Effects of Stator Wakes and Spanwise Nonuniform Inlet Conditions on the Rotor Flow of an Axial Turbine Stage

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Abstract

Detailed measurements have been performed in a subsonic, axial-flow turbine stage to investigate the structure of the secondary flow field and the loss generation. The data includes the static pressure distribution on the rotor blade passage surfaces and radial-circumferential measurements of the rotor exit flow field using three-dimensional hot-wire and pneumatic probes. The flow field at the rotor outlet is derived from unsteady hot-wire measurements with high temporal and spatial resolution.

The formation of the tip clearance vortex and the passage vortices is presented, which are strongly influenced by the spanwise non-uniform stator outlet flow. Taking the experimental values for the unsteady flow velocities and turbulence properties, the effect of the periodic stator wakes on the rotor flow is discussed.

Nomenclature

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<thead>
<tr>
<th>Symbol</th>
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<tr>
<td>b</td>
<td>axial chord</td>
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<td>c</td>
<td>velocity, absolute frame</td>
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<td>static pressure</td>
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<td>random axial velocity fluctuation</td>
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<td>random radial velocity fluctuation</td>
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<td>w</td>
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<td>w'</td>
<td>random circumferential velocity fluctuation</td>
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<td>Z</td>
<td>blade number</td>
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<td>a</td>
<td>flow angle in circumferential direction, absolute frame</td>
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<td>b</td>
<td>flow angle in circumferential direction, relative frame</td>
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<td>d</td>
<td>tip clearance</td>
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<td>t</td>
<td>reduced frequency</td>
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Subscripts

r radial direction
x axial direction
θ circumferential direction
0 stator inlet plane
1 rotor inlet plane
2 rotor exit plane

Superscripts

ω periodic value
- averaged value

Introduction

The efficiency of modern turbine stages is strongly affected by the generation of secondary flows and the development of the profile boundary layers. The first models for cascade flows were presented by Hawthorne (1955) and Klein (1966) and later by Langston et al. (1977) and Marchaland, Sieverding (1977). The turbulence and loss generation in cascades were investigated in detail by Gregory-Smith et al. (1988) and Zunino et al. (1987). Based on these and many other investigations, great progress on the development of numerical methods for aerodynamic calculations has been made, so that the prediction of the steady flow through an isolated blade row is possible with impressive accuracy.

However, the flow in a real turbomachine is unsteady as a result of the relative motion of the blade rows. Potential field interactions between stationary and rotating airfoils and unsteadiness due to the cutting of wakes and secondary vortices shed from upstream blade rows have a profound effect upon the turbine performance. This is amplified by small axial gaps in modern turbomachines. The stator-rotor interaction is known to affect aerodynamic efficiency, heat transfer, structural loading and noise generation of turbine stages. Improvements of high-response measurement techniques provided means for the experimental investigation of the unsteady rotor flow.
Joslyn et al. (1983) demonstrated that the rotor outlet flow at midspan of a turbine changes markedly as the rotor interacts periodically with the stator wakes. The three-dimensional flow through a large scale turbine was measured by Hunter (1982), Sharma et al. (1985) and Joslyn and Dring (1990). They detected considerable variations of flow angles, velocity and pressure distribution for different rotor-stator positions.

Besides the potential interaction the cutting and transport of wakes through a downstream blade row is a major source for the unsteadiness. Meyer (1958) presented an early model of the wake convection, Fig. 1, representing the wake as a negative jet in the downstream blade row. Kerrebrock and Nikolajczak (1970) used the model to explain the temperature redistribution by compressor rotor wakes passing through a stator. Very detailed measurements of Adachi and Murakami (1979) in a cascade with an upstream wake generator confirmed the model further. Additionally, the incoming stator wakes have a strong impact on the rotor profile boundary layers. Hodson (1983) reported a 50 percent increase of the profile losses compared to a cascade with steady inflow. This is explained by the early transition of the suction side boundary layer, which is likely to occur when the turbulent wake impinges on the blade surface. Unsteady transition and the oscillation of the boundary layer between turbulent and laminar state was also found by Evans (1978) for a compressor and by Dring et al. (1982) for an axial turbine rotor. Many other investigators studied the unsteady wake boundary layer interaction with respect to heat transfer, but this is beyond the scope of this paper.

