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# The Effects of Relative Motion, Blade Edge Radius and Gap Size on the Blade Tip Pressure Distribution in an Annular Turbine Cascade With Clearance

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## ABSTRACT

Flow visualisation and microscopic static pressure measurements were done in the tip clearance region of an annular turbine cascade with a rotating outer casing to simulate the relative motion at the tip of an axial rotor. The effect of relative motion did not have a significant effect on the blade gap pressure distributions. As in previous studies the narrow deep pressure depression on a sharp pressure edge was seen. It was confirmed that the width of the gap separation bubble depends on clearance and a correlation with flow visualisation showed that at the reattachment line there is the expected slight pressure peak. The separation bubble, which is thought to contribute a major part of the leakage loss, was shown to disappear when the pressure surface tip is give a radius of 2.5 gap widths.

## NOMENCLATURE

| | |
|---|---|
| $C_{p2}$ | Static pressure coeff = $2\ (p-p_1)\ /\ \rho\ V_2^2$ |
| p | Static pressure |
| V | Absolute velocity |
| $\rho$ | Density |
| 1 | Refers to cascade inlet, tip |
| 2 | Refers to cascade outlet, tip |
| Y | Distances from pressure face tip edge measured normal to chord along clearance gap face |
| Y' | distance down pressure face measured from tip edge |

## 1. INTRODUCTION

The degradation of turbomachine performance with increasing tip clearance is well known for both turbines (see for example Booth et al 1982a, and Ewen et al 1973) and compressors (Ruden 1937 and Rains 1954). The efforts at reducing this loss are dominated by mechanical control of gap size and by aerodynamic methods of limiting the gap leakage mass flow for a given gap size. The leakage related quantities of loss, gap mass flow and tip sealing efficiency are often conveniently described by the gap discharge coefficient. Much of the previous work has sought to predict and measure this quantity or to use it as a basis for evaluating tip performance (Rains 1954, Wadia and Booth 1982, Moore and Tilton 1987). All of these models are related in some way to the driving pressure difference across the profile and are based on a single physical clearance flow model applicable at all axial stations between the leading and trailing edges. These models all acknowledge the separation bubble near the pressure surface and the subsequent reattachment or mixing.

In order to extend the understanding of the flow physics inside the clearance gap Bindon (1986a,b and 1987a,b,c), using miniaturised probes and tappings in a linear turbine cascade with clearance, measured both the complete static pressure field and the boundary layers inside the gap and on the endwall and also visualised the minute flow structures with smoke.

In a companion paper offered, Bindon (1988) analysed the data to obtain the variation with axial distance of the tip clearance losses due separately to internal gap friction, mixing of the leakage flow and normal secondary flow and attempted to relate some of the findings to the observed flow

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phenomena. It is suggested that the separation bubble plays a dominant part in loss formation inside and outside of the gap. Due to the accelerating gap pressure field over the forward half of the blade, a strong chordwise flow is established within the separation bubble. This flow accumulates near mid chord and near the pressure edge where the gap pressure is the lowest. The only way that this flow can leave the gap is by mixing with the leakage jet at inlet thus forming a very intense loss zone due the high inlet jet velocity. On the endwall, mixing losses are insignificant over the forward half of the blade despite the entry of leakage flow there. Mixing losses only appear to emerge and grow very rapidly when the high loss wake containing the mixed out bubble enters at 70% chord. The exceptionally rapid growth of the endwall mixing loss was tentatively attributed to the possibility of the bubble wake having insufficient energy to negotiate the diffusion in the suction corner. The result would then be a "separation" or flow reversal with accompanying mixing loss. Finally it was suggested that since so much entropy is generated within the gap in reducing the discharge coefficient, fewer losses may result from a relatively loss free flow in which the combination of tip deflection and entropy production is optimised. Such a loss free flow may also reduce the mixing loss.

This paper therefore in the first instance begins to address the question of eliminating the clearance gap separation since it appears to play such an important part in loss formation. The sharp edged pressure corner at gap inlet is progressively radiused and the effect of this on the separation bubble as refected by pressure distribution and in surface flow visualisation is investigated. The radiused edge studies also extend the very limited radius studies originally performed by Bindon (1986b, 1987a).

Since much of the understanding of tip clearance flow was obtained in experimental rigs which do not model the relative motion in the tip region, this paper also studies this effect by rotating the casing of an annular test cascade. There is conflicting evidence in the literature regarding the influence of relative motion. Mayle and Metzger (1982) argued that the gap path length was not long enough for the two boundary layers to meet and interact viscously. A spinning disc rectangular surface experiment proved the thesis by showing that the heat transfer coefficient was virtually independant of simulated turbine wall motion. Graham (1985) for a turbine cascade configuration found that wall motion had the effect of significantly reducing the leakage flow, of moving the leakage vortex in towards the suction surface and of virtually eliminating the suction surface depression caused by the vortex. He also provided gap exit velocity traverses which showed that the bubble wake boundary layer almost filled the gap which is in agreemnent with the traverses of Bindon (1987c) in the same general region. Dean (1954) found that the vortex core was moved further away from the suction surface of a compressor rig in which relative motion enhances tip leakage flow. Gearhart (1964), also for a comprssor rig, reported that endwall motion increased the gap static pressure depression. The 2D numerical model of Wadia and Booth (1982) allows for wall motion and a drop in discharge coefficient was calculated with the wall in motion as for a turbine.

