Research Article

Performance and Loss Analysis of a Small-Scale Supercritical CO₂ Turbine

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The supercritical carbon dioxide (S-CO₂) power cycle is widely applicable to the utilization of multigrade heat sources due to its high efficiency and compact structure, in which the turbine is critical to the performance of the system. In this study, the operation pattern and output performance of an S-CO₂ axial turbine with a design power scale of 10 kW power are investigated by experiments and numerical simulations. The loss mechanisms and distributions are analyzed using numerical simulations and loss models. The results show that the turbine output power and efficiency are at 9.66 kW and 36.6% at the designed condition and the output performance depends on the pressure ratio and rational speed. In addition, the flow field distribution reveals that the flow separation phenomenon occurs leading to loss, with the tip clearance loss being the most significant. Furthermore, the loss model validated in this paper is proven to be competent enough to fulfill the prediction of the loss distribution under different operational conditions, which could be very promising in the design of an S-CO₂ axial turbine.

1. Introduction

Supercritical carbon dioxide (S-CO₂) is a stable, nontoxic, and thermal conductivity working fluid [1]. As the properties of supercritical carbon dioxide (S-CO₂), it is applied on the Brayton cycles contributing to compacting facilities and improving efficiency [2]. Due to the advantages of the S-CO₂ Brayton cycle, it is regarded as the most promising technology to generate power widely applied in nuclear reactors [3–5], waste heat recovery systems [6, 7], and concentrated solar power systems [8–11]. It also can be applied to address the challenges in air turbines which face low operating temperatures, the restriction of location, and a relatively limited range of conditions [12, 13]. Various layouts have been studied to cover different power scales and optimal configurations of loop components [14, 15]. In the power plants, the recompression cycle layout is investigated including two compressors, one turbine, and two recuperators [16, 17]. The majority of current research and applications on S-CO₂ power systems focused on the tens of MW of output power [2, 18, 19]. As the S-CO₂ cycles were developed, the small-scale S-CO₂ power systems have been promising strategies with power outputs below several hundred kilowatts or even few kilowatts [20–22]. In order to improve the power output of the cycle utilizing low-grade heat, a transcritical CO₂ power cycle was proposed which condensates under the critical state [23–25]. Pan et al. built a transcritical CO₂ power cycle system and generated the electric power output about 11 kW [26]. Wu et al. established a transcritical CO₂ power cycle and found that the net power output increased by 3.9–26.3% compared to the previous research [27].

In a study by Kulhanek and Dostal, it was concluded that the performance of the recompression cycle was significantly improved compared with the simple Brayton cycle. Through comparative analysis, the performance of the compressors and turbines remarkably impacted the efficiency of the cycle [28]. In the research by Sarkar [17], the effects of changes in the turbine and the compressor on performance were compared. It is drawn that the turbine’s efficiency has a more
noticeable effect on the system. Thus, the turbine is a crucial component in the cycle resulting that the S-CO₂ turbine has emerged as a major topic recently. SNL has been improving the S-CO₂ Brayton cycle to optimize the output efficiency and developed a 0.2 MW radial turbine [29, 30]. Zhang et al. developed an aerothermodynamic design of a 15 MW axial and 1.5 MW radial turbine. Numerical analysis was conducted by taking the real gas properties of S-CO₂ into account [31]. Ji et al. employed a neural network to predict the flow distribution and realized multiobjective optimization of a turbine with a genetic algorithm [32]. Qin et al. used the Kriging model to predict the performance of a turbine and improved the efficiency of multiobjective optimization [33]. Wang et al. established an active learning-based model to realize the optimization for aerodynamic performance which takes into account the effects of the Brayton cycle [34]. Based on their results, the importance of accurate estimation of the performance of turbines in the simulation of loop performance was observed. Research has mostly concentrated on the design and numerical study of turbines above 50 kW scale.

Since a large amount of low- and medium-grade heat sources are unutilized, it requires a specific design for the S-CO₂ cycle and turbine [35]. In this case, a small volume flow rate and high axial force have become challenges to turbomachinery design and manufacture. The partial admission axial turbines provided prospective technological options for small-scale cycles with tens of kilowatt power output.