Another effect causing unsteadiness in the rotor is the cutting of stator secondary vortices. This was examined by Binder (1985a), who measured a sudden increase of turbulence energy when the rotor intercepted the passage vortices of the upstream stator row. By using the laser-technique he was able to detect the stator wakes by increased turbulence and to show the convection of the wakes through the rotor passage (Binder (1985a)).

The objective of the present study is to investigate the flow in a turbine rotor as it is influenced by an upstream stator, including the effects spanwise nonuniform inlet conditions have on the rotor secondary flow and periodically unsteady stator wakes have on the rotor exit velocity and turbulence distribution. The results are of interest for designers using steady-state solutions to estimate the significance of the unsteady effects. On the other hand, the data can be used for a comparison with advanced numerical flow calculations, like the three-dimensional Navier-Stokes solvers presented by Dawes (1990) and Rao and Delaney (1990), which are capable of performing full stage calculations by coupling the grids of stator vanes and moving rotor blades.

Experimental Facility and Instrumentation

The experimental work was carried out in a single-stage axial turbine with untwisted blades (Fig. 2). In the stator, the Traupel profile described by Utz (1972) was used, the rotor consists of modified VKI-profiles. The application of hot-wire probes limited the investigations to subsonic flows. Though some results may be specific to this turbine, the data can improve the understanding of the more complex flow in modern, multi-stage turbines. A cross section of the stage with midspan velocity triangles is shown in Fig. 2.
A turbocompressor set provided a continuous airflow to the test rig. The total temperature at turbine inlet was set to 308 K ± 0.5 by cooling the air at the compressor outlet. The total pressure could be adjusted by bypassing part of the compressor mass-flow. With a shaft speed variation of less than ± 0.2 percent during the run, the tests could be repeated with a flow coefficient variation of less than ± 0.7 percent. The Reynolds numbers were constant with an accuracy of ± 1.0 percent.

The flow in the rotor exit plane was surveyed with pneumatic five-hole probes and three-or hot-wire probes, with the probes mounted in the absolute frame. Close to the endwalls, X-hot-wire and pneumatic boundary layer probes were used. The probes were traversed from hub to tip and circumferentially over two blade pitches. In the stator exit plane, only pneumatic probes were located 18 % and 12 % axial chord downstream of stator and rotor, respectively. The measurement of the static pressure distribution on the rotor blades was accomplished with a rotating scanivalve. Using 10 pressure taps on the suction side and 6 on the pressure side, the distributions were obtained at five radial locations. The measurements were corrected to take into account centrifugal effects on the column of air in the rotating piping, and thus could be repeated with an accuracy of ± 2 percent. In order to obtain the unsteady static pressure field at the casing above the rotor, miniature high response pressure transducers were mounted at 25 axial locations in the casing. To improve the accuracy of the measurements, only the unsteady pressure amplitude of the transducers was recorded and superimposed on the time-mean static pressure measured by steady-state instrumentation. The data acquisition and reduction scheme was essentially the same as described in the following section for the hot-wire probes.

All experimental data is normalized with respect to the International Standard Atmosphere at stator inlet and refers to the design operation point of the turbine.

**Real time data acquisition and reduction**

Since the hot-wire probes are a key feature of this investigation, the calibration and data acquisition is described shortly. A more detailed description was presented by Poensgen and Galus (1990).

The unsteady voltages from the hot-wire bridges were logged with a high-speed (up to 10 MHz) multi-channel data acquisition system. In these experiments, 256 real-time samples were recorded over roughly 4 blade passing periods and averaged over 256 revolutions. For each of the 65000 data points, the velocity vector was calculated based on a polynomial least square fit method. Input for this were the results from a free-jet calibration of the probes, where an accuracy in velocity of 0.5 m/s and in pitch and yaw angle of less than 0.2° was achieved. Phase-locked-averaging (Gostelow (1977)) has been used for the data reduction. The results of the velocity measurements, for example, are presented in the form of the ensemble mean values,

\[ \bar{c}(i) = \frac{1}{N} \sum_{n=1}^{N} c(i,n) \]  

and the ensemble root mean square of the random fluctuations,

\[ \sqrt{\bar{c}^2(i)} = \sqrt{\frac{1}{N} \sum_{n=1}^{N} [c(i,n) - \bar{c}(i)]^2} \]

where \( N \) denotes the number of revolutions and \( i \) the index for the circumferential direction.

