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THE PERFORMANCE OF A LOW SPEED ONE AND A HALF STAGE AXIAL TURBINE WITH VARYING ROTOR TIP CLEARANCE AND TIP GAP GEOMETRY

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Abstract

The performance of a low speed axial turbine followed by a second stage nozzle is measured with particular reference to the understanding of tip clearance effects in a real machine and to possible benefits of streamlined low loss rotor tips.

A radiused pressure edge was found to improve the performance of a single stage and of a one and a half stage turbine at the small tip clearance levels for which the radius was selected. This is in contrast to cascade results where mixing loss reduced the benefits of such tips. Clearance gap flow appears therefore to be just like other turbine flow where the loss mechanism of separation must be avoided. Loss formation within and downstream of a rotor are more complex than previously realized and do not obey the simple rules that have been used to design for minimum tip clearance loss. For example, approximately 48% of the tip leakage mass flow within a rotor appears to be a flat wall jet rather than a wrapped up vortex.

The second stage nozzle efficiency was found to be significantly higher than for the first stage and to even increase with tip clearance. This is a surprising result since it means that not only is there a reduction in secondary flow loss but also that rotor leakage and rotor secondary flows do not generate downstream mixing loss.

Nomenclature

Variables

C	Dimensionless coefficient
h	Specific enthalpy [kJ/kg K]
$\dot{m}$	Mass flow rate [kg/S]
P	Pressure [Pa]
q	Dynamic pressure [Pa]
U	Blade velocity [m/s]
V	Absolute velocity [m/s]
W	Relative velocity [m/s]
α	Absolute angle [degrees] (tangential plane)
β	Relative angle [degrees] (tangential plane)
η	Efficiency
ρ	Density [kg/m^3]
ω	Specific work [Kj/kg]

Subscripts

1	First nozzle inlet
1.5	One and a half Stage
2	Rotor inlet (first nozzle outlet)
3	Rotor outlet (second nozzle inlet)
4	Second nozzle outlet
i,j,m,n	Summation grid variables
is	Isentropic
N1	First nozzle
N2	Second nozzle
o	Total
P	Pressure
ref	Free stream inlet reference
S	Static, Entropy
ts	Total to static (single stage)
tt	Total to total (single stage)
ω	Specific work
x	Axial

1 Introduction

As in the development of the understanding of basic turbomachinery flow, much of the recent knowledge of tip clearance flows has been obtained in cascade models, both linear (Bindon (1987), Dishart & Moore (1989), Yaras et al (1988)) and annular (Morphis & Bindon (1988)).

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None of this knowledge has been applied or verified with fully rotating turbine blades. The only known rotor study relating to tip clearance has been a trenching study by Offenburg et al (1987). Boletis & Sieverding (1991) made a comprehensive comparison of the flows in first and second stage stator and some information can be deduced about the effect of the rotor and its tip clearance.

The present paper describes experiments aimed at relating the nature of tip clearance loss as gained in cascade to the real performance of unshrouded single stage and multistage axial turbines. In a companion paper, Morphis & Bindon (1994), the different flow fields in the first and second stage stators are presented.

In a linear cascade it has been found that tip clearance loss has two main components. The first is the internal gap loss or entropy that is generated within the leakage flow as it passes through the gap. The second is the mixing loss as the leakage flow merges with the mainstream. It in turn should be thought to have two parts, mixing that takes place within the rotor and mixing downstream.

An important part of tip clearance loss control has always been to minimize the gap mass flow. For a fixed area or clearance gap size, the only way to decrease flow is by deliberately introducing separations that "block" the flow and lead to entropy formation. Linear cascade tests (Morphis & Bindon (1992)), show that although this loss be can reduced to 1/5th by simple streamlined tip gap geometries that avoid separation, the leakage mass flow and mixing loss increases to reduce the benefits of the lower internal gap loss.

The development of tip leakage flow and the fully mixed out losses downstream of a linear cascade were measured by Yaras & Sjolander (1989) but in a multistage turbine neither the leakage flow or the secondary flow are fully mixed out before entering the next nozzle. A second stage nozzle will therefore play an important part in the ongoing formation of loss.

