

Flow characteristics in a rotor-stator cavity with cooling air inlet at low radius

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Abstract: Experiments were conducted on a typical rotor-stator system where air entered through an annular slot at low radius and flowed out of the cavity axially through a rim seal between the rotor and the stator. For the seal in this rotor-stator system, the stationary shroud overlapped the rotating one. Pressure distributions at the stator surface and flow resistance coefficients of the rotor-stator cavity with a maximum gap of 67 mm were measured under different dimensionless mass flow rates from 1.32×10^4 to 4.87×10^4 with a large range of rotational Reynolds numbers from 0.418×10^6 to 2.484×10^6 . The results show that pressure on the stator surface decreases with the increase of rotational Reynolds number when the dimensionless mass flow rate is below 1.3×10^4 ; when the dimensionless mass flow rate is above 3.034×10^4 , the trend reverses. This is the so-called “pressure inversion effect”. However, dimensionless pressure does not show the same changes when rotational dynamic pressure is chosen as the denominator. The resistance coefficient of the rotor-stator cavity is determined by the dimensionless mass flow rate and rotational Reynolds number; for practical application, the resistance coefficient can also be estimated by the turbulent flow parameter in the range of turbulent parameter from 0.1 to 1.6.

Key words: cooling system; rotor-stator system; flow characteristics; pressure distribution;
resistance coefficient

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Rotor-stator systems are widely used in modeling complex flow and heat transfer in the areas of cooling turbine or compressor discs, supplying turbine blade cooling air and sealing at the rim of a rotor-stator cavity. The cavities are often simplified into a plane rotating disc and a plane stationary disc with or without a shroud at the outer rim of disc. As there are often multiple inlets or outlets at some radial positions of the rotor or the stator, there are numerous different possible configurations of the rotor-stator cavity. In this paper, a brief review of the studies

related to the flow in rotor-stator systems with or without a shroud is presented.

A rotor-stator system with an inlet at the centre of the stator without a shroud is one of the earliest study models, and the flow mechanics of the laminar and turbulent flow of a large range of cavity gaps were studied by many researchers. Bayley and Owen^[1] carried out a theoretical and experimental study on the effects of mass flow rate, rotational speed and spacing of the rotor and stator on pressure, velocity and moment coefficient. Daily et al.^[2] and Dadkhah^[3] obtained

the relationship of the swirl ratio, the local dimensionless radius and the turbulent flow parameter.

Rotor-stator system with an inlet at the centre of the stator with a shroud at the disc rim is another important study model which is commonly used in modern gas turbine engines. In comparison with the cavity without a shroud, the moment coefficient on the disc surface increases. Most of the research related to the flow in the rotor-stator system has concentrated on the sealing and ingestion of gas, and the sealing performance of the cavity without mainstream was studied^[4-6]. When the mainstream appears, Daniels et al.^[7] claimed that the mainstream mass flow rate and circumferential pressure distribution dominates the minimum value of sealing air. Many other researchers have paid attention to this subject^[8-9].

During the last decade, as the basic knowledge of flow in infinite cavities has been obtained^[10], the emphasis of studies shifted to flow instabilities, and transition and turbulent flow in cavities with a hub and shroud. Two types of linear flow instability are believed to occurred in the cavity. Poncet et al.^[11] obtained the instability patterns of the two types numerically. Cros et al.^[12] conducted an experiment

using visualization and video image analysis, while Séverac et al.^[13] performed numerical studies to investigate the flow transition mechanisms inside the rotor-stator cavities. More studies focused on the turbulent flow in the cavities and numerical methods were extensively adopted^[14-15].

From previous investigations, it appears that flow characteristics in a rotor-stator cavity have been studied in depth. However, the flow characteristics of a rotor-stator cavity with inflow at a low-radial position, which is widely used in actual gas turbine engines, have not been completely investigated. Pressure distributions at the stator surface can help to calculate the axial force on the turbine disk and verify the numerical simulation results, so this paper aims to investigate the pressure distributions on the surface of the stator and the flow resistance coefficients of the rotor-stator cavity are also included.