Finally this paper responds to Moore (1987) who has suggested that the width of the separation bubble increases with the clearance gap and the width is measured for various gap sizes.

## 2. ROTATING CASING CASCADE AND INSTRUMENTATION

An existing one and a half stage low speed research turbine was converted to model the relative motion between a rotor blade and casing. As shown in Figure 1, the stator row becomes the stationary annular test cascade while the rotor row is attached to a segment of the outer annulus. This segment extends forward over the stator and rotates with the rotor to create the required relative motion at the tip.

To provide adequately sized blade profiles for the microscopic pressure measurements needed, relatively low aspect ratio untwisted cascade was chosen. A NACA $A_3K_7$ profile with a slightly thickened trailing edge to model a cooled turbine was used. The profile and test data are summarised in Table 1.

The profiles were cast in epoxy resin using a mould created from a hand-made wooden master blade. As described in Bindon 1987a,

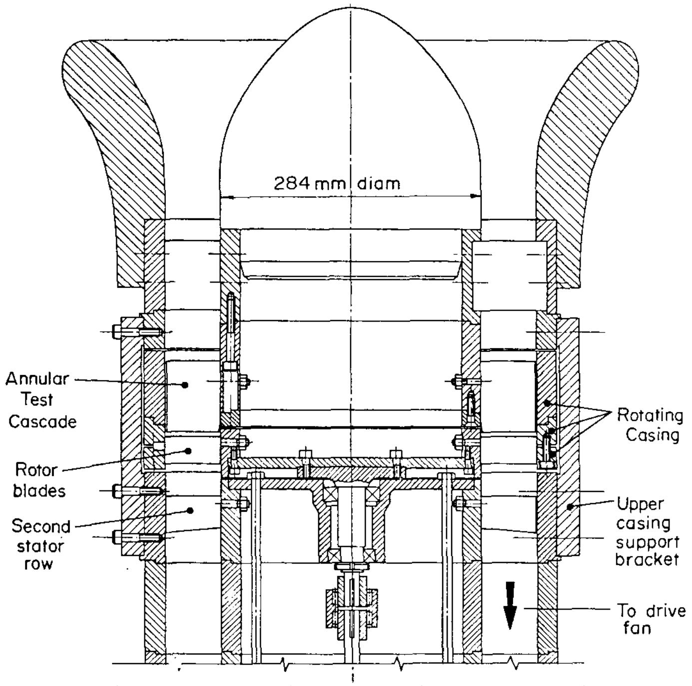

FIGURE 1 ONE AND A HALF STAGE LOW SPEED RESEARCH TURBINE ADAPTED FOR ROTATING CASING STUDY.

the blades were slotted at the tip, the slots taped over and micropunctured for high resolution static pressure measurements of up to 5 tappings per millimeter. After puncturing the position of a tapping was measured using a hand held magnifier with 0.1mm engraved graduations. Each slot was accessed by a spanwise hole drilled through into the hub region and connected to a multitube manometer by a tube bunch exiting through the nose cone.

**TABLE 1 : CASCADE AND TEST DATA**

| | |
|---|---|
| Chord | 115mm |
| Span | 61mm |
| Annulus Tip diameter | 406mm |
| Solidity (tip) | 1.35 |
| Maximum thickness | 20% chord |
| Trailing edge thickness | 5% chord |
| Cx / Ucasing | 0.7 |
| Camber angle | 60 deg |
| Number of blades | 15 |
| Reynolds number (exit) | $3 \times 10^5$ |
| Gas inlet angle | 0 deg |
| Clearance gaps : | 4% chord (4.6mm) |
| | 2% chord (2.3mm) |
| | 1% chord (1.2mm) |
| Tape thickness | 0.06mm |

## 3. THE CLEARANCE GAP PRESSURE DISTRIBUTIONS AND THE EFFECT OF RELATIVE MOTION

In a turbine the relative motion of the casing is in the opposite direction to the leakage flow. The viscous shear on the gap endwall is therefore much more intense than in a compressor. Although gap sizes are small, the flow path even at maximum blade thickness is also short and the internal boundary layers would not normally be fully developed at gap exit. A relatively loss free core layer was seen in Bindon (1987c) over most of the gap length except at mid chord where the mixed out bubble flow is thought to emerge. Thus the casing boundary layer is unlikely to have a marked effect on the blade boundary layer and hence on the pressure distribution.

