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An Unsteady Flow Control Technology Based on Vortex Injection for Centrifugal Compressor

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1. Introduction

The semi-open centrifugal compressor has the advantages of high efficiency, high pressure ratio and wide working range, making it a widely used device in military and civil fields, such as APU for aircraft, propulsion system for unmanned aerial vehicle or missile, turbocharger for vehicle, fuel cell for automobile, etc [1]. However, the presence of blade tip clearance in a semi-open centrifugal compressor can cause leakage flow, which not only reduces the work capacity at the blade tip, but also induces unstable flows such as tip separation vortex and tip leakage vortex (TLV). Many studies have shown that these unstable flow structures at the blade tip of a centrifugal compressor will seriously affect compressor performance and even induce stall [2-5]. In particular, the TLV, together with its generation, propagation, and breakdown, often acts dominantly on compressor performance [6-9].

In order to solve this problem, people are beginning to pay attention to the use of flow control methods to alter the flow field structures at the blade tip, thereby affecting the overall performance of the compressor. These flow control techniques can be broadly divided into two categories, one being the casing treatment method and the other being the tip air jet/suction method.

There are many ways of casing treatment, including self-recirculation casing treatment (SRCT) [10], circumferential grooves [11], bleed slots [12], axial groove [13], ported shroud [14], etc. Although some success has been achieved in the application of casing treatment technology, the following problems still exist in its further development.

1. The diversity of casing treatment technologies leads to differences in the results and conclusions obtained. How to choose when applying them?

2. How does a casing treatment technology actually changes the flow field structures of a compressor? The understanding of these mechanisms is not clear enough, which can lead to difficulties in determining structural parameters when a certain technology is applied. For example, how to select the number of axial grooves? Because when the axial groove acts on the flow field, there will be relative motion between it and the blade, which is actually an unsteady treatment method. Therefore, the selection of quantity will determine the unsteady control frequency. In some studies, people do not even consider the unsteady effects.

3. Some casing treatment technologies have improved the working margin of the compressor, but have a significant impact on pressure ratio and efficiency. Can other technical indicators be improved without affecting the key performance parameters of the compressor?

Currently the tip air jet/suction technologies are often developed on cascades or don't concern to unsteady factors. Therefore, in practice, it is necessary to further consider the relative motion between the jet/suction location and the blade. In addition, many studies have found that unsteady control can often achieve better results with a small amount of energy injection than steady control. Therefore, these jet/suction technologies are also beginning to adopt unsteady schemes. However, when these jet/suction technologies are inherently unsteady, their actual effects depend on both the inherent frequency and the blade rotation frequency, which may complicate the problem. Moreover, centrifugal compressors often have compact structures, especially micro centrifugal compressors, where complex jet/suction schemes are difficult to apply.

Therefore, based on the above issues, we propose a control method using unsteady vortex injection based on our previous research. Its characteristic is to use the relative motion of the blades and the casing to produce an unsteady control effect, and to use vortex injection to enhance any contact between the jet flow and the blade tip flow. In this paper, we have compared the effects of several control schemes by numerical simulations, and the intrinsic mechanism has also been explained. Finally, a proof experiment on a micro centrifugal compressor is conducted.

2. The unsteady flow control method and numerical simulation methods

This unsteady flow control is achieved by means of a bypass attached to the shroud of the flow passage and equipped with a vortex generator device, which we refer to as a VID (vortex injection device). The number of VIDs depends on the frequency of the TLV, which used in subsequent studies in this paper is half of the blade number. A discussion of the exact structure and number of VIDs can be found in our previous studies [15].

The investigated micro centrifugal compressor is used in a distribution power generation system of 30 kW. Table 1 shows the main geometric and aerodynamic parameters of the compressor. Considering the relative motion between the VID and the impeller, we adopt unsteady numerical simulation to study a double-blade-passage with Numeca/Fine. The full non-matching (FNM) connection and rotor-stator (R-S) interaction are employed to connect the computational domains of the mainstream and bypass streams. Since we use the unsteady control frequency of 50% BPF, the number of VIDs should be half of the number of blades. The total number of grids in the compressor calculation domain without flow control is approximately 1,800,000. The Reynolds number is about 3.1×10^5 and the non-dimensional wall distance y^+ varies from 1−3. For details, please see reference [15].

