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EXPERIMENT AND THEORETICAL ANALYSIS OF A TYPE OF NON-SYNCHRONOUS VIBRATION IN A HIGH PRESSURE COMPRESSOR

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Abstract

Non-synchronous vibration (NSV) is a unique blade vibration phenomenon in the fan/compressor. Its main characteristic is that the vibration frequency of the blade and the aerodynamic frequency are not in an integer multiple order relationship with the rotating frequency. Although the researchers have carried out a large number of experiments and theoretical analysis, the causes and mechanism of the formation of non-synchronous vibration are still not fully understood. At present, the internationally accepted theories include the theory of rotating instability and the theory of acoustic feedback. Based on a nine-stage high-pressure compressor with non-synchronous vibration and unique vibration characteristics, a series of targeted experiments were carried out. Through the theoretical analysis of the test results, the unique vibration characteristics of the blade are described from frequency domain and time domain. Combined with similar experiments and theoretical analysis, the intrinsic excitation process of non-synchronous vibration is analyzed.

Keywords: non-synchronous vibration; rotating instability mechanism; characteristic frequency; acoustic resonance

1. General Introduction

With the development of modern aero-engine fan and compressor research technology, the efficiency of compressor/fan is continuously improved, the flow loss is continuously reduced, and the performance parameters of the whole engine are significantly improved.
However, advanced aerodynamic and structural design has significantly increased the aerodynamic load borne by the fan and compressor blades, while the structure itself has become thinner and lighter, and its vibration problems have become more prominent, which seriously threatens the structural strength and lifespan of aero engines. Generally speaking, in the field of fans and compressors, there are two main types of blade vibration induced by fluid: forced vibration and flutter. The forced vibration is generally caused by the unsteady aerodynamic excitation source caused by the circumferential inhomogeneity in the flow field, such as the upstream wake and other factors. In the static coordinate system, this kind of excitation is stable, and the vibration frequency it causes is generally an integer multiple of the rotation frequency of the rotor. This type of vibration is called synchronous vibration. Flutter is generally considered to be a self-excited vibration, caused by the interaction of blades and flow field, and its frequency is not directly related to the rotational frequency, and belongs to a type of non-synchronous vibration.
The non-synchronous vibration described in this article belongs to another type of vibration. This vibration phenomenon has been reported in high-pressure compressors and fans. When this type of vibration occurs, the blade vibration and the pressure pulsation at the blade have obvious characteristic frequencies. This article will refer to this type of vibration as non-synchronous vibration, or NSV if it is not explained later in this article. In some cases, this vibration may be confused with flutter. Figure 1 shows the approximate positions of NSV vibration, flutter and forced vibration on the Campbell diagram. Generally speaking, compared with flutter, this type of asynchronous vibration has the following characteristics:

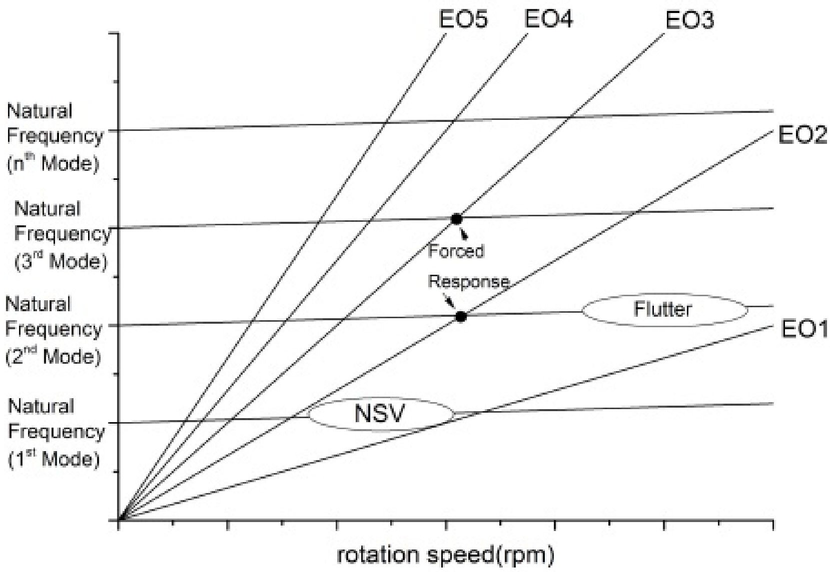

Figure 1 – NSV on Campbell diagram

- NSV vibration occurs in a wide range of stable working state. The sub/transonic stall flutter, supersonic stall flutter and blockage flutter found at present generally occur in the fan/compressor near stall or plugging point and other unstable flow areas. Supersonic non-stall flutter and A100 flutter may also occur in stable operation.
- When NSV vibration occurs, the blade vibration amplitude will remain stable if the working state remains unchanged. However, the most direct characteristic of blade flutter is that the amplitude of the blade increases with the development of time. If it is not restrained, blade fracture will soon occur.
- The damage mechanism of NSV non-synchronous vibration to the engine is long-term fatigue, which eventually leads to blade fracture, Flutter causes the blade to burst suddenly for a short time.

