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Demonstration Test Plant of Closed Cycle Gas Turbine with Supercritical CO_2 as Working Fluid

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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 around 10 kWe and recirculation CO_2 flow rate of 1.2 kg/s. Optimized range of compressor inlet temperatures as well as pressure is investigated under the given turbine inlet conditions of 550 K and 12 MPa 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 30mm for turbine under an optimal rotational speed of 100,000 min^{-1}.

Ispitivanje demonstracijskog postrojenja plinske turbine zatvorenog ciklusa s nadkritičnim CO_2 kao radnim medijem

Izvornoznanstveni članak

U tijeku je razvoj plinske turbine zatvorenog ciklusa s nadkritičnim ugljikovim dioksidom kao radnim medijem za proizvodnju električne energije iz nisko ili srednje temperaturne otpadne topline iz industrije. Opisano je ispitivanje demonstracijskog postrojenja s turbinom u umanjenom mjerilu. Glavne karakteristike su: neto izlazna snaga od približno 10 kWe i protokom CO2 od 1.2 kg/s. Istraživano je optimalno područje ulaznih temperatura i tlakova kompresora za zadane ulazne parametre turbine od 550 K i 12 MPa. Za zadane ulazne uvjete, odabran je kao odgovarajući radijalni tip turbostroja. Na temelju uobičajenih metoda konstruiranja i pretpostavljajući idealni plin određene su optimalne vrijednosti vanjskih promjera rotora kompresora 20 mm i rotora turbine 30 mm uz optimalnu brzinu vrtnje od 100.000 min^{-1}.

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 a low temperature region such as waste heat from industry or renewable energy.

Realization of the cycle, however, necessitates a regenerative heat exchanger made of compact heat exchanger with high thermal performance and with high mechanical strength resistant to supercritical pressure up to 20 MPa. To meet the requirement Tokyo Tech has developed S shaped fin 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 per unit fluid mass obtained from a gas turbine cycle using real gas as a working fluid is expressed formally as

$$Q = W_T - W_C,$$
$$W = \int_{P_1}^{P_2} v dp = \int_{P_1}^{P_2} Z(p,T) \cdot RTdp / p. \quad (1)$$

Symbols/Oznake

A	- heat transfer area, m^2 - površina prijenosa topline
D	- outer diameter, m - vanjski promjer
D_h	- hydraulic diameter, m - hidraulički promjer
d_s	- specific diameter - specifični promjer
f	- pressure drop coefficient - koeficijent pada tlaka
G	- mass flux, $kgm^{-2}s^{-1}$ - maseni protok
H_{ad}	- isentropic enthalpy drop, Jkg^{-1} - izentropski pad entalpije
L	- heat exchanger axial length, m - aksijalna duljina izmjenjivača topline
Nu	- Nusselt number - Nusseltov broj
n_s	- specific speed - specifična brzina
p	- pressure, Pa - tlak
p_{CE}	- compressor discharge pressure, Pa - izlazni tlak kompresora
p_{CI}	- compressor inlet pressure, Pa - ulazni tlak kompresora
Pr	- Prandtl number - Prandtlov broj
Q	- net work output, J - neto rad
R	- gas constant, $J\ kg^{-1}\ K^{-1}$ - plinska konstanta
Re	- Reynolds number - Reynoldsov broj
T	- temperature, K - temperatura
T_{IT}	- turbine inlet temperature, K - ulazna temperatura turbine
U	- overall heat transfer coefficient, $W\ m^{-2}K^{-1}$ - ukupni koeficijent prijenosa topline
V	- volume, m^3/volume flow rate, m^3s^{-1} - volumen
v	- specific volume, m^3kg^{-1} - specifični volumen
W	- work, J - rad
z	- compressibility coefficient - koeficijent kompresibilnosti
α	- heat transfer coefficient, $Wm^{-2}K^{-1}$ - koeficijent prijenosa topline
η	- efficiency - iskoristivost
λ	- heat conductivity, $Wm^{-1}\ K^{-1}$ - toplinska vodljivost
μ	- dynamic viscosity, Pas - dinamički viskozitet
ω	- rotational speed, rad /s - kutna brzina
HWSexp	- hot water supplier CO_2-Water experiment - dobavljač tople vode CO_2 – vodeni pokus
IHX	- intermediate heat exchanger - srednji izmjenjivač topline
MCHE	- micro channel heat exchanger - mikro kanalni izmjenjivač topline
RHX	- regenerative heat exchanger - regenerativni izmjenjivač topline
S-CO_2GT	- Gas turbine with supercritical carbon dioxide as working fluid - Plinska turbina sa superkritičnim ugljikovim dioksidom kao radnim medijem
TIT-3EXP	- CO_2- CO_2 experiment - CO_2 - CO_2 pokus

