Rotating Instabilities in an Axial Compressor Originating From the Fluctuating Blade Tip Vortex

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Rotating instabilities (RIs) have been observed in axial flow fans and centrifugal compressors as well as in low-speed and high-speed axial compressors. They are responsible for the excitation of high amplitude rotor blade vibrations and noise generation. This flow phenomenon moves relative to the rotor blades and causes periodic vortex separations at the blade tips and an axial reversed flow through the tip clearance of the rotor blades. The paper describes experimental investigations of RIs in the Dresden Low-Speed Research Compressor (LSRC). The objective is to show that the fluctuation of the blade tip vortex is responsible for the origination of this flow phenomenon. RIs have been found at operating points near the stability limit of the compressor with relatively large tip clearance of the rotor blades. The application of time-resolving sensors in both fixed and rotating frame of reference enables a detailed description of the circumferential structure and the spatial development of this unsteady flow phenomenon, which is limited to the blade tip region. Laser-Doppler-anemometry (LDA) within the rotor blade passages and within the tip clearance as well as unsteady pressure measurements on the rotor blades show the structure of the blade tip vortex. It will be shown that the periodical interaction of the blade tip vortex of one blade with the flow at the adjacent blade is responsible for the generation of a rotating structure with high mode orders, termed a rotating instability.
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1 Introduction

Extensive investigations of aerodynamic instabilities in axial compressors have been carried out in the last years. The main objective of this research is to avoid critical flow conditions and to extend the stable operating range of compressors.

There are two major different stall inception patterns. The first one is a short length scale disturbance, which is known as ''spike.'' It is a small rotating stall cell which is in its initial phase limited to the blade tip region of only one or a few blade passages. It grows within a few rotor revolutions to a fully developed rotating stall cell.

The second pattern is the modal wave stall inception. In this case a long length scale wavelike disturbance rotates around the annulus. Often it occurs only a short time before exceeding the stability limit. The breakdown of the flow field is initiated in a velocity minima of the modal wave. Camp and Day [1] give a detailed overview on the recent work on these two stall inception patterns.

Another aerodynamic instability, which is quite different to the phenomena described above, can be found already in the stable range of compressors. The disturbance can be described by a group of superimposed modes or a part-span stall with fluctuating cell numbers. This phenomenon is known as rotating instability (RI). It does not necessarily lead to full-span rotating stall inception, but it is responsible for an intensification of the tip clearance noise and for the excitation of rotor blade vibrations.

The first description of this unsteady flow phenomenon is given by Mathioudakis and Breugelmans [2]. They described a group of rotating stall cells with different cell numbers in a single-stage axial compressor. Mongeau [3] and Bent et al. [4] found this instability in centrifugal compressors. Mongeau and Quinlan [5], Kameier [6], Krane et al. [7], Kameier and Neise [8], and März et al. [9] observed this aerodynamic instability in different axial fans. In high-pressure compressors RIs were investigated by Baumgartner et al. [10] and Witte and Ziegenhagen [11].

In pressure or velocity frequency spectra RIs are to be found as a significant amplitude increase within a frequency band below the BPF, usually superimposed by peaks with constant frequency spacing. This pattern is related to a group of consecutively numbered modes. The frequencies of RIs are not harmonically related to the rotor frequency.

The main parameters of RIs in axial machines vary over a wide range. Mode orders of RIs of about 1/3 up to the double rotor blade number have been observed. Thus the circumferential wavelength of this disturbance is only about 0.5–3.0 times the rotor blade pitch. The rotation velocity of RIs in different machines varies between 25 percent and 90 percent of rotor velocity.

Major influence parameters on the origination of RIs axial compressors are the tip clearance of the rotor blades, the pressure difference across the rotor blade row and the axial component of velocity depending on the operating point.

März et al. [9] presumed the periodical interaction of the blade tip vortex with the tip flow of the neighboring blade to be the reason for the origination of RIs. However, recent numerical investigations of Hoying et al. [12] and experiments of Inoue et al. [13] showed the influence of the blade tip vortex on short length scale disturbances and rotating stall inception.

This paper describes experimental investigations of the flow field at the casing, on the rotor blades and within the rotor blade passages respectively the tip clearance. The objective is to show how the blade tip vortex influences the origination of RIs.

2 Experimental Facility

The experiments have been carried out in the Dresden LSRC. The compressor consists of four stages, which are preceded by an inlet guide vane, Fig. 1. Detailed descriptions of the construction of the compressor are given by Boos et al. [14] Table 1 gives a summary of the main design parameters.

It was assumed that RIs are a local phenomenon, restricted to the blade tip region of the rotor blade row. At the nominal tip clearance ratio $s^* = 1.3$ percent (tip clearance/chord length at

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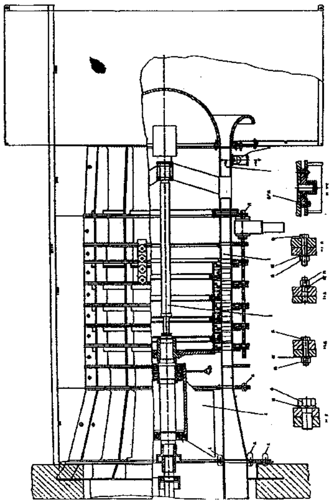

Fig. 1 Sectional drawing of Dresden LSRC

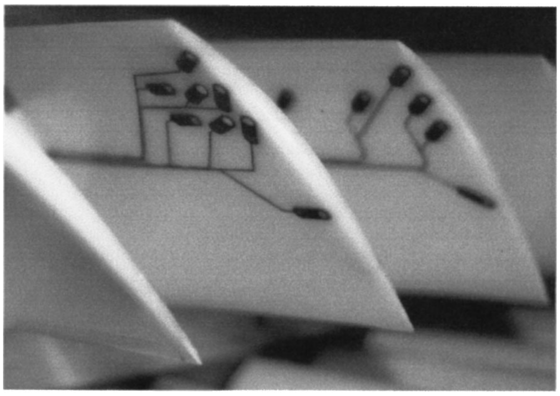

Fig. 2 Time-resolving pressure transducers on the PS of rotor blades

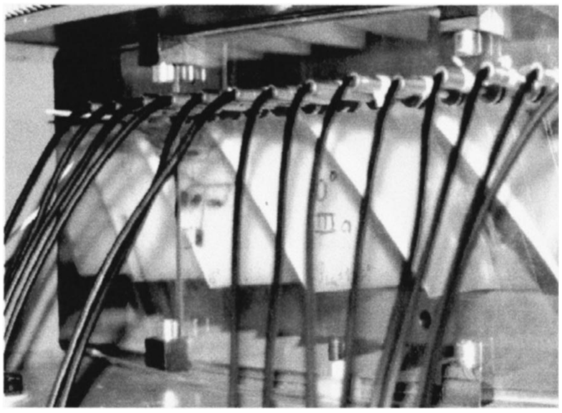

Fig. 3 Configuration C1 of microphones at the casing

blade tip) it could not be observed. To induce it, the tip clearance of the third stage was enlarged step by step from the nominal size to 2.0 percent, 3.0 percent, and finally 4.3 percent. This stage was chosen for the investigations because the behavior of it is typical for a middle stage of a high-speed compressor. Detailed results of conventional flow measurements were also available there.

Different measurement techniques were used to investigate the flow field of the compressor. A 2D-LDA system was applied to determine the axial and circumferential velocity components of the flow field within the rotor blade passage. Measurements within the tip clearance were only possible for a tip clearance ratio of 4.3 percent. The measuring volume was 0.06 mm×0.06 mm×0.63 mm.

For postprocessing the data were ensemble averaged. Time-resolved results of these measurements are not available because of a low tracer particle rate. Detailed descriptions of the LDA system are given by Müller et al. [15].

Due to the large scale of the compressor, many time resolving pressure sensors could be used.

Kulite LQ47 sensors were applied to acquire unsteady pressures on the rotor blade surfaces, Fig. 2. They were mounted at both pressure and suction sides of two adjacent rotor blades of the third stage. By this arrangement correlations between the signals of both sides of the blade as well as between different blades are possible. The signals were transmitted via rotating amplifiers and slip rings into the stationary system. All signals were acquired simultaneously.