In summary, this type of NSV vibration can more conceal the damage to the engine. After the engine is tested or put into production, the consequences of blade fracture failure due to high-cycle fatigue in the high altitude will be more serious. It may even cause a large-scale grounding problem.
Due to the potential hazards of NSV, the generation, existence mechanism, and elimination methods of this problem have always been the most concerned content of relevant researchers. Baumgartner from Rolls-Royce Company and Neise et al. from DLR explained the phenomenon from the perspective of rotational instability. Baumgartner conducted a test on a ten-stage compressor and believed that the problem was caused by rotational instability. Different from rotational instability, Thomssin from Pratt Whitney, Canada, et al. proposed related theories based on acoustic induced vibration, and carried out mechanism test verification and numerical simulation, which is currently a more comprehensive mechanism. theory. However, due to the complexity of this issue, various interpretation theories have not yet formed a unified and widely accepted theory.
Aiming at the characteristics of non-synchronous vibration, this paper designs a series of related tests for a high-pressure compressor with non-synchronous vibration of the first-stage rotor blades, and records the corresponding blade vibration characteristics and aerodynamic characteristics. Comparing the relevant research results at home and abroad, combined with the test results, the performance of pressure pulsation, noise, and blade vibration in the frequency domain and time domain during non-synchronous vibration are analyzed and studied.

2. Experiment Setup

The test platform for the series of experiments in this article is a nine-stage high-pressure compressor. The first-stage rotor blades of the high-pressure compressor will occurs non-synchronous vibration problems under certain conditions. The test facility of the compressor is a single-shaft bypass compressor test rig of Shenyang Engine Research Institute.

As mentioned earlier, when non-synchronous vibration occurs, the most attention is paid to the performance of dynamic pressure and blade vibration characteristics in the frequency and time domains. Therefore, special attention is paid to time synchronization and phase correction. This experiment is based on the dynamic measurement system of HGL Company, and constructs a dynamic unified synchronous test analysis platform for dynamic pressure, blade dynamic stress, and inter-stage flow field measurement. The platform has a complete dynamic measurement solution, which can complete the acquisition, storage, monitoring and analysis of almost all dynamic physical signals in the compressor experiment.

In the process of dynamic signal testing, the test platform involves many dynamic test sensors, amplifying and conditioning systems, data acquisition modules, etc., such as dynamic pressure and dynamic stress. In order to solve the time synchronization problem, the following two parts are mainly completed:

- Obtain the time delay of each sensor and signal amplifier to complete the time delay correction.
- Establish the time synchronization communication mechanism of each acquisition module to ensure the consistency of the time of each module and the acquisition and storage system.

Phase analysis is particularly important in analyzing the propagation characteristics of asynchronous vibration and its internal mechanism, so ensuring the accuracy of phase measurement is one of the primary considerations of the testing system. In order to ensure the reliability of the phase test, the dynamic stress sensor and the dynamic pressure sensor are connected to the signal transmission line to carry out unified phase calibration. In the data analysis and post-processing, phase correction was carried out according to the calibration results of each test channel to meet the needs of phase analysis.

As shown in Fig. 2-3, the distribution diagram of the main measuring points in the test is simplified. Distribution diagram in figure 2 for measuring axial position, in front of the level of the rotor section 1-1 and level 2-2 after the static son cross section of each arrangement has 10 wall fluctuating pressure sensor, arrangement can see in figure 3, this arrangement can be measured through synchronous vibration frequency of circumferential propagation characteristics, at the same time for the acoustic modal analysis provides the convenience.

Three dynamic pressure sensors are arranged along the radial direction of the first stage moving blade to measure the characteristics of pulsating pressure on the blade surface and its radial propagation characteristics. At the same time, twelve blade dynamic stress sensors are continuously installed on the surface of the primary rotor blades, and the dynamic stress sensors are located at the root of the blades. The main purpose of this arrangement is to measure the circumferential propagation characteristics of blade vibration. The pulsating pressure and dynamic stress signals in the rotating coordinate system are drawn to the HGL synchronous dynamic test system through the slip ring primer installed on the shafting system.

After the first-stage rotor, a two-hole dynamic wedge probe was used to measure the flow field to analyze the corresponding situation of the flow field at the rotor blade exit when the non-synchronous vibration occurs, and to provide support for studying the internal mechanism of non-synchronous vibration.

