DEVELOPMENT OF TIP CLEARANCE FLOW DOWNSTREAM OF A ROTOR BLADE WITH COOLANT INJECTION FROM A TIP TRENCH

Debashis Dey
Department of Aerospace Engineering
The Pennsylvania State University
University Park, PA 16802
dxd30@psu.edu

Cengiz Camci
Department of Aerospace Engineering
The Pennsylvania State University
University Park, PA 16802
cxc11@psu.edu

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ABSTRACT

Rotational frame velocity and pressure measurements were made downstream of a rotor blade row in a single stage turbine research rig . Measurements were taken at two axial locations to track the development of the secondary flow in the blade tip region using a five hole probe. Coolant mass flow was injected at several locations on the rotor tip to investigate the effect of coolant air on the secondary flow. The ultimate objective is to reduce losses by the introduction of high momentum air in the tip gap. Results indicate that the passage and the leakage vortices retain their structures up to at least 46 % chord-length downstream of the rotor trailing edge. The cooling air, which is 0.3 % of the total mass-flow, appears to be well mixed with the leakage flow and makes little difference to the flow field downstream of the rotor. It appears that a relatively large amount of cooling air ought to be injected from specific locations on the rotor tip to cause any significant change in the flow field. This study summarizes the early stages of a multi-year investigation on the fluid mechanics and heat transfer aspects of turbine tip leakage flow structures.

NOMENCLATURE

c Rotor axial chord-length at mid-span
h Rotor blade height
U_m Mean wheel speed, i.e. midspan rotor speed
α Absolute flow angle measured from axial direction
β Relative flow angle or yaw angle measured from axial direction
w Relative velocity
v Absolute velocity
N Rotor speed (RPM)
p_t Total pressure
p_s Static pressure
p_{atm} Ambient pressure
p_0 Reference pressure, atmospheric pressure
Re Reynolds number
r, θ, x Radial, tangential and axial coordinates

INTRODUCTION

The flow within turbine blade passages is unsteady and three dimensional. The relative motion between the moving rotor and the stationary casing and stator contribute to the unsteadiness. The high turning that the flow undergoes and the non-uniformities in various flow quantities contribute to the three-dimensionality. The unavoidable tip gap that exists in unshrouded blades complicate the three dimensional picture. The small leakage flow through the tip gap has an effect on the loss and heat transfer disproportional to its size. Tip related losses might account for as much as a third of the total losses in a stage (Booth, 1985). The desire to raise the cycle temperature to maximize cycle efficiencies has made it imperative to understand the flow field and heat transfer distribution in the blade tip region.

Most studies in secondary flows have been performed in cascades. Cascade experiments provide detailed measurements of the flow field, only at the expense of significant idealizations. These experiments, however, do seem to capture most of the flow physics. Experiments in rotating rigs are closer to the actual engine conditions, and are far more desirable. The present study at the Penn State Axial Flow Turbine Research Facility is an attempt to extend some of the observations made in the cascade experiments to rotating rigs.

The motivation for this study comes from the cooling requirement of the rotor tip. Both the nozzle guide vane and the rotor blade need to be cooled adequately in order to maintain the material temperature below the melting point. The mean value of the heat transfer coefficient could be as high as 5 kW/m^2K for the high pressure stage blades. Additional coolant injection from the tip section of the turbine rotor complicates the aerodynamics of the passage flow structure. This complicates the aerodynamics of the passage flow. It is necessary to examine the effect of the coolant air on secondary flow and losses.

The flow inside a turbine passage is fairly well understood, and there is good agreement among investigators regarding the major flow structures to be found inside the passage (Gregory-Smith, 1997). Figure 1 shows a steady snapshot of the local flow near the tip gap with no discrete vane wake effect. D represents the dividing streamline on the end-wall between the flow that leaks through the tip gap and the flow from the pressure side towards the suction side that forms the passage vortex even without tip clearance. Hence D would coincide with the pressure side end-wall corner had there been no tip gap. A small separation bubble (b1) is shown on the blade tip, and its presence is inevitable in tips with sharp corner. Usually there is a second recirculation zone (b2) on the end-wall. This model of the flow field is important in order to understand the various losses related to the tip leakage flow.