1 Experiment apparatus

Experiments were conducted on the multifunctional rotational heat transfer experimental rig at the National Key Laboratory of Science and Technology on Aero-Engine Aero-thermodynamics at Beijing University of Aeronautics and Astronautics. As shown in Fig. 1, the whole

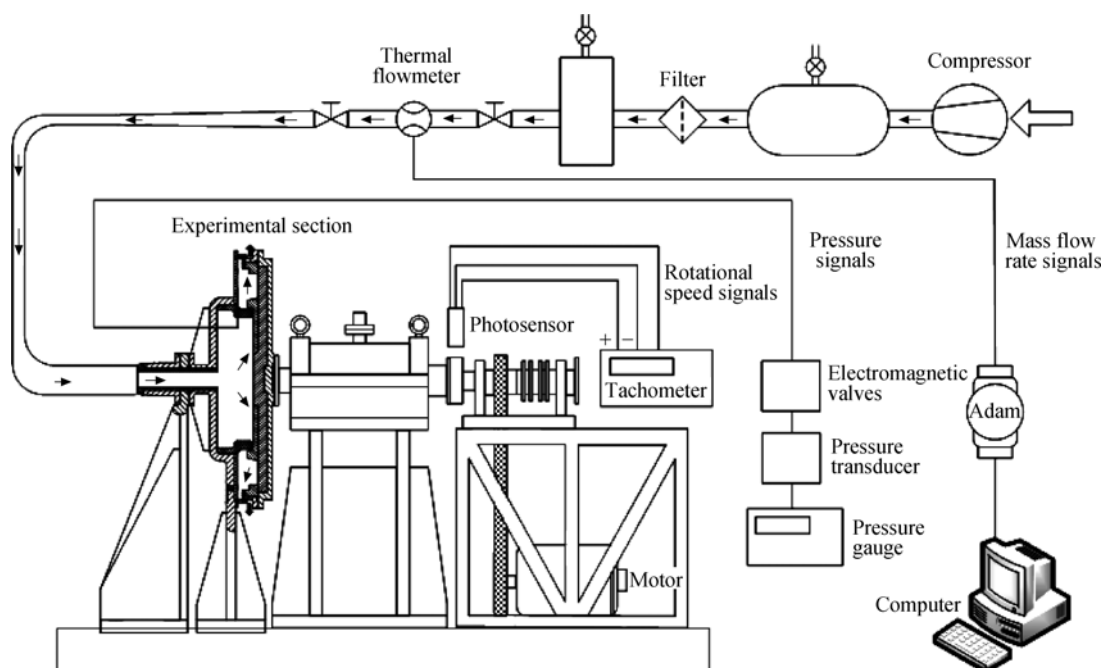


Fig. 1 Schematic diagram of experimental system

experimental system consists of the air supply system, experimental rig, power supply system, and data acquisition system.

The air is supplied by a laboratory compressor with a pressure up to 810.6 kPa and then regulated down to the required mass flow rate by a pressure regulator and flow control valves. An air dryer and filter were installed to remove water mist, oil drops and dust. The clean air flows through a plastic tube and then enters the central collecting cavity between the stator and the

rotor.

The experimental rig is composed of stationary parts (on the left in Fig. 2(a)) and rotating parts (on the right in Fig. 2(a)). $\dot{m}$ is the mass flow rate. The fluid from the collecting cavity is admitted into the cavity through an axially oriented slot at low radius of the stator and leaves it axially through the rim seal between the rotor and the stator. The geometric dimensions of the experimental section are summarized in Fig. 2(b).

