INFLUENCE OF TIP CLEARANCE ON THE INTER BLADE AND EXIT FLOW FIELD OF A TURBINE ROTOR CASCADE

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ABSTRACT
A detailed study of flow through the blade passage and downstream of a linear turbine cascade was carried out for four cases of tip clearance including zero clearance. Apart from inlet traverse, a total of eight stations were chosen for inter-blade flow traversing between 5% and 95% of axial chord from leading edge. Downstream flow surveys were made at distances of 106% of axial chord from the blade leading edge. Pitchwise and spanwise traverses were conducted for each tip clearance at these stations using a small five hole probe. Provision was also made for the measurement of static pressure distribution on the suction and pressure surfaces and also on the blade tip surface when clearance is present.

At about 40% of axial chord from the leading edge, the presence of clearance vortex is identified inside the passage. The growth of the clearance vortex in size, its movement towards the suction surface and its increase in strength with the gap size were observed beyond 55% of axial chord till the trailing edge region. The rate of growth of the losses in the endwall region increased with clearance. Horse shoe vortex was not observed for the highest clearance. The overall losses increase rapidly with clearance in the rear half of the blade.

INTRODUCTION
Many turbomachine impellers are not shrouded and the leakage flow through the tip clearance of blade is an unavoidable factor which deteriorates the performance. Denton and Cumpsty (1987) mentioned about two distinct and equally important aspects to the tip clearance flows. First, there is a reduction in the 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 gap entropy is also produced. The second major aspect is the mixing of the flow that passes through the tip clearance gap with that which passes between the blades. The effect of tip leakage is found to be more severe in turbines as compared to compressors. Since the lift coefficient of a turbine blade is much larger than that of a compressor, the vortex core diameter is likely to be larger, thus influencing the flow over a large region near the tip. Additional losses are noticed when the endwalls of the cascade move with respect to the blades. A strong accumulation of losses at the blade end on the suction side of the blades was observed by many investigators.

Bindon (1987, a, b, c) conducted experiments in a linear cascade with tip clearance to extend the understanding of the flow physics. He measured both static pressure flow field and the boundary layers inside the tip gap and on the end wall. He aided his experiments with smoke flow visualisation. Bindon (1989) from his later experiments concluded that, of the tip clearance flow losses generated until the trailing...
edge, about 40% occurred within the gap, attributed mainly to the separation bubbles formed on the blade tip. Yaras and Sjolander (1992a) have found that the gap loss is relatively small compared to the overall tip clearance loss. Existence of a low pressure zone near the suction side of the blade and the flow separation inside the tip gap are noted for turbines by Graham (1988), Wadia and Booth (1982), Bindon (1987 a,c), Sjolander and Amrud (1987). More recently Bindon and Morphis (1992) have reduced the losses generated within tip gap by radiusing the blade side tip corner, thus removing the separation bubble. Yamamoto (1988a) carried out investigations to understand the mechanism of clearance flow in the tip gap by making direct flow measurements within the gap in a linear turbine cascade at design incidence and discussed the interaction of the clearance flow with the passage vortex. Moore and Tilton (1988) studied the flow in the tip clearance space of linear turbine cascade and deducted a flow model using the concepts of potential flow theory supplemented by flow mixing and boundary layer effects. Dishart and Moore (1990) in their investigations in a linear turbine cascade have brought the phenomena and salient features associated with the loss generation, distribution and mechanism of tip clearance flows.

Yaras and Sjolander (1992b) and Yaras et al (1992) investigated the effects of simulated relative motion on tip leakage in a linear turbine cascade. A substantial decrease in the gap flow rate was observed as a result of the relative wall motion. Reduction in the strength of the tip clearance vortex, enhancement of the passage vortex due to scraping effect of the blades was found due to wall motion.

Yaras and Sjolander (1990) investigated the development of tip leakage flows downstream of a planar turbine cascade vorticity field. Their details provided an insight to understand the influence of tip clearance on the downstream flow field. Sjolander and Amrud (1987) have investigated the effects of tip clearance on blade leading in planar cascade and found an increase of lift near the tip before the final decline takes place right at the tip. Hah (1986) has applied three dimensional viscous calculations to blade passage flows in which tip clearance was present.