One-quarter-inch capacity microphones (Microtech MK301) were used to measure the pressure fluctuations at the casing wall in two measuring configurations.

Table 1 Design parameters of Dresden LSRC

Hub diameter	1260 mm
Hub to tip ratio	0.84
Design speed	1000 rpm
Reynolds number at inlet of rotor, midspan	$5.7 \cdot 10^5$
Mach number at inlet of rotor, midspan	0.22
Mass flow at design point	25.35 kg/s
Number of rotor blades	63
Chord length of rotor blades, midspan	110 mm
Chord length of rotor blades, tip	116 mm
Nominal tip clearance of rotor blades	1.5 mm
Solidity of rotor blades, tip	1.55
Stagger angle of rotor blades, tip	40.5°
Number of stator blades	83
Chord length of stator blades, midspan	89 mm

An array of microphones, distributed over the circumference at constant axial position, was installed to determine the expected high mode orders of RIs, Fig. 3 (configuration C1). The microphones were mounted at the casing wall at the leading edge position of the rotor blades. An equal spacing of 1.5° enables the acquisition of mode orders up to 120. The 30 available microphones allow to observe the pressure fluctuations over a 45° section of the circumference simultaneously. To determine the mode orders one of the sensors always remained at a fixed position as a reference, while the others were moved stepwise over the whole circumference.

A second measuring configuration C2 consists of an array of microphones, covering the area of one rotor blade passage at the casing at different axial and circumferential positions. They were arranged in different traces following the rotor blade contour. This configuration was used for determining the propagation velocity and direction of the perturbation in the fixed frame of reference.

3 Influence of Tip Clearance and Operating Point on RIs

First signs of RIs have been found at a tip clearance ratio of 3.0 percent. At a larger tip clearance ratio of 4.3 percent it was fully developed. It has been found at all investigated rotor speeds (50 percent, 80 percent, and 100 percent design speed) [16,17]. The results shown in this section are referred to the measurements with the largest tip clearance and design speed only.

The formation of RIs is limited to a narrow operating range near the stability limit. At design speed this includes the range of dimensionless mass flow rates of about $\xi = 0.87 - 0.81$, Fig. 4. At the operating point $\xi = 0.87$ a narrow band increase of the ampli-

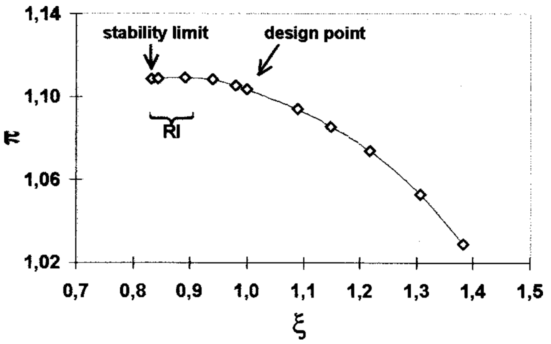

Fig. 4 Compressor characteristic for design speed

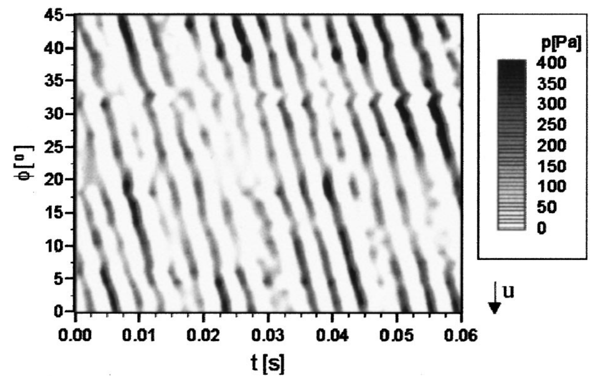

Fig. 6 Propagation of RIs in circumferential direction at the casing wall, axial position at the leading edge of the rotor blades, design speed, $\xi=0.82$

tudes in the frequency spectrum at about 30 percent of blade passing frequency (BPF) can be observed. Approaching the stability limit by throttling the compressor, the RI shifts to slightly lower frequencies while the amplitude of the perturbation grows. Figure 5 shows a typical frequency spectrum measured at the casing wall for an operating point near the stability limit. The largest amplitudes of RIs can be found at a frequency of 265 Hz. This is 25 percent of 1. BPF of rotor blades. Furthermore a modulation of 1. BPF and RI frequencies can be seen. Referred to the rotating system RIs can be detected within about the same frequency range.

At the same time two different modal waves (first and second order) can be observed in the whole compressor. They are represented as discrete peaks at much lower frequencies than RIs (about 1 percent of BPF respectively 50 percent of rotor frequency). An interaction of these long-length disturbances with the short-length RIs could not be detected.

Measurements on the rotor blades show that RIs are limited to the blade tip region. Maximum amplitudes appear at the positions of the upper measuring traces at 92 percent of the blade height and 20–30 percent of chord length. The disturbance is more pronounced on the pressure side (PS) than on the suction side (SS). At the casing highest amplitudes occurs at the same axial position. A second maximum can be observed in the axial gap downstream the rotor blade row at the casing.

The mode orders and the rotating velocity of RIs were determined using the C1 configuration of microphones. Figure 6 shows the pressure fluctuations at the casing over a 45° section of the circumference (≈8 rotor blade pitches) for about one rotor revolution. The signals were band-pass filtered (finite impulse response filter) to suppress higher frequency parts (for instance rotor wakes), low-frequency parts of the signal (like modal waves) and stochastic disturbances. It can be seen that RIs propagate in rotor direction with 50–60 percent of rotor velocity. The estimated mean wavelength in circumferential direction is only about two rotor blade pitches. Hence the mean mode order of RIs is about 30. This is nearly half the rotor blade number (63). Strong fluctuations of the amplitude, frequency and wavelength of the disturbance can be observed.

For an exact determination of the mode orders the microphones were moved stepwise over the whole circumference. One sensor always remains at a fixed position as a reference. In that way evenly distributed signals were achieved. The phase angle of the cross power spectra of each sensor referring to the fixed reference sensor permits the detailed evaluation of the mode order depending on the frequency. A detailed description of this method and the results for different operating points are given by Mailach [16,17].

As an example Fig. 7 shows the mode orders for an operating point at design speed near the stability limit. It complements Figs. 5 and 6, which show results for the same operating point. Figure 7 indicates that RIs can be described by a group of superimposed modes. This is the reason why RIs can be identified as an amplitude increase in a frequency band. The mode orders of RIs rise with frequency; they are consecutively numbered. Mode orders of about 10–40 can be attributed to the disturbance. The highest amplitude of RIs given for a mode order of 28. This agrees very well with the observations in the time domain, Fig. 6.

The strong fluctuation of RIs shown above is the reason for the appearance of RIs as an amplitude increase in a frequency band in the spectrum, Fig. 5. These fluctuations are averaged in the frequency spectrum over a certain time period.

Contrary to that modal waves or rotating stall are flow phenomena with constant mode order respectively cell number. This is the reason for their appearance as a discrete peak in the spectrum, Fig. 5.