In attempting to relate the results to an actual rotor, relative motion between the blade and the outer casing has been simulated in an annular cascade by Morphis & Bindon (1988) and by Kaiser (1993) and in a linear cascade by Yaras & Sjolander (1991). The former found the gap mass flow to be little influenced but the latter reported up to 50% reduction due as much to the change in tip loading as to the change of shear within the gap.
In an attempt to represent actual rotor performance, linear cascade velocity vectors have been converted into rotor vectors so that simulated rotor work transfers and performances are obtained which depend on the actual deflection and entropy rise of the tip region flow (Bindon & Morphis (1992)). Despite this, there is some doubt that the losses measured in a cascade can be accurately translated into a rotor context to provide a reliable statement regarding all the interrelated effects contributing to actual rotor performance. It also appears that tip clearance phenomena are dependent on the blade profile, Yaras & et al (1988) using a true tip section while Bindon (1987) and Dishart & Moore (1989) used existing cascades with thick highly cambered shapes.

This means that the downstream nozzle can be expected to play an important part in the complete formation and definition of tip clearance loss. If the flow can be redirected and expanded without an undue decrease in nozzle efficiency, it may be found that what was previously considered to be mixing loss has been overestimated. Although not specifically noted by Boletis & Sieverding (1991), this may already be concluded to be true since their second stage nozzle did not show any increase of loss due to the negative incidence of the leakage flow or due to the rotor secondary and clearance flows mixing out within the blades.

The behaviour of a one and a half stage turbine have not received much attention from researchers and the only study known is that of Boletis & Sieverding (1991) for a low aspect ratio untwisted design. The results concentrated on flow field measurement rather than on rotor and overall performance.

In order to deepen the understanding of tip clearance loss mechanisms and to assess the possible benefits of reduced entropy generation within the tip gap, this paper therefore measures, at three gap sizes, the performance of three 406mm diameter low speed rotors, each being identical except for the tip gap geometries which are shown in Figure 1. These are the same as those previously used in the linear cascade rotor simulation experiments of Morphis & Bindon (1992).

A standard square tip shape provides a basis of reference since so much of the present understanding relates to it and since it is commonly used in real engines. A tip with a pressure edge radiused to suppress the gap separation bubble is the first of two "streamlined" low gap loss shapes. The second is a gap contoured to avoid a separation bubble while shedding a radial flow component into the gap in an attempt to reduce gap mass flow.

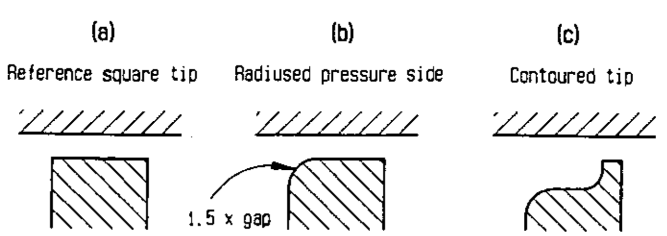

Figure 1 Three rotor tip cross sections as in Bindon and Morphis (1992)

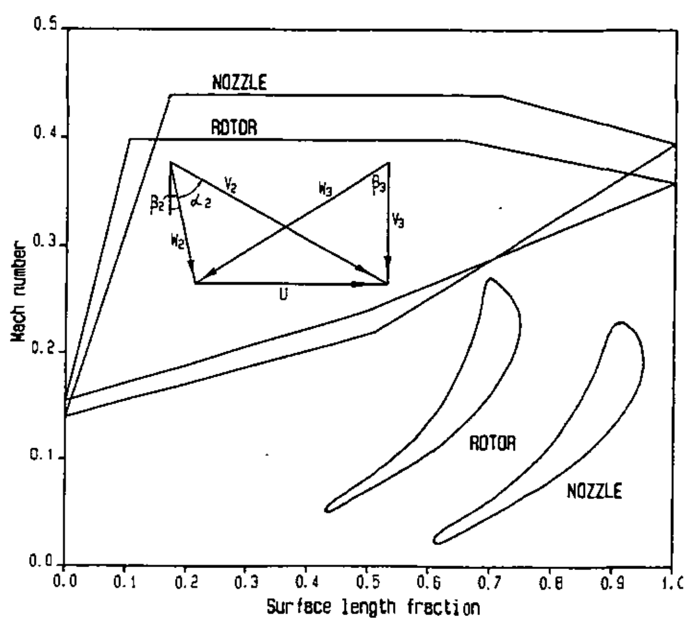

Figure 2 Design mid span Mach number distributions, velocity diagrams and blade profiles

2 The low speed research turbine

A 406mm diameter low speed axial research turbine was used to measure the performance of three rotors. The turbine is followed by a second stator, identical to the first, that is used to measure the second stage nozzle performance and the overall one and a half stage performance. In the companion paper (Morphis & Bindon (1994)) the differing flow phenomena that take place in the second stage are assessed.

