ABSTRACT

As an important component for a pre-swirl air supply system, the performance of the receiver hole directly affects the quality of cooling air supplied to turbine rotor blades of an aeroengine. For a high-radius pre-swirl system with a difference in radius between the pre-swirl nozzle and the supply hole, numerical research and experimental verify were done to research the design problem of the receiver hole for the system. The results of numerical study show that the runway shaped receiver hole and the drilling shaped receiver hole have little difference for the system performance, and the vane shaped receiver hole has the best system performance. The increase of the length-to-diameter ratio slightly improved the system performance. Meanwhile, the circumferential angle of the receiver hole can significantly reduce the system dimensionless specific power consumption of the pre-swirl air supply system. Compared with the system using the runway shaped receiver hole, the system temperature drop efficiency of the system using the vane shaped receiver hole can increase by 14.6% under the design condition, up to 0.63. And the system dimensionless specific power consumption of it can decrease by 44.6%.

Key words: Pre-swirl air supply system; Receiver hole; Numerical simulation

1. INTRODUCTION

The pre-swirl air supply system is a system that can supply sufficient high-quality cooling air to turbine blades, generally including pre-swirl nozzles, a pre-swirl cavity, receiver holes, a cover-plate cavity, supply holes, inner seal and outer seal. A well-designed pre-swirl system not only ensures the normal working of turbine blades, but also effectively extents the working life of the turbine blades. As an important part of the pre-swirl system, the performance of receiver holes is concerned by more and more scholars.

The receiver hole can be regarded as a rotating hole with small aspect ratio. Therefore, the evaluation method of the rotating hole can be applied to the receiver hole. The discharge coefficient can also indicate the flow resistance of the receiver hole, which is affected by the geometry, aspect ratio, incidence angle and rotating speed (Carlen, 1965; Wittig, et al., 1996; Maeng, et al., 1998).

For meeting the needs of engineering applications, and evaluating the influence of the receiver hole on the performance of the pre-swirl air supply system, it is necessary to research the influence of the receiver hole on the pre-swirl system. Chew et al. measured the relative total temperature at the entrance of the receiver hole, and verified a simple one-dimensional flow model for complete mixing in a pre-swirl cavity based on experimental measurement data (Chew, et al., 2003). Bricaud et al. (2007) measured the field between the nozzle out and the receiver hole entrance by 3D PIV measurement technology, and the receiver hole efficiency. Liu et al. (2019) researched the influence of the rotating speed, system pressure ratio and Reynolds number on the system temperature drop of the pre-swirl system using straight
round-type receiver hole. As a low-cost and high-efficiency research tool, numerical simulations have become more and more prominent in design and improvement research. In order to ensure the reliability of the numerical calculation results, Javiya et al. (2010; 2011) studied the influence of the rotor-stator interface, turbulence models, and unsteady-state effects on the numerical calculation results. Jarzombek et al. (2006; 2020) studied the influence of the radius position of the nozzle relative to the receiver hole, the edge shape, the circumferential inclination angle, the area ratio and the number of the receiver hole on the flow temperature drop of the pre-swirl air supply system. Lee et al. (2019) optimized the receiver hole by using a multi-objective genetic algorithm to improve the performance of the pre-swirl system. Zhu et al. (2010) found that the swirl ratio at the receiver hole inlet is a key factor, affecting the temperature drop efficiency and pressure loss coefficient. Chen et al. (2018; 2020; 2019; 2020) studied the influence of the radius position of the nozzle relative to the receiver hole, the edge shape, the circumferential inclination angle, the area ratio and the number of the receiver hole on the flow temperature drop of the pre-swirl air supply system. Lee et al. (2019) optimized the receiver hole by using a multi-objective genetic algorithm to improve the performance of the pre-swirl system. Lei, et al (2020) studied the temperature drop characteristics of a pre-swirl system.