Indices / Indeksi

1	- inlet - ulaz
2	- outlet - izlaz
C	- compressor - kompresor
T	- turbine - turbina

where W_T signifies turbine work, W_C denotes compressor work, p represents pressure, T is temperature, R is the gas constant, v stands for specific volume, and $Z(p,T)$ is the compressibility coefficient with $Z \leq 1$. In fact, Z is equal to unity for an ideal gas. Then, Q becomes maximal if the turbine works in a state near ideal gas and simultaneously, the compressor works near the critical point (7.38 MPa, 304 K), where Z becomes minimum. Furthermore, because the critical temperature of carbon dioxide is closer to room temperature than that of water, a quasi-ideal gas exists at temperatures much lower than those of other media. This characteristic is advantageous for recovering work output from a heat source in a low or intermediate temperature range.

Figure 1 shows the behavior of Z as function of p and T in which preferable working zone of turbomachinery is illustrated..

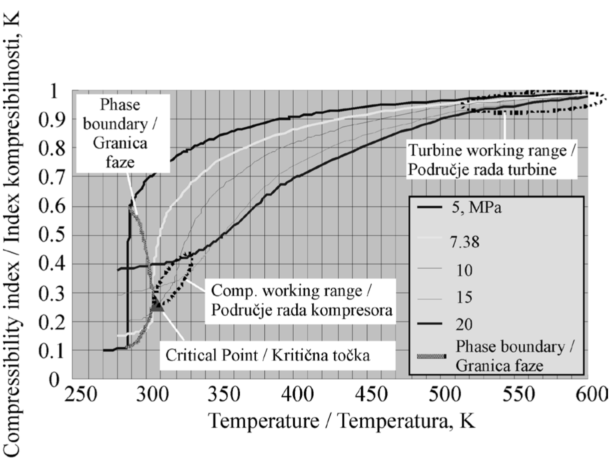


Figure 1. Behavior of compressibility coefficient Z

Slika 1. Koeficijent kompresibilnosti Z u ovisnosti od temperature

Figure 2 portrays the compressor work ratio to turbine work of both the conventional open cycle GT and S-CO_2 GT with the compressor inlet temperature set to 308 K. Compressor work for the S-CO_2 GT was evaluated from cycle optimization study on S-CO_2 part flow cycle given in Fig.3. That is the parameters were optimized under the condition that the peak pressure and temperature are 12 MPa and 800 K. This gave an optimal turbine expansion ratio as 1.42 where compressor work is markedly reduced to about one-third of those for ideal gas coincident to the reduction of Z value.

