PRESSURE SURFACE SEPARATIONS IN LOW PRESSURE TURBINES:
PART 2 OF 2 - INTERACTIONS WITH THE SECONDARY FLOW

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ABSTRACT
This paper describes a study of the interaction between the pressure surface separation and the secondary flow on low pressure turbine blades. It is found that this interaction can significantly affect the strength of the secondary flow and the loss that it creates. Experimental and numerical techniques are used to study the secondary flow in a family of four low pressure turbine blades in linear cascade. These blades are typical of current designs, share the same suction surface and pitch, but have differing pressure surfaces.

A mechanism for the interaction between the pressure surface separation and the secondary flow is proposed and is used to explain the variations in the secondary flows of the four blades. This mechanism is based on simple dynamical secondary flow concepts and is similar to the aft-loading argument commonly used in modern turbine design.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>C_d</td>
<td>dissipation coefficient</td>
</tr>
<tr>
<td>C_X</td>
<td>axial chord (m)</td>
</tr>
<tr>
<td>C_L</td>
<td>= \int C_p d(x / C_X)</td>
</tr>
<tr>
<td>C_p</td>
<td>circumferential loading coefficient</td>
</tr>
<tr>
<td>h</td>
<td>span (m)</td>
</tr>
<tr>
<td>H</td>
<td>=\delta*/\theta</td>
</tr>
<tr>
<td>s</td>
<td>blade pitch (m)</td>
</tr>
<tr>
<td>\dot{s}</td>
<td>specific entropy generation rate (J/kg.K.s)</td>
</tr>
<tr>
<td>V</td>
<td>isentropic velocity (m/s)</td>
</tr>
<tr>
<td>Y</td>
<td>stagnation pressure loss coefficient</td>
</tr>
<tr>
<td>\alpha</td>
<td>yaw angle</td>
</tr>
<tr>
<td>\delta</td>
<td>displacement thickness (m)</td>
</tr>
<tr>
<td>\mu</td>
<td>dynamic viscosity (kg/m.s)</td>
</tr>
<tr>
<td>\theta</td>
<td>momentum thickness (m)</td>
</tr>
<tr>
<td>\rho</td>
<td>density (kg/m^3)</td>
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</table>

Subscripts

P       | profile                                        |

Introduction

The designer of the low pressure turbine in a modern aircraft engine is commonly faced with the choice between thick, hollow and thin, solid blades. Brear et al. [1] point out that thin, solid blades can be preferable since they are considerably less expensive to manufacture. However, separation bubbles often occur near the leading edge on the pressure surface of these thin profiles at design conditions. This separation is referred to as the 'pressure surface separation' and, as Brear et al. [1] show, can significantly increase the profile loss.

Several previous studies have observed some interaction between the pressure surface separation and the secondary flow [2-4]. These studies were primarily concerned with the effects of incidence. As such, they were not able to quantify the interaction between the pressure surface separation and the secondary flow because incidence affects both the size of the pressure surface separation and the stage loading. However, one observation of Yamamoto et al. [4] is of particular interest to the present study. In certain cases, reductions in incidence decreased the strength of the secondary flows only up to a particular value of negative incidence. At incidences lower than this the secondary flow became stronger. Yamamoto et al. [4] explained this as resulting from fluid within the pressure surface separation migrating to the tip of their rotor blade, over the casing and into the secondary flow. However, Yamamoto et al. [4] did not investigate this further.

The continued trends towards reduced blade counts and increased stage loading [5] both infer that secondary flows are becoming more significant in limiting the performance of the modern low pressure turbine. This paper is therefore intended to complement Brear et al. [1] by studying the effect of the pressure surface separation on the secondary flow. A family of four blades, named A, B, C and D, are examined (Figure 1). Blade A is also studied in Brear et al. [1]. Blades B, C, and D all have the same suction surface profile and pitch as blade A but different pressure surface profiles. At
midspan and 0° incidence, blade A has a large pressure surface separation which extends from the leading edge to approximately 50%C_X. Blade D is also spanwise uniform, although this profile has been thickened in order to remove the pressure surface separation (Figure 2). Blades B and C are three-dimensional designs. Both are the same as blade A in the central third of span but are blended out to a thicker profile at the endwalls. Blade C has the same profile at the endwall as blade D while blade B is thinner at the endwalls.