In the past researches, the receiver hole shape researched was always the drilling shaped hole, such as the circular hole, while the vane shaped receiver hole and the runway shaped receiver hole are rarely studied, and the design and improvement method of the receiver hole is rarely reported for a high-radius pre-swirl system with a radius difference between the pre-swirl nozzle and the supply hole. For the system, numerical simulations and experimental verify used to improve the performance of different types of the receiver hole. The research results will provide a way for designing and improving the receiver hole to effectively improve the performance of the receiver hole.

2 METHODS

2.1 Parameters definition

The fluid zones of a typical high-radius pre-swirl system are shown in Fig. 1. The system is usually composed of the pre-swirl nozzle, the pre-swirl cavity, the receiver hole, the cover-plate cavity and the supply hole.

The system pressure ratio is defined as the ratio of the total pressure $p_0^*$ at the system inlet to the static pressure $p_2$ at the supply hole outlet,

$$\pi = \frac{p_0^*}{p_2}$$  \hspace{1cm} (1)

Meanwhile, the static pressure ratio can be used to describe the dimensionless pressure at critical sections, which can be defined as

$$\pi = \frac{p_i}{p_2}$$  \hspace{1cm} (2)

where $p_i$ is the static pressure at the specific section; $i$ can be 0, 1, 1’, 1’’ or 2, representing the system inlet, the nozzle outlet, the receiver hole inlet, the receiver hole outlet and the supply hole outlet (or called the system outlet), respectively.

Similar to the static pressure ratio, the temperature ratio $\tau$ is an indicator suggesting the temperature of the section,

$$\tau_i = \frac{T_{rel,i}}{T_0}$$  \hspace{1cm} (3)
where $T_{rel,i}^*$ is the relative total temperature of the section $i$ (the meaning of $i$ is the same as that in the static pressure ratio). When $i$ is equal to 2, $T_2^*$ is the system temperature ratio.

The rotating Mach number is a dimensionless parameter describing the rotating speed, which is defined as

$$Ma_\varphi = \frac{\omega b}{\sqrt{k R_g T_0^*}}$$  \hspace{1cm} (4)

Where $\omega$ is the angular speed, $b$ is the radius of the turbine disk, $k$ is the specific heat ratio, $R_g$ is the gas constant, $T_0^*$ is the system inlet total temperature.

The relative total temperature is the temperature that can be directly measured on the rotating parts. It can be defined by

$$T_{rel}^* = T + \frac{V_a^2}{2c_p} + \frac{V_r^2}{2c_p} + \frac{(V_\varphi - \omega r)^2}{2c_p}$$  \hspace{1cm} (5)

where, $T$ is the static temperature of the air, $V_a$, $V_r$, and $V_\varphi$ are the axial velocity, radial velocity, and circumferential velocity of the air, respectively, $c_p$ is the specific heat of the air, $r$ is the radius. Where the total temperature and the relative total temperature can be expressed as

$$T_{rel}^* = T^* - \frac{(\omega r)^2}{2c_p}(2\beta - 1)$$  \hspace{1cm} (6)

where $\beta$ is the airflow swirl ratio ($\beta = V_\varphi/\omega r$), which is the ratio of the airflow circumferential velocity to the turbine disk speed at the same radius position.

The system temperature drop can directly indicate the air supply temperature, the greater the system temperature drop, the lower temperature which the turbine blade can feel. The parameter is defined as,

$$\Delta T = T_0^* - T_{rel,2}^*$$  \hspace{1cm} (7)

where $T_0^*$ and $T_{rel,2}^*$ are total temperature at the system inlet and the relative temperature at the supply hole outlet, respectively.

The mass flow rate ratio is defined as the ratio of air supply mass flow rate $m_2$ to the nozzle critical mass flow rate $m_{cr}$,

$$q(\lambda) = \frac{m_2}{m_{cr}}$$  \hspace{1cm} (8)

where the nozzle critical mass flow rate can be expressed as

$$m_{cr} = 0.0404 \frac{p_{cr} A_1}{\sqrt{T_0^*}}$$  \hspace{1cm} (9)

where, $A_1$ is the nozzle throat area. The critical mass flow rate is the maximum mass flow rate at the specific system inlet total temperature and total pressure. In the experimental process, the air supply mass flow rate will be equal to the design value under the condition of ensuring the mass flow rate ratio and the system inlet total pressure equal to the design operation.

