ABSTRACT

Dangerous blade excitation caused by unsteady flow in a high pressure/high mass flow compressor running in a low mass flow region has been investigated. Experiments were carried out for compressors with two different types of vaned diffusers. Blade vibration was measured with strain gauges while simultaneous unsteady pressure was measured with fast response dynamic transducers. All measured results were analysed in detail so that an in-depth understanding of blade excitation mechanism can be obtained. Firstly, the compressor with a straight-channel vane diffuser at reduced rotational speed of 12,300 rpm in an unstable operation region was considered. The analysis of blade vibration and unsteady pressure showed an unusual excitation phenomenon. Besides a strong blade vibration frequency component near the first blade mode frequency excited by the rotating stall cells existed another dangerous resonance excitation with first blade mode component which dominated the blade vibration spectrum. A detailed pressure signal analysis indicated that this blade vibration was excited by a broad band pressure fluctuation due to a strong reverse flow occurring simultaneously with the rotating stall. Further reducing the compressor mass flow to the operation point shortly before surge, the rotating stall was significantly weakened while the reverse flow kept its intensity until surge occurred. In this operation region blades suffered throughout a violent excitation of resonance because of the strong reverse flow. These blade excitation phenomena were also found in the next experiment for the compressor with a cambered vane diffuser at higher rotational speeds of \( n_{rem} = 13,500 \) and 14,000 rpm. The maximum strain values of blade vibration were obtained to quantitatively estimate the danger of blade vibration caused by this excitation.

INTRODUCTION

Rotating stall as an unsteady phenomenon occurring in the part load compressor operation zone was considered to play an important role in the blade excitation due to its periodic occurrence (Haupt et al. 1986; Hagiwara et al. 1987; Ishihara and Funkawa 1980). Recent research results showed that not only the main frequency components of rotating stall can be the cause of dangerous blade vibration, but also the circumferentially asymmetric amplitude profile of rotating stall pressure pattern had a strong additional excitation on blades (Jin et al. 1992a). The blade vibration stress excited by the doubled frequency component of rotating stall due to its non-sinusoidal pressure profile had reached such a high level that it had already exceeded the allowable stress limits of the blade (Hasemann et al. 1991).

The experimental results in the paper presented a blade excitation phenomenon with dangerous vibration stress in a
location of semiconductor transducers:
1) $x = -10 \text{ mm}$  
2) $x/s = 0.1$  
3) $x/s = 0.55$  
4) $x/s = 1.0$  
5) throat of diffuser vanes  
6) diffuser exit  
7) $57^\circ$ circumferentially located

Fig. 2 Cross-sectional view of the test compressor with the indication to the location of strain gages and pressure transducers

high load and high speed centrifugal compressor with vaned diffusers. Blade vibration signals showed an unusually strong excitation in an unstable operation region of the compressor. An intermittent rotating stall with three rotating cells against impeller rotation was determined by a phase analysis of the pressure signals obtained from the pressure transducers circumferentially located in the throats of the diffuser vanes. This rotating stall pressure pattern strongly excited the blades with a frequency component of

$$f_b = 3(f_s + f_{rot}) = 3\cdot f_s + f_p$$

near the first blade mode. Here $f_s$ stands for the real impeller rotational frequency; $f_{rot}$ for the rotational frequency of the stall cells; $f_p$ for the frequency of the rotating stall pressure pattern obtained by high frequency pressure transducers and $f_p = 3\cdot f_{rot}$. In addition, the dangerous resonance of blade excitation with first blade mode component dominated the blade vibration spectra. The observations and the analyses of the flow and the blade vibration behavior presented in this paper aimed to achieve a better understanding of the internal flow structure and the blade excitation mechanism as well as their physical essences.

