Experimental and CFD analysis on the pressure ratio and entropy increment in a cover-plate pre-swirl system of gas turbine engine

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ABSTRACT

The pre-swirl system can offer cooling air to improve thermal shock to the rotating turbine blades in a gas turbine engine. This study focuses on theoretical, experimental, and numerical analyses about the flow and entropy increment for a pre-swirl system at the high-speed test conditions (up to 10000 rpm). Especially, the pre-swirl system entropy increment was firstly measured in the experiment. Then, the experimental test and CFD analysis are conducted to reveal the pressure ratio characteristics and loss mechanism. Research suggests that the non-dimensional temperature drop and the rotating Mach number are major determinants for the ideal pressure ratio. However, the ratio of the actual system pressure ratio to the ideal pressure ratio only depends on the entropy increment. For a given rotating Mach number, the actual system pressure ratio decreases with the non-dimensional temperature drop increasing. The increment of mass flow rate results in an enhancement in the entropy increment. In addition, the entropy increment of the receiver hole can be minimal for the swirl ratio of pre-swirl nozzle outlet close to 1. Therefore, this study can provide the basis for designing and optimizing the pre-swirl system at a high rotation speed.

NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| A      | Area, m²    |
| b      | Outer diameter of the turbine disc, m |
| c      | Speed of sound, m/s |
| ccr    | Critical velocity, m/s |
| cp     | Specific heat capacity, J/(kg·K) |
| m      | Mass flow rate, kg/s |
| mccr   | Critical mass flow rate, kg/s |
| Maφ    | Rotating Mach number, - |
| p      | Static pressure, Pa |
| p*     | Total pressure, Pa |
| r      | Radius, m |
| Rg     | Specific gas constant, - |
| T      | Static temperature, K |
| T*     | Total temperature, K |
| ηs     | Pressure ratio coefficient, - |
| ϵ      | Turbulent dissipation rate, m²/s³ |
| Δs     | Relative entropy increment, kJ/(kg·K) |
| ΔT     | Temperature drop, K |
| θ      | Dimensionless temperature drop, - |
| Π      | Pressure ratio of the system, - |
| ρ      | Density, kg/m³ |
| μ      | Dynamic viscosity, N·s/m² |
| Ω      | Rotating speed, rad/s |
| id     | Ideal |
| ref    | Relative coordinate system |
| φ      | Tangential direction |
| ax     | Axial direction |
| β      | Swirl ratio, - |
| γ      | Isentropic exponent, - |
| γid    | Ideal |
| ref    | Relative coordinate system |
| φ      | Tangential direction |
| ax     | Axial direction |

1. Introduction

The pre-swirl system is one major component of the second air system for a gas turbine engine. The temperature at the combustion chamber exit always exceeds the thermal resistance limit of turbine rotor blades material to improve efficiency (Goldstein et al., 1974; Liu et al., 2012). Then, turbine rotor blades need cooling air to ensure...
their normal work and enough long service life. However, the pre-swirl system can provide high-pressure and low-temperature cooling air for the turbine rotor to avoid thermal shock as soon as possible (Jiang et al., 2021). Hence, the performance of pre-swirl system will directly relate to the cooling effect of downstream turbine rotor blades.