Despite the progress that has been made in the mechanism of tip leakage flow, there is a lot of misunderstanding. The data is not really sufficient to test calculation methods or loss prediction schemes. More over most data has been established towards quantifying losses rather than establishing the basic nature of the flow structure that occurs between inlet and outlet of the cascade. In addition, there is a need to get detailed experimental data on the mechanism of secondary loss especially tip leakage loss data through the blade cascade and at downstream station of a high deflection impulse turbine cascade. The interaction between passage and tip clearance vortices is also studied. The experiments are somewhat idealized with low inlet turbulence intensity (less than 0.4%) and with no wall motion.

**EXPERIMENTAL SET UP**

Experimental investigations were carried out in a linear cascade tunnel available at the Thermal Turbomachines Laboratory, Indian Institute of Technology, Madras, India. Fig. 1 shows the general lay-out of the experimental apparatus used for the present investigations. The salient details of the cascade are given below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chord, ch</td>
<td>175 mm</td>
</tr>
<tr>
<td>Axial chord, e</td>
<td>170.8 mm</td>
</tr>
<tr>
<td>Spacing, S</td>
<td>114.5 mm</td>
</tr>
<tr>
<td>Span, h</td>
<td>400 mm</td>
</tr>
<tr>
<td>Aspect ratio</td>
<td>2.35</td>
</tr>
<tr>
<td>Space/chord ratio, ( \delta )</td>
<td>0.65</td>
</tr>
<tr>
<td>Maximum thickness/chord</td>
<td>35%</td>
</tr>
<tr>
<td>Position of maximum thickness/chord</td>
<td>42%</td>
</tr>
<tr>
<td>Trailing edge thickness/chord</td>
<td>5%</td>
</tr>
</tbody>
</table>

**Angles**

<table>
<thead>
<tr>
<th>Angle</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of blades, ( Z )</td>
<td>8</td>
</tr>
<tr>
<td>Inlet blade angle, ( \beta_1 )</td>
<td>57.5°</td>
</tr>
<tr>
<td>Outlet blade angle, ( \beta_2 )</td>
<td>-62.5°</td>
</tr>
<tr>
<td>Stagger angle, ( \gamma )</td>
<td>12.5°</td>
</tr>
</tbody>
</table>

Angles are referred with respect to cascade axial direction.

As the flow encountered near the endwalls of the cascade is generally expected to be three dimensional, a miniature five hole pressure probe having 2.8 mm head diameter was used to measure the total pressure, static pressure and the flow direction in two perpendicular planes, namely, yaw and pitch planes. The probe was calibrated in non-nulling mode for different pitch and yaw angle combinations. Detailed measurements of flow condition were made at the 5 axial stations, Fig. 2, by traversing the probe. The blade surface static pressures were measured at different spanwise locations from the blade tip for four configurations of the cascade, viz., for four values of the tip clearances. The tip clearance (\( \gamma \)) is expressed as a ratio of blade chord (ch) and the four (\( \gamma/ch \)) values chosen for the present work are 0.000, 0.008, 0.015 and 0.03. The distances of the traverse stations were measured from the blade leading edge along the cascade axis (x direction) and are expressed as a percentage of axial chord \( 'e' (\xi = 0 \text{ at blade leading edge and } \xi = 1.0 \text{ at blade trailing edge}). The axial stations are 5%, 40%, 70%, 95%, and 109% of the axial chord from the blade leading edge, (L.E.). The required blade tip clearances were obtained by machining one end of the blades in a milling machine.

A semi-automatic traversing mechanism was used in the present experiments for traversing the probes at various measuring stations. A digital type micro-manometer, together with a scanning box was used for recording the blade surface static pressures as well as the pressures sensed by the pressure probes during the traverses. The micro-manometer had an accuracy of \( \pm 0.01 \) mm of H\(_2\)O.

**EXPERIMENTAL INVESTIGATIONS**

The following measurements were made at different axial stations of the cascade for all the four cases of tip clearance chosen for the present study.

