Experimental investigation of tip clearance effects on flow field in an annular turbine rotor cascade

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The present paper reports results of an experimental investigation on the effects of tip clearance in a high deflection annular turbine rotor cascade. The measurements are carried out at the inlet, in the blade passage and at the exit of the cascade using a five-hole probe and a seven-hole probe. The results indicate that the incoming flow to the rotor cascade is uniform and the incidence is about ±2°. When the clearance is increased, the size and the magnitude of the tip clearance loss region increase. The flow near the hub experiences overturning, whereas near the casing the flow substantially underturns due to tip clearance. The underturning increases with tip clearance. Secondary and tip clearance losses steeply increase downstream of the blade trailing edge indicating losses due to the wake and clearance flow mixing with the main flow.

Principal internal losses occurring in a turbine are due to secondary flows, endwall flows and tip clearance flows. Tip clearance flow is due to the static pressure difference across the blade surfaces, whereas, endwall flow is due to the presence of boundary layer on the endwall surfaces. There are two distinct and equally important aspects of tip clearance flows. First, there is reduction in blade force, therefore the work done. This occurs because the leakage flow passes over the blade tip essentially without being turned. As a consequence of viscous effects in the tip clearance, entropy is also produced. The second major aspect is the mixing of the flow that passes through the tip clearance with main flow. Since the lift coefficient of a turbine rotor blade is much larger than that of a compressor rotor blade, the vortex diameter is likely to be larger, thus influencing the flow over large region near the tip. The tip clearance losses within the gap are relatively very less compared to overall tip clearance losses due to interaction with secondary and endwall flows. Analysis and understanding of tip clearance and secondary flows would be certainly helpful in improving the efficiency and aerodynamic performance of turbines.

Research on secondary and tip clearance flows in axial turbines is receiving considerable attention, as evidenced by a lecture series devoted to this topic. Morphis and Binod obtained tip static pressure distribution on a turbine rotor blade at various values of tip clearances. Yaras and Sjolander, Sjolander and Amrud studied the influence of tip clearance on the downstream flow filed and the blade loading of a linear turbine rotor cascade. Moore and Tilton studied the flow in the tip clearance space of linear turbine rotor cascade and derived a flow model using the concepts of potential flow and theory supplemented by flow mixing and boundary layer effects. Dishart and Moore in their investigations in a linear turbine rotor cascade have brought the phenomena and salient features associated with the loss generation, distribution and mechanism of tip clearance flows. Sjolander and Cao found multiple vortices on the blade tip, which are probably the reason for the burnout that some times occurs on turbine rotor blade tips near the pressure surface. Govardhan et al. studied tip clearance flows within and exit of a large deflection linear rotor cascade. They found a strong horse-shoe vortex formed for the zero clearance, which disappears for 3% clearance indicating that the pressure forces have dominating influence than the viscous forces for large clearance. The mechanism of tip clearance flow is far from understood. The data available are not sufficient to test calculation methods or loss prediction schemes. The main difficulty in obtaining detailed measurements within the typical small dimensions of the tip gaps found in actual machines or created in the
Experimental Procedure

The measurements were carried out in an annular cascade tunnel available in Thermal Turbomachines Laboratory, Department of Mechanical Engineering, Indian Institute of Technology, Madras. A schematic of facility is shown in Fig. 1. The hub and the casing diameters of the test section are 400 mm and 600 mm respectively. The tunnel consists of 25 numbers of inlet swirl vanes and outlet straightener vanes. The test cascade is mounted in between these two vanes. Other details of the tunnel are available elsewhere. The cascade blade profile along with the measurement stations is shown in Fig. 2 and the major details are given below.