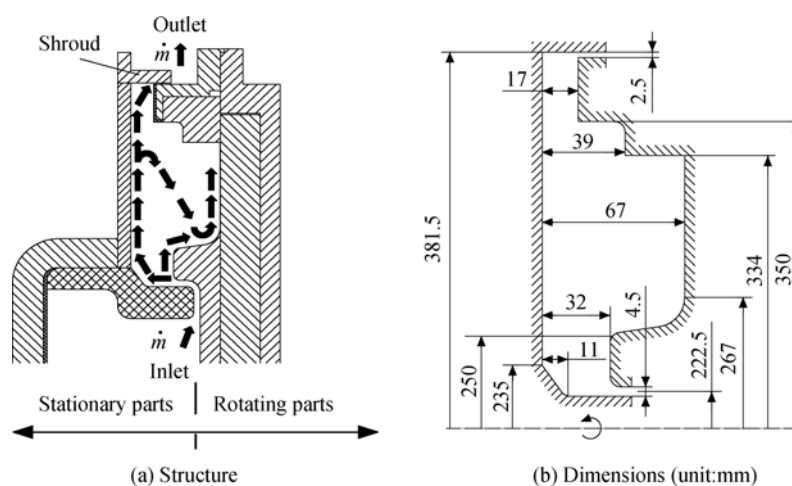


Fig. 2 Experimental rig

All the rotor components are driven by a 30 kW direct current (DC) electric motor which allows the rotational speed of the rotor to be varied up to 3600 r/min, thus providing a maximum rotational Reynolds number Re_ω of 2.491×10^6 .

The data acquisition system consists of measurements of pressure, mass flow rate and rotational speed. The pressure measurement apparatus is a combination of electromagnetically operated valves, pressure transducers and digital instruments. It was calibrated by a pressure calibrator whose full scale error is $\pm 0.05\%$. The calibrating results showed that there seems to be a linear relationship between measured pressure p_b , and calibrated pressure p_i

$$p_i = 0.2028 + 1.00059 p_b \quad (1)$$

The correlation coefficient is 0.99999.

Mass flow rate was measured using a

thermal flowmeter whose error is $\pm 1\%$ readings adding 0.5% full span. A diffusion infrared photosensor was used to obtain information on the rotational speed of the disc, while a YK-23 intelligent revolution meter was used to display the rotational speed ω . Error of rotational speed measurement was ± 8 r/min when the measurement range was 0~3500 r/min.

In order to obtain the flow characteristics in the rotor-stator cavity, a series of pressure taps were located on the surface of the stationary experimental rig (shown in Fig. 3). Tap AP2 was located parallel to the rotating axis facing the inflow to obtain axial total pressure at the inlet. Pressure taps AP8~AP19 were located on the stator surface to obtain pressure distribution. All pressure taps were made of stainless steel tube with 0.8 mm diameter in bore, and their positions are illustrated in Fig. 3.

