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COMPARISON OF CO_2 CRITICAL FLOW MODEL BASED ON HENRY-FAUSKE MODEL WITH TWO-PHASE FLOW

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ABSTRACT

Understanding flows and predicting leakages past seals are important for designing supercritical CO_2 (S-CO_2) rotating machinery. Since the seal leakages can have a direct impact on the cycle efficiency. If pressure and temperature trends of leakage flow can be predicted analytically then recovery system can be optimized for better system's overall efficiency. Furthermore, the authors are interested in the transient behavior of seal leakage flow, since the supercritical CO_2 power system can loose inventory during part-load operation as well as during start-up and shut down sequences which was experienced in a few operating test facilities. Therefore, the off-design seal performance is also equally important while this is not dealt thoroughly yet. Thus, a transient simulation for estimating the critical flow in a turbo-machinery seal is essential to predict the leakage flow rate and to calculate the required total mass of working fluid in an S-CO_2 power system.

This paper describes the test data using a CO_2 critical flow experimental facility with three orifice configurations to model the flow resistance of a rotating shaft labyrinth seal. This data is used for validation of an existing transient analytical tool developed for transient hydraulic system analysis. Contained within this code is the analysis approach of Henry/Fauske from 1971 for two phase critical flow of one-component mixtures. The predictions of transient pressure, temperature, and flow profiles for critical and sub-critical flows of CO_2 relative to blowdown test data are presented to show that reasonable results can be generated from the code.

INTRODUCTION

A S-CO_2 Brayton cycle can improve safety of a 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 ºC) which significantly reduces materials and maintenance related issues, 2) simple layout and physically compact power plant size due to smaller turbo-machinery and heat exchangers [1, 2]. These advantages are possible mainly 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 [1]. Furthermore, the coolant chemistry control and component cooling systems are relatively simpler for the S-CO_2 cycle than 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 via seals since the S-CO_2 power cycle is a highly pressurized system. Leakage can still occur at three points in the S-CO_2 power system for a Sodium-cooled fast reactor (SFR) application because turbo-machinery design layout of the innovative SFR is different from conventional gas turbine which has single shaft [3].

Figure 1 shows the triple shaft design. Three leakage points include turbine-main compressor, turbine-recompressing compressor, and turbine-generator. The parasitic loss caused by the leakage flow should be minimized since this greatly influences the cycle efficiency. The flow past labyrinth seals in a turbo-machinery will be choked (i.e. critical flow) due to high

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pressure difference between supercritical CO_2 turbo-machinery and ambient conditions, while the fluid inside the seal should go through transition from supercritical phase to gaseous phase. Since the leakage flow of turbo-machinery is directly related to the S-CO_2 power cycle efficiency, a computational model of critical flow for a larger power system turbo-machinery seals is essential to predict the amount of leakage flow. From the calculation results, the total required mass of working fluid and the charging rate of the S-CO_2 power system can be determined.

