Abstract: Active flow control methods are commonly used in expanding the operating range of compressors. Indeed, unsteady active control methods are the main focus of researchers due to their effectiveness. For constructing an unsteady active control system, reliable actuators are significant. To compare with conventional actuators such as synthetic jet actuators and rotating valves, fluidic oscillators have structurally robust characteristics and can generate self-excited and self-sustained oscillating jets, which leads to its higher applicability in compressors under severe working conditions. Thus, to explore the feasibility of unsteady active control systems by the usage of fluidic oscillators, a low-frequency and low-speed oscillator is first designed and experimentally studied for improving the stability of a low-speed axial flow compressor. During the experiments, a special casing is designed to install 15 uniformly distributed oscillators in the tip region of compressor. Based on the unsteady micro injections of the rotor tip with rotor rotation frequency, the results indicate that the frequency/period of oscillators are flexible, in which the values are decoupled with the variation of inlet pressure. When the inlet-to-outlet pressure ratio of the oscillator is in the range of 1.1~2.0, the maximum velocity ranges from 30 m/s to 80 m/s. Moreover, the mass flow rate of the single oscillator only varies from 0.017 ‰ to 0.059‰ from the designed compressor mass flow rate. For the improvement of the compressor stall margin, the value is 3.45% when the total mass flow of oscillators is 0.08% of the designed compressor mass flow.

Keywords: fluidic oscillator; pulsating jets; axial compressor; unsteady stall margin improvement

1. Introduction

The compressor is one of the key components of aero engines; it is expected to achieve not only high efficiency and a high-pressure ratio but also high stability. There are two ways to improve the stable operating margin, which are passive flow control and active flow control methods. [1] Passive flow control methods can remarkably extend the operating range of compressors. By changing the structure of the casing or blades, they reallocate the flow rate and blade loading in the tip region and extend the operating range of compressors. Typical passive flow control methods include casing treatment [2–4], blade tip winglets [5,6], and vortex generators [7,8]. However, passive control methods will inevitably cause a significant penalty in the efficiency and the pressure ratio [9]. Thus, the concept of active flow control was put forward by Epstein et al. [10] and Ffowcs Williams et al. [11]; by monitoring the flow field parameters and using the actuating mechanisms, signals can be added to the flow field as needed to inhibit the development of pre-stall disturbances. In this way, active flow control methods can not only delay the instability phenomenon but also reduce the efficiency loss.

Active flow control methods, such as tip injection [12,13], variable guide vanes [14,15], and synthetic jets [16,17], can be divided into steady and unsteady active control methods according to the temporal characteristics of the perturbation signals. Steady tip injection...
is a widely studied steady active control method. It has many technical advantages. By changing the jet parameters [18] or using the over-tip recirculation system [19], steady tip injection can act on the tip region and expand the operating range. Based on the total inject momentum, Lin et al. [20] and Li et al. [21] categorized two kinds of stall margin improvement mechanisms. According to the range of spanwise blade load changes caused by air injection from the steady aspect, these two kinds of mechanisms are micro and macro tip injection. Unsteady active flow control methods have higher control efficiency than steady control methods because they can obtain control effects that are several orders of magnitude higher than the percentage of excitation energy of the mainstream [22]. Li et al. [23] used acoustic excitation, Kefalakis et al. [24] used unsteady jets, and Saddoughi et al. [25] used plasma excitation to conduct unsteady active control of compressors, all of which achieved certain stall margin expansion with small energy inputs. However, there has not been a common understanding of the unsteady stall margin expansion mechanism caused by the unsteady nature of excitation.

Due to the harsh working conditions inside compressors, the difficulty of unsteady active flow control lies in the lack of reliable actuators. The wall-attachment oscillator [26] can generate self-excited and self-sustained unsteady jets based on the Coanda effect. It has a simple internal structure and contains no moving parts. Compared with traditional unsteady actuators, such as synthetic jet actuators, rotating valves, or plasma actuators, fluidic oscillators are structurally robust and resistant to electromagnetic interference. In addition, the wall-attachment oscillator has a large jet velocity range and less added mass flow [27]. Thus, it is a good choice to promote unsteady excitation engineering applications, especially when high frequencies are required. In recent years, wall-attachment oscillators have been widely studied in airfoil lift augmentation [28], cavity noise suppression [29], and heat conduction enhancement [30,31]. In the field of aero-engines, wall-attachment oscillators have been applied to adjust the thrust vector [32]; control flow separation at the s-shaped inlet [33], corner of compressor stator [34,35], and outlet guide vane of high-load turbine [36]; and reduce loss caused by turbine tip clearance leakage vortex [37].

