Numerical simulation of supercritical carbon dioxide turbine thermal stress based on multiphysics coupling method

Jun Wu1, Yubing Li2, Zhenxing Zhao1, Yuansheng Lin1,3, Fan Bai1, Wei Wang1

1Science and Technology on Thermal Energy and Power Laboratory, Wuhan Second Ship Design and Research Institute, Wuhan, Hubei Province, China
2 Wuhan electric power technical college, Wuhan, Hubei Province, China
3 School of Energy and Power Engineering, Xi’an Jiaotong University, Xi’an, Shaanxi Province, China
wu_jun1986@126.com

Abstract. Supercritical carbon dioxide Brayton cycle has significant advantages such as high efficiency, low cost and small size. Turbine is a key turbomachinery in the supercritical carbon dioxide Brayton cycle system. It works in the harsh environment of high pressure, high speed and high temperature gradient. The huge thermal stress is an important cause of the turbine structure failure. In this paper, the thermal stress analysis is conducted for a supercritical carbon dioxide turbine with an output of 200 kW and a rotational speed of 40,000 rpm. In order to consider the coupling relationship between the temperature field and the structural stress field, the flow-thermal-solid coupling analysis method is applied to calculate the thermal stress of the supercritical carbon dioxide turbine impeller. The results of thermal deformation and thermal stress analysis of the turbine impeller are obtained by the numerical simulation. The high stress region not only appears in the geometrical structure, but also in the region with large temperature gradient. The analysis results show that the maximum equivalent stress appears at the fillet between the turbine impeller and the seal with a value of 324 MPa. This value is less than the yield limit of the material at 500 °C. The turbine impeller structure meets the strength design requirements.

1. Introduction
Carbon dioxide has good thermal stability and physical properties, and it also has the advantages of being non-toxic, abundant in reserves, natural in existence, and inexpensive. The power system with supercritical carbon dioxide as the working fluid can obtain satisfactory cycle thermal efficiency without requiring high cycle temperature. Although the design process of supercritical working fluid rotating mechanical components is the same as that of conventional working fluids, its one-dimensional model needs to fully consider the thermal properties of real gases and depends on some related alternative models. SANDIA National Laboratory uses the model to design and manufacture small-size S-CO₂ compressors and turbine prototypes and conducts preliminary experiments and proof of concept. However, it is still an area to be further explored, and there are still many unknown laws.

At present, the United States, the United Kingdom, Germany, Japan, South Korea, Spain and other countries have carried out research on S-CO₂ power generation technology, and some of them have already carried out prototype manufacturing and testing. Domestic and foreign scholars have carried out a lot of research on the S-CO₂ Brayton cycle system and the key turbomachinery. Shintaro
Ishiyama[1], CJ Duan[2], XR Wang[3], and JF Wang[4] focus on the overall performance of the thermal cycle system. The system cycle efficiency is analyzed. The efficiency of the turbine is often assumed to be a constant value during the analysis process. Detailed design and performance analysis studies have not been performed on the S-CO₂ turbine. The one-dimensional preliminary design program TOPGEN is applied to perform a one-dimensional preliminary design space analysis on S-CO₂ turbines of 100kW and 200kW power levels by J H Qi et al [5]. The aerodynamic thermodynamic design, structural geometry design and overall performance prediction analysis with two S-CO₂ turbines of different power levels is investigated by H Z Zhang et al [6]. In summary, a large number of system performance simulations for S-CO₂ Brayton cycle system and simple aerodynamic analysis for S-CO₂ turbomachinery are carried out by researchers. The study on flow-thermal-solid coupling analysis of S-CO₂ turbine impeller is limited.

In this paper, thermal stress analysis is carried out for an S-CO₂ turbine with an output of 200 kW and a rotational speed of 40,000 rpm. In order to consider the coupling relationship between the temperature field of the S-CO₂ working fluid and the structural stress field, the flow-thermal-solid coupling analysis method is applied to analyze the thermal stress of the S-CO₂ turbine impeller.

2. Flow-thermal-solid coupling analysis method of S-CO₂ turbomachinery
Flow-thermal-solid coupling analysis of the S-CO₂ turbomachinery consists of one-dimensional design, quasi-three-dimensional design, CFD flow field analysis, load parameter transfer, and FEA structure analysis. The detailed analysis steps are as follows:

1) The inlet and outlet thermal parameters of the S-CO₂ rotating machinery obtained by the analysis of the system cycle characteristics are used as input conditions. Basic geometric parameters and aerodynamic parameters on the inlet and outlet of the S-CO₂ turbomachinery meridian plane are obtained by the one-dimensional design.