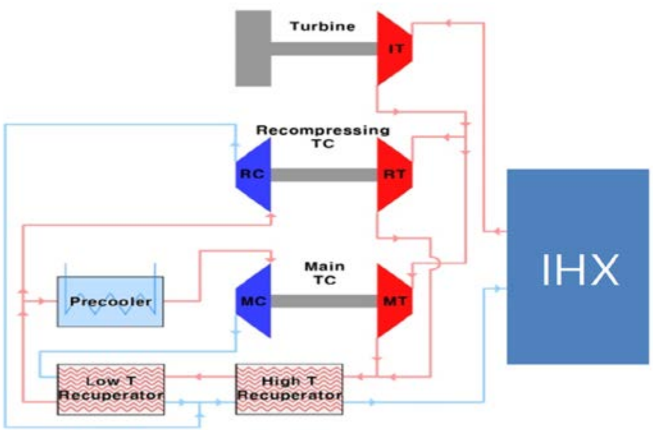

Figure 1: Triple-shaft design of S-CO_2 recompression cycle

Studies on CO_2 leak in a S-CO_2 power cycle were conducted previously. Mignot et al. [4] described the results of an experiment to measure the critical mass flux for numerous stagnation thermodynamic conditions, geometry and outlet tube roughness. 1D homogeneous equilibrium model showed relatively good (less than 10 % error) prediction of the test data but it is not directly related to the critical flow in an S-CO_2 turbo-machinery because the data were used for characterizing the behavior of supercritical fluids during a blowdown or rapid depressurization to validate certain aspects of safety analyses. Hylla et al. [5] gave an overview of numerical and experimental investigations on super-critical CO_2 flow through carbon floating ring seals. Simulation model considers the real gas effect, temperature deformation and the shaft rotation. However, the data were not disclosed since it is proprietary information of MAN Diesel & Turbo SE Inc. and there is a limitation to retrieve the key information for developing a numerical model. For instance, mass flow rate of the leakage flow cannot be obtained from this research as the dynamic behavior of lower pressure stage was not shown in this study. The mass flow rate of leakage flow should be reported for designing an inventory control system. A recent study by Yuan et al. [6] presents a numerical study of S-CO_2 flow using the computational fluid dynamic (CFD) simulation in see-through labyrinth seals. Various designs and conditions have been tested to study the flow characteristic and provide validation data for the numerical model. However, the upstream conditions (10MPa, 45ºC) were fixed and it is only for lower temperature CO_2, which more data for CO_2 leak flow are necessary to understand the full range of critical flow.

This paper presents both numerical and experimental studies of S-CO_2 critical flow while special attention is given to the turbo-machinery seal design. The thermal-hydraulic system transient analysis code MARS and the selected critical flow model are described. Experiments with a simple nozzle geometry were conducted to validate the code first. The comparison of numerical and experimental results of S-CO_2 critical flow will be presented. Furthermore, the validation of the MARS code will be carried out by utilizing the experimental data of CO_2 critical flow facility. In addition, experimental and numerical analysis of two-phase critical flow are conducted while special attention is given to the turbo-machinery seal design. Experimental data obtained from simple nozzle with two-phase condition are reported to study the flow characteristics and provide validation data for the numerical model. Also, comparison with MARS code based on Henry-Fauske model is performed by utilizing the experimental data of CO_2 critical flow facility.

CO_2 CRITICAL FLOW EXPERIMENT

Kim et al. constructed a critical flow test facility to validate the S-CO_2 critical flow model [7]. Figure 2 shows the designed experimental facility for the CO_2 critical flow simulation. The content of the experiment is that CO_2 flows from high-pressure tank (left) to low-pressure tank (right) through the designed nozzle, and pressure and temperature of each position are measured every second. Two tanks are connected by a 1090mm pipe and designed labyrinth seal geometry nozzle is installed between the ball valve and low-pressure tank.

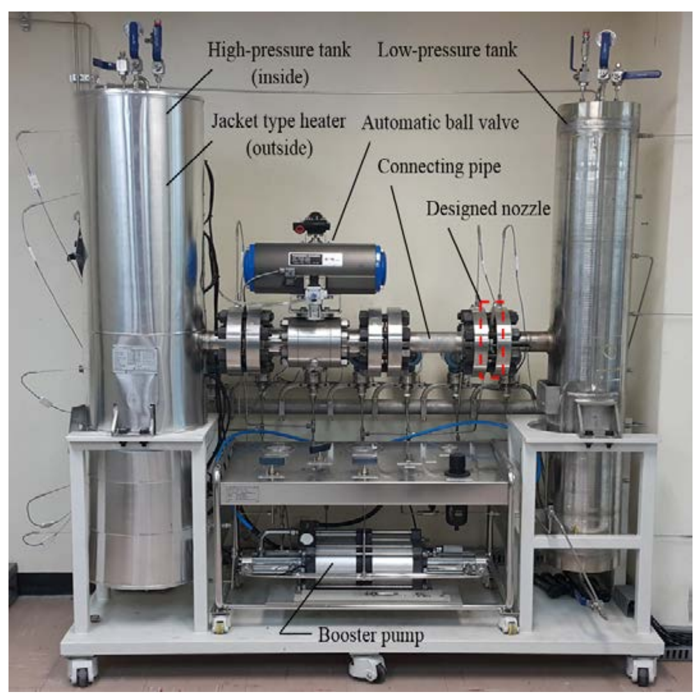

Figure 2: S-CO_2 critical flow experimental facility

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Table 1: Design specifications of the experimental facility

	Design Parameters	
High/Low-pressure tank	Pressure (MPa)	22
	Temperature (ºC)	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 2: Known constant values and uncertainties for calculation