Through careful design, wall-attachment oscillators can generate unsteady micro jets with very little additional flow rate; thus, they have the potential to realize effective unsteady active control while further reducing the efficiency penalty. Moreover, wall-attachment oscillators are more robust and reliable than traditional unsteady actuators. Therefore, when facing severe working environments inside aeroengines, it is essential to explore the feasibility of unsteady stall margin expansion based on fluidic oscillators. Thus, a low-frequency and low-speed oscillator configuration for a low-speed compressor was redesigned based on the models of Wang et al. [38] and Zhou et al. [39,40]. A series of experiments were conducted to investigate the characteristics of this configuration in detail by using hot-wire anemometer and other measurement methods. A specially designed casing was used to anchor 15 uniformly distributed oscillators in the circumferential direction in the tip region of a low-speed axial compressor rotor. Then, the micro tip injection experiments with the unsteady excitation frequency of rotor rotation were conducted.

This paper is organized as follows. First, a brief introduction to the research subjects and methodology is presented in Section 2. The oscillator characteristics and experimental results in active stall control are presented and analyzed in Section 3, which is followed by a discussion in Section 4. Finally, the main findings are concluded in Section 5.

2. Research Subjects and Methodology
2.1. Research Subjects
2.1.1. Low-Speed Axial Compressor Test Rig TA 36

This paper expects unsteady micro tip injection experiments based on fluidic oscillators on the low-speed compressor TA 36, so the test rig will be introduced first.

The schematic of TA 36 and the measurement sections are shown in Figure 1. At Plane A (compressor inlet) and Plane C (50 mm downstream of stator outlet), 4 pressure sensors are embedded in the casing wall at the position shown in Figure 1 to measure the static
pressure of the inlet and outlet. In the meantime, 2 pressure rakes are respectively placed in the inlet and outlet ducts to measure total pressure. At Plane B (0.5 chord length upstream of the rotor inlet), 8 high-frequency response pressure sensors are used on TA 36 in order to capture the pressure perturbations near the rotor tip. All of the pressure sensors are equally spaced around the circumference.

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To design a fluidic oscillator suitable for TA 36, three compressor parameters are highlighted in the present paper, which are the design flow rate, the tip rotational velocity, and the characteristic frequencies associated with unstable phenomena.

Tables 1 and 2 shows the main design parameters of the TA 36 compressor [41]. The design mass flow is 6.5 kg/s. According to the design rotational speed (2930 rpm) and the tip diameter (600 mm), the tip rotational velocity is approximately 92 m/s. Figure 2 shows the static pressure signals of Plane B at 100% design rotational speed. From the change in the propagation velocity of pressure disturbances, it can be judged that TA36 is a spike-type stall. The calculated stall inception frequency and stall cell propagation frequency are about 38 Hz and 27 Hz, respectively. The rotor rotation frequency is 49 Hz since the rotating speed is 2930 rpm (the design speed).

Table 1. The aerodynamic design parameters of TA36 [41].

| Aerodynamic Parameter     | Value |
|---------------------------|-------|
| Design speed (rpm)        | 2930  |
| Design mass flow (kg/s)   | 6.5   |
| Design efficiency         | 92%   |
| Pressure ratio            | 1.0267|
Table 2. The structural design parameters of TA36 [41].

| Geometrical Parameter                | Rotor | Stator |
|--------------------------------------|-------|--------|
| Number of blades                     | 20    | 27     |
| Aspect ratio                         | 1.18  | 1.40   |
| Tip clearance (mm)                   | 0.6   | 0.5    |
| Tip diameter (mm)                    | 600   | 600    |
| Hub clearance (mm)                   | 346   | 401    |
| Stagger angle at hub (°)             | 45    | 0      |

2.1.2. Oscillator Configuration

Combined with the three key parameters of TA36 described above, a low-speed and low-frequency fluidic oscillator was redesigned based on models of Wang et al. [38] and Zhou et al. [39,40] in order to achieve unsteady micro tip injection on TA36.