It is now generally understood that the tip clearance loss has two major components (Morphis and Bindon, 1994). The first is related to the entropy generated inside the gap as the fluid passes through it. The second part is the mixing loss as the fluid merges with the main flow. Part of this mixing occurs inside the passage and part of it occurs downstream of the blade row. Losses inside the gap are primarily the friction losses in the walls, and this part is supposed to be small when there is no cooling. Heyes and Hodson (1993) are of the opinion that the flow from the tip gap forms a jet with a loss-free core. Mixing losses are more significant. Moore et. al. (1995) postulates that this loss proceeds in several stages. The gap mean flow works against the turbulent stresses and this generates turbulent kinetic energy. Thus energy is transferred from the mean field to the turbulent field. The turbulent kinetic energy is subsequently dissipated into heat by viscous action. Energy is, in effect, extracted from the mean flow and this gives rise to losses. Clearly, any technique aspiring to reduce losses has to consider either the reduction of the leakage mass flow or has to manipulate the interaction of the leakage jet with the main flow.

A few studies have focused on the interaction between the passage and the leakage vortex, as this interaction is the main source of loss downstream of the trailing edge. Yamamoto (1988) examined the flow downstream of a cascade in which both the tip clearance and the incidence was varied. His results seem to show that the passage vortex location is unaffected by the presence of the leakage vortex. Chan et al (1994) examined this issue in greater detail in a cascade with low turning and hence with relatively weak secondary flow. Part of their intention was to isolate the tip leakage phenomena and assign loss percentages

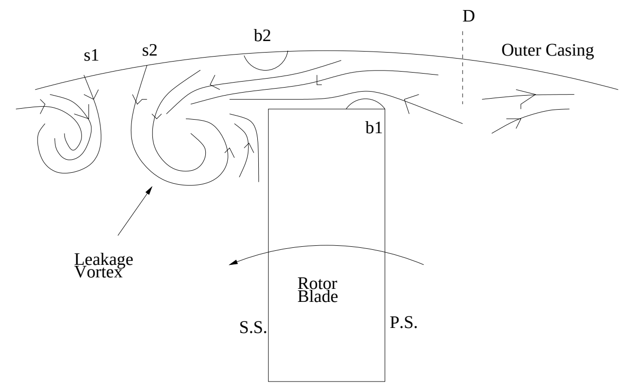

Figure 1. Model of tip clearance flow field

to various sources: inlet, profile, secondary and tip leakage. Peters and Moore made measurements at several locations downstream of a cascade and found results in agreement with Chan et al (1994).

Coolant injected from the tip of the blade, besides aiding the convective heat transfer of the tip, could be expected to play a role in reducing the losses. Coolant from the tip is usually injected at a radially outward direction in most current tip injection systems. However, recent computational studies show that tip injection at an angle to the vertical direction that provides a coolant trajectory towards the pressure side may be beneficial in reducing tip related total pressure losses. The effect of this injection on the separation bubble that clings to the pressure side corner will be investigated in the heat transfer studies in the future. However, one expects that the high momentum coolant jet would block the way of the leakage flow, and this would effectively reduce the tip mass flow. It is well known from many cascade studies that a smaller tip gap significantly reduces the size of the vortex and corresponding losses. The coolant fluid could be expected to end up inside the leakage flow eventually, as the large pressure difference across the blade tip would force the coolant fluid to move with the leakage jet.

Apparatus

The Penn State Axial Flow Turbine shown in Figure 2 was designed to carry out detailed experimental investigations of the flow in a turbine stage. This open circuit type facility has a 1.1 m diameter inlet which smoothly contracts the flow down to 0.916 m diameter. The test section consists of a constant diameter casing containing the first stage stator and rotor. The span of the flow passage h, that is, the distance between the hub and the casing, is 0.123 m. The rotor has a mean chord c of 0.129 m. The stator has 23 blades and the rotor has 29. The flow then passes through an annular row of guide vanes into two axial fans and is discharged from the rig. The power produced by the turbine is

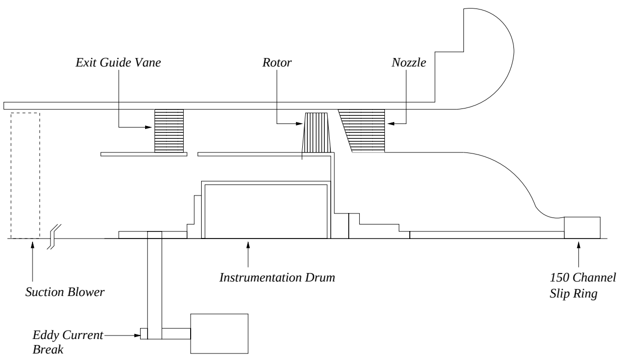

Figure 2. Schematic of the Penn State Axial Flow Turbine Research Facility (AFTRF)

