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TEST PLAN OF A BENCH SCALE CLOSED CYCLE GAS TURBINE WITH SUPER-CRITICAL CO_2 AS WORKING FLUID

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ABSTRACT The development of a closed cycle gas turbine with super-critical carbon dioxide as a working fluid is under way in order to generate power from waste heat source of a low or intermediate temperature range from industry. Its demonstration test plan using a reduced scale turbomachine is described. Principal specifications follow; net power output of 10kWe and recirculation CO_2 flow rate of 1.2kg/s. Optimized range of compressor inlet temperatures as well as pressure is investigated under the given turbine inlet conditions of 550K and 12MPa respectively. Given these inlet conditions, a suitable type of turbomachine is selected to be radial. Based on the conventional design method assuming ideal gas an optimal dimension of the outer wheel diameter is predicted to be 20mm for compressor and 35mm for turbine under an optimal rotational speed of 100000rpm.

Keywords: gas turbine, carbon dioxide, supercritical , microchannel heat exchanger, centrifugal compressor

1. INTRODUCTION

Gas in a supercritical state close to the critical point may lead to the reduction of compression work compared to ideal gas owing to real gas effect. Moreover, carbon dioxide has the critical point around room temperature. Then it is possible to configure an efficient gas turbine power cycle working even in a low/intermediate temperature range[1,2]. The cycle would facilitate power conversion from unused energy in low temperature region such as waste heat from industry or renewable energy.

Realization of the cycle, however, necessitates a recuperator made of compact heat exchanger with high thermal performance and with high mechanical strength resistant to supercritical pressure up to 20MPa. To meet the requirement Tokyo Tech has developed microchannel heat exchanger that is made of plural metal plates with flow channels engraved and integrated by diffusion bonding [3].

On the other hand, the regenerative heat exchanger was found to suffer from heat transfer limitation due to the onset of pinch point at an internal location. This comes from the non-linear variation of the specific heat of CO_2 in the heat exchanger. To mitigate the pinch point, an improved cycle configuration was investigated [4].

Based on these findings, a demonstration test is planned. This paper describes the principle of supercritical CO_2 gas turbine and the facility of demonstration test together with specifications of primary components.

2. PRINCIPLE OF CO_2 GAS TURBINE

Net work Q obtained from gas turbine cycle using real gas as working fluid is formally expressed by

$$Q = W_T - W_C$$
$$W = \int_{P_1}^{P_2} VdP = \int_{P_1}^{P_2} z(P,T) \cdot RTdP \ / \ P \tag{1}$$

where, W_T: turbine work, W_C: compressor work, P: pressure, T: temperature, R: gas constant, V: volume and $z(P,T)$: compressibility coefficient with $z \leqq 1$. In ideal gas z is equal to unity. Then, Q becomes maximal if turbine works in near ideal gas state and at the same time compressor works near the critical point where z is minimal.

Figure 1 shows the behavior of z as function of P and T. It is apparent the compressor should work near the critical point where z is around 0.3.

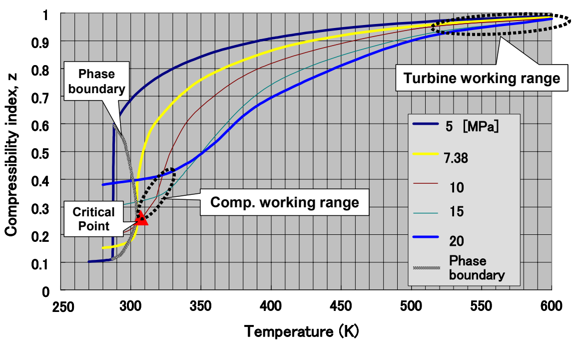


Fig.1 Behavior of compressibility coefficient z

Figure 2 illustrates compressor work ratio to turbine work of both conventional open cycle GT and S-CO_2 GT with compressor inlet temperature set to 308K. It is shown compressor work is significantly reduced in S-CO_2 GT.

