Experimental and Numerical Investigation of Micro Tip Injection in a High-Speed Axial Compressor Rotor

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Abstract: A series of experiments and numerical simulations are carried out in a high-speed axial compressor to systematically investigate the influence and underlying flow mechanisms of micro tip injection on enhancing compressor stability. Different geometric structures of micro tip injection have been investigated, including the axial positions of injector port, injected mass flow rate and injector diameter. First, seven designed micro tip injection structures and one solid wall casing are tested in the compressor test rig to elucidate the influence of different micro tip injection parameters on the compressor stability. Then, numerical simulations are conducted to analyze the underlying flow mechanisms of micro tip injection with different design parameters on enhancing the compressor stability. The experimental and numerical investigation reveal that when the injection port is located upstream of the low-speed region, the compressor stability is significantly enhanced. The tip injection with larger injected mass flow can obtain higher stall margin improvement. Smaller injector diameter produces higher injection momentum and velocity, contributing to greater improvement on the compressor stability.

Key words: axial compressor; high speed; stability; micro tip injection; flow mechanism; experiment; numerical simulation

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0 Introduction

Aeroengines are developing towards higher thrust-to-weight ratios, lower fuel consumption rates, and higher stability, demanding more stringent design and development standards for compressors. Increasing the rotor tip speed is one of the primary measures to enhance the compressor's work capacity. However, as the rotor tip speed increases, the intensity of the rotor tip leakage flow and shock also increases, and the low-velocity fluid generated by their interaction poses a threat to the compressor stability. Therefore, while the compressor is developing towards higher loads, improving the compressor stability remains an urgent challenge today[1-2].

Tip injection[3] is one of the well-known flow control technologies to enhance the compressor stall margin. The high-pressure air injects into the blade tip region, and it changes the flow conditions near the tip region, so the compressor stability is enhanced. Tip injection can enhance the compressor stability without detriment to the compressor efficiency. Due to the potential benefits of tip injection, it has attracted significant attention from researchers. Tip injection has been confirmed to effectively extend the stall margin of axial compressor in most of studies. Day[4] conducted an experimental study on a four-stage low-speed axial compressor, and found that for both spike-stall and modal stall, the tip injection could improve the compressor stall margin. Strazisar et al.[5] conducted an experimental study on a six-stage high-load axial flow compressor, and found that the tip injection could improve both the compressor stability and the total pressure

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ratio at 78% design speed. Researchers have conducted studies to investigate the influence of different tip injection design parameters on the compressor stability. Weigl et al.[6] designed three tip injection schemes with different injected yaw angles, among which the tip injection scheme with an injected yaw angle of $-15°$ obtained the greatest stall margin improvement (SMI). Cassina et al.[7] investigated the effect of different injection mass flow rates (1.19%, 2.7% and 4.3% of overall mass flow) on the compressor stability, and found that a higher injection mass flow rate led to better compressor stability. Ding et al.[8] found that as the injection axial position gradually moves upstream, the ability of tip injection to improve the compressor stability shows a trend of first increasing and then decreasing, and there was an optimal injection axial position for the largest SMI. Numerous studies have also been conducted to understand the stability-enhancing mechanism of tip injection. Suder et al.[9] conducted a parametric study on NASA (National Aeronautics and Space Administration) Rotor 35, and they concluded that the SMI obtained by the tip injection was positively correlated with the injection velocity, and the high-speed injection flow reduced the attack angle and blade load, so the compressor stability was improved. According to Ref.[10], tip injection can enhance the airflow at the rotor leading edge, suppressing the negative impacts of tip clearance backflow and tip leakage vortex formation. Khaleghi et al.[11-12] further suggested that the improvement of the compressor stability was related to the increase in injection momentum, and a higher injection momentum could result in a greater improvement of the compressor surge margin. Chen et al.[13] found that the tip injection could suppress the development of stall cell by performing full-passage numerical simulations on NASA Rotor 35. Wu et al.[14] conducted unsteady numerical simulations on NASA Rotor 35 and concluded that the excitation of the low-speed region near the blade tip by the injection flow was the primary factor to improve the compressor stability, while the unloading effect of the tip injection on the blade tip loads played a secondary role.

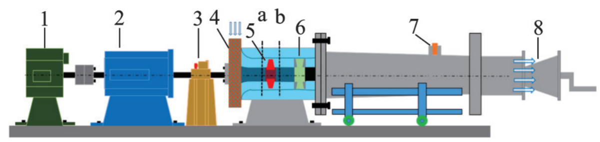
With the extensive researches in the field of tip injection, Nie et al.[15] proposed the concept of micro tip injection, and carried out experimental research on a three-stage low-speed axial flow compressor, and found that a 5.83% SMI was obtained by using an injection mass flow rate of 0.056% of the main flow. Tong et al.[16-17] investigated the micro tip injection in a low-speed axial flow compressor. The injected mass flow rate of micro tip injection was significantly reduced, yet the tip injection still remained great stability-enhancing ability. Besides, they measured the dynamic pressure field in tip clearance, and pointed out that the injection momentum further affected the compressor stability margin compared to the injection mass flow rate. Geng et al.[18-20] suggested that for the subsonic axial compressor, the micro tip injection affected the initiation position of the tip leakage flow and reduced the flow losses in the rotor tip passage, while for the transonic axial compressor, the micro tip injection suppressed the unsteady fluctuation of the tip leakage flow and made the tip leakage vortex trajectory shift to the blade suction surface. Li et al.[21] proposed an automatic stability control method using solid wall pressure signals as the feedback, which achieved nearly an equal SMI (20%) as the steady tip injection with approximately one-fifth of steady injection mass flow. In actual application, the injected flow is bled from the rear of the real compressor. However, a large amount of injected mass flow will lead to significant efficiency losses in the whole compression system. It is essential to minimize the amount of injected mass flow, and micro tip injection with less injected energy can well deal with this problem. In the previous studies about micro tip injection, there also exist some shortcomings. Most studies have only investigated micro tip injection in a low-speed axial flow compressor. However, modern high-load compressors usually operate at extremely high rotational speeds. Therefore, the parametric study about micro tip injection in a high-speed axial compressor needs to be further developed and refined. Additionally, the stability-enhancing mechanism of micro tip injection in high-speed axial compressors also needs to be deeply understood.

In the present study, an experimental and numerical investigation is conducted to evaluate the influence of micro tip injection on a high-speed axial compressor stability. Firstly, the parametric studies of different injection axial positions, injected mass flow rates, and injector diameters are systematically studied by experimental methods, and the influence of different design parameters on the compressor stability are summarized. Then, the flow mechanisms of different design parameters impacting the compressor stability are explored using unsteady numerical simulations. This paper aims to address the limitations of previous studies, further elucidate the underlying flow mechanisms of different parameters impacting the stability-enhancing ability of micro tip injection, and provide support for the development and application of micro tip injection in real compressors.

1 Research Object and Research Methods

1.1 Research object

In this paper, the subsonic high-speed axial compressor rotor at Northwestern Polytechnical University is selected as the research object. Fig.1 shows the cross-sectional diagram of the compressor test rig. The test rig consists of a DC motor, an accelerator, a torque meter, a test section, an outlet pipe, an orifice flowmeter, and a throttle valve. The intake direction is radial. The power comes from the 350 kW DC motor with a rated speed of 1 250 r/min. To meet the operational speed requirements of the compressor rotor, an accelerator with a gear ratio of 15∶1 is used to increase the rotational speed of the rotor. The maximum rotational speed of the rotor is 15 200 r/min. An NC-3 torque meter is installed between the accelerator and the test section to monitor the torque and rotational speed of the rotor. During the experiment, air enters the compressor radially through a metal dust screen, passing through the test section, outlet pipe, and throttle valve before being discharged into the laboratory atmosphere. The test section is designed to accommodate inlet guide vane, rotor, and stator for flexible assembly and disassembly. Multiple test ports are located at various axial and circumferential positions on the external surface, meeting the present compressor performance testing needs. Table 1 shows the basic design parameters of the compressor rotor. The present experimental and numerical investigation are all carried out at 70% of the maximum rotational speed.