As the pre-swirl systems evolve, the integration of high-pressure and low-temperature cooling air sources has become increasingly feasible and interesting. As well as, the swirling flow is of concern for flow field characteristics (Bui, 2008; Liyan et al., 2013; Sergio et al., 2011; Yehia et al., 2009). The variables of temperature drop and power consumption are of importance for evaluating flow field characteristics (Lin et al., 2018, 2019). Then, existing studies in the literature have pointed out the interest of temperature drop for the airflow relative to the rotating wall due to the cooling air expand and accelerate at the static pre-swirl nozzle along the rotating direction of the rotor. Previous researches indicated that the cooling effect of downstream blades is mainly dependent on the temperature drop. Meierhofer et al. (Meierhofer & Franklin, 1981) firstly measured the cooling air temperature at a pre-swirl system exit. The temperature drop was measured at different mass flow rates, rotational Reynolds numbers, and swirl ratios for a direct-transfer pre-swirl system (Geis et al., 2003; Umesh et al., 2010). Then, Benim et al. (2004) and Snowsill et al. (Snowsill & Young, 2006) focused on the 3D computational approaches of pre-swirl systems. They found dimensionless pressure and temperature that show a relatively weak sensitivity to the relative positions of the rotor and stator domains using the frozen rotor modeling. Especially, Umesh et al. (2010) conducted an experimental analysis of pre-swirl system by comparison to the numerical simulation with the $\kappa$-$\epsilon$ turbulence model, which is of a small difference with the experimental result. Then, the variation of geometric structure is also of importance for the flow parameters in the pre-swirl system (Jarzombek et al., 2006; Jarzombek et al., 2007). Bricaud et al. (2007) investigated the influences of geometry (such as the area and shape of nozzles and receivers), the pre-swirl cavity’s shape and size on flow and heat transfer of a pre-swirl system. Since the experimental test and numerical analysis can get more information on the flow field (Lin, Zheng, Jiang, Lin, et al., 2019; Zhang et al., 2020), the CFD analysis is reported for a direct-transfer pre-swirl system (Lei et al., 2021; Paul et al., 2006). Moreover, the numerical results were compared with experimental measurement results and also proved to be sufficiently accurate. Especially, The temperature drop characteristics were studied for a pre-swirl system (Karabay et al., 1999; Popp et al., 1998). And Chew et al. (2003) measured the system temperature drop. Lock et al. (2005) researched the heat transfer in the disc cavity. In addition to the temperature drop, the air supply pressure of a pre-swirl system also has a great influence on the cooling effect and the cooling channel design of the downstream rotor blades. Youyou et al. (2003) discussed the influences of some parameters (such as the nozzle numbers and swirl ratio) on the total pressure of the rotor-stator cavity.

For a cover-plate pre-swirl system, the impeller will also affect the turbocharging results of the rotor part in the pre-swirl system, whose specific effects have been investigated by predecessors (Charles et al., 2011; Gupta et al., 2008; Shuqing et al., 2013). Then, Zhu et al. (2010) researched the influences of rotating Reynolds number, dimensionless mass flow rate, and swirl ratio on the pressure loss characteristics, and analyzed the mechanism of pressure loss in the absence of related experimental data. He et al. (2011) considered the flow characteristics of the turbine disc cavity at an allowable rotating speed. For real case studies of applications of CFD models, more relative researches were studied by Ali Mohammad et al. (2019) and Mahdi et al. (2019).

From the above, many investigations have been performed to prove that the pre-swirl system temperature drop mainly depends on the airflow swirl ratio at the pre-swirl nozzle outlet under the specific rotating speed and radius of the nozzle. It is necessary to enhance the swirl ratio to increase the temperature drop. However, there are fewer experimental investigations for the key performance index of the supply pressure related to the high rotating speed. In the previous experimental study for the pre-swirl system, it is very difficult to measure the system entropy increment directly. This is mainly because of the difficulty of measuring the static temperature. However, the static pressure is easy relatively to measure. Therefore, it is easier to get a temperature drop coefficient. As a continuous work to explore the effect of temperature in a pre-swirl system, this study focuses on theoretically, experimentally, and numerically investigating the pressure ratio and entropy increment characteristics of a cover-plate pre-swirl system under a high rotating speed test condition. In this paper, the static temperature at the system outlet is obtained by the relative total temperature at the system outlet at a small mass flow rate. Especially, it is the first time to define the pressure ratio coefficient for evaluating the pressure loss in the pre-swirl system. The relationship as a function of the ideal pressure ratio and temperature drop of the pre-swirl system is deduced through theoretical analysis. Simultaneously, the main differences between the influencing factors of actual and ideal pressure ratios are revealed. Then, the pressure characteristics are obtained.
by the experimental test with a high rotating speed up to 10 000 rpm. Additionally, the pressure loss mechanism is revealed for the first time to analyze the system entropy increment, based on the comparison of experimental and numerical results. Therefore, the results of this study can provide the basis for optimizing the cover-plate pre-swirl system.

This present study is mainly organized as section 2 shows experiment design and methodology on a novelty experiment test rig (section 2.1), experimental measurement method including the pressure calibration (section 2.2), theoretical derivation (section 2.3), and numerical method (section 2.4). And, results and analysis are presented in section 3. Finally, the conclusions can be addressed in section 4.