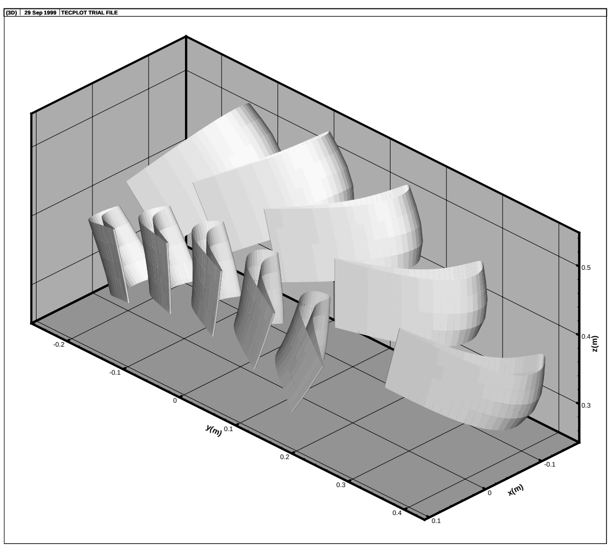

Figure 3. Isometric view of the nozzle and rotor

dissipated by an eddy current brake, which is also used to maintain the rotor at a constant speed. An isometric view of the nozzle and rotor blades is given in Figure 3.

Aerodynamic instrumentation available to the facility for steady state measurements include sensors for the measurement of inlet temperature, total and static pressures, flow velocity and flow direction. A probe traverse mechanism is mounted inside the rotating drum for experiments in the rotating frame of reference . This traverse can move the probe in radial and circumferential directions, while a set of mounting screws can fix the traverse along different axial locations. The data goes out of the rotating frame through a set of slip rings into a dedicated computer for data acquisition. A similar arrangement exists for taking data in the stationary frame of reference. All measurements reported in this paper were taken in the rotating frame of the turbine.

A five hole probe of 1.4 mm diameter was traversed radially and circumferentially at two axial locations, 38% and 46% of the chord length, downstream of the rotor. The axial positions are hereafter referred to as 1.38c and 1.48c measured from the leading edge plane of the rotor. The five static pressures measured by the sensor were converted to the three velocity components, total and static pressure by using a procedure described in Treaster and Yocum (1979).

A test set consisted of a number of circumferential traverses at different radial positions. About 120% of a blade passage was traversed for each radial position. The rig had to be stopped and restarted for every new radial position, for mechanical reasons. The aim of the program was to document the secondary flow over the top half of the passage, and hence traverses were made between 50% and 96% of the span.

Four sets of tests were made. Baseline data at each of the two axial locations consisted of the dry case (no cooling). Subsequently cooling air was turned on and another set of data was taken. The cooling flow was made to choke at the mass flow measuring orifices. This ensured that the coolant mass flow is solely a function of the stagnation pressure and stagnation temperature inside the rotating drum.

The rig rotational speed was maintained at a constant value within ± 1 r.p.m. throughout a traverse using an eddy current break. Inlet air temperature was also maintained within 3 K. Results were cast in a non-dimensional form, as the test conditions like ambient pressure varied from traverse to traverse. The pressures were reduced to pressure coefficients based on turbine inlet total pressure. The velocities were referenced to the midspan wheel speed. Flow angles were measured from the axial direction.

The nominal rotational speed of the rotor is 1320 r.p.m. The the midspan wheel speed U_m is 54.6 m/s. The axial velocity at the rig inlet is 29 m/s. The Reynolds number based on the nozzle vane midspan chord length is 350,000. The nominal turbulence intensity is 1% at the rig inlet, and the boundary layer thicknesses are 0.05h at the hub and 0.1h at the tip. The average tip clearance of the rotor blades is 0.014h (1.72 mm). The trailing edge thickness of the blades is 2.4 mm. The design mass flow is 11 kg/s, and the power output is 60.6 kW.

Figure 4 shows the velocity triangle at the tip section. Figure 5 shows the 2D section of the rotor blade at the tip. Details about the blade design, as well as the rig, could be found in Lakshminarayana, Camci et. al. (1996).