KIER tested the performance of the partial admission axial turbine in the lab scale S-CO₂ cycle and developed a numerical simulation of the partial admission axial turbine. They obtained that the maximum electric power is 10.3 kW [36]. Huang et al. constructed a transcritical CO₂ power cycle system and investigated the performance of a partial admission axial turbine designed for the cycle. The turbine generated a maximum power of 6.92 kW [37]. While previous studies demonstrated the feasibility of applying the partial admission axial turbine to the S-CO₂ cycle of tens of kilowatt power output scale, the turbine did not perform as expected. It is essential to improve the performance of partial admission axial turbines.

The separation and complex vortex occurring on the flow is inevitable, which can lead to large losses in the flow. The heat transfer and phase change that may occur during the flow will deteriorate the flow conditions and influence the aerodynamic performance of the turbine [38–40]. Given that the performance of turbines is crucial to the S-CO₂ cycle, methods are proposed for improvement on the performance of partial admission axial turbines. In the improvement methods, the loss model is a significant solution to the analysis of loss distribution on turbomachinery and optimizing the performance [41]. In the previous studies, the loss models of axial turbines are divided into two categories, one of which is based on enthalpy losses [42] and the other is based on pressure losses [43–45]. Ainley and Mathieson classified the axial turbine loss into the profile loss, the secondary loss, the tip leakage loss, and the trailing edge loss disregarding the effect of the Mach number and the outflow angle [43]. Dunham and Came amended Ainley and Mathieson’s model by taking the Reynolds number, the Mach number, and the aspect ratio of blades into account to adopt the small-scale axial turbine [44]. The loss model established by Kacker and Okapuu has been widely applied in the engineering field to predict turbine performance at both design and off-design conditions. This model takes into account the compressibility of the working fluid. The losses
taken into account are the profile loss, the secondary loss, the tip leakage loss, and the trailing edge loss in this model [45].

In the previous studies, most researchers focus on designing and numerical simulations of over 50 kW power scale $\text{S-CO}_2$ turbines. There is little concentration on the performance and manufacturing of axial turbines below the 50 kW scale. In addition, the majority of studies on the loss distribution of axial turbines apply to conventional working fluids while rare studies on the $\text{S-CO}_2$ axial partial admission turbines. It has resulted in the specific direction of improving the performance of $\text{S-CO}_2$ axial partial admission turbines.

The novelty and purpose of this paper are the investigation of a partial admission axial turbine with the 10 kW scale tested in an $\text{S-CO}_2$ cycle and to have completed numerical simulation. In response to the lack of current literature on turbines with the 10 kW scale, the operational conditions of the turbine were explored when it is in the efficient performance range. In addition, the purpose and novelty of this study are to analyze and investigate the loss causes and loss distribution by researching the flow field and the loss model for the $\text{SCO}_2$ turbines and to explore the factors contributing to the loss distribution, which could be the guide for the optimization of the $\text{S-CO}_2$ turbine.

2. Methods

In this part, a $\text{CO}_2$ power cycle system is built to test the $\text{S-CO}_2$ small-scale partial admission axial inflow turbine performance. The flow field in the turbine is a complex three-dimensional turbulent. The CFD design model is found to simulate the pressure and velocity distribution. Then, the appropriate loss model from previous studies is chosen to analyze the loss distribution of the turbine to optimize the turbine performance.

2.1. Test Bench. A schematic and a picture of the layout of the cycle test rig are given in Figures 1(a) and 1(b), respectively. The system is equipped with a turbine integrated with a high-speed electrical generator. In this cycle, the main parts of the system are the reservoir, the heat exchangers, the turbine, and the pump, with the working fluid used in this cycle being $\text{CO}_2$ at 93% purity. There are five heat exchangers including a preheater, recuperator, condenser, low-temperature heat exchanger, and high-temperature heat exchanger.

The variable speed pump coupled with an asynchronous generator is connected to an inverter that controls the rotational speed of the pump. The waste heat is recovered from the engine coolant and exhaust gas. The low-temperature $\text{CO}_2$ is drawn from the reservoir and pumped to the needed pressure by the pump. The compressed $\text{CO}_2$ is then delivered to the preheater, in which the waste heat of the coolant is absorbed by $\text{CO}_2$. After that, the fluid is divided into two lines after leaving the preheater. In one of the bypasses, a part of the $\text{CO}_2$ flowing into the low-temperature heat exchanger absorbs the heat from the low-temperature exhaust gas. The rest of the $\text{CO}_2$ is transmitted
to the recuperator placed in the other line. After converting, the whole CO$_2$ enters the high-temperature heat exchanger, in which the heat of high-temperature exhaust gas is carried by CO$_2$. The resulting high-pressure working fluid in a supercritical state is then expanded in the turbine where the thermal energy is converted into mechanical energy and then electrical energy through the high-speed generator. Expanded low-pressure CO$_2$ flows into the recuperator optimizing the heat unutilized. CO$_2$ will be further cooled down in the condenser. Finally, it flows back into the reservoir. The thermodynamic analysis, assumptions, and operating conditions range are listed in our previous study [46].