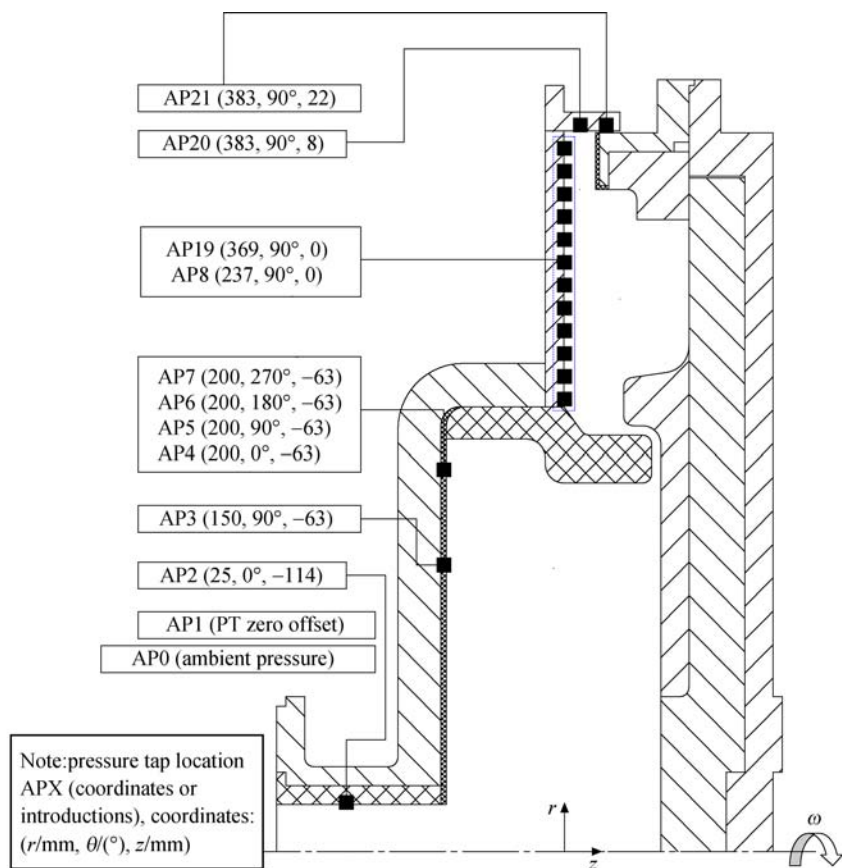


Fig. 3 Instrumented stator

2 Presentation of results

2.1 Pressure distribution on stator

As shown in Fig. 2, for the radial rotor-stator seal in this paper a “pressure inversion effect” occurs inside the cavity when the mass flow rate is relatively large (inertial force is dominant in controlling the flow in the cavity)^[16]. That is to say, the pressure in the rotor-stator cavity increases with the increase of rotational speed.

The effects of rotational speed on the gauge pressure p distribution on the stator at different mass flow rates can be seen in Fig. 4. p_a represents the ambient pressure at exit of cavity, and r represents the radial location. When the mass flow rate was low, centrifugal force was dominant and the inertial force was small. Therefore, the “pressure inversion effect” did not happen inside the cavity. Pressure on the stator decreased with the increase of rotational speed (Fig. 4(a)). When the mass flow rate rose to

500 kg/h and the rotational speed was lower than 2000 r/min, the “pressure inversion effect” began to appear. In other words, pressure on the stator increased along with increasing rotational speed (Fig. 4(b)). If the mass flow rate continued to rise, exceeding 700 kg/h for instance, the “pressure inversion effect” would be obvious (Fig. 4(c)).

Variation of the gauge pressure distributions on the surface of the stator with the change of mass flow rate at different rotational speed can be seen in Fig. 5. The gauge pressure increased monotonously with the increase of mass flow rate. It can be seen from each rotational speed that there was minimum value of gauge pressure at $r \approx 0.261$ mm, whereas the mass flow rate at which the minimum values were reached increased with the increase of rotational speed. This phenomenon probably depended on the relative magnitudes of the inertial force of air inflow and the viscous force from the rotating disk which dominated the flow structure in the cavity.

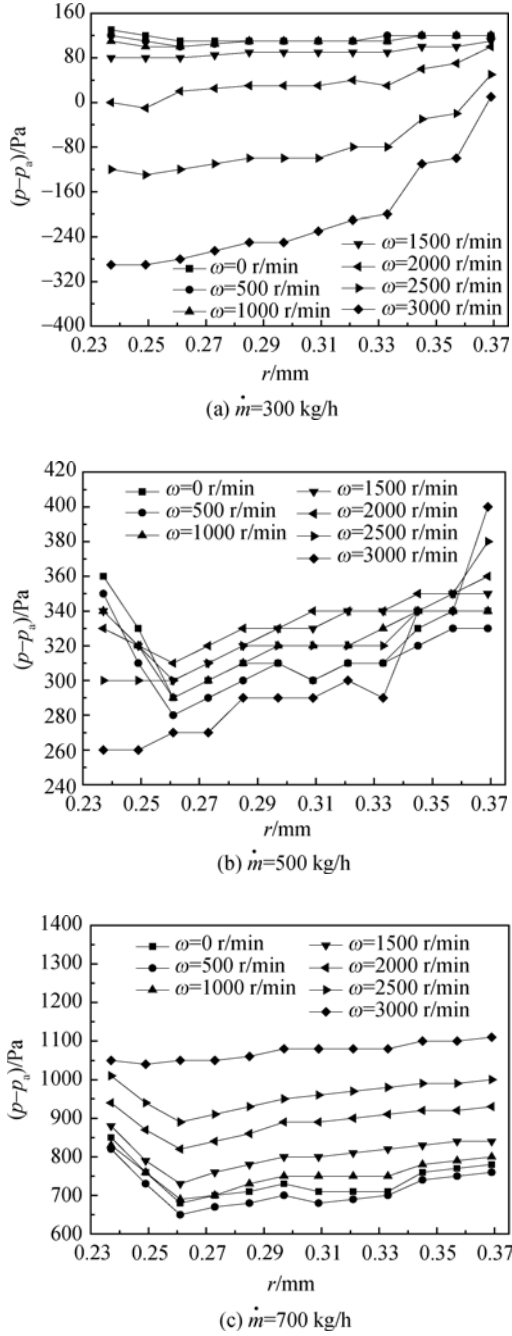


Fig. 4 Variation of gauge pressure distribution with rotational speed

Zhao^[17] numerically studied the effects of inertial force and viscous force on the flow structures in the cavity. The results showed that flow structures in the cavity could be divided into three types: inertial force dominating region, codetermined region and viscous force dominating region (Fig. 6, R is the maximum radius of cavity).