This oscillator is a wall-attachment pulsed oscillator, and its interior and exterior structure are shown in Figure 3. The corresponding key parameters are listed in Table 3; all of the parameters are nondimensionalized by the depth of the interior structure, which is 2 mm. In Figure 3a, the blue arrows mark the planar core channel, and the orange tubes in Figure 3c are feedback channels. Together these two parts are named as the feedback loop. To meet the requirements of subsequent experiments, the oscillator’s response frequency needs to be adjustable. Studies have shown that the response frequency of the wall-attachment pulsed oscillator is mainly controlled by the total length of the feedback loop [38,42]. Therefore, this configuration uses plastic tubes as feedback channels, which makes it easy to adjust its length by changing tubes. The interior structures are processed by the high-precision computerized numerical control method in order to ensure the consistency of the oscillators. In addition, feedback channels connect to the oscillator’s main body by threaded pneumatic connectors.

Table 3. The key dimensions of the fluidic oscillator.

| Symbol | Value | Parameter                                      |
|--------|-------|-----------------------------------------------|
| t      | 0.2   | Throat width and depth ratio                  |
| h      | 0.35  | Control port height and depth ratio           |
| w₁     | 1     | Outlet width and depth ratio                  |
| w₂     | 12    | Outlet distance and depth ratio               |
| L₁     | 40    | Oscillator height and depth ratio             |
| L₂     | 20    | Oscillator width and depth ratio              |
Figure 3. The interior and exterior structure of the fluidic oscillator: (a) interior structure; (b) critical region; and (c) exterior structure.

2.2. Experiment Methods

2.2.1. Oscillator Test Bench

As shown in Figure 4, the oscillator test bench includes an air pump (JUBA, TJ1600X6-3201, Wenling, Zhejiang, China) providing high-pressure air, a pressure regulator valve (SMC, AW20-02BG, Tokyo, Japan), a flow meter (Sevenstar, CS100, Tokyo, China), and a pressure gauge (Asmik, MIK-Y290, Hangzhou, Zhejiang, China). The test system is the StreamLine Pro CTA system. The hotwire (Dantec, 55P11, Skovlunde, Danmark) is placed at the center of the left outlet, less than 1 mm away from the outlet section along the depth direction. The hotwire calibration velocity range is 5 m/s to 100 m/s and the sampling rate is 10 KHz.

Figure 4. The oscillator test bench.

The oscillator outlet pressure is equal to the atmospheric pressure $P_{atm}$. The dimensionless parameter inlet-to-outlet pressure ratio is $P_i / P_{atm}$, where $P_i$ is the total inlet pressure of the oscillator. The feedback channel length is $L_f$ and the feedback channel inner diameter is $d_f$. During the test, $d_f$ is 4 mm and $L_f$ ranges from 2.5 m to 7 m. At each length, $P_i / P_{atm}$ gradually increases from 1.1 to 2.0 in order to obtain thorough frequency and velocity characteristics of this oscillator.
2.2.2. Test Casing

Restricted by the outlet size, pulsating jets from a single oscillator can be quite limited. It is essential to arrange a certain number of oscillators on the casing to achieve the desired control effect. Figure 5 presents the test casing specially designed to fix oscillators. Thirty oscillators are anchored on the test casing, and the number of oscillators can be varied by disconnecting the air supply tubes of some of them. The axial position of oscillators is adjusted by switching the relative position between the inner ring and axial positioning rings. As shown in Figure 5, unsteady jets are 25° away from the mainstream and the axial position of actuation is 0.2 cord length downstream of the rotor tip leading edge.

![Figure 5. The test casing.](image)

Meanwhile, the height of the oscillator is relatively short, so after the oscillator array is installed on the test casing, slot-like chambers still remain in the casing, as marked by the red boxes in Figure 5, in order to protect the rotor.

2.2.3. Oscillator–Compressor Experimental System

The composition of the oscillator–compressor experimental system is shown in Figure 6. It includes an air pump (JUBA, TJ1600X6-3201, China) providing high-pressure air, an air tank (JUBA, JB1.0-8, Wenling, Zhejiang, China) storing the compressed air, a strainer (JUBA, JB-30, Wenling, Zhejiang, China) filtering imported air, a manifold dividing the air 30 ways, a pressure regulator valve (SMC, AW20-02BG, Tokyo, Japan), a flow meter (SMC, PFMB7501, Tokyo, Japan), and a pressure gauge (Asmik, MIK-Y290, Hangzhou, Zhejiang, China). The data acquisition system is a NI-PXI system, which contains 48 channels, and its maximum sampling rate is 100 KHz. TA 36 has a controllable bleed valve to reach the pre-stall region as slowly as possible while recovering from the stall as quickly as possible. This setup ensures that all the steady operating characteristics of TA 36 can be recorded by the standard time–mean instruments.