2. Methodology

2.1. Experiment test rig

The configuration of a novelty test system designed in this study can be shown in Figure 1. Then, the core components include the experimental test section (pre-swirl system), measurement equipment, intake and exhaust air systems, drive systems, lubrication oil system, and cooling water system. In the figure, the air intake pipeline (green line) is divided into two parts: one is the main inlet air path, which is to control the mainstream parameters for the test section; the other is the sealing air path to control the mass flow rate by adjusting pressure. Then, the exhaust air (orange line) is also divided into two parts: one is the emptying air path for the auxiliary adjustment of mainstream parameters; the other is the main exhaust air path, which is responsible for the discharge of experimental waste gas and the adjustment of back pressure of the pre-swirl system. A detailed view of the test section of the pre-swirl system is shown in Figure 2.

In Figure 1, the drive system is responsible for driving the turbine disc and adjusting its rotating speed, which is mainly composed of a bearing box, a high-speed motor (electrical machinery), and an inverter. The rotating speed of the high-speed motor is controlled by a frequency converter, and the speed control accuracy is ±1 rpm. Moreover, the maximum operating speed of the rotating component (cover-plate disc and turbine disc) is 10 000 rpm. To make the rotating disc run stably at high speed and reduce the influence of bearing heat production on the measurement, the motor and bearing should be lubricated and cooled sufficiently. Then, the high-speed motor is lubricated by the grease, which does not need to supply lubricating oil, but only needs a water system for cooling. The lubrication and cooling of the bearing can be achieved by the circulating cooling system. Hence, the flow of cooling water and lubricating oil was monitored in real-time during the experimental test. Especially, the thermocouples are arranged near the

![Figure 1. An experimental test of system configuration.](image-url)
bearing pedestal to monitor the temperature in real-time so as to prevent the bearing from overheating during the experiment.

2.2. Experimental measurement and pressure calibration

The parameters were measured on the static and rotating parts of the test rig, as shown in Ref. (Wu et al., 2018). The mass flow rate, pressure, and temperature were measured on the stator parts. The mass flow rate was metered by high-precision orifice meters. They were calibrated with the maximum uncertainty of 1.0%. The pressures in the inlet cavity, nozzle exit, axial force balance cavity, and pressure balance cavity were measured by a PSI-type pressure scanner with 16 channels and an uncertainty of 0.05%. The total temperatures in the inlet cavity and exhaust cavity were measured by K-type thermocouples with an accuracy of ±1 K. The inlet cavity was designed to be big enough to make the velocity in it smaller than 10 m/s, so the difference between the inlet total temperature and the measured recover temperature by a bare thermocouple is small enough to be ignored.

To obtain the relative total temperature and the outlet static pressure at the specific sections, the measurement points of temperature and pressure on the turbine disc are set in the pre-swirl system. Additionally, two K-type thermocouples were fixed at two different circumferential positions of the turbine disc to measure the system outlet relative to total temperature. The measurement accuracy of the K-type thermocouples is ±1 K. Similarly, there were static pressure measurement points
Figure 3. Measurement of rotor parameters.

at the supply hole outlet at two different circumferential positions in order to measure the static pressure on the turbine disc, which were measured by 2 Kulite tiny transducers that can endure a high centrifugal acceleration. The Kulite pressure transducers were calibrated by the PSI-type pressure scanner, and then, the relative error of which after calibration is 0.1% within the range of experimental pressure change.

As shown in Figure 3, a self-designed data recorder is connected with the temperature and pressure measurement points on the turbine disc by the hollow shaft to collect the data measured on the disc, which has its batteries to reduce the impact of electromagnetic interference. During the experiments, the data recorder rotates synchronously with the shaft and converts the pressure and temperature signals generated by sensors and thermocouples into digital signals for storage. After the experiment is finished, the disc stops rotating and the collected data is exported through the reserved data port.

To reduce the centrifugal load on the pressure sensor at high rotating speeds, a pressure sensor was located in the low radius position, measuring the pressure at a high radius position by the pressure-leading pipe. The pressure measured by the pressure sensor is less than the actual pressure at the measurement point due to the centrifugal forces at a high rotating speed. What’s more, a higher centrifugal load will lead to some changes in sensor characteristics. Therefore, the pressure measured by the pressure sensor should be calibrated before experiments, as shown in Figure 4. When the mass flow rate of supply air is small, the differential pressure at the same radius of the exhaust cavity in the rotor-stator cavity is small. According to that, the pressures at the same radius of the exhaust cavity were measured respectively by a pressure scanner on the stator casing and the pressure sensor on the turbine disc at different rotating speeds. Then, the pressure measured by the sensor at different rotating speeds was corrected according to that of the scanner.