When the rotational speed is relatively low and the mass flow rate is relatively high, the

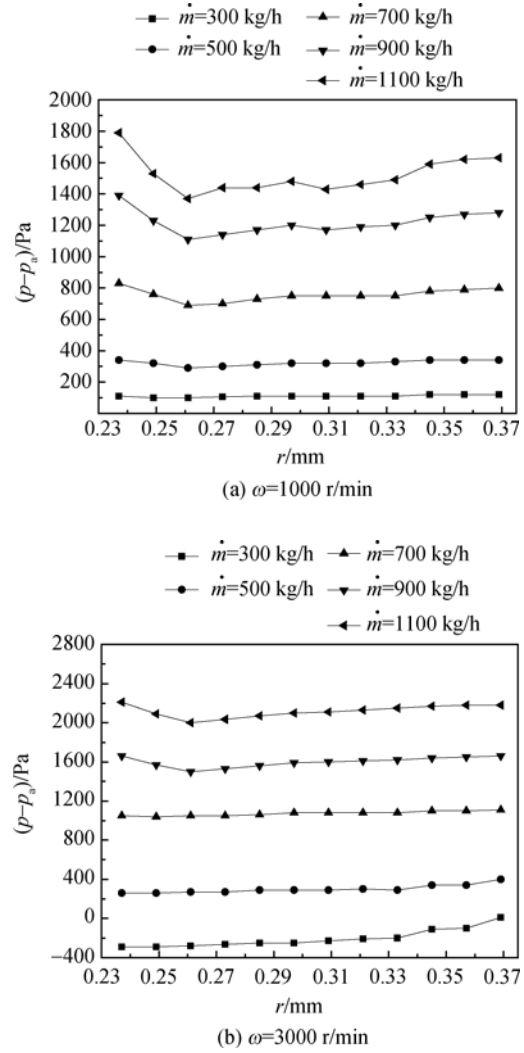
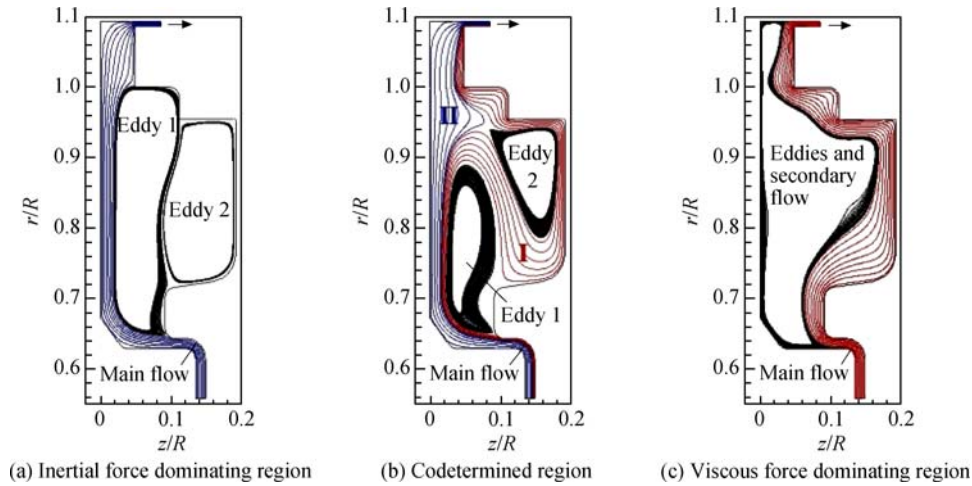


Fig. 5 Variation of gauge pressure distribution with mass flow rate

inertial force of the air inflow is dominant in terms of controlling the flow in the cavity. Thus air impinged onto the low-radius of the stator surface while entering the cavity, then flow radially outwards. At the top section of the impingement area, the flow velocity rose, and then its dynamic pressure increased and static pressure decreased (Fig. 6(a)). The radial location $r \approx 0.261 \text{ mm}$ ($r/R \approx 0.746$) is just within the area where the flow velocity rose. When the rotational speed was relatively high and the mass flow rate was relatively low, the viscous force of the rotating disk was dominant. Inflow into the cavity did not impinge onto the stator, but flow radially outwards adhering to the rotor. At this moment, flow structures near the stator are mainly vortexes and secondary flow (Fig. 6(c)).