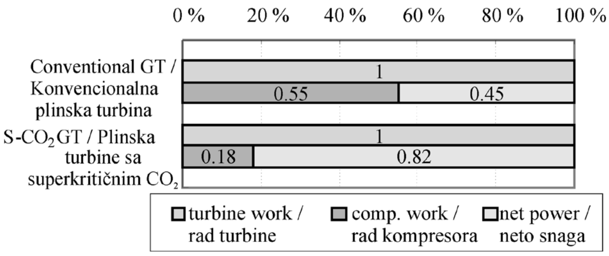


Figure 2. Compressor work to turbine work ratio

Slika 2. Omjer rada kompresora prema radu turbine

Figure 3 presents a schematic diagram of the S-CO_2 part-flow power cycle [4]. The difference from the conventional closed Brayton cycle is the addition of a second compressor. The flow splits at position 9 and a bypass flow enters the second compressor to return to the regenerative heat exchanger RHX1. The part flow ratio ψ is defined as the ratio of flow entering main compressor to the total flow. Where ψ is unity, the part flow cycle coincides with the Brayton cycle. In this configuration, the cycle thermal efficiency is increased because of enhanced heat recovery from the turbine exhaust gas. The division of the regenerative heat exchanger into two—RHX1 and RHX2—enables heat transfer limitation to be mitigated. That is, the confined RHX2, whose hot fluid contains pseudo-critical point, raises the regeneration efficiency. In other words, because ψ<1, the cycle thermal efficiency is enhanced as a result of the reduction of cycle heat loss at the pre-cooler PHX.

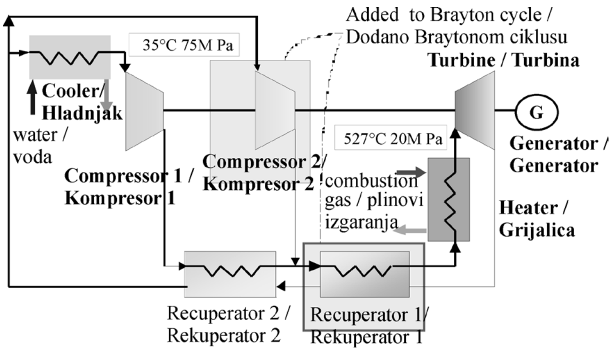


Figure 3. S-CO_2 part flow cycle configuration

Slika 3. Konfiguracija toka S-CO2 dijela ciklusa

The cycle analysis for a prototype plant was referred from an earlier report [4]. Figure 4 shows the cycle thermal efficiency as a function of specific power. Given T_{CI}=308 K, regenerator temperature effectiveness of 0.98 and isentropic efficiencies of compressor and turbines to be 0.9 and 0.93, cycle efficiency to each combination of T_{CI} and p_{CE} was obtained after optimization of turbine expansion ratio ψ and turbine expansion ratio. The major calculation conditions for the prototype machine were T_{CI}=308 K, T_{IT}=800 K, and p_{CE}=20 MPa. The efficiency of the prototype machine is shown as point C. Its capacity was assumed for biomass applications with ca. 5 MWt external heat input at IHX.

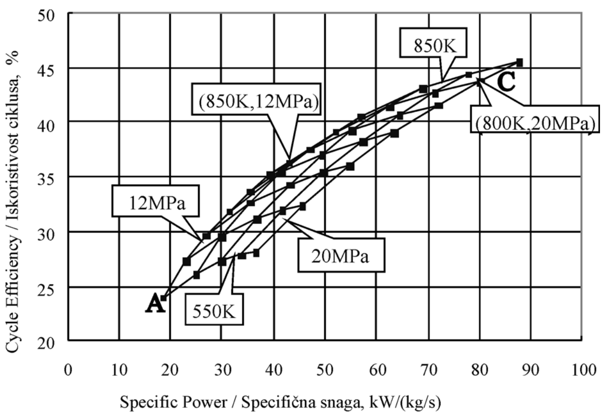


Figure 4. Cycle thermal efficiency as function of specific. power

Slika 4. Toplinska iskoristivost ciklusa kao funkcija specifične snage

If the prototype machine operating at point C were reduced to 1/100 from 2.3 MWe by the similarity law, the optimal rotational speed would be 230,000 rpm and the wheel outer diameter would be 14 mm too small to manufacture at present. Then, the design condition for a reduced scale machine was moderated and shifted to point A in Figure 4: T_{IT} =550 K, p_{CE} =12 MPa and flow rate =1.2 kg/s. The scale was chosen to be as small as possible to put the rotational speed in a practically feasible range from a manufacturing perspective.

3. Regenerative heat exchanger

CO_2 cycle with high thermal efficiency necessitates high performance compact heat exchanger. For example, required temperature effectiveness at the hot side is more than 95 %, which has prevented CO_2 cycle from implementation to date. Figure 5 illustrates S shaped fin microchannel heat exchanger (MCHE) Tokyo Tech has developed [5]. Metal plates with 1mm in thickness each of which has two dimensional flow channel engraved are laminated and bonded by the diffusion bonding method. It has high mechanical strength the same as that of base material and thus suits supercritical CO_2 cycle. S shaped fin has low pressure drop characteristics being one fourth of conventional zigzag flow channel [6].