Five of the rotor blades have provisions for coolant injection from the tip. A view of the cooling passage is given in Figure 6. There are four such cooling holes in each blade, arranged in a straight line. Three of them have a diameter of 0.5 mm and inject towards the pressure side, as shown in the figure. The one near the trailing edge is different from the rest. It injects radially, and

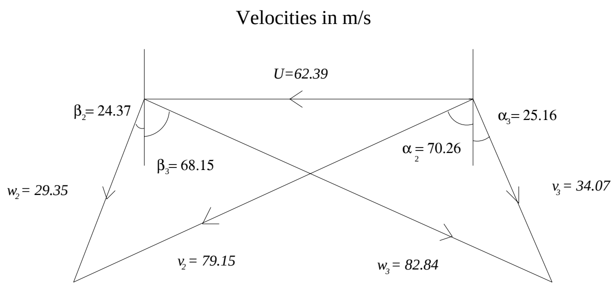

Figure 4. Velocity Triangles at the Tip Section

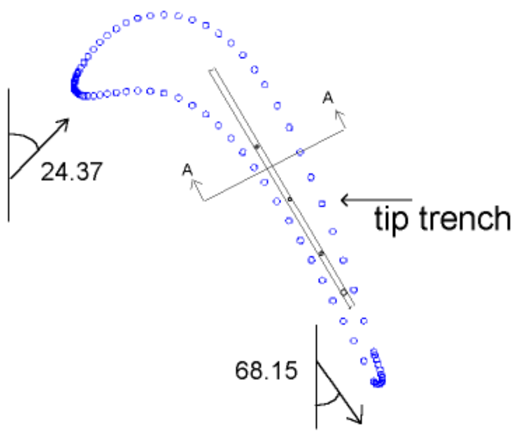

Figure 5. Coordinates of the Tip Section

has a diameter of 1.8 mm. The plenum chamber at the bottom of the blade supplies air for all the four holes.

Figure 7 shows a tip cooled blade in the rotor of the turbine. Tangential probe traverse slots used in the current study are also shown. Figure 8 shows a closeup view of the tip trench.

The rig underwent major modifications to include the rotating air transfer system. A schematic of the air transfer system is given in Figure 9. Regulated air from the plenum (outside the rig, not shown in the figure) was passed on to the stagnation chamber formed by the stationary and the rotating drums. The air from the rotating chamber was then transfered to the blades by flexible tubing. Orifice mass flow-meters were installed in the tubing for measuring mass-flow in the rotating frame. Seals A and B, placed between the stationary and the rotating drums, ensured the minimization of coolant leakage. These seals worked against plasma coated surfaces on the rotating components. The plenum outside the rig was held at 2.5 bars during the runs, and ensured that all the subsequent throats in the air passage were choked. The nominal coolant mass flow in each of the blades was 1.1 g/s. Hence, had there been injection from all the 29 blades, the

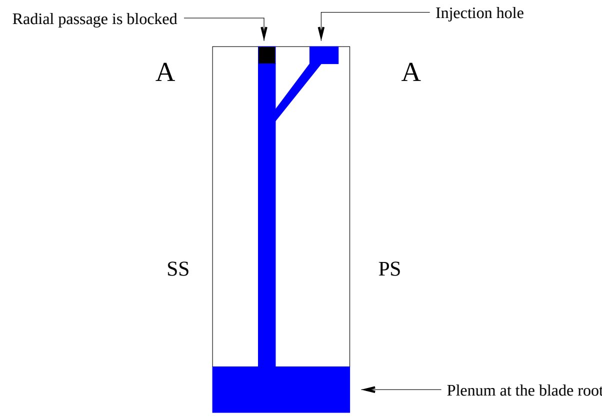

Figure 6. Coolant holes on the blade, corresponding to section AA in Figure 5. (figure not to scale)

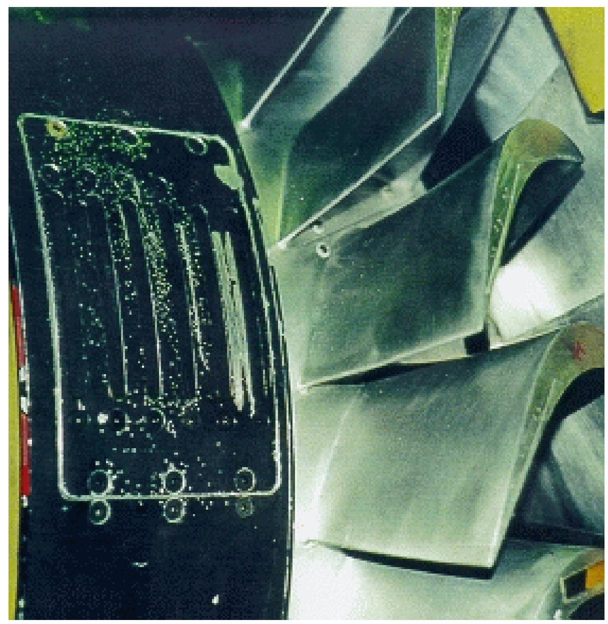

Figure 7. A View of the Rotor Blade Row

coolant mass flow would have been 0.3 % of the total mass flow.