NOMENCLATURE

**Subscripts**

4  
diffuser inlet  
6  
diffuser exit  
b  
blade  
p  
pressure oscillation  
red  
reduced to reference condition  
rs  
rotating stall  
s  
shaft, impeller  
tot  
total

**Fig. 3 Impeller used in the blade vibration investigation**
The experiment was performed for two different vaned diffusers, straight channel vane diffuser and cambered vane diffuser respectively. The former is shown in Fig. 4(a), which has 19 straight wedge vanes. The vane leading edge is located at a radius ratio \( \lambda_4 \) of 1.15 and extends to a radius ratio \( \lambda_6 \) of 1.95. The wedge angle is 9° 57' and is so set that the angle between the pressure side surface and the tangent to the leading edge circle is 17° 48'. The diffuser width is 23.8 mm. The latter, the cambered vane diffuser is shown in Fig. 4(b). The diffuser width is 21.6 mm. The vane leading edge is located at the radius ratio \( \lambda_4 \) of 1.15 and extends to the \( \lambda_6 \) of 1.5. The vane inlet angle is 17°.

MEASURING SYSTEM

The semiconductor strain gages located in the zone of blade inlet shown in Fig. 2 were used to measure blade vibration. The fundamental calculations (Haupt et al. 1985a) and latter tests (Haupt et al. 1985b) indicated that this is the zone where the maximum stresses occur for the vibration in first and second blade modes. These modes were indeed the main modes excited when the compressor operates in the unsteady flow region.

The strain gages on different blades with locations shown in Fig. 2 and 3 were connected by thin wires to a 8-channel FM-telemetry transmitter mounted on the bore of the hollow shaft at the impeller inlet. This transmitter is the rotating part of the telemetry system described in detail by Haupt et al. (1985a). The blade vibration signals transmitted by this system with trigger were recorded on a separate magnetic tape. One of these blade signals was recorded simultaneously with the different unsteady pressure signals on the 14-channel tape recorder.

Flow measurements were made by means of several high frequency response semiconductor pressure transducers to determine the time dependent characteristics of unsteady flow. These transducers were mounted on the compressor shroud along the flow path from the impeller inlet to the diffuser vane exit. The propagation of rotating stall pressure pattern in the meridional direction was examined. The structure of the rotating stall pressure pattern was determined by the phase analysis of the pressure signals obtained from the pressure transducers circumferentially located in the throats of the diffuser vanes. A schematic display of the transducer positions is shown in Fig. 2.

RESULTS AND DISCUSSIONS

- Compressor with straight channel vane diffuser

The investigation of pressure and blade vibration behavior was carried out along constant speed line. For each speed line several operating points were taken before and during the existence of flow oscillations. The flow oscillations behavior characterised by rotating stall were determined and plotted in Fig. 5. The flow characteristics for all operating points at different speed lines were described by Haupt et al. (1988). In this paper a very interesting flow phenomenon as well as its effect on blade vibration occurring at the constant speed of \( n_{\text{red}} = 12,300 \) rpm will be reported.

At the compressor operating point of rotating speed of \( n_{\text{red}} = 12,300 \) rpm and mass flow rate \( \dot{m}_{\text{red}} = 2.99 \text{ kg/s} \), the pressure signals obtained from pressure transducers and the blade vibration signal from strain gages were unstable. The time period of the unstable pressure oscillation signals from the impeller inlet to the diffuser exit and the blade vibration signal taken simultaneously were plotted in Fig. 6 (A). An intense unstable periodic pressure oscillation and violent blade vibration can be observed. The period of the unstable periodic pressure oscillations was about 0.5 second and the time intervals between two unstable pressure oscillation periods were about 1 second. The strongest periodic pressure oscillations occurred at \( x/s = 0.55 \) and the diffuser inlet. No significant periodic pressure oscillation can be observed.

![Fig. 4 Shape of the different types of diffuser vanes used in the blade vibration experiments](image-url)
found at the impeller inlet and the diffuser exit. An evident blade vibration with a maximum strain value of 1.43 mm/m can be determined during this period of time. Figure 6 (B) provides another view of this pressure signals where the periodic oscillation components were filtered and the pressure scale was amplified. This picture shows an evident sustained rise in pressure level at the impeller inlet and a drop in diffuser region when the periodic pressure oscillations appear. These changes in pressure level can be explained as that the compressed air with high pressure and high energy flows along full circumference of the shroud from the diffuser region to the impeller inlet. The pressure value at the diffuser exit drops even more sharply because of the compressed air leaks to the impeller inlet and through compressor outlet tube. When the reverse flow stops, the raised pressure at the impeller inlet drops to the value which is lower than that before occurrence of the pressure oscillation. Then, the pressure in the compressor recover to the normal level. In the recovery period blades are weakly excited and the pressure change profile is similar to that during surge, which was reported by Haupt et al. (1987a) and Jin et al. (1992b). However, the magnitude of the pressure changes here is smaller than in that case.