Table 1

Main parameters of the micro centrifugal compressor

	Parameter	Value
Overall performance	Mass flow rate	0.36, kg/s
	Rotation speed	80 000, RPM
	Total pressure ratio	3.2
Impeller inlet	Inlet tip diameter	59.5, mm
	Outlet tip diameter	19.5, mm
	Relative Mach number of inlet blade tip	0.87
	Relative flow angle of inlet blade tip	30.6°
	Relative flow angle of inlet blade root	59.3°
Impeller outlet	Outlet diameter	98.5, mm
	Outlet blade width	4.08, mm
	Outlet absolute air flow angle	30.0
	Outlet relative air flow angle	71.7

3. Compressor performance with and without VIDs

3.1. Overall performance

In order to have a comparative analysis, we have simulated the cases of the compressor with non-controlled (NC), with hole control (HC, holes are set on the shroud with the same size and number as VIDs), and with vortex injection control (VID, half the width of HC). Fig. 1 shows the sizes of HC and VID. The flow field in one blade passage period after calculation convergence is selected for analysis. The performances of the time-average flow field of the compressor in this period are shown in Fig. 2 and Fig. 3. As can be seen from Fig. 2, both HC and VID control methods can enable the compressor to operate at a smaller flow rate compared to NC. However, with the HC method, the total pressure ratio of the compressor has decreased, with a decrease range of approximately 2.4% to 3.2%. With the VID method, although the pressure ratio decreases at large flow rates (with a maximum decrease of about 2%), it increases when the compressor approaches the design point and stall point (with a maximum increase of about 0.75%). The compressor efficiency comparison shown in Fig. 3 also exhibits similar characteristics to the pressure properties. With the HC method, the efficiency of the compressor has decreased by 0.4% to 1.0%. However, using the VID method, the maximum efficiency reduction of the compressor at high flow rate is about 0.37%, but the efficiency is improved at the design point and near the stall point, with a maximum increase of about 0.56%. A relative stall margin is defined as:

$$SM_{rel} = \frac{\pi_{cs} / \dot{m}_{cs} - \pi_{ocs} / \dot{m}_{ocs}}{\pi_{ocs} / \dot{m}_{ocs}}, \tag{1}$$

where π_{cs} and $\dot{m}_{cs}$ are the total pressure ratio and mass flow rate of the compressor near stall condition with control, π_{ocs} and $\dot{m}_{ocs}$ are the total pressure ratio and mass flow rate of the compressor near stall condition without control. The SM_{rel} is 10.5% and 8.5% corresponding to the compressor with HC and VID respectively.

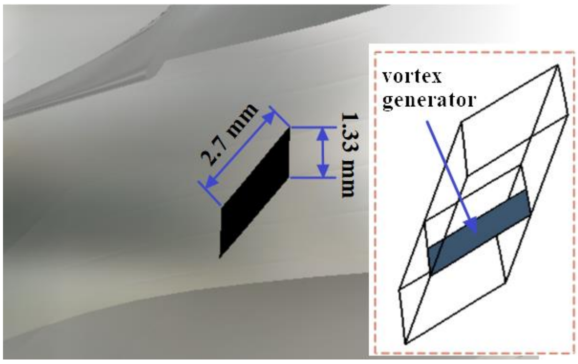

Fig. 1 The sizes of HC and VID (HC – with hole control, VID – vortex injection device)

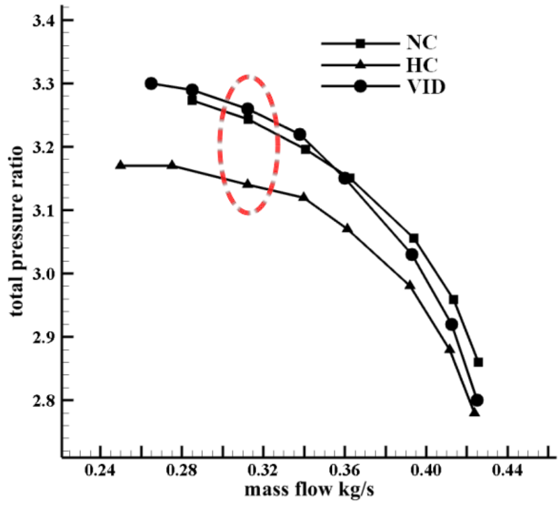

Fig. 2 Compressor total pressure ratio comparison over one blade period (NC – no control, HC – with hole control, VID – vortex injection device)

