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THE FLOW IN A SECOND STAGE NOZZLE OF A LOW SPEED AXIAL TURBINE AND ITS EFFECT ON TIP CLEARANCE LOSS DEVELOPMENT

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Abstract

The flow field in a one and a half stage low speed axial turbine with varying levels of rotor tip clearance was measured in order to compare the behaviour of the second nozzle with the first and to identify the manner in which second nozzle responds to the complex tip clearance dependent flow presented to it and completes the formation of tip clearance loss.

The tangentially averaged flow relative to the rotor blade in the tip clearance region was found to differ radically from that found in cascade and is not underturned with a high axial velocity. There is evidence rather of overturning caused by secondary flow. The axial velocity follows an almost normal endwall boundary layer pattern with almost no leakage jet effect. The cascade tip clearance model is therefore not accurate.

The reduction in second stage nozzle loss was shown to occur near the hub and tip which confirms that it is probably a reduction in secondary flow loss. The nozzle exit loss contours showed that leakage suppressed the formation of the classical secondary flow pattern and that a new tip clearance related loss phenomena exists on the suction surface.

The second stage nozzle reduced hub endwall boundary layer below that of both the first nozzle and that behind the rotor. It also rectified secondary and tip clearance flows to such a degree that a second stage rotor would experience no greater flow distortion than the first stage rotor.

Radial flow angles behind the second stage nozzle were much smaller than found in a previous study with low aspect ratio un-twisted blades.

Nomenclature

Variables

$$C_V = .5\rho V^2/q_{ref}$$
$$C_{Po_{1-2}} = (P_{o1} - P_{o2})/q_{ref}$$
$$C_{Po_{3-4}} = (P_{o3} - P_{o4})/q_{ref}$$

C	Dimensionless coefficient
h	Specific enthalpy [kJ/kg k]
i	Incidence [degrees]
$\dot{m}$	Mass flow rate [kg/S]
N	R.P.M
q	Dynamic pressure [Pa]
R	radius [mm]
U	Blade velocity [m/s]
V	Absolute velocity [m/s]
W	Relative velocity [m/s]
α	Absolute angle [degrees]
β	Relative angle [degrees]
η	Efficiency
ρ	Density [kg/m^3]
ω	Specific work [Kj/kg]
θ	Swirl angle

Subscripts

1	First nozzle inlet
2	Rotor inlet (first nozzle outlet)
3	Rotor outlet (2nd nozzle inlet)
4	Second nozzle outlet
i,j,m	Summation grid variables
is	Isentropic
o	Total
P	Pressure
ref	Free stream inlet reference
s	Static
ts	Total to static (single stage)
V	Velocity
x	Axial
θ	Swirl component

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1 Introduction

The flows in second and subsequent stage nozzles have not received much attention from researchers. The second nozzle flow field of a low aspect ratio untwisted design was investigated by Boletis & Sieverding (1991) and the unsteadiness of the flow was measured by Joslyn et al (1983). Apart from a fundamental interest in the type of flow occurring, the second stage plays an important part in the complete definition and formation of tip clearance and secondary loss because these stator blades provide the only way of correctly quantifying the effect of rotor exit energy and vorticity.

In a linear cascade Bindon (1987) it has been found the tip clearance loss has two main components. The first is the internal gap 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 within and downstream of the rotor. Since the leakage vortex radically effects the suction surface pressure distribution, there could also be tip clearance related losses generated in the boundary layer and by boundary layer separation. Using cascade data construed to represent a rotor with work transfer (Morphis & Bindon (1992)), an approximate estimate was obtained regarding rotor performance. However, before any final conclusions can be made regarding the mixing loss and internal gap loss, not only is a real rotor needed, but a downstream nozzle as well to correctly redirect and expand the rotor exit flow.

This paper therefore examines the flow structure in a full annular nozzle downstream of a low speed unshrouded axial turbine. Because of the importance of tip clearance, the gap size is varied. In a companion paper, Morphis & Bindon (1994) examines the performance (efficiency) of all three blade rows with particular reference to different rotor blade tip shapes. It was found that the second stage nozzle has a distinctly better efficiency than the first stage and that a streamlined rotor tip shape, with a low internal gap loss, benefits the performance when compared with the flat sharp edged reference rotor tip. Most of the results in this paper are for the standard flat rotor tip and only minimal reference is made to the small flow structure variations caused by the different rotor tip shapes.