In the test, the pressure ratio is used to control the load on the turbine. To alter the pressure ratio, it varies the turbine inlet pressure or outlet pressure. Controlling the rotational speed of the pump is used to adjust the mass flow rate in order to vary the turbine inlet pressure. The turbine outlet pressure is modified by adjusting the coolant flow rate. For the system, we also armed a resistive load to consume the electricity produced by the generator and control the rotational speed by adjusting the level of resistance. We also developed a power analyzer for measuring output. A schematic of the load and power analyzer is shown in Figure 2.

2.2. Uncertainty Analysis. The uncertainty of the experimental bench is derived from the errors in the measuring instruments. In the experiment investigation, the inlet and outlet temperature of the turbine, the inlet and outlet pressure of the turbine, the flow rate of the turbine, and the output of the turbine are measured by thermal resistance, differential pressure transmitter, and Coriolis mass flowmeter, photovoltaic and power analyzer, respectively. The specification and precision of these sensors are listed in Table 1.

During the actual measurement, all measured values are directly measured by the sensors. Thus, each of the relative uncertainty is equal to the precision of the measuring instrument. In summary, the experimental bench is reliable in measurements.

2.3. Data Reduction. Isentropic efficiency and pressure loss coefficient are used to evaluate the performance of the turbine during the process. In particular, due to the particularity of the S-CO$_2$ axial turbine, the excessive pressure loss coefficient of the stator and rotor will increase the loss of power generation capacity and worsen the performance of

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G$ (kg/s)</td>
<td>0.5</td>
</tr>
<tr>
<td>$W$ (kW)</td>
<td>13.5</td>
</tr>
<tr>
<td>$T_0$ (K)</td>
<td>493.15</td>
</tr>
<tr>
<td>$P_0$ (MPa)</td>
<td>12.5</td>
</tr>
<tr>
<td>$\eta_{T-S}$ (%)</td>
<td>60</td>
</tr>
<tr>
<td>Rotational speed Ns (rpm)</td>
<td>40,000</td>
</tr>
<tr>
<td>$P_1$ (MPa)</td>
<td>6.4</td>
</tr>
</tbody>
</table>

Table 2: Design parameters of the turbine.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha_1$ (deg)</td>
<td>15</td>
</tr>
<tr>
<td>$\beta_1$ (deg)</td>
<td>70.8</td>
</tr>
<tr>
<td>$\alpha_2$ (deg)</td>
<td>-60</td>
</tr>
<tr>
<td>$\beta_2$ (deg)</td>
<td>60</td>
</tr>
<tr>
<td>$r$ (mm)</td>
<td>24</td>
</tr>
<tr>
<td>$h$ (mm)</td>
<td>4</td>
</tr>
<tr>
<td>$c$ (mm)</td>
<td>0.06</td>
</tr>
<tr>
<td>$b_x$ (mm)</td>
<td>10</td>
</tr>
<tr>
<td>Number of blades of the stator</td>
<td>1</td>
</tr>
<tr>
<td>Number of blades of the rotor</td>
<td>31</td>
</tr>
</tbody>
</table>

Table 3: Geometric parameters.

<table>
<thead>
<tr>
<th>Group</th>
<th>Grid number</th>
<th>Total</th>
<th>$\eta_{T-S}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Stator</td>
<td>Rotor</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>192,300</td>
<td>137,160</td>
<td>329,460</td>
</tr>
<tr>
<td>2</td>
<td>519,240</td>
<td>343,920</td>
<td>863,160</td>
</tr>
<tr>
<td>3</td>
<td>1,082,040</td>
<td>721,800</td>
<td>1,803,840</td>
</tr>
<tr>
<td>4</td>
<td>1,454,520</td>
<td>1,031,040</td>
<td>2,485,560</td>
</tr>
<tr>
<td>5</td>
<td>2,854,980</td>
<td>2,097,900</td>
<td>4,952,880</td>
</tr>
</tbody>
</table>

Table 4: Grid independence verification.
the turbine. Therefore, it is necessary to consider the isentropic efficiency and pressure loss coefficient performance of the turbine at the same time.