The turbine was designed using a commercial package (Northern Research and Engineering Corporation (1975). Table 1 and Figure 2 summarizes the philosophy employed and the main details of the design. Figure 3 shows to scale the cross section of the turbine and defines the position and nomenclature of the measuring stations.

Table 1 Summary of turbine data

	Hub	Mid	Tip
α_1	00.0	00.0	00.0
α_2	66.1	61.7	57.6
α_3	00.0	00.0	00.0
β_2	42.8	13.6	-17.9
β_3	53.1	58.2	62.3
Radius [mm]	142.0	172.5	203.0
Reaction	0.15	0.42	0.58
V_x/U	0.752	0.617	0.526
Nozzle chord [mm]	36.8	41.8	46.5
Rotor chord [mm]	46.3	46.0	46.0
Noz. blade thick. [%chord]	18.5	15.8	14.3
Rot. blade thick. [%chord]	17.3	15.3	15.3

Exit Reynold's number	170 000
Inlet axial velocity [m/s]	29.6
Rotor tip speed [m/s]	56.3
R.P.M	2645
Nozzle Pitch/Chord ratio	0.55
Rotor Pitch/Chord ratio	0.63
Nozzle number of blades	41
Rotor number of blades	43

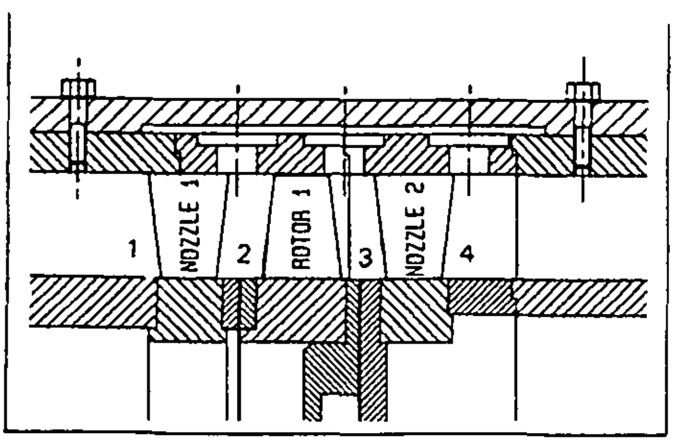

Figure 3 Turbine cross section and measurement planes

3 The turbine and instrumentation

The standard rig tip clearance was 1% of tip chord (.45%mm, .75% span, .11% diam). The two additional clearences of 2% and 3% chord were achieved using two larger diameter rotor casings carefully blended back to the standard dimension in front of and behind the rotor to avoid steps.

Despite the blades being moulded in high strength dimensionally stable epoxy, special curing procedures had to be developed in conjunction with the suppliers to ensure the blade length and hence clearance gap did not change with time. A master rotor (square tip only) and stator blade were NC machined in brass including a pedestal that would form the hub endwall and a chordwise T shaped mounting root. After hand polishing these were used to make multiple moulds for blade production. The moulds for the two additional rotor tip shapes were made by adapting a square tip mould at the tip only thus ensuring that variations only occurred at the tip.

It should be noted that the two low loss blade tips involve radii that are chosen to suppress separation bubbles on the blade tip. These radii have been shown to depend on the gap dimension (Morphis & Bindon (1988)). The experimental program could not afford to make the four extra rotors needed for variations in clearance gap and thus only two rotors were made with radii suitable for the 1% clearence considered to be the dimension more closely resembling current engine practice.

The rotor and stator hubs were made from cast aluminium with milled chordwise T shaped slots for blade mounting.

The rig is driven by a fan powered by a variable speed hydrostatic motor. The turbine power is absorbed by and the speed controlled by another hydrostatic motor. The operating point of the rig can be set and held to within .01% by computer control of the high pressure oil circuits. Turbine torque was measured with a high accuracy (.05%) Himmelstein transducer.

Since all turbine performance definitions involve isentropic quantities that are defined by pressure, the complex fluctuating flow field behind a rotor cannot be determined with a hot wire or a laser. A quick response probe and transducer system was considered but a conventional 5 hole probe that records a mean of the high frequency pressure field was finally adopted because the results from it are widely accepted and used.

The 2D flow field in each traverse plane is measured with a 5 hole United Sensor 3.2mm diameter probe used in the yaw null mode where the yaw angle is set by the automated traverse system responding to the differential yaw pressures and pitch are found from the probe calibration and the pitch hole pressures.