Results

All aerodynamic results from five hole probe measurements are obtained in a plane defined by the radial direction and the tangential direction of the rotor. The measurement plane is normal to the rotational axis of the turbine. The probe is aligned with the relative exit flow direction at the measurement location.

Figure 1 shows our understanding of a typical flow-field n-

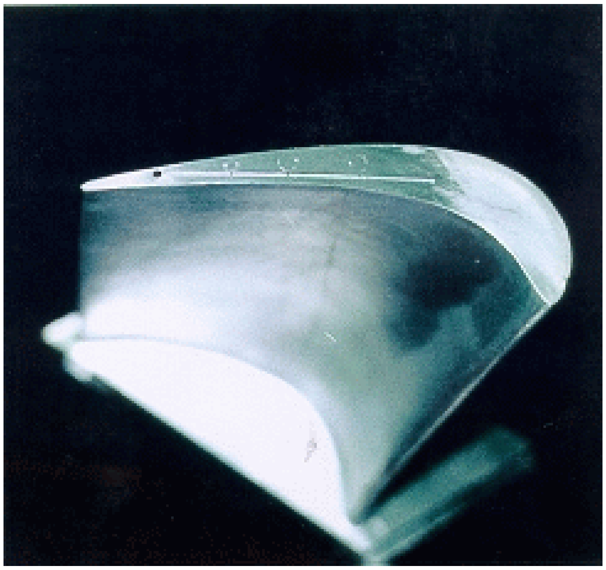

Figure 8. Coolant Holes on the Blade

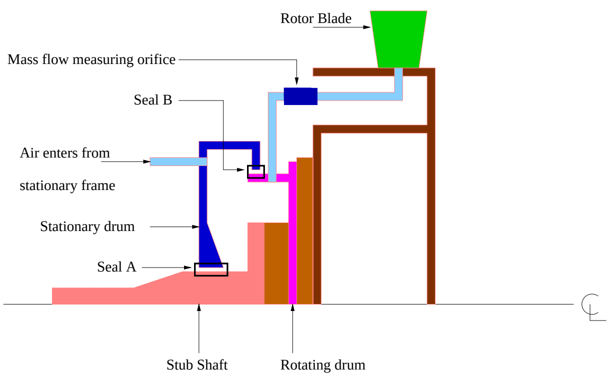

Figure 9. Schematic of the air transfer system

ear the tip end-wall (after Sjolander, 1997). There is a separation streamline at the end-wall (D), and flow on one side accelerates across the tip gap towards the suction surface. The vorticity in this flow contribute to the formation of the leakage vortex. Flow on the other side ends up contributing to the passage vortex. The leakage and the passage vortices rotate in opposite directions. This picture changes somewhat downstream of a blade row. The leakage vortex remains at roughly the same pitch, while the passage vortex keeps moving towards the suction surface and eventually ends up to the right and below the leakage vortex.

Figure 10 shows the measured secondary velocity vectors superimposed on the total pressure loss contours. This represents the measurements taken in a plane 0.38 of the chord length downstream of the rotor exit plane. Pressure loss is indicated simply

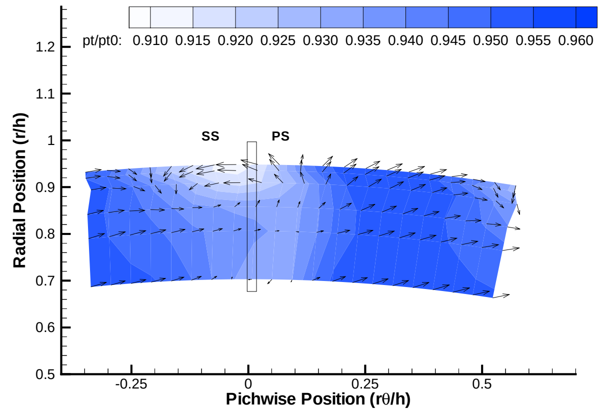

Figure 10. Loss Contours and velocity vectors at $1.38c$ (tip clearance=1.4% h)

