Design of a Supercritical CO₂ Compressor for Use in a 1 MWe Power Cycle

Peng Jiang, Bo Wang, Yong Tian,* Xiang Xu, and Lifeng Zhao

ABSTRACT: Supercritical CO₂ power cycles are considered to be a more effective means to replace the steam Rankine cycle in power generation by power coal in the future. However, CO₂ compressors for this application have not been well developed. A conceptual design of the compressor for a 1 MWe cycle has been summarized and a calculation method of the axial force of a supercritical CO₂ compressor has been introduced. The influences of inlet temperature and pressure near the critical point on compressor performance, the rotor dynamics analysis for this compressor, and the influence of the rotation factor C₀ on compressor axial force were also investigated. The results show that the changes in inlet temperature and pressure near the critical point have great influences on compressor performance and the axial force of the compressor is closely related to the selection of the rotation factor C₀. The pressure ratio and power decrease with the increase of inlet temperature; when the inlet temperature increases by 2 °C, the pressure ratio decreases by 7–24% and the power decreases by 1–9%. With the increase of inlet temperature, the maximum efficiency of the compressor decreases, and the maximum pressure ratio of the compressor decreases with the increase of inlet pressure. When the rotation factor C₀ is equal to 0 and 0.2, the axial force of the compressor decreases with the increase of rotating speed. When the rotation factor C₀ is equal to 0.4, the curves of the compressor with the flow rate at different speeds begin to produce intersections. When the rotation factor C₀ is equal to 0.6, 0.8, and 1, the axial force of the compressor increases with the increase of rotating speed.

1. INTRODUCTION

With the development of society, energy is in the increasing demand, and at the same time, it brings various environmental problems. In recent years, countries have been paying increasing attention to environmental problems, and targets such as carbon reduction and carbon neutralization have been put on the agenda. Renewable energy sources, such as solar, biomass, geothermal, wind, and hydro, can be good alternatives to conventional fuel sources. These sustainable energy sources are available in sufficient quantities and have minimal impact on the environment. Supercritical carbon dioxide (sCO₂) Brayton cycles have emerged as an efficient technology in recent years, with thermal efficiencies reaching about 50%, which is considered a promising way to generate electricity efficiently. Supercritical carbon dioxide Brayton cycles can be applied to solar energy, nuclear energy, geothermal energy, and other occasions.

Much scientific research on supercritical carbon dioxide cycles and power generation has been carried out in the past decade. Le Moullec9 designed a concept of a coal-fired power plant built around a supercritical CO₂ Brayton power cycle and 90% postcombustion CO₂ capture and adapted the power cycle to the coal-fired boiler thermal output. Al-Sulaiman and Atif10 conducted a thermodynamic comparison of five supercritical carbon dioxide Brayton cycles integrated with a solar power tower and found that the regenerative Brayton cycle, although simpler in configuration, shows comparable performance to the recompression Brayton cycle.

At the same time, as the key equipment in the cycle, turbines and compressors have been studied to improve the cycle efficiency and extend the applicability of sCO₂ power generation systems by researchers at Sandia National Laboratories (SNL).11–17 Bidkar et al.18,19 summarizes a scale-up of the 10 MWe Southwest Research Institute (SwRI) and General Electric (GE) Sunshot CO₂ turbine design to about 50 MWₑ size and a clean-sheet conceptual design of the main compressor and recompressor for a 450 MWₑ reheat cycle. The non-reheat recompression cycle can achieve >49% cycle efficiency at ISO conditions with wet cooling. A meanline model of a centrifugal compressor and a method for coupled optimization with a heat pump system were presented by Meroni et al.20 Ameli et al.21 investigated and compared different centrifugal compressor
design methodologies in close proximity to the critical point and suggested the most accurate design procedure based on the findings. A design strategy for very low flow coefficient multistage compressors operating with supercritical CO2 was presented by Lettieri et al.22

There are many types of compressors, which can be divided into positive displacement compressors and speed compressors according to their working principles. The working principle of a positive displacement compressor is to compress the volume of the gas and increase the density of gas molecules per unit volume to increase the pressure of compressed air. The working principle of a speed compressor is to increase the moving speed of gas molecules so that the kinetic energy of gas molecules can be converted into the pressure energy of the gas, thus increasing the pressure of compressed air. As a kind of speed compressor, a centrifugal compressor has the advantages of small size, light weight, and compact structure.