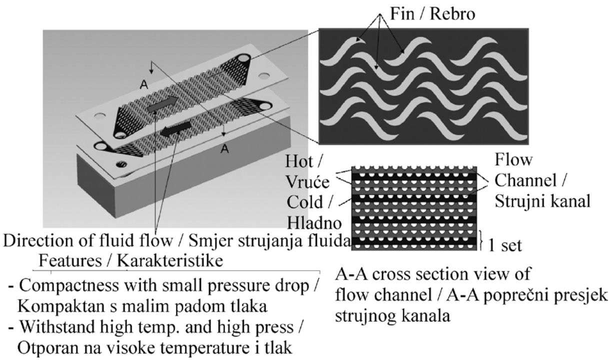


Figure 5. S shaped fin microchannel heat exchanger

Slika 5. Izmjenjivač topline sa S oblikovanim rebrima mikrokanala

Empirical correlations of heat transfer coefficient and pressure drop coefficient for S shaped fin MCHE were derived from S-CO_2 experiments as follows:

$$Nu = \alpha D_h / \lambda = 0.219 Re^{0.619} Pr^{0.358}, \tag{2}$$

$$f = \Delta P(\frac{D_h}{L}) / (\frac{v}{2} G^2) = 2.294 Re^{-0.25}, \tag{3}$$

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

MCHE may be regarded 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) in terms of averaged overall heat transfer coefficient U. Good agreement was obtained.

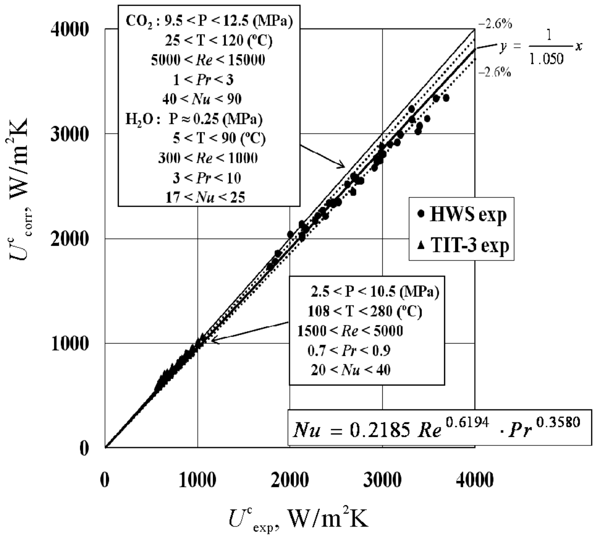


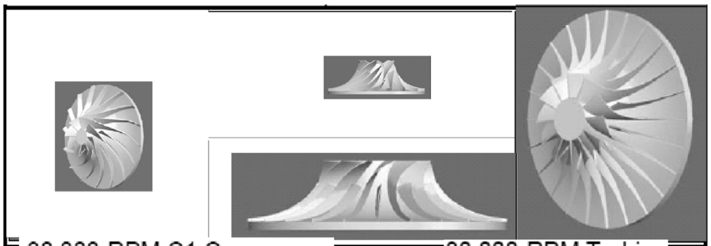
Figure 6. Accuracy of empirical correlation of Nusselt number

Slika 6. Točnost empirijske korelacije za Nusseltov broj

4. Design of turbomachine

4.1. Conceptual design of prototype machine

Compressor and turbine aerodynamic specifications were investigated by use of design software COMPAL and RITAL by Concepts NREC Co. Ltd. [9]. Working fluid was treated as real gas using NIST gas property library. Calculation conditions follows: net power=2.3 MWe, T_{IT}=823.2 K, T_{CI}=305.2 K, p_{CE} =20 MPa, turbine expansion ratio =2.8 and rotational speed = 30,000rpm. Results are shown in Fig.7 about wheel shapes and dimensions of compressor 1 (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.



