Effects of Simulated Rotation on Tip Leakage in a Planar Cascade of Turbine Blades
Part II: Downstream Flow Field and Blade Loading

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ABSTRACT

This paper and its companion paper present experimental results on the effects of simulated rotation on the tip leakage in a linear turbine cascade test. Part II examines the downstream flow field. For clearance sizes of 2.4 and 3.8 percent of the blade chord measurements were made in two planes downstream of the trailing edge using a seven-hole pressure probe. Significant changes in the tip leakage vortex and passage vortex structures are observed with the introduction of relative motion. The effects of clearance size and rotation on the relationship between bound circulation and tip-vortex circulation are discussed. The validity of a previously developed tip-vortex model for the case of rotation is examined in the light of the measurements. Finally, for clearances of 1.5, 2.4 and 3.8 percent of the blade chord the effects of rotation on blade loading are studied through static pressure measurements on the blade surfaces. The distortion of the surface pressure field near the tip is found to be reduced with increasing wall speed. This is consistent with the reduced strength of the tip-leakage vortex as wall speed is increased. For all measurements two wall speeds are considered and the results are compared with the case of no rotation.

NOMENCLATURE

\[ C_P = \frac{P - P_{CL}}{\frac{1}{2} \rho V_{cl}^2} \]

- static pressure coefficient

\[ h = \text{span} \]

\[ H = \frac{\delta^*}{\theta} \]

- boundary layer shape factor

\[ L = \text{blade lift force} \]

\[ P = \text{static pressure} \]

\[ r = \text{radial co-ordinate} \]

\[ Re = \frac{\rho V_{cl} c}{\mu} \]

- Reynolds number based on blade chord

\[ s = \text{streamwise co-ordinate} \]

\[ S = \text{blade spacing} \]

\[ t_{\text{MAX}} = \text{blade maximum thickness} \]

\[ v_t = \text{tangential component of velocity} \]

\[ V = \text{resultant velocity} \]

\[ V_{\text{bld}} = \text{velocity at the boundary layer edge} \]

\[ V_{\text{m}} = \text{mean velocity through the cascade} \]

\[ V_{\text{rel}} = \text{relative tip-wall speed} \]

\[ x, y, z = \text{co-ordinates in axial, tangential and spanwise directions} \]

\[ x' = \text{co-ordinate in chordwise direction} \]

\[ y' = \text{local pitchwise co-ordinate} \]

\[ \alpha = \text{pitch flow angle, measured from axial direction} \]

\[ \alpha_m = \text{mean flow angle, } \tan(\alpha_m) = \frac{\tan(\alpha_t) + \tan(\alpha_s)}{2} \]

\[ \beta = \text{blade metal angle} \]

\[ \gamma = \text{blade stagger angle} \]

\[ \Gamma_B = \text{bound circulation} \]

\[ \Gamma_V = \text{tip leakage vortex circulation} \]

\[ \Gamma' = \text{non-dimensional circulation} \left( \frac{r}{cV_L} \right) \]

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\[ \delta \] = boundary layer thickness

\[ \delta' \] = axial displacement thickness

\[ \delta'' \] = transverse displacement thickness

\[ \theta_s \] = axial momentum thickness

\[ \theta_r \] = transverse momentum thickness

\[ \theta_{\phi} \] = axial interaction momentum thickness

\[ \theta_{\phi} \] = transverse interaction momentum thickness

\[ \Lambda \] = Owen's eddy viscosity constant (Eqn. 8)

\[ \mu \] = dynamic viscosity

\[ \nu \] = kinematic viscosity

\[ \nu_e \] = eddy viscosity

\[ \rho \] = density

\[ \tau \] = tip gap height

\[ \omega \] = vorticity

\[ \omega' \] = non-dimensional vorticity \((\frac{\omega C}{V_1})\)

Subscripts

\[ CL \] = centreline value at inlet

\[ e \] = boundary layer edge value

\[ x,y,z,s \] = component in x, y, z and streamwise direction

\[ 1,2 \] = cascade inlet and outlet

INTRODUCTION

Part II of the paper presents detailed measurements from downstream of the cascade as well as blade-loading measurements. Taken together, the two papers present the most complete set of data of which the authors are aware for the tip-leakage flow in a turbomachinery blade row, with and without relative wall motion.