The flow into a first stage nozzle is relatively "clean" and therefore more closely resembles the ideal cascade model with inlet vorticity evenly distributed on the two casing walls. The other nozzles are always downstream of an axial gap and the flow reflects radial and periodic variations in total enthalpy, incidence and vorticity.

Distinct aspects of these differing flows have been examined. The skewing of the inlet boundary layer has been studied by Bindon (1979) and Gregory-Smith (1987). Boletis & Sieverding (1991), by comparing the real second stage nozzle flow with their previous skewed study (Boletis et al (1983)), showed that the skewed model has almost no validity and some similarity exists only near the inlet and close to the endwall. Periodic effects on rotor blade flow (for example Blair et al (1989)), is an ongoing and active area of research in heat transfer.

In the full comparison of the flows in a first and second stage nozzle, Boletis & Sieverding (1991) explored the flow field in front of, within and downstream of the blades. The general conclusions were that flow angles were significantly different, particularly in the tip region that is affected by tip clearance. Although the radial and tangential loss distributions were different due to the changes in the pattern of migration and generation of low momentum fluid, no change in loss was found. This is in sharp contrast with the present findings presented in the companion paper, Morphis & Bindon (1994), where the losses are found to be much lower. More detailed comparisons with the present measurements will be made below.

2 The turbine, instrumentation and definitions

The complete suction driven low speed single stage axial turbine followed by a second stage stator identical to the first is fully described in Morphis & Bindon (1994). It is shown in Figure 1 together with the various traverse stations while Table 1 summarizes the design data.

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

The standard rotor tip clearance was 1% of tip chord (.45%mm, .75% span, .11% diam). The two additional clearances of 2% and 3% chord were introduced with two slightly larger rotor casings carefully blended back to the standard dimension in front the

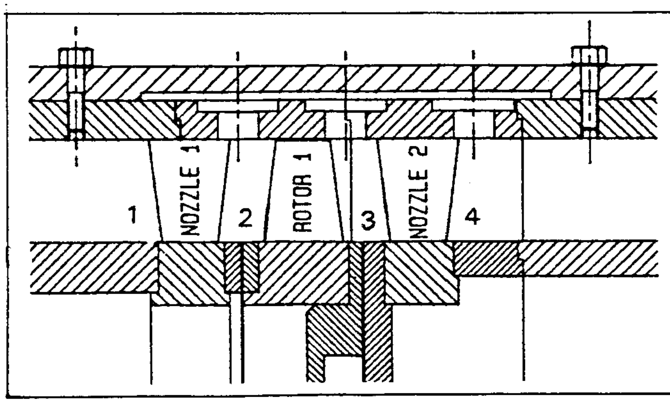

Figure 1 Turbine cross section and measurement planes

second nozzle.

Since the 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 front of and behind the two stators 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 repeatability of the turbine rig and instrumentation 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 0.05% of previous runs. The yaw angle was found to be repeatable to within 0.3 degrees.

The total to static efficiency was defined as follows.

$$\eta_{ts} = \frac{\omega}{h_{o1} - h_{3is}} = \frac{\overline{C_\omega}}{C_{P_{o1}} - C_{P_{s3}}}$$

Where the work coefficient at a particular grid point was

$$C_{\omega\, ij} = \frac{\dot{m}_{ij}\rho_{ij}U_{ij}\Delta V_{\theta_{ij}}}{\dot{m}_{ref_{ij}}q_{ref_{ij}}}$$

$$= \frac{\dot{m}_{ij}}{\dot{m}_{ref_{ij}}} \; \frac{2\pi\rho_{ij}N_{ij}R_{ij}}{60} \; \frac{(V_{2_{ij}}\sin\theta_{2_{ij}} - V_{3_{ij}}\sin\theta_{3_{ij}})}{(P_o - P_s)_{ref_{ij}}}$$

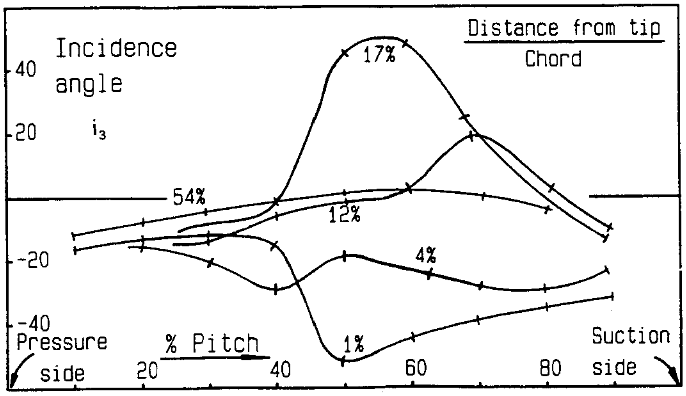

Figure 2 Second stage nozzle incidence angle at various distances from the blade tip simulated from linear cascade data including tip clearance (Morphis (1989))

