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Investigation of CO_2 Recovery System Design in Supercritical Carbon Dioxide Power Cycle for Sodium-cooled Fast Reactor

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1. Introduction

The S-CO_2 (S-CO_2) cycle can improve the safety of Sodium-cooled Fast Reactor (SFR) by preventing the sodium-water reaction by changing the working fluid from water to CO_2. The major benefits are 1) relatively high efficiency under moderate turbine inlet temperature (450~750℃) which significantly reduces materials and maintenance related issues, 2) simple layout and physically compact power plant size due to small turbo-machinery and heat exchangers [1], [2]. These are mainly possible because the S-CO_2 Brayton cycle has lower compressing work than other Brayton cycles due to its high density and low compressibility near the critical point. These attributes make easier to achieve higher turbine inlet temperature [1]. Furthermore, the coolant chemistry control and component cooling systems are relatively simple for the S-CO_2 cycle unlike the steam Rankine cycle, and therefore the total plant footprint can be greatly reduced further.

However, certain amount of leakage flow is inevitable in the rotating turbo-machinery since the S-CO_2 power cycle is a highly pressurized system. A computational model of critical flow in turbo-machinery seal is essential to predict the leakage flow and calculate the required total mass of working fluid in S-CO_2 power system. Before designing a computational model of critical flow in turbo-machinery seal, this paper will identify what the issues are in predicting leakage flow and how these issues can be successfully addressed. Also, suitability of this solution in a large scale S-CO_2 power cycle will be discussed, because this solution is for the small scale.

2. Leakage Flow in Turbo-machinery of S-CO_2 Cycle

2.1 Issues with Leakage Flow

For realizing the S-CO_2 power cycle, the turbo-machinery design is the main challenge because CO_2 properties sharply change around critical point. For using the gas foil bearing, CO_2 leakage is essential but stability cannot be achieved due to the high pressure of leakage flow and high shaft speed. Because the shaft speeds are high and the fluid densities are large approaching the density of water. Due to high fluid

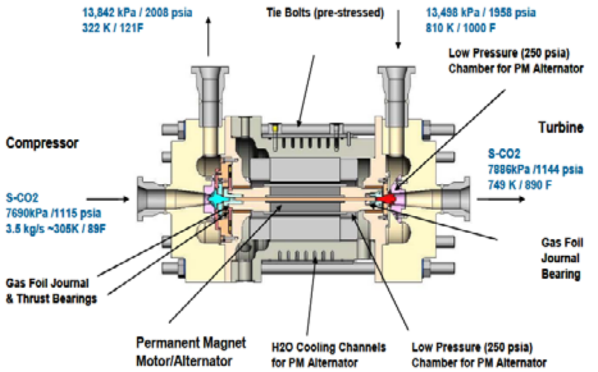

Fig. 1. Turbo-alternator-shaft design for the SNL S-CO_2 test loop [3]

density, significant friction loss is generated within the turbomachinery, which is called windage loss. This occurs especially between the moving shaft and the fluid. The high pressure and high density in the S-CO_2 Brayton cycle can also cause significant thrust loads, and leakage flow rates through seals that must be properly managed by the design. The equation for the windage losses is as follows [4].

$$Pwr = \pi C_d (Re) \rho r_{rotor}{}^4 \omega^3 L_r \tag{1}$$

C_d is the discharge coefficient which is a function of Reynolds number, ρ is the fluid density, r is the radius of the rotor, ω is the angular frequency of the rotor, and L_r is the length of the rotor.

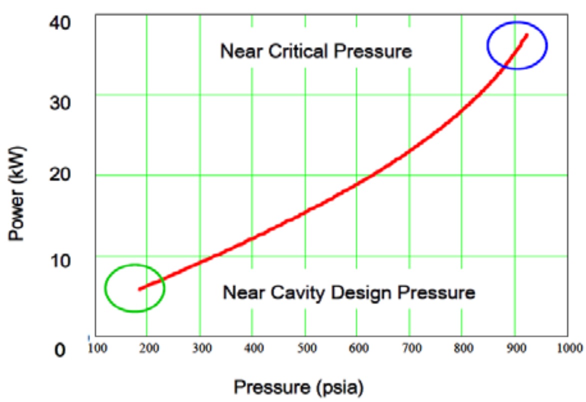

Fig. 2. Calculated windage loss for the S-CO_2 turbo-alternator-compressor as a function of rotor cavity pressure [3]