Figure 1: Endwall and sectional profiles of blades A, B, C and D

EXPERIMENTAL METHODS

The experiments were performed in a linear cascade without varying the inlet boundary layer or blade incidence. Any variations in the secondary flows of the four blades must therefore only be due to the differing pressure surface geometries. The same cascade and tunnels as those presented in Brear et al. [1] are used. Further details are given in Brear [6]. All experiments were performed at 0° incidence and a Re (based on chord) of 210,000. The cascade consisted of seven blades with parameters given in Table 1.

Blades B, C and D were formed by placing an insert on the pressure surface of blade A. These inserts were fixed in place on the pressure surface and allowed a quick change between cascade geometries during a given set of experiments. At all times, an aerodynamically smooth transition between the inserts and the blade surface was maintained. However, the inserts also prevented measurement of the static pressure on the pressure surface of blade D at midspan and at 1% span on blades B, C and D. The suction surface boundary layer on all four blades was tripped at 60% surface length (73% C_X) with a 0.31 mm stainless steel tube in order to limit the effect of variations in the suction surface flow.

**Table 1: Parameters of blades A, B, C and D**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>chord, C (mm)</td>
<td>148.3</td>
</tr>
<tr>
<td>axial chord, C_X (mm)</td>
<td>126.8</td>
</tr>
<tr>
<td>pitch, s (mm)</td>
<td>103.8</td>
</tr>
<tr>
<td>span, h (mm)</td>
<td>375.0</td>
</tr>
<tr>
<td>inlet flow angle at i=0°, (\alpha_i) (°)</td>
<td>30.4</td>
</tr>
<tr>
<td>design exit flow angle, (\alpha_{2D}) (°)</td>
<td>62.8</td>
</tr>
</tbody>
</table>

Care was taken not to vary overall tunnel conditions throughout the experiments. The inlet boundary layer was measured 200mm (158%C_X) upstream of the leading edge of the central cascade blade. Integral parameters are shown in Table 2. The shape factor agrees closely with that of a zero pressure gradient, turbulent boundary layer.

**Table 2: Integral quantities of the inlet boundary layer 158% upstream of the cascade**

<table>
<thead>
<tr>
<th>(\delta/h)</th>
<th>(\theta/h)</th>
<th>H</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00611</td>
<td>0.00421</td>
<td>1.45</td>
</tr>
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</table>

Area traverses were performed with a conical tip, 90° included angle, five hole probe with 0.3mm forward facing probe holes. The probe tip had an outer diameter of 1.7mm. Pressure measurements were taken with an eight channel, ‘Scanivalve Zoc’ differential pressure transducer of ±10”H_2O range. The probe was calibrated over the range ±30° in both yaw and pitch and applied to the experimental data using the methods presented in [7]. The measurement grid consisted of 39 evenly spaced pitchwise points and 50 radial points. The radial points extended from 2.7% to 50% span and were concentrated nearer the endwall in order to obtain better resolution of the secondary flow region. In Table 3, the loss coefficient \(Y_3\) and exit yaw angle \(\alpha_3\) were obtained from mass averaging from 2.7% to 50% span. The profile loss coefficient \(Y_p\) is a mass average from 30% to 50% span, which is the approximate extent of two dimensional flow in Figure 3.

Blade surface static pressure measurements were measured with a ‘Scanivalve’ differential pressure transducer of ±35mbar range.

Langston & Boyle’s [8] method of flow visualization was used on blades A and D. This method uses a liquid to dissolve and transport dots of ink in the local direction of the skin friction. When the tunnel is turned on, the dots of ink become short streaks that reveal the structure of the surface streamlines. Synthetic oil of wintergreen (methyl salicylate) was used to dissolve the ink dots produced with a ‘Staedler Permacolor 317’ pen drawn on a drafting film backing. This method of flow visualization is, of course, least accurate in regions of zero shear. It is therefore emphasized that the results obtained in regions of separated flow should be interpreted with great care. It is also noted that the suction surface trip wire was removed for these experiments only.
NUMERICAL PREDICTIONS

The numerical predictions were performed using 'Multip99', which solves a thin shear layer form of the Reynolds averaged, Navier-Stokes equations supplemented by a mixing length turbulence model [9]. The predictions were performed at a cascade exit Mach number of 0.3 in order to accelerate convergence.

Other than their differing pressure surface geometries, all four blades had the same basic computational mesh. This mesh extended from 33%\(C_X\) upstream of the blade-row to 33%\(C_X\) downstream and had 109 axial, 49 circumferential and 73 radial points. Preliminary numerical predictions showed that this axial extent of the mesh was effectively outside the potential field of the bladerow and that the quality of the solution was not compromised. A plane of symmetry was enforced at midspan with an inviscid boundary condition. Several numerical predictions were performed on coarser meshes and, in keeping with Hildebrant & Fottner [10], mesh independent predictions of the secondary flow were never observed. However, the mesh used in the present study appears to have produced results of sufficient accuracy for the arguments put forward.

