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Investigation on steady aerodynamic performance of a S-CO₂ compressor with different diffusers in solar power system

Qiuwan Du¹, Di Zhang^{1,*} and Yonghui Xie²

¹ MOE Key Laboratory of Thermo-Fluid Science and Engineering, Xi'an Jiaotong University, Xi'an, China

² Shaanxi Engineering Laboratory of Turbomachinery and Power Equipment, School of Energy and Power Engineering, Xi'an Jiaotong University, Xi'an, China

E-mail: zhang_di@mail.xjtu.edu.cn

Abstract. Research on solar power system has become a hot spot recently. The supercritical carbon dioxide (S-CO₂) power cycle has been widely used in the solar power system with excellent advantages of high efficiency and small device size. Centrifugal compressor plays an important role in the S-CO₂ power cycle. The design of a centrifugal compressor is completed according to the requirements of thermal design in this paper. The differences of three kinds of diffusers on the steady aerodynamic performance are further investigated based on the designed model. Finally, the influence of diffuser blade number on the performance is discussed. The results show that the centrifugal compressor with wedge-shaped diffuser holds the best performance under the design and off-design conditions. The diffuser blade number has significant influence on the aerodynamic performance. The current design and performance analysis of centrifugal compressor could provide important reference value for further application of centrifugal compressor in solar power system.

Keywords: solar; centrifugal compressor; design; aerodynamic performance; diffuser

1. Introduction

As a new renewable energy, solar energy has the advantages of rich-reserved, clean, safety and so on. At present, many countries have strengthened the research and development of utilization of solar energy. Brayton cycle systems using S-CO₂ offers advantages such as smaller weight and volume, lower thermal mass and less complex power blocks and has been widely used in Concentrating Solar Power (CSP) plants[1].

Recently a large number of scholars have investigated S-CO₂ Brayton cycle in CSP. Ricardo[2] conducted exergy analysis of four different S-CO₂ Brayton cycle configurations in CSP and found that the thermal efficiency increases monotonically with the temperature. Dynamic characteristics of a S-CO₂ PCS in a solar thermal power plant were investigated by Rajjinesh[3]. R. Chacartegui[4] used three CO₂ cycles in CSP and showed obvious benefits in terms of efficiency and cost. The influence of transient nature of the solar resource on solar thermal power generation was analyzed and a data set for stable S-CO₂ Brayton cycle was established by Iverson[5]. Garg[6] analyzed S-CO₂ Brayton cycle in CSP and selected thermal efficiency, specific work output and magnitude of irreversibility generation as performance indicators to compare with trans- and sub-critical operations.

Centrifugal compressor plays an important role in the S-CO₂ Brayton cycle. The flow characteristics analysis has become a hot pot. A centrifugal compressor required in a solar dish-Brayton system was designed and optimized by Wang[7] and the numerical simulation results showed a great performance. Rinaldi[8][9] numerically analyzed the performance map of a radial compressor operating with S-CO₂. The numerical results were consistent with the experimental data. Guo[10] investigated steady and unsteady performance of a S-CO₂ centrifugal compressor with splitters. A S-CO₂ centrifugal compressor with large input power was designed and numerically investigated by Wang[11] and the



unsteady flow characteristics were revealed in detail. Zhao[12] conducted the aerodynamic design and impeller aerodynamic optimization for a S-CO₂ centrifugal compressor, furthermore, the strength analysis was conducted to obtain the stress and deformation results.

In this paper, a centrifugal compressor is designed and the steady aerodynamic performance is analyzed in detail. The differences of three kinds of diffusers on the aerodynamic performance are further investigated based on the designed model. In addition, the influence of diffuser blade number on the performance is discussed. The current design of centrifugal compressor could provide important reference value for further application of centrifugal compressor in S-CO₂ Brayton cycle for solar power system.

2. Compressor design

The compressor design process is divided into three sub processes, namely 1D flow design, thermodynamic design and geometry design. The thermodynamic parameters of a centrifugal compressor are obtained by the combination of the real gas state equation, the continuity equation, the momentum equation and the energy equation. In addition, the geometry parameters of the impeller and diffuser are determined. The performance estimates and state parameters of the designed centrifugal compressor refer to Zhao's research[12], as listed in Table 1.