**Flow Surveys Through the Cascade**

The Reynolds number was maintained at a constant value of 3.8 \times 10^6 (with respect to blade chord and the mean outlet flow velocity) for the present experimental studies. For each tip clearance condition and at each of the axial stations, Fig. 2, the total pressure, static pressure and flow direction were measured with the help of the 5 hole probe by traversing at 10 to 19 pitchwise locations, depending upon the passage width. The measurements could not be performed very near the pressure and suction surfaces because the probe head radius was about 1.4 mm and the blade curvature also prevented the traverses very close to the surfaces. Apart from these reasons, the probe interference effects were also more near the surfaces and the data was not reliable. Hence measurements were taken at about
Secondary velocities are also plotted and superimposed over the angles. Unlike in low deflection compressor blades, angle observable difference between the pressure surface and suction surface flow by 120° and has large pressure and suction surface curvatures. With the result, at any axial station there is a considerable difference between the pressure surface and suction surface angles. Unlike in low deflection compressor blades, angle obtained from camber line at any axial station does not represent mean angle between pressure and suction surfaces at that axial station. Hence at each axial traverse station, it was decided to use midspan flow angle for each pitchwise location for computation of secondary velocities. The difference between midspan flow angle and local flow angle along the span at that particular pitchwise station gives the deviation of the flow from its two dimensional value. The streamwise and normal velocities are computed as follows.

\[ C' = \sqrt{C'_m^2 + C'_n^2} \]

where \( C_m \) and \( C_n \) are axial and tangential components of velocity

\[ C_{2n} = C' \cos(\alpha' - \alpha) \]

\[ C_{2m} = C' \sin(\alpha' - \alpha) \]

where \( C' \) is the streamwise velocity component and \( C_{2m} \) is the normal or pitchwise velocity component.

\[ \alpha_{sec} = \tan^{-1} \left( \frac{C_{2m}}{C_{2n}} \right) \]

Blade Tip surface static pressure measurements were taken for two clearances \( r/ch = 0.008, \) and 0.030. Static pressure tapings were provided by inserting stainless steel tubes of 1.2 mm diameter for most part of the end surface and 0.8 mm diameter near the trailing edge, through the hollow instrumented blade. About 42 tappings were provided at the 6 axial stations corresponding to 5%, 15%, 25%, 40%, 55%, 70%, 85%, and 95% of the axial distance from the leading edge.

Blade Surface Static Pressure Measurements

Blade surface static measurements were measured by embedding 24 stainless tubes of 1.2 mm diameter on suction surface and 16 tubes on pressure surface with closer spacing of tubes near leading and trailing edges. Static pressure tapings were drilled at 4 spanwise locations, namely; at 200 mm, 35 mm, 10 mm and 5 mm respectively from the blade tip.

RESULTS AND DISCUSSION

Local Loss Contours and Secondary Velocity Vectors

The local loss coefficient at any station is calculated as the difference between midspan inlet total pressure (\( \xi = -40\% \)) and local total pressure non-dimensionalised with the dynamic pressure based on pitch and spanwise mass averaged velocity at the blade exit at \( \xi = 106\% \) for zero clearance. The inlet boundary layer thickness at \( \xi = -40\% \) was about 42 mm and the velocity at this station was about 30 m/s. On the x-axis of the contour plots, the distance from the endwall (\( z \)) non-dimensionalised with full blade span (\( h = 400 \text{ mm} \)) is represented. The y-axis represents the distance between pressure and suction surfaces non dimensionalised by the blade spacing.

To facilitate better understanding of the vortex cores, their movement, magnitude and rotation, the vector plots of the secondary velocities are also plotted and superimposed over the loss contours. The vector plots are plotted as follows. The rotor blade employed in the present investigations deflects the flow by 120° and has large pressure and suction surface curvatures. With the result, at any axial station there is a considerable difference between the pressure surface and suction surface angles. Unlike in low deflection compressor blades, angle obtained from camber line at any axial station does not represent mean angle between pressure and suction surfaces at that axial surface region.
tion of a separation bubble inside the tip gap. At a small clearance, \( \tau/ch = 0.008 \), Fig. 6 in our investigations also the clearance flow did not form into clearance vortex but joined the main secondary flow inside the blade passage up to \( \xi = 40\% \). Graham (1986) based on the flow modeling studies has reported that at smaller clearances the dominance of viscous forces inside the tip gap could result in such a tendency. Bindon (1987a,b), Moore and Tilton (1988), Sjolander and Amrud (1987), and Yamamoto (1988a) stated that the clearance flow passing through the tip gap forms a vortex almost around mid chord position inside the tip gap and the flow rearward to this emerging from the tip gap exit into the blade passage will be in the form of a vortex of fluid which is known as the clearance vortex.