	Known value	Uncertainty		Uncertainty
D_{nozzle} (mm)	1.5/0.5	±0.02	P (kPa)	±(0.00025P)
D_{tank} (mm)	200	±0.5		
H_{tank} (mm)	1600	±1.2	T (ºC)	±(0.15+0.002T)
△Time (sec)	1	±0.03		

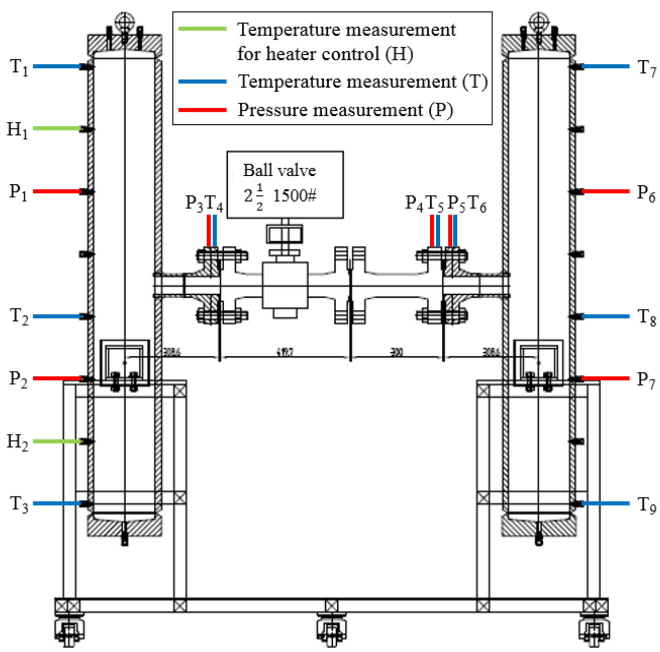

Figure 3: The location of measurements in the experimental facility

Each tank has 200mm of inner diameter, 1600mm of height, and 47 liters of volume and temperature and pressure limits are 150ºC and 22MPa. 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.

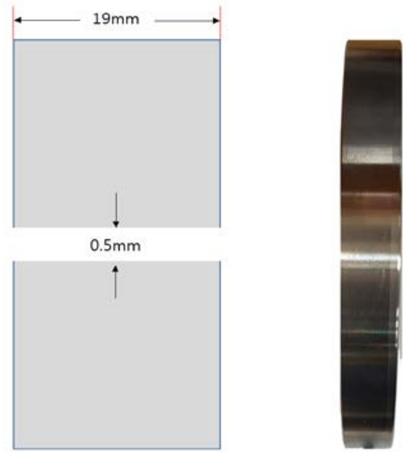

Figure 4: Internal and external geometry of simple nozzle

The nozzle geometry can be modified later to test for different flow conditions. The detail design specifications are shown in Table 1. For accurate measurements, total nine RTDs (Resistance Temperature Detectors) and seven pressure gauges are installed on the critical flow facility as shown in Figure 3. Three RTDs 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 RTD 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. Moreover, one RTD and one pressure gauge are installed between high pressure tank and a ball valve. The known constant values and the uncertainties are summarized in Table 2.

Initial conditions of the low-pressure tank is maintained at room condition (about 15ºC, 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 and temperature of the high pressure tank were set to 15.5MPa and 120ºC. The temperature is high enough to maintain the gaseous state after the expansion.

The Barber-Nichols Inc. which is the first mover of turbo-machinery in S-CO_2 Brayton cycle suggested two applicable mechanical seals [8]: 1) Labyrinth seal, 2) Dry gas seal. Two seals are most widely used mechanical seals in high speed rotating machines. The labyrinth seal is non-contact sealing action and it is composed of many grooves, so that the fluid has to pass through a long and difficult path to escape. Leakage amount is proportional to the gap area and inversely proportional to the tooth number.

The dry gas seal or dry lift off seal is non-contacting, and dry-running mechanical face seal which consists of a mating (rotating) ring and a primary (stationary) ring. Although the dry gas seal is preferred over the labyrinth seal in larger supercritical CO_2 power systems, this paper focused on the labyrinth seal. This is because the authors believe that supercritical CO_2 turbomachinery should be hermetic type to make the system more compact and simpler, thus the labyrinth

DOI: 10.17185/duepublico/46092

seal is more likely to be selected for such machinery. The internal and external geometry of simple nozzle is shown in Figure 4.