According to the error transfer function, the relative error of the static temperature at the supply hole outlet is 1.1%, which depends on the relative error of the relative total temperature and the magnitude of mass flow rate at the cross-section of the supply hole outlet. The measurement accuracy of the temperature drop is ±2 K, which is obtained easily by the difference between the system inlet total temperature and the supply hole outlet relative total temperature. The relative error obtained by the experiment is 0.14% for the system pressure ratio.

2.3. Theoretical derivation and analysis

The cooling air through the pre-swirl nozzle expands and accelerates along the rotating direction, then flows into the rotor-stator cavity. After that, it passes through the receiver hole into the cover-plate cavity. Finally, it reaches the turbine rotor blades after crossing through the supply holes reserved on the turbine disc. At the outlet of the supply hole, the circumferential velocity of the airflow is equal to that of the turbine disc.

The system temperature drop and the system pressure ratio are the key performance parameters. According to Ref. (Wu et al., 2016), the correlation between the temperature drop (ΔT) and the pressure ratio (p2/p0) can be defined as:

$$\Delta T = T_0^* - \frac{V_{2,ax}^2}{2C_p} - T_0 \left( \frac{p_2}{p_0} \right)^{\frac{R_g}{C_p}} \cdot e^{\frac{\Delta s}{n_p}}$$  \hspace{1cm} (1)
At the system inlet, the inlet static parameters can be approximately replaced by the total inlet parameters in Eq. 1 because of the low air velocity. The airflow axial velocity at the pre-swirl system outlet is relatively small, which can be ignored. The system pressure ratio can be expressed as follows:

$$\Pi = \left(1 - \frac{\Delta T}{T_0^*}\right)^{\frac{\gamma}{\gamma - 1}} e^{-\frac{\Delta s}{R_g}}$$  \hspace{1cm} (2)$$

The system dimensionless temperature drop is defined as:

$$\Delta T = \frac{\Omega r_2^2}{2c_p} \Theta$$  \hspace{1cm} (3)$$

where, \(\Omega\) is the rotating speed of the system, \(r_2\) is the radius of the system outlet.

Then, the system pressure ratio can be expressed as:

$$\Pi = \left(1 - \frac{\gamma - 1}{2} Ma_\theta^2 \Theta\right)^{\frac{\gamma}{\gamma - 1}} e^{-\frac{\Delta s}{R_g}}$$  \hspace{1cm} (4)$$

where, \(Ma_\theta\) is the rotating Mach number, which is mainly affected by the rotating speed of the system.

In this study, the system entropy increment only comes from the entropy production of the internal airflow without the consideration of the heat transfer from the wall. Since the entropy production cannot be negative, so the entropy increment \(\Delta s \geq 0\) is satisfied. When the entropy increment is nil, the ideal pressure ratio can be computed as:

$$\Pi_{id} = \left(1 - \frac{\gamma - 1}{2} Ma_\theta^2 \Theta\right)^{\frac{\gamma}{\gamma - 1}}$$  \hspace{1cm} (5)$$

From Eq. 5, the ideal maximum pressure ratio (\(\Pi_{id}\)) of the pre-swirl system depends on the two dimensionless parameters of the rotating Mach number \(Ma_\theta\) and the dimensionless temperature drop (\(\Theta\)) of the system. Previous studies have been indicated that the dimensionless temperature drop is decided by the airflow swirl ratio at the nozzle outlet for the fixed pre-swirl radius and rotating speed.

Figure 5 gives a variety of ideal pressure ratio with the non-dimensional temperature drop. As illustrated in the figure, with the increasing of the dimensionless temperature drop (\(\Theta\)), the ideal pressure ratio (\(\Pi_{id}\)) would decrease sharply at a specific rotating Mach number \(Ma_\theta\). And then, when \(\Theta = 0\), \(\Pi_{id}\) is 0. Because the increasing temperature drop means the decrease of the ideal pressure ratio, so designing the pre-swirl system cannot only consider increasing the system temperature drop indefinitely. This design should consider comprehensively the requirement of what the appropriate parameters should be selected according to the requirements of turbine rotor blades for air supply pressure and temperature.

The ratio of actual pressure ratio to the ideal pressure ratio is defined as the pressure ratio coefficient (\(\eta_s\)), which can be obtained by Eq. 4 and Eq. 5.

$$\eta_s = \frac{\Pi}{\Pi_{id}} = e^{-\frac{\Delta s}{R_g}}$$  \hspace{1cm} (6)$$

As shown in Eq. 6, the actual system pressure ratio of the system depends on not only the ideal pressure ratio but also the entropy increment of the system. For a pre-swirl system with a specific structure, on the one hand, different parameters will lead to the different pressure ratio coefficients. On the other hand, the deviation of the actual system pressure ratio is not the same as the ideal pressure ratio due to the different entropy increments at the corresponding system.