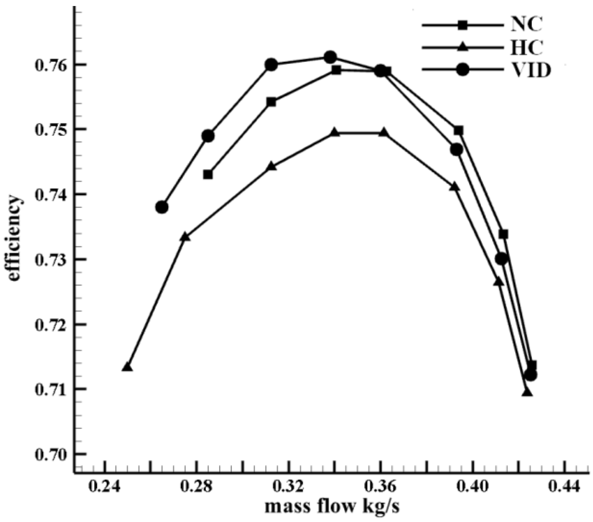

Fig. 3 Compressor efficiency comparison over one blade period (NC – non-control, HC – with hole control, VID – vortex injection device)

By comparing the overall performance of the compressor, we can see both the HC and VID can expand the stable operating range of the compressor. Although the improvement of VID method is slightly lower than that of HC method, it performs better in maintaining the original compressor pressure ratio and efficiency, and even improves slightly at the design point and near the stall point.

3.2. Flow field analysis of the centrifugal compressor

For the analysis we use the same flow field data in one blade pass period as above. Figs. 4 to 6 show the pressure distribution at 95% blade height section of the compressor under no control and two other different controls. These cases are selected from the calculation point before the stall point, as shown in the dotted line in Fig. 2. The mass flow and total pressure ratio of these cases are almost the same, so we consider this comparison to be credible. Furthermore, in order to eliminate the differences of the compressor under different working conditions and to have a better comparability, the dimensionless static pressure coefficient is selected here, which has the following form:

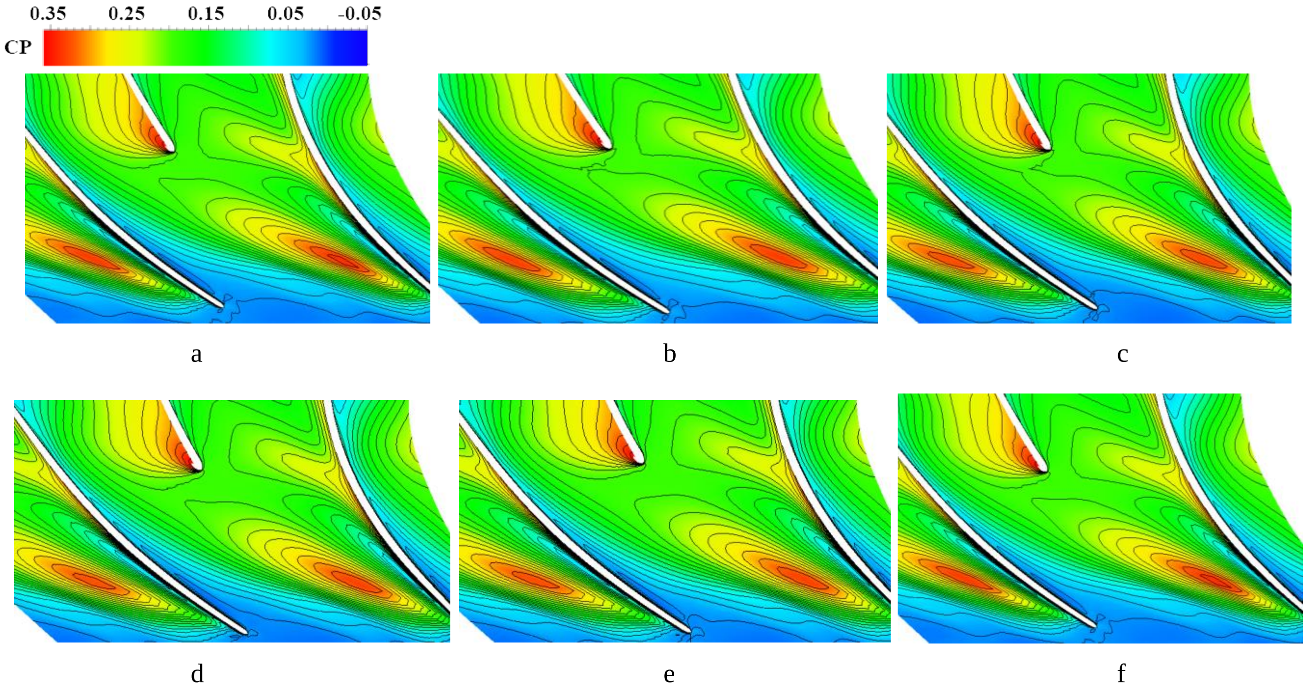

Fig. 4 Transient static pressure distributions at 95% blade height in the case of NC (no control) operation: a – start point of the period, marked as T0; b – 1/5 period, marked as T1; c – 2/5 period, marked as T2; d – 3/5 period, marked as T3; e – 4/5 period, marked as T4; f – the end of the period, marked as T5 (T5=T0)