A supercritical carbon dioxide cycle testbed is being built in Shanghai Lingang District, and its schematic diagram is shown in Figure 1. This paper presented a conceptual design and a preliminary design for a single-shaft back-to-back sCO2 main compressor for a 1 MWe power cycle and introduced a calculation method of the axial force of the supercritical CO2 compressor. The rotor layout with necessary components is presented in Section 2. The commercially available meanline prediction software COMPAL23 is used for compressor design. The results are presented in Section 3. The rotordynamic analysis for this compressor is presented in Section 4. The compressor axial force analysis is presented in Section 5. Finally, a summary of the present work is presented in Section 6.

2. ROTOR LAYOUT

The rotor layout24 with necessary components of the compressor for the back-to-back configuration is shown in Figure 2.

Standard locations for internal components are as follows

1. Nut
2. Thrust collar
3. Journal bearings
4. Positioning shaft sleeve
5. Dry gas seals
6. Second-stage impeller
7. First-stage impeller
8. Coupling
9. Axis

The nut is used for axial fixation of rotors, and the axial force transmission between the rotor and the thrust bearing is completed through the thrust collar. Journal bearings are used to support the rotor. The positioning shaft sleeve is used for axial positioning among various parts of the rotor. Dry gas seals are used to prevent process gas from leaking to the outside. As a working element, impellers are used to increase the pressure of the gas. Coupling is mainly used for connecting the compressor with the speed-increasing box or motor. Various parts of the rotor are connected by the axis.

This compressor is designed according to API 617,25 which covers many of the critical components on the shaft and lists out the required analysis and studies for rotor dynamics.

Component sizing starts with the furthest outboard component, which is the coupling on the drive end of the shaft, which sees the smallest diameter and also peak stresses from torsion. For assembly, it is important that the diameters step up from the coupling diameter to allow for the ease of install and also prevent the damaging of critical surfaces. Compared to more conventional power turbines, one of the big advantages of sCO2 is its compact flow design. With its high density, the airfoils on the impellers can be made relatively small to reduce the overall size of the machine.24

3. COMPRESSOR SIZING

Preliminary sizing of the compressor was performed to determine the feasibility of the compressor for the predicted cycle conditions. The operating conditions used for designing the compressor are based on the thermodynamic cycle and are
listed in Table 1. The compressor sizing was obtained using COMPAL. The first-stage design pressure ratio is 1.55, and the second-stage design pressure ratio is 2.05. The parameters of the impellers are shown in Table 2.

| parameter                     | compressor |
|-------------------------------|------------|
| inlet pressure $P_1$ (MPa)    | 8          |
| inlet temperature $T_1$ (°C)  | 33         |
| inlet mass flow rate ($kg/s$) | ≥26        |
| outlet pressure $P_2$ (MPa)   | 24         |
| max speed (rpm)               | 25 000     |

Table 2. Parameters of Impellers

| parameter                      | first stage | second stage |
|--------------------------------|-------------|--------------|
| impeller-hub inlet diameter (mm) | 70          | 70           |
| impeller exit diameter (mm)    | 104.8       | 115.9        |
| impeller exit width (mm)       | 4.65        | 4.7          |
| clearance gap (mm)             | 0.3         | 0.3          |
| number of impeller main blades | 7           | 7            |
| number of impeller splitter blades | 7         | 7            |
| vaneless diffuser exit diameter (mm) | 136.2 | 150.7 |

The layout of the second-stage compressor design using COMPAL is shown in Figure 3. The compressor was designed with two volutes. The outlet of the first-stage volute and the inlet of the second-stage volute are connected from the outside through pipelines. The first-stage design pressure ratio is 1.95, and the second-stage design pressure ratio is 1.55.

COMPAL can be used to obtain a definition of stage geometry and an accurate estimate of compressor efficiency. The performance maps of the compressor within a certain flow range at constant speed are shown in Figure 4. The pressure ratio of the compressor decreases with the increase of flow rate. The power was found to increase with the increase of flow rate.