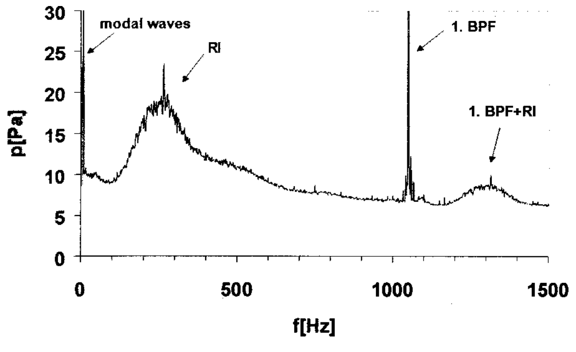

Fig. 5 Frequency spectrum at the casing, design speed, $\xi=0.82$, $s^*=4.3$ percent

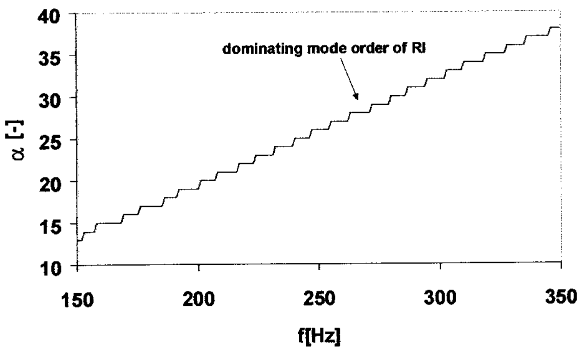

Fig. 7 Mode orders of RIs, design speed, $\xi=0.82$

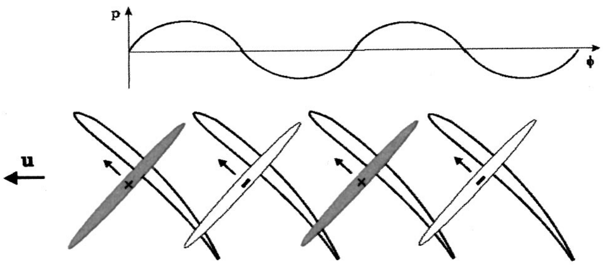

Fig. 8 Propagation of RIs at the casing wall and circumferential pressure distribution at profile leading edge, t=const (fixed frame of reference)

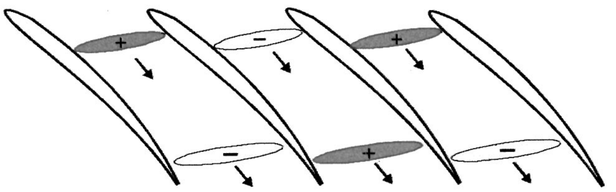

Fig. 9 Propagation of RIs in the blade tip region, t=const (relative frame of reference)

Up to now only the circumferential distribution and propagation of RIs was described. Using statistical evaluation methods Mailach [16,17] could show that RIs propagate in the blade tip region and at the casing wall like wavefronts. The measurements (C2) reveal a periodic reversed axial flow component at the casing, Fig. 8. This corresponds to the findings of Kameier [6] who showed reversed flow due to RIs with hot-wire measurements within the tip clearance. März et al. [9] also confirm this, using measuring data of an array of time-resolving pressure sensors at the casing.

As the dominating mode order is about half the rotor blade number that means, the same process repeats simultaneously at every other blade. This is schematically shown in Fig. 8 (+ and − marks the pressure maxima and minima of RIs.

The propagation direction in the blade tip region is determined by correlating the signals of pressure transducers on PS and SS of two adjacent rotor blades, Fig. 9. With respect to the rotating frame of reference the circumferential propagation takes place

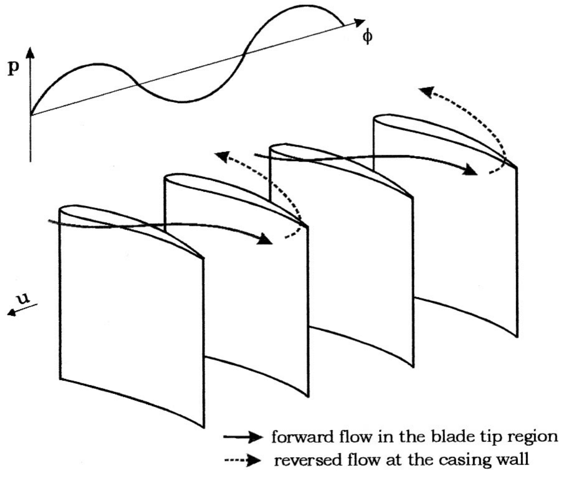

Fig. 10 Propagation direction of RIs in the blade tip region, rotating system

against rotor turning direction. Contrary to the propagation at the casing wall respectively within the tip clearance the axial component of RIs in the blade tip region is directed downstream.

Both in the blade tip region and at the casing wall the phase difference between leading and trailing edge of the profile is about 180° for constant circumferential position. That means if at the leading edge a pressure maximum appears at the trailing edge a pressure minimum can be found, etc.

To use a consistent frame of reference the observations made in the fixed frame at the casing wall have been transformed into the rotating system. With respect to the relative frame the flow within the tip clearance of the rotor can be described by a reversed axial component and a counterrotating circumferential component of RIs. The periodical flow due to RIs in the blade tip region also counterrotates, but propagates downstream. Thus a spiral propagation of the rotating instability modes against the rotor direction takes place, whereby the disturbance propagates like wavefronts, Fig. 10.

4 Influence of the Rotor Blade Tip Vortex on the Formation of RIs

The flow field in the blade tip region of the rotor blades is dominantly influenced by the blade tip vortex. While the blade tip vortex occurs at each rotor blade, these investigations show dominating mode orders of RIs of only half the rotor blade, *i.e.* the disturbance occurs at the same time at each other blade.

The tip vortex is induced by the pressure difference between PS and SS of the rotor blades. The maximum vortex intensity can be expected in the region of the maximum pressure difference between PS and SS, Inoue et al. [18]. The rollup of the vortex can be located there.

The main influence parameter on the blade tip vortex is the tip clearance height. Other parameters are the thickness of casing wall boundary layer, Reynolds number, Mach number, blade thickness, blade loading, and speed.

4.1 Nominal Tip Clearance, $s^*=1.3$ percent. First evidence on the location of the maximum tip vortex strength is given by the time-averaged pressure differences between positions at PS and SS of a rotor blade, Fig. 11. Measuring positions on a line at $r^*=92$ percent were used for this. Only signals of sensors at comparable chordwise positions will be compared.

The largest pressure difference between PS and SS at design point can be seen close to the leading edge of the blade. Approaching the stability limit this difference grows. Thus the vortex intensity increases in this region, while it is decreases in the rear part of the blade.

The results of the LDA measurements are in accordance with these observations. For nominal tip clearance at design point Fig. 12 shows the distributions of the relative flow angle in a measuring plane at $r^*=95$ percent.

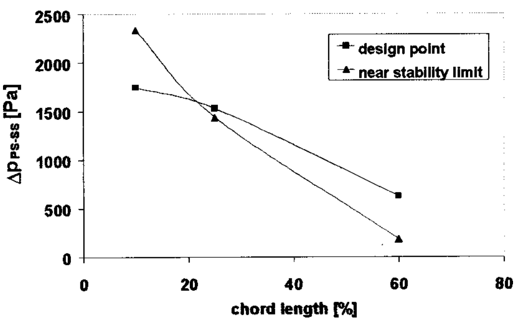

Fig. 11 Pressure difference between PS and SS of a rotor blade vs chord length, $s^*=1.3$ percent, $r^*=92$ percent, design speed

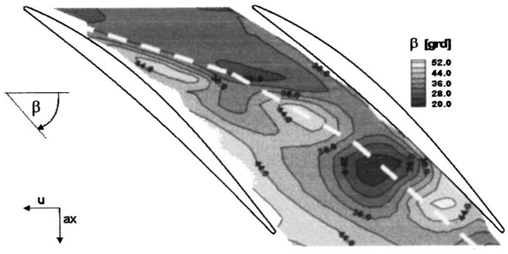

Fig. 12 Relative flow angle in the blade tip region of the rotor blades, nominal tip clearance ($s^*=1.3$ percent), $r^*=95$ percent, design speed, design point ($\xi=1.0$). Dashed line: expected vortex trajectory

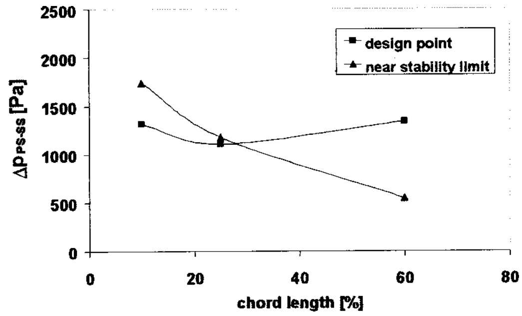

Fig. 14 Pressure difference between PS and SS of a rotor blade vs chord length, $s^*=4.3$ percent, $r^*=92$ percent, design speed

The relative flow angle distribution indicates a spiral blade tip vortex. Three areas of large flow angles and two areas of small flow angles clearly illustrate this. These regions of local under- and overturning show the intersection points of the vortex with the measuring plane. The axial velocity distribution shows the blade tip vortex as a region of low velocities between the front part of the SS of one blade and the rear part of the PS of the adjacent blade, Müller et al. [15]. Extremely low axial velocity components are to be found in the regions of underturned flow.