Further reducing the compressor mass flow rate the pressure oscillation period was extended while the time interval between two pressure oscillation periods was shortened, until the sustained pressure oscillation was formed. At this operation point the parameters of compressor characteristics can not be determined because of the intense pressure oscillation. Figure 7 (A) shows the pressure signals, two long periodic pressure oscillation periods of 4.5 second with a short time interval about 0.3 second in between. The violent blade vibration excited by this pressure oscillation was also plotted. In order to investigate the change in pressure value of this periodic pressure oscillation the component of the pressure signals were also filtered and plotted in Fig. 7 (B). The pressure signals at the time interval between two periodic pressure oscillation periods show a pressure behavior in normal flow. The pressure values at the impeller inlet are lower than that during periodic pressure oscillations. In diffuser, on the other hand, the pressure values are higher. This can be a further indication that the continuous periodic pressure oscillations were accompanied by a reverse flow mentioned above.
significant amplitude of the pressure oscillation with broad band characteristics can be observed. The maximum value of this amplitude at the impeller inlet is 3 mbar. The broad band behavior of the pressure spectra represents the characteristics of reverse flow in compressor, which was described by Haupt et al. (1987b). Blades can be excited by the pressure oscillation with broad band characteristics, especially, at the impeller inlet, where blades are sensitive to excitation.

Frequency analyses were also performed for blade vibration signals. The blade vibration spectrum with average value is shown in Fig. 9. A significant amplitude of the discrete frequency component of 664 Hz was excited by the rotational cells of the rotating stall pressure pattern \((f_p = 3 \cdot f_s + f_p = 3 \times 208 + 38 = 662 \text{ Hz})\). Here \(f_s\) stands for the impeller rotational frequency and \(f_p\) for the measured pressure oscillation frequency of the rotating stall. The excitation with a frequency components of 856 Hz (first blade mode) showing a broad band characteristics is the response of the pressure oscillation to the reverse flow. The discrete frequency components of 832 Hz (4 \(\cdot f_s\)) are possibly excited by the circumferentially asymmetric static pressure distribution with four periodic waves, which are induced by the interaction between the blade vibration and the circumferentially asymmetric reverse flow. This complex excitation mechanism was described by Haupt et al. (1989).

An oil injection experiment was conducted to confirm the reverse flow mentioned above. Oil was injected in the flow channel near the impeller exit \((x/s = 0.8)\) through a hole in the shroud wall, when the compressor was operated at \(n_{red} = 12,300 \text{ rpm}\) and during the occurrence of the continuous rotating stall pressure pattern. The results of this experiment is given in Fig. 10. The coloured dye in the picture represents the reverse flow conditions near the shroud. It is demonstrated by this experiment that the reverse flow extends as far as a location next to the impeller inlet, as shown in this picture, before the shroud wall was removed.

The mass flow of the compressor was further slowly reduced to surge. Shortly before the occurrence of the surge a
very interesting flow phenomenon appeared. The pressure and blade vibration signals at this time are shown in Fig. 11 (A). The pressure oscillations characterizing rotating stall pressure pattern are few in amount and far apart. Pressure pulses at the impeller inlet with a period from 0.25 to 0.3 second and a slight amplitude can be observed. However, the blade vibration signal displays an intense unstable vibration strain. The maximum vibration strain is the same as that during the occurrence of the rotating stall pressure pattern. The pressure and blade vibration signals obtained at the compressor operating point before the occurrence of the rotating stall pressure pattern are shown in the right picture in Fig. 11 (A) to compare the flow behavior and the blade excitation. The picture in Fig. 11 (B) shows the blade vibration strain and pressure signal again. The high pressure frequency components were filtered and the pressure scale was amplified. Comparing the normal flow shown in the right picture in Fig. 11 (B) the pressure level for the compressor operating point shortly before surge at the impeller inlet is higher and at the diffuser exit is lower. This pressure level change characterizes the reverse flow from the diffuser region to the impeller inlet.