3. Experiment Result

After completing the relevant preparations for the experiment, a number of non-synchronous vibration measurement experiments were performed under different inlet temperature conditions. As shown in Figure 4, it is the position of the NSV area on the correct mass flowrate-pressure ratio performance graph when the compressor is at different inlet temperatures under the same hardware conditions.

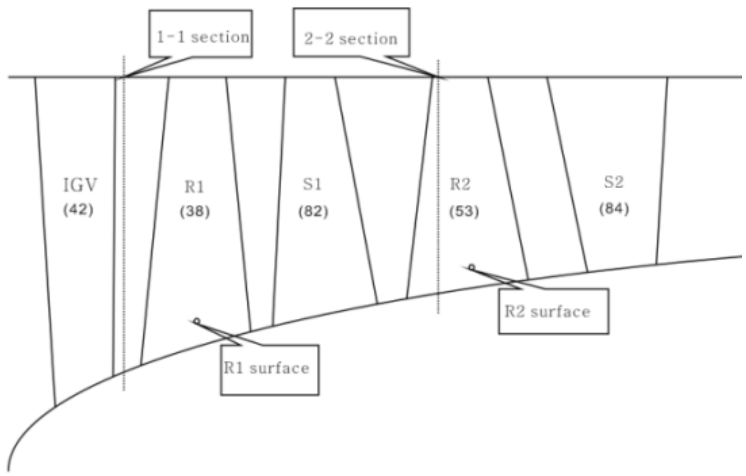

Figure 2 – Sketch of axial position of dynamic pressure transducers and strain gauges

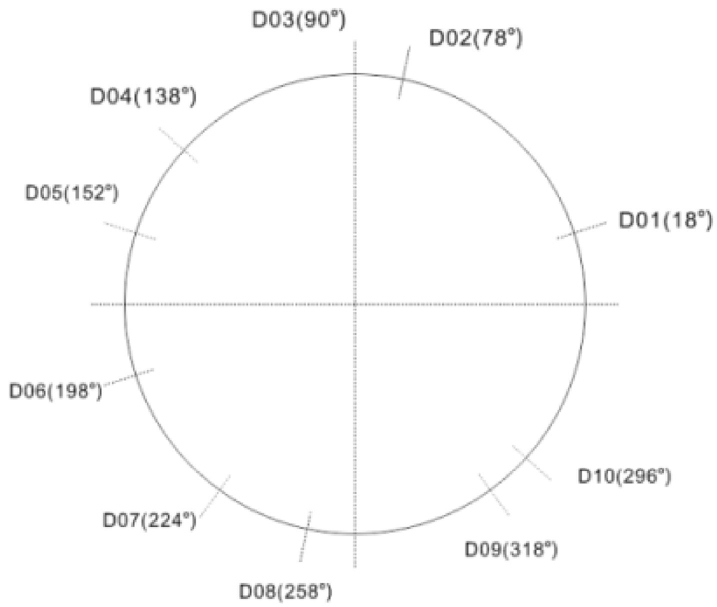

Figure 3 – Circumferential distribution of dynamic pressure transducers in 1-1 section and 2-2 section

The hot day state refers to the experiment completed in summer with an inlet temperature of about 23℃, and the cold day state refers to the experiment completed in winter with an inlet temperature of about -12℃.
By switching the exhaust throttle and the driving speed of the shaft to change the working state of the compressor, the characteristics of blade vibration along the constant speed line, along the working line, along the constant throttle line and along the blockage point are obtained.
Through these series of experiments, the appearance, increase, maximum, decrease and disappear process of the non-synchronous vibration of the nine-stage compressor at different inlet temperatures are obtained, and then the area where NSV occurs can be roughly obtained. The test results show that under different inlet temperatures, the area where NSV occurs mainly has the following characteristics:

- The most important characteristic of NSV vibration is that the dynamic stress on the first-stage rotor appears the blade vibration characteristic frequency. Meanwhile, the aerodynamic characteristic frequency appears at the dynamic pressure measuring points before and after the first-stage rotor, as well as the first-stage stator. Under the condition of constant speed, the blade vibration characteristic frequency and aerodynamic characteristic frequency remain unchanged. The magnitude of NSV vibration characteristics depends on the strength of vibration characteristic frequency and aerodynamic characteristic frequency, which are synchronous.

- When the inlet temperature of the compressor is different, the region where NSV occurs is also different. When the inlet temperature is higher, the conversion speed of the region where NSV vibration occurs is also higher. When the inlet temperature is low, the corrected speed of the NSV vibration region is also low. The NSV vibration region at different inlet temperatures does not overlap in the corrected mass flowrate-pressure ratio map.
- In the process of improving the working state of the compressor test piece from the blocked state point by turning down the exhaust throttle along the constant speed line at the maximum vibration characteristic, the vibration characteristic of NSV first gradually increases to the maximum, then gradually decreases, and finally disappears.
- When NSV occurs, its internal noise is relatively large, and the noise data measured by the sensor placed on the casing above the first-stage rotor shows up to more than 150dB.