Because the inlet condition of the compressor is close to the critical point of carbon dioxide, the fluctuation of temperature and pressure will cause a drastic change in carbon dioxide density, which will affect the performance of the compressor, and it is difficult to ensure the stability of the inlet condition of the compressor in the supercritical carbon dioxide cycle.
Therefore, it is necessary to study the influence of inlet temperature and pressure on compressor performance. Figure 5 shows the predicted performance curves of the compressor at 25 000 rpm, an inlet pressure of 8 MPa, and different inlet temperatures, and Figure 6 shows the predicted performance curves of the compressor at 25 000 rpm, an inlet temperature of 33 °C, and different inlet pressures.

**Figure 5.** Predicted (a) pressure ratio total—total, (b) power, and (c) adiabatic efficiency total—total versus $Q_m$ of the compressor at 25 000 rpm with different inlet temperatures.

**Figure 6.** Predicted (a) pressure ratio total—total, (b) power, and (c) adiabatic efficiency total—total versus $Q_m$ of the compressor at 25 000 rpm with different inlet pressures.
The results show that the pressure ratio and power decrease with the increase of inlet temperature; when the inlet temperature increases by 2 °C, the pressure ratio decreases by 7−24% and the power decreases by 1−9%. With the increase of inlet temperature, the maximum efficiency of the compressor decreases. This is mainly because with the increase of inlet temperature, the inlet condition becomes closer to the critical point of carbon dioxide, which makes it easy to produce condensation at the blade tip of the impeller and affects the performance of the compressor.

With the increase of inlet pressure, the maximum pressure ratio of the compressor decreases and the pressure changes more smoothly with the flow rate. Compressor power increases with the increase of inlet pressure. The higher the inlet pressure, the higher the maximum adiabatic efficiency. This is mainly attributed to the fact that the higher the inlet pressure, the farther the inlet condition deviates from the critical point of carbon dioxide and the more difficult it is to produce condensation in the impeller passage, which is beneficial to the efficient and stable operation of the compressor.

A three-dimensional (3D) layout for the back-to-back configuration and specific size data are shown in Figure 7. While the compressor design is preliminary in nature, it provides a basis for detailed design in future work.

4. ROTOR DYNAMICS ANALYSIS

The rotor dynamics stability of the compressor was analyzed using DYROBES software. The solid model of the compressor (see Figure 7) was used as a starting point for geometry definition. The rotor dynamics model is shown in Figure 8.

Shaf elements were used to model the rotor length, and diameter values of the nut, thrust collar, positioning shaft sleeve, coupling, and axis. Disks were used to model the mass and inertia of impellers. Tilt-pad journal bearings were used in this study with both stiffness and damping changing with speed. The shaft diameter is 40 mm. The radial loads are 61.87 and 74.52 N.

The number of pads of the tilt-pad journal bearing is 5. The offset point is 0.55. The width of the radial pad is 70 mm and the preload value is 0.5. Bearing stiffness and damping calculations were performed using DYROBES software. As shown in Figures 9 and 10, the stiffness and damping of the journal bearings change with the rotating speed.

The goal of the rotor dynamics analysis was to study the forced unbalance response of the rotor and evaluate the stability of the rotor. The first critical speed, based on the imbalance response without seal effects, is predicted to be approximately 14 600 rpm for the nominal rotor. This is lower than the operating speed of 25 000 rpm. The lateral steady-state response curves at stations 5, 9, 11, 13, and 17 are shown in Figure 11. The nominal rotor length is predicted to meet API separation margin requirements for the first critical speed. The subject mode is predicted to have the minimum separation margin of higher than 41% and the maximum peak response of 16.9 μm, meeting API separation margin requirements. Thus, the rotor is an acceptable rotor dynamics configuration.

The influence of rotor length on the first-order critical speed was also studied, and the results are shown in Figure 12. The rotor length increases by 4%, and the first-order critical speed decreases by 8.2%.

5. AXIAL FORCE ANALYSIS

The high pressure and high density of the supercritical carbon dioxide compressor bring high blast loss and high axial thrust, and the axial force per unit area is far higher than that of the ordinary compressor. It is difficult to accurately calculate the axial force of the supercritical CO2 compressor because the axial force of the compressor is affected by many parameters, such as size, flow rate, pressure, temperature, rotation factor, and so on. The calculation method of the axial force of the supercritical carbon dioxide compressor was introduced to facilitate the evaluation of axial force in supercritical CO2 compressor design.