Two aspects of the downstream flow are of interest: the vorticity field and the evolution of the tip-leakage losses. In the present flow, the blade row was stationary and the rotation was simulated using a moving belt. The belt does work on the flow and thus produces a total pressure rise. It appears impossible to separate the total pressure rise due to belt work from the total pressure drop due to losses, based on measurements of the pressure field alone. The same problem would of course also arise in a rotating test rig or an actual turbine. Therefore, the paper does not address the question of losses.

Most previous studies of the tip-leakage vorticity field have been performed in compressor cascades and rotors. An exception is the authors' measurements in the present turbine cascade for the stationary wall case (Yaras and Sjolander, 1989). The compressor studies include those of Lakshminarayana and his co-workers (eg. Lakshminarayana and Horlock, 1962; Lakshminarayana et al., 1987) and Inoue and his co-workers (eg. Inoue et al., 1986; Inoue and Kurokami, 1989). All the studies show that the circulation of the tip-leakage vortex is significantly less than the bound circulation of the blade and is a function of the gap height and the relative wall speed. Lakshminarayana and Horlock introduced the term "retained lift" to describe this effect and offered a physical explanation. Among other things, the present data are used to examine critically the nature of this phenomenon.

As noted, most previous studies, particularly where relative wall motion was included, have considered compressor flows. Significant differences can be expected in turbine flows since the direction of the relative wall motion is reversed. The present study documents some of these differences.

EXPERIMENTAL APPARATUS AND PROCEDURES

Test Section and Test Cascade

The test section and moving-belt endwall were described in Part I. Figure 1 summarizes the geometry of the test cascade and shows the location of the downstream measurement planes. Planes A and C1 are 0.475 and 0.96 axial chord lengths downstream of the trailing edge respectively.

Instrumentation and Data Acquisition

Only instrumentation which is specific to Part II is described here.

The flow downstream of the blades was measured with a seven-hole pressure probe of 2.4 mm diameter and 60 degrees apex angle at the conical tip. In each measurement plane the probe was aligned with the undisturbed flow direction and was traversed in spanwise and pitchwise directions using a motorized traversing mechanism. This arrangement required the probe to be used in a non-nulling mode. Thus, the probe was calibrated in 5 degree steps through all combinations of pitch and yaw out to 50 degrees of misalignment for both angles, as described by Yaras and Sjolander (1989). Inferred flow angles are estimated to be accurate to within 2 degrees, while the error in the measured local dynamic pressure is estimated to be within 5 percent of the actual local dynamic pressure. The probe position is estimated to be accurate to within 0.25 mm in all three co-ordinate directions.

The outer half of the centre blade in the cascade is instrumented with 14 chordwise rows of static pressure taps, each row having 37 taps on the pressure side and 36 on the suction side.
The employment of equations (2) and (3) instead of the conventional expressions in terms of velocity gradients enables the calculation of \( \omega_x \) and \( \omega_z \) from measurements in a single \( y-z \) plane. The accuracy of the approximate values given by equations (2) and (3) was found to be excellent by Yaras and Sjolander (1989). The streamwise vorticity is then determined from the vorticity components in the \( x \) and \( y \) directions using the expression:

\[
\omega_x = \frac{1}{u} \left( \omega_0 x + \frac{1}{\rho} \frac{\partial P}{\partial y} \right),
\]

\[
\omega_z = \frac{1}{u} \left( \omega_0 z + \frac{1}{\rho} \frac{\partial P}{\partial z} \right).
\]

The components of the vorticity vector are obtained in cartesian co-ordinates within a co-ordinate system aligned as shown in Figure 1. The \( x \) component of vorticity is determined using,

\[
\omega_x = \frac{\partial w}{\partial y} - \frac{\partial v}{\partial z},
\]

where the \( y \) and \( z \) components of vorticity are obtained from Euler's equation written in terms of vorticity:

where \( \alpha \) is the local midspan flow direction in the measurement plane of interest.