During the experiment, the throttling process was stopped when the working condition of TA 36 reached the near-stall condition of baseline, the pressure regular valve was opened, and after the oscillators worked stably, we continued the throttling process. Pressure data was continuously collected during the whole process.
3. Results

3.1. Oscillator Characteristics

3.1.1. Frequency Response

Hotwire tests were conducted under the condition that $P_i/P_{atm}$ ranged from 1.1 to 2.0. The feedback channel’s inner diameter $d_f$ was 4 mm, and the length $L_f$ range was 2.5–7 m.

The oscillator characteristic test results of some feedback channel lengths had very little difference, which is difficult to present. Therefore, in order to show the frequency characteristics that changed with $L_f$ as clearly as possible, Figure 7 shows the experimental results with the feedback channel lengths $L_f$ of 2.8 m, 3.0 m, 3.2 m, 3.8 m, 4.6 m, 5.2 m, and 6.2 m. With the decrease of $L_f$, the response frequency of the oscillator increased. All frequency–pressure response curves of different $L_f$ show similar trends with the increase of $P_i/P_{atm}$. When $P_i/P_{atm}$ increased from 1.1 to 1.25, the response frequency increased quickly, and after $P_i/P_{atm}$ reached 1.25, the frequency basically did not change but floated around a certain value. According to the research of Cerretelli et al. [43], wall-attachment pulsed fluidic oscillators have the potential to realize frequency–pressure decoupling by designing interior configurations rationally. When $P_i/P_{atm}$ was 1.25 to 2.0, the oscillator in this paper had response frequencies insensitive to the pressure change, that is, it had the frequency–pressure decoupling characteristics, which can effectively reduce the difficulty of frequency regulation in future experimental work.

![Figure 6. The oscillator–compressor experimental system.](image)

![Figure 7. The oscillator frequency response spectrum.](image)
In addition, three characteristic frequencies related to unstable phenomena are also marked in Figure 7. This configuration can achieve these frequencies by selecting the feedback channel length $L_f$. $L_f$ corresponding to the rotating stall frequency (RSF, 27 Hz), the stall inception frequency (SIF, 38 Hz), and the rotor rotating frequency (RRF, 49 Hz) were respectively 5.2 m, 3.8 m, and 3.0 m. For each $L_f$, frequency deviations to the target frequency were less than 1 Hz, which means the error was no more than 3.70% and is in line with the experimental expectations. When $P_i/P_{atm}$ is from 1.25 to 2.0, the average frequency on each curve is defined as the average decoupled frequency. Further revealing the frequency characteristics of this oscillator, the average decoupled frequency and the average decoupled period $T$ related to $L_f$ of all test results are plotted in Figure 8. An empirical formula was obtained by linear fitting, where $c$ represents the local speed of sound. Compared with the formulas proposed by Wang et al. [38] and Zhou et al. [39], the $T$ of three curves are linear with $L_f$, but the slopes and the intercepts are different. The slope of the formula of this configuration is between Wang’s and Zhou’s formulas, and the intercept is negative while in the other two formulas, it is zero. These differences indicate that the slope and intercept of an oscillator are sensitive to its internal structure. In future work, if target frequencies are known, the feedback channel length $L_f$ can be approximately predicted by this empirical formula.

$$T = 1/f \approx \frac{2.6L_f}{c} - \left(3.14 \times 10^{-3}\right)$$ (1)

Figure 8. Changes of decoupled frequency and period $T$ with $L_f$ [38,39].

3.1.2. Velocity Response

Figure 9 shows the typical velocity response when $L_f$ is 3 m and $P_i/P_{atm}$ is 1.3. In each period $T$, the time of high pulsed speed is $T_{on}$ and the time of low pulsed speed is $T_{off}$. The duty cycle, which is the ratio of $T_{on}$ and $T$, was about 56%. In addition, the low pulsed speed was close to 0 and barely fluctuated, while the high pulsed frequency had obvious fluctuation. For the convenience of the following description, the average value of the high pulse velocity of each waveform is defined as $V_{max}$. 

