Numerical investigation on property effects in a low speed supercritical carbon dioxide centrifugal compressor

J L Gou¹,², K L Zhang¹,²,³, C Ma¹,², C H Huang⁴,², K Chen¹,² and B M Li¹,²

¹Science and Technology on Thermal Energy and Power Laboratory, Wuhan 430205, Hubei Province, P.R. China
²Wuhan Second Ship Design and Research Institute, Wuhan 430205, Hubei Province, P.R. China

E-mail: luckyzkl@163.com

Abstract. Compressor is one of the key components in the closed Brayton cycle operating with supercritical carbon dioxide, but the design method is far from mature. The physical property effects on the 1D isentropic compression process and a low speed centrifugal compressor are numerically investigated in the present work. The results show that with similar inlet conditions, the physical property of the supercritical carbon dioxide results in lower outlet Mach number than the ideal air. Furthermore, the static pressure coefficient distribution is surprisingly almost the same between these two working fluids at low Mach number situations for both the compression process and the low speed centrifugal compressor. It indicates that lots of air compressor design experience can be well applied to the low speed centrifugal supercritical carbon dioxide compressor design.

1. Introduction
Closed Brayton cycle operating with supercritical carbon dioxide (S-CO₂) is widely recognized and studied in these years for its compact layout and high efficiency with modest turbine inlet temperature [1,2]. It was firstly proposed by Feher [3] and then analyzed by Angelino [4]; however, it was not realizable at that time for the manufacturing level. From 2000s, the closed Brayton cycle operating with S-CO₂ attracted attention again, and turned into an attractive option for geothermal, nuclear and solar energy conversion.

Compressor is one of the key components in this system. Many authors investigate the performance of S-CO₂ compressors [5-9]; however, the overall performance and the numerical accuracy are mainly focused, and the property effects of S-CO₂ are not fully clarified. Comparing to the ideal gas, the physical property of S-CO₂ is quite different. This will lead to different compression process and result in different feature structures with similar inlet condition. Baltadjiev et al [10] proposed that the real gas property would greatly affect the compressor performance, and the application of the ideal gas model in the numerical simulation lead to large prediction errors. On the other time, the physical property of the carbon dioxide greatly changes near the critical point and it may also lead to different flow field structures compared to the ideal gas. With these variations, the well-established design method of air compressor may not be fully suitable for the S-CO₂ compressor design, and needs further improvement. Mechanism investigation of the influence of the special physical property in S-CO₂ compressor is necessary. This is beneficial to reveal that whether the design model of the air compressor can be directly used for S-CO₂ design, or needs what kind of correction. These results are...
helpful to fully take advantages of the abundant researches of air compressors [11-15].

In the present work, the physical property effects on the isentropic compression process are numerically investigated. The performance of a low speed centrifugal S-CO₂ compressor is further numerically investigated. The similar air compressors with similar geometry and inlet conditions are also designed and simulated for comparison. The detailed flow field is analyzed to investigate the effects of the real gas property.

This paper is organized as follows: the simulation object and the numerical method are firstly described. The influence of the real gas property of S-CO₂ on the 1D isentropic compression process is then discussed, and the flow field between two similar low speed centrifugal compressors (S-CO₂ and the ideal air) is compared. Finally, the conclusion is given in the last section.

2. Simulation object and numerical method
A low speed S-CO₂ centrifugal compressor stage is studied in the present work. This S-CO₂ compressor is consisted of 12 impellers with half of split blade. 18 diffuser vanes are set up downstream. More parameters of this S-CO₂ centrifugal compressor can be found in [16].

A similar air compressor is further designed with the same important dimensionless parameters, and simulated for better comparison. The peak efficiency point of S-CO₂ compressor with mass flow rate of 4.758 kg/s is set as the similar operation point. Comparing to the S-CO₂ compressor, the air compressor has the same geometry with different scale factors, the same Reynolds number, the same inlet flow angle and the same inlet Mach number at 50% span. With much larger density of S-CO₂, the same Reynolds number results in much larger length scale of the air compressor and the original geometry is magnified about 106 times in the present work.