Using a square wave signal, the corner frequency of the hot-wires was found to be greater than 20 MHz, which is a conservative estimation. At 50 KHz, the ratio between signal amplitude and probe response is still greater than 0.8, while the blade passing frequency is only 2.4 KHz. The recorded real-time signals were filtered by a digital low-pass Blackman filter to eliminate high frequency noise. The Channon factor, which describes the relation between cut-off frequency of the filter process and the sampling frequency is about 0.3.

During the measurements, the probes having a sensitive cone of ± 25° yaw and pitch angle, were always aligned to the mean flow direction. It was found, however, that at some measurement stations the pitch angle fluctuation exceeded the acceptance cone when a rotor wake passed the probe.

Therefore, it was necessary to record additional data after rotating the probe. Data was only accepted if the ensemble mean pitch angle
was smaller than 15°, thus allowing for turbulent fluctuations, and it had to be derived from two measurements with different probe orientations. Fig. 3 compares the circumferentially averaged flow velocity at rotor outlet as detected by hot-wire and pneumatic probes. At locations (2) and (4), where the pitch angle variation is relatively small, good agreement is reached, whereas at locations (1) and (3) with a pitch angle variation of more than 40° during the passing wake, the differences between steady and unsteady measurements amount to more than 10 percent. Based on this, the authors felt that the hot-wire technique yields reliable results, confirmed by the good agreement with the five-hole probe and that the steady-state pneumatic probe readings were not reliable in regions of large unsteady fluctuations.

Experimental Results

Stator Flow. Flow measurements downstream of the stator are presented briefly to document the rotor inlet conditions. The circumferentially mass-averaged exit velocity measured at 20 percent of axial chord downstream of the stator, Fig. 4, indicates the higher acceleration at the hub. The reason for this is the high exit swirl and the resulting higher static pressure at the tip. The overall averaged outlet Mach number $M_{\infty}$ is 0.45. The plot of the exit flow angle, Fig. 4, exhibits a pattern familiar in secondary flow studies, with overturning at the endwalls and the corresponding underturning at a short distance away from the endwalls. In spite of the lower acceleration, the overturning is stronger at the tip. This is explained by the thicker inlet boundary layer, with an estimated 99 percent thickness of 11.8 and 5.5 percent span at tip and hub, respectively. At both end-walls, the boundary layers are natural turbulent. The secondary flow field, Fig. 5, demonstrates the circumferential extension of the passage vortices. At the tip, the overturning occurs over the whole cross-section, whereas at the hub only a smaller passage vortex is observed. In the wake area, the imbalance between centrifugal and pressure forces causes inward directed radial components. In the total pressure loss contour plot, Fig. 6, significant losses are found only in the wake of the profile and in the corner between suction side and tip. Two loss cores can be seen at about 15 and 85 percent span. At these locations, a third vortex at the hub and a somewhat distorted flow region at the tip are observed in Fig. 5. Flow visualization at the passage surfaces with a Ti,0-oil mixture and the results from other authors as for example Gregory-Smith et al. (1988) indicate that, forced by the passage vortices, low momentum fluid from the inlet endwall boundary layers is transported towards the suction side and accumulated in the loss cores. Again, the higher losses and the larger extension of the wake at the tip is due to the thicker inlet boundary layer.

In Fig. 7, the time-averaged rotor inlet flow in the relative frame obtained from the absolute values and a vector addition of the circumferential speed is presented. The incidence angle increases towards the hub because of the higher stator exit velocity and the lower circumferential speed. At the radial locations of 15 and 85 percent span, a local turning of the inlet flow vector of about 10 degrees towards the suction side occurs, which is caused by stator underturning, Fig. 4. The velocity increases nearly linear from tip to hub, with almost no detectable reduction at the hub, whereas the higher losses at the tip lead to a reduced velocity between casing and 90 percent span.

The shaded areas in Fig. 7 mark the velocity and incidence fluctuations as the rotor passes the stator wakes. At midspan, the incidence is reduced by 8.5 degrees and the velocity by 10 percent. The
Fig. 6: Total pressure loss contours at stator exit

highest incidence fluctuations occur at about 85 and 15 percent span. When the rotor blade passes the loss cores in the stator wake, the incidence at these locations decreases 22 degrees and 15 degrees, respectively. At the tip, the incidence fluctuations are generally high, caused by the large extension of the wake. The velocity fluctuations are nearly of the same magnitude across the whole span, showing higher fluctuations only at the locations of the loss cores. The steady-state measurements in the stator exit flow could not detect the potential flow field interaction. However, the upstream influence of the rotor potential field on the steady-state stator flow is only weak, as stated by Dring et al. (1982) and Dunn et al. (1990), who varied the rotor-stator axial spacing between 10 to 65 percent of blade pitch. They also found that the fluctuations in the rotor were of much larger amplitude than could be explained by the stator potential flow field alone and that the decay with increased axial gap was similar to the wake decay. Therefore, the authors believe, that the incidence of the stator wakes on the rotor flow is considerably stronger than that of the potential flow field of the thin trailing part of the stator vanes.