The position of the two nozzle rings relative to each other should be determined by selecting the point where a wake from the first rotor strikes the second rotor. Because of the blurring, skewing and twisting that occurs with axial distance and inside a rotor, the position of a wake is neither exact or unique. Therefore, somewhat arbitrarily, a mid span loss peak, at the traverse plane ahead of the second nozzle, was located at a second stator mid pitch position.

4 Repeatability and optimum V_x/U

The repeatability of the turbine rig and instrumentation is of great importance since small changes in efficiency had to be detected. This was examined by performing full 2D traverses for the same rotor on various days. For a series of 5 runs the efficiencies were found to be repeatable to within .05%. Although encouraging, this result did not however give any indication to the ability of the rig to detect small changes in tip clearance configuration. The second nozzle had a tip clearance of 0.1 mm or .22 %chord necessary for assembly and traversing. An increase in second nozzle efficiency of approximately .3% was measured for all three rotors when this small clearance was eliminated by inserting 0.1 mm shim between the blades and outer casing. The shim was then removed and the efficiency was once again measured to within .05% of previous runs.

A further check on the reliability of the results was provided by comparing the torque transducer based performance with a purely aerodynamic traverse based performance. The two results reflected identical trends and were different only by a constant that was attributed to disc and bearing friction. The single stage efficiency was measured and plotted for various V_x/U ratios. The point of maximum efficiency on the plot was found to correspond to the design V_x/U.

5 Coefficients used to define performance

Two efficiency definitions are commonly used to evaluate turbine performance, namely total to total and total to static. The total to total efficiency assumes that the kinetic energy in the fluid leaving the rotor can be utilized. That would be as if the rotor was able to 'convert' its exit kinetic energy into a useful effect as for example in a perfect down stream nozzle or diffuser. The total to static efficiency assumes that the only the work extracted by the rotor is useful and that all energy in the flow leaving the rotor is lost.

The total to total efficiency was defined as follows.

$$\eta_{tt} = \frac{\omega}{h_{o1} - h_{o3}} = \frac{\omega}{h_{o1} - h_{3is} - \frac{1}{2}V_3^2} \qquad (1)$$

Where ω can be a mechanically averaged quantity based on torque or it can be quantity, like the other terms, that represent an average for the whole flow field. Therefore, for incompressible flow :-

$$\eta_{tt} = \frac{\dfrac{\overline{\rho\omega}}{q_{ref}}}{\dfrac{\overline{P_{o1} - P_{S3}}}{q_{ref}} - \dfrac{\frac{1}{2}\rho V_3^2}{q_{ref}}} = \frac{\overline{C_\omega}}{\overline{C_{P_{o1}-P_{S3}}} - \overline{C_{V_3}}} \qquad (2)$$

Except for torque, each of the various efficiencies and coefficients used to describe the performance of the one and a half stage turbine were formed from a 2D grid (radial and tangential) of probe measurements, each of the 5 hole pressures being recorded simultaneously. Because flow in a turbine varies from moment to moment and because the rig was continuously held at pre-defined values of Reynold's number and V_x/U, each grid data point was converted to a dimensionless coefficient via an upstream reference dynamic pressure recorded at the same time. Thus each grid point may be used to form an integrated result which enables a direct comparison of measurements made at different times and even for different turbines.

The total to static efficiency was defined as follows.

$$\eta_{ts} = \frac{\omega}{h_{o1} - h_{3is}} = \frac{\overline{C_\omega}}{\overline{C_{P_{o1}}} - \overline{C_{P_{S3}}}} \qquad (3)$$

The nozzle efficiencies become.

$$\eta_{N1} = \frac{\frac{1}{2}V_2^2}{h_{o1} - h_{2is}} = \frac{\overline{C_{V_2}}}{\overline{C_{P_{o1}}} - \overline{C_{P_{S2}}}} \qquad (4)$$

and

$$\eta_{N2} = \frac{\frac{1}{2}V_4^2}{h_{o3} - h_{(at\ P_4\ \&\ S_3)}} = \frac{\overline{C_{V_4}}}{\overline{C_{P_{o3}}} - \overline{C_{P_{S4}}}} \qquad (5)$$