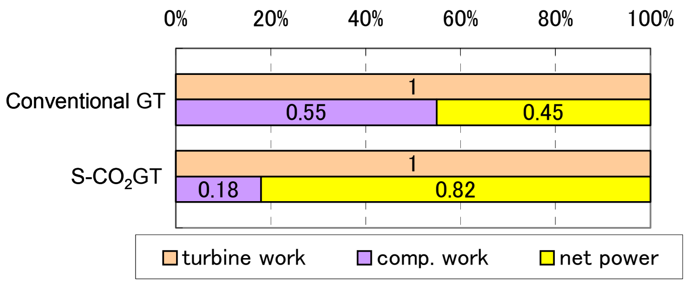


Fig.2 Compressor work to turbine work ratio

Figure 3 shows a part flow cycle configuration to solve pinch problem at regenerative heat exchanger (RHX). RHX is divided into two i.e. recuperator1 (RHX1) and recuperator2 (RHX2) to confine the pinch into RHX2 and moderate the pinch problem as a whole by optimizing flow split ratio. The relation between cycle thermal efficiency and specific power is shown in Fig.4 with turbine inlet temperature 550<TIT<850K and compressor exit pressure 12<CEP<20MPa. Prototype machine is to be designed at design point C and demonstration test at design point A.

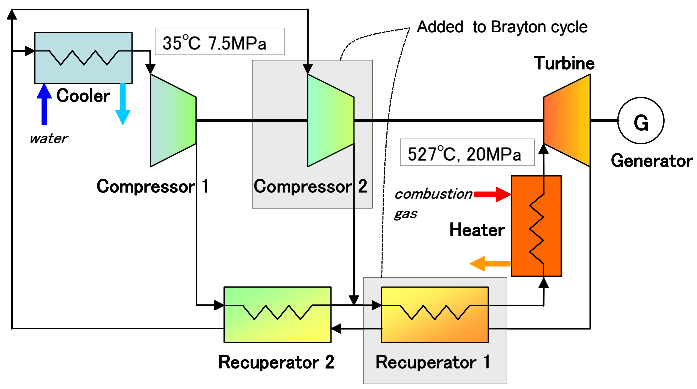


Fig.3 Part flow cycle configuration

3. REGENERATIVE HEAT EXCHANGER

CO_2 cycle with high thermal efficiency necessitates high performance compact heat exchanger. For example, required temperature effectiveness at hot side is more than 95%, which has prevented CO_2 cycle from implementation to date. Figure 5 illustrates X fin microchannel heat exchanger (MCHE) Tokyo Tech has developed. Metal plates with 1mm in thickness each of which has two dimensional flow channel engraved are laminated and bonded by diffusion bonding method. It has high mechanical strength same as that of base material and thus suits supercritical CO_2 cycle. X fin has 1.2 times higher Nu number than that of conventional zigzag flow channel[6] and two times larger than that calculated by Dittus Beolter correlation for turbulent flow in circular tube. Empirical correlations of heat transfer coefficient and pressure drop coefficient for X fin MCHE were derived from CFD numerical experiments as follows:

$$Nu = \alpha D_h / \lambda = 0.6365 Re^{0.4907} Pr^{0.7427} \tag{2}$$

$$f = \Delta P (\frac{D_h}{L}) / (\frac{v}{2} G^2) = 1.425 Re^{-0.0614} \tag{3}$$

$$D_h = 4V / A \tag{4}$$

MCHE may be considered as a conduit network. Thus the value calculated by Eq. (2) was compared with that by Dittus Boelter for forced convection in circular tube[7]. The former is two times larger than the latter [4]. Figure 6 shows accuracy of Eq. (2) compared with CFD results. Good agreement was obtained.