As shown in Figure 13, the axial force on the compressor impeller is mainly divided into three parts, namely, its body Fc1 acting on the compressor inlet face, gas force Fc2 acting on the impeller, and gas force Fc3 acting on the impeller back by the gas in the wheel back gap, and the following relationships exist among each force.

\[ F_c = F_{c3} - F_{c1} - F_{c2} \]  
\[ F_{c1} = F_{c10} + F_{c1m} \]  
\[ F_{c3} = F_{c10} - D_{c1}^2 \int_{c1} p_c 2\pi r \, dr \]

where \( F_{c10} \) is the axial force generated by the static pressure of the inlet gas and \( F_{c1m} \) is the axial force generated by the change of momentum from the axial direction to the radial direction. The calculation formulas are as follows:

\[ F_{c10} = \frac{\pi(D_{c1}^2 - D_{o1}^2)}{4} p_{c1} \]  
\[ F_{c1m} = \frac{\pi(D_{c1}^2 - D_{o1}^2)}{4} p_{c1} \]  
\[ F_{c3} = \frac{\pi(D_{c1}^2 - D_{o1}^2)}{4} p_{c1} + \frac{\pi(D_{c1}^2 - D_{o1}^2)}{4} p_{c1} \]

where \( D_{o1} \) is the diameter of the positioning shaft sleeve, \( D_{c1} \) is the diameter of the impeller inlet, \( p_{c1} \) is the pressure of the
impeller inlet, $Q_m$ is the inlet mass flow rate, and $c_{01}$ is the inlet velocity.

Suppose that the change rule of $p$ is as follows\textsuperscript{26}

$$P_r = P_{c2} \left( \frac{r_c}{r_{c2}} \right)^2$$

where $p_{c2}$ is the pressure of the impeller outlet and $r_{c2}$ is the radius of the impeller outlet.

The following relation is obtained

$$F_{c2} = \int_0^{r_{c2}} P_{c2} \left( \frac{r_c}{r_{c2}} \right)^2 2\pi r_c \ dr_c = \frac{\pi P_{c2}}{2r_{c2}} (r_{c2}^4 - r_{c1}^4)$$

$$= \frac{\pi P_{c2}}{8r_{c2}^2} \left( D_{c2}^2 - D_{c1}^2 \left( \frac{D_3}{D_{c2}} \right)^2 \right)^2$$

(8)

where $C_0$ is the rotation factor and $D_{c2}$ is the diameter of the impeller outlet.

The radial equilibrium equation is as follows

$$\frac{dP}{dr} = \rho(C_0\omega)^2 r$$

(9)

The gas pressure at any radius can be expressed as follows

$$P(r) = \int_0^{r_{c2}} dP = P_{c2} - \frac{\rho(C_0\omega)^2}{8} (D_{c2}^2 - D_3^2)$$

(10)

The second term on the right side of the formula is the pressure drop at any radius, that is, the pressure drop caused by the rotation factor is considered. The axial thrust formula on the back of the wheel disc can be expressed as follows

$$F_{c3} = 2\pi \int_0^{r_{c3}} P(r)r \ dr$$

$$= \frac{\pi}{4} (D_{c2}^2 - D_{c3}^2) P_{c3} - \frac{\pi \rho (C_0\omega)^2}{64} (D_{c3}^2 - D_{c1}^2)^3$$

(11)

where $D_{c3}$ is the diameter of the shaft.

The compressor is a two-stage centrifugal compressor, and the inlet of each stage adopts a dry gas seal. The thrust of the dry gas seal can be calculated by the following formula

$$F = \frac{\pi}{4} (D_{c2}^2 - D_{c3}^2) P_3$$

(12)

where $D_3$ is the dry gas seal top diameter, $P_3$ is the dry gas seal pressure, and $D_1$ is the dry gas seal balance diameter.

The related dimension parameters of axial force are shown in Table 3.

The axial force of the compressor was calculated with rotation factor values of 0, 0.2, 0.4, 0.6, 0.8, and 1, and the relationship between the axial force of the compressor and mass flow under different rotation factors is shown in Figure 14. When $C_0$ is equal to 0 and 0.2, the axial force of the compressor decreases with the increase of rotating speed. When $C_0$ is equal to 0.4, the curves of the compressor with the flow rate at different speeds begin to produce intersections. When $C_0$ is equal to 0.6, 0.8, and 1, the...
axial force of the compressor increases with the increase of rotating speed. When the rotation factor is small, the rotation factor has a slight influence on the axial force. With the increase of rotating speed, the axial forces on both sides are better offset, which makes the axial force of the compressor smaller. With the increase of rotation factor, the influence of the rotation factor on axial force gradually increases. When the rotation factor and mass flow rate increase to a certain value, the axial force direction of each stage impeller changes and then affects the axial force magnitude and direction of the compressor.