1—DC motor; 2—Accelerator; 3—Torque meter; 4—Intake steel cover; 5—Rotor disk; 6—Stator disk; 7—Orifice flowmeter; 8—Throttle valve; a—Rotor inlet probe location; b—Rotor outlet probe location

Fig.1 Diagram of the compressor test rig

Table 1 Basic design parameters of the compressor rotor

Parameter	Value
Mass flow rate/(kg•s^{-1})	5.6
Adiabatic efficiency	0.905
Total pressure ratio	1.245
Rotational speed/(r•min^{-1})	15 200
Hub-tip radius ratio	0.61
Number of rotors	30
Aspect ratio	1.94
Tip gap/mm	0.3

1.2 Experiment measurement method

Before the experiment, the local atmospheric pressure and temperature are measured using a mercury pressure gauge and a mercury temperature gauge, respectively. And the actual rotational speed is calculated according to the conversion rotational speed required by the experiment. The rotational speed and torque are measured by the torque meter. Two three-hole probes are used to measure the inlet and outlet total pressure, static pressure, and airflow angle, respectively. The electronic pressure scanning valve is used for pressure signal acquisition. The inlet measurement axial position is located at four times the axial chord length upstream of the rotor leading edge, and the outlet measurement axial position is located at two times the axial chord length of the rotor trailing edge. The probes can be moved along the radial direction, and the airflow parameters in the seven radial positions are measured in the experiment. Each radial position is collected ten times, and the average-value is used as the measurement result of this radial position. Finally, the rotor total performance is calculated by averaging the flow parameters at the seven radial positions. An orifice plate is installed in the middle of the outlet section to measure the mass flow rate, and the mass flow rate is changed by adjusting the throttle valve.

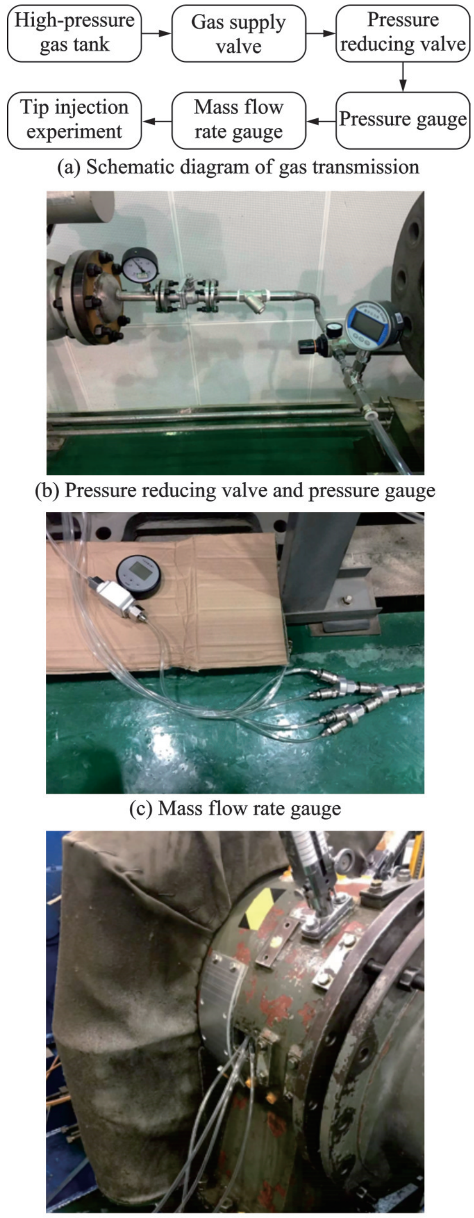
If the temperature increment is used to calculate the compressor efficiency, the absolute error needs to be controlled within 0.1 ℃. However, the temperature increment is small for the rotor, and it is difficult to quickly obtain the high precision temperature in the experiment, so the compressor torque and other parameters are used to calculate the compressor isentropic efficiency η^*. The specific calculation formula is shown in Eqs.(1—3), where $L^*_{ad,\mathrm{K}}$ represents the isentropic compression work, L_K the actual work consumed, k the specific heat ratio, R the gas constant, T^*_1 the inlet total temperature, π^*_K the total pressure ratio, T the torque, n the rotational speed, and m the mass flow rate.

$$\eta^* = \frac{L^*_{ad,\mathrm{K}}}{L_\mathrm{K}} \tag{1}$$

$$L^*_{ad,\mathrm{K}} = \frac{k}{k-1} R T^*_1 \left[(\pi^*_\mathrm{K})^{\frac{k-1}{k}} - 1 \right] \tag{2}$$

$$L_\mathrm{K} = \frac{100\, Tn}{955\, m} \tag{3}$$

In the tip injection experiment, the high-pressure gas comes from the gas tank, with a maximum gas pressure of 1.6 MPa, and the gas tank transports the high-pressure gas to each laboratory through gas pipelines. The whole tip injection device consists of a high-pressure gas tank, a gas supply value, a pressure reducing valve, a pressure gauge, a flowmeter, and several gas pipelines. The flowmeter can measure the volumetric flow rate and temperature of the airflow. By combining these parameters with the pressure value from the pressure gauge, the mass flow rate of the injected flow can be calculated. After the high-pressure gas is filtered, reduced and stabilized pressure, it is divided into the experimental pieces according to the experimental demand. Fig.2 shows the diagram of high-pressure gas transmission.



(a) Schematic diagram of gas transmission

(b) Pressure reducing valve and pressure gauge

(c) Mass flow rate gauge

(d) Airflow pipeline

Fig.2 Diagram of high-pressure gas transmission

1.3 Numerical method and verification

The grid is generated by Autogrid5 module of NUMECA software. The compressor rotor passage adopts the HOH-type grid topology. The rotor tip clearance adopts the butterfly grid topology. The inlet and outlet extensions adopt the H-type grid topology. Fig.3 shows the grid diagram of calculation domain. Based on extensive previous research, our research team obtains a very reliable single-passage rotor grid for numerical simulations[22]. The grid point distribution in the rotor passage is 25×69×209 (circumferential×radial×around the blade). The grid point distribution in the rotor tip clearance is 17×17×209. The total number of single-passage grids is about 970 000, and the three-passage grid is obtained by replicating the single-passage grid. To ensure the dimensionless distance y^+ is less than 1, the distance between the first layer grid and the solid wall is set to 5×10^{-6} m.

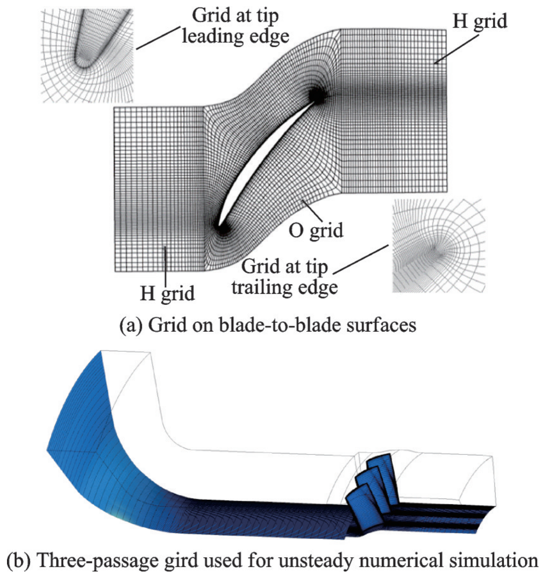


Fig.3 Grid diagram of calculation domain

The interactive geometry modeler and grid generator (IGG) module of NUMECA software is used to generate the grid of tip injection section. The injector port adopts a butterfly-type grid topology. The injector inlet boundary condition is set to the steady mass flow rate, and the injector is connected to the casing surface through the "full non-matching" method.