Fig. 6 Three types of flow structure^[17]

In the meantime, the velocity did not increase remarkably at $r/R \approx 0.746$, and then there were no minimum values of static pressure at this radial position. When the inertial force and the viscous force are dominant together, the flow structure in the cavity is shown in Fig. 6(b).

The parameter to compare the relative magnitude of the inertial force from the air inflow and the viscous force from the rotating disk is turbulent flow parameter λ_t , which was often used by Owen and Rogers^[18] and many other researchers. It is defined as

$$\lambda_t = C_w Re_\omega^{-0.8} \quad (2)$$

Where $C_w = m/(\mu b)$, μ is dynamic viscosity, and b is outer radius of cavity; m is mass, and C_w is the dimensionless mass flow rate. $Re_\omega = \rho \omega b^2 / \mu$, where ρ is density of air. According to Eq. (2), it is easy to see that small λ_t represents a relatively small inertial force and a relatively large viscous force. When λ_t is large, the situation is reversed.

2.2 Dimensionless pressure distribution on stator

Variation of the dimensionless pressure distribution with Re_ω at different C_w can be seen in Fig. 7 (curves when $C_w \approx 2.21 \times 10^4$ which is not presented are similar). The dimensionless pressure is the ratio of the gauge pressure $p - p_a$ on stator and dynamic pressure $0.5 \rho \omega^2 R^2$ on rotor. It is noteworthy that after non-dimensionalization of the gauge pressure on the stator, the “pressure inversion effect” did not appear in

Fig. 7(a) and Fig. 7(b); that is, the dimensionless pressure decreased with the increase of Re_ω . Consequently, if studies are aimed at the effects of rotation on pressure distribution, it is not appropriate to use dynamic pressure $0.5 \rho \omega^2 R^2$ to non-dimensionalize the gauge pressure. Furthermore, it can also be seen from Fig. 7 that the minimum value rose again at the dimensionless radial position $r/R \approx 0.746$ ($r = 0.261$ mm) when Re_ω is small (λ_t is large). Again, Re_ω at which the minimum values arise increases with the increase of the dimensionless mass flow rate.

It can be concluded from the analysis above that the appearance of the minimum value at $r/R \approx 0.746$ indicated whether the air under the balanced control of the inertial force and viscous force flowed inclined to the stator or not. Thus there is a critical value of turbulent flow parameter $\lambda_{t,c}$ or a $\lambda_{t,c}$ range to determine when the flow pattern changed. When $\lambda_t > \lambda_{t,c}$, inflow entering the rotor-stator cavity flows along the stator. Because of the low resolution of the dimensionless gauge pressure in the radial direction and the limited range of λ_t , it is difficult to estimate the exact value of $\lambda_{t,c}$. But according to the gauge pressure distribution on the stator when $\dot{m} = 300$ kg/h ($C_w \approx 1.32 \times 10^4$), the minimum value arises at $r = 0.261$ mm ($r/R \approx 0.746$) when rotational speed $\omega = 1000$ r/min ($Re_\omega \approx 0.845 \times 10^6$) in Fig. 4(a) and Fig. 7(a), so it can be reckoned that the critical turbulent flow parameter

$\lambda_{t,c} \approx 0.24$.