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chord, Ch</td>
<td>95 mm</td>
</tr>
<tr>
<td>Axial chord, e</td>
<td>92.75 mm</td>
</tr>
<tr>
<td>Span, h</td>
<td>100 mm</td>
</tr>
<tr>
<td>Aspect ratio, h/Ch</td>
<td>1.05</td>
</tr>
<tr>
<td>Inlet blade angle, $\beta_{1i}$</td>
<td>57.5°</td>
</tr>
<tr>
<td>Outlet blade angle, $\beta_{1o}$</td>
<td>-62.5°</td>
</tr>
<tr>
<td>Camber angle, C</td>
<td>120°</td>
</tr>
<tr>
<td>Stager angle, $\gamma$</td>
<td>12.5°</td>
</tr>
<tr>
<td>Leading edge radius/ Chord</td>
<td>0.25</td>
</tr>
<tr>
<td>Trailing edge thickness/ Chord</td>
<td>0.05</td>
</tr>
<tr>
<td>Maximum thickness/ Chord</td>
<td>0.38</td>
</tr>
<tr>
<td>Position of max. thickness/ Chord</td>
<td>0.42</td>
</tr>
<tr>
<td>No. of blades, Z</td>
<td>24</td>
</tr>
</tbody>
</table>

All angles are taken with reference to axial direction.

A five-hole probe at upstream and a seven-hole probe within the rotor blade passage and downstream of the cascade are used for flow measurements. Both probes have a head diameter of 2.6 mm only. Both probes are used in non-nulling mode of operation following the method of Treaster and Yocum for the five-hole probe and the method of Venkateswara Babu et al. for the seven-hole probe. The seven-hole probe is calibrated in the high angle range of ± 5° in both yaw and pitch planes. Both probes suffer many measurement errors arising from various sources. The following magnitudes of errors are estimated in the measurements obtained by the five-hole probe.

- Total and static pressures: ± 1% of dynamic head
- Flow and radial angles: ± 1°

Same magnitude of error may be expected in the measurements by the seven-hole probe. In the
boundary layer, wake and vortex regions, where the gradients in velocity and pressure are large, the magnitude of error increase substantially for both probes. The details of the measurements are given in Table 1.

**Results and Discussion**

**Inlet flow**

Fig. 3 shows the non-dimensional total pressure at inlet to the inlet guide vanes. The boundary layer on the hub is thicker ($\delta_v/h = 0.28$) than that on the casing ($\delta_v/h = 0.12$), due to the fact that flow coming from the settling chamber travels a longer distance on the hub compared to that of the flow on casing.

**Rotor blade inlet flow**

The flow that enters the IGV axially is deflected by about 57.5°. Fig. 4 shows the distribution of the total pressure coefficient, static pressure coefficient and flow angle at the inlet of the rotor blades ($X = -0.34$). The $X$-axis represents the non-dimensional tangential distance. $Y = 0$ represents the center of the measurement region. The measurements are taken at

<table>
<thead>
<tr>
<th>Axial Station</th>
<th>Grid Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>$X$</td>
<td>(Span-wise direction $\times$ Tangential direction)</td>
</tr>
<tr>
<td>-0.34</td>
<td>700 grid points ($25 \times 28$)</td>
</tr>
<tr>
<td>0.20</td>
<td>168 grid points ($21 \times 8$)</td>
</tr>
<tr>
<td>0.40</td>
<td>147 grid points ($21 \times 7$)</td>
</tr>
<tr>
<td>0.70</td>
<td>147 grid points ($21 \times 7$)</td>
</tr>
<tr>
<td>0.88</td>
<td>168 grid points ($21 \times 8$)</td>
</tr>
<tr>
<td>1.04</td>
<td>693 grid points ($21 \times 33$)</td>
</tr>
<tr>
<td>1.29</td>
<td>693 grid points ($21 \times 33$)</td>
</tr>
<tr>
<td>1.53</td>
<td>693 grid points ($21 \times 33$)</td>
</tr>
</tbody>
</table>

$\tau/Ch = 0.00$. The total pressure is found to be uniform in the tangential direction, indicating complete mixing of the exit flow from the IGV (Fig. 4a). The total pressure losses are very low except near the hub and the casing. The static pressure increases towards the casing to account for radial equilibrium of the flow (Fig. 4b). The maximum variation in flow angle in the regions away from the boundary layer is $\pm 2°$ from the design value of 57.5°, so the incidence on to the rotor blades is about $\pm 2°$ (Fig. 4c). For the turbine blades, which have thick leading edges, the effect of this small incidence on the flow is negligible.

**Rotor blade passage flow**

Contours of total pressure coefficient in the blade passage at $X = 0.4$ and 0.88 are presented in Figs 5 and 6 for the three values of tip clearances, viz., $\tau/Ch$. 