Rotor Flow. The time-mean static pressure distribution on the rotor blades was measured at five radial locations, Fig. 8. With increasing loading towards the hub, see Fig. 7, the stagnation point moves to the pressure side. From tip to midspan, the suction side flow accelerates smoothly up to the throat at about 50 percent axial chord and decelerates towards the trailing edge. At lower radii, the high positive incidence angle causes a strong acceleration around the leading edge and merely a weak deceleration over nearly the whole suction side. At the pressure side, the flow is only slightly affected by the spanwise varying incidence. The static pressure is nearly constant up to 40 percent axial chord, followed by an acceleration in the last part of the passage. Fig. 9 shows the time-mean static pressure distribution at the casing for a certain rotor position. As expected for an inlet flow with nearly zero incidence, see Fig. 7, the stagnation point at design point is located at the leading edge. If the direction of the local velocity is assumed to be perpendicular to the lines of constant static pressure, Fig. 9 indicates that the influence of the stator wakes on the rotor flow is considerably stronger than that of the potential flow field of the thin trailing part of the stator vanes.

To identify the effect of the tip clearance flow on the blade pressure distribution, the static pressure at the casing was plotted together with the blade pressures at 94.5 percent span in Fig. 8. Both distributions agree well at the leading and the trailing edge, thus confirming the measurement techniques. Starting at about 25 percent axial chord, where the tip clearance flow begins to emerge from the gap, the pressures at the suction side are considerably increased. The corresponding pressure decrease on the pressure side starts at about 50 percent chord, where the intensity of the crossflow through the radial gap is amplified. This leads to clearly visible cores in the tip region. Another interesting fact is...
that even at the casing downstream of the trailing edge the static pressure at the suction
side is considerably lower than at the pressure side, see Fig. 9. As evident in Fig. 10, this
causes intense mixing in the wake downstream of the rotor.

At 12 percent axial chord downstream of the rotor, hot-wire measurements were carried out. The measurements were recorded at 32 radial and 9 circumferential locations. From these measurements, a stop-action sequence was computed, which can be understood as “photographs” of the rotor exit flow. All data presented in this chapter refers to the same position relative to the upstream stator, while the rotor blades pass the stationary observer. The four rotor positions referenced here are equally spaced, i.e. between each position the rotor passes 20 percent of the stator pitch.

Prior to the effects of stator wakes on the rotor flow, the basic flow pattern at rotor exit is discussed. The secondary flow field in the stationary frame is depicted in Fig. 10 for a specific rotor-stator position. The arrows represent the local velocity vectors viewed from the downstream direction. This direction is defined by the overall mass-averaged pitch angle $\alpha$ and a zero yaw angle. Between 2.7 to 9 percent and 92 to 98 percent span only 2D-hot-wire probes could be used. To obtain the radial velocity components in these locations, the Euler equations were solved under the assumptions of steady-state flow and negligible pressure gradients in axial direction. Fig. 10 reveals strong crossflow components from the pressure side into the wake area, as expected from pressure measurements, Fig. 9. In the tip region, the clearance causes outwards directed flow at the pressure side and strong inwards directed components at the suction side. Stator passage vortices persisting throughout the rotor as mentioned by Sharma et al. (1985) could not be detected. At the hub, the exit swirl is directed against the sense of rotation, due to the lower circumferential speed.

A possible explanation for the wavy flow pattern at the hub might be the combination of small measurement errors and the simplifications used for the calculation of the radial components. However, since the computed flow pattern at the tip is very smooth, the hub flow may be influenced by the small gap between the rotor and the downstream stationary hub. The gap, which is sealed by a labyrinth seal, is located right under the measurement plane.

The secondary flow fields in the relative frame, Fig. 11, show two intense passage vortices and a tip clearance vortex. At rotor position 1, which corresponds to Fig. 10, the tip clearance vortex extends to about 20 percent of the blade-to-blade cross-section. This
is in good agreement with the casing static pressures, Fig. 9. The relatively small tip clearance vortex with a radial extension of only 10 percent span is due to the small radial gap of 0.6 percent span.