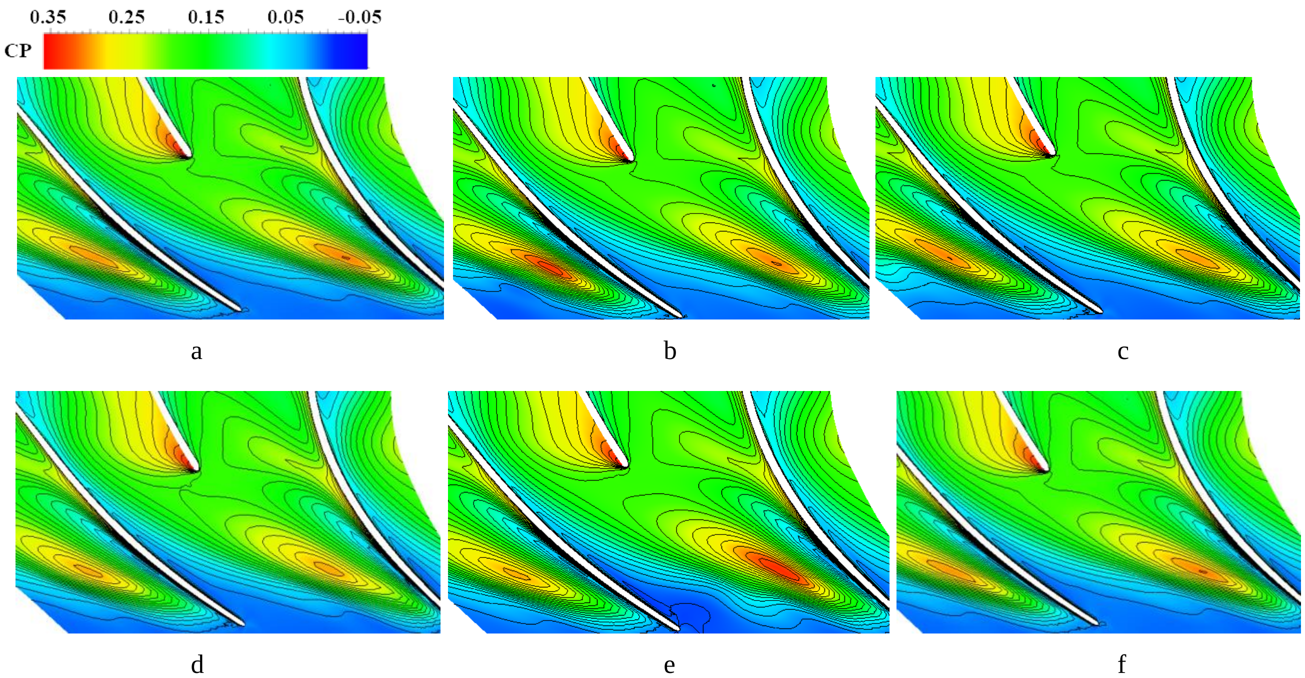

Fig. 5 Transient static pressure distributions at 95% blade height in the case of HC (hole control) operation: a – start point of the period marked as T0; b – 1/5 period, marked as T1; c – 2/5 period, marked as T2; d – 3/5 period, marked as T3; e – 4/5 period, marked as T4; f – the end of the period, marked as T5 (T5=T0)

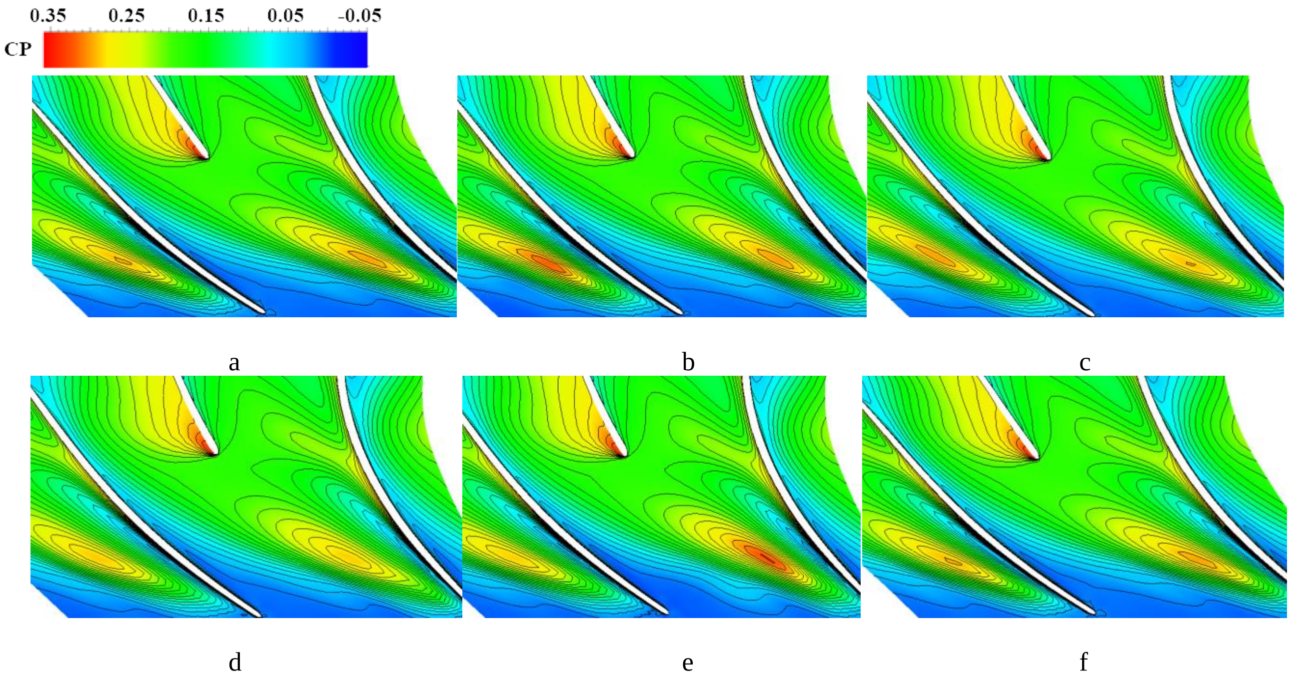

Fig. 6 Transient static pressure distributions at 95% blade height in the case of VID (vortex injection device) operation: a – start point of the period, marked as T0; b – 1/5 period, marked as T1; c – 2/5 period, marked as T2; d – 3/5 period, marked as T3; e – 4/5 period, marked as T4; f – the end of the period, marked as T5 (T5=T0)

$$CP = \frac{p(t) - p_{in}}{p_{in}}, \tag{2}$$

where $p(t)$ is the transient static pressure value and p_{in} is the average static pressure at inlet.