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

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

Slika 7. Konstrukcija rotora kompresora i turbine te njihove karakteristike za 2,3 MWe električni prototip

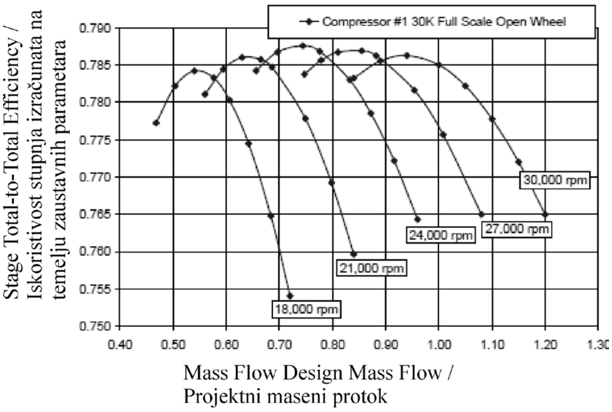


Figure 8. Compressor isentropic efficiency at off design point

Slika 8. Kompresorska isentropska iskoristivost kompresora u vanprojektnoj točki

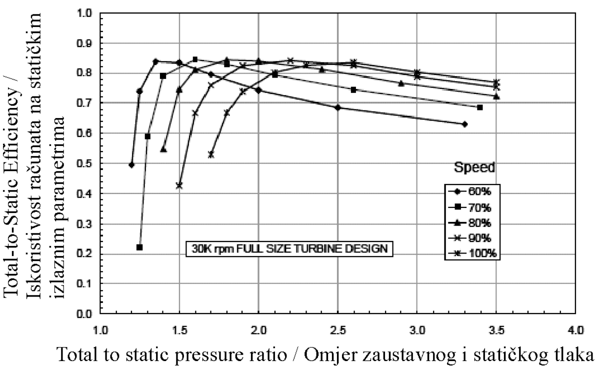


Figure 9. Turbine isentropic efficiency at off design point

Slika 9. Isentropska iskoristivost turbine u vanprojektnoj točki

Figures 8 and 9 show their off design aerodynamic characteristics. Incidence angle and backsweep angles were set 2deg and -35deg respectively. Mean line method based on boundary layer theory was applied to the analyses. That is, physical properties at both inlet and outlet were given and the mean line method was applied to obtain aerodynamic physical properties. Isentropic efficiencies for the compressor and the turbine at rated speed and flow were 0.785 and 0.84 respectively. It would be kept unchanged as low as 60 % rotational speed if the working line is selected properly. Isentropic efficiencies were expressed in terms of maximum possible value i.e. total to total basis for compressor and total to static basis for turbine. The wording total to total means total pressure at both inlet and outlet was used to evaluate efficiency and total to static, total pressure at inlet and static pressure at outlet is applied.

4.2. Size-comparison with conventional steam turbine

The size of prototype machine is shown in Figure 10. In contrast plan view of 2 MWe commercial steam turbine is shown in Figure 11. Steam cycle needs additional components, say condenser, food water pumps, heater and steam generator. 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 the steam turbine. This comes from high density of working fluid. CO_2 density in working range varies 1/5~1/2 of liquid water.