If the flow rate is reduced the vortex is displaced upstream with a slightly stronger inclination toward the circumferential direction, Fig. 13. This coincides with the results of the experiments of Saathoff and Stark [19] and the numerical investigations of Hoying et al. [12]. The reason for this is the reduced axial velocity component. The distances between the intersection points of the vortex with the measuring plane are also clearly smaller.

An interaction of the blade tip vortex and the flow along the PS of the adjacent blade seems to be possible in the rear part of the blade at both operating points. The only evident periodical influence on the pressure difference between PS and SS can be attributed to the stator wakes (compare Fig. 20 for large tip clearance).

4.2 Large Tip Clearance, $s^*=4.3$ percent. At design point the pressure difference between PS and SS is nearly constant along the chord, Fig. 14. Thus the time-averaged tip vortex intensity is not depending on the axial position. However, at an operating point near the stability limit again an increase of this driving pressure difference at the leading edge can be observed, which indicates that an increase of the vortex intensity can be expected.

The results of LDA measurements show significant changes in the nature of the blade tip vortex for large tip clearance.

Already at design point in the measuring plane through the tip clearance large regions of reversed flow can be observed. This seems to be due to the rolling-up of the blade tip vortex. The positive pressure gradient across the rotor blade row intensifies this reversed flow in the plane inside the tip clearance.

Figure 15 shows this region with low axial velocity or even axial reversed flow for design point. In the front part of the blades a zone of low axial velocity which means an underturning of the flow can be seen in the middle of the blade passage. The relative flow angle remains positive. The reversed flow region within the tip clearance is limited to the rear part of the blades and extends over the whole circumference. Minimum axial velocities and relative flow angles can be observed between the middle of the blade passage and the PS. This can be explained by the existence of a large tip vortex. This corresponds to the observations of Inoue and Kuroumaru [18], who also observed the center of the reversed flow region away from the SS at a comparable tip clearance ratio.

In the measuring plane near the blade tips at $r^*=92$ percent a low velocity area is visible in the rear part of the blades towards the PS, Fig. 16. This is caused by the reversed flow due to the tip vortex which radially moves away from the casing. This results in a blockage of the flow in the rear part of the rotor blades, Fig. 19. This blockage region causes a turning of the flow towards the hub and the SS of the blades. Near the SS an overturning of the flow can be seen, Fig. 16.

The reversed flow region within the tip clearance is shifted upstream when approaching the stability limit, Fig. 17. This is due to the reduced axial velocity component and the increase of the pressure difference between PS and SS at the leading edge. In operating points near the stability limit the blade tip vortex can be

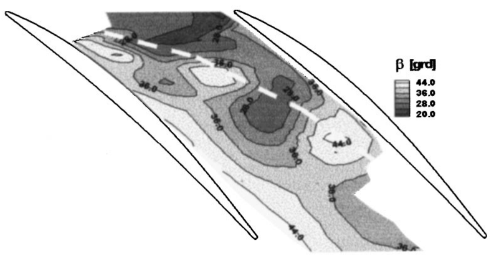

Fig. 13 Relative flow angle in the blade tip region of the rotor blades, nominal tip clearance ($s^*=1.3$ percent), $r^*=95$ percent, design speed, operating point near stability limit ($\xi=0.85$). Dashed line: expected vortex trajectory

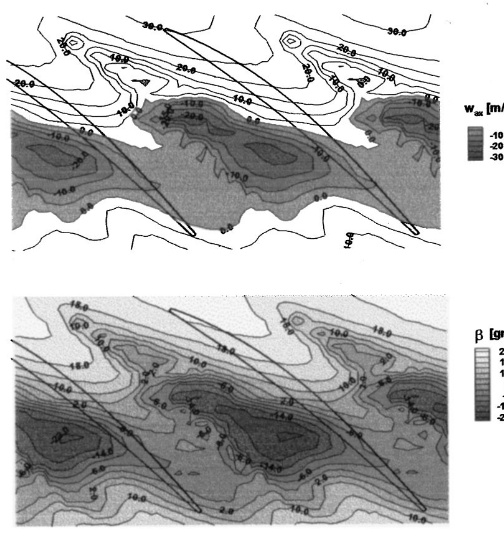

Fig. 15 Axial component of velocity and relative flow angle within the rotor blade tip clearance, large tip clearance ($s^*=4.3$ percent), $r^*=97.9$ percent, design speed, design point ($\xi=1.0$)

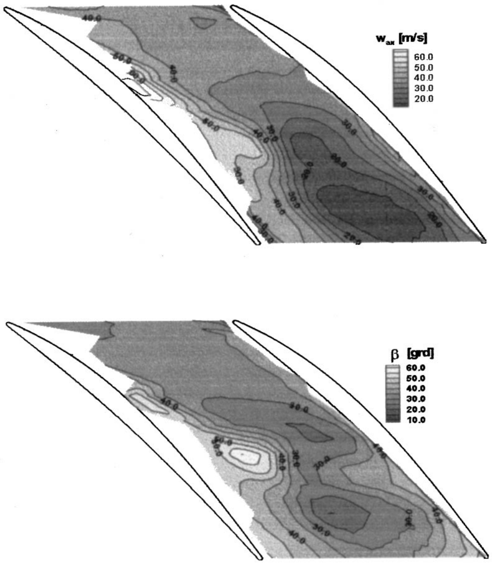

Fig. 16 Axial component of velocity and relative flow angle in the blade tip region of the rotor blades, large tip clearance ($s^*=4.3$ percent), $r^*=92.0$ percent, design speed, design point ($\xi=1.0$)

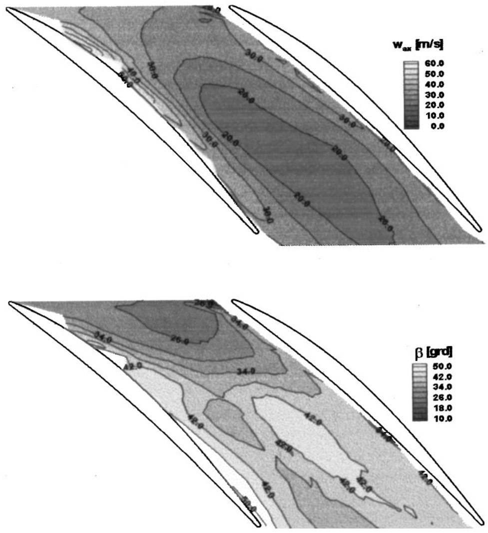

Fig. 18 Axial component of velocity and relative flow angle in the blade tip region of the rotor blades, large tip clearance ($s^*=4.3$ percent), $r^*=92.0$ percent, design speed, operating point near stability limit ($\xi=0.85$)

located in the front part of the blade, Fig. 18. A strong underturning of the flow occurs near the leading edge within the blade passage. Thus an interaction of the tip vortex of one blade and the flow at the tip of the PS of the adjacent blade is likely. According to the shifted reversed flow region the area of low velocity in the blade tip region also moves upstream. Minimum velocities can be located in the middle of the blade passage at about 50 percent of chord length. This low velocity region is restricted to about the upper quarter of the blade height.

One should remind that the results of LDA measurements are ensemble-averaged. It will be shown that the flow field shown for the operating point close to the stability limit is strongly influenced by periodic fluctuations. These parts of the signal disappear due to the averaging of the results explained above.

