MECHANISM AND PROPAGATION OF CHARACTERISTIC FREQUENCY NOISE INNER A HIGH PRESSURE COMPRESSOR

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An acoustic experiment on a high pressure compressor is implemented to investigate noise mechanism and propagation corresponding to high level vibration occurred on rotor blades. The results show that, noise spectrum tested on the first stage rotor blades of high pressure compressor presents typical characteristics for discrete multi-tone in a pre-arranged structure adjustment and specific rotating speed. These discrete frequencies are composed with blade passing frequency of the first stage rotor blades and other special characteristic frequencies. And the amplitude of characteristic frequency 1402Hz will synchronously increase with the sharp vibration of the rotor blades. Rotating speed plays an important role on the characteristic frequency. This frequency merely occur on the specific speed range, and has relationship with structure adjustment of the compressor. With the rotating instability theory, the generation mechanism of the frequency can be explained with a rotating sound source having the same speed with the first stage rotor blades. The frequency of the rotating noise source will match with the characteristic frequency at a specific mode. Through the coherent analysis between the noise measured at different testing positions, this characteristic frequency noise source originates behind the first stage rotor blade at least. Therefore, the propagation state should be from back to forward with a circumferential speed. All these analysis reveal generation mechanism and propagation characteristic of the discrete frequency inner the compressor which may have an important reference value for Engineers.

Keywords: high pressure compressor, acoustic experiment, characteristic frequency, generation mechanism, coherent analysis

1. Introduction

Aeroengine compressor is a high-speed rotating impeller machinery, which usually works in high temperature and high pressure complex environments. Therefore, the strength of rotating blades is the focus of the engine development. Unsteady flow phenomenon occurs when the compressor operated on the specific status. And this unsteady flow phenomenon not only has an important effect on the compressor's performance stability, but also may be an exciting source for the asynchronous vibration of rotor blades. More and more scholars have begun to research the mechanism of unsteady flow and characteristics of sound field inner the compressor. The more prominent research is that the instability mechanism of rotating blades flow field considered as a
noise source, which excites rotor blades to appear high level vibration. Tip clearance flow instability is widely researched as a typical unsteady flow phenomenon. It is found that tip clearance flow instability may be treated as a noise source inner the compressor. Baumgartner et al. [1] detected the high level vibration of rotor blades on a ten stage compressor. The rotating instability was proposed as the noise source which excited the vibration of blades. Mailach et al. [2][3] proposed that the blade tip flow instability which also considered as rotating instability excited the cavity acoustic mode and then blades vibration occurred. Thomassin et al. [4][5] put forward the viewpoint that the combination of rotor tip clearance flow instability and acoustic wave phenomenon induced the high level vibration of rotor blades. Kameier et al. [6] measured the pressure fluctuation close to rotor blades on a low speed axial flow compressor, and found that the characteristic frequency spectrum and the special frequency combination relation presented in the internal pressure spectrum of the compressor under specific status. Moreover, rotor blades mostly appeared non-integral order vibration which simultaneously reflected in the measuring results of unsteady pressure field through experimental studies of a multistage axial compressor under off-design conditions [7][8][9]. However, there is no a reasonable explanation about abnormal noise signal. Recently, an experimental study had been implemented on the acoustic excitation of rotor blades of a high pressure compressor [10]. It was considered that the high intensity noise wave produced inner the compressor in a special and abnormal status was one of the reasons for arousing resonance or rotor blades flutter. But the contribution of acoustic excitation to rotor blades vibration should not be ignored. In order to reveal the characteristics of noise signal inner compressor, relevant experimental research and data analysis had been carried out. The variation law of characteristic frequency noise signal corresponding to the high level blade vibration was summarized [11]. However, the research on the mechanism of noise source inner the compressor has not been implemented.