![Figure 8. Changes of decoupled frequency and period T with L_f [38,39].](image-url)
3.1.2. Velocity Response

Figure 9 shows the typical velocity response when \( P_i / P_{atm} = 1.3 \), the typical velocity response of the oscillator.

Figure 10 shows the comparison of velocity responses with \( P_i / P_{atm} \) of 1.3, 1.5, 1.7, and 1.9, respectively. As the inlet pressure rose, the waveform of the velocity response was almost unchanged, and the \( V_{\text{max}} \) achieved increased as well as the fluctuating amplitude of the high pulsed velocity. Even if \( P_i / P_{atm} \) was the same, there were still some differences in neighboring waveforms due to velocity fluctuations. Therefore, the maximum velocity response of the oscillator was estimated by the arithmetic average \( V_{\text{max}} \) of 10 continuous waveforms, which was 30 m/s to 80 m/s, as shown by the blue curve in Figure 11. The unsteady jet speed generated by this oscillator configuration was lower than the tangential velocity of the rotor tip of TA36, which was larger than 92 m/s, so this configuration meets the requirements of the unsteady micro tip injection used in this paper.

Figure 10. The oscillator velocity response at: (a) \( P_i / P_{atm} = 1.3 \); (b) \( P_i / P_{atm} = 1.5 \); (c) \( P_i / P_{atm} = 1.7 \); and (d) \( P_i / P_{atm} = 1.9 \).
3.1.3. Mass Flow Response

The mass flow response of this oscillator is shown by the red curve in Figure 11. Restricted by the throat area, the flow rate increases monotonically with the increase of the inlet pressure. When the range of $P_i/P_{atm}$ was 1.1 to 2.0, a single oscillator mass flow rate was from 0.115 g/s to 0.388 g/s. The dimensionless parameter $M_n$ is the per millage of the oscillator flow rate to the design flow rate of the compressor TA36. In Figure 11, the $M_n$ of a single oscillator ranges from 0.017 to 0.059, which is far less than the design flow rate of the compressor and meets the micro-injection requirement of the experiments.

In summary, this new oscillator configuration can realize the three frequencies related to the unstable phenomena of TA 36 by adjusting the feedback channel length. It can also achieve response frequency/period-pressure decoupling as an empirical formula between frequency/period and the feedback channel length was established, which is helpful to predict and select the corresponding target frequency. The $P_i/P_{atm}$ range was between 1.1 and 2.0, and the maximum jet velocity range was 30 m/s to 80 m/s, which was smaller than the flow velocity at the rotor tip region. The mass flow range of the single oscillator was only 0.017~0.059‰ of the designed compressor flow rate, which ensures that the excitations were micro-injections, and the jet energy basically does not cause efficiency penalty.

3.2. Stall Control Application

In this study, 15 oscillators distributed evenly above the compressor rotor tip along the circumference were used, with an axial position of 0.2 times the tip cord length downstream of the leading edge. The total inlet pressure of all oscillators was 1.7 times atmospheric pressure, the jet frequency is rotor rotating frequency (RRF), and the maximum actuate velocity is 70 m/s. With a jet flow of only 0.08% of the compressor design mass flow and the maximum jet momentum of 0.27% of the compressor momentum corresponding to design mass flow, the active stall control experiments were conducted on TA 36 to explore the influence on the stability of the low-speed compressor.

3.2.1. Parameter Definition and Experimental Repeatability Verification

In order to quantitatively describe the effect of the unsteady excitation generated by oscillators on compressor stability, the following several parameters are defined.
The total-to-static pressure rise coefficient $\psi$ is usually used to describe the low-speed compressor’s pressure rise characteristics, which is defined as Formula (2). The flow coefficient $\phi$ is defined as Formula (3):

$$\psi = \frac{P_2 - P_{t1}}{0.5\rho U_m^2}$$  \hspace{1cm} (2)

$$\phi = \frac{V_s}{U_m} = \frac{q_v}{U_m A_0}$$  \hspace{1cm} (3)

where $P_2$ is the static pressure of the stator exit, $P_{t1}$ is the total inlet pressure, $U_m$ is the tangential velocity of the mid-span, $q_v$ is volume flow, and $A_0$ is the inlet area.