The present work applied the commercial solver: NUMECA FINE/Turbo. The Reynolds-averaged Navier-Stokes (RANS) equations with Shear Stress Transport (SST) turbulence model are solved. Mixing-plane model is adopted for this steady simulation. The real gas property of the supercritical carbon dioxide is handled using the property tabulation which is generated using the program TabGen. This tabulation contains 301x301 resolution, and more points are clustered near the saturation line.

3. 1D isentropic compression process
The 1D isentropic compression process is one of the key components of the complex fluid field in the compressor passage. The physical property effects on the 1D isentropic compression process is firstly discussed in the present work, and a simple compression case is simulated to reveal the difference between the ideal air and S-CO₂.

![Figure 1. Sketch and the simulation mesh of the 1D isentropic compression case. (a) Sketch of the 1D isentropic compression process and (b) Simulation mesh.](image-url)
Figure 1(a) shows the sketch of this 1D isentropic compression process. The geometry parameters are set as \( L_3=L_5=L, L_4=3L, L_1=0.6L \) and \( L_2=L \). As given in Figure 1(a), the fluid flows through the path with increasing cross area. The Mach number gradually decreases and the static pressure increases in this process. To consider the effects of the physical property and strip other effects, several assumptions are made in this part:

- The same inlet Mach number: \( (Ma_1)_{\text{S-CO}_2} = (Ma_1)_{\text{ideal air}} \).
- The subsonic conditions: \( Ma_1<1 \).
- The same flow area expansion ratio: \( (A_2/A_1)_{\text{S-CO}_2} = (A_2/A_1)_{\text{ideal air}} \).

The 2D Euler equations are solved for this isentropic case, and the simulation mesh is given in Figure 1(b) with the grid resolution of 153×41.

A low Mach number situation \( (Ma_1=0.28) \) is compared in the present work. Table 1 shows the inlet and outlet parameters of these cases. Three S-CO\(_2\) cases are compared with different inlet property state: the inlet of LMa-C case is close to the critical point, the inlet of LMa-A case is away from the critical point and the inlet of LMa-M case is in the middle condition.

As given in Table 1, with the same inlet Mach number and flow area rise ratio, the outlet Mach number of S-CO\(_2\) is lower than the ideal air case. Even for the S-CO\(_2\) cases with different property state, the outlet Mach number is a bit different: LMa-M case which has moderate state results in the lowest Mach number, and the cases which are close to and far away from the critical point both lead to a bit larger Mach number case, as given in Figure 2(a).

On the other hand, with similar Mach number reduction of these three S-CO\(_2\) cases, the static pressure changes quite different. Figure 2(b) shows the distribution of the static pressure for the low Mach number 1D isentropic expansion process. For LMa-C case which is close to the critical point, the static pressure rise is almost the same with the ideal air case, even with much lower outlet Mach number. For the other two cases (LMa-M and LMa-A), the static pressure rise ratio is consequently much larger than LMa-C case, and LMa-A case has the largest value.