The gradients in equations (1), (2) and (3) are determined by central differences except at the boundaries of the measurement grid where simple differences are used. The grid extended one passage width in the pitchwise direction and half the blade span in the other direction. A spacing of 10 mm between adjacent data points was found to give accurate local gradients in both measurement planes.

The circulation around any closed path downstream of the blades is calculated using Stokes' theorem. This circulation is compared with the bound circulation which is obtained from

\[
\Gamma_B = \frac{L}{\rho V_m},
\]

where the lift force, \( L \), is determined using the blade-surface static pressures measured at midspan.

**EXPERIMENTAL RESULTS**

**Operating Conditions**

As noted in Part I, limitations on the belt speed resulted in the measurements being made at a blade Reynolds number of \( 1.4 \times 10^5 \), which is about one third of the value in the actual turbine. As also mentioned, a trip wire was applied to the blades to eliminate a laminar separation which occurred near the leading edge at the lower Reynolds number. To ensure that the downstream flow field was not affected significantly by the
reduced Reynolds number, vorticity measurements were made at different Reynolds numbers for the stationary tip wall. The results were found to be essentially identical.

Downstream measurements were made for two clearances of 2.4 and 3.8 percent of the blade chord. Blade loading measurements were obtained at an additional clearance of 1.5 percent. The tip-wall boundary layer at mid-passage and in the leading-edge plane was traversed for all three clearances. The resulting loci of velocity vectors through the layers are plotted in Figure 2 for the maximum belt speed. As discussed in Part I, if the inlet boundary layer is coplanar in the stationary (wall) frame of reference in the actual turbine, vector plots such as those in Figure 2 will form triangles. As seen from the figure, the experimental flow closely simulated a coplanar inlet flow, particularly at the larger tip clearances. The weak variation of the boundary layer with tip clearance is thought to be due to the small amount of outflow through the varying gap between the leading edge of the belt and the stationary wall. The integral parameters of the boundary layers, in the blade frame of reference (which is equivalent to the wall frame of reference in the engine), are given in Table 1.

As with the gap measurements, downstream and blade loading measurements were obtained at three different wall speeds corresponding to 0, 60 and 100 percent of Engine Equivalent Speed (EES).

**Measurement of the Downstream Flow Field**

The effects of rotation on the downstream flow field is observed from Figures 3(a), 3(b) and 3(c) which present the streamwise vorticity distribution in a plane 0.475 axial chord lengths downstream of the trailing edge (Plane A). The same contour interval is used on all three plots for easy comparison.

**TABLE 1. INLET BOUNDARY-LAYER PARAMETERS.**

<table>
<thead>
<tr>
<th>CLEARANCE, υ/c</th>
<th>0.015</th>
<th>0.024</th>
<th>0.038</th>
</tr>
</thead>
<tbody>
<tr>
<td>EES (%)</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>υ (mm)</td>
<td>3.7</td>
<td>6.0</td>
<td>9.5</td>
</tr>
<tr>
<td>δ* (mm)</td>
<td>2.7</td>
<td>3.0</td>
<td>4.5</td>
</tr>
<tr>
<td>θ (mm)</td>
<td>1.1</td>
<td>2.7</td>
<td>0.3</td>
</tr>
<tr>
<td>H (mm)</td>
<td>2.4</td>
<td>4.3</td>
<td>1.6</td>
</tr>
<tr>
<td>δ* (mm)</td>
<td>4.3</td>
<td>1.4</td>
<td>1.4</td>
</tr>
<tr>
<td>θ (mm)</td>
<td>0.4</td>
<td>0.3</td>
<td>0.4</td>
</tr>
<tr>
<td>H (mm)</td>
<td>3.8</td>
<td>4.7</td>
<td>3.5</td>
</tr>
<tr>
<td>θ (mm)</td>
<td>0.7</td>
<td>0.6</td>
<td>0.8</td>
</tr>
<tr>
<td>θ (mm)</td>
<td>0.2</td>
<td>0.1</td>
<td>0.2</td>
</tr>
</tbody>
</table>