This experiment studied the characteristics of the pressure boost and the system entropy increment at different rotating speeds and mass flow rates. The definition of the rotating Mach number (\(Ma_\theta\)) and the critical mass flow ratio (\(m/m_{cr}\)) can refer to Refs. (Karabay et al., 1999; Popp et al., 1998). The critical flow rate is defined as the following:

$$m_{cr} = \frac{p_0^* A_1}{\sqrt{T_0^*}} \Delta \frac{2}{R_g \gamma} \frac{s^{\frac{\gamma - 1}{\gamma}}}{\gamma + 1}$$  \hspace{1cm} (7)$$

where, \(A_1\) is the area of nozzle throat, \(m_{cr}\) is the system critical mass flow rate, which can achieve under the certain inlet total pressure and the certain inlet total temperature. Thus, the final range of dimensionless parameters can be determined as:

$$0.50 \leq Ma_\theta \leq 0.68, \ 0.56 \leq m/m_{cr} \leq 0.89$$  \hspace{1cm} (8)$$
2.4. Numerical method and model

To obtain more detailed flow field characteristics and compare them with the experimental results, some simplifications were made based on the geometrical structure of the experimental section. Figure 6(a) gives the geometry structure of a cover-plate pre-swirl system. Moreover, the main geometry dimensions of the computational model were consistent with those of the experimental model. Here, the pre-swirl cavity with complex structure was simplified as an annular cavity, ignoring the inward flow and the outward flow. Due to an axisymmetric periodic domain for the cover-plate pre-swirl system (2011), a rotating periodic boundary condition was adopted to save the computation time. Thus, the final computational domain was determined as 1/64 of the actual annular domain, as shown in Figure 6(a).

As shown in Figure 6(b), static domains and rotational domains were merged by an axial interface surface, and structured hexahedral meshes were divided. The grids in the wall adjacent zones were refined to ensure the $y^+$ value within the range of $30 \sim 200$. A grid dependency study is conducted at different grid cell numbers from 0.707–1.11 million, as shown in Table 1. As you can see, the results under the three grid numbers have very little difference. Therefore, 1.11 million nodes were employed in the present numerical study.

Since the heat conduction between the system wall and the airflow is very small, the pre-swirl system can be regarded as an adiabatic system. The effect of flow field near the wall does not care about. This paper pays more attention to the work-to-heat conversion. Since the $\kappa$-$\epsilon$ turbulence model can well predict the flow field with a high Reynolds number (Jiang et al., 2020; Lin, Zhou, Fawzy, Zhang, et al., 2019), this study mainly considered the aerodynamic and thermodynamic in the turbulence core regions, instead of heat transfer near wall region. Then, the standard $\kappa$-$\epsilon$ equations and the standard wall treatment were adopted. The fixed rotor phase method was used to calculate the rotational region. The rotating speed was given for rotating walls and domains in the numerical model. The stationary walls and domains were

| Number of cells | $\Theta$ | $\Pi$ | $\Delta s/R_g$ |
|-----------------|--------|------|--------------|
| 707,167         | -0.006 | 0.835| 0.153        |
| 908,879         | -0.007 | 0.836| 0.153        |
| 1,111,806       | -0.006 | 0.836| 0.151        |

Figure 6. Computational model and mesh of the cover-plate pre-swirl system.
Figure 7. The static pressure distribution contours in the exhaust cavity.

Figure 8. A comparative experimental result in the process of the pressure calibration.

Given 0 rpm. The ideal gas was employed with variable thermal conductivity and viscous coefficient. All walls were set to be adiabatic and no-slop. The system inlet was set as a pressure inlet with a total temperature of 300 K and a total pressure of 0.15 MPa. The outlet was set as a pressure outlet with the target mass flow rate which corresponded to the experimental operation.

Figure 7 shows the static pressure distribution contours in the exhaust cavity when the speed is the maximum experimental operating speed of 9720 rpm. The static pressure difference between the rotor and the stator at the same radius is very small, and the maximum does not exceed 1 kPa. Thus, the calibration measures described in Figure 5 are feasible.