2) The streamline curvature method is applied to analyze the variation of flow parameters along the streamline. The initial geometric model of the blade is obtained by leaf superposition. Furthermore, the quasi-three-dimensional design of the S-CO₂ turbomachinery flow passage is completed.

3) Three-dimensional refined CFD flow field simulation is carried out on the flow passage of the S-CO₂ turbomachinery. The three-dimensional spatial distribution of the detailed flow parameters of the S-CO₂ turbomachinery is obtained.

4) The S-CO₂ turbomachinery structure analysis model is constructed. The temperature load and pressure load on the coupling interface obtained by the previous CFD analysis are transmitted to the structural analysis model.

5) The temperature field distribution of the S-CO₂ turbomachinery structure is obtained by finite element numerical analysis. The temperature load is applied into the structural model as a body load. The flow-thermal-solid coupling thermal stress calculation results of the S-CO₂ turbomachinery structure are obtained by static analysis.
3. S-CO₂ turbine aerodynamic scheme design and flow field analysis

3.1. One-dimensional design
Initial design parameters of the inlet and outlet of the S-CO₂ turbine are shown in Table 1, including the total inlet temperature $T_0$, the total inlet pressure $P_0$, the outlet back pressure $P_1$, the mass flow rate $G$, and the impeller speed $n$. One-dimensional design process is based on the one-dimensional compressible flow control equation of turbomachinery. Physical parameters of the S-CO₂ working fluid at various state points are calculated by the REFPROP software. The shape of the meridian plane and the key geometric parameters and flow parameters of the inlet and outlet are obtained, such as the inlet radius of the blade top and root, the blade tip and the root exit radius and so on.

![Fig. 2 Schematic diagram of the S-CO₂ turbine meridian plane parameters](image)

| Number | Variable | Value          | Number | Variable                  | Value          |
|--------|----------|----------------|--------|---------------------------|----------------|
| 1      | $T_0$    | 500 °C         | 1      | Impeller inlet radius $R_{i1}$ | 63.87 mm       |
| 2      | $P_0$    | 13.85 MPa      | 2      | Impeller outlet hub radius $R_{o2h}$ | 14.63 mm       |
| 3      | $P_1$    | 7.84 MPa       | 3      | Impeller outlet shroud radius $R_{o2s}$ | 22.26 mm       |
| 4      | $G$      | 3.61 kg/s      | 4      | Impeller inlet blade height $B$ | 1.81 mm        |
| 5      | $n$      | 40000 r/min    | 5      | Impeller main blade number $NB_{im}$ | 12             |
|        |          |                | 6      | Impeller Splitter blade number $NB_{is}$ | 12             |
|        |          |                | 7      | Nozzle outer diameter $R_{no}$ | 94 mm          |
|        |          |                | 8      | Nozzle inner diameter $R_{ni}$ | 74 mm          |
|        |          |                | 9      | Nozzle blade number | 13             |
|        |          |                | 10     | Nozzle blade height | 1.81 mm        |

3.2. Quasi-three-dimensional design
Meridian plane parameters are utilized as calculation input conditions. The through-flow solver is applied to analyze the impeller flow design. Flow velocity distributions for different spanwise cross-sections of the turbine impeller are obtained. The flow velocity distribution is applied as the design target to select and design the blade profile of each section. The three-dimensional geometric model of the blade is obtained by superimposing the blade profile. Furthermore, the key parameters of the turbine are determined by structural design.
According to the key geometric parameters of the turbine impeller and nozzle flow section, the impeller meridian plane shape and the B2B section nozzle profile, the three-dimensional geometric design of the turbine impeller and nozzle is carried out by the three-dimensional modeling software. Based on the thermal parameters and geometric parameters, the turbine volute is further designed by the 3D modeling software.

3.3. CFD flow field analysis and optimization

According to the design result of the quasi-three-dimensional scheme, the single turbine flow field performance CFD simulation process includes meshing, boundary condition setting, and nonlinear iterative calculation and result analysis. According to each CFD simulation result, the impeller blade parameters are optimized and iteratively analyzed until the convergence optimal performance scheme is obtained. The specific analysis and optimization steps are shown in Figure 5.