Table 1. Performance estimates and state parameters of the designed centrifugal compressor

Items	Parameters	Value
State parameters	Inlet temperature/K	306.7
	Inlet pressure/MPa	8.96
	Outlet pressure/MPa	18.94
	Rotating speed/(r·min ⁻¹)	15000
Performance estimates	Input Power/ MW	5
	Mass flow rate/(kg·s ⁻¹)	272.6
	Pressure ratio	2.1
	Efficiency/%	79.5

In this paper an impeller including 15 blades and three kinds of diffusers including 15 blades are designed, as shown in Figure 1. The geometry parameters are listed in Table 2.

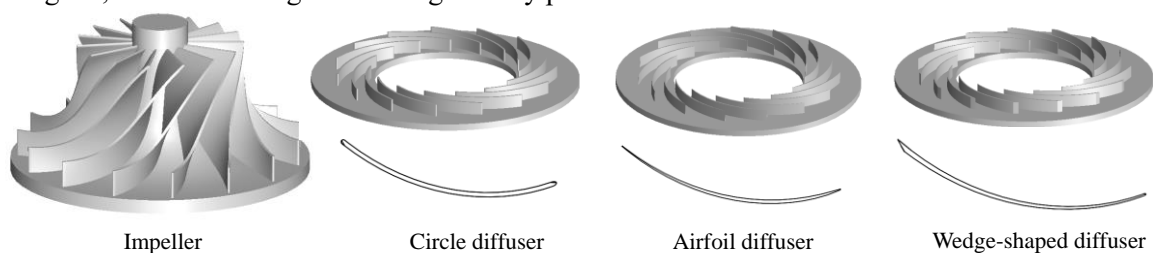
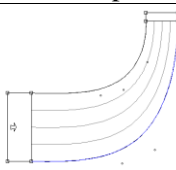
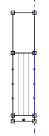


Figure 1. Geometry of the compressor

Table 2. Geometry parameters of the impeller and diffuser

Meridian plane	Parameters	Value
	Blade height at outlet/mm	18.8
	Outer radius of impeller/mm	102
	Outer radius of inlet/mm	60
	Inner radius of inlet/mm	20
	Blade height /mm	18.8
	Outer radius of diffuser/mm	167
	Inner radius of diffuser /mm	112
	Outer radius of outlet /mm	202

3. Numerical method

3.1. Governing equations

The flow characteristics of the compressor were investigated by ANSYS CFX. The continuity, momentum and energy equations are selected as the governing equations. The general form of these equations can be expressed as[13]:

$$\frac{\partial(\rho\varphi)}{\partial t} + \text{div}(\rho u\varphi) = \text{div}(\Gamma \text{grad}\varphi) + S \quad (1)$$

Where φ is the universal variable, and $\varphi = 1$ indicates continuity equation. When φ is temperature, it represents energy equation, and when φ is set as the velocity of an arbitrary direction, Equation (1) corresponds to being a momentum equation. In addition, Γ is generalized diffusion coefficient and S is generalized source term.

In addition, the standard k - ε turbulence model was used for the analysis. The CO2RK database in ANSYS-CFX is adopted to describe the practical property of the working fluid SCO₂. It is based on the RK gas state equation[14] which is shown as:

$$P = \frac{RT}{V-b} - \frac{a}{T^{1/2}V(V+b)} \quad (2)$$

Where, $a = 0.42748 \frac{R^2 T_c^{2.5}}{P_c}$, $b = 0.08664 \frac{RT_c}{P_c}$

Where, R - gas constant, T_c - critical temperature, P_c - critical pressure.

3.2. Mesh and boundary conditions

A single passage model is adopted for flow analysis. The mesh is shown in Figure 2. The hexahedral grid is used in order to improve the calculation accuracy. In addition, the O-H topology was set to the mesh topology. This can greatly reduce the number of grids and further improve the grid quality. The total number of grids for the three models is 201635, 203440 and 288674 respectively after grid independence validation.