Using the ANSYS CFX software, the unsteady numerical simulations are carried out in this paper, and the k-omega turbulence model is selected to solve the 3D Navier-Stokes equation. The temporal discretization scheme used for numerical calculations is dual time stepping, with 20 physical time steps for each blade passage and 20 inner iterations numbers. In the numerical calculations, the inlet/out extension section and the stator passage are set to the stationary domain, and the rotor passage is set to the rotational domain. The inlet boundary conditions are set as follows: The total pressure is 101 325 Pa, and the total temperature is 288.15 K. The outlet boundary condition defines as the average static pressure. By gradually increasing the outlet average static pressure, the complete compressor performance curves are obtained, and the different in outlet average static pressure between the neat-stall point and the first divergent numerical calculation is 50 Pa.

Fig.4 compares the compressor total performance for the numerical simulation and experiment. The numerical simulation accurately predicts the compressor total performance and the distribution of total pressure ratio along the radial direction, and the relative error between the numerical results and the experimental values is smaller than 2%, so the grid independence verification and the turbulence model verification are not shown in this paper. However, there is still a certain error between the experimental values and the numerical results, which may

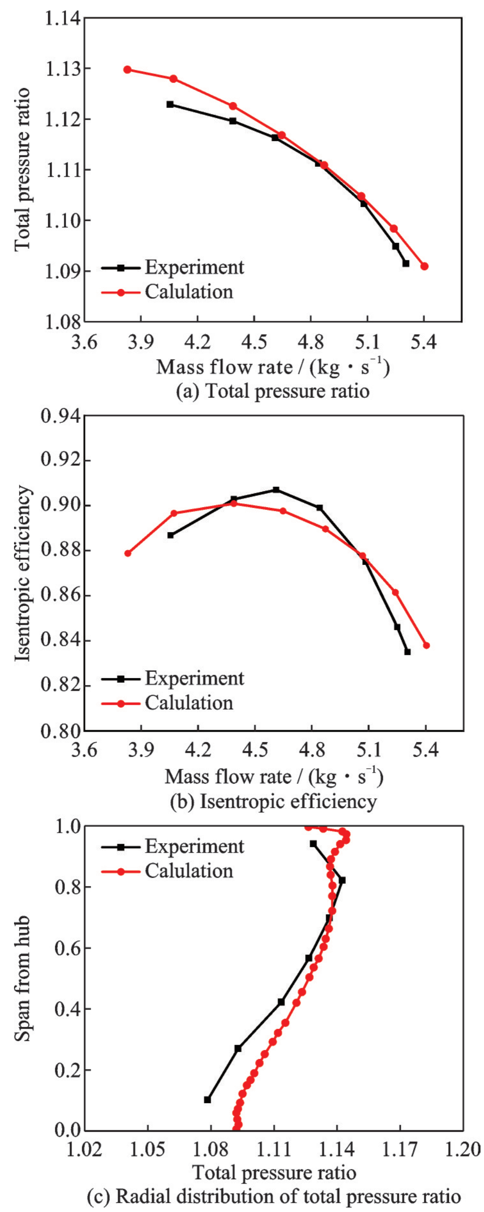


Fig.4 Comparison between numerical and experimental results

be caused by the following reasons: (1)The present numerical simulation techniques cannot fully capture the complex flow inside the compressor, and there will be inevitable errors in experimental measurements; (2)the compressor geometric model used in numerical simulations cannot be completely consistent with the real compressor test rig.

2 Micro Tip Injection Experimental Results and Discussion

In this section, the micro tip injection structures with different design parameters are examined in the high-speed axial compressor test rig. The influence of the different injection axial positions, injected mass flow rates and injector diameters on the compressor stability are analyzed and summarized.

2.1 Micro tip injection structures

Fig.5 shows the diagram of the micro tip injection structure. The injection axial position is the axial distance from the injector port to the rotor tip leading edge. When the injector port is located downstream of the rotor tip leading edge, the injection axial position is a positive value. Using the circle as the injector port cross-sectional shape, four micro tip injection structures are evenly arranged along the circumferential direction. The injected yaw angle and injected skew angle are constant, both set at 0° and 15°, respectively. The injected mass flow rate of each injector is normalized by the compressor design mass flow rate. Seven micro tip injection structures and one solid wall casing test pieces are designed. Table 2 shows the specific design parameters for micro tip injection test pieces, where the No.1 experimental piece represents the solid-wall casing (SW) and C_a represents the rotor tip axial chord length. Fig.6 presents the physical images of part of the experimental test pieces.

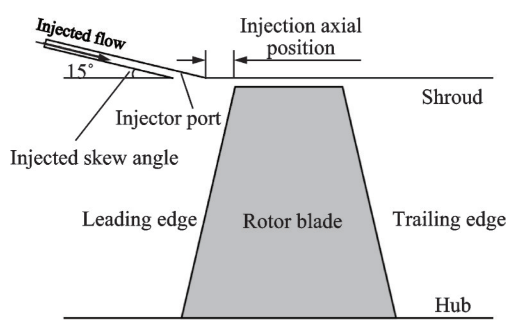
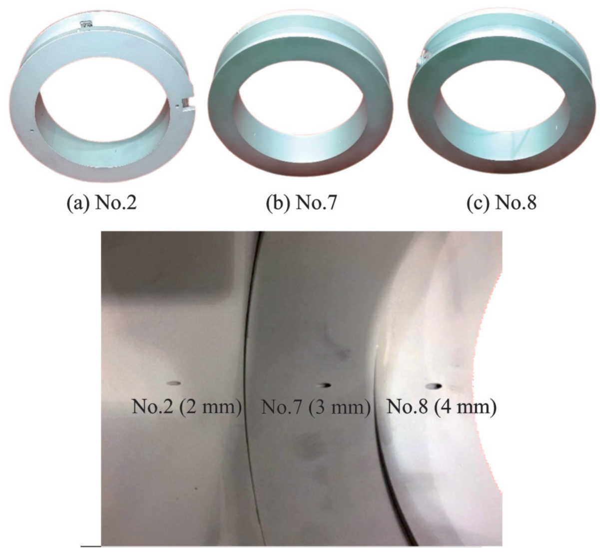


Fig.5 Diagram of the micro tip injection structure

Table 2 Specific design parameters for different experimental test pieces

Number	Injector diameter/mm	Injection axial position	Normalized injected mass flow rate for one injector/%
1	—	—	—
2	2	$12\%C_a$	0.024
3	2	$-21.3\%C_a$	0.024
4	2	$-54.4\%C_a$	0.024
5	2	$12\%C_a$	0.048
6	2	$12\%C_a$	0.012
7	3	$12\%C_a$	0.024
8	4	$12\%C_a$	0.024



(d) Comparison of micro tip injection test pieces for different injector diameters

Fig.6 Images of part of experimental pieces

2.2 Influence of different injection axial positions on the compressor stability

Based on an injector diameter of 2 mm and only an injected mass flow rate of 0.024% of the design mass flow rate of the compressor for one injector, three experimental pieces with injection axial positions of $12\%C_a$, $-21.3\%C_a$, and $-54.4\%C_a$ are discussed (corresponding to the No.2, No.3 and No.4 in Table 2, respectively).