In this paper, according to the experiment data under the condition of a turbofan engine compressor testing rig, and referring to the analysis method of the unsteady flow pressure field inner the compressor mentioned by Baumgartner et al. [1]. The characteristics of noise spectrum tested on the first stage rotor blades of high pressure compressor are analyzed. And the correlation between the noise and the rotating speed is illuminated. The generation mechanism of the frequency is explained with the rotating instability theory, which has the association with the high level rotor blades. Through the coherent analysis between the noise measured at different testing positions, the noise source propagation state is expounded. All the above will provided meaningful data and theoretical analysis reference for the mechanism of rotor blades vibration inner the turbofan compressor.

2. Acoustic experiment and instrumentation

An acoustic experiment is operated on a multistage axial compressor. The noise pressure signal inner the casing wall is measured through an acoustic waveguide system. And the rotor speed is monitored simultaneously. A 1/4 inch condenser microphone is internally installed in the system. One end of the system is connected to a semi-infinite tube which applied to avoid the acoustic wave reflection inner the system, and another end is connected to the compressor with the pipe which installed flush with inner surface of the connotation. Four testing positions are determined along the same circumferential direction and the different axial direction positions in the compressor. The first position is the clearance of inlet guide vanes (IGV), the second is clearance between inlet guide vanes and the first stage rotor blades (1R), the third is directly above the first stage rotor blades and the fourth is the clearance of the first stage stator blades (1S). The schematic diagram of testing positions is showed in Figure 1.
3. Generation mechanism of characteristic frequency noise

3.1 Frequency spectrum analysis

The time-domain wave of noise is transformed into frequency spectrum to reveal the special frequency structures by fast Fourier transform. The frequency interval of spectrum analysis is 25 Hz. The noise frequency spectrum with special frequency structures measured above the first stage rotor blades (position 3) in a pre-arranged structure adjustment and specific rotating speed in the compressor is shown in Figure 2. It is shown that the noise spectrum tested on the first stage rotor blades of high pressure compressor presents typical characteristic for discrete multi-tone in a pre-arranged structure adjustment and specific rotating speed. These discrete frequencies are composed with blade passing frequency (1BPF and 2BPF) of the first stage rotor blades and other special characteristic frequencies.

In the compressor component testing of a turbofan engine, the engine speed was pushed up from 9480 r/min to 10560 r/min in a specific structural adjustment condition. In order to acquire the correlation between characteristic frequencies and the rotor speed, noise frequency spectrums are analyzed at different rotating speed. It is indicated in Figure 3 that the amplitude of characteristic frequency 1402Hz will synchronously increase with the sharp vibration of the rotor blades in combination with the experiment. This characteristic frequency merely occur on the specific speed range, and has the relationship with structure adjustment of the compressor. Rotating speed plays an important role on the frequency.

Figure 3: Frequency spectrum from 9720r/min to 10560r/min.
3.2 Mechanism analysis

It is mentioned in the relevant literature [1] that the non-uniform flow inner the compressor will develop into the rotating instability mechanism under specific conditions, which interacts with rotor blades. And this rotating instability mechanism is just the noise source rotating with rotor blades.

Under the fixed coordinate frame (F), the instability pressure wave $p(\varphi^F, t)$ preventing the circumferential rotation can be described as a series of sine wave with different circumferential mode number $\alpha$. And the amplitude $A$ and the phase angle $\Phi$ change with the time. Thus the pressure wave form can be expressed as

$$p(\varphi^F, t) = \sum_{n=-\infty}^{\infty} \sum_{\alpha=-\infty}^{\infty} A_{\alpha n}^F \cos(\alpha \varphi^F - \omega^F_n t - \Phi^F_{\alpha n}), \quad (1)$$

If the rotating noise source frequency is $\omega^{SM}_{\alpha n} / 2\pi$, in the noise source coordinate frame (S), equation (1) can be the represented as

$$p(\varphi^S, t) = \sum_{n=-\infty}^{\infty} \sum_{\alpha=-\infty}^{\infty} A_{\alpha n}^S \cos(\alpha \varphi^S - \omega^S_n t - \Phi^S_{\alpha n}), \quad (2)$$