The discharge coefficient is a critical non-dimensional parameter to evaluate the flow loss magnitude. The discharge coefficient of the receiver hole usually can be defined as,

$$C_{D,R} = \frac{m_2}{m_{rel,rel}} = \frac{m_2}{\frac{p_{rel,rel} A_R}{R_g T_{rel,rel}^{0.5}} \sqrt[k-1]{2k \left((p_{rel,rel}^{* / p_T})^{2/k} - (p_{rel,rel}^{* / p_T})^{(k+1)/k}\right)}}$$  \hspace{1cm} (10)

Where $p_{rel,rel}^{*}$, $T_{rel,rel}^{*}$, $A_R$ and $p_T$ are respectively the relative total pressure at the receiver hole inlet, the relative total temperature at the receiver hole inlet, the area of the receiver hole outlet, and the static pressure at the receiver hole outlet.

The system temperature efficiency is a dimensionless parameter to evaluate the flow loss in the pre-swirl system, which is the ratio of the system temperature drop to the system ideal temperature drop,

$$\eta = \frac{\Delta T}{\Delta T_{id}} = \frac{T_0^* - T_{rel,2}^*}{T_0^*(1 - \pi^{(1-k)/k})}$$  \hspace{1cm} (11)

As well as the temperature drop efficiency, the system specific entropy increment is a more direct parameter to achieve the flow loss of the system, which is defined as
\[ \Delta s = c_p \ln \left( \frac{T_2}{T_0} \right) - R_g \ln \left( \frac{p_2}{p_0} \right) \]  
where \( T_2 \) and \( T_0 \) are the static temperature at the system outlet and the static temperature at the system inlet, respectively; \( p_2 \) and \( p_0 \) are the static pressure at the system outlet and the static pressure at the system inlet, respectively.

The system dimensionless specific power consumption is also concerned by the pre-swirl system designers, as same as the system temperature drop,

\[ \Phi = \frac{c_p(T_2^* - T_0^*)}{(\omega)^2/2} \]  
where \( T_2^* \) is the total temperature at the system outlet. When \( \Phi \) is positive, it suggests that walls do work on the gas.

2.2 Structure design

2.2.1 Experimental rig

The structural sketch of the pre-swirl system is drawn in Fig. 2, which has a radius difference between the pre-swirl nozzle and the supply hole. The airflow first flows into the pre-swirl nozzle, and then expands and accelerates through the pre-swirl nozzle to produce the high-quality cold air. The cold air sequentially passes through the pre-swirl cavity, the receiver hole, the cover-plate cavity and the supply hole, and finally flows through the outlet chamber. For the pre-swirl system, the number of pre-swirl nozzles, receiver holes and air supply holes are all 48. The radial position of the supply hole is \( r_2 \), the length is \( l/r_2 = 0.127 \), the shape of the supply hole is round, and the total outlet area is \( A_2 \). The radial position of the receiver hole given is \( r_R/r_2 = 0.974 \), and the outlet area of the receiver hole is \( A_R/A_2 = 1.086 \). The radial position of the pre-swirl nozzle using vane shaped hole nozzle is \( r_1/r_2 = 0.961 \), the nozzle height of the pre-swirl nozzle is \( h/r_2 = 0.25 \), the outlet angle of the vane shaped hole nozzle is \( \alpha_N = 76.7^\circ \), and the total throat area of the pre-swirl nozzle is \( A_1/A_2 = 0.312 \). The vane shaped nozzle profile is drawn in Fig. 3.