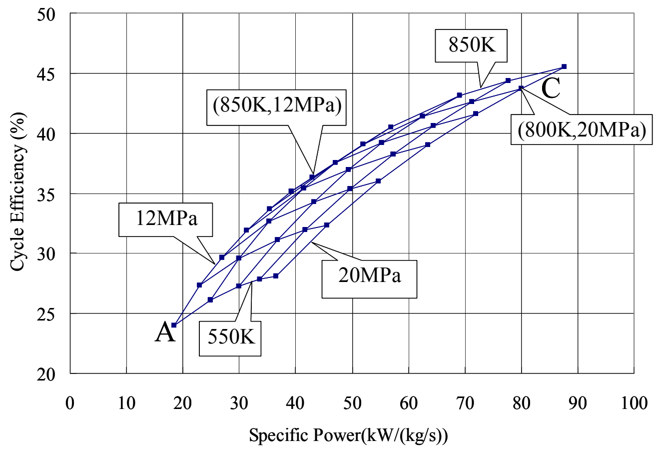


Fig.4 Cycle efficiency and specific power

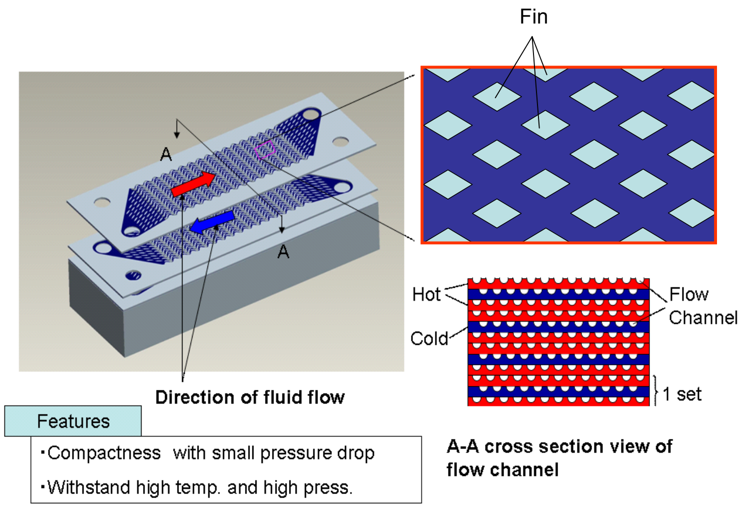


Fig.5 X fin microchannel heat exchanger

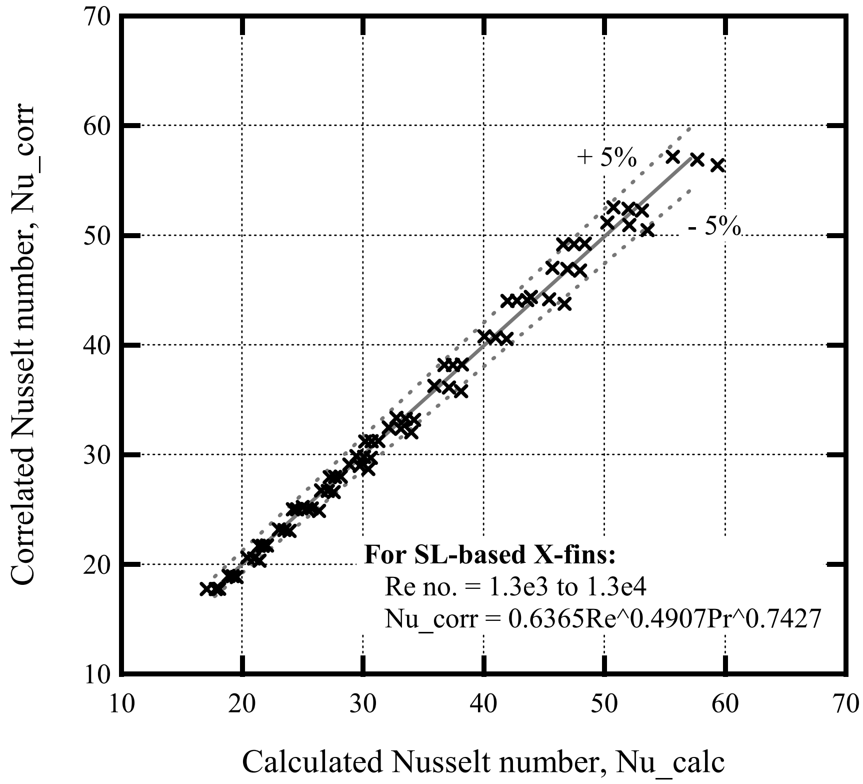
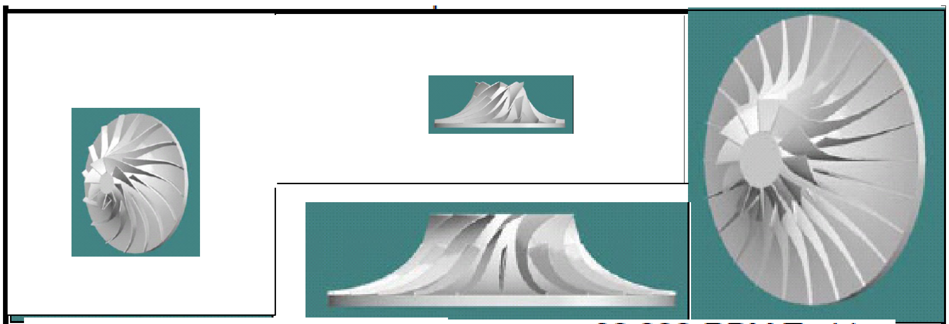