![Fig. 3—Distribution of total pressure at inlet](image-url)

![Fig. 4—Distribution of total and static pressures and flow angle at inlet of rotor blade](image-url)
Center of tip clearance loss region

Fig. 5—Distribution of total pressure in the passage of rotor blade (X=0.40)

![Diagram showing distribution of total pressure](image)

Fig. 6—Distribution of total pressure in the passage of rotor blade (X=0.88)

![Diagram showing distribution of total pressure](image)

= 0.00, 0.01 and 0.03. Suction and pressure surfaces are marked as SS and PS. These contours clearly show the effect of tip clearance, i.e., decrease in total pressure in the clearance region and increased extent of low-pressure region as the clearance increases. The flow over the tip rolls into a vortex, known as tip leakage vortex. The tip leakage vortex causes large losses in the total pressure. Also the extent of low-pressure region increases with axial distance (Figs 5 and 6). At X = 0.4 (Fig. 5), the boundary layer on the suction surface is found to be thicker than that on the pressure surface due to the cross channel deflection of the boundary layer fluid from the pressure surface to the suction surface. At X = 0.88 (Fig. 6), the tip clearance loss region near the casing is slightly pushed away from the suction surface. The boundary layer on the casing has become thin. This suggests movement of the low energy fluid of the boundary
layer into this loss region. The losses are more concentrated and are higher in their magnitude in the tip clearance region. The centre of the tip clearance loss region at \( X = 0.88 \) for \( \tau/Ch = 0.003 \) is away from the casing and towards the suction surface compared to that for \( \tau/Ch = 0.01 \). This indicates a tip leakage vortex of larger diameter and of higher strength. For the clearance of \( \tau/Ch = 0.00 \), no such loss region is clearly seen. However, passage vortex near the hub and the casing can be seen at zero clearance as well as non-zero clearances.

**Rotor blade exit flow**

Figs 7 and 8 show the contours of total pressure coefficient at \( X = 1.04 \) and 1.53 for the values of \( \tau/Ch = 0.00, 0.01 \) and 0.03. As the trailing edge thickness for the present cascade is thicker (about 5% of the blade chord), the wakes are broader and the losses in the wake are considerable (Fig. 7). Although the blade trailing edge is radial, the wake is inclined towards suction surface due to cross channel pressure gradient. The wake center-lines are shown by dashed straight-lines. However near the hub and the casing, the wakes
interact with passage and tip leakage vortices and the wake centres are no longer straight. The losses due to the hub and the casing passage vortices are clearly seen. The loss region due to hub passage vortex is found to be more and occupies about 50% of the blade passage where as the loss region due to the casing passage vortex is found to occupy only 33% of the passage. The total pressure loss region near the hub is much broader in the tangential direction than near the casing. The loss region due to the hub vortex remains unaltered for all values of tip clearances. For \( \tau/Ch = 0.01 \) (Fig. 7b), the tip clearance vortex is located nearer to the casing. When the clearance is further increased to \( \tau/Ch = 0.03 \) (Fig. 7c), the size as well as magnitude of the tip clearance loss region increases. At \( X = 1.53 \) (Fig. 8), the wake mix with the main flow. At \( X = 1.53 \), the total pressure loss is much higher than that obtained at \( X = 1.04 \). The loss increases substantially with increase in tip clearance (Figs 8b and 8c).

**Pitch-wise averaged flow angle**

Fig. 9 shows the spanwise variation of the pitchwise averaged flow angle from \( X = 0.20 \) to 1.53 for

![Fig. 9](image)

**Fig. 9**—Span-wise variation of pitch-wise averaged flow angle at different axial stations

![Fig. 10](image)