Turbulent onset on the pressure surface was fixed at 25%\(C_X\), which was the approximate location of turbulent onset in experiments (see Brear et al. [1]). On blade A, this gave a reattachment location of 59%\(C_X\) compared with the experimental result of 50%\(C_X\). It is also noted that specifying transition onset at the pressure surface leading edge completely removed the pressure surface separation. Transition on the suction surface was specified at the same location as the boundary layer trip in experiments (73%\(C_X\)). The transition locations were then fixed for all subsequent predictions on the four blades. The endwall boundary layer was considered turbulent throughout the computational domain. Since the measurement of the inlet boundary layer was performed 158%\(C_X\) upstream of the blade-row, the boundary layer at the inlet to the computational domain was ‘grown’ from the measured inlet boundary layer using zero pressure gradient, turbulent boundary layer relations. Thus, the integral quantities given in Table 2 were increased by 20% when used as inlet boundary conditions for the numerical predictions.

RESULTS AND DISCUSSION

Area traverses

Pitchwise mass averaged experimental and computational results for the area traverses are shown in Figure 3. These show that from approximately 25% to 50% span, the exit yaw angle is effectively the same on all four blades and matches the design intent closely. Of note are the spanwise uniform profiles A and D. Even though blade D is considerably thicker than blade A, both blades have the same exit yaw angle and hence loading at midspan. It therefore follows that any spanwise uniform profile made from any section of the four blades studied will have the same loading since the profiles of blades B and C fall between those of blades A and D. The observed variations from 0% to 25% span are therefore not due to an intentional spanwise variation of loading. In this sense, this work differs from most three dimensional design studies where spanwise variations in loading are deliberately introduced. Thus, the present work is not a typical three-dimensional design study even though blades B and C are three-dimensional designs.

Table 3 shows that the variations in pressure surface geometry have not changed the profile loss significantly. Therefore, the added thickness of blade D serves little aerodynamic purpose around midspan at 0° incidence in addition to the blade being considerably more costly to manufacture than blade A. This supports the findings of Brear et al. [1] and highlights the motivation behind the use of thin, solid blades: if aerodynamic requirements do not dictate the use of a thick blade, a thin blade is preferable because they are less costly.

Figure 3: Pitchwise mass averaged a) stagnation pressure loss (experiment), b) stagnation pressure loss (predicted), c) exit yaw angle (experiment) and d) exit yaw angle (predicted) for blades A, B, C and D at 125%\(C_X\)

The importance of achieving realistic predicted transition onset locations should be emphasized. Specifying transition at the leading
edge on the suction surface resulted in profile losses of approximately 0.04. Other studies of turbine profiles that compared numerical predictions with experiments showed similar behavior [10,11]. Specifying transition at the leading edge on the suction surface is clearly physically unrealistic as the acceleration from the leading edge to peak suction on low pressure turbine blades should keep the boundary layer laminar [12]. The considerable extent of laminar flow on low pressure turbine blades therefore necessitates the use of realistic transition modeling if reasonable predictions of profile loss, and hence efficiency, are sought.

Figure 4 shows that the structure of the secondary flow at 125%C_x is the same as that reported in other studies of low pressure turbine blades in linear cascade [2,3,10,11]. Furthermore, Figure 3 and Figure 4 show that there is considerable variation in stagnation pressure loss and exit yaw angle in the secondary flow region (from approximately 0% to 20% span). A trend is clearly apparent in Figure 3: the peaks in stagnation pressure loss and underturning reduce in magnitude and move closer to the endwall as the blade thickness is increased. Experiments show that blade D, which has the lowest loss, achieved 10.2% lower mass averaged stagnation pressure loss (Table 3) and approximately 2° less peak underturning than blade A. Numerical predictions behave similarly, although the predicted stagnation pressure loss in the secondary flow region is significantly less than that found in experiment. It is emphasized, however, that even though the absolute value of the mass averaged stagnation pressure loss is significantly underpredicted, Table 3 shows that the relative variation between the results for blades A and D (0.0047) is the same as that found in experiment.