The passages vortex in the tip region is generated by the action of the blade-to-blade pressure gradient on the low-momentum fluid at the casing as it enters the rotor passage. The overturning, also observed in Fig. 9, is further amplified by the relative movement of the casing. Close to the suction side, the interaction between passage vortex and tip clearance vortex leads to strong inwards directed radial components.

At the hub, the maximum overturning occurs at about 15 percent span and not, as expected, close to the endwall. A similar effect was observed by Binder et. al. (1985b) in a transonic turbine stage and by Hunter (1982) in a large-scale turbine test rig, but presented without detailed explanation. Hunter found that the hub boundary layer was extremely thin and that overturning occurred only in the first part of the rotor passage. The measured stator exit flow, Fig. 6 and 7, shows also very thin boundary layers at rotor inlet.

Fig. 11: Secondary flow field at rotor exit for 4 different rotor-stator positions, relative frame. Areas with increased turbulence shaded.
It is unlikely that these thin boundary layers cause such strong crossflow. The separation of the passage vortex from the hub is also unlikely, since low momentum fluid tends to migrate towards the hub due to the radial pressure gradient. Therefore, it was assumed that the distorted rotor inlet flow strongly influences the flow through the rotor passage. As already mentioned, in Fig. 7 the incidence angle at rotor inlet increases towards the hub with a local decrease of about 10 degrees at 15 percent span. The circumferentially mass-averaged outlet flow angle in the relative frame, Fig. 12, shows the highest overturning at the same location. Without remarkable reduction, the distortions of the stator outlet are visible in the rotor outlet flow. The profile pressure distributions, Fig. 8, measured at 3.7 and 10.9 percent span remain basically the same; only the blade pressures at the leading edge are influenced by the local displacement of the stagnation point to the suction side. In spite of the reduced incidence angle at 10.9 percent span the low pressures at the leading part of the suction side indicate a strong acceleration and increased circumferential velocity components towards the suction side. This could be explained by the fact that the profile pressure distribution in the three-dimensional flow cannot respond to the local change of incidence as expected from the two-dimensional theory. It seems, as if the pressures from higher and lower radial locations superimpose the local unloading of the blade, thus causing the observed overturning in the hub region at 15 percent span.

The measured exit velocity distribution in the relative frame is shown in Fig. 13. The wake can be clearly distinguished, with sharp gradients at the pressure side edge. The favourable pressure gradient on the pressure side leads to the development of a thin profile boundary layer. On the suction side, the wake is more extended. At the present stage of the investigation, no measurements of the rotor boundary layers were made. Hodson and Addison (1988) found that the transition of the profile boundary layer in their turbine rotor occurred shortly before the peak suction point. This point is located at about 60 percent chord at the tip, 40 percent chord at midspan and close to the leading edge at the hub, see Fig. 8. The transitional and turbulent nature of the suction side boundary layer accounts for the thicker wake at the suction side, which is even more extended at the hub than at the tip. The two minima in the velocity distribution at about 60 and 40 percent span seem to be accumulations of low energy fluid, which is transported to these locations by the passage vortices from higher and lower locations, respectively. This leads to a small rotor wake close.
to the endwalls, where the wake is energized by the passage vortices.

As a consequence of the different number of airfoils in stator and rotor, the flow field in two adjacent rotor passages is slightly different. For the sake of clarity, only the flow of the left rotor blade passage, Fig. 11, which corresponds to the contour plot of Fig. 13 and Fig. 14, is discussed here. The wakes are characterised by increased turbulence and velocity defects. Fig. 14 shows the turbulence intensity measured downstream of the rotor. Close to the endwalls, where only 2D-measurements were carried out, the radial turbulence intensity was assumed to be of the same order as the other two components. This seemed to be justified from the analysis of the 3D-measurements close to the endwalls.

To explain the flow behavior, the convection of the stator wakes through the rotor passage was calculated using a simplified, non-viscous, 2D-flow model. The modelling effort concentrated on predicting the path of the stator wake centerline as it encounters the blade row and progresses through the passage. The data did not allow the accurate prediction of leading and trailing edge propagation rates of the stator wakes. The results agreed well with locations of increased turbulence in the rotor exit flow. These areas, showing a turbulence level of more than 10 percent outside or 15 percent inside of the rotor wake, are shaded in Fig. 11. The positions of the stator wakes remain nearly unchanged in the stationary frame. Because of the higher axial velocity and the negative swirl at the hub, the leading edge of the stator wake is detected first at the hub (rotor position 3), then at midspan (position 1) and later in the tip region (position 3).