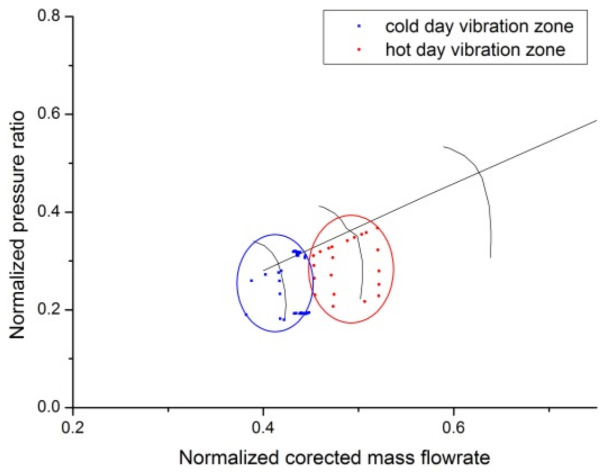

Figure 4 – Diagram of process of NSV experiment

3.1 Overall performance

As shown in Figure 5, the non-synchronous vibration appears when the test state point changes along the choke point, the dynamic pressure and blade dynamic stress changes in the process of increase and disappearance. The Y coordinate is the normalized dynamic pressure frequency at the 1-1 section, the physical speed of the test piece, and the dynamic stress frequency measured on the first-stage rotor blade, and the X coordinate is the time.

As can be seen from the figure, when the inlet temperature is fixed, the NSV region appears in the fixed speed section. The occurrence of aerodynamic characteristic frequency is synchronized with the occurrence of blade vibration characteristic frequency.

In the test process, all the dynamic pressure characteristics measured by the dynamic pressure sensor located near the first-stage rotor at the casing are basically consistent. Therefore, unless specified in the following text, the dynamic pressure or aerodynamic frequency refers to the dynamic pressure measured by the sensor at this position.

As can be seen from Fig. 5 and Fig. 6, NSV vibration has the following characteristics:

- At the same inlet temperature, the starting and ending speeds of NSV in the process of speed increase and speed decrease are not completely consistent, and the starting speed of NSV is slightly lagging behind the ending speed of NSV.

- At the same inlet temperature, the rotation speed at which the strongest NSV vibration occurs during the increase and decrease of the rotation speed is basically the same.

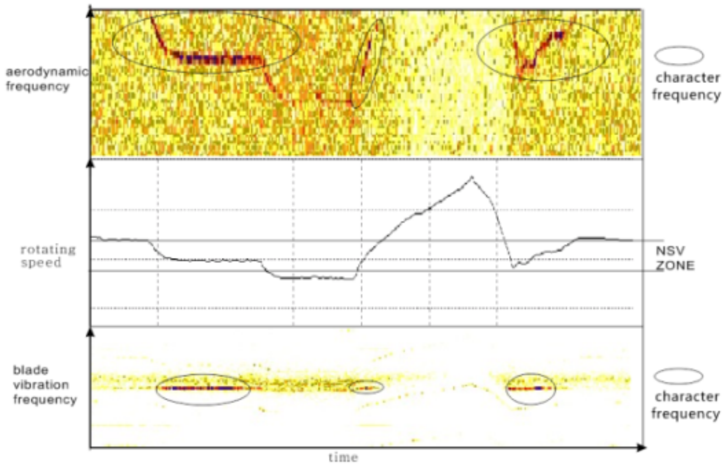

Figure 5 – Process diagram of NSV

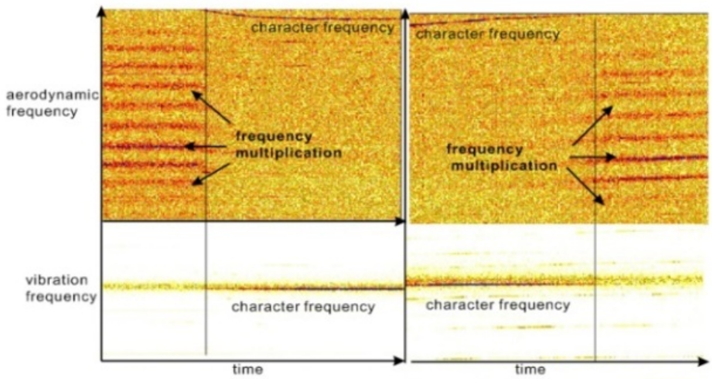

Figure 6 – Occurrence and disappearance process of NSV

In combination with Fig. 5 and Fig. 6, there is another phenomenon, that is, the occurrence of aerodynamic characteristic frequency indicates the occurrence of NSV vibration phenomenon. The vibration frequency of blades is different. When NSV vibration phenomenon does not appear, the vibration component of blade dynamic stress at the characteristic frequency is relatively low, but it is already obvious.
Figure 7 shows the change process of the measurement results of the dynamic stress sensor installed on the primary rotor blade when the compressor passes through the NSV vibration region while the speed is promoted along the working line. It can be seen from the figure that the dynamic stress has an obvious peak region with the change of rotational speed. When NSV vibration is formally formed, the vibration amplitude of blade dynamic stress at the characteristic frequency will greatly increase, with a maximum increase of more than 20 times. Then it decreases rapidly with the increase of rotational speed.