Figures 3(a) and 3(b) show that as the wall speed is increased from 0 to 60 percent EES the strength of the tip-leakage vortex is reduced considerably. At the same time the passage vortex adjacent to the tip-leakage vortex is somewhat enhanced by the scraping effect of the blades on the tip-wall boundary layer. It is worth pointing out that the blade tips do not have to intrude physically into the tip-wall boundary layer for this scraping effect to occur: the pressure field created by the blades is sufficient to introduce this effect. This was noted earlier by Allen and Kofskey (1955) who undertook smoke visualization experiments in a low-speed turbine rotor. Figures 3(a) and 3(b) also show that the viscous dragging effect of the wall motion moves the passage vortex, and thus the tip-leakage vortex, toward the suction side of the passage. In the process, part of the enhanced passage vortex squeezes between the tip wall and the tip-leakage vortex. As the wall speed is increased further to 100 percent EES (Figure 3(c)), the tip-leakage vortex strength continues to decrease while the passage vortex is further strengthened. In addition, the two vortices are moved further towards the suction side. However, the displacement is considerably less than that observed between 0 percent and 60 percent EES. The same trend was also observed for 2.4 percent clearance. Of course for very large clearances the blade or its pressure field cannot produce a scraping effect on the tip-wall. In that case the observed downstream flow field with rotation is likely to approach that of stationary configuration (e.g. see Lakshminarayana et al., 1987).

These observations for turbine blades can be contrasted with the observed effects of rotation in compressors, where the direction of rotation is opposite to that for turbines. With rotation one would expect the tip-leakage vortex in compressors to be shifted towards the pressure side of the passage. This appears to be confirmed by Dean’s (1954) results in a linear cascade and by Lakshminarayana et al.’s study (1987) in a compressor rotor. In addition, for practical clearance sizes the wall scraping effect...
should produce a scraping vortex on the leading (pressure) side of the blade. Phillips and Head (1980) and Inoue et al. (1986) found this and observed that the sense of the rotation of this scraping vortex is the same as that of the tip-leakage vortex. Thus, the merging of the scraping vortex with the passage vortex which is evident in turbines is not likely to occur in compressors.

As described in the accompanying paper, measurements inside the tip gap suggested a global reduction with increasing wall speed in the pressure difference which drives the tip-leakage flow. The mechanism by which this reduction is created seems evident from the downstream flow field. As wall speed is increased, the passage vortex and thus the tip-leakage vortex move toward the suction side of the passage, partly blocking the tip-gap exit. This blockage seems to raise the pressure at the gap outlet, thereby reducing the driving pressure difference for the leakage flow. No attempt was made to obtain static pressures from the gap measurements. Some rise in the pressure with rotation was observed on the suction side of the blade close to the tip (see Fig. 10(b)). However, Sjolander and Amrud (1987) had earlier concluded that the distorted pressures near the blade tip were a very localized effect and were not representative of the pressure field experienced by the bulk of the gap flow. Thus, the observed change in the blade pressures is thought to be a consequence of the weaker tip-leakage vortex which follows from the reduced gap flow rate. The interpretation given here is supported by the results of Morphis and Bindon (1988) who concluded from blade tip measurements that the gap outlet pressure increased with increasing wall speed. By contrast, in compressors the scraping vortex would be expected to influence conditions on the pressure side of the blade. Inoue and Kuroumaru (1989) did in fact note an increase in the gap inlet pressure with rotational speed from their measurements in a compressor rotor. This would have the effect of increasing the gap flow rate.