Isentropic efficiency $\eta_{is}$ is expressed as

$$\eta_{is} = \frac{W}{h_{in} - h_{out}},$$  

where $W$, $h_{in}$, and $h_{out}$ represent electric power, the inlet enthalpy, and the outlet enthalpy of the isentropic expansion, respectively.

Pressure loss coefficient $Y$ is expressed as follows:

For stator,

$$Y = \frac{p_{0}^* - p_{1}^*}{p_{0}^* - p_{1}},$$  

where $p_{0}^*$, $p_{1}^*$, and $p_{1}$ represent the total pressure for the stator of the inlet, the total pressure for the stator of the outlet, and the static pressure for the stator of the outlet, respectively.

For rotor,

$$Y = \frac{p_{2}^* - p_{2}}{p_{2}^* - p_{2}},$$  

where $p_{2}^*$ and $p_{2}$ represent the total pressure for the rotor of the outlet and the dynamic pressure for the rotor of the outlet, respectively.

2.4. The Structure of CO$_2$ High-Speed Axial Turbine. Referring to Figure 3, the prototype of the turbine is a turbogenerator. The structure of the CO$_2$ high-speed axial turbine is shown in Figure 4. The prototype is divided into an aerodynamic structure and a high-speed synchronous generator. Since the nature of CO$_2$, the mass flow rate is low. Therefore, the prototype was designed as a high-speed axial turbine for efficient operation. The axial overload will induce damage to the structure, so a partial admission stator was adopted in the aerodynamic structure. High-speed synchronous generator shares the shaft with the turbine. Three ball bearings were installed to prevent damage to the rotating shaft by excessive axial forces. When the turbine is operating, the rotor drives the shaft resulting in power output.

2.5. Three-Dimensional CFD Simulation Method. When a turbine is working, approximately the working fluid in the turbine is adiabatic. Referring to Figure 5, the working fluid enters the turbine via the inlet and flows into a stator through a passageway. The stator adjusts the direction and velocity of working fluid so that it flows into the rotor at the designed angle and velocity. In this part, A CFD model is found to depict the three-dimensional flow field. A CFD model also contributes to predicting the turbine performance and discovering the loss’s location. The axial turbine design parameters and geometric parameters are shown in Tables 2 and 3.

CFD focuses on revealing the flow inside the fluid, but the fluid interior has highly complex flow conditions, mainly caused by irregular pulsations inside the flow field.

In the simulation, the flow satisfies Equations (1)–(4). Continuity equation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \left( \rho \vec{v} \right) = 0.$$  

Energy equation:

$$\frac{\partial}{\partial t} \left( \rho \vec{v} \right) + \nabla \cdot \left( \rho \vec{v} \vec{v} \right) = -\nabla p + \nabla \cdot (\tau).$$  

Momentum equation:

$$\rho \vec{v} \cdot \nabla T = \nabla \cdot (k_{eff} \nabla T).$$
where $\bar{\tau}$ represents the stress tensor, which can be calculated as

$$\bar{\tau} = \mu \left[ (\nabla \vec{v} + (\nabla \vec{v})^T) - \frac{2}{3} \nabla \cdot \vec{v} \vec{I} \right].$$

(7)

The S-CO$_2$ small-scale partial admission axial turbine is simulated using ANSYS CFX. The first step is 3D blade profile generation which was established in the Bladegen module, according to the design parameters for the axial turbine. Then, the 3D blade model is imported into TurboGrid to generate the structured mesh of the single-flow passage. The grid independence analysis is shown in Table 4.
A grid of 2,485,560 nodes (1,454,520 for the stator and 1,031,040 for the rotor) is selected to accelerate calculation speed and ensure accuracy. The average $y^+$ is 1.21 for the stator and 1.41 for the rotor, which is in the reasonable range. The computational grid of the stator and rotor is presented in Figure 6.

