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Flow In An Idealized Turbine Tip Gap

by

DEKUN CAO

B. Eng. (Mechanical Engineering)

A thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfilment of
the requirements for the degree of
Master of Engineering
in Mechanical Engineering

**Department of Mechanical and Aerospace Engineering
Ottawa-Carleton Institute
for Mechanical and Aerospace Engineering
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FLOW IN AN IDEALIZED TURBINE TIP GAP

submitted by Dekun Cao
in partial fulfilment of the requirements for
the degree of Master of Engineering
in Mechanical Engineering

Thesis Supervisor

Chairman,

Department of Mechanical and Aerospace Engineering

Carleton University

May 1993

Abstract

A survey of the open literature on the flow within the tip clearance in turbomachines reveals the lack of understanding of the flow within the tip gap. Consequently, a large scale single blade test section was designed and constructed to pursue this subject further. Simplicity and versatility were prime consideration in the design, and the performance of the test section was found to be satisfactory.

Detailed surveys of the velocity magnitude, flow direction, and total pressure within the gap were supplemented by surface flow visualization, and static pressure measurements on the endwall and blade tip. Four clearances of from 29.2% to 66.7% of the blade thickness were examined.

The experimental results reveal that for the tip clearance size used in practice, the static pressure distributions are very similar. Thus the driving pressure applied to the gap flow is essentially independent of the tip gap height.

Substantial losses were observed within the separation bubble on the blade tip. The bubble appeared to cover about 30% to 40% of the tip gap height. It was found that before the separation bubble on the blade tip reaches maximum, the tip gap flow is almost free stream flow. After that, there is considerable mixing between the inviscid fluid and the fluid from the shear flow near the blade tip.

Based on the experimental results, Moore and Tilton's (1988) Model was examined, and it was found that the model can provide a good prediction on the tip gap flow.

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Dedication

To my mother

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Nomenclature

A	= area
C_d	= discharge coefficient
C_p	= $(P - P_d)/(\frac{1}{2}\rho V_d^2)$ = static pressure coefficient
$C_{p_{exit}}$	= static pressure coefficient on the endwall at the gap exit
$C_{p_{min}}$	= minimum static pressure coefficient on the endwall
C_{p_o}	= $(P_o - P_{oin})/q_{gap}$ = total pressure loss coefficient
C'_{p_o}	= mass-averaged total pressure loss coefficient (Equation 5.5)
C_q	= Q/q_{gap} = dynamic pressure coefficient
H	= δ^*/θ = boundary layer shape factor
h	= span
l	= characteristic length
$\dot{m}$	= mass flow rate
$\dot{M}_{gap}$	= non-dimensionalized mass flow rate through the gap (Equation 5.3)
P_d	= static pressure at the test section inlet centerline
P_{exit}	= static pressure on the endwall at tip gap exit
P_o	= total pressure
P_{oin}	= total pressure at test section inlet centerline
$(P_{sc1} - P_{sc2})$	= contraction pressure difference of the wind tunnel
q_{gap}	= $\frac{1}{2}\rho V_d^2(1 - C_{p_{exit}})$ = effective gap dynamic pressure
Q	= dynamic pressure

R	= radius from the curvature centre of the blade to the static taps
Re	= $\rho V_{cl} t / \mu$ = wind tunnel operating point
$Regap$	= $\rho V_{gap} t / \mu$ = gap Reynolds number
t	= blade thickness
u, v, w	= components of velocity in x, y , and z directions, respectively
$\bar{u}_e$	= average velocity at gap exit
$\bar{u}_{max}$	= average velocity at the interior plane
u_{max}	= velocity at the interior plane
V_{cl}	= test section inlet centerline velocity
V_{gap}	= $V_{cl} \sqrt{1 - C'_{pexit}}$ = nominal gap velocity
x'	= $R - 438.5(mm)$ = co-ordinate along $\alpha = 45^\circ$
x, y, z	= co-ordinates in tip gap (Figure 5.3.1)
α	= angle measured from inlet flow direction
τ	= tip gap height
ρ	= density
μ	= viscosity
$\xi = u - iw$	= complex velocity in the tip gap
δ^*	= boundary layer displacement thickness
θ	= boundary layer momentum thickness
σ	= contraction ratio
ζ	= $(C_p - C_{pmin}) / (1 - C_{pmin})$ = non-dimensional coefficient for static pressure on the endwall

Chapter 1

Introduction

In turbomachinery stages, a clearance must be kept between the casing and the blade tip so as to prevent the casing from interfering with the blade's rotation. The existence of a tip clearance is known to result in a significant reduction in the performance of a machine due to the following reasons:

- (1) Unloading toward the blade tip results in a reduction in the blade force and therefore, the work done;
- (2) Mixing of the tip gap flow with the main stream causes substantial total pressure losses;
- (3) Off-design inlet flow conditions in the subsequent stage reduces the machine performance in a multi-stage turbomachine.

In addition to these effects, in compressors the tip leakage flow can reduce the machine's stability and lead to early surge. Also, the high heat transfer rates, associated with regions of high local velocity in the tip gap can result in 'burnout' at the tip of turbine blades in cooled engines.