The experimental procedure is as follows. Firstly, the ball valve was initially closed to separate the high and low pressure tanks before pressurizing the high pressure tank to the target pressure. To pressurize the high pressure tank with CO_2, the air compressor and the booster pump were used. With the hydraulic power of compressed air, the high pressure tank was filled with CO_2 from a storage tank until the pressure reaches the maximum (~13 MPa) pressure. After reaching the pressure, the temperature of the high pressure tank was set to meet the target point by controlling the heater. A venting process is needed during heating stage because the pressure of high pressure tank is increased during the heating of CO_2. By controlling the heater and the vent valve, the target initial conditions can be obtained. After setting the initial conditions, the heater was turned off and the ball valve was opened by hydraulic power of compressed air. The high pressure CO_2 started to flow to the low pressure tank through the nozzle as soon as the ball valve was opened. All temperatures and pressures in each point were measured every second until both tanks reach equilibrium.

NUMERICAL MODEL

To perform a transient simulation for estimating the critical flow in a turbo-machinery seal, we conducted analysis with the Multi-dimensional Analysis of Reactor Safety (MARS) code which is a thermal-hydraulic system transient analysis code developed by Korea Atomic Energy Research Institute (KAERI). MARS code has been developed for the realistic thermal-hydraulic system analysis of light water reactor transients. The bases of MARS are the RELAP5/MOD3.2.1.2 (Reactor Excursion and Leak Analysis Program) and the COBRA-TF (Coolant Boiling in Rod Arrays – Two Fluid) codes of USNRC (United States Nuclear Regulatory Commission). The RELAP5 code is a versatile and robust system analysis code based on one-dimensional two-fluid model for two-phase flows whereas the COBRA-TF code is based on a three-dimensional, two-fluid, three-field model. The two codes were consolidated into a single code by integrating the hydrodynamic solution schemes, and unifying various thermal-hydraulic models, EOS and I/O features. The one-dimensional conservation equations for mass, energy, and momentum of the flow are solved in MARS. The numerical solution method for the MARS hydrodynamic model is semi-implicit scheme. The flow model for two-phase flow is two-fluid model (1D module), and two-fluid, three-field model (3D Vessel module). The sources of the code were fully restructured using the modular data structure and a new dynamic memory allocation scheme of FORTRAN (Formula translation). In addition, a new multidimensional fluid model has been developed and implemented to the RELAP5 system analysis module in order to overcome some limitations of COBRA-TF 3D vessel module. Now MARS became currently a popular thermal-hydraulic tool in use for the analyses of nuclear reactor transients, experimental facility simulations and various safety research purposes in Korea.

To analyze the behavior of S-CO_2 critical flow at the nozzle, Henry-Fauske critical flow model [9] which is the default critical flow model in MARS was applied. It was developed for “nozzles, orifices, and short pipes” and includes an effect of thermal non-equilibrium upon the critical flow rate. By combining the mass flux at the throat for high velocities (1) and boundary condition at critical flow (2), the critical mass flux for an isentropic homogeneous mixture with flashing can be written as Eq. (3).

$$G_t^{-1} = -\left[\frac{d\{xu_v + (1-x)u_l\}}{dP} \right]_t \tag{1}$$

$$\left. \frac{dG_c}{dP} \right|_t = 0 \tag{2}$$

$$G_c^{\,2} = \frac{-1}{\left\{ x\dfrac{\partial v_v}{\partial P} + (1-x)\dfrac{\partial v_l}{\partial P} + (v_v - v_l)\left.\dfrac{\partial x}{\partial P}\right|_s \right\}_t} \tag{3}$$

G_t is mass flux at the throat, x is quality, $u_{v.l}$ is velocity of vapor and liquid, P is pressure, G_c is mass flux of critical flow, and $v_{v.l}$ is vapor and liquid specific volume. Embedded in Eq. (3) is the assumption that the two phases move with the same velocity, that is, the slip ratio is unity. The amount of thermal non-equilibrium at the throat is more important in determining the critical flow rate than the amount of mechanical non-equilibrium. Thus, it is assumed that the phase velocities are equal. Henry-Fauske then argued that for normal nozzle configurations, there is little time for mass transfer to take place, and it is reasonable to assume that the amount of mass transferred during the expansion is negligible. Similarly, the amount of heat transferred between the phases during the expansion is also negligible, so that the liquid temperature is essentially constant. Since wall shear, heat transfer with the environment, and interfacial viscous terms were neglected, the system entropy during the expansion was assumed constant. This result along with the assumptions of negligible amounts of interphase heat and mass transfer imply that each phase expands isentropically.