From Fig. 4, we can see that when the compressor does not use flow control, there is a significant high pressure region at the blade tip. From References [16-19], we can know that this high pressure region mainly originates from the TLV, which may cause flow blockage at the compressor blade tip. When using the HC method (Fig. 5), under the influence of periodic excitation, periodic changes occur in the high-pressure region of the compressor blade tip, which are much weaker than in NC operation. High pressure zones alternate in the two adjacent blade passages. The maximum high pressure region in the left passage appears at time T1, and the maximum high pressure zone in the right passage appears at time T4. The range and intensity of this high pressure region will gradually change over time. When the VID method is used (Fig. 6), we can find that the high pressure region at the blade tip is weaker than other operating conditions, indicating that the blockage of the tip flow will be greatly weakened at this time. The law of pressure fluctuations is similar to the HC method, but the overall strength is much weaker.

Based on the above results, we can see that the VID method can effectively weaken the strength of the high pressure region at the blade tip, thus improving the stable working range while maintaining the high efficiency and high total pressure ratio of the compressor. The HC method can also obtain good results, which is very helpful to broaden the stall margin of the compressor with sacrificing the efficiency and pressure ratio of the compressor. The HC method uses the unsteady effect formed by the relative motion of the impeller and the hole to realize periodic excitation. However, it does not take into account the efficiency of each excitation. The VID method not only takes advantage of the same unsteady property as HC, but also considers the efficiency of each excitation. Here we use the concept of negative circulation to force the momentum exchange in the original flow field. Therefore, from the perspective of flow control, we suggest to strengthen the efficiency of each unsteady incentive, which is very beneficial to improve the overall effect of unsteady control.

4. A proof experiment

In order to verify the effectiveness of the VID method, we conducted experiments on a commercial micro centrifugal compressor. The structure of the test piece mainly consists of a centrifugal impeller rotor, a radial diffuser, and an axial guide, which is driven by a motor as shown in Fig. 7. The impeller contains 7 rotor blades with an inlet height of 10.1 mm and tip clearance of 0.3 mm. The design rotational speed is 90,000 RPM, and the design mass flow rate is 0.013 kg/s. The design total pressure ratio is 1.08.

The HC method was also tested for comparison. According to the relationship between the controlled object and the control frequency, the number of the control holes can be 3 or 4. Here, we select 4 holes for testing, and the number of VIDs corresponds to this, as shown in Figs. 8 and 9. For ease of processing, we have used circular holes for testing, and the diameter of the hole is 2.0 mm. In addition, in order to ensure the uniformity of the air intake, a trumpet-shaped transition section is provided at the inlet (Fig. 10), and the entire test apparatus also includes a throttle structure.

A PSI pressure scanning system is used to measure the static pressure at the inlet and the total pressure after the axial guide to obtain the overall performance parameters of the centrifugal compressor. By adjusting the opening of the exhaust throttle and changing the back pressure at the outlet of the test compressor to achieve the purpose of changing the test state. We recorded the parameters of the compressor at different operating points at full rotational speed to obtain the compressor characteristic diagram. The surge is assessed on the basis of the large amplitude pulsation of the compressor inlet and outlet pressures measured dynamically, as well as abnormal compressor noise and pulsation of the data. The results are shown in Figs. 11 and 12.