Figure 8 shows the results of the pressure calibration experiment. The abscissa is the pressure ($p_r$) measured by the rotating measurement system and the ordinate is the pressure ($p_i$) measured by the static measurement system. It can be seen that due to the effect of speed, $p_r$ is smaller than $p_i$, and the greater the speed, the greater the difference. Therefore, when processing experimental data, the pressure data measured by the rotating measurement system should be corrected according to the results shown in Figure 7 to obtain the actual pressure at the measuring point.

The numerical simulations of cover-type pre-swirl system configurations are conducted with commercial code Fluent 16.0. According to the research results of Liang et al. (2019), compared with the unsteady-state time-averaged results, the steady-state space-average results of multiple rotor positions show that the pressure is 0.2% higher and the temperature is 0.1% lower. The difference between steady-state time-average results and unsteady-state results is small. Therefore, the steady-state Navier-Stokes equations are solved to solve the three-dimensional steady-state turbulent flow equation and the energy equation. The second-order upwind scheme was selected for the equations’ discretization (Yang et al., 2020), and the SIMPLE algorithm was adopted as the pressure-velocity coupling scheme. Hence, the residual convergence is set $1 \times 10^{-6}$ for the equations of continuity, energy, $\kappa$, and $\epsilon$.

### 2.5. Definition of entropy production rate

It is necessary to obtain the system entropy production distribution for analyzing the specific area where the entropy increment occurs. The entropy production distribution was studied for a pre-swirl system by using the formula proposed by Fabian and Heinz (2005). Entropy production can be divided into direct dissipation term and turbulent dissipation term, and the following are the specific formulas.

\[
\begin{align*}
    s_d &= \frac{\mu}{T} \left\{ 2 \left[ \left( \frac{\partial V_x}{\partial x} \right)^2 + \left( \frac{\partial V_y}{\partial y} \right)^2 + \left( \frac{\partial V_z}{\partial z} \right)^2 \right] + \left( \frac{\partial V_x}{\partial y} + \frac{\partial V_y}{\partial x} \right)^2 + \left( \frac{\partial V_x}{\partial z} + \frac{\partial V_z}{\partial x} \right)^2 + \left( \frac{\partial V_y}{\partial z} + \frac{\partial V_z}{\partial y} \right)^2 \right\} \\
    s_d' &= \frac{\rho \varepsilon}{T}
\end{align*}
\]

where, $s_d$ is the direct dissipative entropy production rate caused by time-averaged velocity, $s_d'$ the turbulent dissipative entropy production rate caused by time-averaged fluctuating velocity, $T$ is the temperature, $V_x, V_y$ and $V_z$ are respectively the velocities of airflow in the direction of $x, y$, and $z$, $\rho$ is the density, $\mu$ is the dynamic viscosity of airflow, $\varepsilon$ is the turbulent dissipation rate.

### 3. Results and analysis

#### 3.1. Analysis of actual pressure characteristics

Figure 9 shows the variations of the actual system pressure ratio (the ratio of the static pressure at system outlet to the total pressure at system inlet, $\Pi$) with the
change of $m/m_\text{cr}$. The solid points and the hollow points respectively represent the experimental results and the numerical results, meanwhile, the square points and the circle points are respectively the results at $Ma_\phi = 0.68$ and $Ma_\phi = 0.50$. As shown in the figure, although the pressure ratio of numerical simulation is smaller than that of the experiment, the trend of numerical results is basically consistent with that of experimental results. The system pressure ratio decreases gradually with the mass flow rate increment, and increases with the rotating speed increases.

Previous researchers have done a lot of studies on the system temperature drop. The system dimensionless temperature drop depends on the swirl ratio at the nozzle outlet, which increases with the swirl ratio increasing. According to the previous theoretical deduction, the ideal pressure ratio of the system and the dimensionless temperature drop satisfy a specific correlation under a specific rotating Mach number. Figure 10 shows the variations of the actual system pressure ratio with the dimensionless temperature drop, compared with the ideal result given in Eq. 5. The hollow points and the solid points are respectively the results of numerical simulations and experiments, and the dashed line is the theoretical results under the ideal conditions.