**Fig. 10**—Axial variation of total, gross secondary, net secondary and clearance losses
\[ \tau/Ch = 0.00, 0.01 \text{ and } 0.03. \] At all axial stations, the variation in the flow angle variation is small near the hub but substantially increases near the casing with increase in tip clearance. For \( \tau/Ch = 0.01 \) and 0.03, large variation in the flow angle is seen near the casing. Secondary flows cause flow overturning near the hub and the casing. Away from the hub and the casing, flow overturns due to secondary flows. The effect of tip leakage flows is to cause flow overturning. The magnitude of overturning near the casing increases with increase in tip clearance. For \( \tau/Ch = 0.01 \), tip clearance effects are pre dominant, as overturning associated with secondary flows in not seen at all for \( X=0.70 \) and beyond. From the total pressure contours and averaged flow angle distributions, it may be inferred that the tip leakage vortex is formed between \( X=0.20 \) and 0.40 and becomes stronger between \( X=0.40 \) and 0.70. The position of maximum overturning does not coincide with the position of centre of loss region (Figs 7 and 8). This means that the point of lowest total pressure may not be accompanied by the largest flow angle deviation.

**Loss coefficients**

Fig. 10 shows variation of total loss coefficient, \( (Y_T) \), gross secondary loss coefficient, \( (Y_{SC}) \), net secondary loss coefficient, \( (Y_S) \) and tip clearance loss coefficient, \( (Y_{CL}) \) with axial distance for \( \tau/Ch = 0.00, 0.01 \) and 0.03.

They are defined as follows:

\[
Y_T = \sqrt{\frac{\int_0^h (P_{net} - P_0)}{0.5 \rho \overline{C}_2^2}}
\]

\[
Y_{SC} = Y_T - Y_S
\]

\[
Y_S = \text{Profile loss coefficient obtained from 2-D cascade tests} \times 4 \% \text{ of } 0.5 \rho \overline{C}_2^2 \text{ downstream of the cascade}
\]

\[
Y_{NL} = Y_{NL}, Y_{IN}
\]

\[
Y_{NL} = \text{Inlet boundary layer loss coefficient} = 6 \% \text{ of } 0.5 \rho \overline{C}_2^2
\]

\[
Y_{CL} = Y_S - Y_{CL},\text{at} \tau/Ch = 0.00
\]

Fig. 10 shows that secondary and tip clearance losses increase steeply downstream of the blade trailing edge. However, it must be remembered that the loss coefficient in the passage does not include profile losses, as measurements could not be taken close to the blade surfaces. Even then, it may be inferred that tip clearance starts between \( X=0.20 \) and 0.40, as there is a large increase in the magnitude of loss coefficient from \( X=0.20 \) to 0.40. Also tip clearance vortex is observed at \( X=0.40 \) as seen from the contours of total pressure coefficient (Fig. 5). At \( X=1.04 \), there is again a large increase in the total secondary and tip clearance losses. It indicates that the tip clearance effect is less in the passage region but increases downstream of the cascade. This is due to the interaction of tip clearance flow with secondary and endwall flows.

**Conclusions**

From the present investigations the following conclusions are drawn:

1. The total pressure is found to be uniform in the tangential direction, indicating complete mixing of the IGV exit flow; (2) At \( X=1.04 \), the loss region due to hub passage vortex is found to be more and occupies about 50% of the blade passage whereas as the loss region due to the casing passage vortex is found to occupy only 33% of the passage; (3) At \( X=1.53 \), wake mixes with the main flow and losses are found to be higher than those obtained at \( X=1.04 \); (4) When the clearance is increased, the size and the magnitude of the tip clearance loss region increase; (5) The flow near the hub experiences overturning followed by overturning, whereas near the casing, the flow substantially underturns due to tip clearance. The overturning increases with tip clearance; and (6) Secondary and tip clearance losses steeply increase downstream of the blade trailing edge indicating additional losses due to the wake and clearance flow mixing with the main flow.

**Nomenclature**

- \( Ch \) = chord (m)
- \( e \) = axial chord (m)
- \( C \) = velocity (m/s)
- \( C_v \) = axial velocity (m/s)
- \( b \) = blade span (m)
- \( P_r \) = total pressure (N/m²)
- \( P_{ch} \) = settling chamber pressure (N/m²)
- \( s \) = spacing (m)
- \( x, y, z \) = distances in axial, tangential and spanwise directions (m)
- \( X \) = non-dimensional axial distance from blade leading edge, \( x/e \)
- \( Y \) = non-dimensional tangential distance, \( y/s \)
- \( Y_{CL} \) = tip clearance loss coefficient
References
1. VENKATESWARA BABU et al.: ANNULAR TURBINE ROTOR CASCADE

References