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Study of Critical Flow Model Development for Supercritical CO_2 Power Cycle

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1. Introduction

Supercritical CO_2 (S-CO_2) has the potential to be used as the working fluid in power cycle since S-CO_2 has a density as high as that of its liquid phase while the viscosity remains closer to its gaseous phase. Thus, it requires much less work due to its low compressibility as well as relatively small flow resistance.

However, leakage in S-CO_2 turbo-machinery is inevitable since S-CO_2 power cycles are highly pressurized over 7.4MPa. S-CO_2 turbo-machinery needs higher rpm to obtain the equivalent enthalpy change. These characteristics significantly increase windage loss, friction loss in the turbo-machinery, especially between a moving shaft and the fluid. The parasitic loss caused by the leakage flow should be minimized since this can substantially influence the cycle efficiency. Therefore, estimating the critical flow in a turbo-machinery seal is essential to predict the leakage flow rate and calculate the required total mass of working fluid in a S-CO_2 power system to minimize the parasitic loss. This paper presents experimental and numerical analyses of two-phase critical flow while special attention is given to the turbo-machinery seal design. Experimental data obtained from a simple nozzle under two-phase condition are reported to study the flow characteristic and provide validation data for the numerical model. Also, comparison of data with the developed isentropic critical flow model is performed by utilizing the experimental data of CO_2 critical flow facility.

2. CO_2 Critical Flow Model

The CO_2 flow in a turbo-machinery through seal will be mostly choked flow due to a large pressure difference between the rotor cavity and lower pressure station. However, only limited information is available for estimating CO_2 critical flow [1, 2]. In this work, the final goal is to develop a model that can successfully represent transient response of the high pressure side and low pressure side during the CO_2 leak process realistically. In a real operation, the CO_2 flow is a compressible flow, which can be choked depending on the pressure difference. The mass flow rate is determined from the seal size and the mass flux determined from the flow resistance and pressure drop. To model the CO_2 leak flow mechanism realistically, several complex factors should be considered and a few assumptions are necessary for model simplification. Thus, simplified flow model was developed as the first step toward the complete model to predict the CO_2 leak flow process. The base calculation mechanism of CO_2 critical flow model for calculating the leak rate of CO_2 in a turbo-machinery was referred from the CO_2 leak model of Na-CO_2 heat exchanger in the S-CO_2 power cycle [3]. It was assumed that CO_2 flows through a nozzle from a high pressure CO_2 tank to a low pressure CO_2 tank, and the nozzle diameter plays the same role as the seal size.

For an isentropic flow, the frictional pressure loss and heat transfer are neglected thus the flow state can be easily calculated with the following governing equations (i.e. continuity equation, critical-pressure ratio equation, Mach number equation with pressure ratio, and mass flux equation from continuity equation) (John and Keith, 2006):

$$G = \rho V_{elocity} = constant \tag{1}$$

$$\frac{P_0}{P_{critical}} = (1+\frac{\gamma-1}{2})^{\gamma/(\gamma-1)} \tag{2}$$

$$M = \sqrt{\frac{2}{\gamma-1}[(\frac{P_0}{P})^{(\gamma-1)/\gamma}-1]} \tag{3}$$

$$G = \frac{P_0}{\sqrt{RT_0}}\sqrt{\gamma}M(1+\frac{\gamma-1}{2}M^2)^{-\frac{\gamma+1}{2(\gamma-1)}} \tag{4}$$

$$G_{max} = \frac{P_0}{\sqrt{RT_0}}\sqrt{\gamma}(\frac{\gamma+1}{2})^{-\frac{\gamma+1}{2(\gamma-1)}}(M_{exit}=1.0) \tag{5}$$

Based on the above governing equations, the critical pressure obtained from Eq. (2) is compared to the low pressure side at every time step. The choked condition is then checked. If the flow is not choked, Mach number is calculated from Eq. (3) and it is applied to Eq. (4). On the other hand, Eq. (5) with Mach number of unity is used to calculate the choked mass flux.

To simplify the model, it was assumed that temperature and pressure of CO_2 at seal exit are at equilibrium with CO_2 in the low pressure tank. This assumption actually neglects expansion process of CO_2 at the nozzle exit although the CO_2 pressure at the exit is higher than that of CO_2 in the low pressure tank when the flow is choked. Fig. 1 shows a flowchart of calculation process of the CO_2 critical flow model.