From a survey of the open literature, it is evident that a lot of research has been done on the tip leakage flow. However, due to the very small scale of the tip gap in the test facilities of the past studies, either in a rectilinear cascade or for a single-blade configuration, very detailed and accurate data on the flow within the tip gap have not yet been obtained. This is surprising since information from within the tip gap

is crucial for understanding the complex tip leakage flow, leading to more realistic prediction models and more effective design. From the literature, it also can be found that one of the best ways to reduce the negative effects of the tip leakage flow is blade tip treatment. However, few studies have been conducted in this area.

The mechanism of the tip leakage flow is known to be influenced by the following factors:

- (1) The size of the tip gap;
- (2) The driving pressure difference of the tip leakage flow;
- (3) The relative motion between the blade and the casing;
- (4) The thickness of the endwall boundary layer;
- (5) The clearance gap geometry and the presence of measures to control the tip leakage, e.g. tip injection;
- (6) The existence of the radial pressure gradients in the blade passage.

As can be seen, it is almost impossible to simulate all the aspects of the real tip leakage flow in a rectilinear or a single-blade test section. Therefore, it is advantageous to simulate the governing factors of the flow. As to the flow within the tip gap, the most important feature is believed to be the driving pressure difference across the gap, together with the chordwise mainstream velocities on each side of the gap.

With the intention of making a meaningful contribution in understanding the tip leakage flow, a large scale single-blade test section was designed and constructed. The design was intended specifically for the detailed investigation of the tip-leakage flow within the tip gap, but an attempt was also made to make the test section as versatile as possible, so that it could be used for other flow studies.

The test section consists of a single-blade, which is essentially a quarter of a circle with rectangular extensions on both ends. The wind tunnel Reynolds number operating point based on the blade thickness and the test section inlet velocity was kept at 7.8×10^4 except for the experiments where Reynolds number effects were explored. It must be noted that the compressibility effects were not simulated since the test section inlet velocity is about 15m/sec when the wind tunnel operating point is at 7.8×10^4 . Other idealized features of the current arrangement include the absence of adjacent blades with tip leakage flow, the relative motion between the blade and

the endwall, and the centrifugal effects that exist in real machines.

Surface flow visualization was conducted on the blade tips and the endwall. The static pressure measurements were performed on the blade tip and the endwall. Detailed seven-hole pressure probe measurements were performed within the tip gap, giving the tip gap velocity magnitude, flow angles and total pressure distribution. Using the results of seven-hole probe measurements, the mass flow rate, the mass-averaged total pressure loss through the tip gap, and the size of the separation bubble were calculated and the variation of these parameters with tip gap height was examined.

Finally, on the basis of the experimental results, one of the tip leakage flow models available in the literature (Moore and Tilton, 1988) was examined.

Chapter 2

Literature Review

2.1 Introduction

It is widely recognized that tip leakage has a significant effect on the aerodynamic and thermal performance of gas turbines. For example, losses from the leakage flow can account for as much as one third of the losses through a turbine stage (Waterman 1986, as referred to by Yaras et al. 1989). Also, high heat transfer rates, associated with local regions of high velocity in the gap flow, have been a source of 'burnout' of turbine blades (Bindon, 1987b).

The flow in the tip clearances is very complex, involving vortices and separated regions, and has been investigated by a number of researchers. Studies of this flow have been conducted in this Department over a number of years, e.g., Sjolander and Amrud (1987), Yaras et al. (1989), Yaras and Sjolander (1990), Yaras and Sjolander (1992), Yaras et al. (1992).

In a survey of the known studies, it has been found that although lots of experiments have been conducted, most research has concentrated on the problem of loss prediction and correlation, rather than on the basic flow mechanisms. Yaras et al. (1989) appear to be the first researchers who took detailed measurements within the tip clearance gap. More limited measurements were made within the tip gap by Graham (1986), Bindon (1986, as referred to by Bindon 1989), Storer and Barton (1990),

and Heyes and Hodson (1991). However, measurements within the tip gap are still quite few. Information obtained is insufficient to develop a full understanding of the tip leakage flow.

A very complete review of the studies of tip leakage flow has been completed by Yaras and Sjolander (1991). Therefore, only a brief summary of earlier work is given here. Interest is focused mainly on the work after 1991 with particular attention given to work directly related to the present study.

2.2 Tip Leakage Flow and Tip Gap Height Effects

It is known that the flow in the tip gap is very complex, involving vortices and separated regions. Sjolander and Amrud (1987) appear to be the first authors to propose the structure of tip leakage flows, as shown in Figure 2.2.1.