Similarly, in the rotor blades coordinate frame (R), the above pressure fluctuation can be expressed as

$$p(\varphi^R, t) = \sum_{n=-\infty}^{\infty} \sum_{\alpha=-\infty}^{\infty} A_{\alpha n}^R \cos(\alpha \varphi^R - \omega^R_n t - \Phi^R_{\alpha n}), \quad (3)$$

The schematic diagram of the rotating direction of the rotating noise source coordinate frame and the rotor blades coordinate frame relative to the fixed coordinate frame is shown in Figure 4.

![Schematic diagram of the rotating direction in the different coordinate frame.](image)

**Figure 4:** Schematic diagram of the rotating direction in the different coordinate frame.

In Figure 4, in the different coordinate frame, the noise frequency conversion relation can be expressed as

$$\varphi^F = \varphi^S + \Omega^F_S t, \quad (4)$$

$$\varphi^F = \varphi^R + \Omega^F_R t, \quad (5)$$

$$\varphi^S = \varphi^R + \Omega^S_R t, \quad (6)$$

$$\Omega^F_R = \Omega^F_S + \Omega^S_R, \quad (7)$$

Equation (2) can be the represented with (4) as
p(\phi^r, t) = \sum_{n=-\infty}^{\infty} A_n^r \cos(n \omega_s^r - (\omega_n^s + \Omega_s^r) t - \Phi_n^r) \quad (8)

For the rotating noise source frequency $\omega_{SM}^r/2\pi$, comparing equation the (8) and (1), this frequency can be obtained in the fixed coordinate frame

$$\omega_{SM}^r = \omega_{SM}^r + \alpha \Omega_s^r, \quad (9)$$

Through investigating rotor blades vibration in the compressor component experiment induced by rotating instability (RI), it is found that the spectral structure of the noise signal measured above the rotor blade as shown in Figure 2 is very similar to the analogical studies when high level vibration occurs in high pressure compressor rotor blades. Therefore, this research refers to the theoretical method of rotating instability which used to explain the generation mechanism of the characteristic frequency noise source corresponding to high level rotor blades vibration.

![Figure 5: Noise spectrum corresponding to high level rotor blades vibration](image)

As shown in Figure 5, the characteristic frequency 1402Hz is revealed as 1410Hz with 1Hz frequency interval of spectrum analysis. Based on the theory of rotating instability, the frequency 1410Hz and its side-band frequencies with equal frequency interval engendered by the modulation frequency of the rotating noise source in the presence of rotational instability are the rotating noise source frequency $\omega_{SM}^r/2\pi$ measured in the fixed coordinate frame. Since high level vibration with the first order vibration frequency occurred on the first stage rotor blades of the high pressure compressor. And there were no other excitation sources related to blades vibration inner the compressor. The unsteady pressure fluctuation near to rotor blades may lead to this vibration phenomenon. In other words, there is a rotating instability noise source with the first order vibration frequency of the first stage rotor blades inner the compressor. So this frequency is evaluated as $\omega_{SM}^r/2\pi=746Hz$ based on the results of blades vibration monitoring. The rotating speed of this frequency is expressed as $\Omega_s^r=166Hz$ in the fixed coordinate frame synchronized with rotor blades. The circumferential fluctuation of the rotating instability noise source will lead to different circumferential mode number $\alpha$. The frequency of the rotating noise source measured in the fixed coordinate frame can be calculated inversely though substituting the above $\omega_{SM}^r/2\pi$ and $\Omega_s^r$ into the equation (9). The computed results are presented in Table 1.

