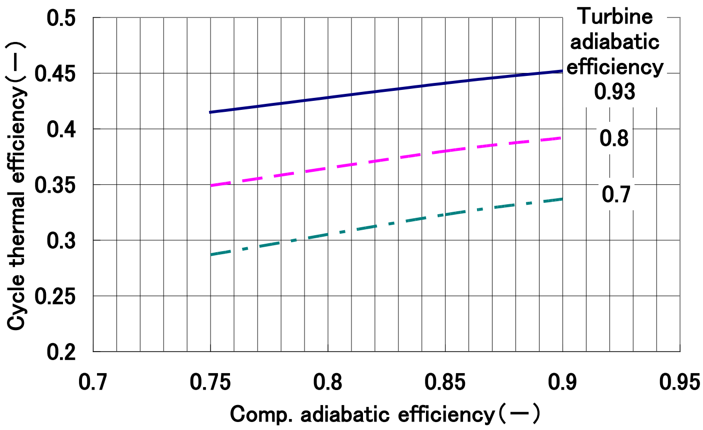

Fig.14 Effect of adiabatic efficiency

5. CONCLUSIONS

Cycle characteristics of a closed gas turbine with super-critical carbon dioxide as a working fluid have been analyzed. Special emphasis was placed upon the deterioration of thermal efficiency due to heat transfer limitation of heat exchangers inherent to the cycle. Its cause and remedy were clarified. Following results were obtained:

1) Conventional closed Brayton cycle suffers from reduction of thermal efficiency η by no more than 7% (absolute) due to the appearance of pinch point inside regenerative heat exchanger.
2) A part flow cycle was a promising candidate to suppress pinch effect. An optimal set of part flow ratio ψ and turbine expansion ratio φ allowed η to be recovered by 6% (absolute) yielding to 45%.
3) Newly developed S-fin microchannel heat exchanger (MCHE) is a promising candidate for regenerative heat exchanger.
4) Temperature effectiveness of regenerative heat exchanger as well as adiabatic efficiency of compressor and/or turbine is not sensitive to η. This feature allows S-CO_2 cycle to apply for distributed generation with small capacity.

ACKNOWLEDGEMENT

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