The coefficients and angles were tangentially averaged for a full nozzle pitch as follows

$$\overline{Y_j} = \frac{\sum_{i=0}^{m} Y_{ij} \frac{\dot{m}_{ij}}{\dot{m}_{ref_{ij}}}}{\sum_{i=0}^{m} \frac{\dot{m}_{ij}}{\dot{m}_{ref_{ij}}}}$$

(where Y_{ij} is a particular coefficient or angle at the (i,j)th grid point)

This has the effect of averaging out the tangential variation that is caused by any stationary upstream blade wakes.

3 Rotor determined second stage nozzle inlet conditions

The second stage nozzle differs from the first in that it is downstream of the rotor. The ability of the rotor to deflect the flow will therefore largely determine the inlet to the second nozzle.

The flow leaving a rotor is periodic and will fluctuate with each blade passing with respect to flow angle and, in the incompressible case, total pressure because of varying degrees of loss and work extraction. Since the present measurements are time averaged behind the rotor, it will be helpful to review the cascade results of Morphis (1989) which are able to show the periodic flow variations that a typical second stage stator must deal with.

Figures 2 and 3 presents the measured cascade exit flow field results transformed to simulate that leaving a rotor by including the hypothetical blade speed. The incidence onto the second stage nozzle (Figure 2) is shown at varying distances from the tip endwall. Remote from the endwall (a distance of 54% chord), the incidence changes from mildly negative to slightly positive as would be expected at "mid-span". Near the tip (a distance of 1% chord) where the leakage jet exists as a strong poorly deflected jet over half the blade pitch, the incidence has almost a mid- span value near the pressure side and then drops sharply to around -40 deg, also for half of the blade pitch. The leakage jet appears as an extremely high velocity zone as reflected in Figure 3 where the velocity coefficient peaks at 3.5 (ie an inlet velocity 1.9 times the mid-span value.

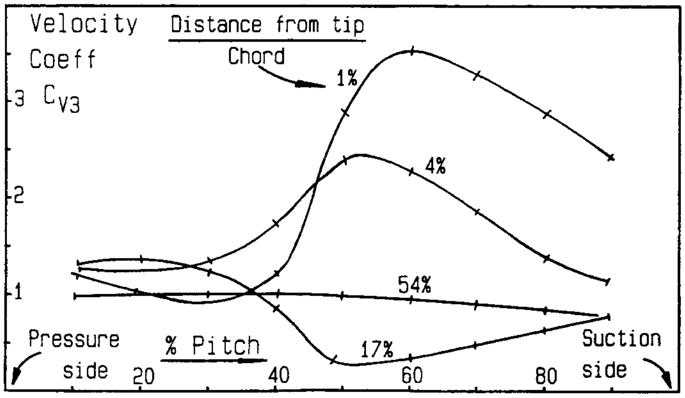

Figure 3 Second stage nozzle inlet velocity at varying distances from blade tip simulated from linear cascade data including tip clearance (Morphis (1989))

At increasing distances from the tip, the incidence changes through zero before becoming positive by as much as 40 deg on the other side of the leakage vortex. The velocity coefficient at this point shows that the nozzle will receive a velocity half the freestream value.

When the above blade to blade simulated rotor flow is tangentially averaged, it can be compared with the present measurements taken at the inlet to the nozzle where the probe responds with a close approximation to the mean of the periodic flow presented to it by the rotor. This is done in Figure 4 as nozzle inlet angle rather than incidence. Both curves are seen to have similar high positive flow angles (negative incidence) near the wall. At a distance of 17% of chord from the tip where Figure 2 shows the positive incidence associated with the leakage vortex, the velocity here is so low that the mass based average does not even show a dip and the angle curve does not even fluctuate at that point.

By design, the first stage inlet angle was zero. The second nozzle is seen to have an inlet angle of near 10 degrees at mid span that increases sharply to 25 degrees near the tip at a clearance gap of 2% chord. At a 1% gap, the increase near the tip is much lower due to the smaller poorly deflected leakage flow. There is also an increase of flow angle near the hub due to the rotor hub secondary flow.