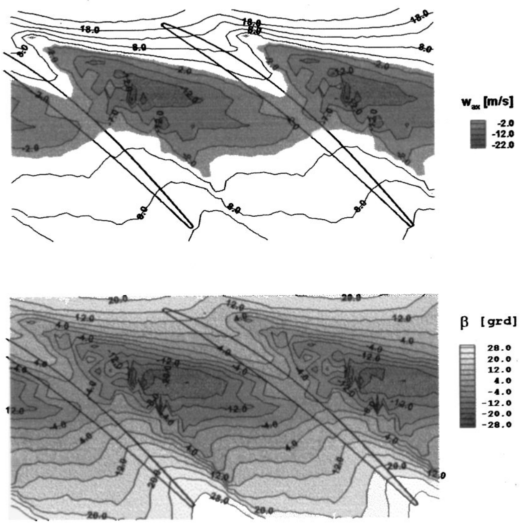

Fig. 17 Axial component of velocity and relative flow angle within the rotor blade tip clearance, large tip clearance ($s^*=4.3$ percent), $r^*=97.9$ percent, design speed, operating point near stability limit ($\xi=0.85$)

The time dependence of the blade tip vortex can be described using once more the data of the unsteady pressure measurements on the rotor blade surface. At design point, where no signs of RI have been found, the pressure difference between PS and SS is strongly periodically influenced by the stator wakes, Fig. 20.

At an operating point near the stability limit a totally different behavior can be seen. In addition to the stator wakes another much stronger influence appears, Fig. 21. Especially in the tip region of the front part of the blade strong pressure decreases can be seen for short time periods. This seems to be caused by the reversed flow of the blade tip vortex through the tip clearance. These pressure drops, which are stronger at the PS, strongly diminishes the pressure difference between PS and SS. For short moments the pressure on the SS is larger than on the PS, which means that the tip clearance flow is stopped or even a reversed flow trough the tip clearance from the SS to the PS, Fig. 21. The strongest influence can be observed at positions where also maximum amplitudes of RIs have been found.

Contrary to the design point the reversed flow affects the flow in a region where the maximum pressure difference between PS and SS is to be found. Thus the consequences of the reversed flow of one blade tip vortex on the formation of the tip vortex of the adjacent blade are much stronger.

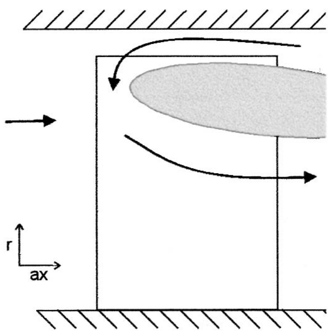

Fig. 19 Blockage in the blade tip region induced by the blade tip vortex, large tip clearance ($s^*=4.3$ percent)

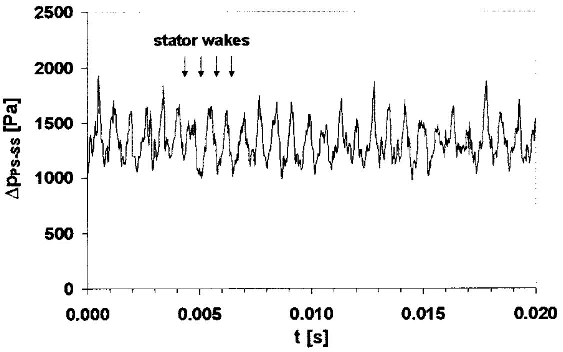

Fig. 20 Periodical influence of stator wakes on the pressure difference between PS and SS of a rotor blade ($s^*=4.3$ percent), $r^*=92$ percent, 10 percent chord, design speed, design point

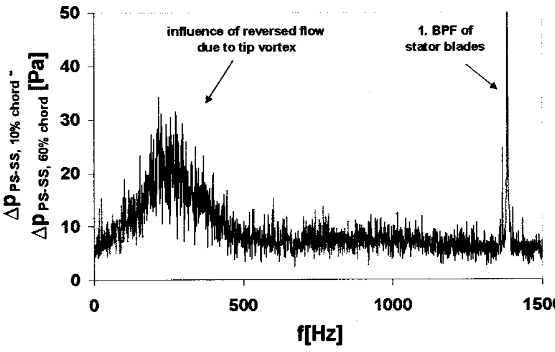

Fig. 22 Fluctuation of tip vortex along the blade chord, frequency spectrum of the difference of the pressure differences between PS and SS near the leading edge (10 percent chord) and the rear part (60 percent chord) of a rotor blade, $r^*=92$ percent), $s^*=4.3$ percent, design speed, operating point near stability limit ($\xi=0.83$)

The vortex intensity in the front part of the blade strongly fluctuates. As a result of this the location of the strongest vortex intensity periodically moves along the blade chord. This can clearly be seen in the time trace of the difference of the pressure differences between PS and SS near the leading edge (10 percent chord) and in the rear part (60 percent chord) of the blade. A frequency spectrum of this time trace is shown in Fig. 22. It shows the periodical fluctuation of the maximum pressure difference between PS and SS along the blade chord, caused by the periodic characteristic of the tip clearance flow. This can be seen as a hump in the spectrum, which agrees very well with the typical pattern of RIs shown in Fig. 5. This periodical fluctuation along the blade chord is also according to the results for the propagation of RIs in the blade tip region, Fig. 9. The discrete peak at 1383 Hz in Fig. 22 shows ones more the periodical influence caused by the stator wakes.

This fluctuation of the blade tip vortex presumably propagates into circumferential direction [17]. At the time t_1 a blade tip vortex with strong intensity near the leading edge occurs at blade 1, Fig. 23. The large reversed flow region of this vortex affects the flow in the front region of the neighboring blade 2 at the time t_2. At this time the pressure difference in this region of blade 2 is diminished and the maximum pressure difference is shifted downstream to the rear part of the blade. However, the pressure difference in this region is considerably smaller than at the leading edge without the influence of the reversed flow. This means the blade tip vortex at time t_2 at blade 2 is weaker than that at time t_1 at blade 1 and can be localized to the rear part of the blade. The vortex presumably leaves the passage and does not affect the flow at time t_3 at blade 3, which again produces a strong tip vortex, etc.

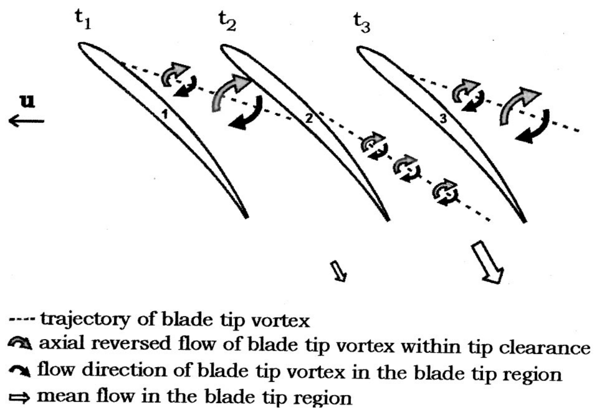

Fig. 23 Time-dependent development of blade tip vortex, rotating system

Thus the fluctuating blade tip vortices propagate in circumferential direction against the rotor turning direction along the rotor blade row (referred to the relative frame). At the same time a

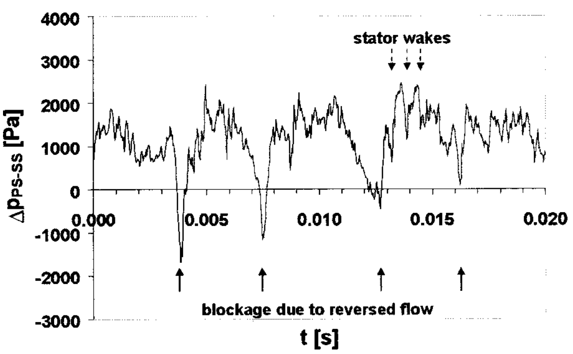

Fig. 21 Pressure difference between PS and SS of a rotor blade, $s^*=4.3$ percent, $r^*=92$ percent, design speed, operating point near stability limit ($\xi=0.83$), sensors at nearly the same axial position: 20 percent chord at PS, 30 percent chord at SS

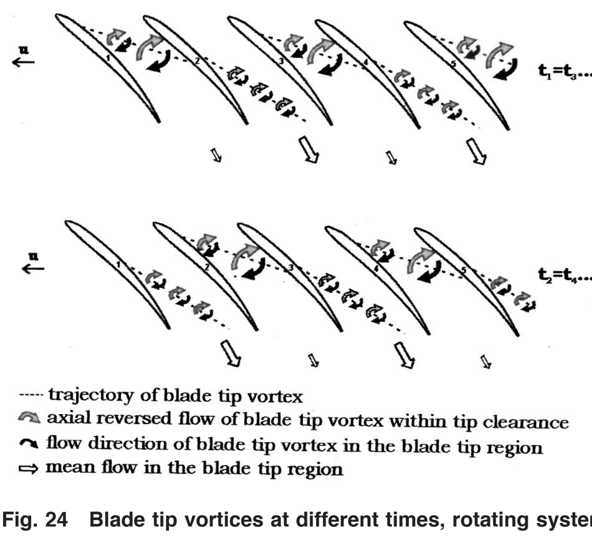

Fig. 24 Blade tip vortices at different times, rotating system

strong blade tip vortex can be found at every second blade, Fig. 24. At this time the other blades can only develop a weak tip vortex. Thus the mode order of the disturbance is about half the rotor blade number. This rotating structure in the blade tip region is the rotating instability (RIs) which was to be investigated.