Effects of Wall Motion and Clearance Size on Tip-Vortex Strength

The tip-vortex circulation is employed as a variable in numerous tip-leakage loss models (e.g., Ainley and Mathieson, 1951; Lakshminarayana, 1970) as well as in the tip-vortex model developed by the authors (Yaras and Sjolander, 1989). Therefore, it is essential to identify the dominant variables which determine the magnitude of this circulation. Figures 3(a), 3(b), and 3(c) suggested that there was considerable reduction in the tip-vortex strength with increasing wall speed. To quantify this effect, the tip-vortex circulation was determined by integrating over the area the x-wise component of vorticity within the tip-vortex flow field in Plane A. The results are shown in Figure 4. Significant reduction in the vortex strength with wall motion is evident. The reduction is non-linear with wall speed, as noted earlier. The trend in the tip-vortex strength with clearance size appears to remain the same regardless of the magnitude of the wall speed.

The effect of clearance size on tip-vortex strength has been examined by a number of authors (e.g., Lakshminarayana and Horlock, 1962; Lewis and Yeung, 1977; Inoue et al., 1986; Yaras and Sjolander, 1989). There is general agreement that the circulation in the tip-leakage vortex appears to be substantially less than the bound circulation of the blade. A physical explanation for this was offered by Lakshminarayana and Horlock (1962) who suggested that some of the vortex lines of the bound circulation lines jump across the tip gap and terminate on the tip wall instead of shedding downstream as part of the tip vortex; the authors
FIG. 4. EFFECT OF TIP-WALL RELATIVE MOTION ON THE TIP-VORTEX CIRCULATION.

introduced the term "retained lift" to describe this effect. The same argument was used by, for example, Lewis and Yeung (1977) to explain their results. However, convection of vorticity across the gap requires a viscous flow field inside the gap. The stationary-wall results of Yaras et al. (1989) showed that a significant portion of the gap height contained loss-free fluid. Likewise, the results of Part I suggest that a large inviscid core is present in the gap flow when there is relative wall motion as well. These observations clearly undermine Lakshminarayana and Horlock’s argument. An alternative argument can be formulated based on the vorticity generated on the tip-wall.

Figures 3(a) to 3(c) show very little streamwise vorticity associated with the blade wakes. Thus, all the bound vorticity is expected to shed along the blade tip. This chordwise vorticity shed from the tip can be viewed as being generated on the blade tip under the influence of the pressure gradient between the suction and pressure sides of the blade. Specifically, if the blade tip is square, as is the case for the turbine blade, the shed vorticity will be generated at the corner formed by the pressure surface and the blade tip (Figure 5). Yaras and Sjolander (1989) found that for clearance sizes up to about 5 percent of the blade chord, the undisturbed blade pressure difference is impressed onto the tip wall with little reduction in magnitude. This pressure gradient produces a strong pitchwise acceleration and chordwise vorticity must then be generated on the tip wall, with a sense of rotation opposite to that of the bound vorticity shedding from the blade tip and of comparable magnitude. The presence of this wall vorticity layer is evident in Bindon’s (1987) flow visualization studies. Interaction of these chordwise vorticity fields of similar strength but opposite sign provides a feasible explanation for the difference between the tip-vortex circulation and the bound circulation. In the absence of viscous forces this interaction does not occur inside the gap; rather, it occurs after the fluid has left the tip gap. The measurements of Yaras and Sjolander (1989) suggest that the majority of this interaction occurs inside the blade passage, with some additional interaction taking place downstream of the blades.

The reduction of the tip-vortex strength with wall motion can also be explained in terms of the chordwise vorticity generated on the tip-wall. In the case of turbines this vorticity layer is further strengthened by the relative motion of the wall. In addition, the wall motion influences the amount of vorticity shed from the blade tip by altering the pressure field in its vicinity. As discussed earlier, wall motion enhances and relocates the passage vortex and reduces the pressure difference driving the tip-leakage flow. This in turn reduces the velocity in the gap main stream and thus the vorticity generated at the blade tip. On the other hand, this effect should tend to give an equal reduction in the generation of chordwise vorticity of the opposite sense on the tip wall. Perhaps the main effect of wall motion is its direct effect of strengthening the vorticity layer on the tip wall. By comparison, in compressors the direct effect of the wall motion will be to reduce the strength of wall layer. This is consistent with the values of tip-vortex circulation observed by Inoue et al. (1986) in their compressor rotor: they were higher than the values obtained at the same clearances for stationary walls by Yaras and Sjolander (1989), Lakshminarayana and Horlock (1962) and Lewis and Yeung (1977).