![Fig. 2 Structural Sketch of the Pre-swirl System](image)

![Fig. 3 The Vane Shaped Hole Nozzle](image)

2.2.2 Design and improvement of the receiver hole

For the pre-swirl system with a radius difference between the pre-swirl nozzle and the supply hole, the drilling shaped receiver hole parallel to the axis was designed according to the traditional design ideas, as shown in Fig. 4. As illustrated in Fig. 5, the runway shaped receiver hole also was designed to contrast with the drilling shaped receiver hole, which has been applied to some aeroengines in recent years. The influence of both the inclination angle and the length-to-diameter ratio was also concerned in design, so it is necessary to design the drilling shaped receiver hole with an inclination angle and the higher length-to-diameter ratio respectively, as given in Fig. 6 and Fig. 7 respectively. They were considered for the influence of both the incidence angle at the receiver hole inlet and the radius difference between the pre-swirl nozzle and the supply hole. Therefore, a drilling shaped receiver hole with both a circumferential angle and a radial angle was designed, as drawn in Fig. 8. In order to further improve the performance of the receiver hole, a detailed aerodynamic design was performed for the receiver hole, considering the airflow angle of both the inlet and outlet of the receiver hole. The receiver hole by detailed aerodynamic design is called the vane shaped receiver hole, which is shown in Fig. 9. In order to distinguish every receiver hole conveniently, the drilling shaped receiver hole, the runway shaped receiver hole, the drilling shaped receiver hole with an inclination angle, the drilling shaped receiver hole with a higher length-to-diameter ratio, the drilling shaped receiver hole with both a circumferential angle and a radial angle, and the vane shaped receiver hole are abbreviated as DSRH, RSRH, aDSRH, hDSRH, bDSRH, and VSRH.
The parameters about the geometrical structure of the receiver hole are listed in Table 1. The radial positions of both the DSRH and the RSRH are higher than the average of both the pre-swirl nozzle radius and the supply hole radius. The
diameters for the DSRH, aDSRH and hDSRH are $d/r=0.038$. The differences for them are the inclination angle and the length-to-diameter ratio. The hDSRH has a greater length-to-diameter ratio up to 2.011, increasing by 2.7 times compared with that of the DSRH. For the aDSRH, hDSRH, bDSRH and VSRH, the radial angles between the receiver hole centerline and the radial direction are all 65°, whose extension line passes through the pre-swirl nozzle outlet center (B point position in Fig 6~Fig. 9) and the supply hole entrance center (C point position in Fig 6~Fig. 8). Meanwhile, the circumferential angle for the bDSRH is 55.6°. The outlet areas for the 5 different receiver holes are all equal to $A_{r}/A_{2}=1.086$.

2.3 Numerical method

To obtain the performance of these receiver holes in the pre-swirl system, numerical simulations were carried out. For saving computing resources and time, the periodic boundary is adopted, 1/48 of the full-annulus was accepted as the computational domain. The computational domain is mainly consisting of the inlet chamber, the pre-swirl nozzle, the pre-swirl cavity, the receiver hole, the cover-plate cavity and the supply hole. The pre-swirl systems using the DSRH, RSRH, aDSRH, hDSRH, and VSRH respectively were established, as successively illustrated in Fig. 10(a)~Fig. 10(f). The geometry parameters are consistent with value listed in Table 1. In the following pictures, the surfaces colored blue are the static parts, the surfaces colored red are rotational parts, the surfaces colored green are periodic surfaces, and the surfaces colored brown are inlet or outlet surface. Date exchange between the static domains and rotational domains of the pre-swirl system was performed by the rotor-stator interface in numerical simulations. As shown in Fig. 10, the rotor-stator interface is located in the middle of the pre-swirl cavity in the axial direction.

![Fig. 10 Computational Models of the Pre-swirl Systems](image-url)
Table 2 gives two operating conditions for the pre-swirl system. For the operating condition, the rotating Mach numbers are respectively 0.54 and 0.53, the system pressure ratio is 1.6, and the system inlet total temperature is 298K. The operating condition was used as calculation condition to evaluate the performance of these receiver holes.