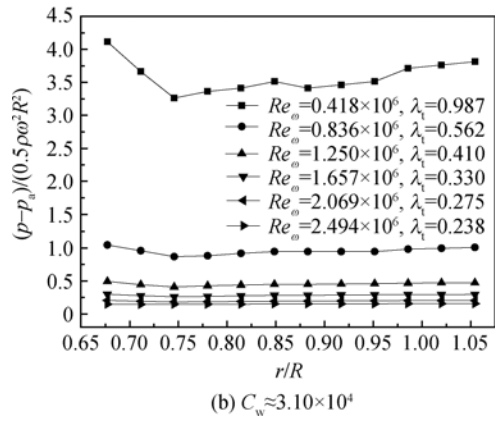
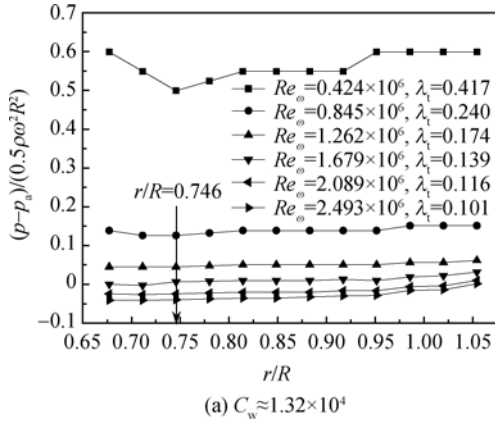


Fig. 7 Variation of dimensionless pressure with Re_ω

Variation of the dimensionless pressure distribution with C_w at different Re_ω is illustrated in Fig. 8. Although the exact value of $\lambda_{t,c}$ cannot be obtained in Fig. 8, the minimum values of dimensionless pressure (at $r/R \approx 0.746$) are all under the condition that $\lambda_t > 0.24$.

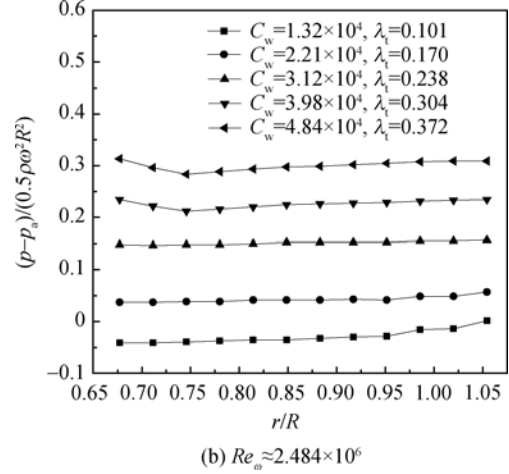
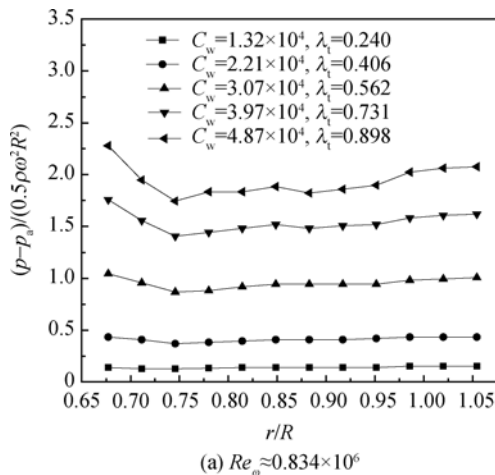


Fig. 8 Variation of dimensionless pressure with C_w

2.3 Flow resistance coefficient of whole cavity

In this paper, the flow in a rotor-stator cavity with a cooling air inlet at low radius of the stator, which is similar to the flow in a rotor-stator system with a central inflow, can be seen as one throttle unit. The overall pressure loss of this kind of cavity can be evaluated using the flow resistance coefficient ξ , which is defined as

$$\xi = \frac{\Delta p}{0.5 \rho \omega^2 R^2} = \frac{p_{in}^* - p_a}{0.5 \rho \omega^2 R^2} \quad (3)$$