by the total pressure referenced to the ambient pressure. The vertical axis in this, as well as all subsequent contour plots, represents the non-dimensional radial location, referenced to blade height h. Hence the casing is along an arc at 1.0 in this units, and the hub is at 0.0. The horizontal axis represents non-dimensional pitch, again referenced to the rotor span. About 120% of the passage is visible. The location of the blade trailing edge is shown. There are two well defined vortical structures. The more prominent tip leakage vortex is centered around 88 % span. This vortex is highly elongated, and appears as an ellipse rather than a circle. This elongation is possibly due to the large stresses in this part of the flow that stretches the vortex in the pitch-wise directions. The less prominent passage vortex is centered around 70 % span. The passage vortex is weaker, but is spread out over a much greater area compared to the leakage vortex. Only the top half of this vortex is visible in this figure.

The two counter-rotating vortices are clearly interacting, and this interaction leads to stresses, turbulence generation and entropy generation, all of them leading to significant losses in mean kinetic energy of the passage flow.

The peak of the loss contour do not coincide with the center of the leakage vortex. The loss peak occurs at a higher span, closer to the end-wall. This may be due to the fact that the leakage vortex has a strong interaction with the outer casing of the turbine.

The location of the wake from the trailing edge can be predicted from the data shown in Figure 10. The wake related losses are spread over a larger area at this downstream location. The losses measured in the tip vortex area are relatively higher than the losses in the wake region. In any case the vorticity in the wake is supposed to get dissipated at downstream axial locations.

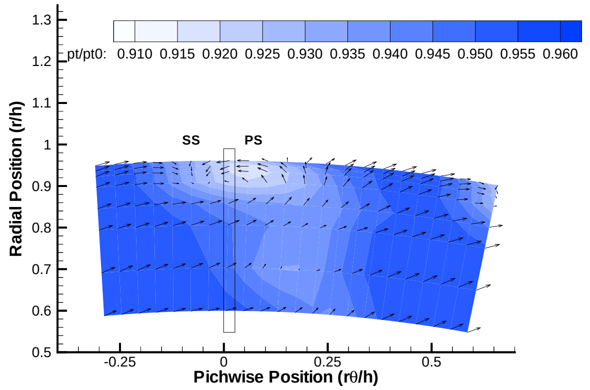

Figure 11. Loss Contours and velocity vectors at $1.46c$ (tip clearance=$1.4\%\ h$)

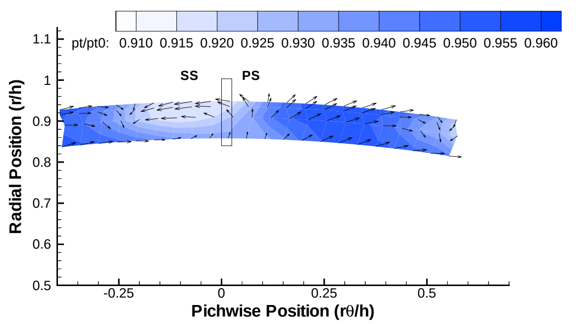

Figure 12. Loss Contours and velocity vectors at $1.38c$ with cooling (tip clearance=$1.4\%\ h$)

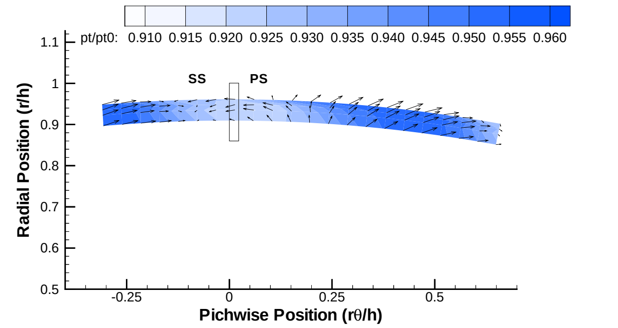

Figure 13. Loss Contours and velocity vectors at $1.46c$ with cooling (tip clearance=$1.4\%\ h$)