The pressure spectra with average value from the impeller inlet to the diffuser exit are shown in Fig. 12, which show a significant broad band characteristics of the pressure signals. Furthermore, the frequency spectra with the average value of the pressure signals measured at the position 10 mm before the impeller inlet are shown in Fig. 13 which displays the process of the pressure behavior changing with mass flow rate reduction at the rotational speed of \( n_{\text{red}} = 12,300 \) rpm. At the compressor operating point of a large mass flow rate, for example \( n_{\text{red}} = 3.87 \) and \( 3.48 \) kg/s, the pressure spectra show a obvious amplitudes of the discrete frequency component of 2920 Hz which are caused by the blade pressure difference between the pressure and the suction side while the pressure oscillation amplitude showing
waves induced by the interaction between blade vibration and the circumferentially asymmetric reverse flow (Haupt et al. 1989). The frequency component at 856 Hz (first blade mode) with broad band characteristics demonstrates the response of the reverse flow.

---

Similar investigations were performed for the compressor with a cambered vane diffuser. The compressor operating points displaying pressure oscillation characteristics of rotating stall pressure pattern were plotted on the compressor map (Fig. 15). At the compressor operating points near surge at the rotational speed of \( n_{\text{red}} = 12,300 \) and \( 12,900 \) rpm rotating stall pressure pattern with three rotating cells directed against the impeller rotation can be determined. However, the pressure oscillation amplitudes of this pressure pattern are not significant and the blade vibration strains excited by this pressure oscillation are also very weak. On the rotational line of \( n_{\text{red}} = 14,500 \) rpm at the point before surge a significant periodic pressure oscillation can be observed, which was defined as rotating stall pressure pattern with two rotating cells directed against the impeller direction (Jin et al. 1992a).

At the rotational speed \( n_{\text{red}} = 13,500 \) and \( 14,000 \) rpm the pressure signals display the intense pressure oscillations. The phase analysis defined that it is caused by the rotating stall with three cells mentioned above. The blade vibration and pressure signals at \( n_{\text{red}} = 13,500 \) rpm and \( n_{\text{red}} = 3.42 \) kg/s are shown in Fig. 16. The periodic pressure oscillations with frequency value of 20 Hz were defined as the rotating stall pressure pattern. The pressure level changes at the

---

The blade vibration signal shortly before surge was also analysed. The blade vibration signal spectrum with average value was plotted in Fig. 14. The discrete frequency component with small amplitude of 664 Hz was excited by the rotating stall pressure pattern \( f_b = 3 \cdot f_s + f_p = 3 \times 208 + 38 = 662 \) Hz. The significant amplitude of the discrete frequency component at 832 Hz (4 \( \cdot f_s \)) was excited by a circumferentially asymmetric static pressure profile with four periodic waves.

---

Fig. 13 Pressure spectra at \( n_{\text{red}} = 12,300 \) rpm with mass flow rate reduction

Fig. 14 Frequency analysis of the blade vibration signal at \( n_{\text{red}} = 12,300 \) rpm and at the operation point shortly before surge
Fig. 16 Blade vibration and pressure characteristics during intermittent rotating stall at \( n_{\text{red}} = 13 \), 500 rpm \( \dot{m}_{\text{red}} = 3.42 \) kg/s impeller inlet and in the diffuser region mentioned above are not easy to be identified. However, the very rough pressure signals with chaotic pressure oscillations at the position of 10 mm before the impeller inlet and at the diffuser exit during the occurrence of the rotating stall pressure pattern can be observed. The pressure and blade vibration signal analyses were performed for this case. The pressure spectra at the compressor operating point during the occurrence of continuous rotating stall pressure pattern, which is directly followed by surge, were plotted in Fig. 17. It can be found that the pressure oscillations with significant broad band characteristics dominated the pressure spectra. The maximum amplitude value at the impeller inlet reaches 5 mbar in a broad frequency range. This broad band pressure characteristics caused by reverse flow can be a main motivation of blade vibration.