The propagation direction of the periodically reversed flow due to the strong tip vortices fluctuates, Mailach [17]. This seems to be the reason for the fluctuation of the circumferential wavelength of the disturbance respectively the excitation of different mode orders. The time-dependent fluctuations are averaged in the spectrum. This is the reason why RIs are spread over a certain frequency range in a spectrum.

At design point and large tip clearance RIs have not been found because only the rear part of the blade is affected by the tip clearance flow of the neighboring blade. Stronger fluctuations at measuring positions on the PS near the trailing edge and the blade tip give evidence for that. Because the pressure difference between PS and SS near the blade tip is relatively constant along the chord the effect on the formation of the tip vortex seems to be small and does not result in a rotating structure.

5 Conclusions

Rotating instabilities have been observed near the stability limit of the compressor provided a relatively large rotor blade tip clearance. This phenomenon can be responsible for the intensification of clearance noise and the excitation of large amplitude blade vibrations.

Measurements both in stationary and rotating system reveal that this phenomenon is limited to the blade tip region within a rotor blade row and the axial gaps upstream and downstream of it. The largest pressure fluctuations of the disturbance could be observed at measuring positions next to the blade tip within the area of the largest blade thickness, whereby the disturbance on the PS is more pronounced than on the SS. The circumferential structure and the spatial development of this unsteady flow phenomenon has been described. RIs comprise a group of modes of much higher order than those of rotating stall. The dominating mode order of RIs is nearly half the rotor blade number. The circumferential velocity is 50–60 percent of rotor velocity.

The results of LDA measurements within the rotor blade passages and the tip clearance show the structure of the blade tip vortex. For a large tip clearance at operating points with RIs a strong blade tip vortex with a reversed flow region within the tip clearance and a blockage in the tip region of the rotor blades has been observed.

Unsteady pressure measurements in the blade tip region of the rotor blades reveal a strong fluctuation of the blade tip vortices. It is caused by periodical interactions of the tip clearance flow of one blade and the flow at the adjacent blade. The fluctuating blade tip vortices propagate in circumferential direction along the rotor blade row and yield a rotating structure with high mode orders, which is the investigated rotating instability.

Acknowledgments

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Nomenclature

f = frequency
p = pressure
r^* = dimensionless passage height
s^* = dimensionless tip clearance
t = time
u = rotor velocity
w = relative velocity
α = mode order
β = relative flow angle (from circumferential direction)
ϕ = circumferential position
π = total pressure ratio
ξ = mass flow/design mass flow

Abbreviations

ax = axial
BPF = blade passing frequency
C1/C2 = measuring configurations
LDA = laser doppler anemometry
LSRC = Low Speed Research Compressor
PS = pressure side
SS = suction side
RI = rotating instabilities

References

[1] Camp, T. R. and Day, I. J., 1998, ''A Study of Spike and Modal Stall Phenomena in a Low-Speed Axial Compressor,'' ASME J. Turbomach., **120**, pp. 393–401.
[2] Mathioudakis, K., and Breugelmans, F. A. E., 1985, ''Development of Small Rotating Stall in a Single Stage Axial Compressor,'' ASME paper 85-GT-227.
[3] Mongeau, L., 1991, ''Experimental Study of the Mechanism of Sound Generation by Rotating Stall in Centrifugal Turbomachines,'' Ph.D. dissertation, Pennsylvania State University.
[4] Bent, P. H., McLaughlin, D. K., and Thompson, D. E., 1992, ''The Influence of Discharge Configuration on the Generation of Broadband Noise in Centrifugal Turbomachinery,'' 14th Aeroacoustics Conference, May 11–14, 1992, Aachen, Germany, DGLR/AIAA 92-02-099.
[5] Mongeau, L., and Quinlan, D. A., 1992, ''An Experimental Study of Broadband Noise Sources in Small Axial Fans,'' International Symposium on Fan Noise INCE, Senlis, France, Sept. 1–3, 1992.
[6] Kameier, F., 1994, ''Experimentelle Untersuchung zur Entstehung und Minderung des Blattspitzen-Wirbellárms axialer Strömungsmaschinen,'' Fortschritt-Berichte VDI Reihe 7 Nr. 243 Düsseldorf, Germany.
[7] Krane, M. H., Bent, B. H., and Quinlan, D. A., 1995, ''Rotating Instability Waves as a Noise Source in a Ducted Axial Fan,'' ASME Winter Annual Meeting, Turbomachinery Noise Symposium, San Francisco.
[8] Kameier, F., and Neise, W., 1997, ''Experimental Study of Tip Clearance Losses and Noise Generation in Axial Turbomachines and Their Reduction,'' ASME J. Turbomach., **119**, pp. 460–471.
[9] März, J., Neuhaus, L., Neise, W., and Gui, X., 1998, ''Circumferential Structure of Rotating Instability under Variation of Flow Rate and Solidity,'' VDI-Tagung: Turbokompressoren im industriellen Einsatz, Oct. 6–7, Hannover, Germany.
[10] Baumgartner, M., Kameier, F., and Hourmouziadis, J., 1995, ''Non-Engine Order Blade Vibration in a High Pressure Compressor,'' 12th International Symposium on Airbreathing Engines, Sept. 10–15, Melbourne, Australia.
[11] Witte, H., and Ziegenhagen, S., 1998, ''Beurteilung von strömungserregten Schaufelschwingungen eines Flugtriebwerks-Axialverdichters mittels statistischer Analysemethoden,'' VDI-Tagung: Turbokompressoren im industriellen Einsatz, Oct. 6–7, Hannover, Germany.
[12] Hoying, D. A., Tan, C. S., Huu Duc Vo, and Greitzer, E. M., 1998, ''Role of Blade Passage Flow Structures in Axial Compressor Rotating Stall Inception,'' ASME paper 98-GT-588.
[13] Inoue, M., Kuroumaru, M., Tanino, T., and Furukawa, M., 1999, ''Propagation of Multiple Short Length-Scale Stall Cells in an Axial Compressor Rotor.'' ASME paper 99-GT-97.
[14] Boos, P., Möckel, H., Henne, J. M., and Selmeier, R., 1996, ''Flow Measurement in a Multistage Large Scale Low Speed Axial Flow Research Compressor,'' Proceedings of the 43rd Gasturbine & Aeroengine Technical Congress, Exposition and Users Symposium, June 2–5, Stockholm, Sweden.
[15] Müller, R., Mailach, R., Lehmann, I., and Sauer, H., 1999, ''Flow Phenomena Inside the Rotor Blade Passages of Axial Compressors,'' AIAA99-IS-084, 14th International Symposium on Airbreathing Engines, Sept. 6–10, Florence, Italy.
[16] Mailach, R., 1999, ''Experimental Investigation of Rotating Instabilities in a Low-Speed Research Compressor,'' 3rd European Conference on Turbomachinery—Fluid Dynamics and Thermodynamics, March 2–5, London, GB.
[17] Mailach, R., 1999, ''Früherkennung rotierender Instabilitäten,'' Abschlußbericht zum BMBF-Vorhaben 0327041L, July 1999, Dresden, Germany.
[18] Inoue, M., Kuroumaru, M., 1989, ''The Structure of Tip Clearance Flow in an Isolated Axial Compressor Rotor,'' ASME J. Turbomach., **111**, pp. 250–256.
[19] Saathoff, H., and Stark, U., 1998, ''Endwall Boundary Layer Separation in a High-Stagger Compressor Cascade and a Single-Stage Axial-Flow Low-Speed Compressor,'' 11. DGLR-Fach-Symposium, Strömungen mit Ablösung, Nov. 10–12, Berlin, Germany.