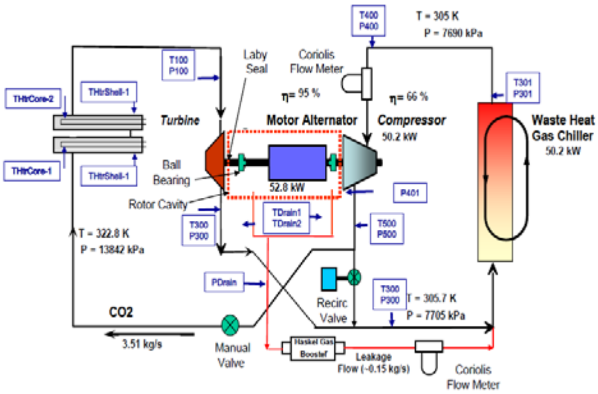

Fig. 3. Simple diagram of the heated un-recuperated S-CO_2 Brayton loop [3]

Note that the windage loss is proportional to the first power of fluid density, third power of shaft speed, and fourth power of rotor radius. This makes the actual power losses very sensitive to the shaft speed and size.

Fig. 2 shows the rotor windage loss and how it varies as a function of cavity pressure. Unfortunately, at 320 K (the maximum assumed cavity temperature) and 900 psia (the maximum assumed cavity pressure that is very near the critical pressure, 1115 psia), the losses are high. At these values the rotor cavity windage losses are expected to be nearly equal the capability of the motor control system and the pumping power for the compressor. Clearly, this magnitude of power loss is unacceptable.

2.2 Solution to Deal with the Problems

To avoid these windage losses, Sandia National Lab (SNL) used a turbo-machinery design that lowers the rotor cavity pressure. At ~200 psia, when the density of the CO_2 is about 0.25 kg/liter, the windage losses are on the order of 4-5 kW. At the design operating conditions, the rotor should be producing approximately 125 kW of alternator power. This then puts the windage loss estimate at 4% of the generated power which seems to be a reasonable value for this small scale test-loop.

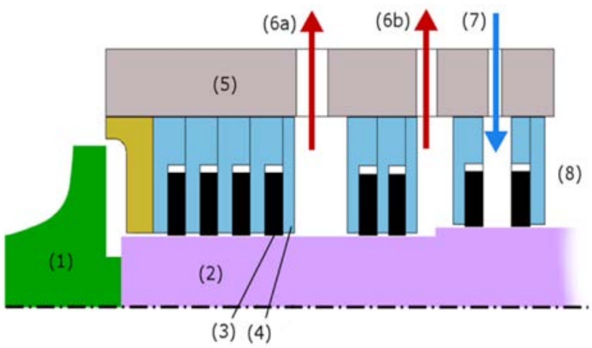

Fig. 4. Design of a high pressure CRS system: impeller (1), shaft (2), carbon ring (3), ring housing or chamber (4), compressor housing (5), first and second lantern with leakage return (commonly to a stage of a lower pressure level) (6a/b), seal gas injection (7), atmosphere (8) [5]

Of course the pressure that can actually be achieved depends on the rate of leakage into the rotor cavity and the capability of the booster pump used to scavenge the gas in the rotor cavity. Fig. 3 shows the diagram of CO_2 recovery system whose leakage flow goes from turbo-machinery to precooler inlet. However, this system is not optimized for CO_2 recovery system because they did not conducted sensitivity study to find the optimal process for decompression and re-pressurization processes. Since upstream pressure of leakage flow is about 14MPa which is almost the same with turbine inlet pressure, directly connecting to the precooler inlet is inefficient.

The compressor inlet and outlet properties including enthalpy and entropy are shown in Table I. Compressor isentropic efficiency 66% is re-confirmed by calculating the data.

Table I: Compressor inlet and outlet properties

	T (K)	P (kPa)	h (kJ/kg)	s (kJ/kg-K)
C inlet	305.00	7690	304.60	1.341
C outlet	322.80	13842	318.82	1.3559
Isentropic	321.38	13842	314.03	1.341

$$\eta_{comp} = \frac{h_{real,out} - h_{in}}{h_{isen,out} - h_{in}} = \frac{(314.03 - 304.60)kJ/kg}{(318.82 - 304.60)kJ/kg} = 0.663 \cong 66\% \qquad (2)$$

The additional compressor work by leakage flow is 2.13kW, which puts the additional compressor work estimate at 1.7% of the generated power and 4.2% of the compressor work.

$$W_{comp,leak} = 0.15kg/s \times (318.82 - 304.60)kJ/kg = 2.13kW \qquad (3)$$

To decrease the rotor cavity pressure, labyrinth seals are good solution. Fig. 5 shows that the pressure of leakage flow is sharply decreased whenever it passes the carbon rings. It means that if pressure of leakage flow can be calculated with a numerical model, optimized turbo-machinery can be designed with stability and no large windage losses.