The above assumptions eliminate the need to calculate the liquid specific volume and the quality at the throat, and also provide a relation for the vapor specific volume in terms of the throat pressure and the upstream conditions. Putting all of the above assumptions into Eq. (6), the final expression for the critical flow value of the mass flux is:

DOI: 10.17185/duepublico/46092

$$G_c^{\,2} = \left[\frac{x_0 v_v}{nP} + (v_v - v_{l,0}) \left\{ \frac{(1-x_0)N}{(s_{v,eq} - s_{l,eq})} \frac{ds_{l,eq}}{dP} - \frac{x_0 C_{p,v}(1/n - 1/\gamma)}{P(s_{v,0} - s_{l,0})} \right\} \right]_t^{-1} \tag{4}$$

If the thermal non-equilibrium factor, N, is taken to be unity, the prediction of Eq. (4) is close to that of the homogeneous equilibrium model, and if equals zero the solution is approximately the homogeneous frozen model. As noted by Henry & Fauske, the experimental results of Starkman et al. indicate that the critical flow rates are in relatively good agreement with the homogeneous equilibrium model for stagnation qualities greater than 0.10 [10].

Before the transient simulation for estimating the critical flow in a turbo-machinery seal, the validation of MARS results by comparing to experimental data from a simple nozzle was performed. The simple nozzle has the diameter of 1.5mm and length of 5.0mm and initial conditions of high-pressure tank is 13.5MPa and 160ºC.

In the MARS analysis, the authors assumed that the actual mass flow rate is the same as the theoretical mass flow rate. Also the homogeneous equilibrium model was utilized. For this reason, discharge coefficient was set to unity and throat equilibrium quality was set to 0.14 [9, 11]. Figure 5 shows the geometry modeled in MARS, and it consists of five pipes, one valve and one nozzle. Each tank is divided to three parts, so it has total nine volumes. The heat structures with initial temperature were added to model the thermal inertia effect.

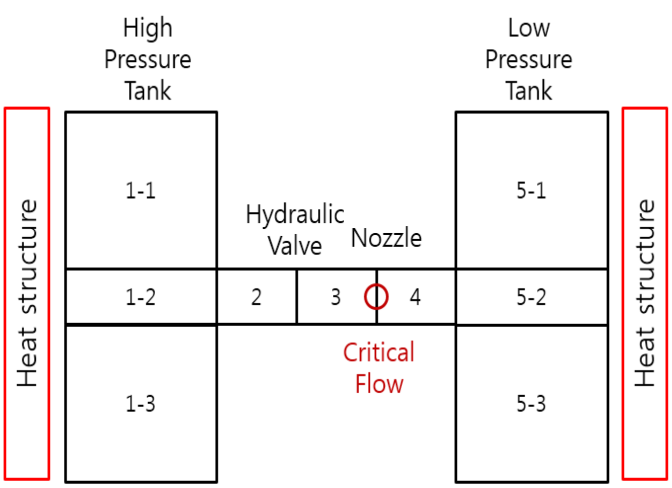

Figure 5: Input geometry and nodalization of MARS code

As shown in Figure 6, it is shown that the MARS results are within 4.7% of experiments and have a good accuracy for predicting the S-CO_2 critical and subcritical flows. Thus, it is reasonable to conclude that the MARS code is sufficient to predict the critical and subcritical flows of S-CO_2.