<table>
<thead>
<tr>
<th>System pressure ratio</th>
<th>Rotating Mach number $Ma_\phi$</th>
<th>System inlet total pressure $p_0$ /kPa</th>
<th>System inlet total temperature $T_0^*$ /K</th>
<th>System outlet static pressure $p_2$ /kPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.6</td>
<td>0.54</td>
<td>160</td>
<td>298</td>
<td>100</td>
</tr>
</tbody>
</table>

Considering that the flow Reynolds number can reach more than 1,000,000 which belong to the high turbulent region, so the standard k-epsilon model was adopted. The enhanced wall treatment was also applied to solve the flow in the boundary layer. What’s more, the influence of the viscous heating was considered in the model. The SIMPLE scheme was employed to solve the problem. The heat transfer between the walls and the airflow was ignored because of the small exchange in actual physical system. Therefore, all walls were set to be adiabatic and no-slip. Both the inlet and outlet of the system were set to as pressure boundary conditions. While the outer seal was set as mass flow outlet in order to adjust the air supply mass flow rate satisfying the design value. Furthermore, the fixed rotor phase method was used to calculate the rotational region. The fluid working medium selected the ideal gas with variable specific heat, thermal conductivity and viscosity. The detailed solution process is conducted by Fluent 16.0.

Fig. 11 shows the mesh generation of the pre-swirl system using the RSRH. The hexahedral structured mesh is adopted. The mesh near the wall has increased the mesh density in order to meet the need of near-wall functions. The wall Yplus for most of the wall region is near 1, the maximum value is less than 10. The mesh independency was studied for reducing the influence of mesh density on the computation results adopting the numbers of the grid nodes about 664,000 nodes, 1,025,000 nodes, and 1,552,000 nodes. The results of grid independency study were illustrated in Table 3.

For the operating condition of the system pressure ratio 1.6, increasing the number of nodes from 0.664 million to 1.55 million, the maximum deviation for the discharge coefficient of the receiver hole $C_{DR}$, the system temperature drop $\Delta T$ and the specific system work consumption $w$ is less than 0.5%, as shown in Table 3.

<table>
<thead>
<tr>
<th>Number of nodes</th>
<th>$C_{DR}$</th>
<th>$\Delta T$ /K</th>
</tr>
</thead>
<tbody>
<tr>
<td>$0.664 \times 10^6$</td>
<td>0.541</td>
<td>20.4</td>
</tr>
<tr>
<td>$1.03 \times 10^6$</td>
<td>0.539</td>
<td>20.4</td>
</tr>
<tr>
<td>$1.55 \times 10^6$</td>
<td>0.539</td>
<td>20.3</td>
</tr>
</tbody>
</table>

![Fig. 11 Grid Generation of the System with the RSRH](image)

The grid density of the pre-swirl systems with other receiver holes is same as that of the system with the RSRH.
Therefore, the grid generation of the systems with other receiver holes will no longer be displayed in this paper.

2.4 Experimental method

2.4.1 Experimental rig

To ensure the reliability of numerical calculations and verify the performance of the receiver hole, the experiments and measurements are performed on the pre-swirl air supply system test rig of Northwestern Polytechnical University. The main components of the test rig are shown as in Fig. 12. It is composed of a compressor, air tanks, intake and exhaust valves, computers, motors and other equipment. The maximum pressure supplied by the compressor is 1.6MPa, and the maximum continuous mass flow rate is 2.5kg/s. The test rig driven by a high-speed motor whose maximum rotating speed can achieve 10,000rpm, is cooled by the circulating water. The pressure, the temperature and the mass flow rate measured on the stator can be collected in real-time. Whereas the pressure and the temperature measured on the rotor will be stored in the data recorder. The air supply mass flow rate $m_2$ can be indirectly achieved by subtracting the sealing mass flow rate $m_1$ from the pre-swirl nozzle mass flow rate $m$. When the experiment is finished, the data in the data recorder will be imported into the computer for post-processing. A more detailed introduction about the experiment can read the literature written by Wu, et al. (2019).