3.2 Frequency Performance

Exploring the relationship between structural vibration and gas pressure pulsation is very important to explore the nature of NSV vibration. The experiment phenomenon shows that the characteristic frequency of aerodynamics and the characteristic frequency of blades have slight changes with the change of rotation speed. In order to distinguish the relationship between aerodynamics and blade vibrations, an order tracking analysis was carried out.

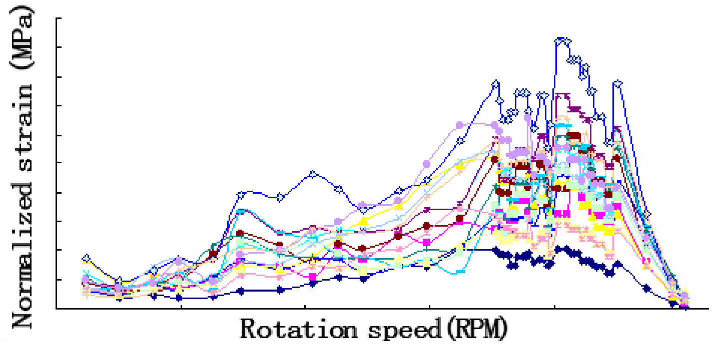

Figure 7 – The stress of different blades changes with the speed during the occurrence of NSV

Figure 8 shows one of the results. In the figure, the Y coordinate is the normalized dynamic pressure frequency at the 1-1 section and the dynamic stress frequency on the first-stage rotor blade, and the X coordinate is the rotation frequency.
The results show that the relationship between aerodynamic frequency, blade characteristic frequency and rotational frequency is approximately linear. The characteristic frequency of blade vibration increases slowly with the increase of speed, but the increase is very small, and it is difficult to distinguish without discrimination.
The main relationship between aerodynamic characteristic frequency and blade vibration characteristic frequency is as follows:

$$\Omega_R^F = |\frac{\omega_S^F + \omega_S^R}{m}| \tag{1}$$

Where $\omega_S^R$is the characteristic frequency in the rotating coordinate system, that is, the characteristic frequency of dynamic stress measured on the first-stage rotor blade. $\omega_S^F$is the frequency in the static coordinate system, that is, the characteristic frequency of pulsating pressure, and $\Omega_R^F$is Rotation frequency.
m is a positive integer, and the value of m is different at different inlet temperatures. For this compressor, when the inlet temperature is 23℃, m is 12, and when the inlet temperature is -12℃, m is 13.as shown in Table 1, The aerodynamic and structural characteristic frequencies and their relationships under the maximum vibration state at different inlet temperatures are shown.

Table 1 – aerodynamic and vibration character frequency at different inlet temperature

Inlet temperature(℃)	Rotating frequency	Vibration frequency	Aerodynamic frequency	multiple
23	178.37	752.25	1387.51	12.00
-12	165.78	744.68	1410.88	13.00

Referring to Figure 8, when NSV vibration occurs, the aerodynamic characteristic frequency and the blade vibration characteristic frequency have an approximately linear relationship with the rotation frequency. The relationship after fitting is as follows:

$$\Omega_S^F = a\omega_R^F - c \tag{2}$$

$$\Omega_S^R = b\omega_R^F + c \tag{3}$$

$$m = a + b \tag{4}$$

Among them, a and b are correlation coefficients, c is the intercept after fitting, a and b are not integers, and have no direct relationship with the number of leaves and their combination.From the above relationship between the characteristic frequency of aerodynamics and the characteristic frequency of blade vibration, they have obvious internal correlation.