| Case Description       | \( p_1 \) (MPa) | \( p_2 \) (MPa) | \( T_1 \) (K) | \( Ma_1 \) | \( p_2 \) (MPa) | \( T_2 \) (K) | \( Ma_2 \) |
|------------------------|----------------|----------------|--------------|-----------|----------------|--------------|-----------|
| Ideal air              | 0.1013         | 0.09594        | 305.2        | 0.280     | 0.09946        | 308.4        | 0.163     |
| LMa-C (S-CO\(_2\))    | 8.0            | 7.570          | 305.4        | 0.280     | 7.850          | 307.1        | 0.150     |
| LMa-M (S-CO\(_2\))    | 9.0            | 7.931          | 306.2        | 0.280     | 8.625          | 308.7        | 0.148     |
| LMa-A (S-CO\(_2\))    | 13.0           | 10.780         | 314.7        | 0.280     | 12.230         | 318.2        | 0.151     |
The commonly used parameter pressure coefficient \( C_p = \frac{p - p_1}{p_{t1} - p_1} \) is further compared, where \( p_1 \) is the inlet static pressure and \( p_{t1} \) is the inlet total pressure. Figure 2(c) shows that even with quite different pressure growth distribution, the pressure coefficient distribution is surprisingly almost the same among these cases, which is quite favorable for the extension of the air compressor research experience. The change of pressure coefficient is related to two factors: the growth rate of the static pressure and the initial difference between the inlet static pressure and inlet total pressure. Similar pressure coefficient distribution shown in Figure 2(c) indicates that these two factors are consistent at low Mach number situation. Larger static pressure growth rate of LMa-M case and LMa-A case appears with larger initial difference between the inlet static pressure and inlet total pressure.

4. **Low speed centrifugal compressor stage**

The flow field in a low speed centrifugal compressor stage [16] with S-CO\(_2\) and ideal air as working fluids respectively is discussed in this section. Figure 3 shows the computational mesh for this low speed centrifugal compressor case. For these two compressors whose working fluids are respectively the ideal air and S-CO\(_2\), the same simulation grid is adopted with different scale factors. The rotating region of one main impeller blade and one split impeller blade contains 1.39 M grid points. Only one vane passage is simulated in the present work and the grid number is 1.08 M. The wall \( y^+ \) is below 1.

The absolute inlet Mach number of this low speed centrifugal compressor is about 0.15 at mid-span, and the relative Mach number is ranging from 0.12 to 0.32 at inlet. This inlet Mach number is low and compatible with the above discussed 1D isentropic compression process case. It is believed that the above variation trend of the main parameters will be suitable for this case, while compression process is one of the key processes in the compressor passage.

Figure 4 shows the relative Mach number distribution at 50% span in the impeller passage. With similar inlet condition, the basic distribution form of the relative Mach number between S-CO\(_2\) case and the air case. Further looking at the detailed values, S-CO\(_2\) compressor generates a bit lower relative Mach number value in the impeller passage than the ideal air. This phenomenon is consistent with the 1D isentropic compression process.
Figure 3. Computational mesh for the low speed centrifugal compressor case.

Figure 4. Relative Mach number distribution at 50% span in the impeller passage. (a) S-CO$_2$ case and (b) air case.

Further looking at the commonly used key parameter $C_p$ (static pressure coefficient) in these two compressors. The static pressure coefficient is defined as follows:

$$C_p = \frac{P - p_1}{p_{t1} - p_1}$$

where $p_1$ is the inlet static pressure at mid-span and $p_{t1}$ is the inlet total pressure at mid-span. Figure 5 shows the static pressure coefficient distribution around the impeller blades at 50% span for S-CO$_2$ and the air compressors. As illustrated in Figure 5, the static pressure coefficient is almost the same...
between these two compressors, which is compatible with the above results in the 1D isentropic compression process. This indicates that the blade load is also same between the ideal air and S-CO₂ cases, and the abundant research results of the air compressor related to the blade load in the literature can be directly used for the low speed centrifugal compressor.

Figure 5. Static pressure coefficient distribution around the impeller blades at 50% span.

5. Conclusions
The physical property effects on the 1D isentropic compression process with S-CO₂ as working fluid are numerically investigated in this work. The variation discrepancy of the Mach number and the pressure rise between the ideal air and S-CO₂ is fully discussed in a wide range of property state at low Mach number situations which are commonly encountered in the Brayton cycle, and several trends are present. With the same inlet Mach number and flow area rise ratio, the present results show that S-CO₂ cases always generate lower outlet Mach number than the ideal air case. The variation trend of the static pressure rise ratio is related to the specific property state of S-CO₂. What’s more, the distribution of the static pressure coefficient is surprisingly almost the same between S-CO₂ and the ideal air case at low Mach number.