Fig. 12 Schematic Diagram of the Test Rig

A schematic diagram of the high-radius pre-swirl system test rig is given in Fig. 13. The system adopts the method of central air intake, as shown in Fig. 13. The pre-swirl system test rig mainly consists of an inlet casing, a pre-swirl nozzle inner and outer ring, a cover-plate disk, a turbine disk and an outlet casing. These casings and disks make up the cavities and holes needed in the experiment. The air flowing into the inlet chamber, passes through in sequence the pre-swirl nozzle, the pre-swirl cavity, the receiver hole, the cover-plate cavity, the supply hole, and the outlet chamber, and finally is discharged into the atmosphere. A picture of the installed test rig is illustrated in Fig. 14. Considering the purpose of experimental researches, the pre-swirl system with the RSRH was manufactured, as displayed in Fig 15.

Fig. 13 Assembly Drawing of the High-Radius Pre-swirl System
2.4.2 Measurement and calibration

Temperature and pressure measuring points were arranged on stationary and rotating parts of the test rig. A total temperature measuring point and two total pressure measuring points were located in the inlet chamber; four static pressure measuring points were set at the pre-swirl nozzle outlet; two static pressure measuring points and three relative total temperature measuring points were placed at the receiver hole outlet; two static pressure measuring points were situated at the supply hole inlet; two static pressure measuring points, two relative total temperature measuring points and a total temperature measuring points were set at the system outlet. Measurement work was carried out on the rotor for the static pressure and the relative total temperature at the receiver hole outlet and the supply hole inlet and outlet. All temperatures were measured by K-type thermocouples calibrated by a PT1000 platinum resistance. The maximum uncertainty of thermocouples is less than ±1K. All pressure on the stator was measured by a PSI pressure scanner with 16 channels and the maximum uncertainty less than ±0.05%. All pressure on the rotor was measured by Kulite™ tiny transducers. In addition, the mass flow rates at the system inlet and the outer seal outlet were measured by high-precision orifice meters whose maximum uncertainty less than ±1.0%. The air supply mass flow rate $m_2$ can be achieved by subtracting the sealing mass flow rate $m_1$ from the pre-swirl nozzle mass flow rate $m$, as shown in Fig. 11. The method detailed measurement points layout can refer to the reference written by Wu, et al. (2019).

3 RESULTS AND DISCUSSION

As illustrated in Fig. 16, the numerical results (CFD) of the system temperature drop ($\Delta T$) with the rotating Mach number ($Ma_\phi$) were compared with the experimental results (EXP) to verify the reliability of numerical simulations. The circles colored green represent the numerical results, and the blocks colored red represent the experimental results. Compared with the experimental results, the maximum deviation of the system temperature drop by numerical simulation is less than 2.0%. The main reasons caused the deviation are the simplifications of the CFD model based on the physical model. However, the calculation accuracy has met the design and improvement needs.
As given in Fig. 17, the deviation of the discharge coefficient is less than 2.2% for the DSRH and the RSRH, both of which are about 0.53. This shows that the shape of the receiver hole has little influence on the performance of the receiver hole. Compared with the DSRH, the discharge coefficient of the aDSRH has a significant improvement, increasing by 7.7%, which is greater than the increasing amplitude 4.0% by increasing the length-to-diameter ratio of the receiver hole. Meanwhile, increasing the length-to-diameter ratio will result in an increase for the weight of the cover-plate. If both the receiver hole incidence angle and the radius difference are considered at the same time, the flow loss of the receiver hole will be further reduced, the discharge coefficient of the bDSRH can achieve about 0.696, which increases by 18.1%, compared with the discharge coefficient of the hDSRH. For the pre-swirl system, this shows that the incidence angle has a greater influence on the performance of the receiver hole. This conclusion is consistent with the research results in references written by Jarzombek, et al. (2006) and Lee, et al. (2020). Compared with other types of receiver holes, the VSRH considering the incidence angle at the receiver hole, the influence of outlet angle of the receiver hole on the supply hole, and the influence of the radius difference, has a minimal flow loss in the receiver hole. The discharge coefficient of the VSRH increases by 10.9%, up to 0.772, compared with that of the bDSRH. But manufacture costs will also rise because of the vane structure of the VSRH.