The results show that both VID and HC can operate the compressor operate at a lower flow rate compared to NC. The SM_{rel} is 13.3% with HC and 6.6% with VID. Although the VID method is slightly inferior to the HC method in expanding stable working capacity, the VID method is far superior to the HC method in maintaining total pressure and efficiency. With the VID method, the pressure characteristics curve and efficiency characteristic curve of the compressor are relatively close to those of the prototype compressor. From the pressure characteristic diagram (Fig. 11),

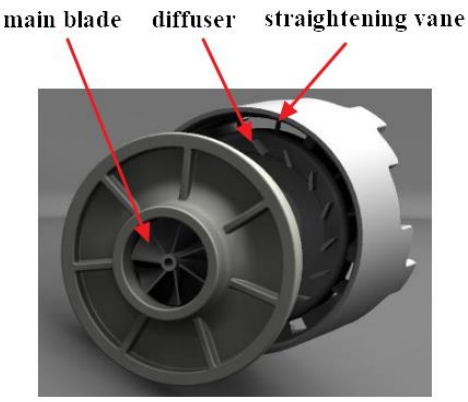

Fig. 7 Three-dimensional model of the tested centrifugal compressor

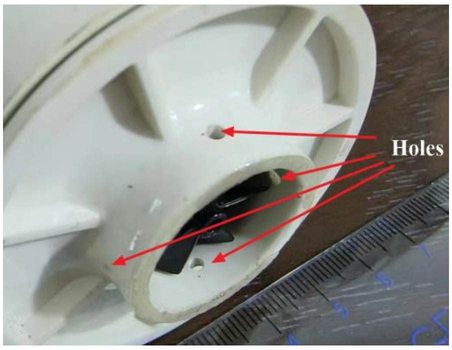

Fig. 8 The tested centrifugal compressor with HC

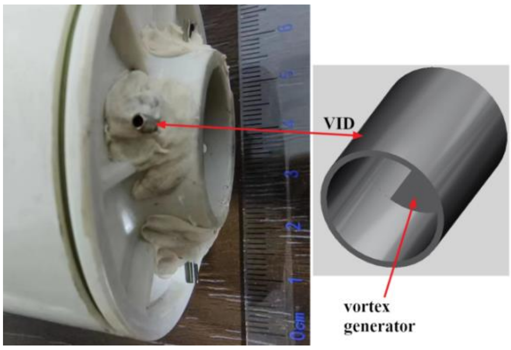

Fig. 9 The tested centrifugal compressor with VID (vortex injection device)

we can see that at low flow rates the pressure can be increased using the VID method, which is very similar to our previous calculation (Fig. 2). The maximum relative pressure increases by 2.6%. The law of the efficiency characteristic diagram is also relatively close to that of the calculation

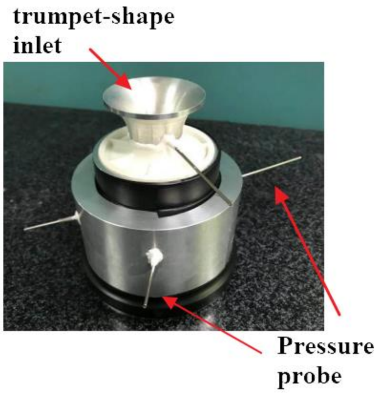

Fig. 10 The assembly of the tested centrifugal compressor with trumpet-shape inlet

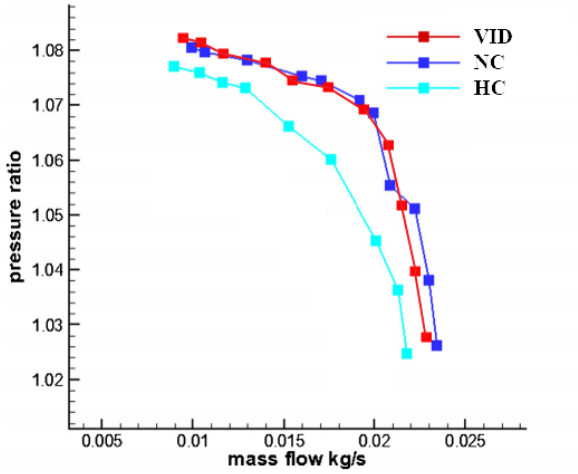

Fig. 11 Total pressure ratio comparison of the micro centrifugal compressor with NC, HC, and VID (NC – no control, HC – with hole control, VID – vortex injection device)