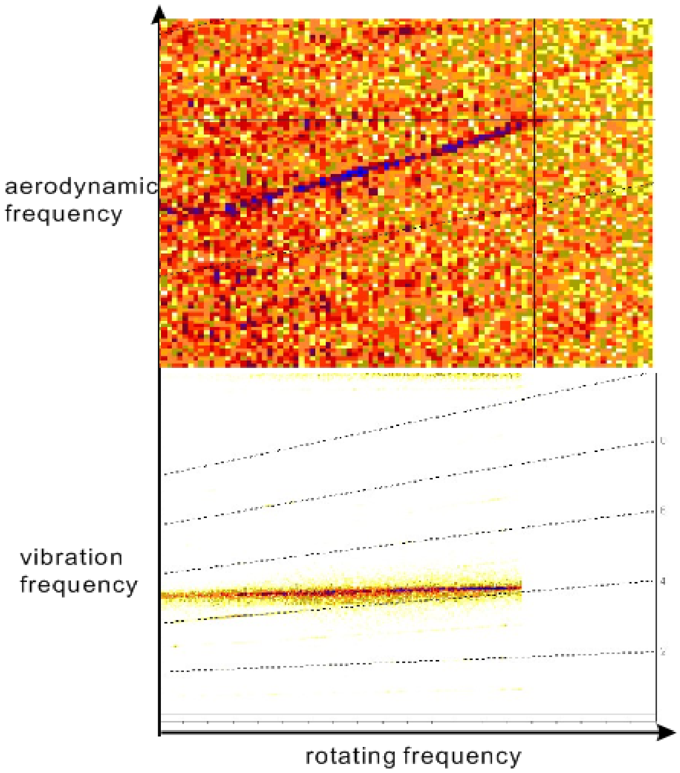

Figure 8 – Order track of aerodynamic frequency and vibration frequency

3.3 phase distribution

Frequency analysis accurately correlates structural vibration with aerodynamic pressure. In order to verify the more accurate relationship between them, the dynamic pressure sensor uniformly arranged in front of the first stage rotor and the dynamic stress sensor continuously distributed are used to analyze the propagation of dynamic pressure wave and structural vibration wave when NSV occurs. The test data of the dynamic pressure sensor installed above the first-stage rotor is subjected to cross-spectrum processing. The relationship between the phase difference and the position of the sensors is shown in Figure 9. The results of linear fitting show that there are 13 pressure wave systems with uniform distribution that propagate uniformly with steering on the first stage blade. The dynamic stress on the first-stage blade also shows similar propagation characteristics, and its phase correlation can be seen in Fig. 10. The above analysis results show that the traveling wave in the airflow and the vibration wave of the blade propagate in a direction consistent with the direction of rotation. The number of traveling waves at this inlet temperature is 13.
In acoustics testing, acoustic modal analysis can not only obtain the main sound mode orders (ie the circumferential traveling wave number) inside the compressor, but also give the sound pressure level of each acoustic mode, which is an effective means to evaluate the noise propagation characteristics inside the compressor. In this experiment, the acoustic modal analysis method is used to analyze the information of 10 dynamic pressure sensors set up in the 1-1 section. However, since the accuracy of the acoustic mode test is heavily dependent on the number of sensors (generally, $2m$+1 sensors are required to achieve the accurate test of the m-order acoustic mode), limited to the complex structure of the actual compressor, it is impossible to install a larger number of sensors. Therefore, based on the measurement point arrangement shown in Figure 2, a new "$m+n$ acoustic modal test and analysis method" is adopted to improve the identification ability of acoustic modal orders. Through 10 sensors, the sound mode orders in the 30th order range are obtained, and the sound pressure level of the maximum 5th order sound mode is given. First, the sound mode analysis method with six points uniformly distributed is used to complete the calculation and analysis of the sound mode. Figure 11 is the analysis result of a piece of data. The Y coordinate is the sound pressure level, and the X

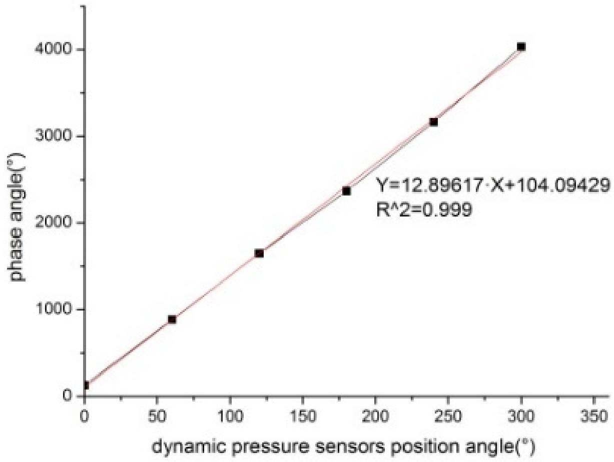

Figure 9 – Cross spectrum phase angle between wall circumferential dynamic pressures