The one and a half stage efficiency was defined as

$$\eta_{1.5} = \frac{\omega + \frac{1}{2}V_4^2}{h_{o1} - h_{(at\ P_4\ \&\ S_1)}} = \frac{\overline{C_\omega} + \overline{C_{V_4}}}{\overline{C_{P_{o1}}} - \overline{C_{P_{S4}}}} \qquad (6)$$

Each coefficient can be found from a summation of the 2D flow field, for example

$$\overline{C_V} = \frac{\sum_{i=0}^{m}\sum_{j=0}^{n} C_{V_{ij}} \frac{\dot{m}_{ij}}{\dot{m}_{ref_{ij}}}}{\sum_{i=0}^{m}\sum_{j=0}^{n} \frac{\dot{m}_{ij}}{\dot{m}_{ref_{ij}}}} \qquad (7)$$

It should be noted that the mass flow term has also been rendered independent of operating condition by making it dimensionless with a simultaneously recorded upstream reference mass flow.

6 Effect of tip shape and clearance on single stage efficiency

The total to total and total to static efficiency variations for the three tip clearance gaps of 1%, 2% and 3% chord are given in Figure 4 and Figure 5 for the reference square tip and for the two low internal loss shapes, one with a radiused pressure edge and one with the contoured pressure side.

The most important result is that the tip with the radiused pressure edge has the highest performance and shows an improvement over the square tip blade, 0.2% for total to total and 0.5% for total to static. This means that the reduction in internal gap loss benefits the performance of even a single stage turbine.

This result has widespread implications regarding the understanding and control of tip clearance loss. It appears that internal gap loss (ie the entropy generated within the tip clearence flow) dominates while mixing loss takes place within the rotor to e lesser degree than measured in cascades. Alternatively, the real tip flow involving real relative motion, real blade profiles and real pressure fields gives rise to a real gap mass flow and mixing loss.

Tip clearance loss does not therefore appear to be simply linked to gap mass flow as has been believed for so long and that, as Denton (1993) points out, it is much more complex end depends on entropy generation. Tip leakage flow is not merely flow that "does no work" and a much better conceptual model is needed to manage tip clearance loss. The flow through the clearance gap must be treated like eny other flow within blades and the entropy production minimized rather than maximized as becomes the case when separations are deliberately introduced into the leakage path in order to "block" the flow.

If tip leakage flow provides no work output and if it suffers no entropy increase or mixing, it will leave the rotor undeflected and will not decrease the rotor efficiency since its work potential becomes fully available at outlet. Alternatively stated, loss free undeflected tip clearanca flow reduces the work output but also reduces the driving pressure ratio (ie the isentropic work) and thus the efficiency is not affected.

The superiority of the radiused tip is maintained from the smallest to largest tip clearance evan though the radius chosen for the pressure edge was sized to prevent separation only at 1% chord. As shown by Morphis & Bindon (1988), a radius of 2.5 times the gap width is needed end thus a completely new rotor would be required at each clearance. If the correct radius were used, a slight improvement in performance at the larger clearances may result.

Regarding the contoured blade tip, except at the largest clearance, the performance is better than (total to static) or equal to (total to total) the square tip and below that of the radiused tip. This blade therefore confirms the finding that it is necessary to streamline the tip clearance flow. Previous cascade tests (Morphis & Bindon (1992)) had shown this tip shape to be the best with the same internal gap loss as the radiused tip (1/5th of the square tip) and the 11% lower gep mass flow seemed to reduce the mixing loss by 25%.

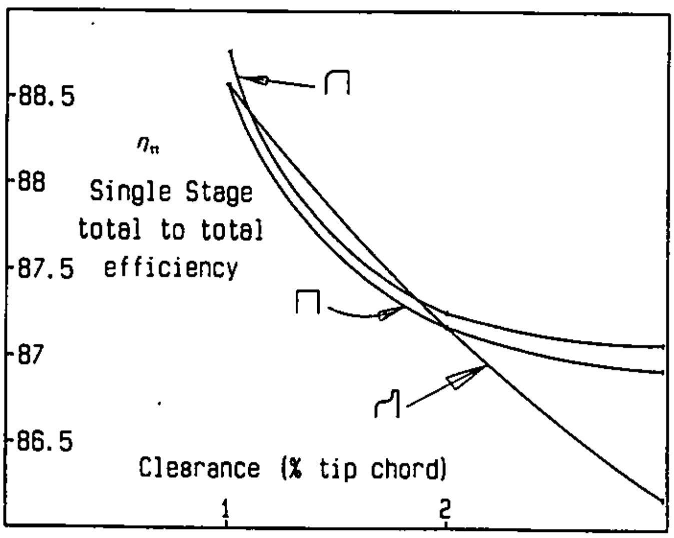

Figure 4 Single stage total to total efficiency for 3 rotors at varying tip clearanca

Since mixing losses can be expected to be lower, the reduced efficiency could be due to the gross destruction of the tip profile necessary to create the large radii required for the contouring and the increase in profile loss offsets some of the gains from internal gap loss and mixing loss.