Fig.6 Accuracy of empirical correlation of Nusselt number



30,000-RPM C1 Compressor

Parameter	Value	Unit
Proposed Material	17-4 SS	
Maximum Diameter	106.54	mm
Minimum Hub Diameter	15.76	mm
Blade Count	8	
Blade Thickness Hub LE	1	mm
Blade Thickness Hub MID	2	mm
Blade Thickness Hub TE	1.5	mm
Blade Thickness Shroud LE	0.5	mm
Blade Thickness Shroud MID	1	mm
Blade Thickness Shroud TE	0.75	mm
Splitters	Yes	
Rotational Speed	30000	RPM
Tip Speed	167.35	m/s
Power	395.4	kW
Torque	125.86	N-m
Estimated Tip Clearance	0.2	mm
Estimated Seal Clearance	0.15	mm

30,000-RPM Turbine

Parameter	Value	Unit
Proposed Material	Inconel	
Maximum Diameter	280.32	mm
Minimum Hub Diameter	50	mm
Blade Count	12	
Blade Thickness Hub LE	1.5	mm
Blade Thickness Hub MID	4.4	mm
Blade Thickness Hub TE	2.6	mm
Blade Thickness Shroud LE	0.6	mm
Blade Thickness Shroud MID	0.6	mm
Blade Thickness Shroud TE	0.6	mm
Splitters	Yes	
Rotational Speed	30000	RPM
Tip Speed	440.33	m/s
Power	4048	kW
Torque	1288.52	N-m
Estimated Tip Clearance	0.2	mm
Estimated Seal Clearance	0.15	mm

Fig.7 Compressor and turbine wheel shapes and their specifications for 2.3MWe prototype machine

4. DESIGN OF TURBOMACHINE

4.1 Conceptual design of prototype machine

Compressor and turbine aerodynamic specifications were investigated with working fluid treated as real gas using property library NIST. Calculation conditions follows: net power=2.3MWe, TIT=823.2K, CIT=305.2K, CEP=20MPa, turbine expansion ratio =2.8 and rotational speed = 30000rpm. Results are shown in Fig.7 about wheel shapes and dimensions of compressor1 (C1) and turbine. Compressor wheel diameter is smaller being one third of turbine wheel. This is because CO_2 density at inlet to compressor is larger. The number of pairs of wheel and splitter is eight and twelve for the compressor and the turbine respectively.

Figures 8 and 9 show their aerodynamic characteristics at part load. Adiabatic efficiencies for the compressor and the turbine at rated speed and flow are 0.785 and 0.84 respectively. It would be kept unchanged as low as 60% rotational speed if working line is selected properly.