\( \xi = 70\% \), Figs. 8 -10

It can be observed for \( \tau/ch = 0.00 \), Fig. 8, that the passage vortex has clearly moved towards the suction surface endwall corner region. The losses are more concentrated and are higher in their magnitude in this region. The endwall boundary layer has become thin. This suggests the movement of the low energy fluid of the wall boundary layers into this loss core. There seems to be only a thin new boundary layer all along the pressure surface in the spanwise direction. For \( \tau/ch = 0.008 \), Fig. 9, the clearance vortex is concentrated very near the endwall and passage vortex is away from the endwall and is spread in both spanwise and pitchwise directions but still attached to the suction surface suggesting that all the fluid from the blade and endwall boundary layers does not go into clearance vortex, a good proportion of it still forms a part of the passage vortex.

Referring to Fig. 10, \( \tau/ch = 0.03 \), the clearance vortex is situated at the end wall corner and the passage vortex is pushed away from the end wall due to the action of the clearance vortex. As stated earlier, horse shoe vortex was not observed for this clearance and subsequently the magnitude of the passage vortex up to this axial station is less than for other cases of clearances.

\( \xi = 95\% \), Figs. 11 - 13

When the flow moves to \( \xi = 95\% \), Fig. 11 for \( \tau/ch = 0.00 \), the passage vortex occupies almost half the blade passage width and about 20% of the span. The losses along the suction surface have gone up considerably. One of the reasons is that the flow decelerates on the suction surface at this measuring station resulting in thicker boundary layer growth and added to this the cross channel deflection of the fluid results in accumulation of pressure surface boundary layer fluid along with wall boundary layer fluid. The magnitude of the loss at the core centre is very high and is trying to convect away from the endwall.

When \( \tau/ch = 0.008 \), Fig. 12, the flow pattern near the suction surface is very complex. The magnitudes of the losses due to passage and tip clearance vortices have gone up with near absence of the end wall boundary layer. Pressure surface end wall corner has mostly through flow fluid. However the size of the passage vortex is still larger than tip clearance vortex. Yaras and Sjolander (1992) mentioned from their earlier investigations that secondary loss which is attributed to passage vortex fell off quickly with clearance and at 2% clearance it accounted for an insignificant fraction compared with loss in the tip leakage.
flow. However their cascade had thin boundary layer with a low turning of only 45° only. In the present investigations the inlet boundary layer thickness is about 42 mm with a cascade turning of 120°, therefore the secondary losses are very strong. The passage vortex did not diminish in size with increase in tip clearance.

When clearance is increased to \( r/ch = 0.03 \), Fig. 13, the tip clearance vortex is strong and spreads to almost entire passage width. A new skewed boundary layer is beginning to form on the end wall and its interaction with the vortices could be clearly seen.

\[ \gamma = 106\% , \text{Figs. 14 - 16} \]

When the flow moves to downstream it is observed from Fig. 14 for \( r/ch = 0.00 \), the passage vortex is larger in magnitude and is located at about 18% of the spanwise distance from the endwall. The endwall boundary layer is thin as most of the boundary layer fluid is swept into this loss core.

As the trailing edge thickness for the present cascade is thicker (about 5% of the blade chord), the wakes are broader and the loss magnitudes in the wake are considerable. Strong inward wake flows are observed to exist along the trailing edge. It can be seen that the wake flow runs into the loss core with peaks on either side. The strong inward flows transport low energy fluids to the suction corner and interact with the flows rolling up from the endwall surface. The transportation of the low energy boundary layer fluid into the loss regions and the wake, appears to be the cause for the redistribution of the losses and their rapid increase in the downstream. This redistribution of the losses will certainly affect the flow mixing downstream.