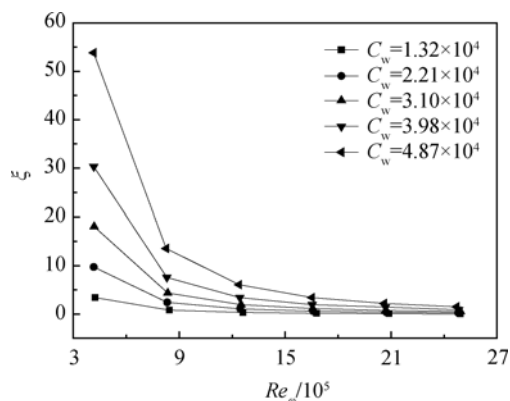
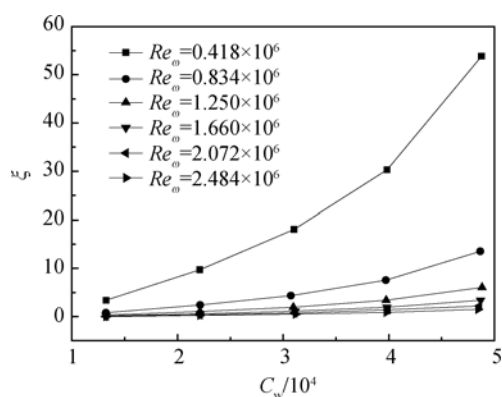
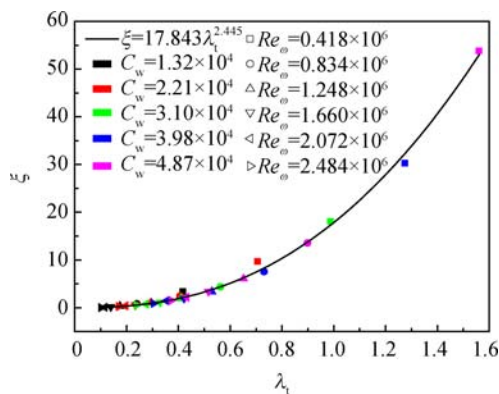
where the denominator indicates the dynamic pressure of the rotating disk; Δp is the overall pressure difference between the inlet and outlet of the total cavity; p_{in}^* is the total pressure at the inlet.

Variation of ξ with C_w (or Re_ω) is shown in Fig. 9 (or Fig. 10). According to Fig. 9 and Fig. 10, ξ increased monotonously with C_w , and decreased monotonously with Re_ω . The experimental results can be fitted into the following relationship:

$$\xi = 73.597 C_w^{0.367} Re_\omega^{-2.001} \quad (4)$$

The experiments were conducted for $1.3 \times 10^4 \leq C_w \leq 5 \times 10^4$ and $4 \times 10^6 \leq Re_\omega \leq 2.5 \times 10^6$ and the correlation coefficient of the fitting Eq. (4) is 0.9969. The relative error of Eq. (4) is $\pm 4.5\% \sim \pm 14\%$.

The effect of λ_t on ξ is given in Fig. 11. It is obvious that ξ increased mainly with the increase of λ_t , but fluctuation exists to some extent. The slight fluctuations in the curves are similar to the

Fig. 9 Variation of ξ with Re_ω Fig. 10 Variation of ξ with C_w Fig. 11 Variation of ξ with λ_t

former argument that λ_t cannot totally control the total pressure loss in the cavity. If the slight effect of C_w or Re_ω on the relationship between ξ and λ_t is neglected, the experimental results can be fitted as follows:

$$\xi = 17.843\lambda_t^{2.445} \quad (5)$$

Experiments were conducted for $0.1 \leq \lambda_t \leq 1.6$, and the correlation coefficient of the fitting Eq. (5) is 0.9966. The relative error of Eq. (5) is $\pm 1.9\% \sim \pm 6.0\%$.

3 Conclusions

Experiments on the flow characteristics of a rotor-stator cavity with a low-positioned air inflow were carried out, and pressure distributions on the stator surface as well as flow resistance coefficients are obtained.

1) When the mass flow rate is large ($\dot{m} > 700 \text{ kg/h}$), the pressure on the stator goes up with the increase of rotational speed, and a “pressure inversion effect” appears in the cavity. The radial pressure increases with the increase of the mass flow rate as former researchers found.

2) The dimensionless pressure increases monotonously with C_w and decreases monotonously with Re_ω . The dimensionless pressure distributions are found to be mainly controlled by λ_t . When $\lambda_t > 0.24$, a minimum dimensionless pressure on the stator at about $r/R = 0.746$ arises.

3) The flow resistance coefficients in the cavity are also found to be mainly controlled by λ_t . For engineering application, it can be estimated by Eq. (5).

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