The results at $Ma_\phi = 0.5$ are shown in Figure 10(a). The system pressure ratio decreases from about 0.94–0.43 with the increase of the temperature drop from about −0.17–0.53. The experimental results and the numerical results correspond well. However, the actual system pressure ratio is smaller than the ideal pressure ratio. The maximum deviation between the actual and ideal system pressure ratio is 36% at $\Theta = 0.96$, which is due to the entropy increment. As shown in Figure 10(b), the experimental results are in accordance with the numerical results at $Ma_\phi = 0.68$ and $Ma_\phi = 0.50$, but both of them are relatively smaller than the ideal values. The computation results suggest that with the dimensionless temperature drop increasing from around −0.38–0.44, the system pressure ratio decreases from about 0.97–0.58. Furthermore, the experimental results indicate that with the increase of dimensionless temperature drop from around −0.45–0.16, the system pressure ratio decreases from about 0.98–0.74. The minimum deviation between the experimental and ideal results is 13% at $\Theta = −0.07$, and the maximum deviation between the numerical results and the ideal results is 33% at $\Theta = 0.44$.

### 3.2. Analysis of ideal pressure characteristics

According to the static pressure and the static temperature obtained from the experiments and numerical simulations at both the inlet and outlet of the system as well as the entropy increment formula at the constant specific heat capacity. The entropy increment at the outlet of the system relative to the inlet of the system can be expressed as:

$$\Delta s = C_p \ln \frac{T_2}{T_0} + R_g \ln \frac{p_2}{p_0}$$

\[ (11) \]
Based on Eq. 6 and the entropy increment, the pressure ratio coefficient can be obtained. Figure 11 presents the variations of $\eta_s$ with the dimensionless temperature drop $\Theta$. A good agreement can be seen between the experimental and numerical results. $\eta_s$ decreases gradually with the $\Theta$ increasing under the same $Ma_\phi$, and $\eta_s$ is relatively larger at the lower rotating speed under the same temperature drop.

The ideal pressure ratio $\Pi_{id}$ can be deduced from the actual system pressure ratio $\Pi$ and the pressure ratio coefficient of the system based on Eq. 6.

$$\Pi_{id} = \frac{\Pi}{\eta_s} \quad (12)$$

What’s more, according to Eq. 5

$$\Pi_{id}^{\gamma - 1} = 1 - \frac{\gamma - 1}{2} Ma_\phi^2 \Theta \quad (13)$$

According to the results of the experiments and the numerical simulations in Figure 10 and Figure 11, a series of data points are obtained from Eq. 12 ($\Pi_{id}$ computed from $\Pi$ and $\eta_s$) and compared with the theoretical line described by Eq. 13. The variations of the ideal pressure ratio with the temperature drop are plotted in Figure 12. The data points present a very good linear trend and accord well with the theoretical line. The maximum aberrancy of them is 2.7%. Although a series of hypotheses are adopted in the theoretical deduction process, the coefficients $\eta_s$ depending on the entropy increment still can better represent the deviation between the actual and ideal system pressure ratio. In addition, the larger the abscissa value, the larger deviation, which should be because of ignoring the axial velocity of the system outlet airflow (the larger axial velocity caused by the relatively larger mass flow rate).

### 3.3. The characteristics of the entropy increment

Although the variation trend of the actual boost pressure ratio corresponds with the dimensionless temperature drop, there is a big deviation between the actual and ideal system pressure ratio, and the change range of the difference is not consistent with the temperature drop. To acquire the relationship of the actual system pressure ratio with the system temperature drop, it is still necessary to consider the system entropy increment according to the previous introduction.

The distribution of the system entropy production can be calculated by Eq. 10, which at a specific operation is plotted in Figure 13. The regions where entropy production is produced along the flow direction are in turn the suction surface of the pre-swirl nozzle, the static wall of the pre-swirl cavity, the inside of the receiver hole, the downstream cover-plate cavity, and the supply hole inlet. Either the direction or the magnitude of velocity changes greatly in these regions, which leads to the flow dissipation. Especially at the inlet of the supply hole, the circumferential velocity of the airflow changes significantly and the swirl ratio rapidly increases to 1.0 under the action of torque, which brings about a larger entropy production.

Figure 14 shows the change situation of the total entropy increment with the dimensionless mass flow rate under different rotating speeds. The hollow points are the results of numerical simulations and the solid points are the results of experiments. The results at $Ma_\phi = 0.50$ are plotted in Figure 14(a). The numerical results indicate that $\Delta s/R_\phi$ increases from about 0.09–0.34 with the increase of dimensionless mass flow rate from about 0.49–0.93; the experimental results make clear that $\Delta s/R_\phi$ increases from around 0.10–0.25 with the increase of dimensionless mass flow rate from around 0.60–0.89.
Figure 13. Streamline and distribution of entropy production in the computational model \((m/m_{cr} = 0.73, Ma_\phi = 0.68)\).