In this study, the Reynolds-averaged Navier-Stokes equations (RANS) are used to simplify calculation problems of turbulence using CFX. For high-accuracy simulation, shear stress transport (SST) $k$-$\omega$ model is chosen to perform the steady-state numerical simulation. The CFX solver condition settings and the boundary conditions are shown in Table 5. CFD was conducted to compute performance at selected pressure ratios.

2.6. Small-Scale Partial Admission S-CO$_2$ Axial Turbine Loss Prediction Model. In this part, the Mach Numbers, pressures, and velocity of the inlet and outlet of the stator and rotor are calculated. The results and geometric parameters are applied in this model. The model results are presented as the loss of total pressure in the stator and rotor. Each loss is calculated as follows:

Profile loss has resulted from the boundary layer on the blade surface, wake, and turbulent dissipation.

$$Y_p = 0.914 \left( \frac{2}{3} K_p x_1 Y_{p(i=0)} + Y_{\text{shock}} \right),$$

$$Y_{\text{shock}} = 0.75 \left( M_{\text{in},H} - 0.4 \right)^{1.75} \left( \frac{\rho_H c}{\rho_m c} \right) \left( \frac{P_m}{P_{\text{out}}} \right) \frac{1}{1 - (1 + ((r - 1)/2)M_{\text{in}}^2)^{1/2}} \frac{1}{1 - (1 + ((r - 1)/2)M_{\text{out}}^2)^{1/2}}.$$

Secondary loss occurs due to the fluid viscosity and pressure gradient in the flow passage.

$$Y_s = 0.048 \chi_{AR} \left( 4(\tan \alpha_1 + \tan \alpha_2)^2 \left( \frac{\cos^2 \alpha_2}{\cos \alpha_m} \left( \frac{\cos \alpha_2}{\cos \beta_{ik}} \right) K_s, \right)^{1/2} \right),$$

$$\chi_{AR} = \begin{cases} 1 - 0.25 \sqrt{2 - h/c} \frac{h/c}{h/c}, & h/c \leq 2, \\ \frac{1}{h/c}, & h/c > 2, \end{cases}$$

$$K_s = 1 - \left( \frac{1}{h/c} \right)^2 (1 - K_p).$$

Trailing edge loss is caused by the vortex and the mixing of the wake with the main flow.

$$Y_{Te} = \frac{1 + ((\gamma - 1)/2)M_{out}^2 \left( (1/1 - \Delta E_{Te}) - 1 \right)^{\gamma - 1} - 1}{1 - (1 + ((\gamma - 1)/2)M_{out}^2)^{-\gamma - 1}} - 1. \ (10)$$

The viscous effect of flows in the tip clearance leads to tip leakage loss.

$$\Delta \eta_t = 0.93 \left( \frac{r_T}{r_m} \right) \left( \frac{1}{h \cos x_2} \right) \eta_{t,0} \Delta k. \ (11)$$

The total loss coefficient of a turbine is expressed as

$$Y = \chi_{Re} Y_p + Y_s + Y_{Ti} + Y_{Te}. \ (12)$$
3. Results and Discussion

3.1. Flow Field Analysis at Design Condition. When the maximum residuals observed were well below $10^{-5}$, all 3D simulations converged. The flow field in the small-scale axial turbine was analyzed in this part. The velocity field and streamlined distribution of the turbine single passage at different spans are shown in Figure 7. At the 5% span of the stator, the working fluid flows smoothly through the passages, and the velocity increases in the meridional direction uniformly. In the stator and rotor, there is no obvious separation of the streamlines. At 50% span of the stator and rotor, there is no significant separation occurring. At 90% span of the stator, the streamline does not separate, while

whereas

$$
\chi_{Re} = \begin{cases} 
\left(\frac{Re}{2 \times 10^5}\right)^{-0.4}, & \text{Re} \leq 2 \times 10^5, \\
1.0, & 2 \times 10^5 > \text{Re} > 10^6, \\
\left(\frac{Re}{10^6}\right)^{-0.2}, & \text{Re} > 10^6.
\end{cases}
$$

Figure 10: Variation in the power at different relative rotational speeds under 1.7-2.0 total-to-static pressure ratio.

Figure 11: Variation in the isentropic efficiency at different relative rotational speeds under 2.0 PR.

Figure 12: Comparison in the power at different relative rotational speeds under 1.7 total-to-static pressure ratio.