The flow field between two similar low speed centrifugal compressors (S-CO₂ and the ideal air) is further numerically compared to reveal the physical property effects on the compressor performance. The results show that the analysis about the 1D isentropic compression process is suitable for the low speed centrifugal compressor. With the same inlet Mach number, flow coefficient, Reynolds number and the same geometry, S-CO₂ compressor generates lower outlet Mach number. The static pressure coefficient distribution is almost the same in this low speed centrifugal compressor. This indicates that lots of air compressor design experience about the blade load can be well applied to S-CO₂ compressor design.

The real gas property effects of S-CO₂ on the compressor performance will be further studied in the future work. These studies will be beneficial to understand the flow field in the S-CO₂ compressor and improve the design method.

Acknowledgments
The author wishes to thank the long term support from the National Natural Science Foundation of
China (Grant NO. 51806154) and Hubei Province Natural Science Foundation of China (NO. 2018CFB317, NO. 2017CFB325, NO. 2016CFA019).

References

[1] Ahn Y, Bae S J et al 2015 Review of supercritical CO\(_2\) power cycle technology and current status of research and development Nucl Eng Technol \textbf{47} 647-61

[2] Sarkar J 2015 Review and future trends of supercritical CO\(_2\) Rankine cycle for low-grade heat conversion Renew Sust Energ Rev \textbf{48} 434-51

[3] Feher E G 1968 The supercritical thermodynamic power cycle Energ Convers \textbf{8} 85-90

[4] Angelino G 1968 Carbon dioxide condensation cycles for power production Eng Power \textbf{90} 287-95

[5] Wright S A, Fuller R et al 2005 Operational results of a closed Brayton cycle test-loop Proceedings of the Space Technology and Applications International Forum (New York)

[6] Wright S A, Radel R F et al 2010 Operation and analysis of a supercritical CO\(_2\) Brayton cycle Sandia Report 2010-0171

[7] Noall J S and Pasch J J 2014 Achievable efficiency and stability of supercritical CO\(_2\) compression systems Proceedings of the S-CO\(_2\) Power Cycle Symposium (Pittsburgh, USA)

[8] Utamura M, Fukuda T and Aritomi M 2012 Aerodynamic characteristics of a centrifugal compressor working in supercritical carbon dioxide Energ Procedia \textbf{14} 1149-55

[9] Pecnik R, Rindldi E and Colonna P 2012 Computational fluid dynamics of a radial compressor operating with supercritical CO\(_2\) J of Eng Gas Turb Power \textbf{134} 122301

[10] Baltadjiev N D, Lettieri C and Spakovszky Z S 2015 An investigation of real gas effects in supercritical CO\(_2\) centrifugal compressors J Turbomach \textbf{137} 091003

[11] Vo H D, Tan C S and Greitzer E M 2008 Criteria for spike initiated rotating stall J Turbomach \textbf{130} 1-9

[12] März J, Hah C and Neise W 2002 An experimental and numerical investigation into the mechanisms of rotating instability J Turbomach \textbf{124} 367-74

[13] Wang H, Lin D et al 2017 Entropy analysis of the interaction between the corner separation and wakes in a compressor cascade Entropy \textbf{19} pp 324

[14] Gou J, Yuan X and Su X 2017 Adaptive mesh refinement method based investigation of the interaction between shock wave, boundary layer, and tip vortex in a transonic compressor P I Mech Eng G J Aer \textbf{232} 694-715

[15] Su X and Yuan X 2017 Improved compressor corner separation prediction using the quadratic constitutive relation P I Mech Eng A J Pow \textbf{231} 618-30

[16] Ma C, Wang W et al 2018 Analysis of unsteady flow in a supercritical carbon dioxide radial compressor stage Proceedings of the 26\(^{th}\) International Conference on Nuclear Engineering ICONE26-82183 (London, UK)