Fig.7 shows the compressor performance curves for the experimental micro tip injection structures with different injection axial positions. Due to the limitation of experimental measurement condition, the energy introduced by the injector cannot be measured. Therefore, the compressor total performance discussed in this paper does not take into account the energy brought by the injected flow. As

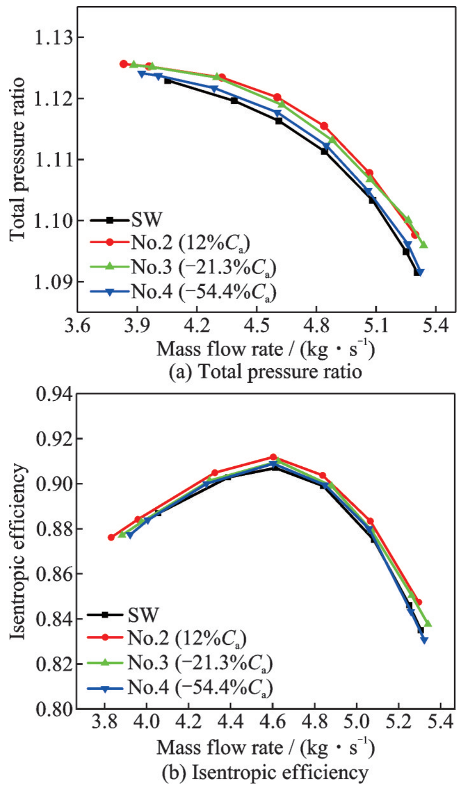


Fig.7 Experimental compressor performance curves for different injection axial positions

shown in Fig.7, the three micro tip injections with different injection axial positions improve the compressor stability to varying degrees, and the total pressure ratio and efficiency are also improved. Among them, the tip injection with the injection axial position of 12%C_a has the strongest ability to improve the compressor stability, and the near-stall mass flow rate is reduced to about 3.84 kg/s. And the improvement in the total pressure ratio and efficiency is also the most significant. Compared to the previous studies, the improvement in the compressor stability is considerable. With the gradual forward movement of the injection axial position, the ability of tip injection to improve the compressor stability gradually weakens, while the improvement in the total pressure ratio and efficiency gradually decreases.

To quantify the effects of different micro tip injections on the compressor stability and performance, the SMI and peak efficiency improvement (PEI) are introduced in the paper. The SMI and PEI are defined by Eq.(4) and Eq.(5), where π^* represents the compressor total pressure ratio and m the mass flow rate. The subscript of injection represents the tip injection. The subscript of SW represents the solid-wall casing and the subscript of s represents the near-stall condition.

$$\mathrm{SMI}=\left[\left(\frac{\pi^*_{\text{injection}}}{m_{\text{injection}}}\right)_{\mathrm{s}}-\left(\frac{\pi^*_{\mathrm{SW}}}{m_{\mathrm{SW}}}\right)_{\mathrm{s}}\right]\Bigg/\left(\frac{\pi^*_{\mathrm{SW}}}{m_{\mathrm{SW}}}\right)_{\mathrm{s}}\times 100\% \tag{4}$$

$$\mathrm{PEI}=\left[(\eta^*_{\text{peak}})_{\text{injection}}-(\eta^*_{\text{peak}})_{\mathrm{SW}}\right]\times 100\% \tag{5}$$

Table 3 gives the values of SMI and PEI for tip injections with different injection axial positions. As shown in Table 3, the micro tip injections not only improve the compressor stability, but also improve the peak efficiency. The tip injection with the injection axial position of 12%C_a obtains the largest SMI and PEI, with values of 6.07% and 0.49%, respectively. The farther the injection axial position is from the rotor tip leading edge, the smaller the benefits brought by the tip injection. Due to the size limit of the compressor test rig, the injection axial position did not move further into the rotor tip passage. Therefore, in the practical application of aero-engines, the injection axial position should not be arranged at a long distance upstream of the rotor tip.

Table 3 Values of SMI and PEI for tip injection with different injection axial positions

Number	Injection axial position	m_s/(kg·s^{-1})	SMI/%	PEI/%
1	—	4.054 8	—	—
2	12%C_a	3.831 9	6.07	0.49
3	−21.3%C_a	3.883 4	4.65	0.30
4	−54.4%C_a	3.922 5	3.48	0.20

2.3 Influence of different injected mass flow rates on the compressor stability

Based on an injector diameter of 2 mm and an injection axial position of 12%C_a, three experimental pieces with injected mass flow rates of 0.012%, 0.024%, and 0.048% of the design mass flow rate of the compressor for one injector are discussed (corresponding to the No.6, No.2 and No.5 in Table 2, respectively).

Fig.8 shows the compressor performance curves for three micro tip injection structures with different injected mass flow rates. The three tip injections with different injected mass flow rates all improve the compressor stability. When the injected mass flow rate is too small, the compressor stability

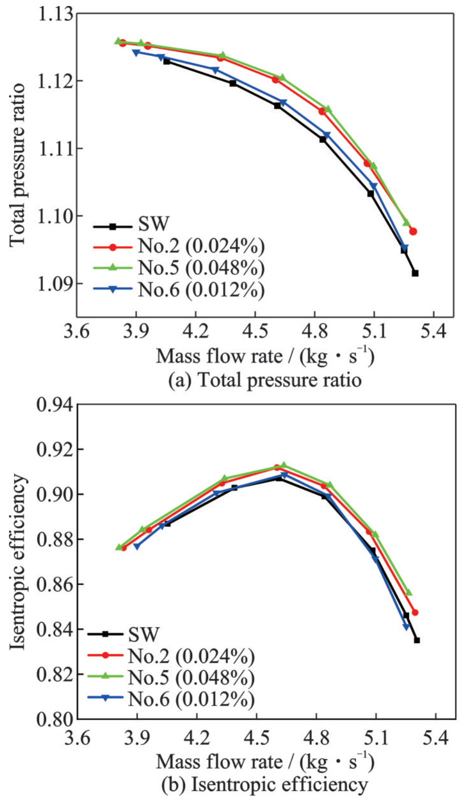


Fig.8 Experimental compressor performance curves for different injected mass flow rates

is not improved much. With the increase of injected mass flow rate, the effect of enhancing the compressor stability is more obvious. However, after the injected mass flow rate increases to a certain value, further increase in the injected mass flow rate results in a minimal improvement in the compressor stable working range.

To quantify the influence of different injected mass flow rates on the compressor stability and performance, Table 4 gives the values of SMI and PEI for tip injections with different jet mass flow rates. The micro tip injection with an injected mass flow rate of 0.048% of design mass flow rate obtains the largest SMI and PEI, with values of 6.71% and 0.59%, respectively. When the injected mass flow rate reaches 0.024%, the influence of further increasing the injected mass flow rate on compressor stability becomes significantly smaller. When the injected mass flow rate increases from 0.012% to 0.024%, the increment of SMI is 1.96%. However, when the injected mass flow rate increases from 0.024% to 0.048%, the increment of SMI is only 0.64%. Therefore, in practical applications, the appropriate injected mass flow rate should be selected

Table 4 Values of SMI and PEI for tip injection with different injected mass flow

Number	Injected mass flow/%	m_s/(kg•s^{-1})	SMI/%	PEI/%
1	—	4.054 8	—	—
2	0.024	3.831 9	6.07	0.49
5	0.048	3.809 6	6.71	0.59
6	0.012	3.899 5	4.11	0.18

to balance the improvement in the compressor stability with the consumption of external high-pressure air.

2.4 Influence of different injector diameters on the compressor stability

Based on an injection axial position of 12% C_a and only an injected mass flow rate of 0.024% of the design mass flow rate of the compressor for one injector, three experimental pieces with injector diameters of 2, 3, and 4 mm are discussed (corresponding to the No.2, No.7 and No.8 in Table 2, respectively).

Fig.9 shows the compressor performance curves for three micro tip injection structures with different injector diameters. The three tip injections all expand the compressor stable operating range.