As the flow moves radially up the pressure surface and then turns sharply through 90° to enter the tip clearance gap, a separation bubble would form just at the corner of the pressure side, as shown in Figure 2.2.1. However, on the basis of careful observation of smoke flow visualization and detailed static pressure measurements on the blade tips, Bindon (1987a, 1987b) suggested that the separation bubble on the blade tip did not form at the edge but a short distance inside the clearance gap. This would force the leakage flow at entry to have small radii of curvature and give rise to low pressures, high surface velocities and high heat transfer. Thus, this flow phenomenon could explain the blade tip 'burnout' occurring in cooled turbines. Another aspect of the separation bubble on the blade tip observed by Bindon is that the internal bubble flow on the blade tip might have a chordwise flow due to the fairly strong pressure gradient in the chordwise direction. This observation was confirmed by Yaras et al.'s (1989) measurements, which showed that the separation bubble on the blade tip was in fact a separation vortex. However, in many aspects, the behaviour of the vortex is very similar to that of a separation bubble. Bindon also noted that at small tip clearances no separation bubble appeared inside the blade tip.

Depending on the tip gap height, and the flow conditions in the gap, the separated flow on the blade tip may or may not reattach onto the blade tip. Storer and Barton

Figure 2.2.1 Tentative Interpretation of the Tip Wall Structure for 2.86% Clearance (reproduced from Sjolander and Amrud (1987))

(1991) investigated the tip gap flow of a compressor cascade by varying the blade thickness to gap height ratio t/τ . Their results showed that the separated shear layer behaved as if it were a turbulent layer at the point of separation, and therefore reattachment was dictated primarily by pressure gradients and inertia forces. Thus, the reattachment length was independent of Reynolds number. They also found that the flow was unlikely to reattach if the blade thickness to gap height ratio t/τ was less than about 1.5. Gearhart (1966) considered a tip leakage flow of shrouded underwater propulsors and also found flow reattachment to appear at 2.5 tip gap heights from the pressure side corner. Graham (1986) varied the tip clearance height on a water flow turbine cascade rig. Once again, separation occurred on the blade tip, and the velocity profiles obtained near the gap exit showed the flow having reattached onto the blade tip for the three tip clearance settings, $t/\tau = 2.5, 3.5$, and 5.2 . These results are consistent with those of Yaras et al. (1989) who reported that reattachment occurred on the blade tip for clearances from $t/\tau = 1.79$ to 6.56 .

Figure 2.2.2 shows the development of the separation bubble as proposed by Bindon (1989). He suggested that high losses would appear in the region where the bubble leaves the gap. Yaras (1987) did observe a small region of high mass-averaged total pressure loss located about 30% blade chord from the leading edge. However, because the blade was strongly forward loaded, this high loss actually resulted from the inlet boundary layer fluid driven into the gap.

In addition to the separation bubble on the blade tip, a second bubble has been observed within the tip gap by a few researchers. Through oil flow visualization, Sjolander and Amrud (1987) found a pair of small, nearly two-dimensional separation bubbles on the endwall. They ascribed this bubble to the adverse pressure gradient toward the exit of the gap which results from the convergent-divergent passage formed by the separation bubble on the blade tip. The flow visualization of Moore and Tilton (1988) also revealed a separation bubble on the endwall within the clearance space. Based on hot wire probe measurements inside the gap, Heyes and Hodson (1991) speculated about the presence of a separation bubble within the tip gap on the endwall. These three studies seem to be the only ones of the known studies reporting the existence of the endwall separation bubble.

Figure 2.2.2 Development of Clearance Gap Separation Bubble
(reproduced from Bindon (1987b))

Figure 2.2.3 Schematic of the Vortex Structure around the Tip
(reproduced from Shun Kang and Hirsh (1991))

Outside the separation bubble on the blade tip and before the *vena contracta*, the flow is essentially two-dimensional and inviscid. This was noted first by Rains (1954) and was confirmed by Booth et al. (1982), Yaras et al. (1989), and Storer and Barton (1991). In the remainder of the tip gap there is considerable mixing between the inviscid fluid and the fluid from the shear flow near the tip. This appears to be the most difficult part of the flow to model.

After emerging from the gap, the leakage flow mixes with the main stream flow and wraps into a vortex. This is the leakage vortex, the size and the strength of which is related to the tip gap height. Yaras and Sjolander (1990) found that the size and the strength of the vortex were reduced with the decreasing of the tip clearance height.

As to the rotation of the tip leakage vortex, the conclusions of Bindon (1987a), and Shun Kang and Hirsh (1992) are consistent with Sjolander and Amrud's (1989) observations. However, three different versions of the location of a tip leakage vortex and the formation of a secondary vortex were obtained. Sjolander and Amrud (1987) observed that the tip leakage vortex separated on the endwall along S1, and the passage vortex, i.e., the secondary vortex, separated along S2, as shown in Figure 2.2.1. They also suggested the existence of another vortex near the suction side of the blade. The directions of the vortices shown in Figure 2.2.1 were determined by smoke trace flow visualization. In discussion to Sjolander and Amrud (1989), Williamson (1982) presented his and Moustapha et al. (1987) observation of flow visualization, which supported Sjolander and Amrud's (1989) vortex structure. Bindon's observations on the tip leakage vortex and the secondary vortex, however, were quite different, as shown in Figure 2.2.4. It should be noted that the streamlines shown in Figure 2.2.4 converge, which clearly contradicts the continuity of the flow. Based on smoke trace flow visualization in a turbine cascade, Bindon (1987a) observed the tip leakage flow joining both the leakage vortex and the secondary vortex. With the results of surface flow visualization in a compressor cascade, Shun Kang and Hirsh (1992) proposed the vortex structure, as shown in Figure 2.2.3. Surprisingly they suggested that the secondary vortex was located near the suction corner and rotated in the opposite direction to the tip leakage vortex. From their description, this secondary vortex

Figure 2.2.4 Schematic of the Vortex Structure around the Tip
(based on Bindon (1987a))

separated along the suction edge of the blade, since the trajectories of ink traces on the blade tip converged along the suction edge. Compared with Sjolander and Amrud's (1987) results (Figure 2.2.1), the secondary vortex of Shun Kang and Hirsh (1992) seems to be the vortex near the suction side suggested by Sjolander and Amrud, with the exception that the rotation is reversed.