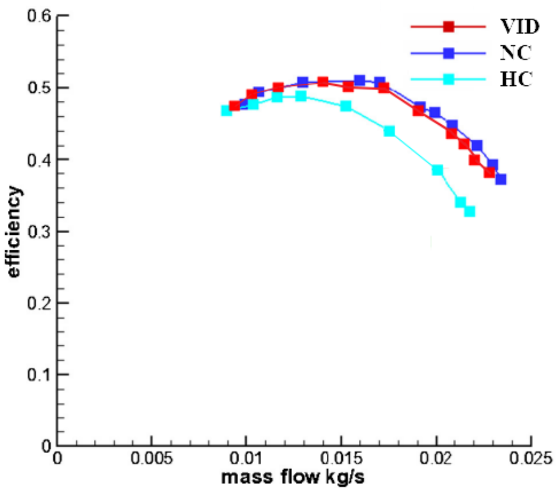

Fig. 12 Efficiency comparison of the micro centrifugal compressor with NC, HC, and VID (NC – no control, HC – with hole control, VID – vortex injection device)

(see Fig. 3 and Fig. 12). Compared to the calculated results, the efficiency improvement of VID at low flow rates is not as good as the calculated results. This may be due to the smaller VID used by the micro centrifugal compressor, which increases the viscosity loss as the air flows through the VID tube, offsetting the effect of VID in improving efficiency. Overall, the test results show that VID can effectively stabilize the operating range, while improving or not reducing the pressure ratio and efficiency of the compressor over a wide flow range.

5. Conclusions

In this study, a passive unsteady flow control method using vortex injection devices (VIDs) is proposed. A VID method realised by equal-circumference tubes with vortex generators connected to the shroud through holes is investigated by numerical studies. A proof experiment is also carried out on a micro centrifugal compressor. The following conclusions are drawn:

1. The numerical calculations show that the VID method can increase the stall margin of the compressor by 8.5%, which is very close to HC (10.5%). However, as the stall point is approached from the design point, VID has significant pressure and efficiency advantages over HC and NC. The maximum efficiency increase and the maximum pressure ratio increase are 0.56% and 0.75% respectively. At high flow rates, the impact on the pressure ratio and efficiency of the compressor is very small. The maximum pressure drop is only 2.0% and the maximum efficiency drop is only 0.37%.

2. Both VID and HC can effectively weaken the strength of the high pressure region at the blade tip and force the blade tip pressure to fluctuate periodically. However, compared to HC, VID weakens the high pressure region more significantly, thus improving the stable working range while maintaining the high efficiency and high total pressure ratio of the compressor.

3. The proof of the VID method on a micro centrifugal compressor shows that the VID can increase the stall margin by 6.6% and the maximum relative pressure by 2.6%. Meanwhile, the efficiency of the compressor is almost unaffected.

4. The overall effect of VID is better than HC. The former uses both the unsteady excitation and the injection of negative circulation to improve the momentum exchange. The latter uses only the unsteady effect. Therefore, improving the efficiency of each unsteady jet/suction and separation flow interaction is highly recommended.

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AN UNSTEADY FLOW CONTROL TECHNOLOGY BASED ON VORTEX INJECTION FOR CENTRIFUGAL COMPRESSOR

S u m m a r y

The flow structures at the blade tip in a centrifugal compressor usually have an important impact on compressor performance, especially the TLV, which often plays a dominant role in its performance. In order to weaken TLV and improve compressor performance, an unsteady flow control method based on the concept of vortex injection is proposed. The numerical simulation shows that VID method can enlarge the stall margin of the compressor by 8.5%. Meanwhile, the total pressure ratio and efficiency of the compressor can be slightly improved from the design point to the stall point. The maximum efficiency increment and the maximum pressure ratio increment are 0.56% and 0.75%, respectively. A proof experiment on a micro centrifugal compressor shows that the VID can enlarge stall margin of 6.6% and the maximum relative pressure can increase by 2.6%. Therefore, both the simulation and experiment prove that the VID method is effective which utilizes both unsteady excitation and injection of negative circulation to weaken TLV. Finally, enhancing the efficiency of each unsteady jet/suction and separation flow interaction is highly recommended.

Keywords: centrifugal compressor, unsteady flow control, tip leakage vortex, compressor stall, vortex injection.

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