Since the inlet angle is caused by the rotor, it is helpful to examine the same flow in the relative frame and Figure 5 shows the rather surprising fact that near the hub secondary flow creates a large amount of overturning. Even more surprising is the overturning at the tip for 1% clearance where leakage flow is expected to be poorly deflected. Even at the larger 2% clearance there is almost no underturning. Linear cascade results showed the exact opposite and the mean exit flow angle decreases sharply near the wall due to the contribution of the undeflected leakage jet.

The interpretation of this unexpected result is also shown in Figure 5. The tangentially averaged flow near the tip is not a poorly deflected high velocity jet but a low velocity rather well deflected flow. A possible reason for this is that true relative motion in the small gap reduces the leakage effect to such a degree that normal secondary flow can still flourish. This however is probably too simplistic an explanation of this complex periodic tip boundary layer flow in which fluid in the laminar sublayer may even be exposed to more than one blade passing.

The radial variation of efficiency of the single stage will be helpful in understanding the part played by tip clearance as well as in understanding why one streamlined rotor tip was found to improve the performance. Figure 6 shows, at different tip clearance gaps, the "streamsurface" efficiency variation radially where the inlet flow at one radius is related to the outlet flow at the same radius. At the tip the sharp drop off in performance is seen at the larger clearances. In Figure 7, the different efficiency reductions near the tip for the different rotor tip shapes are shown. The gain in performance of the tip with a radiused pressure edge over the reference square tip is seen up to 40% span from the tip.

4 Comparisons between first and second stage nozzles

The response of the second stage nozzle to the flow presented to it may now be discussed and also compared to the behaviour of the first stage nozzle.

In Figures 8 the exit flow angle is shown for the two nozzles and for design. The first nozzle is within one degree of design except for differing secondary flow effects at hub and tip. At the tip some underturning on the inner side of the vortex is evident but no overturning against the endwall. At the hub there is very little underturning while right on the wall there is a distinct angle increase.

The angle behind the second nozzle is shown for 1% and 2% rotor tip clearance. As can be expected, increasing clearance increases the overturning near the tip. Near the hub, the rotor induces a large zone of underturning, probably due to its own hub secondary flow.

When the nozzle exit angles are viewed as the relative angles presented to a downstream rotor, the effect on rotor inlet can be seen by assuming that if a second stage rotor were present, it would be identical to the first. The results appear in Figure 9 and the two curves are very similar with a slightly better angle near the tip of rotor 2 due to tip clearance. Rotors in multistage machines will therefore experience slightly less flow angle distortion when compared to the first rotor.

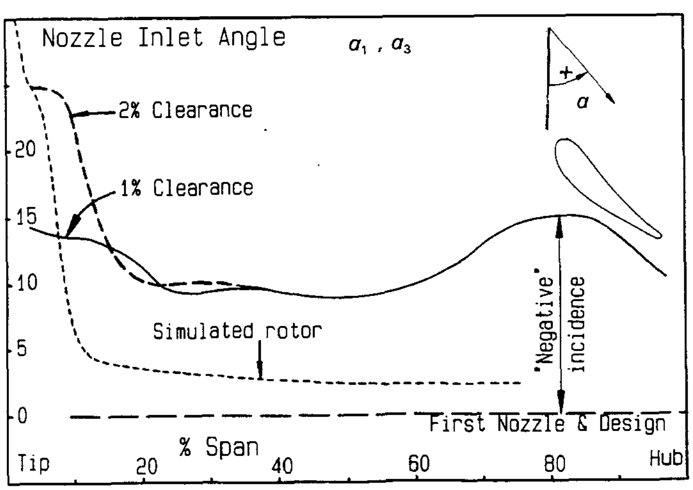

Figure 4 Tangentially averaged nozzle inlet angle (absolute rotor exit angle) for first stator (α_1 = 0 = design), second nozzle (β_3) at varying clearance and for a nozzle behind the simulated rotor of Bindon & Morphis (1992)

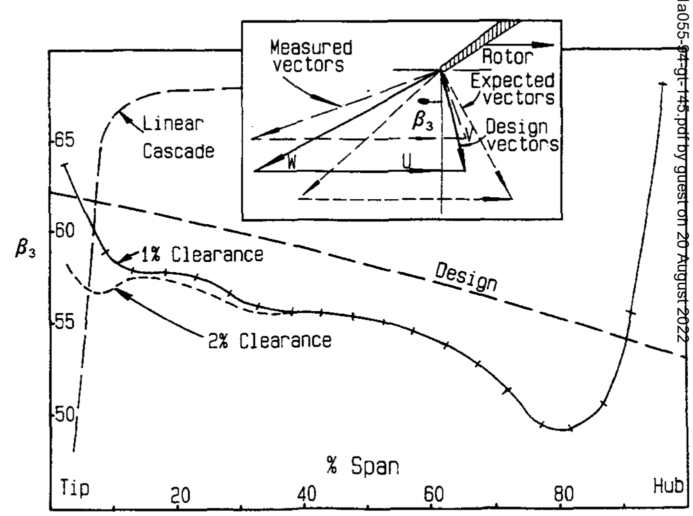

Figure 5 Radial variation of tangentially averaged relative rotor outlet angle at various clearances