At the design speed (n = 25,000 rpm), the relationship between the rotation factor and the axial force of the compressor is shown in Figure 15. At the same mass flow rate, the axial force of the compressor increases with the increase of rotation factor. The larger rotation factor is more obvious to the axial force of the compressor. Because the specific value of the rotation factor cannot be given accurately, it increases the difficulty of balancing the axial force of the compressor, and sufficient margin should be considered when designing the compressor.

6. RESULTS AND DISCUSSION

This paper presented a preliminary design for a single-shaft back-to-back sCO₂ compressor and introduced a calculation method of the axial force of the supercritical CO₂ compressor. The influences of inlet temperature and pressure on compressor performance, the rotor dynamics analysis for this compressor, and the influence of the rotation factor on compressor axial force were also investigated. Specific conclusions can be summarized as follows:

1. The pressure ratio and power decrease with the increase of inlet temperature; when the inlet temperature increases by 2 °C, the pressure ratio decreases by 7—24% and the power decreases by 1—9%. With the increase of inlet temperature, the maximum efficiency of the compressor decreases. With the increase of inlet pressure, the maximum pressure ratio of the compressor decreases, and the pressure changes more smoothly with the flow rate. Compressor power increases with the increase of inlet pressure. The higher the inlet pressure, the higher the maximum adiabatic efficiency.

2. The first critical speed, based on the imbalance response without seal effects, is predicted to be approximately 14,600 rpm for the nominal rotor, which is lower than the operating speed of 25,000 rpm. The nominal rotor length is predicted to meet API separation margin requirements for the first critical speed. The subject mode is predicted to have the minimum separation margin of higher than 41% and the maximum peak response of 16.9 μm, meeting API separation margin requirements. The rotor length increases by 4%, and the first-order critical speed decreases by 8.2%.

3. When C₀ is equal to 0 and 0.2, the axial force of the compressor decreases with the increase of rotating speed. When C₀ is equal to 0.4, the curves of the compressor with the flow rate at different speeds begin to produce intersections. When C₀ is equal to 0.6, 0.8, and 1, the axial force of the compressor increases with the increase of rotating speed. With the increase of rotation factor, the influence of the rotation factor on axial force gradually increases. At the same mass flow rate, the axial force of the compressor increases with the increase of rotation factor. The larger rotation factor is more obvious to the axial

| Table 3. Dimension Parameters | first stage | second stage |
|------------------------------|------------|-------------|
| D₀₀/mm | 65         | 65          |
| D₀₁/mm | 81         | 80          |
| D₀₂/mm | 104.8      | 115.9       |
| D₀₃/mm | 60         | 60          |
| D₁₀/mm | 63.5       | 63.5        |
| D₁₁/mm | 100        | 100         |
Figure 14. Relationship between the axial force and flow rate of the compressor under different rotation factors: (a) $C_0 = 0$, (b) $C_0 = 0.2$, (c) $C_0 = 0.4$, (d) $C_0 = 0.6$, (e) $C_0 = 0.8$, and (f) $C_0 = 1$.

force of the compressor. Because the specific value of the rotation factor cannot be given accurately, it increases the difficulty of balancing the axial force of the compressor, and sufficient margin should be considered when designing the compressor.

Future work includes verifying the accuracy of the compressor performance prediction results and axial force calculation.
through experiments. In addition, since this research is based on the design of a supercritical CO2 compressor in a 1 MWe power cycle, the future work should also include the design of a supercritical CO2 compressor with higher power to meet high-power applications.