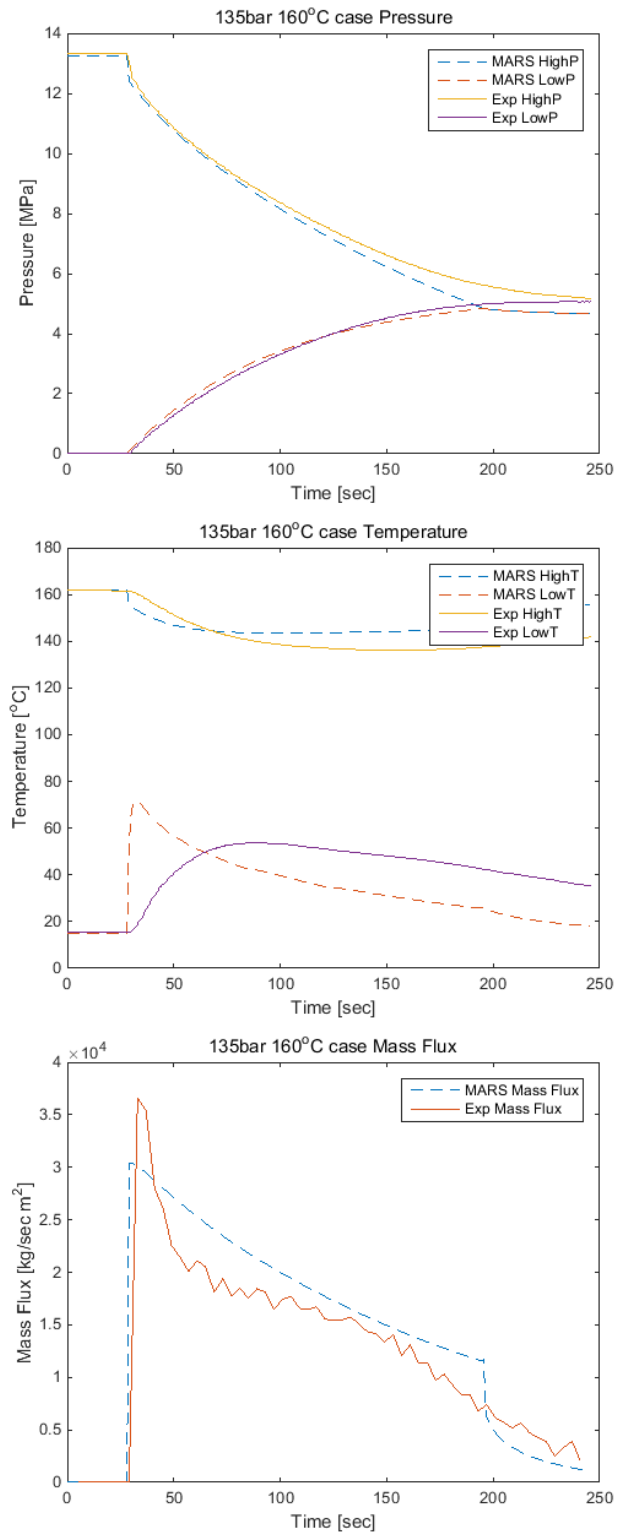

Figure 6: Pressure, temperature and mass flux plots between the experimental results and the MARS results (with simple nozzle)

DOI: 10.17185/duepublico/46092

EXPERIMENTAL RESULTS

To further evaluate if the MARS code can estimate the dynamic behavior under two-phase condition as well, more experiment was conducted. The initial temperature and pressure of the high pressure tank were set to 120ºC and 15.5MPa, respectively, to maintain the gaseous state of the high-pressure tank after the expansion. The CO_2 phase of low- pressure tank will be changed from gaseous state to liquid state after the expansion, which infers that during the critical flow experiment two phase conditions can be obtained. The quality of CO_2 in the low-pressure tank was calculated by using pressure and temperature in each time step. By maintaining the high-pressure tank in supercritical or gaseous state, mass difference can be calculated with the properties of high-pressure tank. Then, the quality of CO_2 in low-pressure tank is calculated with equation (8) since the volume of each tank is already known value, which is 47 liters.