2.3 Configurations and Instrumentation

The tip clearance flow is very complex, and there are a number of parameters influencing the flow. In order to investigate various effects of tip leakage flow, different configurations have been used.

The configuration which is closest to the real turbine is rotating blade rows. However, because it is very difficult to take measurements within the blade rows, before 1986 information obtained from this type of configuration was limited to inlet and exit of the blade rows. After 1986, with the technology of periodic sampling with Laser Doppler Velocimeters, hot-wire and high frequency response miniature pressure transducers, detailed information on the leakage flow within compressor and turbine rotors have been obtained by Murthy and Lakshminarayana (1986), Lakshminarayana et al. (1987), Inoue and Kuroumaru (1989), and Kingcombe et al. (1990).

The most widely used configuration for tip clearance studies is the rectilinear cascade. It consists of a single row of blades. In most cases, the tip clearance was simulated with a gap between the blade tips and the endwall. The relative endwall motion could be introduced by using moving belts. One such configuration (Yaras et al, 1990) is shown in Figure 2.3.1.

The simplest configuration is a single airfoil mounted on a wall at one end, with a gap at another. Such a configuration was used by Sugiyama (1971, as referred to by Yaras and Sjolander 1991) and Lewis and Yeung (1977). In some cases, the airfoil is split at midspan so as to simulate the tip clearance flow without the effect of endwall boundary layers. It is also possible to simulate the relative motion of the endwall while still using an isolated airfoil.

Undoubtedly, the rotating blade rows and the rectilinear cascade can simulate

Figure 2.3.1 Cascade Test Section and Moving Belt
(reproduced from Yaras and Sjolander (1992))

more aspects of the tip clearance flow than the isolated airfoil configuration. However, because at least 3 blades (generally 5 or more) would be included in the test section for rotating and rectilinear configurations, the size of the blade and then the tip gap height are of very small scale. The small size of the tip gap height results in a small scale of the flow phenomena in the tip gap, which makes them very difficult to observe visually and to study experimentally. The shortcoming of the single-blade configuration is that some features of the tip leakage flow, such as adjacent blades with tip leakage flow, can not be simulated. However, it is believed that the governing feature for the flow within the tip gap is the driving pressure difference, together with the chordwise mainstream velocities on each side of the gap. With appropriate design, the above feature of the tip gap flow can be simulated in an isolated airfoil test section. Thus, for a detailed study of the flow within the tip gap, it is advantageous to choose the isolated airfoil configuration, since the relatively large size of the tip gap height available allows more detailed and accurate measurements to be conducted.

From the literature, it can be seen that most measurements taken within the tip clearance are static pressure on the endwall and blade tip, e.g., Bindon (1987b), Moore and Tilton (1988), Storer and Barton (1991), Yaras et al. (1989). Among them Bindon's (1987a) measurements appear to be the most comprehensive ones. With the micro-tapping method, very detailed static pressure distributions were obtained, particularly at the edge of the pressure corner. Based on the results of the static pressure measurements, he suggested the reason for blade tip 'burnout' in cooled turbines.

The measurements of flow angle, velocity magnitude and pressure within the tip gap are very important, since information within the tip gap is crucial to reveal the behaviour of the tip leakage flow, leading to more accurate prediction models and more effective design. However, in a survey of the known studies, it was found that some efforts were made but little success has been achieved in this area. This is understandable since the measurement inside the tip clearance gap is a formidable task. Not only is the gap small thus precluding a five-hole or a seven-hole pressure probe, but at the inlet all gradients are very strong and the zone is particularly inaccessible. This is compounded by a reversal in the flow direction across the separated

shear layer. Graham (1986), Yaras et al. (1989), Storer and Barton (1991), and most recently Heyes and Hodson (1991) did try to take measurements within the tip clearance. Graham used hot-film anemometry for mean velocity measurements at the gap exit. Storer and Barton employed a flattened pitot probe to measure the total pressure, and a hot-wire probe to measure the turbulence intensity. A single axis hot-wire probe was used by Heyes and Hodson to take the clearance gap measurements. Among these, the measurements of Yaras et al. (1989) appear to be the most detailed ones. These authors used a three-hole pressure probe of 0.7mm thickness.