The full-circumferential fluid domain model is applied to conduct the S-CO$_2$ turbine CFD simulation, including twelve impeller flow passages, thirteen nozzle flow passage, and one volute flow passage. The grid model of 8.5 million units is finally selected after grid-independence verification. The Shear Stress Transport (SST) turbulence model is chosen for the calculation. The total inlet and total temperature of the volute inlet are set to 500 °C and 13.85 MPa. The turbine impeller fluid domain is set at a rotational speed of 40000 r/min around the Z axis. The outlet pressure of the volute outlet is set to 7.84 MPa. The interface between the volute outlet and the nozzle inlet, and the interface between the nozzle outlet and the impeller inlet are arranged as coupling interfaces. A hybrid model is used to couple the flow of the vane and the vane. The volute and the nozzle wall are arranged as an absolute stationary wall surface, and the impeller wall surface is a relatively static wall surface. The upper and lower walls of all areas are set as insulated walls to meet the no-slip flow conditions.
Figure 6 shows the overall flow field distribution of the turbine. Carbon dioxide with high temperature and high pressure flows out from the volute and is evenly distributed in each nozzle flow channel. The overall flow field distribution is relatively uniform. There is no large vortex and the flow is smooth. Due to the existence of the tip clearance, part of the carbon dioxide flows from the moving blade pressure surface side to the suction surface side under the action of the pressure gradient. Due to the influence of the blunt front edge at the impeller inlet, the high-speed fluid flowing from the nozzle impacts on the leading edge of the impeller blade. A relatively clear flow separation phenomenon occurs at the leading edge of the pressure surface side of the impeller blade. Overall, there is no large vortex in the entire flow passage and the flow loss is small.

The pressure and temperature are uniformly distributed along the circumferential direction. The carbon dioxide accelerates in the nozzle and flows into the impeller to expand. The pressure and temperature gradually decrease along the flow direction.

4. S-CO$_2$ turbine load parameter transfer
The impeller is the largest load bearing component of the turbine. The impeller works in the harsh environment of high speed and high pressure. Its structural safety is especially critical for the safety and reliability of the turbine. In this paper, a turbine flow-thermal-structure multiphysics coupling model is established. The influence of working fluid pressure and temperature load on the structural strength of the impeller is considered. The multiphysics coupling analysis of the turbine impeller structure is completed.

4.1. Three-dimensional structure model
Based on the turbine aerodynamic design scheme, the turbine impeller structure scheme is designed in detail. The impeller has twelve main blades and twelve split blades. In order to simplify the analysis of the object, a tapered block is used instead of the actual structure of the seal. The turbine impeller
material is titanium alloy TC4. Properties of the material at operating conditions of 510 °C are shown in Table 2.

Table 2 Properties of the turbine impeller material

| Number | Variable                              | Value                |
|--------|---------------------------------------|----------------------|
| 1      | Density                               | 4430 kg/m³           |
| 2      | Poisson's ratio                       | 0.327                |
| 3      | Elastic Modulus                       | 80.6 GPa             |
| 4      | Linear expansion coefficient          | 9.7×10⁻⁶ K⁻¹         |

4.2. Flow-thermal coupling analysis

The temperature and pressure loads transmitted to the impeller structure are obtained by the flow-thermal coupling analysis of the turbine. The mesh model is shown in Figure 11. The blue area in the figure is the fluid domain with 2.65 million grid elements. The red area in the figure is the solid domain with 4.69 million elements and 0.92 million nodes.

The nozzle inlet total pressure and total temperature are set to 13.85 MPa and 500 °C respectively. The flow direction is perpendicular to the inlet section. The static pressure at the impeller outlet is set to 7.84 MPa. The "fluid-fluid" coupling surface is constructed at the interface between the nozzle flow path and the impeller flow path. The coupling surface interpolation is performed by the frozen rotor method. The "fluid-solid" coupling surface is constructed at the interface between the impeller flow path and the impeller solid domain. Heat transfer is performed in a heat flux conservation manner. Both the fluid domain and the solid domain are provided with a global rotational speed of 40,000 rpm in the positive direction of the z-axis.