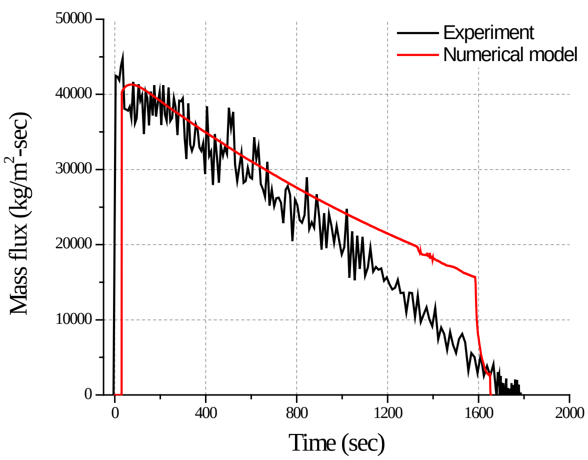

Figure 7: Comparison of mass flux between the experimental and numerical results

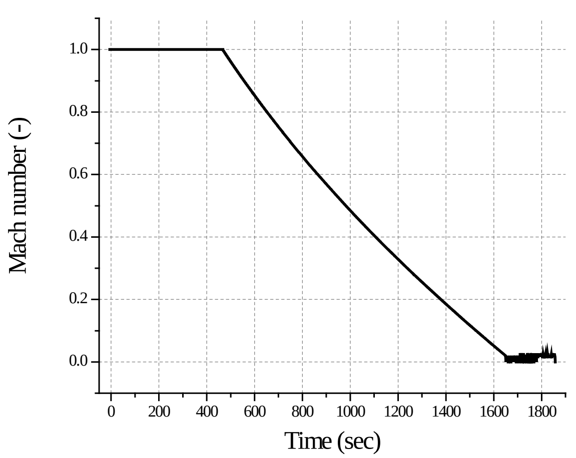

Figure 8: Mach number of experimental result

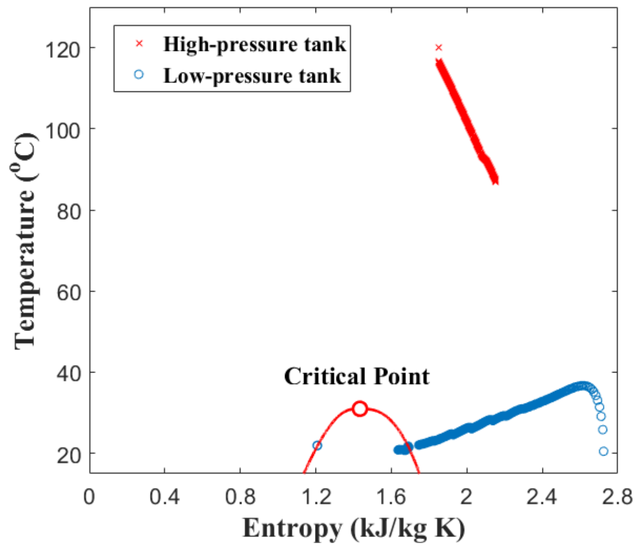

Figure 9: T-s diagram of two-phase flow experimental result

$$x = \frac{v_l(t) - v(t)}{v_l(t) - v_v(t)} \tag{8}$$

where, x is quality of CO_2 in low-pressure tank, and $v_{l,v}$ is specific volume of liquid and vapor.

Figure 7 shows the comparison of mass flux between the experimental data and numerical results, Figure 8 shows the Mach number of experimental result, and Figure 9 shows the T-s diagram of the two-phase flow experimental data. As shown in Figure 7, the difference of equilibrium reaching time between numerical and experimental results is within 1.45%. This indirectly confirms that Henry-Fauske model can estimate the equilibrium reaching time of CO_2 critical flow experimental result quite well. However, the deviation of mass flux is bigger from the experiment around the equilibrium as it can be observed from Figure 7. The main reason of this deviation is may come from the error of property database.

The temperature and pressure of the two tanks were measured during the experiment and the CO_2 properties can be obtained from the NIST reference fluid thermodynamic and transport properties database (REFPROP) after the experiment [13]. The NIST database was also examined in the previous study [14] for the loopback error which was determined to be below 10^{-7} order. However, the fluid thermodynamic and transport property tables use linear interpolation for segments between table search argument values in the MARS code.

Due to the characteristics of thermodynamic property variation of CO_2 near the critical point as shown in Figure 10, the design and analysis methodologies of the CO_2 power cycle components which operate near the critical point should be precise.

DOI: 10.17185/duepublico/46092

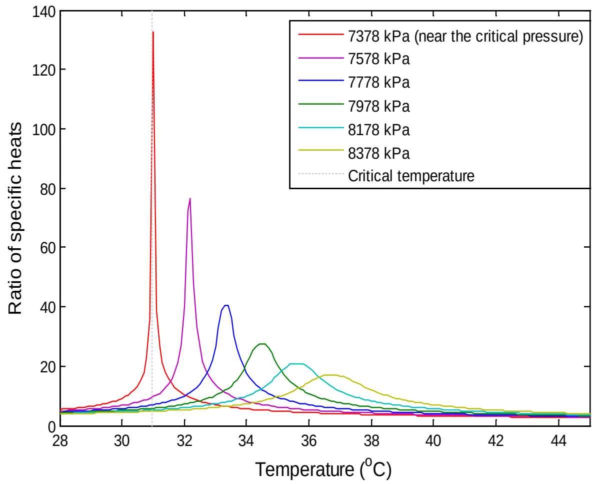

Figure 10: Specific heat ratio variation of S-CO_2 near the critical point [12]