Figure 13: Comparison in the isentropic efficiency at different relative rotational speeds under 2.0 PR.
at 90% span of the rotor, flow separation occurs on both the high-pressure and the suction sides caused by tip leakage flow. It is a substantial reason why the turbine performs below expectation. Tip leakage flow is induced by the pressure difference between the high-pressure side and the suction side. The pressure difference is generated at the radial tip clearance when the blade rotates. At all spans, the velocity of the fluid drops sharply close to the blade surface and endwall. It occurs simultaneously with the trailing edge of the suction surface. The reason for this phenomenon is that a small aspect ratio results in boundary layer separation in blade surfaces, endwall, and the trailing edge of the suction surface. It deteriorates the turbine performance.

The temperature, Mach number, and pressure contours at 50% span are presented in Figure 8. The temperature drops at a uniform gradient in the stator. But there is an irregular decline of temperature in the rotor. The temperature decreases along the flow passages. A temperature explosion occurs at the trailing edge of the stator and rotor. For the pressure contours, it decreases from the inlet of the stator until it drops to a minimum at the outlet of the rotor. The abrupt pressure drop occurs at the trailing edge of the stator and rotor. The same phenomenon occurs on the suction side at 2/3 chord length in the rotor. For the Mach number contours, the fluid in the turbine is subsonic in the flow passage, the Mach number increases gradually from the inlet to the outlet in the stator. There is a sudden drop in the inlet of the rotor caused by intershaft leakage, and then, the Mach number gradually increases until the outlet. A low Mach number boundary layer is formed around the stator and rotor blades. Meantime, the Mach number of the trailing edge of the stator and rotor suddenly decreases. The reason for this phenomenon is that a small aspect ratio results in flow loss near endwalls.

The static pressure along the stator and rotor at different spans is illustrated in Figure 9. For the stator blade, the static pressure distributions are almost the same at different spans. The static pressure distributions are also regular along the stator blade. Because the stator blade is designed as a straight type. There is a pressure drawdown at the suction surface’s leading edge, led by secondary flow near the endwall. Meantime, pressure fluctuation occurs on the trailing edge of the suction surface because of the boundary layer. In addition, the loading of the three spans is centralized after streamwise region 0.75. Pressure loss resulting from intershaft clearance appears at the outlet of the stator and the inlet of the rotors. Similar to the stator, the pressure distributions are almost identical in the different rotor blade spans. The pressure drop gradient is uniformly distributed in the pressure distributions of three spans. It illustrates that the load distribution is average on the rotor blade, which contributes to better
turbine performance. For the rotor blade, there is pressure fluctuation on the suction surface at 50% span different from other spans. The loading of the 5% span and the 50% span is almost the same, but the 90% span is obviously different on the suction surface.

3.2. Off-Design Performance Analysis. In order to research the off-design performance of the S-CO₂ part admission inflow axial turbine and analysis of loss composition, the 3D simulation performance is compared with the experimental results, and 1D prediction of loss is performed. The output form of the 3D simulation results is shaft work, while the experimental output form is electrical power. The shaft work is transformed into electrical power through the drive shaft, generator, and wires. The transmission efficiency of the shaft is usually set as 0.9 [47]. The efficiency of the generator is 0.93 [48]. The traverse loss is calculated by electrical resistance and electric current. In this part, the relative rotational speed is applied to off-design performance analysis which is the ratio of actual value to design value. 3D simulation results converted will be compared with the experimental results. Variation in the power at different relative rotational speeds under 1.7-2.0 total-to-static pressure ratio is shown in Figure 10. The experimental results are mostly consistent with the 3D simulation results. With the speed increasing, the power trend increases first and then decreases. The turbine is operated in the lower speed range and has low power output at the low PR. The turbine performance is reversed at the high PR. The turbine is only operated at the relative rotational speed of the range from 0.45 to 0.85 in the 1.7 and 1.8 PR. The difference at the high PR may be due to larger flow rates than low PR. Variation in the isentropic efficiency at different relative rotational speeds under 2.0 PR is shown in Figure 11. The experimental results at different relative rotational speeds are in good agreement with the 3D simulation values. The power output efficiency varies from 30% to 45%. The experimental results are marginally less than the 3D simulation values due to losses underestimated when mechanical work is converted to electrical output. There is a steady turbine performance in off-design conditions.