Efficiency is expressed as Formula (4), $P_{t2}$ is the total pressure at stator exit, and $\tau$ is the torque.

$$\eta = \frac{q_v(P_{t2} - P_{t1})}{\tau}$$  \hspace{1cm} (4)

The comprehensive stall margin is adopted in this study to evaluate the compressor stability. The stall margin (SM) and stall margin improvement (SMI) are described as follows:

$$SM = \left(\frac{\psi_s}{\psi_d}/\frac{\phi_s}{\phi_d} - 1\right) \times 100\%$$  \hspace{1cm} (5)

$$SMI = SM_{exp} - SM_{ref}$$  \hspace{1cm} (6)

where subscripts $s$ and $d$ refer to stall and design operation points, respectively, and the working conditions corresponding to the subscripts $exp$ and $ref$ are selected as needed.

In Figure 12, a repetitive result of the baseline characteristics tested at the design speed is shown. The baseline cases are tested with the unmodified TA 36 casing. The results show good consistency, and the subsequent experimental results can be considered to be true and credible.

![Figure 12](image-url)

(a) (b)

**Figure 12.** The repeated experiment results of TA36 under baseline condition: (a) pressure rise characteristics; and (b) efficiency characteristics.

### 3.2.2. Effect of Oscillators on Compressor Stability

Results of the unsteady actuation using oscillators are shown in Figure 13. In legends, EXP refers to the usage of test casing only, due to the assembly with oscillators; it has 30 slot-like chambers over the rotor tip region compared with the baseline case. EXP_RRF refers to the usage of test casing and RRF unsteady excitation, and SP refers to stall point.
When only using the test casing, the stall margin expansion (SP 1 to SP 2) completely weaker than the test casing itself. Although the efficiency of case EXP_RRF is obviously selected after considering both safety and its possible stall margin improvement ability.

As mentioned above, 30 oscillators are always installed on the test casing, and the number increases, and the trend is basically first fast and then slow. The oscillators used in this experiment have two outlets, and the width of each is only 2 mm. When 15 oscillators are applied, the circumferential coverage of air injection is only 1.592%, and considering that the flow rate of air injection is only 0.08% of the compressor’s designed flow rate, chambers are left on the casing. In these chambers, the momentum of the injected air is averaged, thereby expanding the circumferential coverage of the excitation to 20 times the original. At the same time, chambers on the casing are similar to the axially slotted casing treatment. According to previous research experience [45], with the increase of the porosity, the SMI and efficiency loss of the slotted casing treatment increase significantly. As mentioned above, 30 oscillators are always installed on the test casing, and the number of oscillators is adjusted by adjusting the air supply. Therefore, from the perspective of casing treatment, the porosity of the casing is independent of the number of oscillators used, which is always 63.67%, and the efficiency loss may be large. Therefore, it is necessary to reasonably design the depth of the oscillator insertion into the casing to reduce the efficiency loss as much as possible. At present, this paper only explores the feasibility of the application of fluidic oscillators on the stall margin control. This form of test casing was selected after considering both safety and its possible stall margin improvement ability. In future work, the influence of unsteady excitation of the oscillator directly acting on the rotor tip will be explored.

Moreover, in Figure 13a, after performing an unsteady excitation with the frequency of RRF, the flow coefficient of SP 3 was slightly smaller than that of SP 2, which was significantly less than the flow coefficient difference between SP 1 and SP 2. This illustrates that unsteady excitation generated by oscillators can have a certain effect. However, it is weaker than the test casing itself. Although the efficiency of case EXP_RRF is obviously lower than the baseline case, there is barely any efficiency loss compared with the case of EXP, and even some efficiency gain in near-choke and near-stall areas.

In this study, the combination of casing treatment and unsteady tip injection is technically realizing the decoupling of steady and unsteady stall margin improvement effects. When only using the test casing, the stall margin expansion (SP 1 to SP 2) completely

![Figure 13. The results of the oscillator–compressor experiments: (a) pressure rise characteristics; and (b) efficiency characteristics.](image)
reflected the steady mechanism, and adding the unsteady excitation on this basis, the extra stall margin expansion gain (SP 2 to SP 3) fully reflected the unsteady mechanism. To quantify the steady and unsteady effects in this work, the three stall points in Figure 13a are individually plotted in Figure 14. According to Formula (5), the stall margin of case Baseline was 32.68%, the stall margin of case EXP was 51.65%, and the stall margin of case EXP_RRF was 55.10%. According to Formula (6), if you select $SM_{ref}$ as the stall margin of case Baseline and $SM_{exp}$ as the stall margin of case EXP, the steady stall margin improvement of the case EXP is 18.97%. Select $SM_{ref}$ as the stall margin of case EXP and $SM_{exp}$ as the stall margin of case EXP_RRF, and the unsteady stall margin improvement of case EXP_RRF is 3.45%. Select $SM_{ref}$ as the stall margin of case Baseline and $SM_{exp}$ as the stall margin of case EXP_RRF, the steady and unsteady coupling improvement is 22.42%.