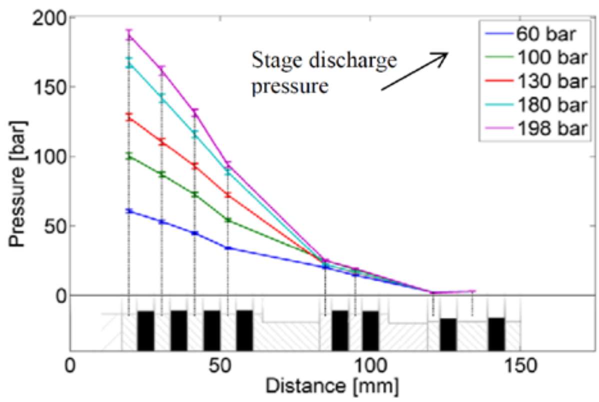

Fig. 5. Measured pressure developments within the seal for relevant points [5]

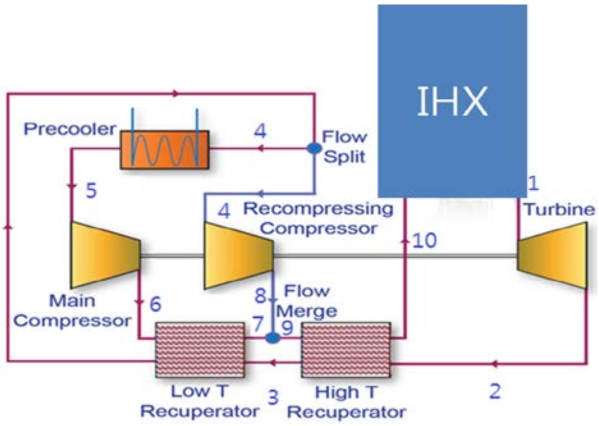

Fig. 6. S-CO_2 recompression cycle single shaft design

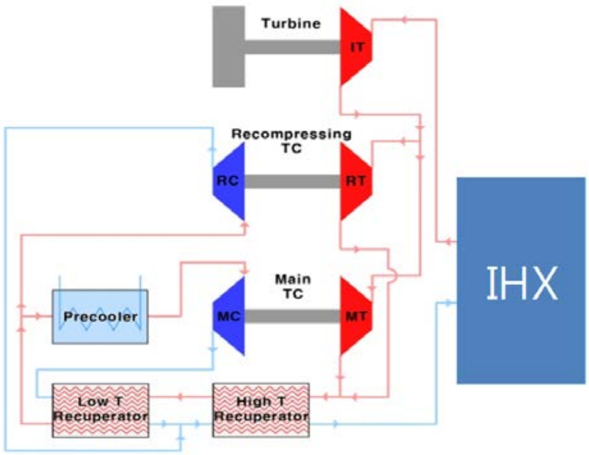

Fig. 7. S-CO_2 recompression cycle triple shaft design

3. Large scale S-CO_2 power cycle

3.1 Difference with Small and Large S-CO_2 Power Systems in Turbo-machinery Leakage

The past study for startup of S-CO_2 Brayton cycle indicates that the fractional power losses due to windage losses would not be expected in larger scale systems used for commercial power generation [6]. Therefore, turbo-machinery problems identified in a small scale system may not be critical problems in a large system.

However, leakage can still occur at three points in the S-CO_2 power system concept for SFR application developed by KAIST research team because turbo-machinery design layout of the innovative SFR is different from conventional gas turbine which has single shaft. Figs. 6 and 7 show the single shaft design and triple shaft design respectively. Three leakage points include main turbine-main compressor, turbine-compressor, and turbine-generator. Since the leakage of turbo-machinery is directly related to S-CO_2 power cycle efficiency and leakage points are more than a small scale S-CO_2 power cycle, an optimization of CO_2 recovery system design is more important.

4. Conclusions

S-CO_2 power cycle has gained interest especially for the SFR application as an alternative to the conventional steam Rankine cycle, since S-CO_2 power cycle can provide better performance and enhance safety.

This paper discussed what the problem in leakage flow is and how to deal with this problem at present. High cavity pressure causing instability of gas foil bearing and large windage losses can be reduced by booster pump used to scavenge the gas in the rotor cavity. Also, labyrinth seals can be another good solution to decrease the rotor cavity pressure. Additionally, difference between large and small scale S-CO_2 power cycle in turbo-machinery leakage is addressed. It is shown that optimization of CO_2 recovery system design is more important to large scale S-CO_2 power cycle.

For economics of the system, designing a process for CO_2 recovery to maintain the system mass at constant is important because this is directly connected to the cycle efficiency. Calculating the leakage amount per day through a turbo-machinery and the ratio of leaked to total CO_2 mass will be conducted for accurate system design and analysis. The comparison of numerical and experimental investigations on S-CO_2 flow through a turbo-machinery seal will be given. Moreover, CO_2 recovery system design for the S-CO2 cycle will be optimized through a sensitivity study to find the optimal process for decompression and re-pressurization processes.

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