The total pressure measurement in the tip gap can be made with a flattened pitot tube aligned with the flow direction (Moore and Tilton 1988, Storer and Barton 1991). Compared with the conventional hemispherical tip model, the flattened pitot tube has a better spatial resolution. Another advantage of the flattened pitot tube is the greater cross sectional area. This is because the radius of the hemispherical tip has to be equal to or less than, the distance between the wall and the closest station of interest, while the cross sectional area of the flattened pitot tube can be increased by increasing the width of the probe. The cross sectional area is related to the response time; for example, as the total pressure decreases in an air flow, the probe must release some air from the measurement system volume. This process can be finished more quickly if the cross sectional area is larger. Furthermore, the flattened pitot tubes are most often smaller than their hemispherical counterparts and would therefore allow measurements closer to a wall. The former will also experience less velocity gradient across its height.

A hot-wire probe is used to measure values of mean velocity and fluctuations in the mean flow direction. Detailed information on the operating principles of hot-wire anemometers is plentiful and easily available, e.g. Lomas (1986); therefore, only a brief description will be presented here. Every hot-wire anemometer, regardless of type, contains the same basic parts: a probe with its cable, and an electronics package. The sensor of the typical hot-wire probe is a wire, usually made of tungsten or platinum, about 1mm long and 5 μ m in diameter (Lomas, 1986). It is convection cooled by fluid passing over it, and this cooling effect is a measure of the fluid velocity. Three types of electronics packages are used: the constant temperature anemometer,

the constant current anemometer, and the pulsed wire anemometer, each controlling the sensor heating current in a different way. The most common is the first one. In this case, the anemometer contains a feedback loop, and a bridge circuit with the sensor as one arm of the bridge. When the power is applied, the sensor is heated and the sensor resistance is increased until the bridge becomes balanced. An increase in velocity cools the sensor and unbalances the bridge. This causes the feedback amplifier to increase the sensor heating current and to bring the bridge back into balance. Since the feedback amplifier responds rapidly, the sensor temperature remains virtually constant as the velocity changes. The voltage difference across the bridge is proportional to the fluid velocity.

From a single measurement with a pitot tube or a single hot-wire probe, it is not possible to distinguish flow directions. However, with a proper traverse gear, it is possible to determine the flow angle by rotating the probe. If the probe is aligned with the flow, the total pressure reading of a pitot tube or the anemometer reading of a hot-wire reaches a maximum. This method was used by Graham (1986), Moore and Tilton (1988), and Storer and Barton (1991). However, aligning the probe with the mean flow direction is time-consuming and tedious. In addition, the effects of flow pitch on the measurements were not taken into account in the above studies. This could be very significant near the separation bubble, the entry and the exit of the tip gap. Thus, as pointed out by Storer and Barton (1991) such measurements are somewhat approximate and should be used mainly to help visualize the flow field.

Champagne et al. (1967) (as referred to by Lomas, 1986) showed that the response of a hot-wire to yaw angle follows the *cosine law* if the sensor aspect ratio is greater than about 600. However, Lomas (1986) also pointed out that because few realistic probe designs have such a large aspect ratio, the *cosine law* can be used only as an approximation. On the basis of the *cosine law*, Heyes and Hodson (1991) employed a single hot-wire probe and a traverse gear to traverse within the tip gap. The probe could be rotated about its spanwise axis, varying the yaw angle of the flow onto the probe. At a given probe location, the measurements were taken over a range of angular settings. By applying the *cosine law*, the flow angle can be deduced. Heyes and Hodson (1991) claimed the measurement accuracy of the flow angle to be within

2°. However, the flow pitch could not be determined, and the velocity magnitude measured is in fact the resultant velocity magnitude. In addition, no information about pressures can be obtained.

Yaras et al. (1989) used a three-hole probe that is 0.7mm thick and 2.0mm wide to take measurements within the tip gap in a rectilinear cascade. The probe was used in the non-nulling mode to measure the pressures as well as the velocity magnitudes. It was calibrated over a range of misalignment angles of $\pm 70^\circ$ in yaw. When the flow was in the plane of the probe, the uncertainty in the measured yaw angle was shown to be $\pm 3^\circ$, and in the dynamic and total pressure 4% of the actual local dynamic pressure. However, for pitch misalignments the uncertainty becomes larger. Yaras (1987) showed that for pitch angles of $\pm 30^\circ$ the measured yaw angles are accurate to within $\pm 4^\circ$, and the local total pressure is underestimated by about 30% of the local dynamic pressure. Once again, no information on the pitchwise flow angle could be obtained.

2.4 Modelling of the Flow Inside the Gap

As reviewed by Yaras and Sjolander (1991), Rains' work (1954) is probably the first study where a reasonably accurate modelling of the flow and a good correlation of the losses were achieved. By studying the flow through the tip gap of a compressor blade, he noted that the pressure gradient across the blade was much larger than that along the blade. He also suggested that in section the flow in a tip gap resembles half of the two-dimensional potential flow through a square-edged orifice. The exit velocity of the tip leakage flow is determined with Bernoulli's equation from just upstream of the gap inlet to the exit. No experimental confirmation of this potential flow theory was provided.