By allowing the casing to either rotate or remain stationary, the pressure distribution on the blade within the clearance gap was measured along lines roughly normal to the blade chord and hence the direction of the leakage flow. In Figure 2 results are shown at 30%, 40%, 50% and 60% chord and for the stationary wall, are similar to those obtained from the linear cascade of Bindon (1987a). A low pressure region exists right on the pressure corner which is thought to be due to an attached flow around the small blade edge radius. This diffuses sharply before separating to form the bubble. With relative motion, this pressure trough is less evident and edge pressures are higher. No explanation is advanced for this since very little is known about the microstructure of the corner flow. These new results for the corner are however significant because in Bindon (1987a) only a single measurement had been available to define the phenomena. An improvement in the micro tapping technique has added 3 more holes close enough to the corner to record the pattern. It can therefore be stated with more confidence that these narrow regions of low pressure do exist on sharp edged pressure surfaces.

The pressure level in the bubble zone is little affected by wall motion as is the apparent width of the bubble. The pressures reached in the fully attached area behind the bubble are significantly higher (about 14%) with rotation. Since wall motion would increase the viscous shear action within the clearance gap, both the leakage flow quantity and the size of the resulting vortex, could be slightly reduced. This could increase the suction surface (i.e. gap exit) pressure which

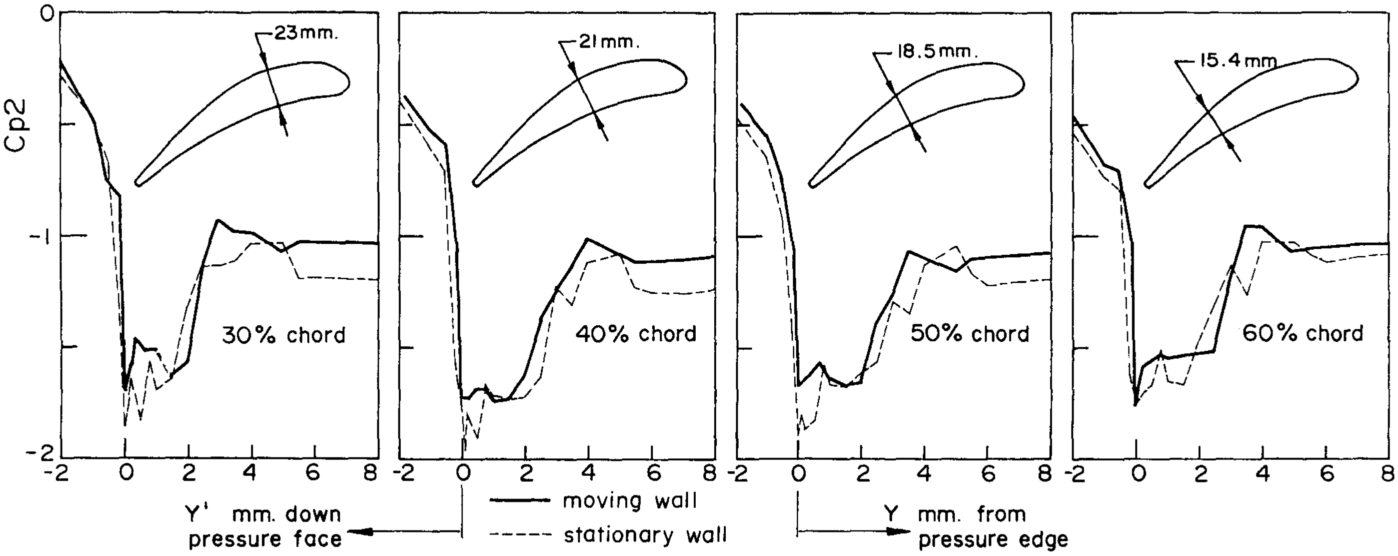

FIGURE 2 EFFECT OF RELATIVE MOTION ON THE BLADE PRESSURE DISTRIBUTIONS IN THE TIP CLEARANCE GAP

has been found to be radically affected by the leakage vortex blockage (Graham 1985, Bindon 1987a).

The remainder of the study was done with the outer annulus rotating.

In order to highlight the low pressure levels encountered in the separation bubble and therefore the kinetic energy that needs to be imparted to the separation bubble flow, Figure 3 shows the mean of the lowest part of the pressure trough related to the bubble in comparison to the suction surface distribution. It should be noted that this bubble pressure is perhaps the quantity most representative of the pressure side distribution. The gradient on the pressure face at the tip is too intense to realistically define a tip pressure and the actual edge pressure is too narrow a zone for it to be regarded as defining the tip loading. It is also difficult to use the pressure surface loading some distance away from the tip, before the gap acceleration starts, because in an annular cascade, radial effects are present. The clearance gap was therefore sealed with felt and, with a stationary casing, a zero clearance distribution recorded and presented in Figure 3 as the "2D" or zero clearance reference loading.