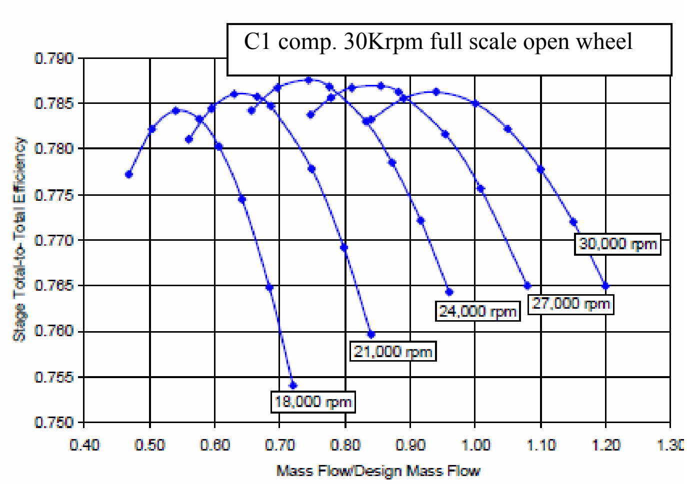


Fig.8 Compressor adiabatic efficiency at off design point

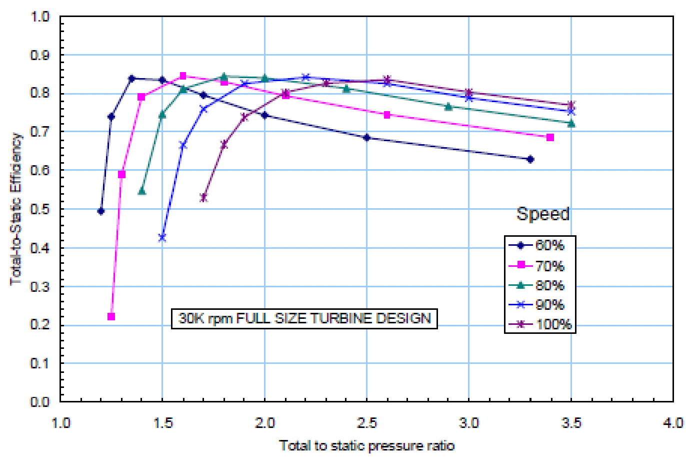


Fig.9 Turbine adiabatic efficiency at off design point

4.2 Size comparison with conventional steam turbine

The size of prototype machine is shown in Fig.10. In contrast plan view of 2MWe commercial steam turbine plant is shown in Fig.11. Steam cycle consists of many components, say condenser, feed water pumps, cooler and so on. Then space factor of S-CO_2 GT is much smaller than steam cycle. It should be noted that the turbomachine itself is very small compared with steam turbine. This comes from high energy density (J/m^3) due to dense working fluid. CO_2 density in working range varies 1/5~1/2 of liquid water.

5. DEMONSTRATION TEST

5.1 Reduced scale model for demonstration test

If the prototype machine is reduced to 1/100 by power according to similarity rule, rotational speed would be 230000rpm and wheel outer diameter 14mm whose manufacturing would be extremely difficult at present. Then, design condition was moderated and shifted to point A in Fig. 4 i.e. TIT=550K, CEP=12MPa and flow rate =1.2kg/s. Based on this fundamental conditions, turbine expansion ratio was optimized by the cycle calculation.

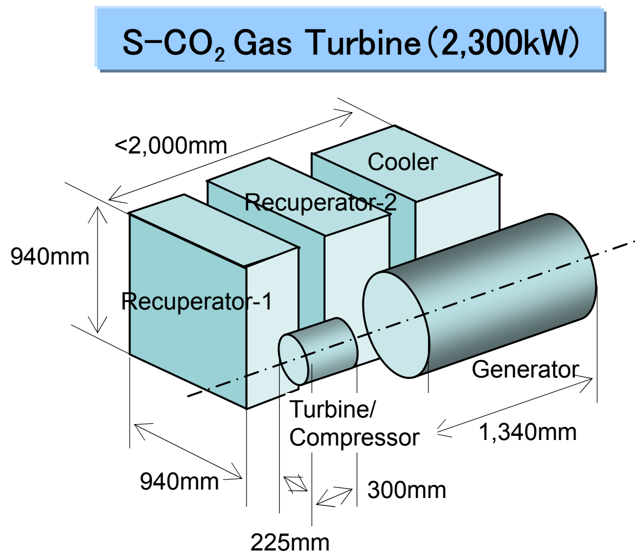


Fig.10 Size of 2.3MWe prototype S-CO_2 gas turbine

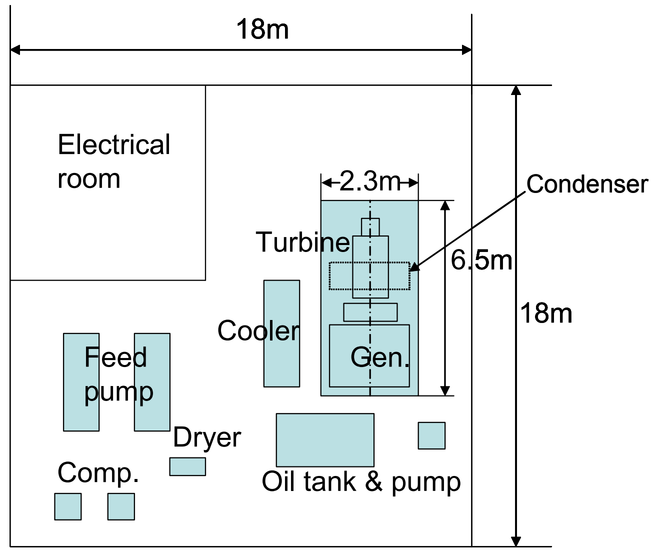


Fig.11 Plan view of 2MWe steam turbine plant

Based on the above cycle calculation results the type of turbomachines and their aerodynamic parameters were determined following Balje methodology in terms of specific speed n_s as follows