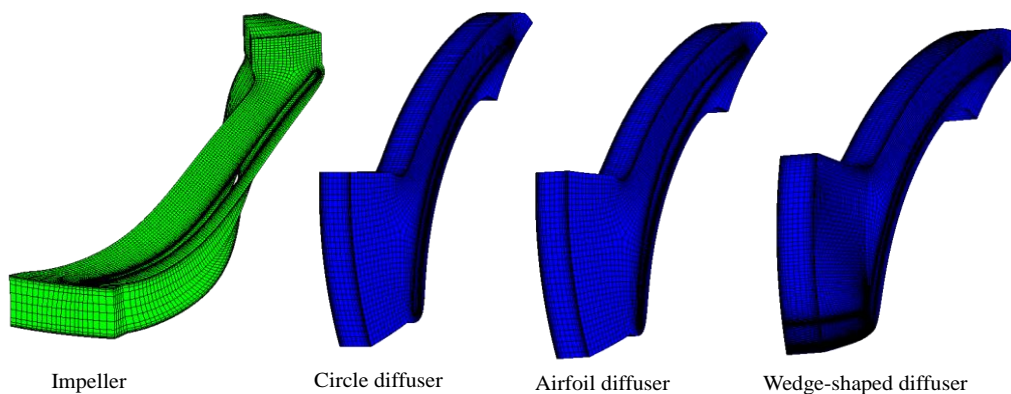


Figure 2. Mesh of the compressor

The boundary conditions employed in this paper for steady aerodynamic performance are as follows: mass flow rate of the inlet (18.17kg/s), static temperature of the inlet (306.7K), and static pressure of the outlet (18.94MPa). Meanwhile, the rotating speed was 15000rpm.

4. Results and discussions

4.1. Steady aerodynamic performance comparison

The pressure contours at mid-span of blades for three models are shown in Figure 3. It can be seen that the pressure distributions of the three models are almost the same and the pressure reaches the minimum at the inlet of the impeller. Due to the centrifugal effect in impeller and the transformation from kinetic energy into pressure energy, the pressure enlarges along the flow passage. In addition, there exists a stagnation point at the leading edge of diffuser blades. Moreover, it is further obtained that the maximum pressure of the airfoil model is slightly larger than that of the other two models. However, the pressure distribution of wedge-shaped diffuser is more uniform.