Figure 3 shows that the suction surface pressure is much lower than the "sealed" or semi 2D value and corresponds to the linear cascade measurements of Bindon (1987a) and Graham (1985). While the suction surface pressure is reduced by clearance, there is no evidence of the significant distortions seen by Bindon (1987a) and Graham (1985) and attributed to the influence of the leakage vortex in the suction corner. Since endwall motion is present this could due to the reduced strength and penetration of the leakage flow as found by Graham (1985) and by Sjolander and Amrud (1987). The bubble pressure is lower than that measured on the suction surface, or clearance gap exit value, from 30% chord onwards. The nature of the pressure distribution indicates that the loading is weighted towards the leading edge, a factor which may have a significant influence on the flows inside the gap. Sjolander and Amrud (1987) found, for a different profile, that the suction surface had two depressions which were associated with two leakage vortices emerging from the gap. Thus very different flows could occur inside the gap depending on the profile, the important factors being perhaps camber and thickness.

The actual bubble pressures and the remainder of the blade gap surface pressure distributions are given in Figures 4 and 5. The first figure shows 3 distributions over the forward half of the blade in the zone where it can clearly be seen that the flow within the bubble undergoes a strong acceleration towards the trailing edge. A new result seen in these curves is that the pressure towards the suction side begins to fall well within the parallel walls of the clearance gap. This effect reduces with chord and, as Figure 5 shows, the distributions are virtually flat from mid chord onwards.

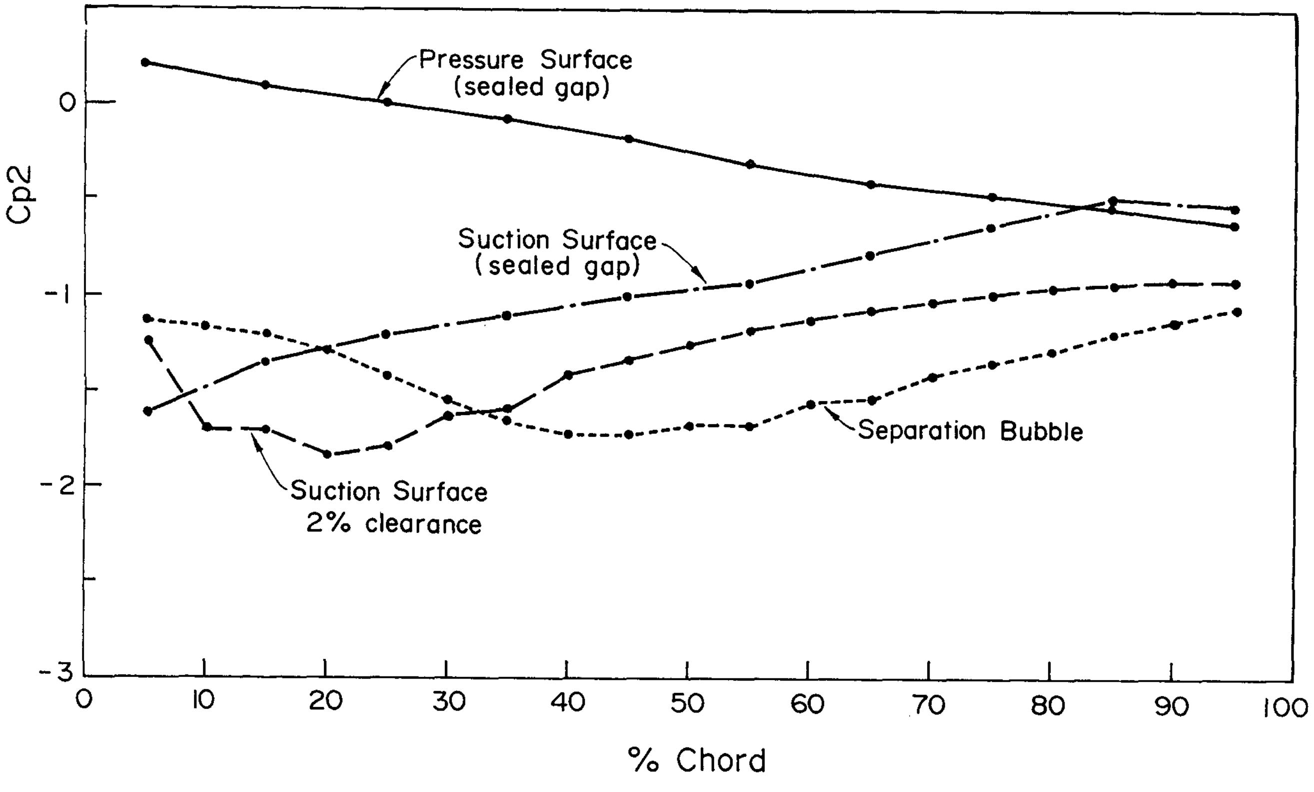

FIGURE 3 BLADE PROFILE PRESSURE DISTRIBUTIONS COMPARED TO THE SEPARATION BUBBLE PRESSURE

Figure 5 shows the pressures over the rear half of the blade and the steep pressure rise required of the flow inside the bubble can be seen. It is this gradient which is thought to separate the internal bubble flow that is established over the forward half of the blade and to force the accumulated fluid to mix with the gap inlet jet in order to move towards the reltively high pressure at gap exit. This mixing process may form the major part of the internal gap loss and appears to be a significant factor in reducing the gap discharge coefficient (Bindon 1988).