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</tr>
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<tbody>
<tr>
<td>A</td>
<td>62.3</td>
<td>62.4</td>
<td>0.0450</td>
<td>0.0347</td>
<td>0.0282</td>
<td>0.0250</td>
<td></td>
<td></td>
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<tr>
<td>B</td>
<td>62.4</td>
<td>62.6</td>
<td>0.0418</td>
<td>0.0334</td>
<td>0.0286</td>
<td>0.0248</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>62.5</td>
<td>62.8</td>
<td>0.0404</td>
<td>0.0312</td>
<td>0.0282</td>
<td>0.0245</td>
<td></td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>62.5</td>
<td>62.7</td>
<td>0.0403</td>
<td>0.0300</td>
<td>0.0281</td>
<td>0.0244</td>
<td></td>
<td></td>
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</tbody>
</table>

Table 3: Mass averaged results of blade A, B, C and D at 125%C_x from experiment and numerical predictions

Entropy generation rates

The entropy generation rate throughout the entire computational domain was calculated for each of the four blades. The entropy generation rate per unit volume (ρs) was calculated from the rate of shear work performed by the fluid:

\[
\rho s = \frac{1}{T} \left( \mu + \mu_t \right) \left[ \frac{\partial v_0}{\partial x} + \frac{1}{r} \frac{\partial v_x}{\partial \theta} \right]^2 + \frac{1}{T} \left[ \frac{\partial v_0}{\partial x} + \frac{\partial v_x}{\partial \theta} \right]^2 + \frac{1}{T} \left[ \frac{\partial v_0}{\partial \theta} + \frac{\partial v_x}{\partial x} \right]^2.
\]

(1)

Figure 3 shows that the secondary flow region extends from approximately 0% to 20% span. This region was divided into a number of volumes that are defined in Figure 5. The pitchwise extent of the ‘pressure surface’ (PS) region was chosen such that this volume would contain the pressure surface separation on blades A and B. Similarly, the radial extent of the ‘endwall’ (EW) region contained the inlet boundary layer. The circumferential extent of the ‘suction surface’ (SS) and ‘core’ regions were less important as the core region contained nearly isentropic flow. The remaining region is termed the ‘downstream’ (DS) region. Since the flow is steady and adiabatic, the total entropy generated in each of these domains can be expressed as an exit stagnation pressure loss coefficient:

\[
Y = \frac{2T_{01}}{\rho s h V^2} \cos \alpha \int p dV.
\]

(2)
Results of this analysis are shown in Table 4. The total loss reduction from blade A to blade C is 0.0044, which is similar to the relative variations in both the experimental and predicted area averaged loss results in Table 3 (0.0047). However, it is also noted that this method of analysis predicts that blade D has a slightly larger loss than blade C. This is not in keeping with the results presented in Table 3 and is thought to expose the limitations of the numerical modeling in discriminating between the expected small differences in loss for these two designs.

<table>
<thead>
<tr>
<th>Region (EW, SS, PS, core, DS, total)</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0.0013</td>
</tr>
<tr>
<td>B</td>
<td>0.0011</td>
</tr>
<tr>
<td>C</td>
<td>0.0009</td>
</tr>
<tr>
<td>D</td>
<td>0.0009</td>
</tr>
</tbody>
</table>

Table 4: Predicted stagnation pressure losses in each region of the secondary flow on blades A, B, C and D

In order to reveal the mechanisms that cause these variations in entropy generation rate, a dissipation coefficient for the flow near the endwall was calculated:

\[ C_d = \frac{T_{01}}{\rho V^2} \int_0^{0.1b} \rho d\theta. \]  

where the spanwise extent of the integration was chosen to include the entire inlet boundary layer. It is noted that this quantity is normalized by the exit velocity of the cascade, and is therefore an indicator of the absolute entropy generation rate from 0% to 10% span at a given (x,θ) location.

Figure 6 shows contours of this dissipation coefficient on blades A and D. As discussed earlier, transition onset on the pressure surface was specified to occur at 25%C_x. Figure 6 shows that there is a clear peak in dissipation coefficient at this axial location on blade A. As with the model presented in Brear et al. [1], numerical predictions therefore show that the pressure surface separation dissipates mean flow energy primarily through turbulent shear. The added thickness of blade D has removed the pressure surface separation, thereby removing a loss production mechanism. The loss produced in the PS region in Table 4 correspondingly decreases. The dissipation at mid-pitch is also noticeably different on blades A and D. On blade A, the region of high dissipation near the pressure surface extends across the blade passage and towards the suction surface. In contrast to this, the mid-pitch flow on blade D is characterised by lower levels of dissipation. Given that the entropy generation rate (equation 1) is defined as a function of shear strain rates only, the variations shown in Figure 6 must arise from variations in shear strain rates. Surface flow visualizations presented in the next section suggest the mechanisms by which this occurs.