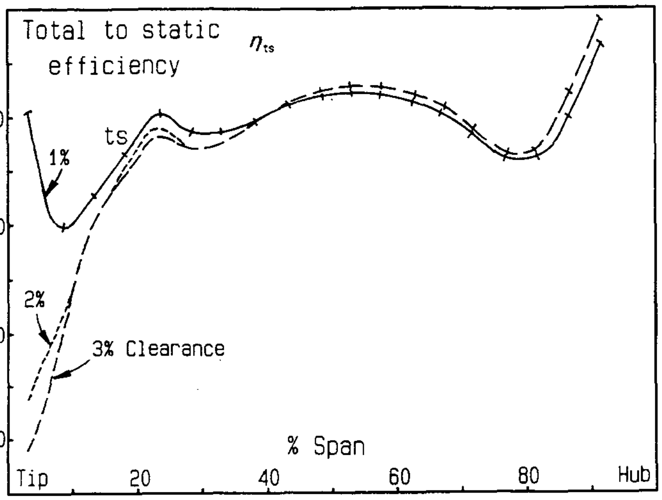

Figure 6 Radial variation of single stage efficiency (ts) for varying tip clearance (square tip rotor)

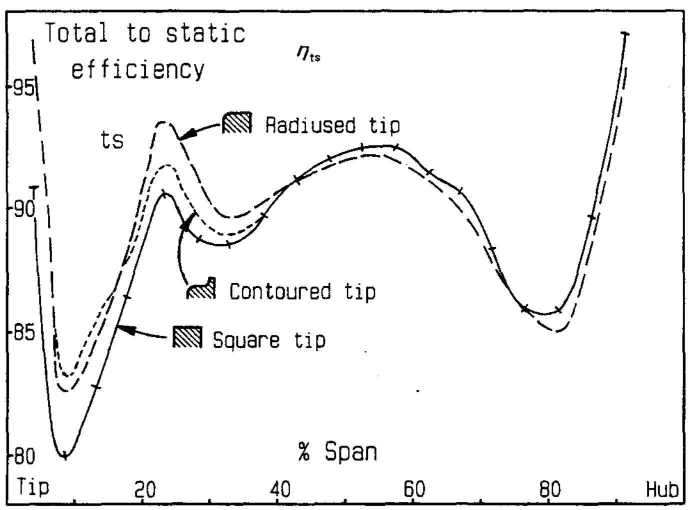

Figure 7 Radial variation of single stage efficiency (ts) for 3 different tip shapes at 1% clearance

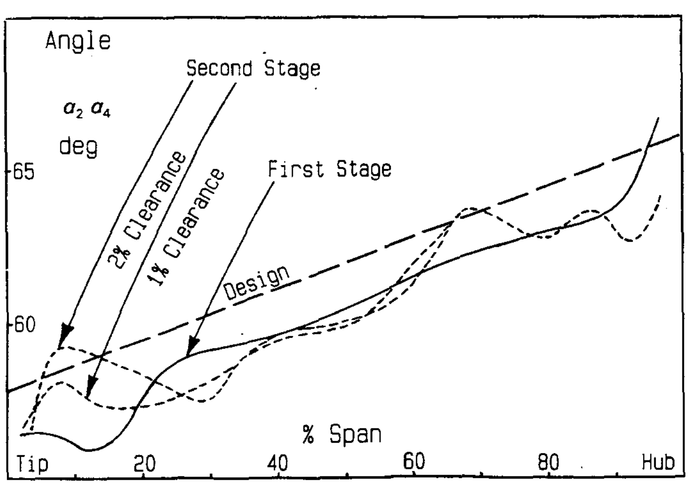

Figure 8 Tangentially averaged radial variation of first and second stage nozzle outlet angle at varying clearance and compared with design