This pressure gradient raises the interesting question of whether the flow within the bubble is not in actual fact towards the leading edge. A careful study was made using oil flow visualisation and at a clearance gap of 4% chord, oil could clearly be seen moving sluggishly from the trailing edge towards mid chord. Unfortunately, the turbine rig was vertical and the oil was moving against gravity. Since oil therefore could be moving towards mid chord from both directions, it explains why oil is seen in Figure 6 to collect near mid chord and break away towards

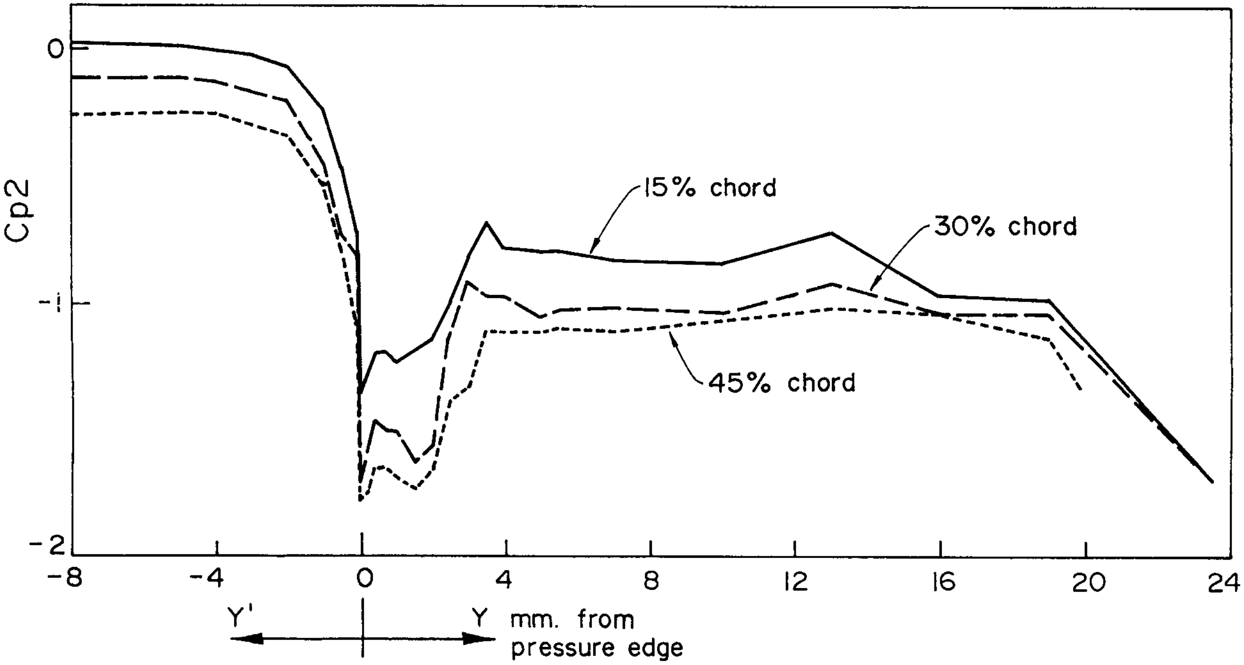

FIGURE 4 BLADE PRESSURE DISTRIBUTIONS IN THE CLEARANCE GAP OVER THE FORWARD REGION OF PROFILE WHERE PRESSURES ARE FALLING

