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Preliminary Validation and Verification of TURBO_DESIGN for S-CO2 Axial Compressor

Jekyoung Lee [a*], Jeong Ik Lee [a], Yoonhan Ahn [a], Seong Gu Kim [a],
Ho Joon Yoon [b], Yacine Addad [b]
[a]*Dept. Nuclear & Quantum Eng., KAIST, 373-1, Guseong-dong, Yuseong-gu, Daejeon, 305-701, Republic of Korea*
[b]*Dept. Nuclear Enginerring, KUSTAR, PO.box:127788, Abu Dhabi, United Arab Emirates*
**Corresponding author : leejaeky85@kaist.ac.kr*

1. Introduction

To use the advantages of Supercritical CO_2(S-CO_2) Brayton cycle for nuclear power plant, KAIST-Khalifa University joint research team has been focusing on S-CO_2 turbomachinery development. TURBO_DESIGN code is one of the products of our researches to design a turbomachinery. The major feature of TURBO_DESIGN is that the formulation is based on the real gas and none of the ideal gas assumption was applied to the code. Thus, TURBO_DESIGN has high flexibility regarding the type of gases. In this paper, preliminary code validation and verification of TURBO_DESIGN will be discussed for axial type compressor design.

2. Axial compressor design and comparison

Yong Wang has suggested an S-CO2 axial compressor design for the main compressor of S-CO_2 Brayton cycle [1]. He set operating conditions for 300MWe nuclear power plant and major operating parameters are in Table. 1.

Table 1. Operating Conditions of the Main Compressor

Mass Flow Rate $(\mathbf{kg/s})$	1915
Shaft Rotational Speed $(\mathbf{RPM})$	3600
Total to total Pressure Ratio	2.6
Inlet Total Temperature $(^{\mathbf{o}}\mathbf{C})$	32
Inlet Total Pressure $(\mathbf{MPa})$	7.69

The critical point of carbon dioxide is at 30.98°C , 7.38MPa. Total cycle efficiency is higher when the operating conditions approaches closer to the critical point. Because high density fluid with low compressibility can achieve lower compressing work, which results in higher total cycle efficiency. But we cannot set 30.98°C as the compressor inlet temperature. Fluid flows with certain velocity so that static condition can be below the critical point when the inlet stagnant temperature is 30.98°C. The velocity is highly related with geometry. At certain mass flow rate, turbomachinery inlet size is calculated with inlet velocity. So, operating inlet temperature should be above 30.98°C to ensure reasonable inlet velocity to reduce the size. In this manner, 32°C of inlet total temperature is reasonably selected. In Yong Wang's paper, unfortunately, there were some difficulties on his modified version of existing code when the compressor is designed close to the critical point, he had to re-set operating temperature to 42°C. The difficulties Yong Wang suffered are caused by property variation of S-CO2 especially specific heat ratio near the critical point. Since S-CO2 doesn't show linear property variation with pressure and temperature, commercially available turbomachinery design codes which are based on ideal assumption cannot be used for S-CO2 turbomachinery design. This is the motivation to develop TURBO_DESIGN. TURBO_DESIGN is not based on ideal gas assumption but based on energy conservation and basic principles without any simplification.

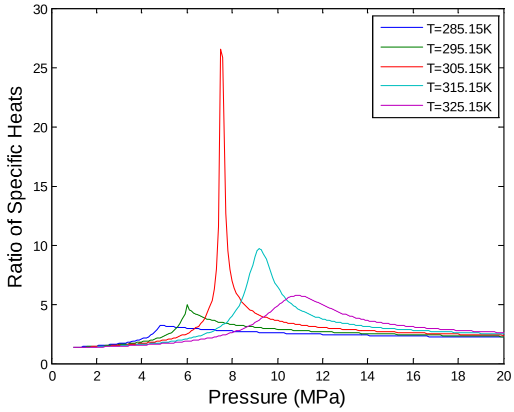

Fig 1. Ratio of specific heats variation near the critical point

Yong Wang generated preliminary design with the operating conditions in Table. 1 and its geometry is shown in Fig. 2.

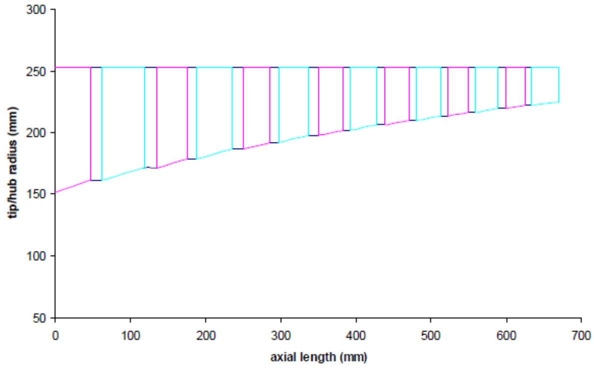

Fig 2. Main compressor geometry with inlet temperature of $\mathbf{32^oC}$ in Yong Wang's report

Yong Wang successfully modified CSPAN for S-CO_2 so called CSPAN_MOD. And the main compressor was successfully designed using CSPAN_MOD with the operating conditions in Table. 1.