<table>
<thead>
<tr>
<th>$\omega_{SM}^r / 2\pi$ (Hz)</th>
<th>746</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\omega_{SM}^f / 2\pi$ (Hz)</td>
<td>166</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>-3</td>
</tr>
<tr>
<td>$\omega_{SM}^f / 2\pi$ (Hz)</td>
<td>248</td>
</tr>
</tbody>
</table>
Through the above analysis, it is found that the frequency 1410Hz and its side-band frequencies with equal frequency interval are recovered by using the rotating instability theory. Therefore, the rotating instability theory can be applied to expound the mechanism of the characteristic frequency noise source corresponding to high level rotor blades vibration. However, it should be noted that the noise source frequencies of other circumferential mode numbers have not appeared in the noise signal spectrum. That may be associated with the inner structure size of the compressor, which resulted in the frequency cutoff of other modes. Furthermore, the circumferential modal distribution of the rotating noise should be measured tentatively. That is to say, the amplitude and the pressure fluctuation frequency of the noise should be captured on the blade. Thereby, the mechanism of the characteristic frequency noise source inner the compressor can be verified though the rotating instability theory.

4. Propagation state of characteristic frequency

In order to reveal the propagation regularity of characteristic frequency noise signal corresponding to high level rotor blades vibration inner the compressor, noise signals at four testing positions have been analyzed through cross-spectrum analysis. And the propagation regularity of the characteristic frequency 1402Hz is expounded by its phase relationship at different testing positions. The phase difference results of the characteristic frequency at different testing points are shown in Table 1.

<table>
<thead>
<tr>
<th>Testing position</th>
<th>(1-2)</th>
<th>(1-3)</th>
<th>(2-3)</th>
<th>(3-4)</th>
<th>(2-4)</th>
<th>(1-4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Phase difference (°)</td>
<td>-115</td>
<td>-142</td>
<td>-242</td>
<td>-28</td>
<td>-128</td>
<td>-99</td>
</tr>
<tr>
<td>Phase relationship (°)</td>
<td>(1-2)+(2-3)=(1-3)</td>
<td>(2-3)+(3-4)=(2-4)</td>
<td>(1-2)+(2-3)+(3-4)=(1-4)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

As shown in Table 1, for example (1−2) = −115° is expressed as the phase of the characteristic frequency 1402Hz at position 1 is ahead of the phase at position 2 about 115°. That is mean that the characteristic frequency noise propagates from position 2 to position 1, and similar to others. The phase results of the characteristic frequency at different testing positions have the following relationship (1−2) + (2−3) = (1−3), (2−3) + (3−4) = (2−4) and (1−2) + (2−3) + (3−4) = (1−4). Therefore, the characteristic frequency noise signal corresponding to high level rotor blades vibration at the four testing positions are the identical noise source. The phase relation of the characteristic frequency is lagging in turn from position 1 to position 4. So this characteristic frequency noise source should originate behind the first stage rotor blades at least. Combining with the results in 3.2, the propagation state should be from back to forward with a circumferential speed.

5. Conclusion

In this paper, an acoustic experiment on a high pressure compressor is implemented. The analysis results show that, noise spectrum tested on the first stage rotor blades of high pressure compressor presents typical characteristics for discrete multi-tone in a pre-arranged structure adjustment and specific rotating speed. And these discrete frequencies are composed with blade passing frequency of the first stage rotor blades and other special characteristic frequencies. The amplitude of characteristic frequency 1402Hz will synchronously increase with the sharp vibration of the rotor blades. Rotating speed plays an important role on the characteristic frequency. This
frequency merely occur on the specific speed range, and has the relationship with structure adjustment of the compressor.

The characteristic frequency generation mechanism can be explained with a rotating sound source having the same speed with the first stage rotor blades based on the rotating instability theory. The frequency of the rotating noise source will match with the characteristic frequency at a specific mode. This characteristic frequency noise source should originate behind the first stage rotor blade at least. And the propagation state should be from back to forward with a circumferential speed. All these analysis reveal generation mechanism and propagation characteristic of the discrete frequency inner the compressor, and will have an important reference value for Engineers.

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