The variation of efficiency with tip clearance matches very closely with the figure of 1% for each 1% of tip clearance reported by Booth (1985).

In an attempt to explain the superior performance of the two streamlined blade tips, it is necessary to explain why the mixing losses do not appear to take place within the rotor as previously found in cascades. When the cascade exit plane flow fields of Morphis (1989) and Morphis & Bindon (1992) are re-examined, (one example, for the square tip, is re-plotted here in Figure 6) some espects not previously noticed can be reported.

Firstly, there is no evidence of the classical secondary flow vortex and only leakage induced rotation can be seen. It would appear that the leekage vortex is not only much stronger but also that the inlet boundary layer is swept towards the pressure side by the leakage jet on the one side and by the flow entering the gap on the other side. Thus there is far less low momentum fluid on the endwall to respond with cross flow to the tangential pressure gradient.

When viewed from the top as in Figure 7 at one gap width from the endwall, the leakage wall jet can be distinctly seen with a clearly demarcated directional change at the edge of the jet. This edge demarcation is even more distinct in a recent tip clearance computation (Basson & Lakshminarayana (1993) of the same cascade end even the predicted leakage jat directions are similar to a remarkable degree. The leakage flow over the first 74% of chord has obviously lifted up and wrapped into the vortex while the flow over the trailing 37% of the gap is seen to flow diractly to the outlet of the cascade without lifting. The latter 36% of clearance gep carries 49% of the leakage mass flow and this tallies almost exactly with the flow identified by the envelope shown in Figure 6 that is thought to correspond to the unlifted leakage jet. A further 30% of leakage mass flow lies immediately above the flat portion in the other envelope shown in Figure 6.

The above observation is thought to be important because it means that the majority of the leakage flow has not mixed out at rotor exit. The mixing loss and leakage vortex that does occur involves the shear flow at the interface between the jet and the main flow,

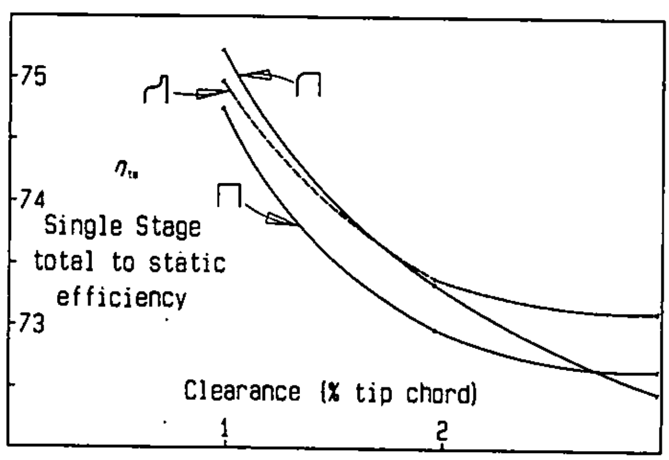

Figure 5 Single stage total to static efficiency for 3 rotors at varying tip clearance

something that is not dominantly mass flow related. The unmixed leakage flow that leaves the rotor will not appear as a rotor debit and the rotor efficiency will be more dependent on internal gap loss than previously thought.

7 The performance of the second stage nozzle

The efficiencies of the first and second stage nozzles for the three tip shapes and for the three tip clearances are given in Figure 8. The second stage nozzle efficiency is seen to be significantly higher in all situations. This is in contrast to Boletis & Sieverding (1991) who found the two loss coefficients to be almost identical.

The second stage nozzle efficiency shows an improvement with an increase of tip clearance between 1% and 2%, so much so that the subsequent decrease at 3% has only brought the performances back to the levels at 1%.