The average values of blade vibration frequency spectra with a reduction of mass flow rate are shown in Fig. 18. At the operating point of \( \dot{m}_{\text{red}} = 3.42 \) kg/s the blade vibration spectrum due to the intermittent rotating stall shows a significant frequency component. The discrete frequency component with the value of 700 Hz is the response of the excitation caused by the rotating stall pressure pattern \( f_b = 3 \cdot f_s + f_p = 3 \times 226 + 20 = 698 \) Hz). The frequency components of 900 Hz are excited by the circumferentially asymmetric static pressure profile with four periodic waves \( f_b = 4 \cdot f_s = 904 \) Hz\(\text{(Haupt et al. 1989). However, the broad band excitation with the maximum value at both 840 Hz (first blade mode) and 1720 Hz (second blade mode) dominates the frequency spectra at the operating point with intermittent rotating stall pressure pattern. Here it must be payed attention that the natural frequency of the measured blade in this case is lower than that for the experiment of the compressor with straight channel vane diffuser. At the operating point with continuous rotating stall pressure pattern this excitation is even stronger.

At the rotational speed of \( n_{\text{red}} = 14,000 \) rpm only the intermittent rotating stall pressure pattern with three cells directed against the impeller rotation, which is immediately followed by surge, can be determined. The blade vibration and pressure signals during the rotating stall were plotted in Fig. 19.
Table 1. Summary of unsteady flow and blade vibration data

<table>
<thead>
<tr>
<th>Diffuser</th>
<th>N_{red} (rpm)</th>
<th>Mass flow rate (kg/s)</th>
<th>Measured pressure frequency (Hz)</th>
<th>Cell of rotating stall (mm)</th>
<th>Cell relative speed (m/s)</th>
<th>Max. blade strain (mm/m)</th>
<th>Max. blade strain during surge (mm/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Straight wedge vane</td>
<td>12, 300</td>
<td>2.99</td>
<td>-3</td>
<td>3.8</td>
<td>-3</td>
<td>6.1</td>
<td>1.43</td>
</tr>
<tr>
<td>Continuous R.S.</td>
<td>200</td>
<td>3.42</td>
<td>-3</td>
<td>2.0</td>
<td>-3</td>
<td>2.9</td>
<td>0.89</td>
</tr>
<tr>
<td>before surge</td>
<td>14, 000</td>
<td>3.42</td>
<td>-3</td>
<td>1.8</td>
<td>-3</td>
<td>3.4</td>
<td>1.35</td>
</tr>
</tbody>
</table>

The maximum strain of the blade vibration excited by the pressure oscillation patterns mentioned above and that during surge are shown in Table 1. The maximum blade vibration strains in the compressor with straight channel diffuser at the rotational speed of 12, 300 rpm are very high. During the intermittent and continuous rotating stall the maximum blade strain values are 1.43 and 1.30 mm/m, respectively. At the compressor operating point shortly before surge, when the rotating stall weakens, the maximum blade strain excited simply by the reverse flow can also reach 1.34 mm/m. The maximum blade strain during surge at this rotational speed is also 1.34 mm/m. (Jin et al. 1992). The danger of this blade vibration becomes evident, if the material data of the impeller is considered. According to the Goodman-diagram for the type of aluminum alloy of the impeller shown in Fig. 21 the degree of danger can be evaluated by a rough estimation. The strain value of σ_{max} = 1.43 mm/m represents blade stress of σ = 101 N/mm² for this material. The diagram shows that even for a low value of static load on the blade due to the centrifugal force, the considered vibration represents a case near the allowed stress limit and safety margin. The maximum values of blade strains in the compressor with cambered vane diffuser are also significant. In the case of rotational speed N_{red} = 13, 500 rpm the maximum blade strain is 0.89 mm/m because of the slightly weak rotating stall and reverse flow. At the rotational speed N_{red} = 14, 000 the maximum blade strain due to the intermittent rotating stall pressure pattern and the strong reverse flow is 1.34 mm/m which is the same as that during surge.

CONCLUSIONS

The rotating stall pressure patterns with three cells directed against impeller rotation are always accompanied by the reverse flow from the diffuser exit to the impeller inlet.

This rotating stall pressure pattern can develop into a simple reverse flow when compressor mass flow rate is further reduced to the point shortly before surge.