![Figure 14. The summary of stall points of TA36.](image)

4. Discussion

This oscillator configuration has the following advantages when applied to the low-speed compressor TA 36:

- The response frequency is adjustable and decoupled with pressure. An empirical formula between frequency/period and the feedback channel length $L_f$ is established, which is helpful to predict and select the $L_f$ of corresponding target frequency.

- The oscillators generate micro tip injections, which have almost no impact on the compressor characteristics compared with cases only using test casing, while it can obtain an unsteady stall margin expansion effect of two magnitudes greater than the excitation itself. In this study, the injected mass flow is only 0.08% of the compressor’s design mass flow, and the unsteady stall margin expansion is 3.45%.

At the same time, because of the special structure of the test casing, the unsteady stall margin expanded by oscillators can be compared with that of the chambers. In Figure 13, the chambers are similar to the conventional passive flow control method and can produce up to 18.97% stall margin expansion, but at a cost of 2.77% efficiency loss. Unsteady micro jets have a weaker effect on stall margin than that of steady methods, but they produce basically no efficiency loss, perhaps because the influence mechanism is changed. Based on previous research experience, [20,21,23], the unsteady micro jets in this study input very small total energy, which can only act on the tip leakage vortex. Therefore, on the one hand, the jets themself suppress the advance and development of the leakage vortex. On the other hand, the unsteady frequency in the jet may interact with some frequencies in the
leakage flow, stimulating the low-energy fluid in the tip region and delaying the leakage vortex generation, thus improving the stable working margin of the compressor.

This paper only verifies the feasibility of the innovative application of the oscillator in the compressor’s active stall control field, and there are still many shortcomings. First, the unsteady excitation was not directly acting on the rotor tip flow field, and the stall margin expansion limit was not reached. Second, only the RRF unsteady excitation experiment was carried out, and the influence of other frequencies has not been clarified. Third, the data tested in the flow field were very limited, so it is difficult to fully explore and reveal the unsteady stall margin expansion mechanism. Lastly, compared with synthetic jet actuators or rotating valve actuators, oscillators have more application advantages in generating high excitation frequencies in the harsh operating environment of aeroengines, and their effects on the stability of high-speed compressors have not been explored. On the one hand, future work will optimize the appearance design of the oscillator and explore the influence of other excitation parameters on stability. Numerical simulation will be used to explore the unsteady stall margin mechanism combined with experimental results. On the other hand, oscillators will be applied on a high-speed compressor rig to explore the application value.

5. Conclusions

In this paper, the characteristics of a new pulsed fluidic oscillator with low frequency and low speed are experimentally studied. Fifteen unsteady oscillators are applied at the rotor tip of a low-speed axial flow compressor (TA36). Based on the usage of low-response steady transducers and high-response dynamic transducers, the influence of the unsteady excitation of oscillators on compressor stability is explored. The concluding remarks can be summarized as follows:

1. The oscillator configuration meets the requirements of unsteady micro tip injection experiments, which can cover the instability characteristic frequencies of TA36 by changing the feedback channel length. When \( P_i/P_{atm} \) ranges from 1.1 to 2.0, the maximum velocity ranges from 30 m/s to 80 m/s, which is smaller than the flow velocity in the rotor tip region. The flow rate of a single oscillator only varies from 0.017‰ to 0.059‰ of the compressor design flow rate, which causes low flow loss in the compressor.

2. The frequency/period of oscillator configuration is decoupled with the variation of inlet pressure. Furthermore, an empirical formula between frequency/period and the feedback channel length \( L_f \) is summarized, which is helpful to select the \( L_f \) corresponding target frequency in actual application.

3. The unsteady active stall margin control with the usage of fluid oscillators is feasible. When the excitation frequency of oscillators is RRF, the improvement of compressor stall margin is 3.45%, in which the excitation flow and excitation momentum of oscillators are 0.08% and 0.27% of the compressor designed flow rate, respectively.

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