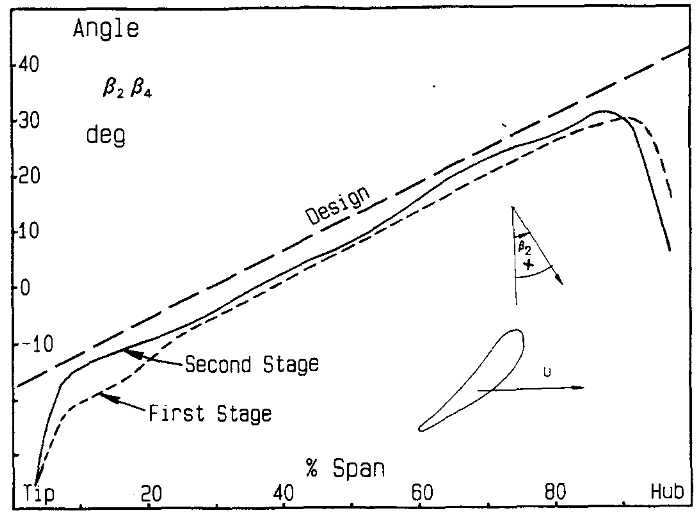

Figure 9 Tangentially averaged radial variation of first and second stage nozzle deflection viewed as relative inlet angle to rotor

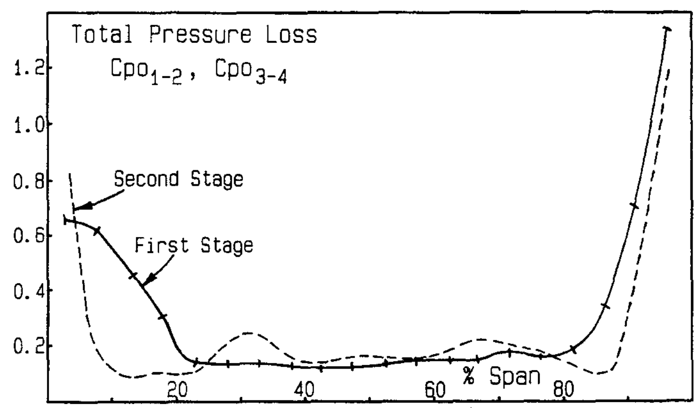

Figure 10 Radial variation in nozzle total pressure loss for first and second stage

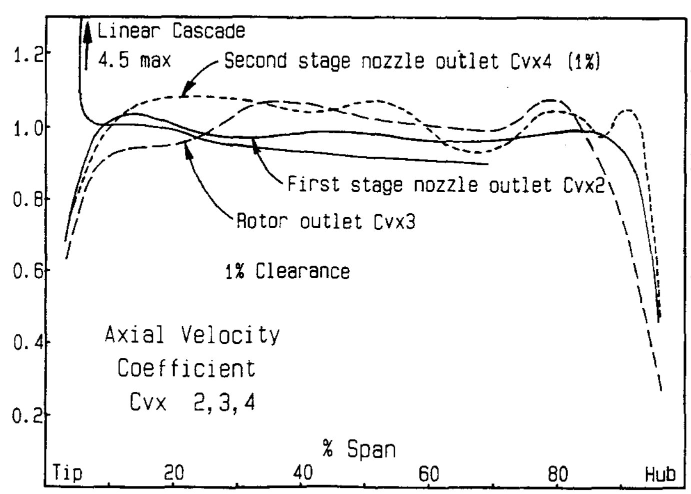

Figure 11 Radial variation of tangentially averaged axial velocity coefficient behind each blade row for 1% rotor tip clearance

In the companion paper, Morphis & Bindon (1994), the losses in the second nozzle were shown to be in the region of 30% below those of the first nozzle. When the radial variation of loss is examined in Figure 10, (on a stream tube basis as with efficiency in Figures 8 and 9), the improvements are seen to be concentrated in the hub and tip regions. This tends to confirm the reasons advanced that the improvements are due to reductions in secondary flow loss. The factors which generate secondary flow are all radically different. The blade loading is lower because at inlet the endwall boundary layers, the leakage flow and rotor secondary flows are skewed and therefore are already partially deflected. The inlet vorticity is not evenly spread across the inner and outer casings but the majority is some distance from the endwalls.

The unexpected relative rotor flow angles discussed in the previous section were deduced from the absolute frame measurements of angle and velocity. The radial variation of axial velocity coefficient is given in Figure 11 at the outlet to each of the three blade rows for 1% tip clearance and also for the linear cascade of Morphis & Bindon (1992). The linear cascade data is radically different from the real rotor and the velocity increases dramatically against the endwall while the real distribution follows the normal endwall boundary layer pattern. It is this discrepancy that also created the unexpected relative flow angles at rotor exit. Thus the linear cascade provides a very poor model of real turbine tip processes.