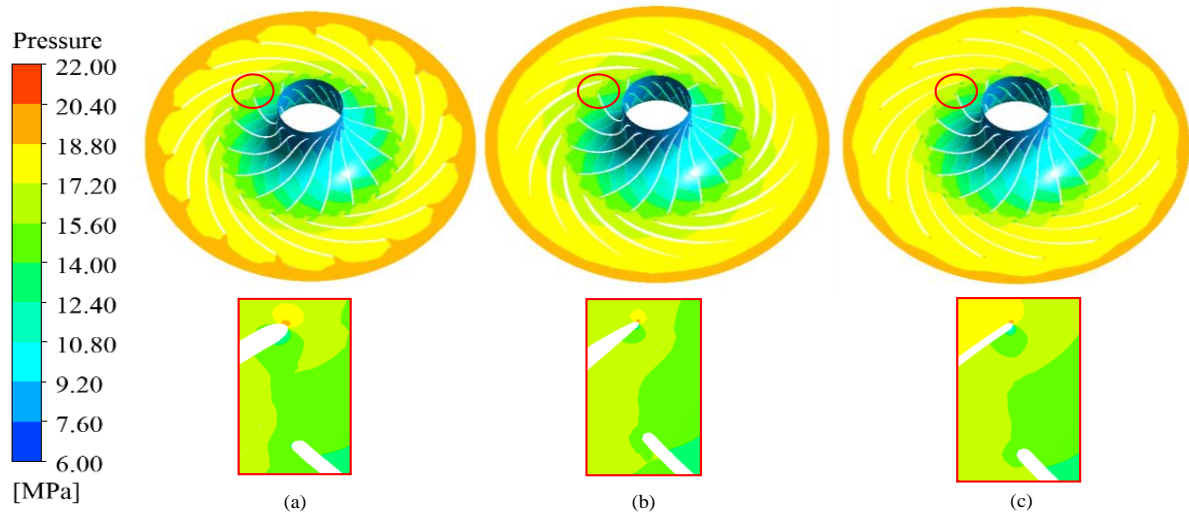


Figure 3. Pressure contours at mid-span of blades: (a) circle; (b) airfoil; (c) wedge-shaped. The temperature distributions are shown in Figure 4. It can be observed that the temperature distributions of the three models are almost the same and the minimum temperature occurs at the inlet of the impeller. The temperature gradually increases along the passage with the increase of pressure. The temperature reaches the maximum value at the leading edge of the diffuser blade. Moreover, it is further obtained that the maximum temperature of the circle model is slightly larger than that of the other two models.

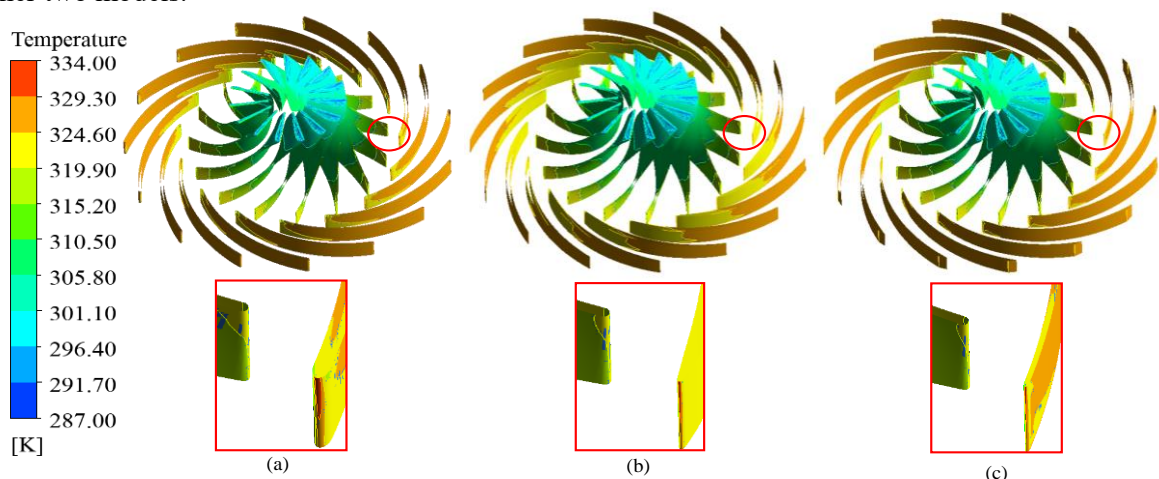


Figure 4. Temperature contours at mid-span of blades: (a) circle; (b) airfoil; (c) wedge-shaped. Figure 5 shows the streamline distributions of the three diffusers. As shown in Figure 5, the streamline distributions of the three diffusers remain almost identical. In addition, there is a velocity stagnation point located at the leading edge of the diffuser blade. The velocity gradually decreases through the diffuser passage resulting in the increase of pressure which is shown in Figure 3. Moreover, it is further obtained that the velocity of the wedge-shaped diffuser is larger than that of the two diffusers and the flow field is more uniform.

In addition, the performance parameters of the three models are also obtained. The powers of the circle model, airfoil model and wedge-shaped model are 4.98MW, 5.01MW and 5.11MW respectively. The pressure ratios are 2.134, 2.126 and 2.154 respectively. Combined with the calculation results of the pressure, temperature and flow field, it concludes that the wedge-shaped model holds the best aerodynamic performance under the design condition.