Based on the algorithm of the CO_2 critical flow model, the sensitivity study of the transient response during the leak process was performed while varying nozzle diameter and initial conditions (temperature and pressure).

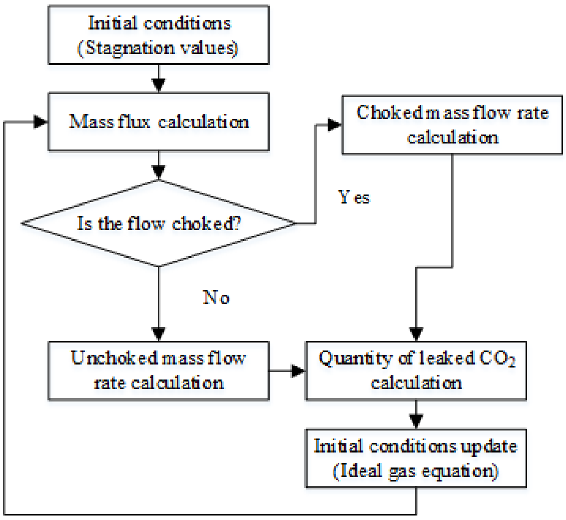

Fig. 1. Flow chart of CO_2 critical flow model

3. CO_2 Critical Flow Experiment

A critical flow test facility was constructed to validate the S-CO_2 critical flow model. Fig. 2 shows the designed experimental facility for the CO_2 critical flow simulation and the design specifications are shown in Table I. For accurate measurements, total nine thermocouples and seven pressure gauges are installed on the critical flow facility as shown in Fig. 3.

Table I: Design specifications of the experimental facility

	Design Parameters	
High/Low-pressure tank	Pressure (MPa)	22
	Temperature (℃)	150
	Volume (L)	47 (I.D.:200 mm, H: 1600mm)
Pipe connecting two tanks	Internal diameter (mm)	57
	Length (mm)	1090
Heater (Jacket-type)	Electric capacity (kW)	5
Valve type	Ball valve	

Table II: Summary of experimental cases

		Case 1	Case 2	Case 3
High-pressure tank	Pressure (MPa)	10.04	13.43	20.16
	Temperature (℃)	103.3	161.5	151.2
Low-pressure tank	Pressure (MPa)	0.101	0.101	0.101
	Temperature (℃)	14.5	15.6	14.1