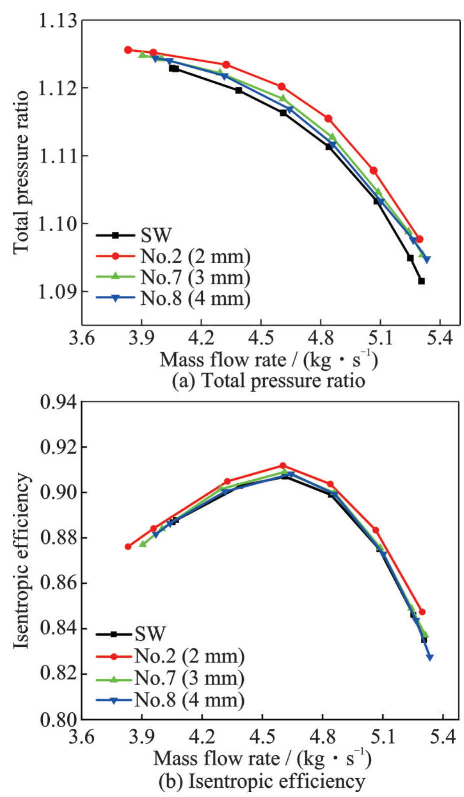


Fig.9 Experimental compressor performance curves for different injector diameters

The improvement in the compressor stability decreases with the increase of injector diameter. The micro tip injection with injector diameter of 2 mm has the greatest ability to improve the compressor stability, while the tip injection with injector diameter of 4 mm has the weakest ability to improve the compressor stability. When the injected mass flow rate is the same, the change of injector diameter impacts the momentum and velocity of the injected flow. The injected flow with higher momentum and velocity achieves a larger SMI. Table 5 gives the values of SMI and PEI for tip injections with different injector diameters. Due to the higher momentum and velocity of the injected flow, the scheme with a nozzle diameter of 2 mm achieves the highest SMI (6.07%).

Table 5 Values of SMI and PEI for tip injection with different injector diameters

Number	Injector diameter/mm	m_s/(kg·s^{-1})	SMI/%	PEI/%
1	—	4.054 8	—	—
2	2	3.831 9	6.07	0.49
7	3	3.904 7	4.02	0.21
8	4	3.969 9	2.28	0.13

3 Numerical Simulation and Analysis of Flow Mechanisms

In section 2, the influences of different injection axial positions, injected mass flow rates and injector diameters on the compressor stability are analyzed based on the experimental results of eight experimental test pieces. The macroscopic rules of the influence of different tip injection parameters on the ability of tip injection to enhance the compressor stability are revealed, but a deeper understanding of the underlying flow mechanisms is necessary. Therefore, the multi-passage unsteady numerical simulations are carried out in this section to further analyze the flow mechanism of the influence of different parameters on the ability of micro tip injection to enhance the compressor stability.

3.1 Research schemes

It is worth noting that the experimental test pieces involved in section 2 are all arranged with four injectors in the circumferential direction. The ratio of the number of rotor passages to the number of injectors is 15∶2. If fifteen rotor passages are used for the unsteady numerical simulations, the total grid number is very large and the unsteady numerical simulation will cost lots of time and computing resource. Considering after the uniform adjustment of the number of injectors, the flow mechanism of different parameter impacting the ability of micro tip injection to enhance the compressor stability can still be analyzed. Therefore, to reduce the cost of time and computing resource on numerical simulations, the number of injectors is changed to 10 in the section 3, while the ratio of the number of rotors passages to the number of injectors is changed to 3∶1. Only three rotor passages need to be used for the unsteady numerical simulations, greatly reducing the cost of time and computing resource.

The following flow field comparisons are discussed at the calculated near-stall point for the solid wall casing. In order to be consistent with the experiment, the energy brought by the injected flow is not considered when calculating the compressor performance.

3.2 Flow mechanisms of different injection axial positions on the compressor stability

In section 2, the influence of different injection axial positions on the ability of tip injection to enhance the compressor stability is explored only in the upstream of the rotor tip leading edge. In this section, based on an injector diameter of 2 mm and an injected mass flow rate of 0.024% of the design mass flow rate of the compressor for one injector, three tip injections with injection axial positions of $0\%C_a$, $33\%C_a$ and $50\%C_a$ are analyzed.

Fig.10 shows the compressor performance curves for three micro tip injection structures with different injection axial positions in numerical simulations. When the injection axial position is located at $0\%C_a$ and $33\%C_a$, the compressor stall margin increases by 7.9% and 8.1%, respectively. When the injection axial position is located at $50\%C_a$, the improvement of compressor stability is marginal. In addition, under the near-stall conditions, the tip injection with the injection axial position of $33\%C_a$ has the highest efficiency, and the tip injection with the injection axial position of $50\%C_a$ has the lowest efficiency improvement.

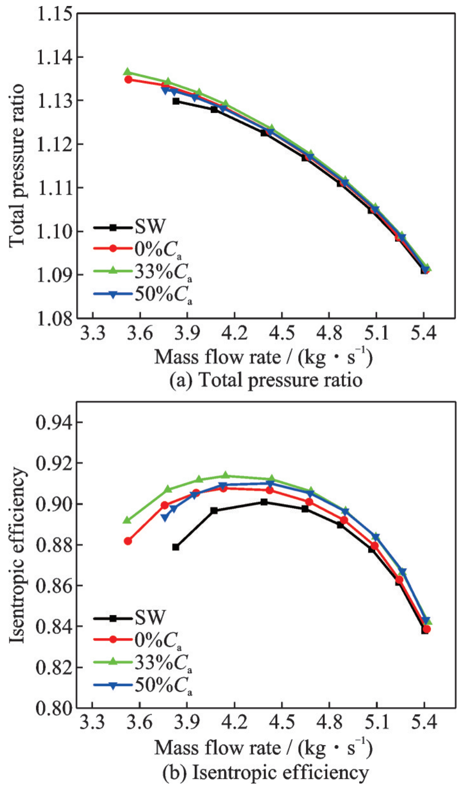


Fig.10 Numerical compressor performance curves for different injection axial positions

Fig.11 shows the distribution of the relative Mach number at 99% blade span under the near-stall condition of SW at the same instant. For the two tip injections with the injection axial positions of 0%C_a and 33%C_a, the injection position is located upstream of the low-speed fluid, and the low-speed fluid in the rotor tip passage is completely eliminated. Therefore, the two tip injections obtain the larger improvement of stall margin and the less compressor efficiency loss. However, for the tip injection with the injection axial positions of 50%C_a, the injection position is downstream of the low-speed fluid. Although the distance between the injector port and the blockage area is not far, there is still an obvious low-speed fluid in the rotor tip passage, so the compressor stability is hardly enhanced.

The impact of tip injection on the compressor efficiency is determined by the reduction of flow losses within the low-speed region and the increase in mixing losses between the injected flow and mainflow. The axial position of the injection has less influence on the degree of mixing losses but has a greater influence on the degree of flow losses in the low-speed region. When the injection position is located upstream of the blockage area, the tip injection is more effective in clearing the blockage area and the ability to reduce the flow loss is stronger. Therefore, the tip injections with the injection axial positions of 0%C_a and 33%C_a have a greater improvement on the compressor stability and efficiency. When the injection position is located downstream of the blockage area, it is difficult for the injected flow to eliminate the low-speed fluid in the rotor tip passage. Therefore, the tip injection with the injection position of 50%C_a has little improvement on the compressor stability and efficiency.