Discussion: ''Rotating Instabilities in an Axial Compressor Originating From the Fluctuating Blade Tip Vortex'' (Mailach, R., Lehmann, I., and Vogeler, K., 2001, ASME J. Turbomach., 123, No. 3, pp. 453–460)

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It is very welcome to see papers appearing with results obtained on the superb compressor facility in the Technical University of Dresden. One feels confident that these have been obtained to the highest standards in a machine that models the flow in high-performance compressors.

I am less happy about some aspects of the paper. I believe that the phenomenon that is the basis of this paper, which the authors call ''rotating instability,'' is part-span rotating stall. This phenomenon is very common in high-speed multistage axial compressors in which the front stages quite normally go into a multicell, part-scan rotating stall when the machine is operating at speeds below the design value. This phenomenon was described in some detail in the famous NACA report ''The Aerodynamic Design of Axial Flow Compressors,'' first published in 1956 [1]. The present authors are quite mistaken to say that this unsteady flow phenomenon was first described by Mathiodakis and Breugelmans in 1985 [2]; indeed these authors referred to the phenomena they saw as rotating stall, either small stall or big stall.

The particular flow that the authors have investigated is the mismatched stage in a multistage compressor. It seems probable to me that over a narrow range of flow coefficients their third stage stalls, since the tip clearance for this stage is larger than the others, but the other three stages maintain the whole machine stable. What results is a part-span rotating stall in the third stage. A very similar flow was investigated by Silkowski [3], who increased the tip clearance of the first stage, so for his tests it was the first stage that was entered into part-span rotating stall. There are particular reasons why the stall of a single stage in a multistage compressor is likely to be both multicell and part span. By having a relatively large number of stall cells, there can be rapid decay in the axial direction, which is necessary since the adjoining stages are unstalled. The part-span nature of the stall allows some axial flow through the blades that are stalled at the tip so that the stages upstream and downstream can be unstalled at the same circumferential position. This makes possible the close proximity of stalled and unstalled stages at the same circumferential position. To be more definite about what was going on when operating in the condition the authors refer to as ''rotating instability,'' one needs to see the raw data (i.e., signals showing, as a function of time, the wall static pressure signals, or the upstream and downstream velocities). The processing to give Figs. 5, 6, and 7 in the paper necessarily gets rid of important information that might clarify what the phenomenon.

The authors make a number of definite statements and conclusions regarding the role of the tip vortex and its part in the ''rotating instability.'' I would prefer that these were seen as highly tentative hypotheses for the behavior, since they are based on inference rather than direct observation. It is well known that increasing rotor tip clearance increases the flow across the tip and (beyond a very small minimum clearance) reduces pressure rise and efficiency. It is less clear that the clearance flow forms a well-defined vortex; what appears to matter most is the formation of flow blockage, with the consequent reduction in static pressure rise. The existence of flow close to the wall flowing in the reverse direction over the rotor tips is a real effect and the first published example that I know was by Koch [4].

One of the most confusing features of the paper, including its title, is the term ''rotating instability,'' which appears to have been coined relatively recently by Kameier [5]. I am confident that what has been investigated is fully developed (i.e., steady) rotating stall with multiple part-span cells. One of the features of rotating stall, whether it is part span or full span, is its stability; indeed sometimes it can be hard to get a compressor out of rotating stall. A compressor operating in stall can be stable, but on opening the throttle the compressor will become eventually become unstable in the stalled condition and it will then change to become unstalled. The use of the words ''rotating instability'' to denote ''rotating stall'' is unfortunate and its use can only lead to confusion where none is necessary. (It may also be added that the term itself is without meaning; it is applied to a nominally steady process, yet the word ''instability'' implies that it is in the process of change from one state to another.)

The paper refers back to the original work by Kameier, which was performed on a single stage fan. Single-stage compressors have a tendency to go into part-span, multicell stall; see Day [6]. At design speed the well-matched multistage machine tends to go into full-span stall with a very small number of stall cells, most often only a single cell, extending the whole length of the compressor. Only when the multistage machine is mismatched (either because its speed is below design or because of geometric changes) does a multicell, part-span stall occur.

References

[1] NACA, 1956, ''The Aerodynamic Design of Axial Flow Compressors,'' NACA TN 3662.
[2] Mathioudakis and Breugelmans, 1985.
[3] Silkowski, 1995.
[4] Koch, C. C., 1975.
[5] Kameier, 1994.
[6] Day, I. J., 1976, ''Axial Compressor Stall,'' PhD Dissertation, University of Cambridge, United Kingdom.

Closure to ''Discussion of 'Rotating Instabilities in an Axial Compressor Originating from the Fluctuating Blade Tip Vortex' (2000-GT-506)'' (2001, ASME J. Turbomach., 123, p. 461)

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The authors thank Prof. Cumpsty for his critical comments. After the discussion in Munich (ASME Turbo Expo 2000), it is disappointing that a major part of this comment still focuses on the chosen term ''rotating instability'' (RI). It is understood that a common term for a certain effect is needed in order to ensure efficient communication. This is the reason the authors used the questioned term, as it was already in use (''coined'') by several investigators to name the observed characteristic flow unsteadiness.

The authors agree that there might be better names for this effect. They do not agree that the term ''rotating instability'' is

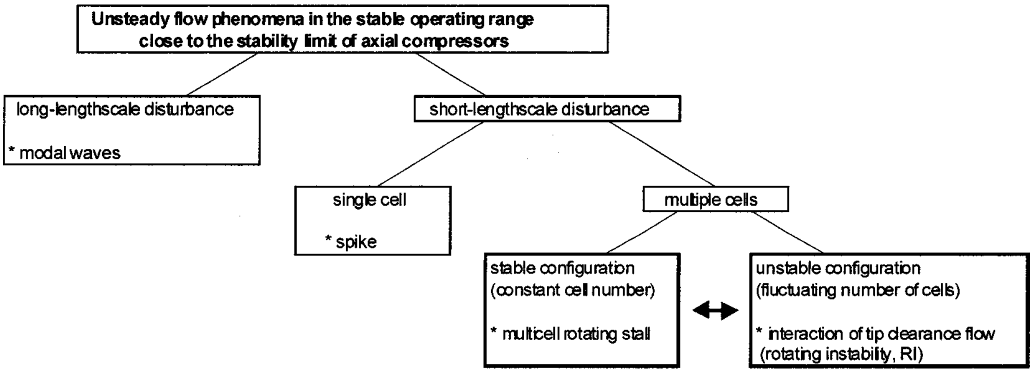

Fig. 1 Overview of the main unsteady flow phenomena in the stable operating range close to the stability limit of axial compressors

totally wrong. They are still convinced that the physics of the observed flow unsteadiness is different from the typical part-span rotating stall caused by overloaded blades.

Figure 1 shows the classification of different flow effects that can be observed close to the stability limit of a compressor. In addition to modal waves and spikes, multicell disturbances can appear. It is well known that part-span multicell disturbances occur in the front stages of multistage compressors at speeds below the design speed while the compressor works well. In multistage low-speed compressors, this normally does not take place as all stages are loaded nearly equally. In this case, multicell disturbances can be found only in mismatched stages. This could be a single mis-staggered blade row or a single blade row with larger tip clearance. Increasing the tip clearance of one rotor blade row was our way to excite a part-span multicell disturbance in the four-stage low-speed research compressor of Dresden (Dresden LSRC). The reason for this was the assumption that the observed unsteady characteristic of the RI has its driving force in the physics of the tip vortex.

Nevertheless, the authors propose to distinguish between stable and unstable multicell configurations. Although this classification cannot be found in the literature, it is thought to be necessary because of the different behavior of the cells and presumably different origination mechanisms of the disturbances.