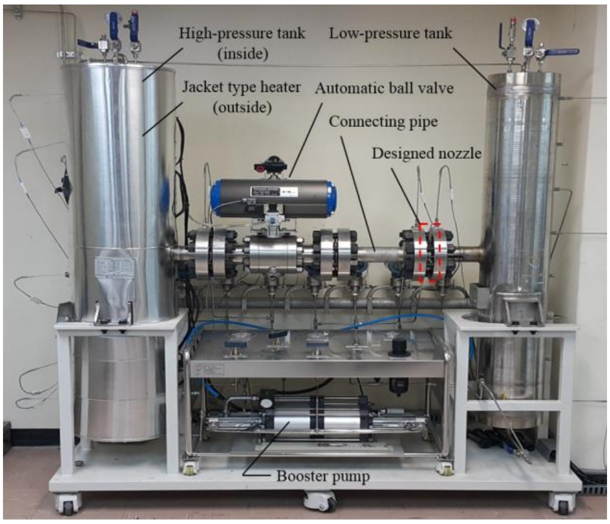

Fig. 2. S-CO_2 critical flow experimental facility

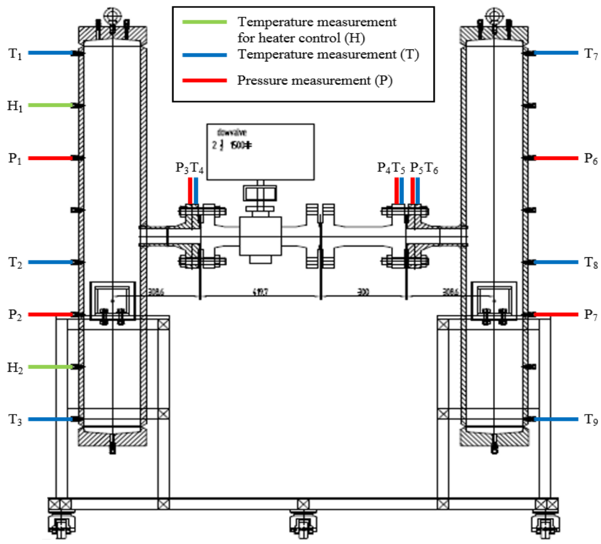

Fig. 3. The location of measurements in the experimental facility

Three thermocouples and two pressure gauges are installed on the high-pressure tank (left) and the low-pressure tank (right), respectively. They are located at top, middle, and bottom sections of each tank. One thermocouple and one pressure gauge are installed at the inlet and the outlet of the nozzle as well as between high-pressure tank and a valve, respectively.

To control the initial temperature of the high-pressure tank, the heater covers the external surface of the high-pressure tank. The pressure of the high-pressure tank can be controlled by injecting the CO_2 gas through a booster pump. The ball valve is automatically opened and driven by hydraulic pressure (1MPa) from an air compressor to minimize the valve opening time. The nozzle geometry can be modified later to test for different flow conditions.

Initial conditions of the low-pressure tank is maintained at room condition (about 15℃, 0.101MPa) to maximize the pressure difference and have a long depressurization time for stable measurement of the CO_2 critical flow. The initial pressure of the high-pressure tank is set at three different values for the testing; 10MPa, 13MPa, and 20MPa. The initial temperature of the high-pressure tank is set between 100℃ and 150℃. The temperature is high enough to maintain the gaseous state after the expansion. The nozzle has the diameter of 1.5 mm and length of 5.0 mm as shown in Fig. 4. Experimental conditions are summarized in Table II.

Fig. 4. Feature of nozzle (with the diameter of 1.5 mm and length of 5.0 mm)

4. Experimental Results

It was verified that the developed S-CO_2 critical flow model based on an isentropic flow assumption has a good accuracy for predicting the S-CO_2 critical and subcritical flows and the critical flow model predictions are all within the uncertainty band of the experimental data as shown in Fig. 5, 6, and 7.

Thus, it was reasonable to conclude that the isentropic critical flow model is sufficient to predict the critical and subcritical flows of S-CO_2 to the low pressure side.

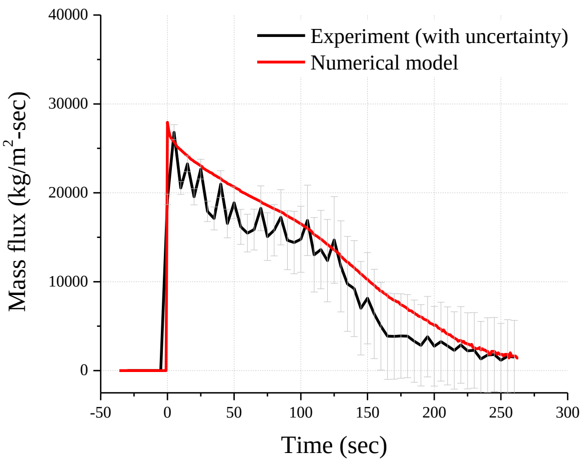

Fig. 5. Comparison of mass flux between the experimental results and CO_2 critical flow model (Case 1)

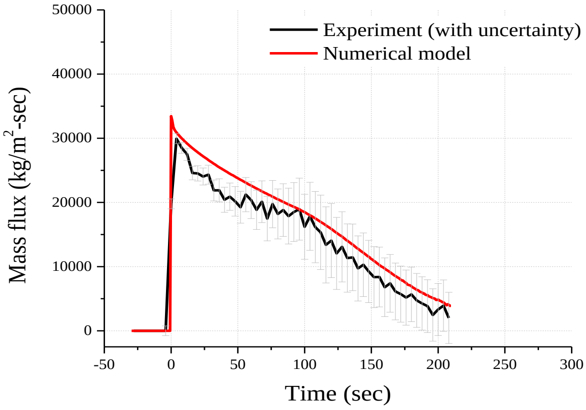

Fig. 6. Comparison of mass flux between the experimental results and CO_2 critical flow model (Case 2)