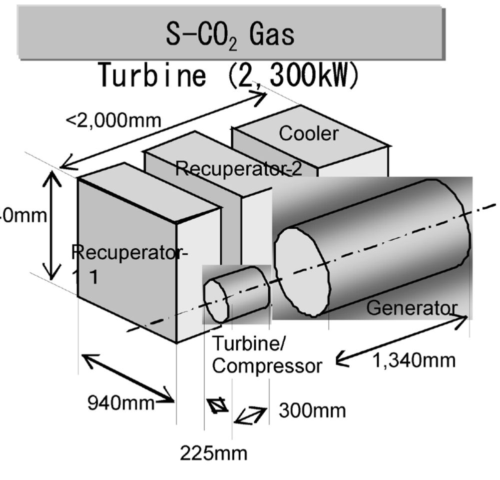


Figure 10. Size of 2.3 MWe prototype S-CO_2 gas turbine

Slika 10. Veličina prototipa S-CO_2 plinske turbine od 2,3 MWe

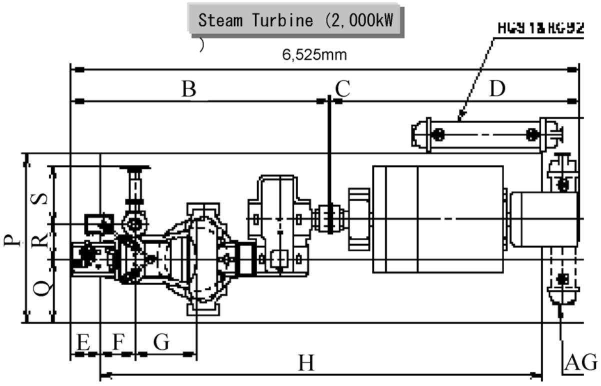


Figure 11. Size of 2 MWe steam turbine commercially available

Slika 11. Veličina komercijalno dostupne parne turbine od 2 MWe

5. Demonstration test

5.1. Reduced scale model for demonstration test

If the prototype machine is reduced to 1/100 power by similarity rule, rotational speed would be 230,000rpm and wheel outer diameter 14 mm, whose manufacturing would be impossible at present. Then, design condition was moderated and shifted to point A in Figure 4 i.e. T_{IT}=550K, p_{CE}=12 MPa and flow rate =1.2 kg/s. Based on this fundamental conditions, turbine expansion ratio was optimized by the cycle calculation.

Based on 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

$$
\begin{aligned}
n_{\mathrm{s}} &= \frac{\omega\sqrt{V}}{\left(H_{\mathrm{ad}}\right)^{3}} \\
\eta &= g\left(n_{\mathrm{s}}\right), \\
d_{\mathrm{s}} &= f\left(n_{\mathrm{s}}\right), \\
D &= \frac{d_{\mathrm{s}}\sqrt{V}}{\left(H_{\mathrm{ad}}\right)^{1/4}}.
\end{aligned} \tag{5}
$$

Functions *f* and *g* are referred to 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 are shown in Figure 12. Cycle efficiency becomes highest at the rotational speed of 100,000 rpm. Outer wheel diameters of compressor and turbine are 20mm and 30mm respectively. The test machine is so small that further aerodynamic loss unique to small scale test should be taken into account because effects of tip clearance, thickness of wheel and the precision of manufacturing becomes dominant as scale becomes smaller. Thus, turbine adiabatic efficiency was assumed to be 0.6 in the calculation shown in Fig.12. Then, predicted cycle efficiency was reduced to 7 % and power output from the machine was expected to be 13 kW.

The compressor and the turbine are to be assembled and integrated into one. To achieve planned aerodynamic performance, sealing and bearing become important design items. Effective sealing against big pressure difference of 4 MPa between compressor part and turbine one is a design issue to be solved. Gas bearing is under consideration, generator/motor with permanent magnet will be adopted.

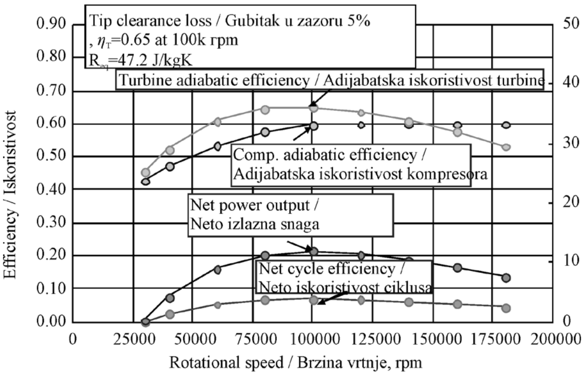


Figure 12. Efficiency and power as function of rotational speed for optimum design point

Slika 12. Iskoristivost i snaga kao funkcija brzine vrtnje za optimalnu projektnu točku

5.2. Test facility

Figure 13 shows schematic diagram of demonstration test facility. Main comp. (compressor1) and turbine are arranged co-axial and manufactured integrated. Re-comp. (compressor2) is of reciprocating type. Piping outer diameters for high and low pressure sides are 25mm and 40mm respectively. Heater capacity is 160 kW.