On the basis of Rains' inviscid analysis and their own experimental observations, Booth et al. (1982) introduced a prediction scheme for determining the velocity and flow angle distribution in the tip clearances. In their model, the velocity component tangential to the blade meanline is determined by the pressure side static pressure, and the driving pressure difference determines the velocity component normal to the

blade meanline. The driving pressure difference was taken as the difference between the pressure and suction sides of the blade along the trajectory of the streamline in question. Furthermore, the magnitude of the velocity component normal to the blade meanline was taken to be reduced by the discharge coefficient. Since the trajectory of the streamline depends on the velocity component normal to the blade meanline, Booth's model is iterative. An empirical discharge coefficient of 0.8 was used in his prediction.

Later Yaras et al. (1987) studied the assumptions built into the model of Booth et al. (1982), and then suggested a modified, simpler and more accurate model. The basis of the model was again to use Bernoulli's equation to determine the tip gap velocity component normal to the blade meanline from the driving pressure difference. From their experimental results, the driving pressure difference was found to be approximately equal to the local undisturbed blade pressure difference, with a rearward shift by about 20% of the chord. With the experimentally justified assumption that the tangential velocity is conserved through the gap, the chordwise distribution of gap velocity and flow angle was then determined. Comparison with experimental results indicated that the model was able to predict the flow field quite accurately. Unlike for Booth's models, no iteration was needed for Yaras et al.'s model. However, the discharge coefficient obtained was about 0.55 to 0.7, which was lower than the value of 0.8 suggested by Booth et al. (1982), and 0.851 measured by Wadia and Booth (1982).

On the basis of Rains' model of a two-dimensional potential flow through a square-edged orifice, Moore and Tilton (1988) proposed two theories for predicting the tip leakage flow within the tip clearances. They predicted and measured a local discharge coefficient of about 0.85, in agreement with Booth and his co-workers. A detailed description of these two theories can be found in Section 5.4, since they are compared with the present data.

Most recently, Heyes and Hodson (1991) employed the model proposed by Moore and Tilton (1988) but represented the tip gap exit flow by a combination of an isentropic jet and a wake. In agreement with Moore and Tilton (1989), they found that any mixing occurring over the blade tip increased the discharge coefficient. They

suggested that mixing was likely to increase as the gap height to the blade thickness ratio decreased although this was not verified by any measurements.

2.5 Blade Tip Geometry and Moving Wall

Blade tip treatment is known to be one of the ways to minimize the tip leakage mass flow and reduce the underturning at the blade tip region.

Patel (1980) performed experiments on the effects of winglets on aerodynamic performance. The experiments were conducted with an axial gas generator turbine with a winglet. Patel's results show that, with the exception of a very small region close to the casing, the underturning of in the outer 50% of the annulus is considerably reduced.

Booth et al. (1982) conducted a number of experiments with different blade tip geometries. The experiments were performed in a single-side discharge rig, and a water-table cascade with turbine blades. The blade geometries examined are shown in Figure 2.5.1. They noted that the grooved blade tip configuration appeared to be the most effective one in a single-side discharge rig, since it had the smallest discharge coefficient, as shown in Figure 2.5.2. However, since the discharge coefficient used was based on the actual pressure drop across the tip, using the discharge coefficient as a basis to compare the configurations may have given misleading results, because a reduction in the tip leakage flow due to reduced pressure drop across the gap would not be reflected in the value of the discharge coefficient. This fact was considered in the later water-table cascade experiments, where the optimal tip treatment was further investigated. This time in the analysis of the results, the authors used the dimensional value of the leakage as a basis for comparison. The results indicated that the winglet was better than the grooved configuration in reducing the leakage mass flow, because in addition to the reduced discharge coefficient, the driving pressure difference across the gap was also reduced. The results of water-table cascade experiments for the configurations shown in Figure 2.5.3, are shown in Figure 2.5.4.

Wadia and Booth (1982) continued the study on blade tip treatment both experimentally and analytically. The blade tip treatment mainly concentrated on tip

Figure 2.5.1 Blade Tip Treatments Investigated by Booth et al.
(Discharge Rig Experiments) (reproduced from Booth et al. (1982))

Figure 2.5.2 Results of the Discharge Rig Experiments
(reproduced from Booth et al. (1982))

Figure 2.5.3 Tip Configurations Investigated by Booth et al.
 (Water-Table Cascade Experiments) (reproduced from Yaras and Sjolander (1991))

Figure 2.5.4 Results of the Water-Table Cascade Experiments
(reproduced from Booth et al. (1982))

injection on different blade tip geometries, as shown in Figure 2.5.5. The numerical and experimental results suggested superior performance with the winglet configuration. However, except for the case of no tip treatment, the experimental discharge coefficients for various blade tip treatments were not predicted accurately.

In order to confirm that the low pressure on the edge of the blade was caused by the high curvature of the streamline, Bindon (1987b) explored the effects of radius at the pressure side corner. The results suggested that the increased radius did limit the amount of pressure reduction on the blade tip. Later, Bindon and Morphis (1990) reduced the losses generated within the tip gap by applying a radius to the blade pressure side tip corner, thus removing the separation bubble, and by contouring the blade tip to form, essentially a suction side squealer.