A stable configuration is characterized by a constant number and a nearly constant size of the rotating cells. This stable multicell configuration is commonly known as multicell rotating stall. It was reported by Day et al. [1] and Inoue et al. [2]. In a pressure or velocity signal versus time, one can observe nearly equal time intervals between consecutive passing cells (e.g., Day et al. [1], Fig. 14). In the frequency spectrum of this signal the disturbance is reflected by a discrete peak.

The multicell configurations are truly unstable if their numbers and size of cells are strongly fluctuating. The phenomenon observed in the Dresden LSRC has this unstable characteristic. The term ''rotating instability'' was taken from an ongoing discussion in the literature to avoid confusion with stable unsteady stall phenomena. To the knowledge of the authors, the first description of this unstable, unsteady behavior of a multicell disturbance is reported by Mathioudakis and Breugelmans [3] in a single-stage compressor. However, at that time they used another term for it (''small rotating stall''). Unstable behavior in multistage compressors is also reported by Longley and Hynes [4], who used the name ''rough running'' and Baumgartner et al. [5], who termed it ''rotating instability.''

Professor Cumpsty claims that information is lost due to the postprocessing used. The authors do not believe that this is the case. Figure 2 shows raw data from a sensor positioned close to the tip. Very strong nonperiodic fluctuations of the pressure signals due to RI occur. Some periodicity can be observed but is not dominating. This is where the observed flow shows its unstable behavior.

Much additional information can be obtained when the analysis of the raw signals is coupled with filtering of the time-dependent signals and statistical evaluation methods like autopower and cross-power spectra. This allows us to isolate the information on the disturbance in which we are interested from the influence of wake passages and from other disturbances like modal waves.

The time-dependent fluctuations of the amplitudes, frequency,

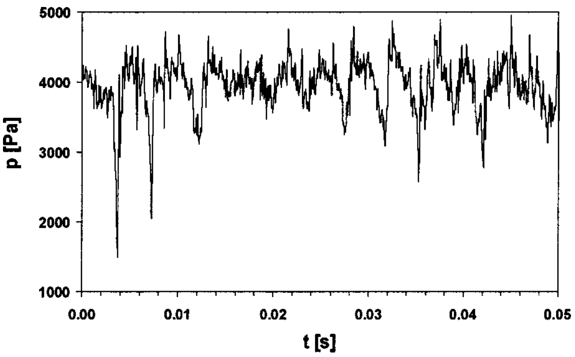

Fig. 2 Signal of a pressure sensor on a rotor blade, sensor near blade tip (92 percent of channel height) and leading edge (10 percent of chord length), operating point near stability limit (design speed, ξ=0.82), tip clearance ratio s^*=4.3 percent

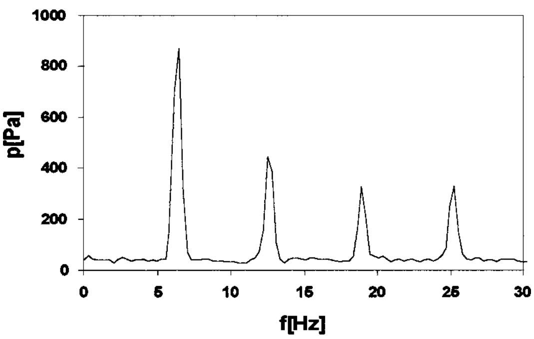

Fig. 3 Frequency spectrum with typical sign of a single-cell full-span rotating stall in Dresden LSRC (measured at the casing)

and wavelength of the disturbance can clearly be seen in Fig. 6 of the paper. These signals are bandpass-filtered to get rid of the influence of modal waves and passing wakes.

The time-averaged results of the statistical evaluation methods confirm the unstable behavior. It can be seen that many frequencies as well as mode orders (respectively cell numbers) of the disturbance are present (Figs. 5 and 7 in the paper). This typical behavior was also shown by other authors (e.g., Kameier [6]). Contrary to that a stable single-cell or multicell rotating stall, with its stable number of cells and constant circumferential velocity, would be recognized by a dominating discrete peak and possibly its harmonics in the frequency spectrum (Fig. 3).

From Fig. 7 of the discussed paper it follows, that the wavelength, that is, the distance between adjacent cells, varies from 1.5 to 5.0 pitches of the rotor blades (63 rotor blades). (In the model presented in Fig. 24 in the paper only the dominating wavelength of 2.0 pitches is considered.) Hence the disturbance is far from being stable. It is not a frozen rotating pattern but changes permanently. Because of this, the term ''rotating instability'' is absolutely correct for this disturbance. However, Prof. Cumpsty is correct in this point, the compressor unit is running stable.

Additional remark: The strong fluctuations of the disturbance, which were investigated in the third stage of the Dresden LSRC, are not caused by the passing blade wakes and the high turbulence of the flow. This is shown in the recent investigations of Mailach et al. [7] in a plane cascade. Even in a plane cascade, this disturbance propagates along the tip region of the blade row. In spite of the low turbulence intensity of the incoming flow and without passing wakes, the wavelength of the short-length-scale disturbance is strongly fluctuating. Of course we cannot talk about a ''rotating'' instability in the plane cascade anymore. But the mechanism of the disturbance, which propagates along a blade row, is the same.

It is the tip clearance flow that is responsible for the characteristic of this kind of disturbance. There is no doubt that it results in a strong tip vortex. This can clearly be seen from the results of LDA measurements for the design tip clearance ratio of 1.3 percent (Figs. 12 and 13 in the paper). As shown in the discussed paper, for larger tip clearance ratios, the tip vortex fluctuates strongly. Due to the ensemble-averaging, its structure is less pronounced in the results of the LDA measurements (Figs. 15–18 in the paper).

Both for the stable multicell configuration (Inoue et al. [2]) and—as shown in this paper—for the unstable multicell configuration, the tip clearance flow is responsible for the resulting disturbance. As shown in this paper, the axial reversed flow only occurs within the tip clearance and causes a reduction or a blockage of the incoming flow in the blade tip region. It is the opinion of the authors that this is definitely not a classical stall, even if we observe reversed flow within the tip clearance with influence into the outer blade passage.

It is agreed by the authors that the term ''rotating instabilities'' could be substituted by a more precise one. We did agree on that already in Munich. But considering the above said, it is definitely not wrong. However, in order to avoid confusion with stalled blades due to separated flow on the profiles, one can discuss whether the expression ''stall'' should be used in the eventually improved term at all. In the opinion of the authors a term like ''propagating tip vortex instability'' (''rotating tip vortex instability,'' ''interaction of tip clearance flow'' . . .) would be more suitable, since it applies more specifically to the physical mechanism of the disturbance, described in the discussed paper.

References

[1] Day, I. J., Breuer, T., Escuret, J., Cherett, M., and Wilson, A., 1999, ''Stall Inception and the Prospects for Active Control in Four High-Speed Compressors,'' ASME J. Turbomach., **121**, pp. 18–27.

[2] Inoue, M., Kuroumaru, M., Tanino, T., and Furukawa, M., 2000, ''Propagation of Multiple Short Length-Scale Stall Cells in an Axial Compressor Rotor,'' ASME J. Turbomach., **122**, pp. 45–53.

[3] Mathioudakis, K., and Breugelmans, F. A. E., 1985, ''Development of Small Rotating Stall in a Single Stage Axial Compressor,'' ASME Paper No. 85-GT-227.

[4] Longley, J. P., and Hynes, T. P., 1990, ''Stability of Flow Through Multistage Axial Compressors,'' ASME J. Turbomach., **112**, pp. 126–132.

[5] Baumgartner, M., Kameier, F., and Hourmouziadis, J., 1995, ''Non-Engine Order Blade Vibration in a High Pressure Compressor,'' Proc. 12th International Symposium on Airbreathing Engines, Sept. 10–15, Melbourne, Australia.

[6] Kameier, F., 1994, ''Experimentelle Untersuchung zur Entstehung und Minderung des Blattspitzen-Wirbellärms axialer Strömungsmaschinen,'' Fortschritt-Berichte VDI Reihe 7 No. 243, Düsseldorf, Germany.

[7] Mailach, R., Sauer, H., and Vogeler, K., 2001, ''The Periodical Interaction of the Tip Clearance Flow in the Blade Rows of Axial Compressors,'' ASME Paper No. 2001-GT-0299.