Modelling of the Tip Vortex with Relative Wall Motion

While rotation results in a significant reduction in the tip-vortex strength, Figures 3(a) to 3(c) show that the axisymmetric nature of the vortex is not altered significantly in the process. This was particularly true for the smaller clearance size of 2.4 percent of the blade chord. Thus, it may be possible to extend the tip-vortex model proposed by Yaras and Sjolander (1989) to the case with rotation. This model could be used, for example, to estimate the cross flow that will be induced by the tip-leakage vortex at a subsequent blade row.
FIG. 6. TIP-LEAKAGE VORTEX AS A DIFFUSING LINE VORTEX.

The model proposed by Yaras and Sjolander is an adaptation of Lamb's (1932) solution for the laminar diffusion of a simple line vortex. From Lamb's analysis the distribution of tangential velocity through the vortex is obtained from,

\[ v_0 = \frac{\Gamma}{2\pi r} \left( 1 - e^{-\frac{r}{s}} \right) \quad (6) \]

where \( \Gamma \) is the circulation of the tip vortex, \( s \) is the distance from the effective origin of the vortex (Figure 6), \( V_2 \) is the undisturbed velocity at the cascade exit and \( v \) is the molecular viscosity. The turbulent nature of the tip vortex was incorporated by replacing the molecular viscosity \( v \) by \( v + v_e \) where \( v_e \) is an eddy viscosity. This simple approach has been reasonably successful in modelling the downstream evolution of turbulent wing-tip vortices. A number of correlations have been proposed for the eddy viscosity. Yaras and Sjolander (1989) evaluated two of them against the effective eddy viscosities obtained in the experiment. The experimental values were calculated from the streamwise variation of the vorticity on the centreline of the tip-leakage vortex. The first correlation examined was the expression suggested by Squire (1965):

\[ \frac{v}{v_e} = \alpha \frac{\Gamma}{v} \quad (7) \]

and the second that due to Owen (1970):

\[ \frac{v}{v} = \Lambda^2 \frac{\Gamma}{\sqrt{v}} \quad (8) \]

where \( \alpha \) was strongly dependent on the clearance size whereas \( \Lambda \) showed relatively little variation. The variation of the vortex parameters with wall speed is shown in Figure 7. While \( \Lambda \) appears to depend somewhat on wall speed, this dependence is considerably less than for \( \alpha \). Thus, for case of rotation as well, Equation (8) appears to be the preferred correlation for the eddy viscosity. The figure also shows that the effective origin shifts somewhat towards the trailing edge with both increasing wall speed and decreasing clearance size. In both cases, the shift is associated with a reduction in the strength of the tip-leakage vortex. Details of the procedures by which the vortex
parameters are obtained from the measurements are given by Yaras and Sjolander (1989).

To apply the model, only estimates of A and the axial position of the effective origin, $x_0$, are required. The streamwise variation of the vortex diameter and the magnitude of the cross-flow velocities are then given by the model. Therefore, a comparison between the calculated and measured cross-flow velocity distributions represents a partial test of the model.

For comparison purposes, the measured tangential components of velocity were extracted for spanwise lines passing through the vortex centres. For the stationary configuration, Yaras and Sjolander (1989) defined the tangential velocity as the component in the plane normal to the vortex centreline. This plane was also essentially normal to the mean velocity at the cascade outlet. However, as wall motion is introduced the tip-vortex structure downstream of the blades was found to be modified such that the vortex centreline is no longer aligned with the outlet flow (see Figure 8). Therefore, the alternative of using the local midspan flow angle as the reference direction has been adopted here for convenience. It was found that the choice of reference direction alters the cross flow distribution in a global fashion only: it has very little effect on the peak to peak variation of the inferred cross flow, which is the quantity of particular interest. The experimental and calculated results are shown in Figures 9(a), 9(b) and 9(c). The results are presented with the vortex centres coincident and the relative position of the tip-wall is indicated. The predictions are slightly shifted with respect to the experimental data partly as a result of the aforementioned choice of reference direction. Also, the velocities induced by the tip vortices of the other blades have not been taken into account. Nevertheless, the peak to peak variation in the cross flows and the diameter of the vortices are seen to be reasonably well predicted for all speeds.