It appears therefore that not only are there no additional losses due to unmixed clearance and secondary rotor flows and due to negative incidence onto the blades near the tip, but that there are strong mechanisms, at times enhanced by tip clearance, which reduce the loss. Since there does not appear to be an obvious mechanism that could affect profile loss, the most likely candidate is the secondary flow loss. This depends on the blade loading and the inlet vorticity which is radically different to that normally associated with the development of endwall loss in cascades. It is not evenly spread across the inner and outer casings but the majority is concentrated in wakes, vortices and leakage jets some distance from the endwalls. The vorticity in the boundary layers on the endwalls are fed to the stator as skewed (overturned) and therefore highly energetic flows. The blade loading, and hence the driving force for the generation of secondary flow, must be lower since both rotor leakage flow as well as rotor secondary flow will appear as negative stator incidence which will unload the blades in comparison with the first stage which has to fully turn the flow. As is discussed in the companion paper, Morphis & Bindon (1994), the reduction in loss is in the hub and tip regions where secondary flow losses are normally seen.

The different shaped rotor blade tip produce different degrees of improvement over the first nozzle. From 1 to 2% clearance, the radiused blade is the best but at 3 % it is well below the other two tip shapes. As with the drop off in performance of the rotor alone at large clearances, this could also be due to the incorrect radius being used on this tip. However, a more important observation is perhaps that flows of this complexity are not easily explained by simple rules and experience gained in cascades.

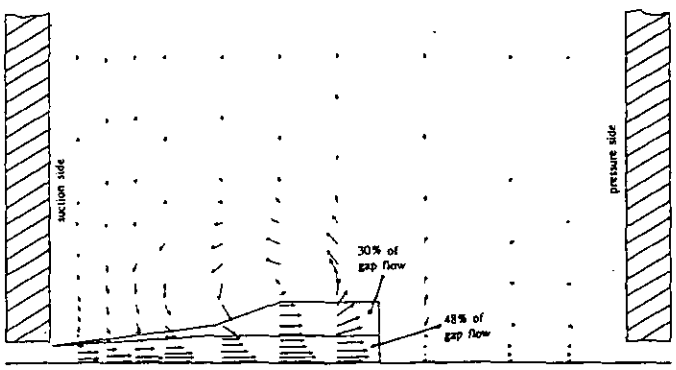

Figure 6 Velocity vectors normal to free stream flow at cascade exit plane (Bindon & Morphis (1992)) identifying possible envelopes that contain the leakage flow

8 The overall performance of the one and a half stage turbine

Unlike the rotor performance which is described by two quantities that have somewhat different meanings, a single unique one and a half stage efficiency has been defined which reflects the losses in all three blade rows without any points of argument as to whether certain effects are included or excluded. This efficiency is plotted in Figure 9, again for all clearances and blade tip shapes.

At the smallest clearance, the radiused tip is again distinctly superior but this drops well below the standard square tip blade at large clearances. Since the rotor alone does not show this, the reason is the poor performance of the second stage nozzle at this clearance.

The contoured tip that outperformed the standard square tip blade at the smaller clearances, is now well below at all clearances. As was ventured to explain its poor performance for the rotor alone, perhaps the loss of blade shape with this tip is simply too gross and increased profile loss is responsible. It is finally worth underlining here the fact that this tip shape had seemed in cascade to be the best, even when the cascade data was specially adapted to reflect rotor processes. It thus points out the complexity of the flows and the dangers of some simplistic models used to make decisions in turbine design.

9 Conclusions

The low speed research turbine and the associated instrumentation and data analysis are believed to be sufficiently reliable and accurate to resolve the minute performance differences resulting from variations in tip clearance flow.

The effect of tip clearance flow and loss phenomena identified in cascades have been explored in a real rotor and a downstream second stage nozzle. Receiving particular attention was the complex interrelationship between gap mass flow, leakage mixing

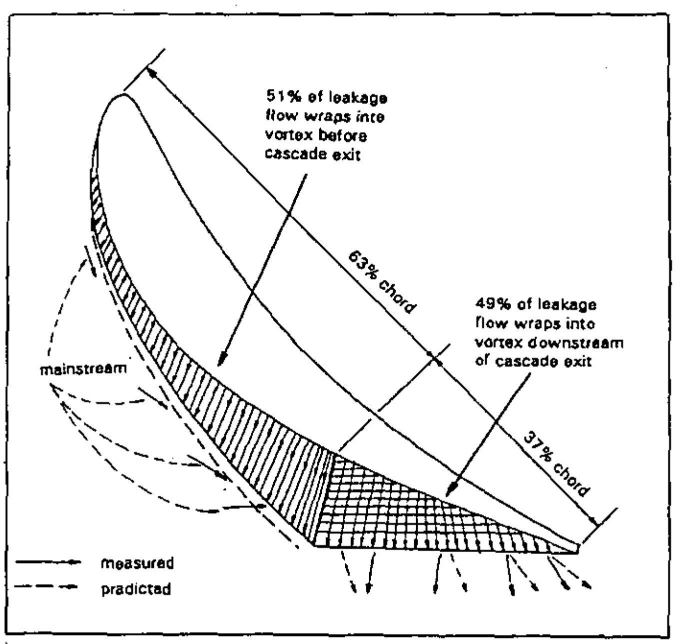

Figure 7 Downstream portion of leakage flow does not wrap into a vortex within the cascade. A remarkably similar flow was predicted by Basson & Lakshminarayana (1993)