As mentioned above, MARS code especially focus on the steam tables since it has been developed for the realistic thermal-hydraulic system analysis of light water reactor transients. Therefore, correlations and models for CO_2 two-phase flow should be re-examined, which will be the next step of this research.

CONCLUSIONS AND FURTHER WORKS

Predicting the leak flow rate is imperative to sustain the high performance of S-CO_2 power cycle. Thus, transient simulation of S-CO_2 critical flow with MARS code which is a thermal-hydraulic system transient analysis code for the realistic thermal-hydraulic system analysis of light water reactor transients was performed to identify the mass flow rate of CO_2 leakage in turbo-machinery.

The one-dimensional conservation equations for mass, energy, and momentum of the flow are solved in MARS. The numerical solution method for the MARS hydrodynamic model is semi-implicit scheme. The flow model for two-phase flow is two-fluid model (1D module), and two-fluid, three-field model (3D Vessel module). To analyze the behavior of S-CO_2 critical flow at the nozzle, Henry-Fauske critical flow model was applied. This study does not immediately contribute for reducing the leakage flow, but it gives useful information for further understanding of labyrinth seal leakage performance and eventually design optimization.

To validate the MARS code with experimental results, experiments of CO_2 critical flow with simple geometry nozzle were performed. The temperature, pressure, and mass flux of MARS code has similar trend with the result of experimental results in all cases. It was identified that the MARS code can simulate the critical and subcritical flow behavior of S-CO_2 reasonably.

Lastly the two phase flow conditions were experimented and compared to the model. The results shows that the MARS code based on Henry-Fauske model can estimate the equilibrium reaching time of CO_2 critical flow experimental result quite well. However, the deviation of the model is bigger from the experiment around the equilibrium due to the different property database. Due to the characteristics of thermodynamic property variation of CO_2 near the critical point, the design and analysis methodologies of the CO_2 power cycle components which operate near the critical point should be precise. Since the MARS code especially focus on the steam tables, the correlations and models for CO_2 two-phase flow should be re-examined, which will be the next step of this research.

NOMENCLATURE

SFR	Sodium-cooled fast reactor
S-CO_2	Supercritical CO_2
SWR	Sodium-water reaction
IT	Individual turbine
TC	Turbine-compressor
RC	Recompression compressor
RT	Recompression turbine
MC	Main compressor
MT	Main turbine
IHX	Intermediate heat exchanger
SNL	Sandia National Laboratory
1D	One-dimentional
Eq.	Equation
ID	Internal diameter
MARS	Multi-dimensional Analysis of Reactor Safety
RELAP5	Reactor Excursion and Leak Analysis Program
COBRA-TF	Coolant Boiling in Rod Arrays – Two Fluid
USNRC	United States Nuclear Regulatory Commission
KAERI	Korea Atomic Energy Research Institute
EOS	Equation of state
I/O	Input/output
FORTRAN	Formula translation
CFD	Computational Fluid Dynamics

SUBSCRIPTS

C_d	Discharge coefficient
ρ	Density of fluid
r	Rotor radius
ω	Rotor angular frequency
L_r	Rotor cylinder length
Re	Reynolds number
t	Air gap length
υ	Kinematic viscosity
μ	Dynamic viscosity
L_{tooth}	Tooth length
L_{cavity}	Cavity length
N	Tooth number
A	Cross section
P	Pressure
$\dot{m}_{v,l}$	Mass flow rate of vapor and liquid
$u_{v.l}$	Velocity of vapor and liquid

DOI: 10.17185/duepublico/46092

F_w	Wall shear stress
G	Mass flux
x	Quality
$v_{v.l}$	Specific volume of vapor and liquid
n	Thermal equilibrium polytropic exponent
γ	Ratio of specific heats
M	Mach number
R	Gas constant
H	Height
T	Temperature
N	Thermal non-equilibrium factor
x_0	Stagnation quality
h	Enthalpy
s	Entropy

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