The reverse flow is characterised by a broad band behavior of the pressure frequency spectra in the compressor, pressure rises at the impeller inlet and pressure drops in the diffuser region.

Fig. 20 Frequency analyses of the blade vibration signals at N_{red} = 14, 000 rpm

Fig. 19. The pressure signals display a strong periodic pressure oscillation with a frequency of 13 Hz in the impeller channel as well as a significant chaotic pressure oscillation with an evident pressure level change at the impeller inlet and the diffuser exit. This pressure behavior is similar to that described for the case of straight channel diffuser which indicates a strong reverse flow from the diffuser exit to the impeller inlet accompanied by the intermittent rotating stall pressure pattern.

The frequency analysis of the blade vibration signals performed for the time periods with and without rotating stall pressure pattern respectively. In the time period with rotating stall pressure pattern in Fig. 20 the frequency component at 720 Hz (second blade mode) is excited by the reverse flow. The spectrum at the time period without rotating stall shows no significant frequency component amplitude excited by rotating stall and reverse flow. The frequency component of 930 Hz is excited by the circumferentially asymmetric static pressure profile (fb = 4\cdot f_s = 940 Hz).

The maximum strains of the blade vibration excited by the pressure oscillation patterns mentioned above and that during surge are shown in Table 1. The maximum blade vibration strains in the compressor with straight channel diffuser at the rotational speed of 12, 300 rpm are very high. During the intermittent and continuous rotating stall the maximum blade strain values are 1.43 and 1.30 mm/m, respectively. At the compressor operating point shortly before surge, when the rotating stall weakens, the maximum blade strain excited simply by the reverse flow can also reach 1.34 mm/m. The maximum blade strain during surge at this rotational speed is also 1.34 mm/m. (Jin et al. 1992). The danger of this blade vibration becomes evident, if the material data of the impeller is considered. According to the Goodman-diagram for the type of aluminum alloy of the impeller shown in Fig. 21 the degree of danger can be evaluated by a rough estimation. The strain value of σ_{max} = 1.43 mm/m represents blade stress of σ = 101 N/mm² for this material. The diagram shows that even for a low value of static load on the blade due to the centrifugal force, the considered vibration represents a case near the allowed stress limit and safety margin. The maximum values of blade strains in the compressor with cambered vane diffuser are also significant. In the case of rotational speed N_{red} = 13, 500 rpm the maximum blade strain is 0.89 mm/m because of the slightly weak rotating stall and reverse flow. At the rotational speed N_{red} = 14, 000 the maximum blade strain due to the intermittent rotating stall pressure pattern and the strong reverse flow is 1.34 mm/m which is the same as that during surge.

CONCLUSIONS

The rotating stall pressure patterns with three cells directed against impeller rotation are always accompanied by the reverse flow from the diffuser exit to the impeller inlet.

This rotating stall pressure pattern can develop into a simple reverse flow when compressor mass flow rate is further reduced to the point shortly before surge.

The reverse flow is characterised by a broad band behavior of the pressure frequency spectra in the compressor, pressure rises at the impeller inlet and pressure drops in the diffuser region.

Fig. 21 Goodman-diagram for aluminium alloy showing allowable alternating bending stress values
Besides the excitation of the blade vibration caused by rotating stall pressure pattern near the blade resonance frequency, the blades are also intensively excited by the reverse flow with broad band characteristics. The blade vibration strain can reach such a value that it is near the allowed values of the impeller material. The reverse flow without rotating stall pressure pattern can also excite dangerous blade vibration.

When the engine order frequency is located near the blade mode, the blades can be excited by the circumferentially asymmetric static pressure profile with periodic waves which is induced by the interaction between the blade vibration and the circumferentially asymmetric reverse flow.

The blade vibration strain due to these unsteady flow phenomena mentioned above can reach such a high level that it is near the allowable values of the impeller material.

ACKNOWLEDGEMENT

The research described in this paper was funded by the Germany Research Association (DFG). The authors are grateful to DFG for their support. The advice of Mrs. X. Huang of Washington State University is gratefully acknowledged. The authors would also like to thank Mr. P. Tanneberg for his contribution in running the tests and Ms. X. Yu for her drawing work in this paper.

REFERENCES