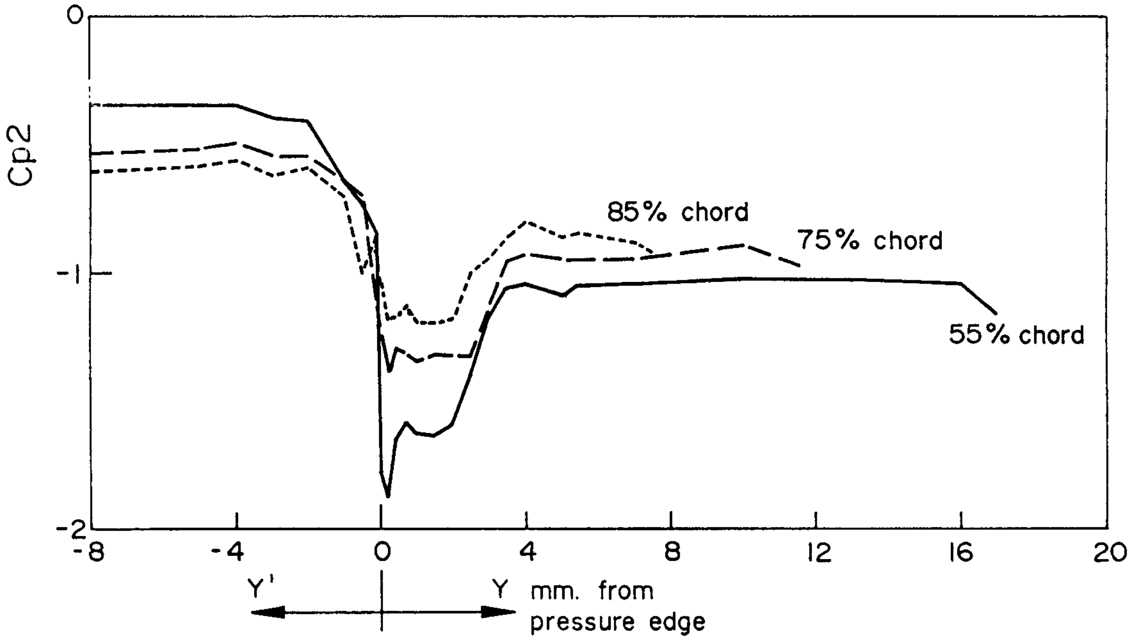

FIGURE 5 BLADE PRESSURE DISTRIBUTIONS IN THE CLEARANCE GAP OVER THE REAR OF THE PROFILE WHERE PRESSURES ARE INCREASING.

the gap exit. The oil movement over the forward half of the blade, although aided by gravity, is not solely due to gravity as can be demonstrated by the strong chordwise corkscrew motion clearly seen in the smoke flow visualisation of Bindon (1986a, 1987b).

The possible counter flow of separation bubble fluid is thought to be important since more fluid would be convected towards mid chord and would add to the internal gap loss generation and to the reduction in gap discharge coefficient.

## 4. THE EFFECT OF GAP SIZE ON CLEARANCE FLOW

The smoke flow visualisation of Bindon (1986a, 1987b) showed that the width of a separation bubble became wider with increased clearance gap. However, the pressure distributions seemed to indicate that the bubble width was independent of gap size. In an oil surface flow study, Moore (1987), also claimed that the bubble increased with clearance.

In the annular cascade, the gap size was set at 1%, 2% and 4% chord and the pressure distribution recorded and the oil surface flow visualisation photographed through a transparent rotating outer casing. Figure 6 shows the pressures at 40% chord and the complete flow visualisation except for the trailing edge, which was hidden by the metal casing.

It is quite obvious that the width of the bubble increases with clearance gap. It also tends to become wider with axial distance and, in the case of the largest clearance, almost breaks away near mid chord, probably due to the accumulation of oil which is fed there from both directions. The bubble width at 40% chord was measured and the positions indicated on the pressure distributions. The edge of the visible bubble coincides almost exactly with termination of the pressure rise which has been associated with the reattaching flow behind the bubble. When this interpretation of the pressure distribution is applied to the results of Bindon (1987a), then the bubble width in that study can also be said to vary with clearance gap size. Since at the edge of the separation bubble (ie where the flow reattaches) there is a flow of considerable strength towards the blade surface, a slight pressure peak can be seen which immediately falls away in the direction of the leakage flow.

The minimum pressure within the bubble is also strongly dependent on gap size but is not monotonic and the 2% clearance shows a very much lower pressure than the values at lower and higher clearances. A somewhat similar result can be seen in the pressure distributions of Bindon (1986b, 1987a)

## 5. THE EFFECT OF BLADE EDGE RADIUS ON GAP FLOW

In Bindon (1987a), a brief study was made of the effect of pressure edge radius and it was seen that the sharp pressure dip on the edge was reduced as the radius was increased. Since the pattern of tip leakage loss development inside the clearance gap and on the endwall is thought to be strongly connected to the gap separation bubble, the effect of increasing edge radius on the formation of the bubble was investigated. If the formation of the separation bubble could be prevented there could be a significant reduction overall loss.

The inlet to the clearance gap is equivalent to a sudden contraction and the corner radius dimension which will keep the flow attached and avoid a separation bubble will most likely depend on the gap size. Tests were therefore done at a fixed gap width of 2% chord and once the critical radius was found, a single experiment at a smaller gap was made for confirmation. By monitoring the bubble using surface flow visualisation and the gap