Most recently Heyes and Hodson (1991) performed a number of experiments on two turbine cascades with three different blade tip shapes: flat tip, suction side squealer and pressure side squealer. The results showed that the use of squealers, particularly suction squealers, can provide a benefit over the flat tip. Their model and data confirmed that the radius of the corner from which the flow separates to form the *vena contracta* is vitally important and should be kept as small as possible to minimize the leakage flow.

When the turbine is in operation, there is relative motion between the rotor blade and the casing. This relative motion can be simulated by using a moving-belt as the endwall in a rectilinear cascade of blades or a single-blade test section. Dean (1954) and Gearhart (1966) used linear cascades with a moving-belt endwall and concentrated on the effects of wall motion on the tip leakage flow rate. Graham's (1986) study in a water-table cascade provided further insight into the gap flow through dye injection. Morphis and Bindon (1988) used an annular cascade with a rotating outer wall to examine the effect of tip wall motion on the pressure distribution on the blade tip. A detailed description on this aspect can be found in Yaras and Sjolander's (1991) review.

Finally, Yaras and Sjolander (1992), and Yaras et al. (1992) performed experiments using a rectilinear cascade with a moving-belt to simulate the relative endwall motion. As in other experiments, they observed that a substantial decrease in the

**Figure 2.5.5 Tip Configurations Investigated by Wadia and Booth
(reproduced from Wadia and Booth (1982))**

gap flow rate occurred as a result of the relative wall motion. They showed that with the introduction of the wall motion, the strength of the tip vortex was reduced considerably while the passage vortex was enhanced by the scraping effects of the blades. Both vortices were dragged toward the suction side of the passage where they appeared partially to block the outlet flow from the tip gap. This might account for the reduction in the driving pressure difference of the tip gap flow and consequent reduction in the gap velocity. They also observed that blade suction-side pressures near the tip were altered considerably by the wall motion, which result in a reduction of the blade loading near the tip. A tip-vortex model for the case of rotation was extended from an earlier model (Yaras et al., 1989) and was found to give good predictions.

Chapter 3

Description of Apparatus

As one can see from the literature review of tip leakage studies, there are still some unsolved problems with regard to tip leakage flows. Since there seems to be a lack of full understanding of the flow inside the tip gap, it was decided to study further this subject experimentally. Because the existing facilities do not allow the flow inside the tip gap to be studied in detail, a large scale tip gap test section and a transition duct were designed and manufactured. This test section was designed to be used with one of the turbomachinery wind tunnels located in the Turbomachinery Laboratory at Carleton University.

An attempt was made to make the test section operationally convenient as well as inexpensive, and incorporate as many features as possible into the design. At the same time, simplicity and ease of manufacture were primary considerations. In the following section, the test section and its components, and other equipment used in this study are described in detail.

3.1 Wind Tunnel

The wind tunnel supplies an open circular jet and uses a centrifugal fan driven by a variable speed 40 HP DC motor. This centrifugal fan can supply a maximum mass flow rate of 4 kg/sec at a total pressure rise of 280 mm of water. Ambient air enters

the fan through a bell-mouth intake and is driven into a large settling chamber with a honeycomb, and three screens in series. The outlet of the contraction at the end of the settling chamber is a 406mm diameter circle.

For a more complete description of the wind tunnel, see previous work on the same wind tunnel by Yaras (1987).

3.2 Transition Section

The test section used in the present study is a two dimensional rectangular channel, as shown in Figure 3.3.1, and the inlet to the test section is a $365 \times 365\text{mm}$ square. However, the outlet of the wind tunnel is a 406mm diameter circle, as mentioned above. Therefore, a transition from circular to square cross section had to be designed.

The transition section consists of a circular flange, a square flange and a transition duct, as shown in Figure 3.2.1. The two steel flanges were made by welding steel sheets together and then machining to the desired sizes. The transition part was made of two equal-sized galvanized steel sheets with 26 gauge (0.551mm) thickness, which were fastened together by bolts and nuts. Triangulation method was used to lay out the two steel sheets. By carefully controlling the angles of bending the steel sheets, a very good transition from circular to rectangular was obtained. The transition duct was slid into the circular flange and then both were riveted together in the radial direction. The other end of the transition duct was bolted to the square flange. In order to achieve a smooth transition and good sealing, all steps and gaps between these parts were smoothed or filled with putty. To prevent rusting, the two steel flanges were painted.

The overall length of the transition was chosen to be 405mm based on consideration of the space available in the lab. An area ratio of 1.03 in the transition section results in slight diffusion before the flow enters the test section.

Figure 3.2.1 Transition Section

3.3 Test Section

The test section, shown in Figure 3.3.1, was specifically designed for the detailed investigation of large scale tip leakage flows. However, it was also attempted to make the test section as versatile as possible, so that it can be used for other flow studies. In fact, most components shown in Figure 3.3.1 were used by Hetu (1993) to study two-dimensional tip leakage flow in a straight channel. With such a large scale test section, it is possible to investigate the flow inside the tip gap with instrumentation, such as the seven-hole pressure probe.