When \( r/ch = 0.008 \), Fig. 15, the magnitudes of losses in tip clearance and passage vortices have gone up which could be due to the further accumulation of boundary layer fluid into vortices which is rather expected from the total absence of end wall boundary layer. When the clearance is further increased to \( r/ch = 0.03 \), Fig. 16, the clearance vortex is strong and spreads to almost entire passage width. The passage vortex is slightly shrunk in size though the magnitude of the loss inside the vortex remains same.
Pitchwise Mass Averaged Loss Coefficient

At $\xi = 106\%$, Fig. 17, the loss peak at zero clearance is evidently due to the passage vortex. When the clearance is present, the first loss peak is due to the clearance vortex and is located very close to the endwall, at about 2\% of spanwise distance. The magnitudes of this loss peak and the loss peak due to passage vortex have increased when compared with that at the zero clearance. The size of the loss peak shows the effect of the clearance vortex and its size increases with the clearance. The centre of these peaks also convect slightly away from the endwall with clearance.

At $\xi = 106\%$, Fig. 18, the blade wakes are wider in Figs. 14 - 16, consequently pitchwise averaged total pressure loss coefficient has increased beyond 20\% of the blade span compared to $\xi = 95\%$. The loss at midspan for this station represents profile loss of the blade. As the new boundary layer growth on
Though there is no blade present, the pressure and suction surfaces are expected to persist at this station and with increased boundary layer growth on the endwall, the strength of the loss peak due to tip leakage vortex has grown especially for $\tau/ch = 0.03$. When $\tau/ch = 0.008$ and $0.03$, the total pressure loss coefficient values of the first peak are 0.44, 0.54, and 0.85 and the values of the second peak are 0.30, 0.31, and 0.35 respectively. The spreading tendency of the second loss region could also be seen to be lesser with the increase in clearance. It can be seen that the loss coefficient in general has increased in magnitude and this includes the wake loss.

**Blade Tip Surface Pressure Distribution**

In Figs.20 a and 20 b where blade tip surface static pressure contours for $\tau/ch = 0.008$ and 0.03 are shown. It can be seen that a low pressure region exists right on the pressure corner which appears to be due to an attached flow around the blade edge radius. The diffusion of this could lead to the formation of the clearance vortex inside the tip gap around the mid chord region. The low value of the static pressure coefficient occurs on the blade tip near the pressure surface between 60% to 80% of axial chord and this is seen to increase with the clearance. The very low pressure on the edge of the pressure surface of the blade is caused by the high curvature of the clearance flow moving radially up the pressure surface and then turning sharply through 90° as it enters the clearance gap. The low pressure thus results in high velocity at the surface. This is further aided by the thin boundary layers normal to the pressure surface flows. These boundary layers would become even thinner due to the acceleration of the flow at the gap entry. The
velocity at gap entry was found to be about twice the velocity at $\frac{c}{r} = 106\%$ and the velocity at $\frac{c}{r} = 100\%$ was about 35 m/s. It can be seen that the zone of low pressure is around 60% of the axial chord for this blade geometry. Graham (1986), Wadia and Booth (1982), Bindon (1987a), Sjolander and Amrud (1987), Yamamoto (1988a,b), Moore and Tilton (1988) have also reported the existence of the low pressure region at the blade tip near the pressure surface. While the decrease in the pressure values near the suction surface could be attributed to the presence of the clearance, the distortions indicated by the contours appear to be due to the influence of the clearance vortex that has formed inside the tip gap.

CONCLUSIONS

The clearance flow issues as a thin jet through the tip clearance. The influence of the tip clearance on the flow far upstream of the cascade is very insignificant. The clearance effect is significant on the tendency of formation of the horseshoe vortex. This vortex is not clearly noticed for the clearance $r/ch = 0.03$, indicating that the pressure forces across the blade surfaces showed a dominating influence affecting this tendency. The secondary losses in the cascade do not show appreciable increase till about a distance of 40% of the axial chord from the leading edge even when the clearance is increased to $r/ch = 0.030$. This could be due to the fact that the clearance fluid passing through the tip gap appears to be less up to this axial station as it does not roll up into tip clearance vortex, instead joined the secondary flows inside the blade passage.

The tip clearance vortex forms inside the gap around 40% chord with less generation. This vortex interacts intensely with the passage vortex inside the blade channel. The effect of secondary flows and tip clearance flows is considerable especially in the downstream flow field of the turbine. The losses due to tip clearance form significant portion of the total loss. The mixing flow patterns of the leakage flow with other vortices depend strongly on the clearance sizes.

REFERENCES