Figure 6: Predicted dissipation coefficient for 0% to 10% span on blades A and D

Surface flow visualisation

The pressure surface flow is strongly modified by the different blade designs (Figure 7, Figure 8). Experiments and numerical predictions show that the pressure surface separation on blade A reattaches at 50%C_x and 59%C_x respectively at midspan. In comparison to this, blade D has no pressure surface separation and most of the pressure surface flow follows approximately straight lines. Blades B and C, because of their increasing thickness, tend to suppress the pressure surface separation near the endwall.

Blade A also has the most circumferential endwall motion (Figure 9, Figure 10). Although less obvious in experiments, the results for blade A show a region of the endwall flow near the pressure surface with negative axial velocity. The passage vortex separation line also impinges onto the suction surface slightly further downstream as the blade thickness increases. These trends suggest that the passage vortex becomes weaker from blades A through to D.

There are also significant variations in the suction surface flow (Figure 11). As in Figure 9 and Figure 10, the passage vortex separation line intersects with the suction surface earliest on blade A. This line reaches the trailing edge at approximately 15% and 10% span on blades A and D respectively. The difference in the area of...
surface flow bounded by these separation lines on the two blades is considerable.

Figure 7: Experimental flow visualization on the pressure surfaces of blades A and D

Figure 8: Predicted flow on the pressure surfaces of blades A, B, C and D

Figure 9: Experimental flow visualization on the endwalls of blades A and D

The structure of the interaction between the pressure surface separation and the passage vortex is now apparent. On blade D, the surface flow is typical of many turbines [2,3] and suggests that the inlet boundary layer and some of the attached pressure surface flow roll up into the passage vortex. In contrast to this, blade A has a more complex flow structure. Figure 12 shows streamlines that originate away from the endwall impinging onto the pressure surface near the reattachment point of the pressure surface separation. As a part of the recirculating flow, these streamlines then travel along the pressure surface and upstream whilst migrating radially towards the endwall. Once they have reached the endwall, these streamlines cross towards the suction surface and roll up into the passage vortex. This overall process gives rise to the surface flow visualizations presented in Figures 7-11 and shows that the flow within the passage vortex is composed of fluid from both the inlet boundary layer and the pressure surface separation.

Figure 10: Predicted flow on the endwalls of blades A, B, C and D

Figure 11: Experimental flow visualization on the suction surfaces of blades A and D

The trend of reduced secondary flow strength with increased blade thickness is also in agreement with the area traverses presented earlier (Figure 3). Furthermore, the surface flow visualisations also give a qualitative understanding of the physical mechanisms responsible for the variations in loss production. For example, the strongly circumferential endwall motion observed on blade A in Figure 9 and Figure 10 suggests that, relative to the other blades, the endwall boundary layers will be more highly skewed. As Figure 6 showed, the entropy generation rate over the endwall on blade A is correspondingly increased. Table 4 showed that the most significant loss reduction occurs in the suction surface region defined in Figure 5. Figure 11 also suggests a reduction in boundary layer skew from blades A through to D. Therefore, the secondary flow strength and the loss produced within the bladerow appear to be directly related.
Figure 12: Predicted streamlines close to the endwall on blade A

Static pressure distributions

Figure 13 shows experimental and predicted isentropic velocity distributions of the four blades. At midspan, the isentropic velocities along the suction surface foreblade (0% to 50% C_X) become higher as the blade thickness increases. This is thought to be due to blockage effects. As discussed in Brear et al. [1], the location of the local minimum in the isentropic velocity on the pressure surface is close to the reattachment point of the pressure surface separation at midspan. Comparison of the local minima in Figure 13 with the reattachment points in Figure 7 and Figure 8 confirm this both experimentally and in the numerical predictions. This also suggests that the overprediction of the length of the pressure surface separation is the main cause of the different experimental and predicted isentropic velocities along the pressure surface.

All four blades have similar loading at midspan (Table 5). This is expected as the blades have the same pitchwise mass averaged yaw angle at midspan (Figure 3). Table 5 shows that predictions of both the foreblade loading and overall loading at 1% span increase with a decrease in secondary flow strength. As a result, the blade loading cannot be the determinant of the observed variations in secondary flow because all four blades have nominally the same loading at midspan and the trend in the loading at the endwalls is of the opposite sense to that required for secondary flows to strengthen.