$$
\begin{aligned}
n_s &= \frac{\omega\sqrt{V}}{(H_{ad})^{3/4}} \\
\eta &= g(n_s) \\
d_s &= f(n_s) \\
D &= \frac{d_s\sqrt{V}}{(H_{ad})^{1/4}}
\end{aligned}
\qquad (5)
$$

Functions f and g are generated referring to the diagram by Balje.O.E.[8]. As this method assumes ideal gas, real gas effect was expressed by approximating equation of state using a modified gas constant as a product of gas constant and compressibility coefficient z.

Calculation results to seek an optimum design point are shown in Fig. 12. Cycle efficiency becomes highest at the rotational speed of 100000rpm. The test machine is so small that mechanical loss as well as further aerodynamic loss unique to small scale test should be taken into account because effects of bearings, tip clearance, thickness of wheel and the precision of manufacturing become dominant as scale becomes smaller. Thus, the equivalent adiabatic efficiencies of the compressor and the turbine were assumed to be 0.6 and 0.65 at 100000rpm. Then, predicted cycle efficiency was reduced to 7% and power output from the machine was expected to be 12kW.

The compressor and the turbine are to be assembled coaxially. To achieve planned performance, sealing and bearing are important design items. Effective sealing against big pressure difference of 4MPa between compressor part and turbine one is a design issue to be solved. Adoption of oil free gas bearing is planned to avoid inclusion of bearing oil in CO_2 loop. Generator/motor with permanent magnet will be adopted for starter.

Main compressor was also designed with real gas characteristics considered using property software library NIST. Calculation conditions were that flow rate is 1.2kg/s,

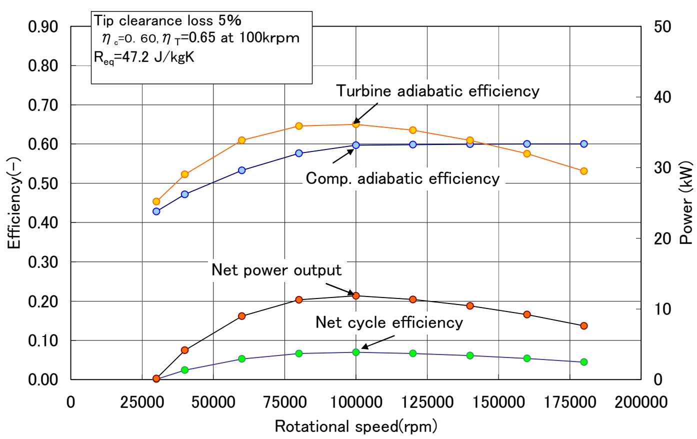


Fig.12 Parameters as function of rotational speed of a bench scale turbo-machine

total temperature at inlet to compressor is 308K, TIT is 550K, total pressure at inlet to compressor is 8.23MPa, compressor exit pressure is 12MPa. The number of blades is 13. Optimum rotational speed was reconfirmed in this method to be 100000rpm.

Table 1 lists the results of both designs by Balje and Meanline methods.

Table 1 Specifications and performance of a bench scale turbomachine

Component	Item	Design method	
		Balje(Quasi-ideal gas)	Meanline (Real gas)
Optimum rotational speed(rpm)		100000	100000
Comp.	Wheel dia.(mm)	19.6	19.9
	n_s(−)	0.65	0.68
	$\eta_{adiabatic}$(−)	0.6(0.80)	0.6(0.77)
	Req. work(kW)	12.9	13.0
Turbine	Wheel dia.(mm)	31.4	35.2
	U/C_0(−)	0.64	0.71
	$\eta_{adiabatic}$(−)	0.65(0.75)	0.65(0.87)
	Work(kW)	25.9	26.1
Cycle	Shaft power(kW)	13	13.1
	η_{cyccle}(%)	6.9	7.8