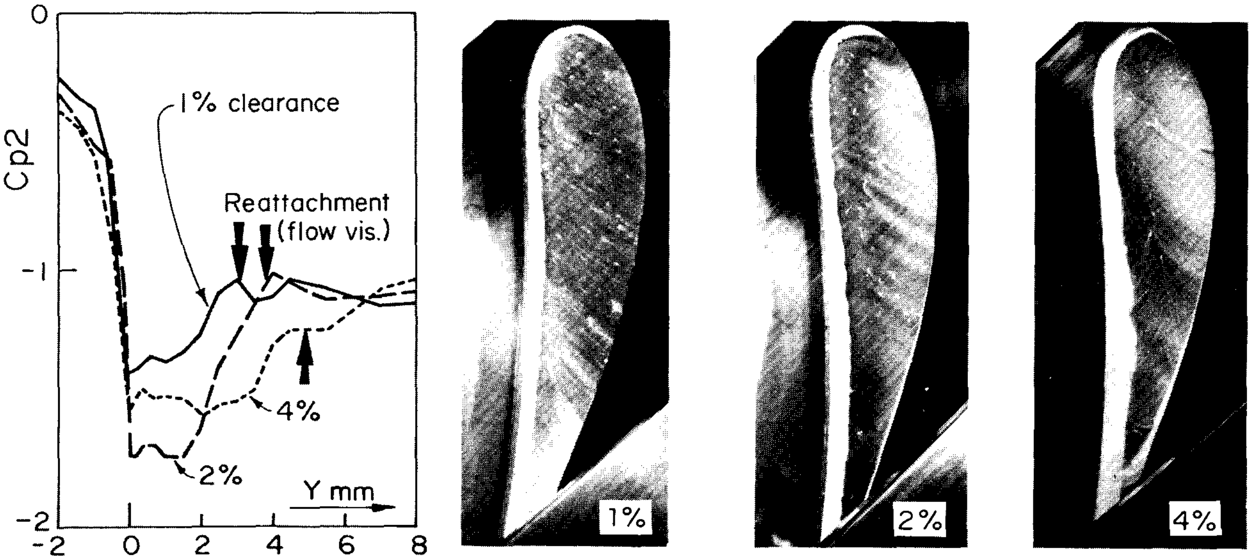

FIGURE 6 EFFECT OF GAP SIZE ON PRESSURE DISTRIBUTION AND SEPARATION BUBBLE

pressure distribution, the edge radius was progressively increased from the original "sharp" value of 1/4 gap width (0.5% chord) until the signs of separation vanished.

Figure 7 shows the surface flow visualisation for pressure edge radii of 0.25, 1, 1.5 and 2.5 gap widths (0.5%, 2%, 3% and 5% chord). For the smallest radius the bubble is very wide and starts right at the edge of the blade and runs unbroken from leading to trailing edge. At a radius of 1 gap width the pattern is quite different. The oil indicates that the bubble starts part way around the radius and the attached flow off the pressure surface can be distinctly seen. The bubble appears to be very narrow. Near mid chord the bubble itself separates and thereafter the flow does not appear to reattach. At a radius of 1.5 gap widths, the flow has remained attached even further around the edge before forming a narrow bubble. At 2.5 gap widths the flow is clearly attached around the edge and right across the gap surface.

In Figure 8 the pressure distributions for various edge radii are presented for 30%, 40% and 50% chord. The characteristic depression attributed to the bubble can be clearly seen at the sharp edge radius of 0.25 gap widths and also at 0.5 gap widths. At the next radius of 1.5 gap widths, the edge pressure is as much as 35% higher and there is a distinct rise slightly before the termination of the radiused section. This coincides exactly with the position of the thin separation line seen in the flow visualisation.

At an edge radius of 2.5 gap widths where the flow visualisation indicated attached flow, the minimum pressure occurs at the termination of the radius. There is a slight pressure rise which could indicate either a small separation but a boundary layer traverse would be needed to ascertain the nature of the flow. The pressure level within the clearance gap is the lowest for the apparently completely attached flow and this low pressure persists right through to gap exit (not shown in Figure 8). This would indicate that the suction surface pressure is actually decreased by the more energetic nature of the leakage flow as also found by Graham (1985).

The clearance gap was finally set at 0.8% chord and a blade with an edge radius of 2% chord (2.5 gap widths) was tested using flow visualisation. No signs of separation were seen thus confirming that separation can be avoided by using edge radii of 2.5 gap widths.