Fig.12 shows the distribution of tip leakage flow streamline under near-stall condition of SW at the same instant. When injection position is located at 33%C_a, the tip injection has the most obvious suppression effect on the tip leakage flow, which significantly reduces the tip blockage. However, when injection positions are located at 0%C_a and 50%C_a, the suppression effects on the tip leakage flow are weaker. According to Fig.11, there should

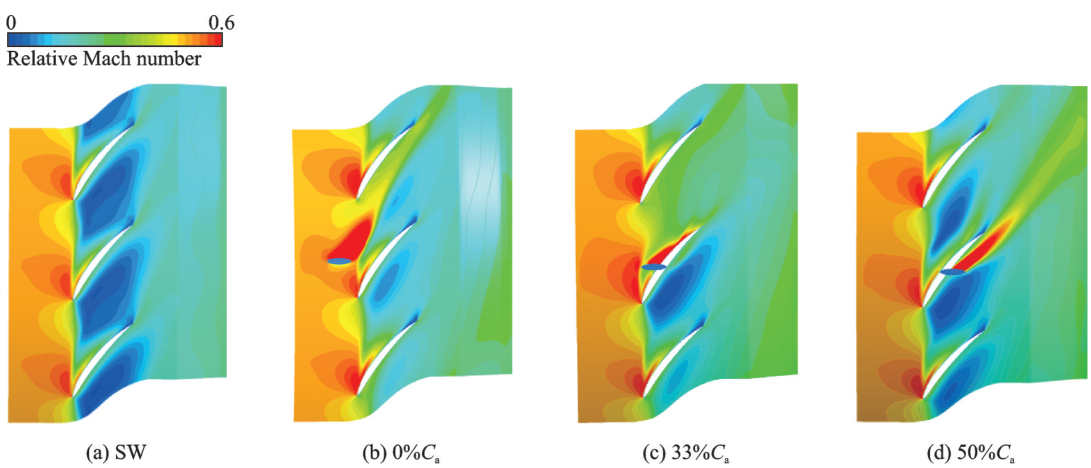


Fig.11 Distribution of the relative Mach number at 99% blade span for different injection axial positions under near-stall condition

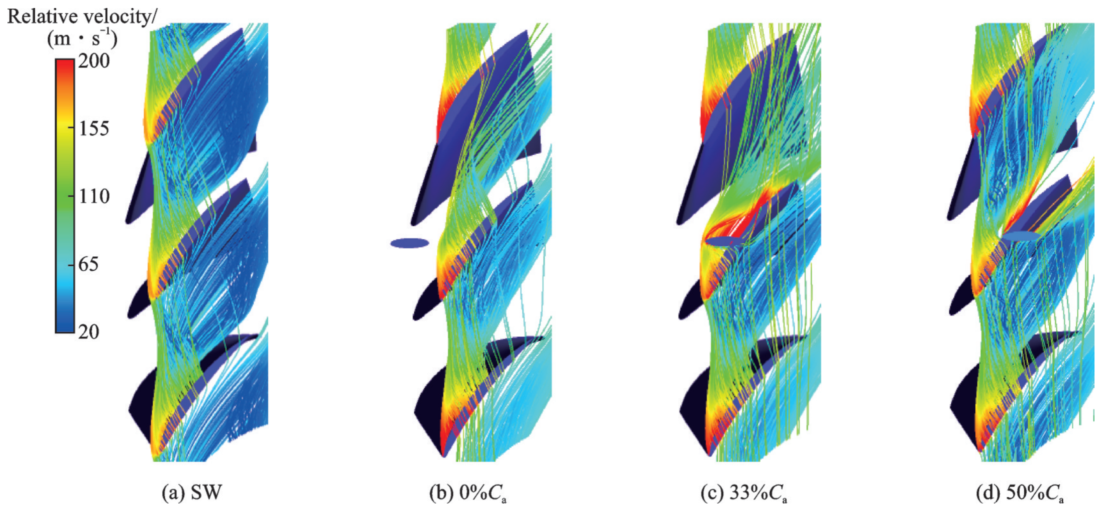


Fig.12 Distribution of tip leakage flow streamline for different injection axial positions under near-stall condition

be an optimal injection axial position that exhibits the best ability to improve compressor stability. This optimal axial position is related to the blockage region in the rotor tip passage.

The diffusion factor can be used to reflect the rotor radial load distribution to a certain degree[9,23]. The diffusion factor (D) is defined by Eq.(6), where w_1 represents the inlet relative velocity, w_2 the outlet relative velocity, Δw_u the variation of tangential velocity, and τ the solidity.

$$D = 1 - \frac{w_2}{w_1} + \frac{\Delta w_u}{2 w_1 \tau} \tag{6}$$

Fig.13 shows the distribution of the rotor diffusion factor along the radial direction under the near-stall condition of SW. After applying the tip injection, the rotor radial load distribution in the 85% to 100% blade span range is significantly reduced. Tip injection with an axial position of 33%C_a results in a more significant reduction on the rotor load, thereby facilitating a wider compressor stable operating range.

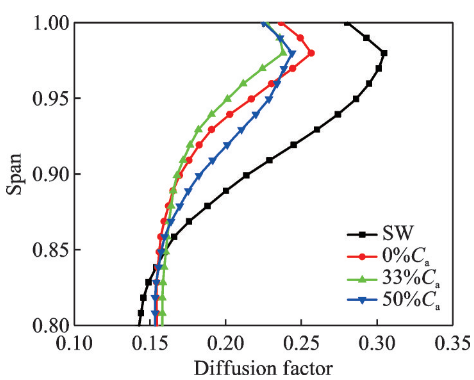


Fig.13 Radial distribution of the diffusion factor for different injection axial positions under near-stall condition

3.3 Flow mechanisms of different injected mass flow rates on the compressor stability

Based on an injector diameter of 2 mm and an injection axial position of 0%C_a, three tip injections with the injected mass flow rate of 0.012%, 0.024% and 0.048% of the design mass flow rate of the compressor for one injector are analyzed. Fig.14 shows the compressor performance curves for three micro

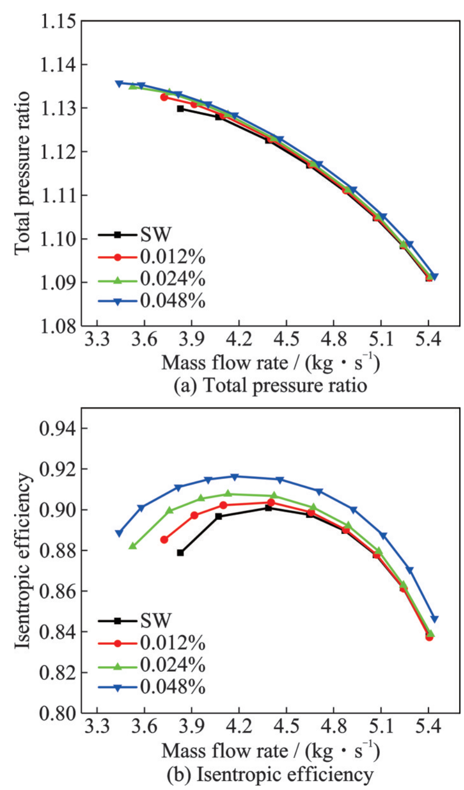


Fig.14 Numerical compressor performance curves for different injected mass flow rates

tip injection structures with different injected mass flow rates in numerical simulations. The three tip injections all improve the stability and efficiency of the compressor. The tip injection with the injected mass flow rate of 0.012% obtains the SMI of 2.7%. The tip injection with the injected mass flow rate of 0.024% obtains the SMI of 7.9%, and the tip injection with the injected mass flow rate of 0.048% obtains the SMI of 10.1%. This shows that the gradual increase of the injected mass flow rate is conducive to improve the compressor stability. Beyond a certain threshold of injected mass flow rate, increasing it further yields diminishing returns in terms of compressor stability enhancement.

Fig.15 shows the distribution of the relative Mach number at 99% blade span under near-stall condition of SW at the same instant. The larger the injected mass flow rate, the smaller the area of low-speed fluid in the rotor tip passage. Due to the small injected mass flow rate and speed, it is not enough to eliminate the low-speed fluid in rotor tip passage for the tip injection with an injected mass flow rate of 0.012%. When the injected mass flow rate increases to 0.024%, the injection velocity increases, and the low-speed region is significantly reduced. When the injected mas flow rate increases to 0.048%, the flow state of tip passage is further improved.