Figure 14. Experimental and numerical results of entropy increment with mass flow rate.

Figure 14(b) displays clearly the results at \(Ma_\phi = 0.68\). The numerical results declare that with the increase of dimensionless mass flow rate from 0.49–0.93 approximately, \(\Delta s/R_g\) increases from 0.15–0.30 approximately; the experimental results show that with the increase of dimensionless mass flow rate from about 0.58–0.91, \(\Delta s/R_g\) increases from about 0.18–0.25. The numerical results correspond well with the experimental results under different rotating speeds.

The entropy increments of the main components relative to the system inlet can be calculated by Eq. 11 according to the numerical results. Figure 14 shows the variation of the entropy increment of the main components with the mass flow ratio and rotating Mach number. Different color blocks represent different pre-swirl system elements. For most of the operating conditions, the entropy increment at the supply hole is larger than that in the other regions, which is consistent with the results in Figure 13. In addition, with the mass flow ratio increasing, the entropy increment of both the system and the main components increase gradually. However, with the mass flow ratio increasing, the entropy increment of the receiver hole decreases first and then increases. The minimum entropy increment is different under different rotating Mach numbers, showing that the higher the rotational Mach number, the larger the mass flow ratio corresponding to the minimum entropy increment. As shown in Figure 14(a), the entropy increment in the receiver hole is larger than that in the supply hole at low rotating Mach numbers.

Under different rotating speeds, Figure 15 shows the variations of both the entropy increment in the receiver hole and the swirl ratio at the nozzle outlet. The results at \(Ma_\phi = 0.50\) are plotted by the red line, while the results
at $Ma_0 = 0.68$ are plotted by the blue line; the solid line gives the system entropy increment in Figure 15(a), while the dashed line shows the swirl ratio in Figure 15(b). The curves of both the swirl ratio and the entropy increment under the same rotating speed indicate that $\beta_1$ is close to 1 under the mass flow ratio corresponding to the minimum entropy increment. The greater deviation of $\beta_1$ from 1, the faster the circumferential velocity between the airflow in the receiver hole and the rotating wall, it will lead to the bigger entropy increment. With the mass flow ratio increasing, $\beta_1$ gradually increases from less than 1 to more than 1, and the mass flow ratio corresponding to $\beta_1 = 1$ increases with the rotating Mach number. This is because that the entropy increment at the receiver hole first decreases and then increases with the mass flow ratio increasing, and the mass flow ratio corresponding to the minimum entropy increment is larger with the rotating Mach number increasing.

4. Conclusions

This study focuses on the pressure ratio and the entropy increment characteristics of the cover-plate pre-swirl system at high rotating speed, based on the theoretical derivation, experimental analysis, and numerical simulation. The main conclusions are as follows:

For the pressure characteristics of a pre-swirl system, the ideal pressure ratio as a function of temperature drop is firstly deduced through theoretical analysis. Then, the experimental test and numerical simulation are conducted to conclude the variations of the system pressure ratio with the mass flow ratio at different rotating speeds. Especially, the system pressure ratio decreases gradually as the mass flow ratio increases, while increases slightly as the rotating speed increases.

Since the system pressure ratio decreases gradually with the dimensionless temperature drop increasing, a comprehensive variable equation can be deduced as a function of the pressure ratio, the entropy increment, and the temperature drop; moreover, the maximum deviation of this theoretical equation is only about 2.7% with comparison to the results of experimental measurement and numerical simulation. This theory can accurately present the relationship among the system pressure ratio, the system entropy increment, and the system temperature drop, indicating that the deviation between the actual and ideal system pressure ratio depends on the system entropy increment.

For the system entropy increment characteristics, the system entropy increments increase gradually as the mass flow rate enhances. Especially, the receiver hole entropy increment has the minimum value when the swirl ratio is close to 1 at the nozzle outlet. The result and conclusions of this study can provide the basis for optimizing the flow-pressure characteristics for a pre-swirl system.

It is necessary to highlight one limitation in the section 2.3. In the theoretical derivation, the static temperature can be approximately equal to the relative total temperature at the supply hole outlet due to a small mass flow rate. However, it is worth investigating the sensitivity of a large mass flow rate of supply air on static temperature at the supply hole outlet and the pre-swirl system entropy increment from the theoretical derivation in the future study.

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