Studies of S-CO2 Power Plant Pipe Design for Small Modular Sodium-cooled Fast Reactor

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1. Introduction

Because of low cost and low carbon emission characteristics of a nuclear power plant, the roll of nuclear energy will be increased for the main energy source in the future. However, with current light water reactor (LWR) technology the sustainability of the nuclear energy is questionable due to the spent fuel issue and the limited uranium resources. To resolve these issues and consistently utilize the nuclear energy in an economic way, many countries have conducted some research works on the Sodium-cooled Fast Reactor (SFR) which can recycle the existing LWR's spent fuel. Furthermore, if SFR can be developed into the economical small modular reactor (SMR) for an export from Korea, the expected value can be greater.

However, current SFR design may face difficulty in public acceptance due to the potential hazard from sodium-water reaction (SWR) when the current conventional steam Rankine cycle is utilized as a power conversion system for a SFR. In order to eliminate SWR, the Supercritical CO_2 (S-CO_2) cycle has been proposed. Although there are many researches on S-CO_2 cycle concept and turbomachinery, very few research works considered pipe selection criteria for the S-CO_2 cycle. As one of the most important parts of the plant, this paper will discuss how to select a suitable pipe considering thermal expansion for the S-CO_2 power plant and perform a conceptual design of SFR type SMR.

2. S-CO_2 Power Plant Pipe Design for PG-SFR

2.1. S-CO_2 Brayton cycle layout and properties

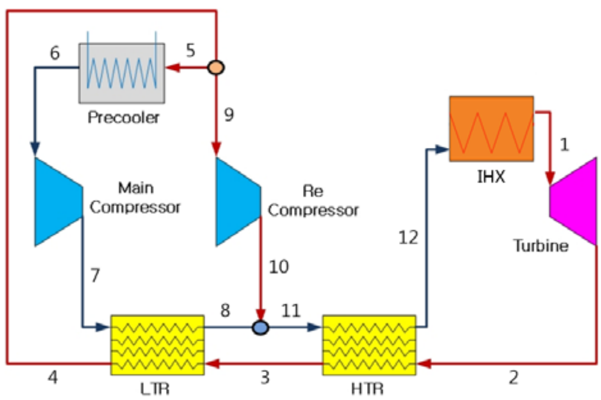

Fig. 1 S-CO_2 recompressing cycle layout

Table. 1 Cycle design variables and specification

Cycle design		
Layout	Recompressing cycle	
Compressor outlet pressure	20	MPa
Turbine inlet temp.	505	℃
Turbine efficiency	92	%
Main and Re-compressor efficiency	88/90	%
Recompressing fraction	36	%
Recuperator effectiveness	95	%
HTR hot side pressure drop	150	kPa
HTR cold side pressure drop	75	kPa
LTR hot side pressure drop	150	kPa
LTR cold side pressure drop	75	kPa
Precooler CO_2 pressure drop	75	kPa
IHX CO_2 pressure drop	75	kPa
CO_2 mass flow	912.75	kg/s
Net output	75.0	MW
Cycle thermal efficiency	43.55	%

The S-CO_2 cycle has lower compressor work than other gaseous state working fluid because density of CO_2 is higher when S-CO_2 is compressed around the critical point. As shown in Fig. 1, recompressing some portion of the flow without heat rejection to increase the thermodynamic efficiency of the cycle is known as the most effective layout.

The KAIST research team developed an in-house code to calculate the S-CO_2 recompressing cycle performance, and the fluid properties are obtained from the NIST database. The cycle design variables and specification of the S-CO_2 cycle system are given in Table. 1. With the in-house code developed by KAIST research team, the properties at each station are shown in Table. 2.

2.2. Determination of pipe diameter and thickness for S-CO_2 cycle

Typical considerations such as Energy costs, Corrosion, Erosion, Noise, Vibration, System requirement (pump inlet/outlet etc.), pressure loss, and thermal expansion should be considered at the same time when determining the pipe diameter. [1]

However determining the pipe diameter after reviewing all the above considerations requires a lot of effort and time. Therefore, to minimize these efforts, most of the engineering companies establish the criteria of proper flow velocity for design guideline shown in Fig. 2.