Adiabatic efficiencies in the blacket show total to static ones and contain only aerodynamic loss with ideal manufacturing process. In reality further aerodynamic loss due to manufacturing precision, mechanical and electrical losses would be superposed. Then, the design of auxiliary components as well as plant performance assumed smaller adiabatic efficiencies as listed. Both results by Baldje and meanline agree well each other, which ensures the validity of the present design. Outer wheel diameters of compressor and turbine predicted by meanline method are 19.9mm and 35.2mm respectively. It is interesting to note that the wheel outer diameter of the compressor is about two thirds of that of the turbine. This results from the fact that the densities of the incoming fluid are 619kg/m^3 for the compressor and 89kg/m^3 for the turbine.

5.2 Compressor aerodynamic performance

Figure 13 shows static pressure profile along the meridian. Sub-critical region appears locally at inlet due to acceleration. It is seen It should be noted there is static pressure drop at inlet. If it decreases below critical pressure, then structural integrity may suffer from erosion due to either cavitation or the collision of liquid droplets. It is noted that pressure recovery at diffuser is significant, which implies that diffuser design is important.

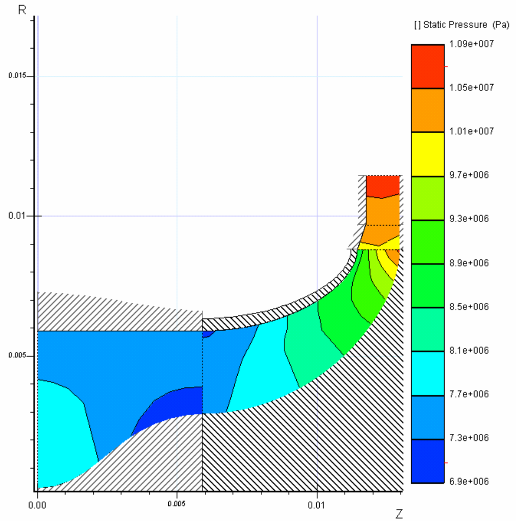


Fig.13 Predicted static pressure profile along meridian of compressor for demonstration test

6. CONCLUSIONS

The principle and demonstration test plan of the supercritical CO_2 gas turbine were described. Conceptual design of 2.3MWe class prototype machine was carried out and its layout was obtained. Area required for installation is about one half of the conventional steam turbine. The scale of demonstration was investigated, which led to it that net power output was around 10kWe, CO_2 design recirculation flow rate was 1.2kg/s and thermodynamic state at compressor inlet was 308K, 8.23MPa and at turbine inlet 550K, 12MPa respectively. Under the condition, detailed aerodynamic design of the centrifugal compressor as well as radial turbine was executed, which resulted in that optimum rotational speed was 100000rpm.

ACKNOWLEDGMENT

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NOMENCLATURE

A : heat transfer area(m^2)
CEP: compressor discharge pressure
CIP: compressor inlet pressure
D : outer diameter (m)
D_h : hydraulic diameter (m)
d_s : specific diameter(-)
f : pressure drop coefficient (-)
G : mass flux (kg/(m^2s))
H_{ad} : isentropic enthalpy drop (J/kg)
HWSexp: Hot water supplier CO_2-Water experiment
L : heat exchanger axial length (m)
Nu : Nusselt number (-)
MCHE: micro channel heat exchanger
n_s : specific speed (-)
P : pressure (Pa)
Pr : Prandtl number (-)
Q : net work(J)
R : gas constant (J/(kgK))
Re : Reynolds number (=GD_h/μ) (-)
RHX: regenerative heat exchanger
T : temperature(K)
TIT: turbine inlet temperature
TIT-3exp: CO_2- CO_2 experiment
U: overall heat transfer coefficient(W/(m^2K))
V : volume (m^3)/ volume flow rate(m^3/s)
v : specific volume (m^3/kg)
W : work (J)
z : compressibility coefficient (=Pv/(RT))

Greek letters
α: heat transfer coefficient (W/(m^2K))
η: efficiency(-)
λ: heat conductivity (W/(mK))
μ : dynamic viscosity (Pas)
ω: rotational speed (rad/s)

Subscript
C : compressor
T : turbine
1 : inlet
2 : outlet

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