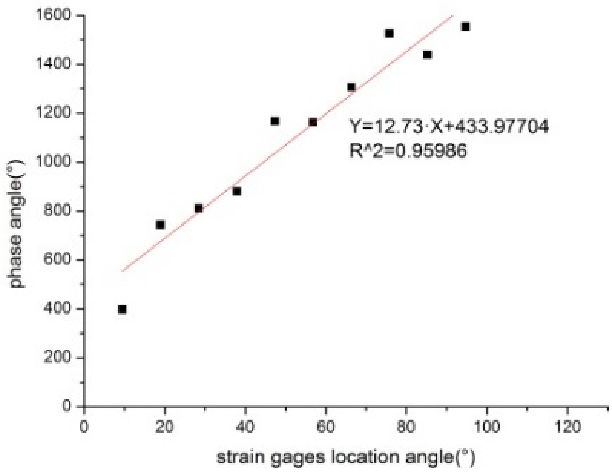

Figure 10 – Cross spectrum phase angle between blade strains

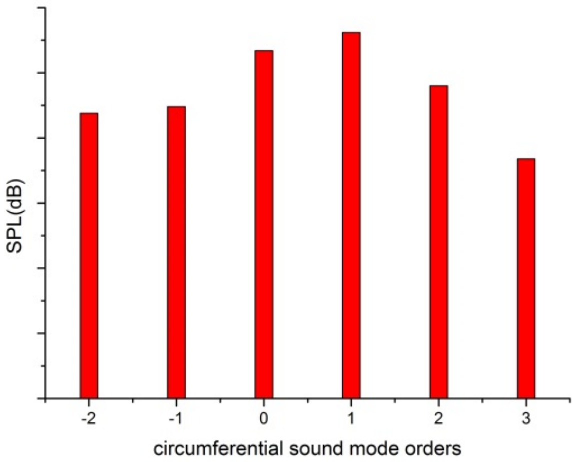

Figure 11 – 6 measurement points Sound mode analysis result in 1-1 section

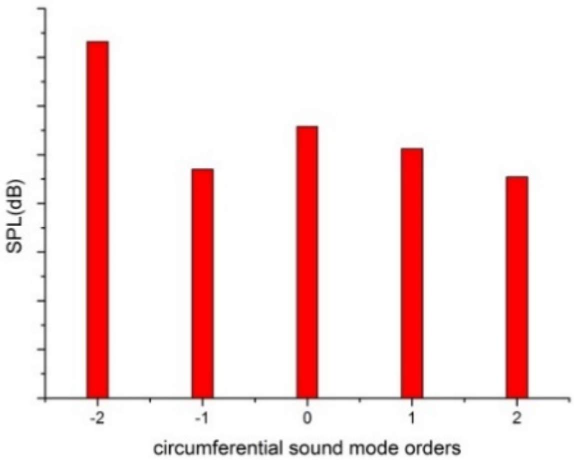

Figure 12 – 5 measurement points Sound mode analysis result in 1-1 section

coordinate is the circumferential acoustic mode order. As can be seen from the figure, limited to the number of sensors, the six-point uniform acoustic modal analysis can only realize the recognition of -2 to +3 acoustic modes, and the multiple acoustic modes have been mixed in these acoustic modes. On the above, the order of the largest acoustic mode cannot be accurately identified.
Then, the acoustic mode analysis method with five points uniformly distributed is used to complete the calculation and analysis of the acoustic mode. The analysis result is shown in Figure 12. Similarly, the five-point uniform acoustic modal analysis only realizes the recognition of the -2 to +2 acoustic modes, and the multiple acoustic modes have been mixed on these acoustic modes, and the loudest sound cannot be accurately identified. The order of the mode.
Finally, combined with the data in Figure 11 and Figure 12, using the "m+n-order acoustic modal test and analysis method", the calculation and analysis of the acoustic modal are completed, and the analysis results are shown in Figure 13. Through this method, the recognition of the sound mode orders within the 30th order is realized, and the maximum 5th order sound mode order and the corresponding sound pressure level are obtained. It can be found from the figure that the circumferential modal order of noise propagation inside the compressor is an acoustic mode with +13 order as the main characteristic, and the result is completely consistent with the phase analysis result.

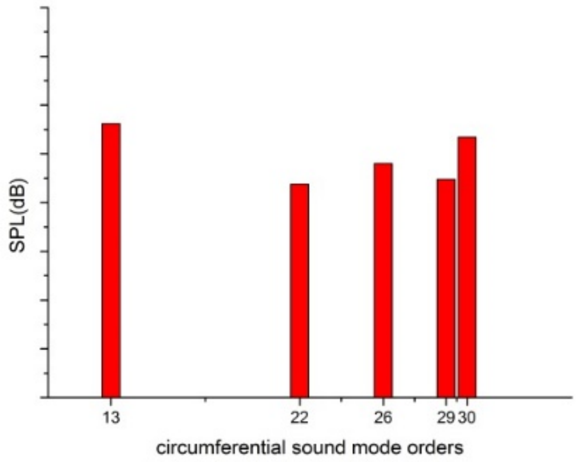

Figure 13 – 5+6 measurement points Sound mode analysis result in 1-1 section

3.4 Flow field measurement results

In NSV vibration conditions, the dynamic pressure double hole wedge probe was used to measure the rotor exit flow field. The axial Mach number of the measured results can be shown in figure 14. t The result shows that the static pressure in the rotor tip area is relatively large. If the speed is too small, the angle between the speed direction and the axial direction is larger, indicating that the secondary flow state in the tip area is active, the blockage is serious, and the axial Mach number is lower.