Table. 2 Properties at each station in 75MWe S-CO_2 recompressing cycle

Section Condition	ṁ (kg/s)	T (℃)	P (MPa)	ρ (kg/m3)	h (kJ/kg)
① Turbine Inlet	912.7	505.00	19.775	130.89	979.6
② HT Recuperator HS Inlet	912.7	396.68	7.875	62.44	863.7
③ LT Recuperator HS Inlet	912.7	164.00	7.725	103.46	599.6
④ LT Recuperator HS Outlet	912.7	65.19	7.575	167.06	473.0
⑤ Precooler Inlet	584.2	65.19	7.575	167.06	473.0
⑥ MC Inlet	584.2	31.25	7.5	594.19	306.5
⑦ LT Recuperator CS Inlet	584.2	61.28	20	715.60	327.7
⑧ LT Recuperator CS Outlet	584.2	151.24	19.925	323.40	525.6
⑨ RC Inlet	328.6	65.19	7.575	167.06	473.0
⑩ RC Outlet	328.6	153.42	19.925	319.46	529.1
⑪ HT Recuperator CS Inlet	912.7	152.03	19.925	321.97	526.8
⑫ IHX Inlet	912.7	351.21	19.85	170.24	791.0

Although there is an optimal flow velocity for water, a similar value is not determined for S-CO_2 cycle.

To define the pipe dimensions for the S-CO_2 cycle, following equation was first applied and tested. The equation is an empirical formula suggested by Ronald W. Capps. [3]

$$V = f_{pv} / \rho^{0.3} \tag{1}$$

PIPE VELOCITY FACTORS	
Motive Energy Source	$m(kg/m^3)^{0.3}/s$
Centrifugal pump, Blower	14
Compressor Pipe dia<6in. Pipe dia>6in.	24 29
Steam Boiler	63~68

V : optimal flow velocity [m/s]

f_{pv}: pipe velocity factor [$m(kg/m^3)^{0.3}/s$]

ρ : density of flow [kg/m^3]

In the case that the diameter of pipe is larger than 6 in, optimal velocity factor is 29.

To determinate the pipe diameter and thickness in accordance with the ASME standard, temperature and pressure should be considered. In addition, as the selection of pipe material affects the minimum thickness and the cost of a pipe, the overall economy of the pipe material selection has to be studied further.

$$t_m = \frac{PD_o}{2(SE + Py)} + A \tag{2}$$

Where t_m : minimum required wall thickness [m], P: internal design pressure [Pa], D_o: outside diameter of pipe [m], S: maximum allowable stress [Pa], E: weld joint efficiency, Y: coefficient, A: additional thickness [m]

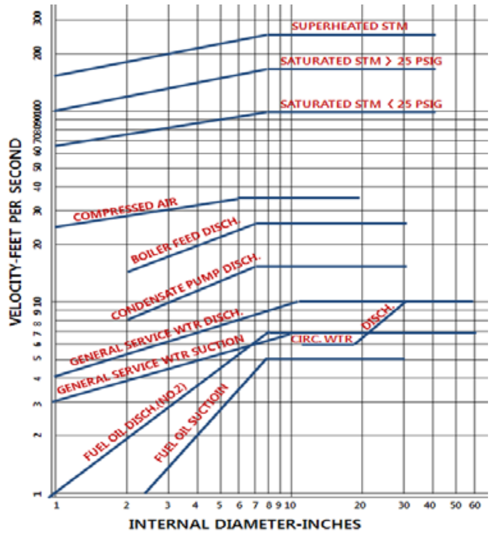

Fig. 2 The optimal flow velocity of various piping systems [2]

The optimal diameter and thickness in accordance with the ASME standard were calculated for the 75MWe S-CO_2 power conversion system and are shown in Table. 3. All the additional thicknesses of pipes are 2.5mm for the safety margin. Also the minor pressure loss of all the elbows and confluence loss of mixing tee are considered [4]. The used materials are high nickel alloys and alloy steels and all the figures of S, E, y are found in the ASME B31.1 [5].

To minimize the pressure drop and footprint, optimal arrangement of components and pipes is being found. The length of the highest pressure drop sections(①, ②, ③ and ⑫) are reduced as much as possible. The total pressure drop compared to the overall system pressure is 1.22%. And after considering the pressure drop by pipe design, cycle thermal efficiency drops from 43.55% to 43.07%.

2.3. Design to compensate for thermal expansion

To compensate the thermal stress by thermal expansion, pipes need some expansion joints.

Table. 3 The optimal diameter and thickness of 75MWe S-CO_2 cycle in accordance with the ASME standard

S.C.	Nominal Pipe Size	External Diameter(m)	Internal Diameter(m)	Schedule No.	Thickness (mm)	Pressure drop (kPa)
①	24	0.610	0.553	60	28.58	39.27
②	28	0.711	0.679	30	15.88	25.67
③	28	0.711	0.682	20	14.27	48.47
④	28	0.711	0.676	30	17.48	11.91
⑤	28	0.711	0.676	30	17.48	5.61
⑥	24	0.610	0.578	30	15.88	0.85
⑦	22	0.559	0.502	80	28.58	3.25
⑧	24	0.610	0.556	60	26.97	17.52
⑨	28	0.711	0.676	30	17.48	2.70
⑩	24	0.610	0.556	60	26.97	2.46
⑪	24	0.610	0.556	60	26.97	4.82
⑫	28	0.610	0.556	60	26.97	82.44
Total pressure drop (kPa)						244.97