<table>
<thead>
<tr>
<th></th>
<th>C_L_fb</th>
<th>C_L_ov</th>
</tr>
</thead>
<tbody>
<tr>
<td>expt</td>
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<td>expt</td>
</tr>
<tr>
<td>A</td>
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<td>0.35</td>
</tr>
<tr>
<td>B</td>
<td>0.35</td>
<td>0.34</td>
</tr>
<tr>
<td>C</td>
<td>0.37</td>
<td>0.34</td>
</tr>
<tr>
<td>D</td>
<td>n/a</td>
<td>0.34</td>
</tr>
</tbody>
</table>

Table 5: Loading coefficients at midspan and 1% span from experiment and numerical predictions ('fb' is the foreblade from 0% to 50% C_X, ‘ov’ is the overall blade from 0% to 100% C_X)

A number of investigators have found that the overall development of the secondary flow in a turbine is particularly sensitive to the flow around the intersection of the foreblade and the endwall [11,14]. This can be explained using simple dynamics. Because the boundary layer momentum on the endwall and pressure surface is at its lowest around the foreblade, the flow is most easily forced towards the suction surface of the adjacent blade by a given blade-to-blade pressure gradient. For a given blade loading, it is therefore best to impose the largest blade-to-blade pressure gradient in regions that are most resistant to cross-passage transport i.e. regions where the flow has highest momentum. This is commonly referred to as ‘aft-loading’ since, for a given amount of turning, profiles designed using this argument will tend to have as much loading as possible in the aft region of the blade.
impingement of the secondary flow from across the blade passage [11,14]. Therefore, the causal link between endwall loading and secondary flow strength is not one way: the endwall loading drives the secondary flow, but the secondary flow also modifies the endwall loading. Using endwall isentropic velocities to explain some observed variation in secondary flow may result in a circular argument.

This difficulty can be avoided by using the isentropic velocity distribution of the two-dimensional blades A and D at midspan (Figure 13a). Because of its added thickness, blade D has higher isentropic velocities over most of its surface. Near the endwall, blade D still represents the greater blockage to the incoming flow. The momentum of the fluid through the bladerow should therefore be higher on blade D. Fluid passing blade D is therefore less likely to be turned toward the suction surface and the secondary flow should be weaker.

Figure 7 and Figure 8 showed clearly that the progressive increase in blade thickness from blades A through to D is matched by a decrease in the size in the pressure surface separation. This is expected since the blockage added onto the pressure surface should tend to reduce the intensity of the deceleration around the pressure surface foreblade and hence encourage the pressure surface boundary layer to remain attached. The existence or suppression of the pressure surface separation is therefore inseparable from the choice of isentropic surface velocity. Because the argument put forward makes no reference to whether any of the boundary layers are separated or attached, the pressure surface separation should then be viewed as part of an overall low momentum flow near the endwall. The results for blades B and C are also easily explained using this argument. Because they represent intermediate blockage of blades A and D, the momentum of the fluid near the endwall on blades B and C will lie between that on blades A and D. The observed trend in secondary flow over the four blades then follows.

CONCLUSIONS

The interaction of the pressure surface separation with the secondary flow can significantly affect the development of the secondary flow and the loss that it creates. This has been shown in a series of experiments performed on a family of four low pressure turbine blades in linear cascade. Numerical predictions of these four blades show the same trend as that found experimentally.

A mechanism for this interaction has been suggested. This employs simple dynamics and is similar to the aft-loading argument commonly used in modern turbine design. The foundation of this argument rests on the momentum of the flow near the endwall. In particular, the existence of a pressure surface separation near the endwall is an indicator that the flow in this region has low momentum and will therefore respond to the imposed blade-to-blade pressure gradient by migrating strongly across the endwall. The strength of the secondary flow and the skewing of the blade surface boundary layers are therefore increased. Since the entropy generation rate is a function of the varying shear strain rates inside these skewed boundary layers, the loss production throughout the secondary flow region varies with the strength of the secondary flow.

Using this argument, the secondary flow strength and loss production can be reduced by raising the momentum of the fluid near the endwall. This has been shown by progressively thickening the blade on the pressure surface. As part of an overall increase in the momentum of the fluid near the endwall, additional blade thickness has the inevitable effect of reducing the size of the pressure surface separation, thereby suppressing the interaction between the pressure surface separation and the secondary flow.

ACKNOWLEDGMENTS

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REFERENCES