In most cases, S-CO₂ power systems will perform under off-design conditions due to different operational conditions. In order to operate the turbine in the high output performance range for application in low-degrade heat sources, the pressure ratios and rational speeds of turbines are required to be sustained at a high level. Specifically, when the system load is adjusted, it is possible to guide optimizing
output performance by modifying the pressure ratio and rational speeds.

Compared with our previous study [37], the output of the optimized turbine is remarkably improved. The output and efficiency of the comparison are shown in Figures 12 and 13. At 1.7 PR, the output power is increased by 3.1 kW on average. Simultaneously, the efficiency is increased by 9.94% on average. It is mainly due to the improved sealing performance of the turbine which decreased the occurrence of leakage.

The temperature, Mach number, and pressure contours at 50% span at the rotational speed of 30,000 rpm and 50,000 rpm are presented in Figures 14 and 15. At the rotational speed of 30,000 rpm, efficiency and power decline are compared with the rotational speed of 40,000 rpm. With the rotational speed decreasing, the pressure distribution and its drop gradient remain essentially unchanged at the stator, and the dramatic fluctuation of pressure occurs on the suction surface on the rotor compared with the design point. Especially in the trailing edge of the rotor, the pressure has violently dropped resulting in incomplete energy conversion of the working fluid which causes loss to increase. In the temperature and Mach number contour, the same phenomenon occurs on the trailing edge of the rotor.

At the rotational speed of 50,000 rpm, efficiency and power increase compared to the rotational speed of 40,000 rpm. With the rotational speed increasing, the fluctuation of pressure occurring on the suction surface on the rotor has been weakened. The velocity of flow increases with the rotational speed increasing leading to a larger Mach number. So the drop gradient in the trailing edge of the rotor is improved. The pressure distribution is the same as at 40,000 rpm in the stator. Meantime, uneven drop gradient is improved in the temperature and Mach number contours compared with 40,000 and 30,000 rpm. At the rotational speed of 50,000 rpm, the degree of loss is reduced.

3.3. Loss Distribution Analysis. In this section, validation of the accuracy of the loss model established by Kacker and Okapuu and loss distribution was analyzed. The loss of total pressure is expressed in terms of the pressure loss coefficients of the stator and rotor to analyze the loss distribution.

Figure 16: Stator, rotor, and total pressure loss coefficient at different relative rational speeds under 2.0 PR.
Figure 16. As can be seen from the pressure loss coefficient of the stator and rotor, it can be observed that the results of the model are extremely simulated with CFD under 2.0 PR which is designed pressure ratio. And the maximum difference is only 4% in the stator. Also, the maximum difference is only 6% in the rotor. The pressure loss coefficient of the stator varies from 0.35 to 0.4. The majority of the pressure loss coefficient of the rotor is in the range of 0.1. In the stator and rotor, most values of the pressure loss coefficient of CFD are slightly higher than in the model. In the total pressure loss coefficient comparison, most values of the pressure loss coefficient of the model are generally consistent with the experimental values. The pressure loss coefficient of the stator is significantly higher than the rotor. As shown in the results, the model can be adapted to the tens of kilowatt partial admission S-CO₂ axial turbines.

Loss distribution at different relative rational speeds under 1.7–2.0 PR is shown in Figure 17. By analyzing the loss distributions, it can be drawn that the obvious proportion of the total losses is tip clearance loss, profile loss, and trailing edge loss. At the designed flow rate, tip clearance loss accounts for a significant portion. This can be explained by the fluid through the rotor blade producing a pressure difference between the suction side and the high-pressure side resulting in a leakage flow at a low mass flow rate. The tip clearance loss was predicted to slightly decline as the rotational speed level increased. This can be attributed to the increase in the speed resulting in the reduction of differential pressure at the rotor blade. The tip clearance losses at different pressure ratios are mostly the same depending on the speed and tip clearance size. The variation of profile loss is not significantly related to the variation in pressure ratio and speed. This can be explained that the profile loss mainly depends on the parameters of the blades. The fluctuation range of trailing edge loss is small. Secondary loss has only a rather small account of the total losses.