loss and the possible benefits of low loss rotor tips without the entropy generating flow separations normally used to "block" leakage.

The rotor blade tip with a simply radiused pressure edge to suppress a separation bubble and reduce the internal gap loss was found to improve the performance of a single stage rotor and of a one and e half stage turbine at the small tip clearance levels for which the tip radius was selected. This is in contrast to what was found in cascade and the result leads to some important conclusions. Tip clearance flow, like any other flow, needs to avoid the entropy generating mechanisms of flow separation. Mixing loss, and indeed all flows in the rotor and downstream of it, are complex and do not readily respond to the simple rules that have been used to design for minimum tip clearance loss.

With respect to the rotor alone, mixing loss appears to be less dominant than was deduced in cascade, probably due to all the effects being correctly modeled. The more likely form of the tip leakage jet was elso seen to be a flat high energy wall jet containing most of the leakage fluid rather than a tip leakage vortex. This jet appears to be readily accepted by the second stage and expanded without further loss formation.

The efficiency of the second stage nozzle was found to be significantly higher than that for the first stage nozzle. This is a surprising result since it was expected that secondary and tip leakage mixing would be evident. An increased clearance actually decreased the loss. Thus the manner in which unmixed rotor flow is processed by a second stage nozzle needs a fuller explanation. The higher efficiencies appear to be due to a reduction in secondary flow loss at the tip and the hub which results from a combination of the lower blade loading necessitated by the pre-deflected flow (ie poorly deflected by the rotor) and of the low momentum fluid being remote from the endwall.

The superiority of the low tip loss rotor blades was not maintained at large clearances and while details such as using the correct blade tip radii for the increased gap sizes needs to be corrected, the increased loss was seen primarily in the second stage nozzle and this complex flow requires a great deal of investigation.

Other future work in this area must obviously be to establish the applicability of the results to machines with real engine Reynold's Numbers and Mach Numbers. The two low loss tips studied were edaptations to en existing blade tip and perhaps advanced design techniques are required that involve not only the whole tip region geometry but also the complex downstream flows that stream periodically from a rotor and interact with the downstream nozzle.

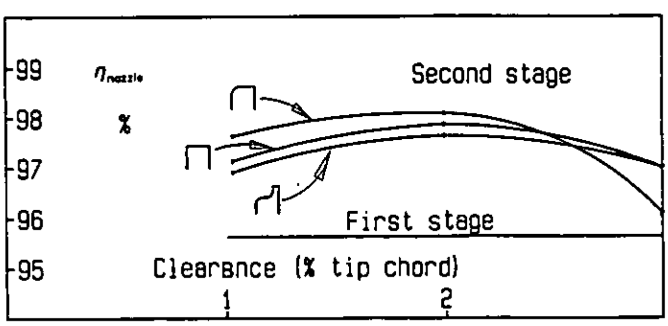

Figure 8 First and second stage nozzle efficiency for different rotor tip shapes and different clearences

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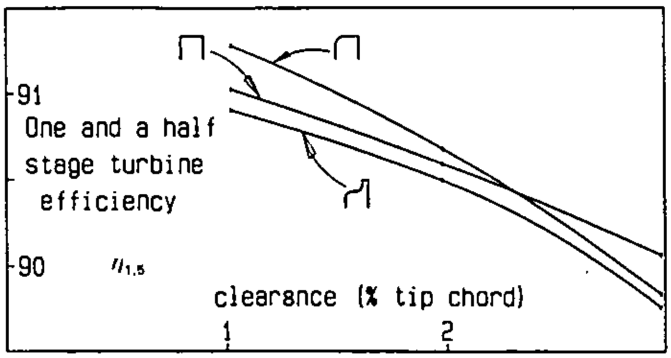

Figure 9 Overall performance for the one and a half stage turbine with different rotor tips and different clearances

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