Our team also designed main compressor with the operating conditions based on Table-1 and its geometry is shown in Fig. 3.

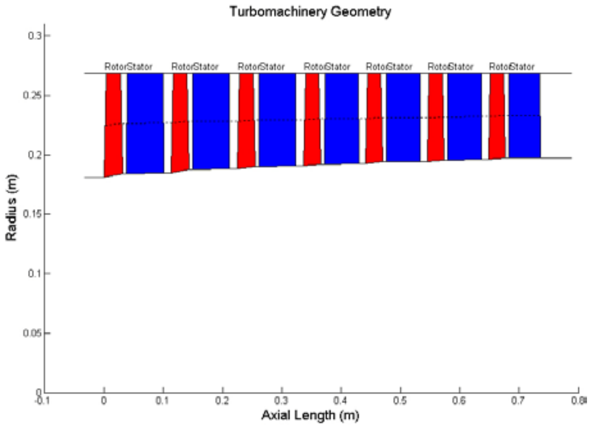

Fig 3. Main compressor geometry provided by TURBO_DESIGN

Table 2. Size comparison between two results from TURBO_DESIGN and CSPAN_MOD

		TURBO_ DESIGN	CSPAN_ MOD	Ratio (T/C)
Inlet	Tip	0.268 m	0.253 m	1.06
	Hub	0.181 m	0.152 m	1.19
Outlet	Tip	0.268 m	0.253 m	1.06
	Hub	0.197 m	0.224 m	0.88
Overall length		0.737 m	~0.7 m	1.05

TURBO_DESIGN result and CSPAN_MOD result are compared in Table 2. TUBO_DESIGN generally designs larger geometry than CSPAN_MOD (up to 20%) . However, the annular flow area is the same for results from both codes. Fig. 4 and Fig. 5 are off design performance maps calculated by TURBO_DESIGN. The efficiency of main compressor provided by CSPAN_MOD is 91.08% for total to static efficiency and TURBO_DESIGN predicted 92.24% for total to static efficiency. The difference in geometry mainly depends on velocity and density. Since both code predicted almost equal efficiency (which means entropy generation are quite similar in each stage), S-CO_2 density in both codes are more or less the same in each stage. This means that the geometry difference is mainly caused by different velocity. From simple calculation with above geometry and property database, velocity at the outlet in CSPAN_MOD case is about 3 times higher than TURBO_DESIGN case. Our team will focus on this issue and improve our code TURBO_DESIGN by comparing our result to Yong Wang's 42°C case. Furthermore, comparison with helium turbomachinery design will be performed to verify the flexibility of TURBO_DESIGN code in the ideal gas region as well.

3. Summary

Preliminary code validation and verification (V&V) of TURBO_DESIGN is being carried out, and as a part of this effort previous design result for axial compressor is compared in this paper. TURBO_DESIGN provides similar geometry and on design performance compared to the reference results. However, TURBO_DESIGN can predict the off-design for axial compressor near the critical point which the reference admitted it was not possible in its case. Further V&V will be carried out for various cases in the future.

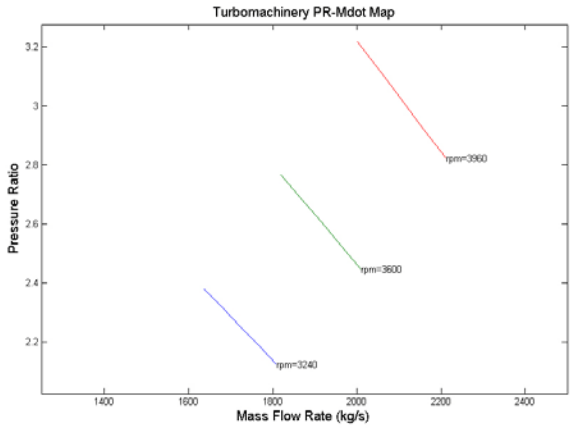

Fig 4. Pressure ratio predicted by TURBO_DESIGN

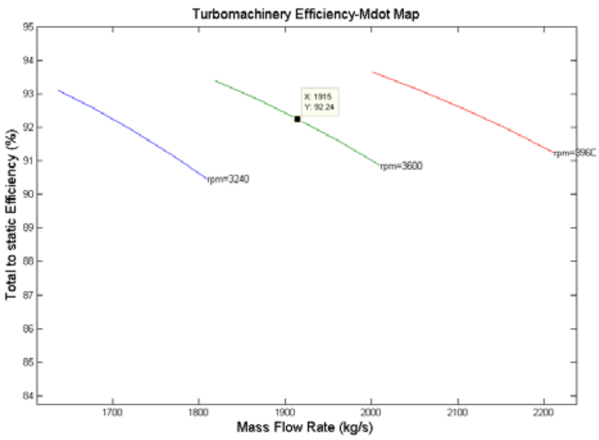

Fig 5. Total to static efficiency predicted by TURBO_DESIGN

ACKNOWLEDGMENTS

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