Four series of tests are to be conducted; 1) Test to confirm the reduction of required compressor work with Re-comp. alone without Main-comp. and Turbine 2) Test to obtain thermal performance of the regenerative heat exchanger (RHX1) 3) Brayton cycle operation to aim self-operation without external work input 4) part flow cycle operation with Re-comp. incorporated in order to generate power 10 kW.

5.3. Compressor aerodynamic performance

The main compressor was designed using COMPAL. Real gas characteristics were considered using property library NIST. Calculation conditions were that flow rate is 1.5 kg/s, rotational speed is 150,000rpm, total temperature at inlet to compressor is 310 K, T_{IT} is 550 K, total pressure at inlet to compressor is 8 MPa, p_{CE} is 12 MPa. The number of blades is 15. Outer diameter of the wheel is 17.5mm being much the same size of that evaluated due to the method by Balje. The blade height is 3mm at inlet and 2mm at exit respectively.

Figure 14 shows static pressure profile along the meridian. Sub-critical region appears locally at inlet due to acceleration. The observed seen pressure recovery at diffuser is significant, which implies that diffuser design is important. Mach number was confirmed to be less than 0.8 in the entire region.

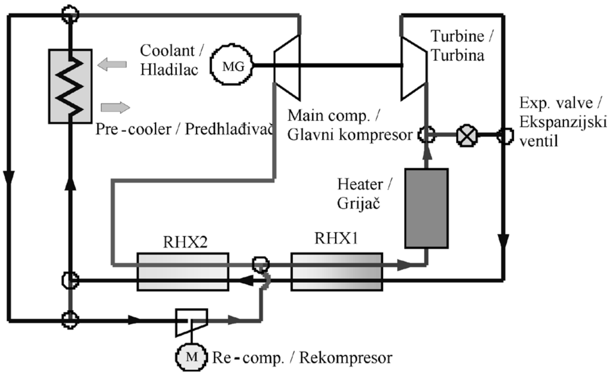


Figure 13. Schematic diagram of demonstration test plant

Slika 13. Shematski dijagram ispitnog demonstracijskog postrojenja

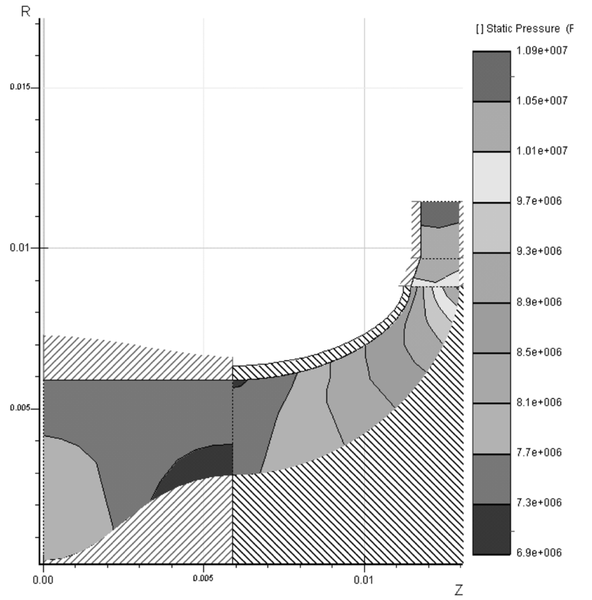


Figure 14. Predicted static pressure profile along meridian of compressor for demonstration test

Slika 14. Distribucija statičkog tlaka duž meridionalneravnine kompresora kod demonstracijskog ispitivanja

6. Conclusion

The principle and demonstration test plan of the supercritical CO_2 gas turbine were described. Conceptual design of 2.3 MWe 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 the fact that net power output was around 10 kWe, CO_2 design recirculation flow rate was 1.2kg/s and thermodynamic state at compressor inlet was 308 K, 8 MPa and at turbine inlet 550 K, 12 MPa 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 100,000 rpm .

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