Relatively high-pressure ratios can be reached with slightly lower pressure loss. With the rational speed increasing, the pressure loss coefficient decreases when the turbine operates at higher pressure ratios than 1.7. This is mainly caused by the decrease in the profile loss and tip clearance loss with high-pressure ratios increasing rational speed. This
can be mainly explained by the effect of higher stator and rotor outlet velocities with high-pressure ratios. The improved design speed and pressure ratio provide a better output performance of the turbine.

4. Conclusion

The high-power density of S-CO₂ results in high rotational speeds for 10 kW scale turbomachine designs. In this paper, a 10 kW scale partial admission axial turbine is tested with an S-CO₂ power cycle for recovering engine waste heat. Simultaneously, numerical simulations were completed. The turbine is designed to generate a power output of 13.5 kW at 40,000 rpm and 2.0 pressure ratio.

The maximum electric power measured reached 9.66 kW when the turbine rotated at 38,000 rpm and 2.0 pressure ratio. At the 10 kW scale, the design mass flow rate of turbines is lower compared to high-power turbine designs. The performance of turbine output is based on pressure ratio and rotational speed. The turbine has superior output performance with operation at higher pressure ratios and rational speed.

For the stator and rotor blade, the static pressure distributions at different spans are consistent. The stator had obvious aft-loaded characteristics which relieve the separation phenomenon and reduce the separation loss effectively. The rotor had obvious average loading characteristics which can facilitate the improvement of the output power of the turbine. The distributions of the flow field parameters were basically reasonable. There was a slight separation in the turbine. The phenomena of separation are primarily attributed to the irregular pressure distribution of the rotor, which resulted in the flow loss. The leakage phenomenon developed at the clearance tip, which contributed to the inefficient performance of the turbine.

The loss model established by Kacker and Okapuu is adopted to find out the loss distribution of the scale part admission axial turbines at the level of tens of kilowatts. With low turbine design powers, the tip clearance loss is observed as the main loss, which accounts for nearly 50% of the total pressure losses.

In future work, it is suggested that a comprehensive of the effects of power scale and design-specific speeds on S-CO₂ turbines should be investigated. In addition, it is important to develop turbine loss models as design guidelines for improved designs for S-CO₂ turbines and extended potential application to other medium-temperature heat sources, including solar, industrial waste heat, and geothermal. Further, it is recommended that the optimization of the stator and rotor be studied more in detail and applied.

Nomenclature

Symbols

W: Output power (kW)
P: Pressure (MPa)
G: Mass flow rate (kg/s)
T: Temperature (K)

Ns: Design rotational speed (rpm)
N: Operational rotational speed (rpm)
Ma: Mach number
h: Blade height (m)
c: Clearance height (m)
bx: Axial chord length (m)
Re: Reynolds number
Yₚ: Profile loss
Yₛ: Secondary loss
Yₚₑ: Trailing edge loss
Yₜₑ: Tip leakage loss
r: Radius
K: Mach number correction factor.

Greek Symbols

α: Absolute flow angle (deg)
β: Relative flow angle (deg)
η: Efficiency.

Subscripts

0: Turbine inlet
1: Turbine outlet
in: Stator/rotor inlet
out: Stator/rotor outlet
T-S: Total-to-static
H: Hub
T: Shroud
m: Mean.

Abbreviations

CFD: Computational fluid dynamics
PR: Pressure ratio
S-CO₂: Supercritical carbon dioxide
SNL: Sandia National Laboratories.

Data Availability

Data will be made available on request.

Conflicts of Interest

The authors declare that they have no competing interests.

Authors’ Contributions

Hua Tian was responsible for the validation, formal analysis, data curation, writing the original draft, investigation, and visualization. Xin Lin was responsible for the validation, formal analysis, data curation, and visualization and wrote, reviewed, and edited the manuscript. Ligeng Li was responsible for the conceptualization, resources, and supervision and wrote, reviewed, and edited the manuscript. Xianyu Zeng was responsible for the resources and supervision and wrote, reviewed, edited the manuscript. Yurong Wang was responsible for the resources and wrote, reviewed, and edited the manuscript. Lingfeng Shi was responsible for the resources and wrote, reviewed, and edited the manuscript. Xuan Wang was responsible for the conceptualization,
resources, and supervision and wrote, reviewed, and edited the manuscript. Xingyu Liang was responsible for the resources and supervision and wrote, reviewed, and edited the manuscript. Gequn Shu was responsible for the conceptualization, resources, and supervision and wrote, reviewed, and edit the manuscript.

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