**FIG. 8. EFFECT OF WALL MOTION ON TIP VORTEX**

**FIG. 9. DISTRIBUTIONS OF TANGENTIAL VELOCITY THROUGH CENTRE OF TIP VORTEX (t/c = 0.038).**

**Effect of Wall Motion on Blade Loading**

The effects of rotation on blade pressures near the tip and at midspan are shown in Figures 10(a) and 10(b). Very little influence is evident at midspan. However, near the tip the suction-side pressures are altered considerably as wall speed is increased. This results in reduced loading near the tip with increasing wall speed, as was also inferred by Graham (1986) from somewhat sparse blade pressure measurements. The tests for clearance sizes of 2.4 and 1.5 percent showed that the effect of wall motion on blade-tip loading increases with increasing...
CONCLUSIONS

Detailed data have been obtained for the effect of relative wall motion on the downstream flow field and blade loading in a linear turbine cascade.

It was concluded that the strength of the tip vortex was reduced considerably with the introduction of wall motion. At the same time, the passage vortex was enhanced by the scraping effect of the blades. Both vortices were dragged toward the suction side of the passage where they appeared to partially block the outlet flow from the tip gap. This seems to explain the apparent reduction in the pressure difference driving the flow through the gap and the consequent reduction in the gap velocity (as described in Part I).

The circulation of the tip vortex was found to decrease with decreasing clearance size and increasing wall speed and to be substantially less than the bound circulation of the blade. However, the results do not support the concept of "retained lift" introduced earlier to explain this effect. For both stationary and moving tip walls, the flow inside the gap appears to have a large, essentially inviscid core. Thus, all of the bound vorticity must be shed at the blade tip. At the same time, a chordwise vorticity layer of opposite sense is generated on the tip wall by the pressure field which is impressed onto it. This layer undergoes a strong interaction with the tip vortex shortly after the flow leaves the gap. This appears to account for the reduced circulation observed downstream of the trailing edge.

The tip-vortex model developed by Yaras and Sjolander (1989) was extended to the rotating configuration. Once the effects of rotation on the tip-vortex circulation, the location of the effective origin of the vortex and the eddy viscosity were accounted for, the model gave good predictions for the cross flow associated with the vortex.

Blade suction-side pressures near the tip were altered considerably by the wall motion. The net effect was a reduction of the blade loading near the tip, which is consistent with the observed decrease in the strength of the tip vortex with relative wall motion.

ACKNOWLEDGEMENTS

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FIG. 10. BLADE PRESSURE DISTRIBUTIONS (v/c = 0.038).

clearance size. The measurements at the other spanwise rows of static taps showed that the dominant effects of wall motion were confined to a region within a few gap heights of the tip.

The distortion of the suction-side pressure field near the blade tip (Figure 10(b)) is a consequence of the tip vortex. Thus, the reduced blade loading with increasing wall speed is consistent with the weakening of the tip vortex inferred from the reduced core velocity in the gap and observed from the downstream measurements.

The relative unloading near the tip due to rotation should not necessarily be viewed as detrimental. Sjolander and Amrud (1987) found that the increased loading caused by the tip vortex is accompanied by a rotation of the load vector toward the axial direction. Thus, in an actual machine the additional loading would appear as an axial thrust on the rotor instead of a tangential force augmenting the rotor shaft torque. As was previously noted, in compressors the tip-vortex strength is increased with rotation while the pressure level on the pressure side of the blade tip is increased by the scraping effect. Thus, one would expect the blade loading near the tip to increase with rotation. This was observed by Dean (1954).
REFERENCES