Fig.16 shows the distribution of tip leakage flow streamline under the near-stall condition of SW at the same instant. The tip injections effectively suppress the development of tip leakage flow and reduce the tip blockage. Among them, the tip injection with a injected mass flow rate of 0.048% suppresses the tip leakage flow to the greatest extent and make the tip leakage flow streamline closer to the suction surface. Therefore, the tip injection with a injected mass flow rate of 0.048% obtains the largest stall margin improvement.

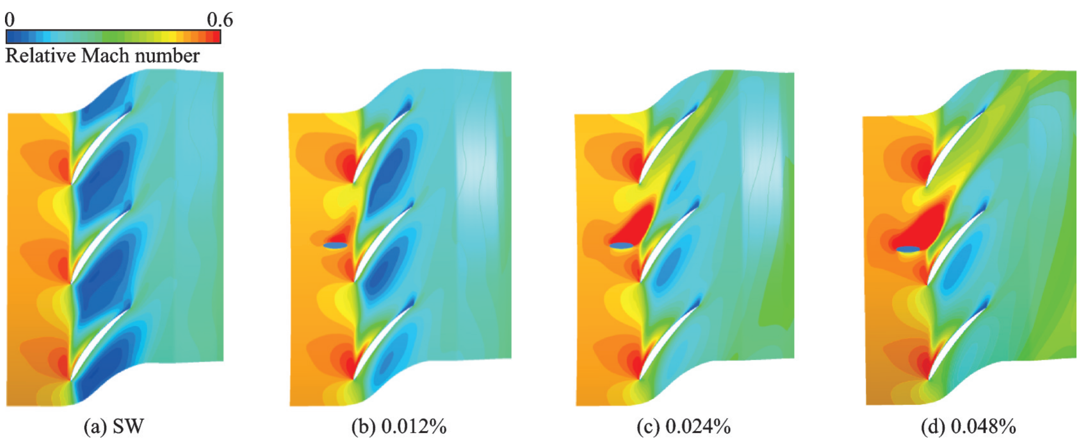


Fig.15 Distribution of the relative Mach number at 99% blade span for different injected mass flow rates under near-stall condition

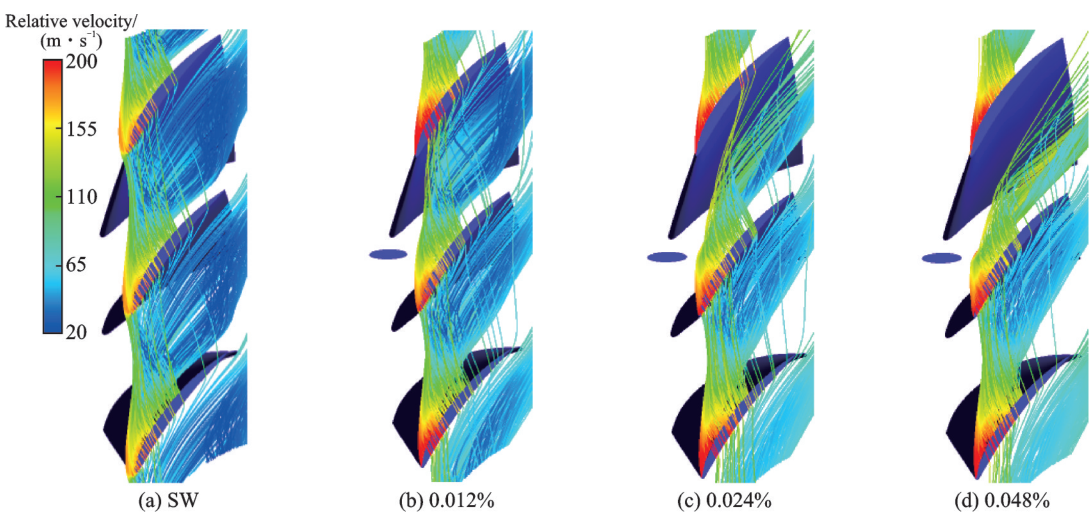


Fig.16 Distribution of tip leakage flow streamline for different injected mass flow rates under near-stall condition

Fig.17 shows the distribution of the rotor diffusion factor along the radial direction under the near-stall condition of SW. As shown in Fig.17, after applying the tip injection, the rotor load in 85% to 100% blade span range is significantly reduced. Moreover, the larger injected mass flow rate, the smaller diffusion factor, which indicates that the tip injection can effectively reduce the rotor load and expand the compressor stability.

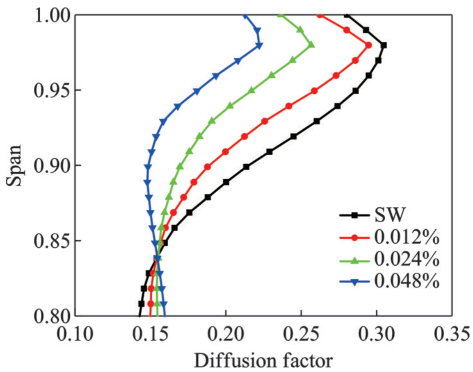


Fig.17 Radial distribution of the diffusion factor for different injected mass flow rates under near-stall condition

3.4 Flow mechanisms of different injector diameters on the compressor stability

Based on an injection axial position of $0\%C_a$ and an injected mass flow rate of 0.024% of the design mass flow rate of the compressor for one injector, three tip injections with injector diameters of 2, 3, and 4 mm are analyzed. Fig.18 shows the compressor performance curves for three tip injections with different injector diameters. The three micro tip injections all improve the stability and efficiency of the compressor. With the reduction of injector diameter, the compressor stability gradually increases, but the improvement in the compressor efficiency gradually decreases. For the tip injection with an injector diameter of 2 mm, the ability to improve the compressor stability is the greatest, and the tip injection with an injector diameter of 4 mm has the weakest ability to improve the compressor stability. When the injected mass flow rate is constant, the change of injector diameter affects the momentum and velocity of the injected flow. The injected flow with higher momentum and velocity can achieve a greater improvement on the compressor stability.

Fig.19 shows the distribution of the relative Mach number at 99% blade span under near-stall

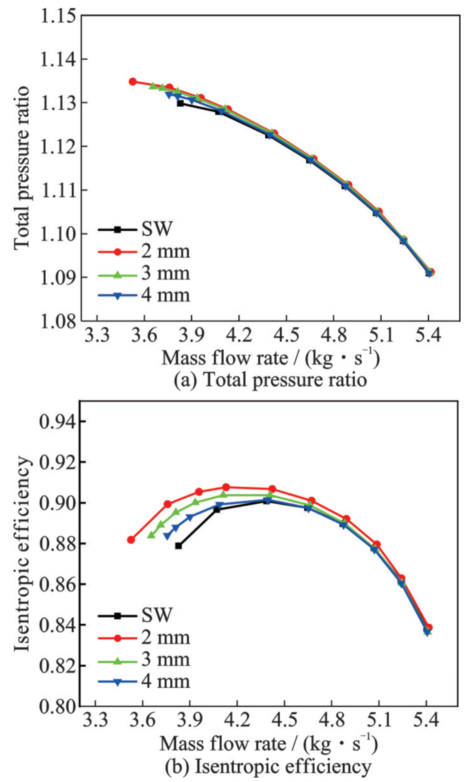


Fig.18 Numerical compressor performance curves for different injector diameters

condition of SW at the same instant. Because the tip injection with an injector diameter of 2 mm has a higher injection momentum and velocity, the injected flow almost eliminates the whole low-speed region in the rotor tip passage. However, for the other two tip injections, the low-speed fluid in the tip passage is larger, and the momentum and velocity of injected flow for the two are lower. So, the low-velocity region in the tip passage is more difficult to be cleared.