At the hub, the second stage nozzle has a thinner boundary layer than the first nozzle, a reduction that takes place despite being presented with a relatively thick layer by the rotor. At the tip, all three curves are similar up to 10% span from the tip. Between 10 and 30% the effects of tip clearance are seen by a reduction at rotor exit that is more than restored by the second nozzle. Although not as quite as marked, this was also seen at 2% and 3% clearance. Thus the secondary and tip clearance flows fed to the second stage nozzle appears to benefit the axial velocity distribution considerably. At mid span, the second nozzle shows an increase over the first nozzle that matches well with that expected due to compressibility effects in a constant area annulus.

When a comparison is made between the two nozzles of the radial variation in absolute axial velocity, Figure 12 also shows the effects of compressibility. At the tip, the second nozzle demonstrates a region energized by the tip clearance flow by an amount that shows almost no variation with tip clearance.

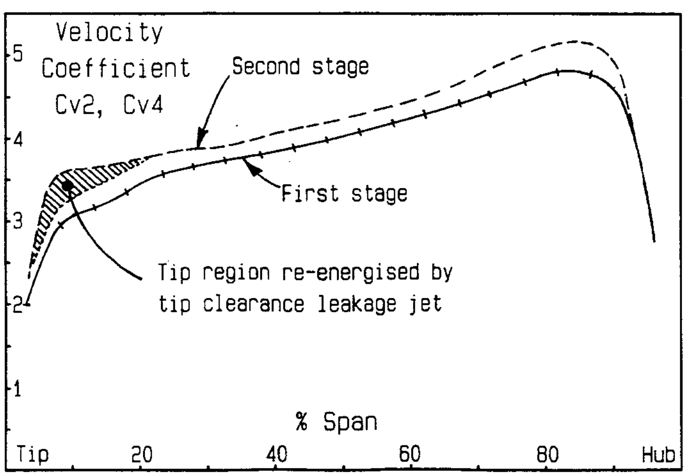

Figure 12 Radial variation of first and second stage nozzle outlet velocity coefficient (tangentially averaged, curves similar for all clearances)

5 Radial flow angles

Boletis & Sieverding (1991) reported large radially inward (negative) flow angles behind the second stator as high as -32 degrees, the magnitude of which was partly attributed to the radial gradient of total pressure arising from constant angle blading. Such blading will also create an inward streamline shift due to radial equilibrium. The blades also has a low aspect ratio which would form more secondary flow fluid that responds to the radial pressure gradient.

The maximum angles in the present study are only from -5 to -6 degrees and all occur at 27% span from the tip at rotor exit and second nozzle exit (Figure 13). There is very little increase in radial flow in either of the nozzles and the largest increase takes place as the first nozzle wake moves through the rotor.

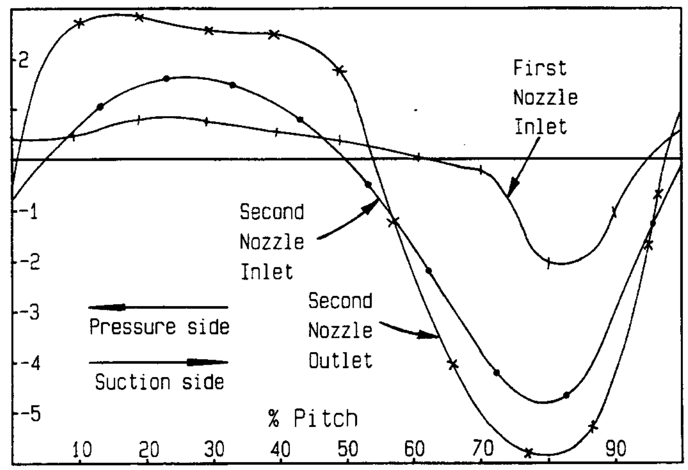

Figure 13 Tangential variation of second stage nozzle pitch (radial flow) angle at 27% span from tip where angles are the highest

6 Nozzle exit plane loss contours

Figure 14 compares the total pressure loss contours at the exit planes of the two nozzles, the latter showing the effects of tip clearance. The plots are based on an inlet averaged total pressure.

The significant reduction in overall loss coefficient found in the second stage nozzle (Morphis & Bindon (1994)) as compared to the first was attributed to differences in secondary flow loss. As already seen in the radial loss variation, the differences lie within 20% span zones at hub and tip. The familiar loss areas associated with the corner vortices can be seen in the first nozzle and are almost absent in the second stage nozzles.