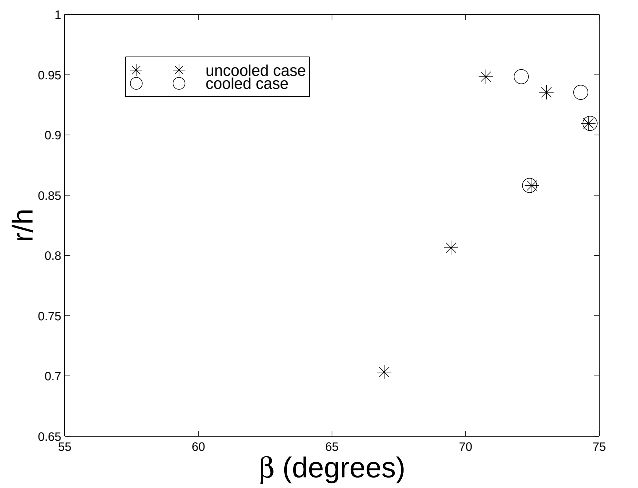

Figure 14. Yaw Angle β distribution in radial direction at $1.38c$

Figure 11 is the traverse made at 46 % of the chord from the trailing edge. The pattern in loss contour as well as the velocity vectors are similar. The whole pattern have shifted towards right, which indicates that the vortical structures, while being convected by the main flow, retains their coherency. The leakage and wake related losses are distinctly visible. The center of the leakage vortex has shifted to about 85% blade height, so it is moving away from the outer casing.

The leakage vortex core is away from the loss core. The loss is clearly spreading: same color contour lines encircle larger areas.

Figure 12 shows the area traverse at the $1.38c$ location with cooling turned on. The cooling flow, reasonably enough, does not have any effect away from the tip, and the velocity vectors at 90% radial location downwards seem to be identical to the no cooling case. Hence one assumes that the passage vortex is unaffected by the cooling flow. The recirculation is somewhat less severe. The leakage vortex seems to be slightly elongated in the pitch-wise direction. The loss contour seems to be a shallower, may be by 0.5 %. One gets the impression that a shallower loss core is occupying a larger area, and the area averaged total loss might still be the same as the uncooled case.

Figure 13 shows the flow-field at $1.46c$ with cooling on. The flow-field is similar to the one at $1.38c$, and one finds a fast spreading of the loss core.

Figure 14-15 shows the pitch averaged yaw angles at various radial locations. The findings support the above observation that the cooling flow does not alter the overall characteristics of the three dimensional mean flow measured downstream of the rotor.

Figure 16 shows the flow-field at the upstream location with a tip gap reduced to 0.6 % span (as opposed to the tip gap of 1.4% span in all the tests cited above). The vorticity content in