Fig.20 shows the distribution of tip leakage flow streamline for different tip injections under the near-stall condition of SW at the same instant. When the injector diameter is 2 mm, the tip injection can effectively suppress the development of tip leakage flow. And there is no secondary leakage phenomenon in the rotor tip passage. However, with the increase of the injector diameter, the secondary leakage cannot be effectively suppressed and the improvement of the compressor stall margin also decreases.

Fig.21 shows the distribution of the rotor diffusion factor along the radial direction under the near-stall condition of SW. After applying the tip injection, the rotor load in 85% to 100% blade span

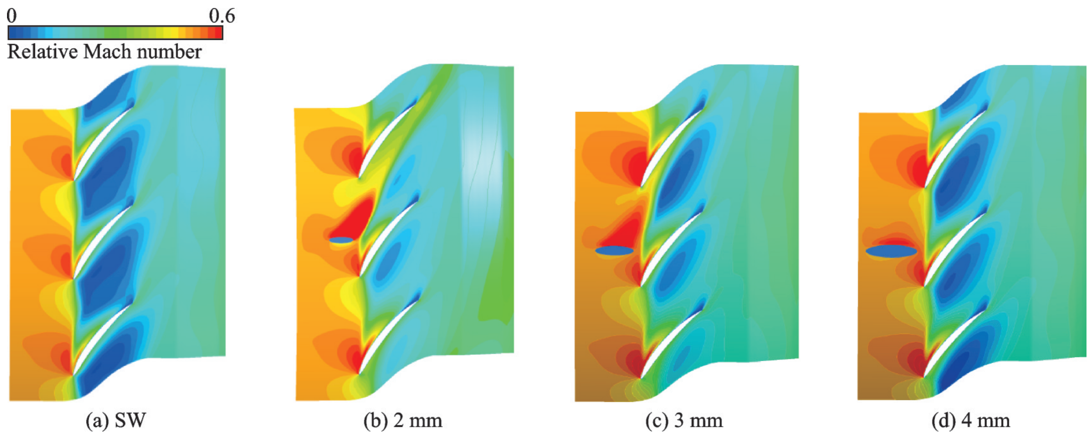


Fig.19 Distribution of the relative Mach number at 99% blade span for different injector diameters under near-stall condition

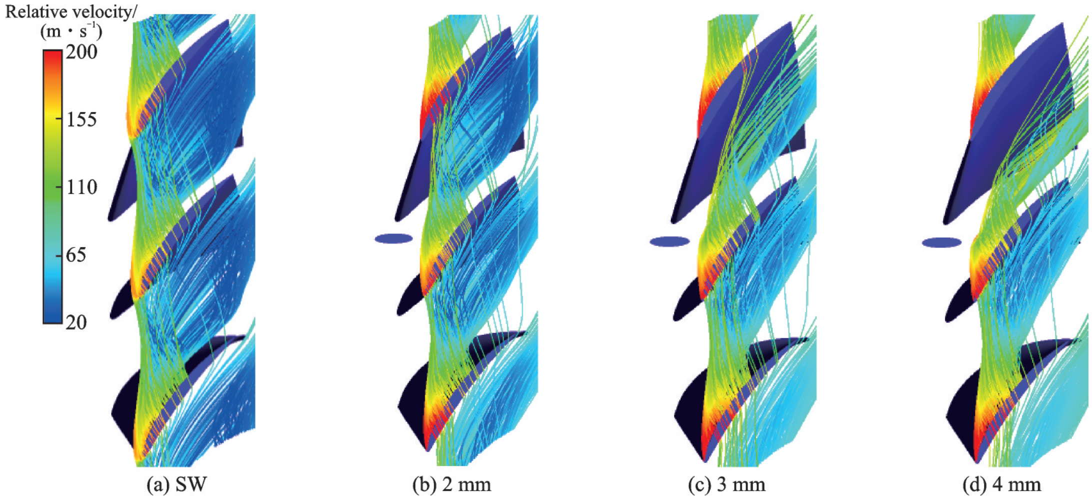


Fig.20 Distribution of tip leakage flow streamline for different injector diameters under near-stall condition

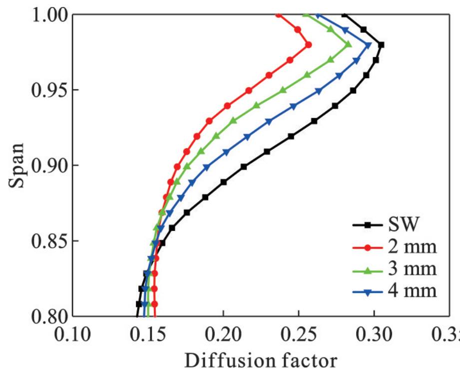


Fig.21 Radial distribution of the diffusion factor for different injector diameters under near-stall condition

range is significantly reduced, and the smaller the injector diameter, the smaller the rotor load, which is more conducive to expanding the compressor stable operating range.

4 Conclusions

This paper investigates the influences of micro tip injection on the compressor stability in a subsonic high-speed axial flow compressor. The tip injection design parameters, including the injection axial position, injected mass flow rate, and injector diameter are studied by an experimental method. And the macroscopic rules of the influences of different tip injection design parameters on the compressor stability are obtained. Subsequently, three-passage unsteady numerical simulations are employed to reveal the underlying flow mechanism of micro tip injection on the compressor stability. The findings of this work provide a comprehensive and valuable reference for the engineering application of micro tip injection. The main conclusions of this paper are as follows:

(1) There is an optimal injection axial position of micro tip injection that maximizes the compressor stability enhancement. This optimal injection position is closely related to the tip blockage location. When the injection axial position is located upstream of the tip blockage region, the tip injection can effec-

tively enhance the compressor stall margin.

(2) A higher injected mass flow rate leads to more effective suppression of tip leakage flow development, resulting in a greater improvement on the compressor stall margin. However, when the injected flow rate reaches a certain value, further increasing the injected mass flow rate result in diminishing returns in terms of the compressor stability improvement.

(3) With a constant injected mass flow rate, variations in injector diameter result in changes in the momentum and velocity of the injected flow. A higher injection momentum and velocity can more effectively clear the low-speed region and suppress tip leakage flow development in the rotor tip passage. So, a smaller injector diameter can lead to a greater improvement on the compressor stability.

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Author contributions Mr. LIU Wenhao designed the study, performed the experiment and numerical simulation, conducted the analysis, interpreted the results and wrote the manuscript. Prof. CHU Wuli contributed to the discussion and background of the study. Prof. ZHANG Haoguang contributed to the discussion and background of the study. Mr. WANG Hao contributed to the data processing and analysis, and refined the manuscript. Dr. WANG Guang revised the article according to the reviewer's comments. All authors commented on the manuscript draft and approved the submission.

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(Production Editors: XU Chengting, WANG Jie)

高速轴流压气机转子叶顶微喷气的试验及数值研究

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摘要：以某高速轴流压气机为研究对象,利用试验和数值模拟方法,系统地研究了叶顶微喷气不同设计参数对压气机稳定性的影响,并揭示了相关的流动机理。研究的叶顶微喷气不同设计参数主要包括喷嘴轴向位置、喷气流量和喷嘴直径。首先,在高速轴流压气机试验台上试验测试了实壁机匣试验件以及7种具有不同设计参数的叶顶微喷气试验件,以阐明叶顶微喷气不同设计参数对压气机稳定性的影响规律。然后,开展了数值模拟研究,通过分析不同设计参数叶顶微喷气对压气机叶顶流场的影响,揭示了叶顶微喷气不同设计参数影响压气机稳定性的流动机理。研究结果表明,当喷嘴位于叶顶低速区上游时,压气机的失速裕度能够得到明显提升。对于喷气流量而言,喷气流量越大,压气机的失速裕度改善越大。对于喷嘴直径而言,更小的喷嘴直径可以形成更高的喷气动量和速度,可以更大程度地提升压气机的失速裕度。

关键词：轴流压气机；高速；稳定性；叶顶微喷气；流动机理；试验；数值模拟