An unusual loss pattern is seen on suction side of the second nozzle blade wakes at a 30% span distance from the tip. It increases in size and prominence with increasing tip clearance. It was also present, slightly more distinctly, with the special rotors with streamlined tip shapes. It is possibly a separation pattern induced by the tip leakage flow. Its nature and importance were not studied and adaptations are required to provide data within these stator blades.

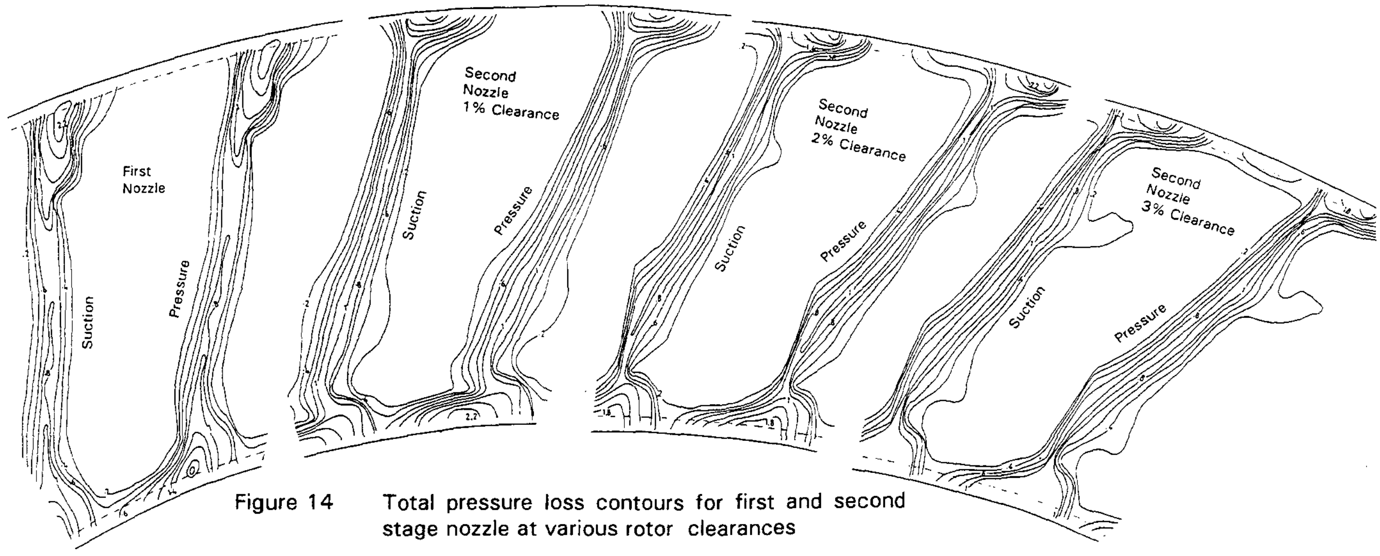

Figure 14 Total pressure loss contours for first and second stage nozzle at various rotor clearances

7 Conclusions

The study of the flow field in a one and a half stage low speed axial turbine with varying levels of tip clearance has revealed a number of interesting aspects of tip clearance flow and of flow in a multistage machine.

At the rotor tip the tangentially averaged flow as was found to differ radically from that seen in cascades where a region of underturned flow with a high axial velocity was found. Instead, the real rotor shows a region of overturned flow with a normal endwall boundary layer pattern. The cascade model of tip clearance cannot therefore accurately represent the real flow and a great deal of further investigation is needed.

The most important result of the comparison between the first stage and second stage nozzles was the confirmation that the significant reduction in loss in the second stage is in fact due to reductions of secondary flow. The radial variation of total pressure showed reductions in loss at both hub and tip. The nozzle exit loss contours showed that leakage suppressed the formation of the classical secondary flow pattern. Also seen in the contours at second nozzle exit was a new tip clearance related loss phenomena on the suction surface that needs further investigation.

The second stage nozzle also showed that it was able to thin the hub endwall boundary layer below that of both the first nozzle and that behind the rotor. It also was able to rectify secondary and tip clearance flows to such a degree that a second stage rotor would experience no greater flow distortion than the first stage rotor. The small amounts of leakage related overturning that persists right through the second stage nozzle near the tip thus has very little effect of rotor inlet angle.

Radial flow angles were much smaller (-6 as compared to -32 deg) than previously found by Boletis & Sieverding (1991). The difference is thought to be due to the present use of lower aspect ratio twisted nozzles blades.

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