■ AUTHOR INFORMATION

Corresponding Author

Yong Tian — Institute of Engineering Thermophysics, Chinese Academy of Sciences, Beijing 100190, People’s Republic of China; orcid.org/0000-0003-1153-1643; Email: jiangpeng@iet.cn, tianyong@mail.etp.ac.cn

Authors

Peng Jiang — Institute of Engineering Thermophysics, Chinese Academy of Sciences, Beijing 100190, People’s Republic of China

Bo Wang — Institute of Engineering Thermophysics, Chinese Academy of Sciences, Beijing 100190, People’s Republic of China

Xiang Xu — Institute of Engineering Thermophysics, Chinese Academy of Sciences, Beijing 100190, People’s Republic of China

Lifeng Zhao — Institute of Engineering Thermophysics, Chinese Academy of Sciences, Beijing 100190, People’s Republic of China

Complete contact information is available at: https://pubs.acs.org/10.1021/acsomega.1c05023

Notes

The authors declare no competing financial interest.

■ ACKNOWLEDGMENTS

The authors are grateful to Effects of Near-critical Non-equilibrium Phase-change on the Internal Flow and Aerodynamic Performance of Supercritical CO2 Compressor (No.: S2006216) for the financial assistance for this study.

■ NOMENCLATURE

- $P_1$: inlet pressure (MPa)
- $P_2$: outlet pressure (MPa)
- $\text{nimpepprotating speed (r/min)}$
- $F_{C,\text{in}}$: axial force acting on the compressor inlet face (kN)
- $F_{C,\text{back}}$: axial force acting on the impeller back (kN)
- $F_{C,\text{momentum}}$: axial force generated by the change of momentum from the axial direction to the radial direction (kN)
- $D_{c1}$: diameter of the impeller outlet (m)
- $D_{c2}$: dry gas seal balance diameter (m)
- $P_{d}$: dry gas seal pressure (Pa)
- $K_{C1}/K_{C2}$: stiffness of the journal bearing (N/mm)
- $F_{C,\text{impeller}}$: axial force acting on the impeller (kN)
- $C_{rot}$: rotation factor
- $F_{C,\text{axial}}$: axial force generated by the static pressure of the inlet gas (kN)
- $r_{d1}$: radius of the impeller outlet (m)
- $c_{01}$: inlet velocity (m/s)
- $D_{c3}$: dry gas seal top diameter (m)
- $D_{c4}$: diameter of the shaft (m)
- $C_{p1}/C_{p2}$: damping of the journal bearing (N·s/mm)
- $P_{c1}$: pressure of the inlet gas (MPa)

■ REFERENCES

(1) Al-Sulaiman, F. A.; Atif, M. Performance comparison of different supercritical carbon dioxide Brayton cycles integrated with a solar power tower. Energy 2015, 82, 61–71.

(2) Turchi, C. S.; Ma, Z.; Neises, T.; Wagner In Thermodynamic Study of Advanced Supercritical Carbon Dioxide Power Cycles for High Performance Concentrating Solar Power Systems, Proceedings of the ASME 2012 6th International Conference on Energy Sustainability, San Diego, CA; July 23–26, 2012.

(3) Utamura, M.; Tamaura, Y.; Hasuie, H. In Some Alternative Technologies for Solar Thermal Power Generation, International solar energy conference, Denver, CO; July 8–13, 2006, 2007; pp 75–84.

(4) Ma, Z.; Turchi, C. In Advanced Supercritical Carbon Dioxide Power Cycle Configurations for Use in Concentrating Solar Power Systems, NREL/CPS00-50787, National Renewable Energy Laboratory: Golden, CO; 2011.

(5) Dostal, V.; Hejzlar, P.; Driscoll, M. J. The supercritical carbon dioxide power cycle: comparison to other advanced power cycles. Nucl. Technol. 2006, 154, 283–301.

(6) Sabharwall, P.; Kim, E. S.; Patterson, M. Fluoride High Temperature Reactor Integration with Industrial Process Applications; TEV-1160, March 29, Idaho National Laboratory: Idaho Falls, ID; 2011.

(7) Dostal, V. A supercritical carbon dioxide cycle for next generation nuclear reactors. Ph.D. thesis, Nuclear Engineering, Massachusetts Institute of Technology, 2004.

(8) Chen, H.; Goswami, D. Y.; Stefanakos, E. K. A review of thermodynamic cycles and working fluids for the conversion of low-grade heat. Renewable Sustainable Energy Rev. 2010, 14, 3059–3067.

(9) Le Moullac, Y. Conceptual study of a high efficiency coal-fired power plant with CO2 capture. Energy 2013, 49, 32–46.