**Fig. 16 Comparison of the System Temperature Drop Between Experimental and Numerical Results**

**Fig. 17 Discharge Coefficient of Different Receiver Holes**

Fig. 18 gives the variation of the static pressure ratio (\(\pi\)) at some critical sections. The variations of the static pressure ratio along the flow direction generally show a decreasing trend for the pre-swirl systems using other types of receiver hole respectively. For the pre-swirl systems using the DERH, the RSRH and the aDSRH, respectively, every cross section almost has the same static pressure ratio, and the maximum deviation of the static pressure ratio at every section is less than 0.9%. Three of them have a slight rise in the static pressure ratio at the supply hole inlet. For these three types of the receiver hole, their contours of the swirl ratio and the relative velocity streamlines are shown in Fig. 19(a) – 19(c). It can be seen from these figures, because of a smaller length-to-diameter ratio, the airflow with high velocity from the pre-swirl nozzle directly passes through the receiver hole, and part of airflow hits the turbine disk surface, which result in airflow velocity decreasing, pressure increasing. However, a higher length-to-diameter ratio receiver hole will increase significantly the centrifugal boost effect which will decrease the pressure at the receiver hole inlet by about 3.9, compared with that of the aDSRH. The contours of the swirl ratio and the relative velocity streamlines of the receiver hole is illustrated in Fig. 19(d). Compared with the Fig. 19(c) and the Fig. 19(d), it can be found that the flow situation in cover-plate cavity is improved by increasing the length-to-diameter ratio, but failure to improve the flow situation at the receiver hole inlet. Therefore, the large flow loss in the pre-swirl cavity results in a large pressure loss,
which is not good for reducing the pressure of the pre-swirl nozzle outlet. After considering the incidence angle at the receiver hole inlet, the pressure at the pre-swirl nozzle outlet is obviously lower for the systems using the bDSRH or the VSRH. Compared with the system using the DSRH, the pressure at the pre-swirl nozzle outlet is respectively reduced by 2.4% and 5.3%. The contours of the swirl ratio and the relative velocity streamlines of both of them are drawn in Fig. 19(e) and 19(f). Compared these two figures with the Fig. 19(d), it can be discovered that the flow situation is evidently improved under the condition of considering the incidence angle of the receiver hole inlet, and the flow loss at the receiver hole inlet is also reduced. What's more, for the system using VSRH, because of improving the outlet of the receiver hole simultaneously, this type of the receiver hole not only reduces the flow loss of the receiver hole inlet, but also improves the flow situation of the supply hole inlet, and decreases the flow loss at the supply hole. Thus, the system using the VSRH has the best aerodynamic performance in all types of the receiver holes.

Fig. 18 Variation of the Static Pressure Ratio at Some Critical Sections

Fig. 20 Variation of the Temperature Ratio at Some Critical Sections

(a) DSRH

(b) RSRH
The variations of the temperature ratio at every cross section are displayed in Fig. 20. The temperature ratio is lower in Fig. 20; the relative temperature is lower. The variation of the temperature ratio of every cross section is slightly different from the variation of the static pressure ratio. For the systems using the DSRH, the RSRH, and the aDSRH, the variations of the temperature ratio of the rotating parts (that is equal to the difference of the temperature ratio between the receiver hole inlet and the system outlet) are almost identical, the maximum deviation less than 0.1%. The system temperature ratios of three of them are approximately 0.931, and the corresponding system temperature drops are 20.6K. For these six types of the receiver holes, the maximum deviation of the temperature ratio is located at the pre-swirl nozzle outlet. The temperature ratio at the pre-swirl nozzle outlet is the smallest for the system using the VSRH, about 0.913, the corresponding pre-swirl nozzle temperature drop around 26.0K; the temperature ratio at the pre-swirl nozzle outlet is the largest for the system using the DSRH, about 0.921, the corresponding pre-swirl nozzle temperature drop about 23.6K. This is because that the temperature drop of the pre-swirl nozzle is decided by the pre-swirl nozzle pressure ratio under the condition of the specified system inlet temperature and system inlet pressure. Furthermore, the system temperature ratio is dependent on the pre-swirl nozzle. Therefore, the minimum system temperature ratio is the system using the VSRH in all of the systems, approximately 0.922, the corresponding system temperature drop about 23.4K.