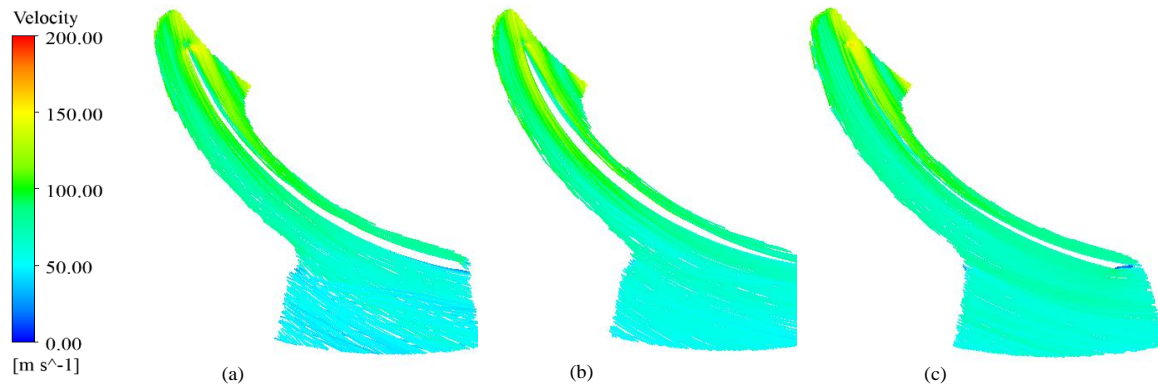


Figure 5. Streamline distributions of the three diffusers: (a) circle; (b) airfoil; (c) wedge-shaped

4.2. Off-design performance comparison

The off-design performance of the three models is investigated in this chapter by changing the mass flow rate which varies 17.80kg/s, 18.17 kg/s, 18.50 kg/s, 19.00 kg/s and 19.50 kg/s. The calculation results are listed in Table 3. It can be seen that the power of the three models increases as the mass flow rate increases while the pressure ratio decreases. It is worth mentioning that the calculation with the airfoil model is not convergent under the conditions with 17.80kg/s and 19.50kg/s, which indicates that the off-design performance of the airfoil model involved in this paper is the worst. Further comparison shows that the power and pressure ratio of the wedge-shaped model is higher than those of the other two models under the same working condition.

Table 3. The off-design performance parameters

Mass flow rate/(kg/s)	Power/MW			Pressure ratio		
	Circle	Airfoil	Wedge-shaped	Circle	Airfoil	Wedge-shaped
17.80	4.90	/	5.02	2.138	/	2.158
18.17	4.98	5.01	5.11	2.134	2.126	2.154
18.50	5.06	5.08	5.18	2.130	2.123	2.151
19.00	5.17	5.20	5.30	2.124	2.116	2.146
19.50	5.28	/	5.41	2.117	/	2.140

4.3. Influence of diffuser blade number

In order to further compare the differences between the three models, the influence of diffuser blade number on the performance is discussed in detail in this chapter. The numbers of diffuser blade are selected as 13, 14, 15, 16 and 17 respectively.

Table 4. The performance parameters with different diffuser blade number

Diffuser blade Number	Power/MW			Pressure ratio		
	Circle	Airfoil	Wedge-shaped	Circle	Airfoil	Wedge-shaped
13	4.96	4.95	5.02	2.100	2.102	2.124
14	4.94	4.99	5.07	2.120	2.115	2.142
15	4.98	5.01	5.11	2.134	2.126	2.154
16	5.02	4.96	5.06	2.144	2.134	2.158
17	5.04	4.97	5.15	2.155	2.139	2.164

The calculation results are listed in Table 4. It can be seen that the power and pressure ratio show an increase trend as a whole with the increase of the diffuser blade number. Further comparison shows that the power and pressure ratio of the wedge-shaped model are higher than those of the other two models under all working conditions.

5. Conclusions

In this paper, the design of S-CO₂ centrifugal compressor is carried out and the steady aerodynamic performance is analyzed in detail. The differences of three kinds of diffusers on the aerodynamic performance is further investigated based on the designed model. In addition, the influence of diffuser blade number on the performance is discussed. The conclusions are as follows:

- (1) The distributions of pressure, temperature and velocity for the circle model, airfoil model and wedge-shaped model remain almost identical. The pressure, temperature and velocity reach the maximum value for the airfoil model, the circle model and the wedge-shaped model respectively.
- (2) The power and pressure ratio of the wedge-shaped model are larger than those of the two models under the design and off-design conditions.
- (3) The power and pressure ratio show an increase trend as a whole with the increase of the diffuser blade number. In addition, the wedge-shaped model show the best performance under the same conditions.

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Acknowledgments

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