(10) Al-Sulaiman, F. A.; Atif, M. Performance comparison of different supercritical carbon dioxide Brayton cycles integrated with a solar power tower using a supercritical CO2 Brayton cycle. Energy 2015, 82, 61–71.

(11) Wright, S. A.; Fuller, R.; Pickard, P. S.; Vernon, M. E. In Initial Status and Test Results from a Supercritical CO2 Brayton Cycle Test Loop, International Conference on Advances in Nuclear Power Plants, Anaheim, CA; June 8–12, 2008; p 768–5.

(12) Wright, S. A.; Fuller, R.; Noall, J.; Radel, R.; Vernon, M. E.; Pickard, P. S. In Supercritical CO2 Brayton Cycle Compression and
Control Near the Critical Point, International Conference on Advances in Nuclear Power Plants, Anaheim, CA, June 8–12, 2008; pp 810–819.

(13) Wright, S. A.; Pickard, P. S.; Vernon, M. E.; Radel, R. F.; Fuller, R. In Description and Test Results from a Supercritical CO2 Brayton Cycle Development Program, 7th International Energy Conversion Engineering Conference, Denver, CO, August 2–5, 2009.

(14) Wright, S. A.; Vernon, M. E.; Radel, R. F.; Fuller, R. L. In Supercritical CO2 Main Compressor Performance Measurements, American Nuclear Society Annual Meeting and Embedded Topical Meeting: Nuclear and Emerging Technologies for Space, June 14–18, Atlanta, GA, 2009; pp 499–500.

(15) Wright, S. A.; Conboy, T. M.; Parma, E. J.; Lewis, T. G.; Rochau, G. A.; Suo-Anttila, A. J. In Summary of the Sandia Supercritical CO2 Development Program, Supercritical CO2 power cycle symposium, Boulder, CO, 2011.

(16) Wright, S. A.; Pickard, P. S.; Fuller, R.; Radel, R. F.; Vernon, M. E. In Supercritical CO2 Brayton cycle Power Generation Development Program and Initial Results, Albuquerque, NM, 2009.

(17) Wright, S. A.; Radel, R. F.; Vernon, M. E.; Rochau, G. E.; Pickard, P. S. Operation and Analysis of a Supercritical CO2 Brayton Cycle; SAND2010-0171, Sandia National Laboratories: Albuquerque, NM, 2010.

(18) Bidkar, R. A.; Mann, A.; Singh, R.; Sevincer, E.; Cich, S. Conceptual Designs of 50MWe and 450MWe Supercritical CO2 Turbomachinery Trains for Power Generation from Coal, 2016.

(19) Bidkar, R. A.; Musgrove, G.; Day, M.; Kulhanek, C. D.; Allison, T. In Conceptual Designs of 50MWe and 450 MWe Supercritical CO2 Turbomachinery Trains for Power Generation from Coal, Part 2: Compressors, The 5th International Symposium - Supercritical CO2 Power Cycles March 28–31, 2016.

(20) Meroni, A.; Zhlsdorf, B.; Elmegaard, B.; Haglind, F. Design of centrifugal compressors for heat pump systems. Appl. Energy 2018, 232, 139–156.

(21) Ameli, A.; Afzalifar, A.; Turunen-Saaresti, T.; Backman, J. Centrifugal compressor design for near-critical point applications. J. Eng. Gas Turb. Power 2018, 141, 1–8.

(22) Lettieri, C.; Baltadjiev, N. D.; Casey, M.; Spakovszky, Z. S. In Low-Flow-Coefficient Centrifugal Compressor Design for Supercritical CO2, Proceedings of the ASME Turbo Expo 2013: Turbine Technical Exposition, 2013, San Antonio, Texas, USA, June 3–7, 2013.

(23) Concepts NREC. [Online]. Available: http://www.conceptsnrec.com/, 2015.

(24) Cich, S. D.; Moore, J.; Mortzheim, J. P.; Hofer, D. In Design of a Supercritical CO2 Compressor for Use in a 10 MWe Power Cycle, The 6th International Supercritical CO2 Power Cycles Symposium, March 27–29, 2018.

(25) API Standard 617, Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services; American Petroleum Institute: Washington, D.C., 2002.

(26) Hang, Z.; Wang, X.; Wang, W. Turbine Compressor; Chemical Industry Press: Beijing, 2013; pp 112–150.