As shown in Fig. 20 and Fig. 21, the system temperature drop efficiency ($\eta$) and the system dimensionless specific power consumption ($\Phi$) were displayed. As the same as the above analysis, the system using the vane shaped receiver hole has the best performance, compared with other systems using other receiver holes. Its system temperature drop achieves about 0.63, increasing by 14.6%, compared with the system using the vane shaped receiver hole. Meanwhile, its system dimensionless specific power consumption decreases up to -0.63, decreasing by 44.6%, compared with the system using the runway shaped receiver hole. However, its special vane structure will also increase the manufacture costs. In contrast with the system using the vane shaped receiver hole, the system using the receiver hole with both a circumferential angle and a radial angle (bDSRH) has only 6.0% system temperature drop efficiency loss and 12.7% system dimensionless specific power consumption increase, second only to the system using the vane shaped receiver hole.
hole. Therefore, the receiver hole using both a circumferential angle and a radial angle (bDSRH) is also a good choice in design.

From the above analysis, it can be found that, for the system with a small radius difference between the pre-swirl nozzle and the supply hole, using the receiver hole with a small length-to-diameter ratio, the shapes and radial angles have little influence on the flow performance of the receiver hole. What's more, increasing the length-to-diameter ratio of the receiver hole cannot effectively improve the system temperature drop, due to failure to improve the flow situation of the pre-swirl cavity, but it will increase the centrifugal boost effect and decrease the pressure of the receiver hole inlet. Unfortunately, a higher length-to-diameter ratio receiver hole means that a heavier cover-plate disk, although not much improvement on the system performance. Therefore, it not recommended to directly increasing the length-to-diameter ratio of the receiver hole. If the incidence angle of the receiver hole inlet is considered in the design, the performance of the receiver hole will be significantly improved, increasing the discharge coefficient of the receiver hole by 18.1%, up to around 0.7, increasing the system temperature drop to 22.0K, increasing the system temperature drop efficiency up to 0.59. For the system, the flow loss of the supply hole inlet restricts the further improvement of the system performance. If a further improvement is made at the receiver hole outlet, then the receiver hole with good aerodynamic performance is designed, which is called the vane shaped receiver hole (abbreviated as VSRH). The system using the VSRH has the best system performance, increasing the system temperature drop to 23.4K, and the discharge coefficient of the receiver hole can be achieved to 0.77. At the same time, the manufacturing costs will also rise for the system using the VSRH.

4 CONCLUSIONS

For the pre-swirl system with a small radius difference between the pre-swirl nozzle and the supply hole, by numerical analysis, the performance of all systems using six types of the receiver holes was compared, including the system temperature drop efficiency, the dimensionless system specific power consumption, the flow field and the discharge coefficient of the receiver hole. To verify the numerical results, the typical system using the runway shaped receiver hole was studied by experiment. The main research conclusions as follows.

In designing and improving the pre-swirl system, the vane shaped receiver hole with a good aerodynamic performance is the best choice in terms of the system performance. However, when there is no high requirement for the system performance, but also hope to reduce the manufacturing costs, the drilling shaped receiver hole with both a circumferential angle and a radial angle is a good choice. It also has a relatively good aerodynamic performance and a lower manufacturing costs. What’s more, its weight increase by increasing the length-to-diameter ratio can be minimized through a careful design.

Compared with the system using the runway shaped receiver hole with a small length-to-diameter ratio, the system performance has a significant improvement for the system using the vane shaped receiver hole. According to the numerical results, the system temperature drop efficiency increases by about 14.6%, respectively, for the two design conditions of the system pressure ratio 1.6 and 1.9; the dimensionless system specific power consumption is reduced by 44.6%, under the same design condition. The maximum value of the system temperature drop efficiency can reach up to 0.63 for the system.

DISCLOSURE STATEMENT

No potential conflict of interest was reported by the authors.

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