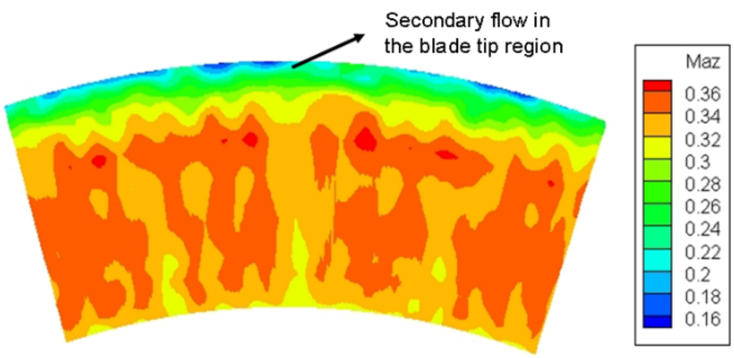

Figure 14 – Axial mach number distribution contour after the 1st rotor

4. Preliminary analysis of NSV induction mechanism

At present, researchers mainly explain the vibration mechanism of NSV in two directions, namely, the non-synchronous vibration of the blade induced by aerodynamic rotation instability and the acoustic feedback theory that the reciprocating oscillation of the tip gap leakage flow between the blades induces the non-synchronous vibration of the blade. The non-synchronous vibration of this series of experiment has similarities with them and has its own unique characteristics. Both will be discussed in this section.

4.1 Analysis of the phenomenon of rotation instability mechanism

All the signs above indicate that the structural characteristic frequency and the aerodynamic characteristic frequency are closely related, It is even speculated that they come from the same source. The theory rotational instability mechanism is similar. The theory speculates that the blade vibration caused by the leakage vortex system in the tip area . leakage vortex causes high-frequency pressure pulsation, and then interact with the blade to induce the blade vibration. In a nutshell, the rotational instability of the blade tip is the multiple circumferential traveling waves formed by the mutual influence of the tip leakage vortex system. This vibration source can be analogous to a horn emitting a fixed frequency rapidly rotating around a certain center, and the frequency in the static and static coordinate system conforms to the Doppler law in the rotating coordinate system. However, this theory cannot provide an effective explanation for different NSV speed regions and different travelling wave number m at different inlet temperatures.

4.2 Phenomenon analysis of acoustic feedback theory

Another widely accepted theory is the acoustic feedback theory proposed by Thomassin et al, from PW Corporation. The basic principle is that the tip leakage flow will impact the adjacent blades, and the adjacent blades will rebound the leakage flow. If the impact flow coincides with the natural frequency of the blade, it will cause the blade to interact with the aerodynamic frequency, resulting in non-synchronization vibration. The mechanism experiment, verification experiment and numerical simulation of this vibration theory have been completed. It is currently a relatively complete theory explaining non-synchronous vibration. Its core formula is as follows:

$$U_{tipc} = 2(C - \frac{2sf_b}{n}) \tag{5}$$

Where U_{tipc} is the tangential velocity of the rotor blade tip,C is the local sound speed, s is the pitch of the blade tip, f_b is the natural frequency of the blade, and n is any positive integer. This formula shows that when the tangential velocity of the blade tip reaches a critical value, it will excite non-synchronous vibration. The critical value is directly related to the blade pitch and the natural frequency of the blade. However, the tangential velocity of the blade tip at the beginning of the non-synchronous vibration in this paper is not consistent with the criticality, and the vibration behavior is not completely consistent. The theory does not give an explanation for the characteristic frequency in the static coordinate system.

5. Conclusion

In summary, this article mainly completed the following tasks:

• After many experiments, the dynamic stress and pressure on the rotor at the moment of NSV vibration are measured in detail, and the dynamic pressure on the inner wall of the casing is described in detail. The generation, increase, and disappearance process of NSV vibration characteristics and its circumferential propagation characteristics are described in detail;

• Through tests at different inlet temperatures, the NSV vibration speed and regional migration characteristics have been completed.

• The characteristics of forced vibration, flutter, and NSV vibration are analyzed in detail. Improved awareness of this type of NSV vibration.

• The mechanism of aerodynamic rotation instability and the mechanism of acoustic echo causing blade vibration are successively discussed. The analysis thinks that the obvious characteristics of the vibration phenomenon in this article are difficult to explain by the two.

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