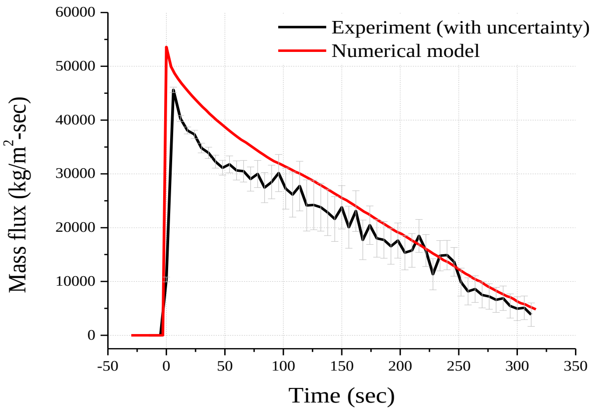

Fig. 7. Comparison of mass flux between the experimental results and CO_2 critical flow model (Case 3)

To identify that the developed S-CO_2 critical flow model can estimate the dynamic behavior for critical flow with two-phase condition, further experiment was conducted. The initial temperature and pressure of the high pressure tank were set to 34.5℃ and 8.1MPa which is similar to the compressor inlet conditions of S-CO_2 Brayton cycle. The CO_2 phase of high pressure tank will be changed from supercritical state to liquid state, and then to the gaseous state after the expansion.

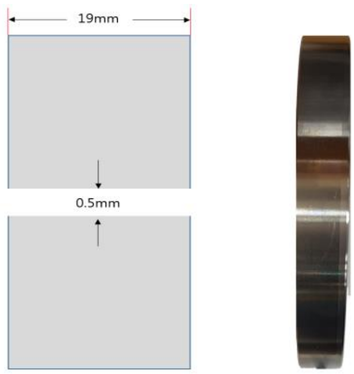

Fig. 8. Internal and external geometry of simple nozzle

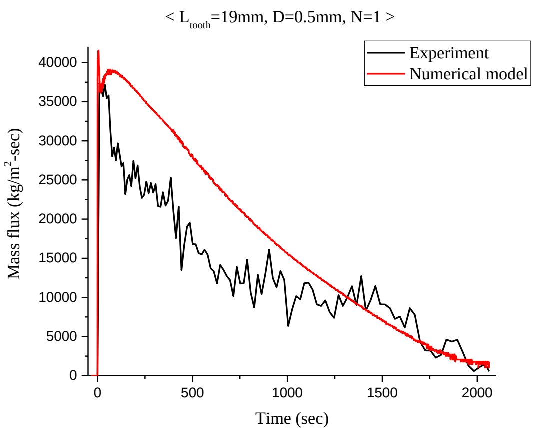

Fig. 9. Comparison of mass flux between the experimental results and CO_2 critical flow model with two-phase flow

The internal and external of the nozzle are shown in Fig. 8, and the length and nozzle diameter are 19mm and 0.5mm, respectively.

Although the end region has small difference between experimental and numerical results, there is a big mass flux difference especially near the choked flow region as shown is Fig. 9. The main reason is the ideal gas assumption. The developed CO_2 critical flow model based on the isentropic flow was assumed that CO_2 in operating condition behaves like ideal gas. (Compressibility factor ≈ 1). To estimate the dynamic behavior of CO_2 with two-phase condition, ideal gas assumption should be revised since CO_2 has the low compressibility near the critical point.

5. Conclusions

Experimental and numerical study of critical flow with two-phase condition was conducted for transient simulation model while special attention is given to the turbo-machinery seal design. The experiment results identified that the developed S-CO_2 critical flow model based on an isentropic flow assumption cannot estimate the dynamic behavior for critical flow with two-phase condition. To estimate the dynamic behavior of CO_2 with two-phase condition, ideal gas assumption should be revised since CO_2 has the low compressibility near the critical point.

To reflect the real gas effect, more complex assumptions will apply to the CO_2 critical flow model. Also, further experiments with labyrinth seal geometry nozzle under various conditions will be conducted to validate the revised CO_2 critical flow model.

REFERENCES

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