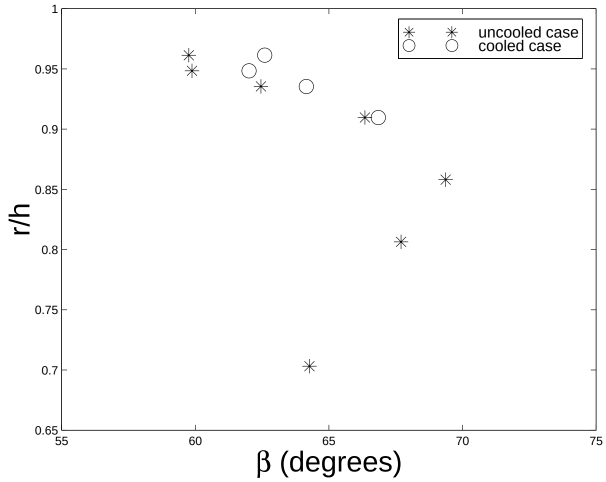

Figure 15. Yaw Angle β distribution in radial direction at $1.46c$

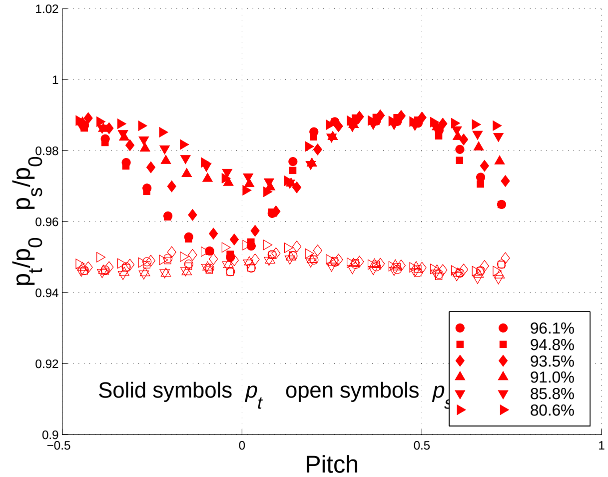

Figure 17. Total and static pressure distributions at $1.38c$

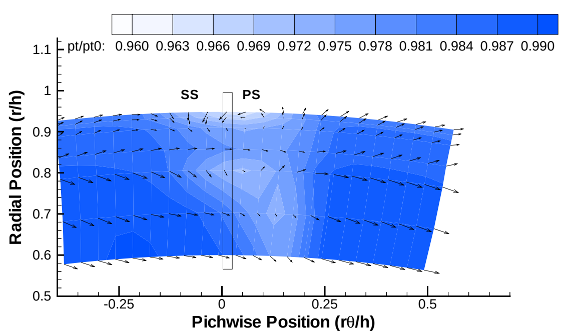

Figure 16. Loss Contours and velocity vectors: tip clearance = $0.6\%\ h$

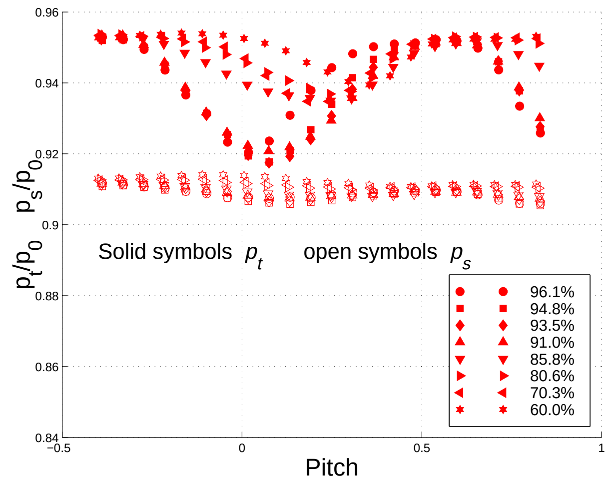

Figure 18. Total and static pressure distributions at $1.46c$

the tip vortex is much smaller as expected from this enlarged tip gap case. The loss core is closer to the end-wall. The loss core may also be a lot shallower compared to above, as shear stresses are less.

Figure 17 shows the pitch-wise distribution of total and static pressures at $1.38c$. The reluctance of the static pressure to change across the passage is obvious. The difference between the two pressures, especially near the tip, points to the very low kinetic energy of the flow near the leakage vortex core. Figure 18 shows the pitch-wise distribution of total and static pressures at the $1.46c$ location. Figure 19 shows the velocity vectors at the $1.38c$ traverse plane. The high turning that the flow experiences near the leakage vortex is evident.

Concluding Remarks

Cooling is indispensable in turbine rotors. It is also expensive, since the cooling air is high pressure air that does no work. One would like to use cooling air for reasons other than heat transfer if possible; for instance, it could be used for the reduction of total pressure losses. The present study attempted to find whether coolant could be used for reducing the losses associated with tip leakage flow. The idea was to block the way of flow entering the tip gap, hence reducing the leakage mass flow. Results indicate that the concept might prove beneficial, but a relatively large amount of coolant is necessary to make significant improvements. Results from rotating frame measurements indicate that the leakage vortex is much stronger than the passage vortex,

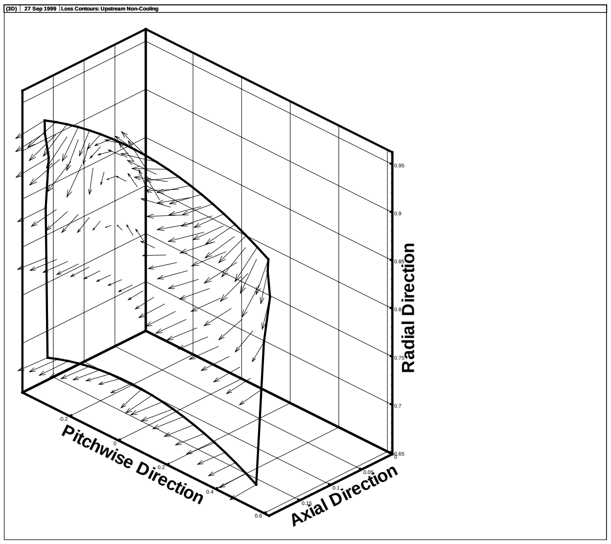

Figure 19. Velocity Vectors at $1.38c$

and is expected to be the most dominant vortical structure entering the downstream stage. Efforts are being made to increase the coolant mass-flow rate as well as to measure heat transfer characteristics of the flow. This study summarizes the early stages of a multi-year investigation of the fluid mechanics and heat transfer aspects of turbine tip leakage flow structures.

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