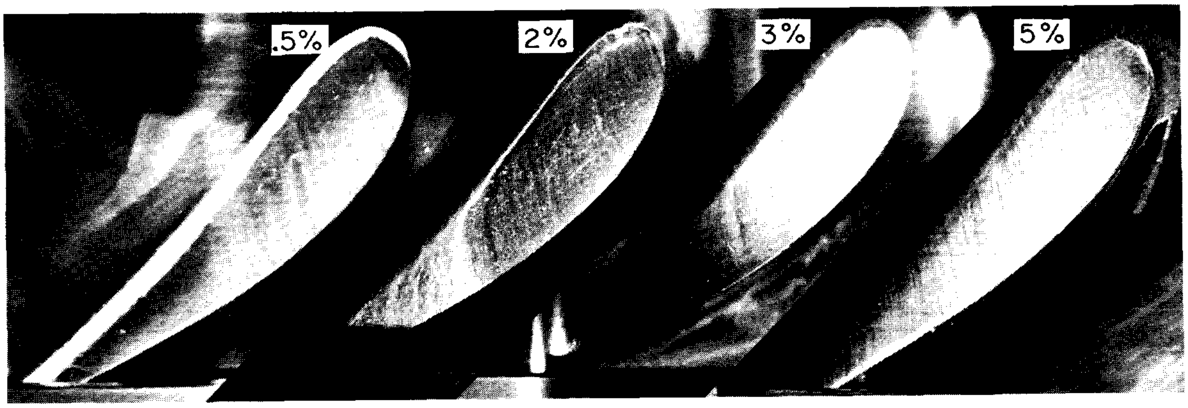

FIGURE 7 SURFACE OIL FLOW VISUALISATION SEEN THROUGH MOVING CASING SHOWING GAP SEPARATION BUBBLE FOR INCREASING PRESSURE EDGE RADIUS AND FOR A CLEARANCE GAP OF 2% CHORD

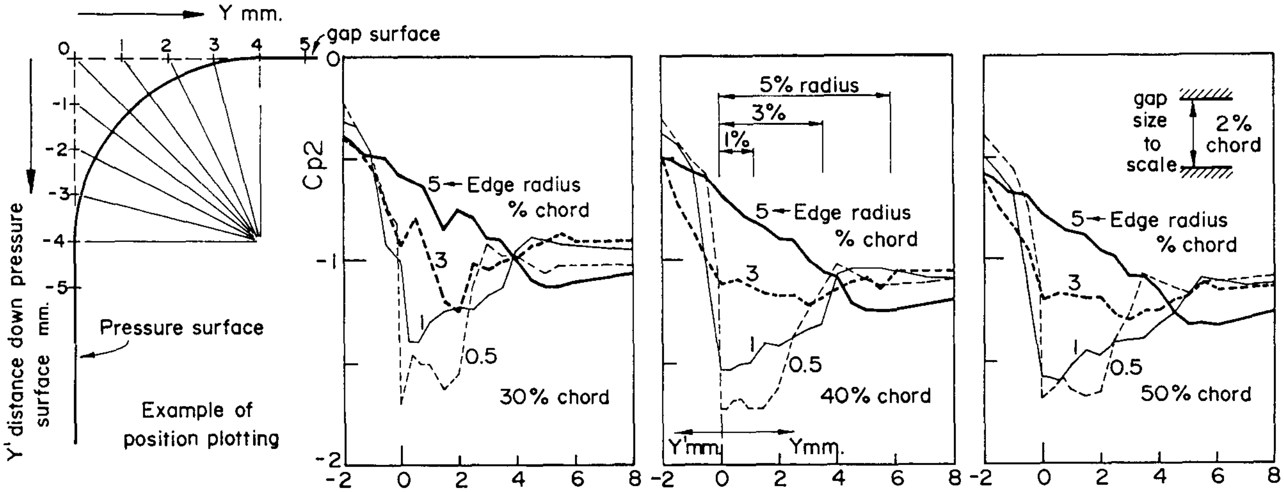

FIGURE 8 EFFECT OF PRESSURE SURFACE EDGE RADIUS ON THE SEPARATION BUBBLE AND THE BLADE GAP PRESSURE DISTRIBUTION

## 6. CONCLUSIONS

The annular rotating casing rig has provided a relatively simple vehicle for creating a relative motion in the tip clearance region of a cascade. Due to the layer of virtually loss free flow between the casing boundary layer and the blade gap surface boundary layer, the effect of relative motion was found to have little influence over the general shape of the pressure distributions measured in the tip clearance region and hence over the basic flow mechanisms in the gap. The low pressure associated with the separation bubble was little influenced by relative motion but in the reattached regime behind the bubble, the pressures were higher, probably as a result of a reduced leakage flow and reduced distortion of the suction surface pressure. This reduction in leakage flow means that relative motion is more important with respect to loss formation and gap discharge coefficient than in determining the basic nature of the gap flow.

The existence of what is thought to be a flow attachment around the pressure edge leading to very low pressures and high heat transfer, was confirmed by a higher resolution pressure tapping procedure which provided additional data in the microscopic zone of interest.

It was confirmed that the width of the separation bubble depends on gap size and the reattachment line was seen to coincide with a slight pressure peak. A pressure gradient which would accelerate flow within the bubble was again seen over the forward half of the blade. Since a separation bubble clearly also exists over the latter half of the blade where the pressure gradient is such as to move flow from trailing edge towards mid chord, it is suggested that both bubble flows meet at mid chord to mix out with the incoming leakage jet and greatly enhance the internal gap loss and reduce the gap discharge coefficient.

Increasing the radius of the pressure edge of the profile to 2.5 times the clearance gap width prevented the formation of the separation bubble and its associated deep pressure depression within the gap. This also had the effect of apparently strengthening the leakage flow and therefore reducing the suction side pressure and hence the pressure inside the gap.

The next part of the investigation will deal with the effect of eliminating the separation bubble on internal gap and endwall loss generation, on tip region flow deflection and on a deduced rotor output when the stationary cascade results are converted into the rotational frame.

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