According to a reduced frequency of

\[
\omega_R = \frac{n \times z v \times b_R}{c X}
\]

(n shaft speed, \(z\) number of stator vanes, \(c\) axial velocity, \(b\) axial chord of rotor blades), about two stator wakes are simultaneously present in each rotor passage.

The flow at midspan is only little affected by the stator wakes. The average free-stream velocity decreases from 115 m/s to 90 m/s at the rotor wake centerline, the turbulence intensity increases from 6 percent to 10 percent. The highest turbulence of 12 percent at midspan is detected at rotor position 1, with a velocity defect of 20 percent in the wake. At positions 2 and 3, the turbulence in the rotor wake is reduced to 8 percent, with a velocity defect of 26 percent. At position 4 the turbulence level is low, i.e. 8 percent, and the velocity defect amounts to 20 percent. The higher turbulence and the higher velocity in the wake is caused by the early transition of the suction side boundary layer, which is likely to occur when the stator wake impinges the blade surface, Hodson and Addison (1988).

With no stator wake visible at the hub, the leading edge of a stator wake intercepts the lower passage vortex, which coincides with increased turbulence. A similar interception was observed by Herbert and Tiedemann (1989) in a linear cascade with an upstream wake generator. In the upper part of the channel the turbulence level is generally higher than at the hub. One explanation for the high turbulence are the high losses and incidence fluctuations, which arise as the rotor passes the stator wake area. Another explanation was suggested by Binder (1985c), who found that the cutting of the stator secondary vortices by the rotor blade was associated with high turbulence energy of about the same magnitude as measured here. The cutting of the passage vortex occurs shortly after the rotor blade has passed the centerline of the stator wake. Since intensity and size of the stator passage vortex at the casing fairly
exceed that of the hub passage vortex, Fig. 5, the turbulence in the outer region is considerably higher. It is further amplified by the interaction between passage and tip clearance vortex.

Because of the strong radial components, the region with the highest turbulence and lowest velocity has moved towards midspan.

The analysis of the data showed, that the tip clearance vortex is only slightly affected by the passing stator wakes, which was also observed by Sharma et al. (1985). Its size, velocity and turbulence were not significantly different at the various rotor positions, although, the intensity seemed to be reduced at position 3 and 4, when the stator wake was visible in the tip region.

Summary and concluding remarks

The measurements of the unsteady, three-dimensional rotor flow field have demonstrated that the development of the rotor secondary flow, the rotor wake and the outlet flow angles are mainly influenced by the circumferentially-averaged, nonuniform stator exit flow. But, as observed by several other investigators, also the periodically unsteady rotor inlet flow caused by the upstream stator influences the rotor flow significantly. The present results can be summarized as follows:

1) The passage vortices in the rotor are strongly influenced by the nonuniform stator outlet flow and cause the accumulation of low energy fluid in the rotor wake close to mid-span.

2) The stator wakes are discernible in the rotor outlet. Data of fast-response hot-wire probes traversed circumferentially relative to the stator show a variation of the time-averaged rotor exit velocity of 5 percent at midspan and more than 9 percent in the regions of the passage vortices. The time-averaged outlet flow angle varies between 3 deg. at midspan and 7 deg. in the outer regions. This emphasizes the need to traverse probes circumferentially, if the probes are mounted at a short axial distance downstream of the rotor, even if only steady-state values will be recorded.

3) The stator wakes act as "negative jets" in the rotor passages and amplify the crossflow components of the passage vortices towards the suction side.

4) The stator wakes impinging on the rotor blade surface have a significant effect on the rotor wakes at midspan. The early boundary layer transition increases the turbulence intensity in the wake and causes a lower freestream velocity at the edge of the wake, thus leading to higher profile losses.

5) The highest fluctuations of the velocity, the flow angle and the turbulence intensity are detected in the hub and tip region. Here the deep stator wakes and the cutting of stator secondary vortices lead to periodically high turbulence levels and intensified crossflow components towards the suction side.

These results suggest that the radial distribution of the flow properties has to be included into the design process. Although no correlation for a quantitative assessment of the unsteady effects has been derived from the data, the results can be very useful for estimating the importance of these effects, if a steady-state design method is applied.

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