Fig. 7  Flow-thermal coupling analysis model grid and results of the turbine impeller

The surface temperature distribution of the impeller obtained by flow-thermal coupling analysis is shown in Fig. 7(b). The highest temperature point appears on the surface of the hub of the impeller inlet, which is 747.0K. The lowest temperature point appears at the axial end of the impeller's working
outlet, which is 703.7K. The maximum temperature difference of the impeller structure is 43.3K (°C). The impeller temperature gradually decreases along the flow direction of the working fluid. The surface pressure distribution of the impeller obtained by flow-thermal coupling analysis is shown in Fig. 7(c). The pressure gradually decreases along the flow direction and is basically matched with the inlet and outlet parameters of the working fluid.

5. S-CO₂ turbine structure analysis
A turbine thermal-solid coupling analysis model is constructed, in which the total number of elements is 5.12 million and the total number of nodes is 920,000.

The boundary conditions for the thermal-solid coupling analysis are set as follows:
1) Pressure load. A corresponding aerodynamic pressure load is applied to the coupling surface. The pressure at the inlet of the seal is applied to the back of the impeller: 10.97 MPa. The pressure at the seal outlet is applied to the seal outlet end face node: 0.4 MPa.
2) Temperature load. According to the results of the flow-thermal coupling analysis, the temperature load of the impeller model is set.
3) Displacement constraint. Tangential displacement constraints are imposed on the inner surface nodes of the impeller. Axial displacement constraints are applied to the upper end nodes of the impeller. Axial degrees of freedom are coupled to the back nodes of the impeller.
4) Centrifugal load. A centrifugal force load at a working speed of 40000 rpm is applied to the model.

![Deformation and Stress](image)

**Fig. 8** Thermal-Solid coupling analysis results of the turbine impeller

Figure 8(b) shows the equivalent stress distribution of the impeller. The maximum equivalent stress appears at the fillet of the turbine impeller and the seal with a value of 324 MPa. Figure 8(c) shows the equivalent stress distribution of the blade hub. The maximum equivalent stress appears at the root radius of 60% axial chord of the of the turbine main blade with a value of 95.7 MPa. Due to the leakage of the working fluid at the back of the impeller, the large bending moment of the radial section of the impeller causes a large stress on the top of the main blade. The yield limit of the material at 500 °
C is 441 MPa. The maximum equivalent stress value of the turbine impeller is much smaller than this value, indicating that the turbine impeller design meets the strength design requirements.

6. Conclusions
The S-CO₂ turbine flow-thermal-solid coupling analysis model is constructed in this paper. The flow-thermal coupling analysis results are applied as boundary conditions to the finite element model of the S-CO₂ turbine structure. Moreover, the strength analysis of the S-CO₂ turbine is conducted by thermal-solid coupling analysis. The maximum equivalent stress of the S-CO₂ turbine impeller at the working condition appears at the rounded corner between the turbine impeller and the seal with a value of 324 MPa. This value is less than the yield limit of the material at 500 °C. Therefore, the impeller structure meets the strength design requirements.

Acknowledgements
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. 2016CFA019, NO. 2017CFB325).

References
[1] Shintaro Ishiyama, Yasushi Muto, Yasuyoshi Kato, et al. Study of steam, helium and supercritical CO₂ turbine power generations in prototype fusion power reactor[J]. Progress in nuclear energy, 2008, 50: 325-332.
[2] C J Duan, X Y Yang, J Wang. Parameters optimization of supercritical carbon dioxide brayton cycle[J]. Atomic Energy Science and Technology, 2011, 42(12): 1489-1494.
[3] X R Wang, Y Wu, J F Wang, et al. Thermo-economic analysis of a recompression supercritical CO₂ cycle combined with a transcritical CO₂ cycle[C]. Proceedings of ASME Turbo Expo 2015, Montreal, Canada, GT2015-42033.
[4] J F Wang, Y P Huang, J G Zhang, et al. Recent research progress on supercritical carbon dioxide power cycle in china[C]. Proceedings of ASME Turbo Expo 2015, Montreal, Canda, GT2015-43938.
[5] J H Qi, T Reddell, K Qin, et al. Supercritical CO₂ radial turbine design performance as a function of turbine size parameters[C]. Proceedings of ASME Turbo Expo 2016, Seoul, South Korea, GT2016-58137.
[6] H Z Zhang, H Zhao, Q H Deng, et al. Aerothermodynamic design and numerical investigation of supercritical carbon dioxide turbine[C]. Proceedings of ASME Turbo Expo 2015, Montreal, Canda, GT2015-42619.