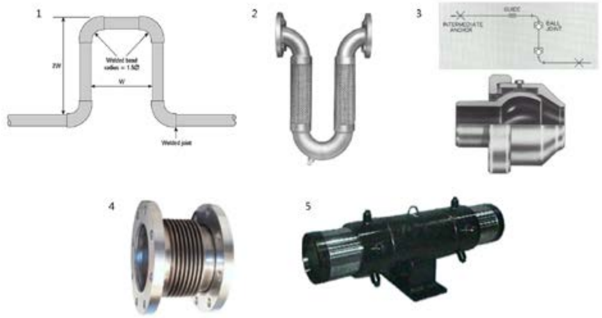

Fig. 3 Hard U-shape loop, flexible loop, bend, bellows and sliding [6]

The typical type of expansion joints are hard U-shape loop, flexible loop, bend, bellows and sliding shown in the Fig. 3.

The flexible loop has many advantages including very compact, no maintenance, minimal guiding requirements, lowest anchor loads, almost no structural considerations and large movement. However it isn't suitable to recompressing S-CO_2 cycle as it is very vulnerable to high pressure. And bellows have low pressure drop and compactness, but it must be replaced if damaged. Moreover, hard loop can be made without any expensive parts, but it needs lots of space.

On the other hand nonlinear expansion devices such as ball joints can accommodate movements in multiple directions. In addition, they have lower anchor loads than those associated with either bellows or slip type expansion joints. Due to ball joints construction, the internal pressure tends to aid in sealing and ball joints are therefore less likely to develop leaks during the service. Most of all, a ball joint is compact and available in sizes ranging from 3/4" through 30" NPS, which means that this can be applied to the system.

The final pipe design of S-CO_2 recompressing cycle applying ball joints is shown in the Fig. 4. Total volume is approximately 9.76m * 7.16m * 3.95m.

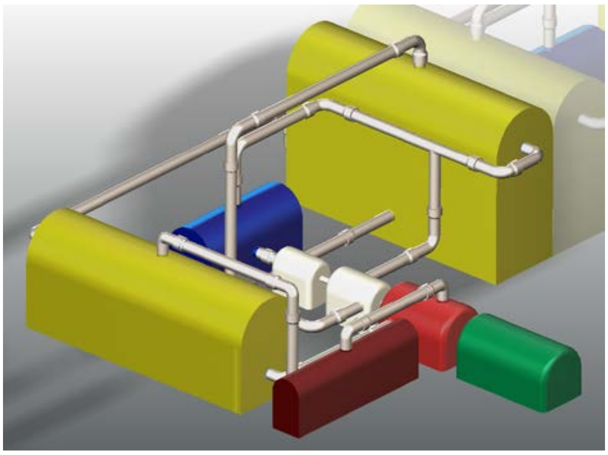

Fig. 4 Conceptual pipe design of S-CO_2 recompressing cycle

Additional study for a larger system such as 300MW class system in MIT report will be conducted in the future study. From the preliminary estimation when the S-CO_2 system becomes large, the pipe diameter may exceed the current ASME standard. This means that more innovative approach will be needed for the S-CO_2 pipe design.

To economically design the pipe of S-CO_2 recompressing cycle, optimal flow velocity for S-CO_2 that can be obtained through the process engineering. Although the Ronald W. Capps equation offers an optimal flow, this equation is optimized for a water or steam system. As the S-CO_2 cycle is not commercialized yet and it is being developed actively at present, procedure for the S-CO_2 pipe design is not fully established. Thus, further study and accumulation of operating experiences are salient for the further development and realization of the S-CO_2 cycle.

3. Conclusions

The S-CO_2 cycle can improve the safety of SFR as preventing the SWR by changing the working fluid. Additionally, not only the relatively high efficiency with 450~750℃ turbine inlet temperature, but also the physically compact footprint are advantages of the S-CO_2 cycle. However the pipe design is more complicated than existing power plant because it has high pressure and temperature conditions and needs high mass flow rate.

By designing the piping system for a small modular - SFR, the compactness and simplicity of the S-CO_2 cycle are re-confirmed. Moreover, in this paper, realistic and safe pipe design was conducted by considering thermal expansion in the high pressure and temperature conditions. Although total pipe pressure drop is somewhat high, the cycle thermal efficiency is still higher than the existing steam Rankine cycle.

ACKNOWLEDGEMENT

This research was supported by the National Research Foundation of Korea (NRF) and Sodium cooled Fast Reactor development Agency (SFRA).

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