The tip leakage flow study might involve many variables, such as changes of the tip clearance, changes of the driving pressure difference, changes of the blade tip geometry, and the control of the tip leakage. Thus, the test section was designed to allow as many of these changes as possible.

The geometry of the test blade is a quarter of a circle with rectangular extensions on both ends, as shown in Figure 3.3.2. From two-dimensional flow predictions neglecting the tip clearance, the pressures on the pressure and suction sides are uniform. Thus, a constant driving pressure difference across the gap would be obtained. However, due to the tip leakage the actual driving pressure difference varies somewhat along the chord length.

Since the flow inside the tip gap is of primary interest, no attempts were made to simulate all aspects of the turbine flow. It was believed that the most important features to simulate were the driving pressure difference across the gap, together with the chordwise mainstream velocities on each side of the gap.

The tip clearances from 0 to 52.1mm can be achieved by replacing the shims between the backplate and the blade (see Figure 3.3.1). The shims were cut into two pieces along the central lines, which allows the tip clearances to be changed by simply inserting the shims without taking the blade off the backplate.

The mainstay of the test section is a 25.4mm (1.00inch) thick particle board which is mounted on a frame made of steel angles. All other components are supported by this component. The advantage of choosing particle board instead of aluminum plate is that it is very cheap and easy to drill holes into for mounting experimental

Figure 3.3.1 Assembled Test Rig

Figure 3.3.2 Test Section and Its Main Dimensions(mm)

instrumentation and modified components.

The endwall of the test section is made of 12.70mm thick plexiglass. This facilitates flow visualization in addition to reducing the fabrication cost. With two handles on it, the endwall allows for quick and easy mounting and dismounting. Nine clamps are then used to secure the endwall to the blade passage and obtain good sealing. The endwall is instrumented with 46 static pressure taps of 0.5mm diameter, which are along the $\alpha = 30^\circ$, $\alpha = 45^\circ$ and $\alpha = 60^\circ$ lines, as shown in Figure 3.3.3. The coordinates of these static taps can be found in Appendix A. The endwall could be replaced with a moving belt to simulate the relative motion between the blade and the endwall.

In order to change the driving pressure difference across the tip gap, screens can be installed at the exit of the inner passage, as shown in Figures 3.3.1 and 3.3.2. The screen is clamped to the frame of the upper sidewall on one side. Another side is held by bolts through a steel rod mounted on the backplate. By changing the open area of the screen, various driving pressure differences can be achieved. In the present study, two types of screen were used: a piece of perforated steel sheet with 40% open area and a piece of solid steel sheet with which the inner passage of the test section was completely blocked.

The sidewalls of the test section were made of aluminum frames with galvanized steel sheets. These two components were glued together with Hysol Epoxy-Patch Adhesive 1105. This approach of fabricating the sidewalls provides a very smooth surface with a good transition from the straight part to the circular part and vice versa, and is inexpensive. With the sidewalls being a quarter of a circle, the flow leaves the test section at 90° compared with the test section inlet flow direction.

A shortcoming of the test section is the wear of the particle board. In the present study, it was found that the holes for holding the test blade became a little bigger with use. Although the wear on the holes is small, care was taken to locate the test section components accurately each time they had to be remounted.

Figure 3.3.3 Static Pressure Taps on the Endwall

3.4 Test Blade

As seen from Figure 3.4.1, the method of manufacturing the test blade is similar to that used for making the sidewalls, *i.e.* gluing two pre-shaped galvanized steel sheets to the two upper ribs and the two plates of the aluminum frame with Hysol Epoxy-Patch Adhesive 1105. The bottom rib was then fastened to the galvanized steel sheets with screws. The ribs of the frame were cut from a 12.7mm thick aluminum sheet by a band saw following a template. This procedure combined with several hours of finishing the cutting surface provided a very smooth curvature.

As one can see from Figure 3.4.1, slots were milled into the two upper ribs and an air pipe was installed on the bottom rib. Thus, air could be blown into the interior of the test blade. Using this, the effects of the tip injection on the tip leakage flow could be explored. The tip of the test blade is replaceable. By changing the cross section of the cover, the effects of the blade tip geometry could be investigated.

In the present study, the blade tip has a simple rectangular cross section. It is instrumented with 45 closely-spaced static pressure taps of 0.5mm diameter which are located along the $\alpha = 30^\circ$, $\alpha = 45^\circ$ and $\alpha = 60^\circ$ lines, enabling static pressure measurements to be made on the blade tip. The coordinates of these static taps can be found in Appendix B. The static taps are connected to Tygon flexible plastic tubing by brass fittings. The flexible tubing is then passed through the hollow blade centre and out through the air pipe.

3.5 Seven-hole Probe

The main flow field measurements were made with a seven-hole pressure probe in a non-nulling mode. This probe has been used successfully in several three-dimensional flow studies: for example, in the tip leakage flow studies of Yaras and Sjolander (1992), Yaras et al. (1992), and more recently by Hetu (1993).

As discussed earlier, the present study has concentrated on the flow behaviour inside the tip gap, where the flow is very complex and three-dimensional. Because three-hole probes are sensitive to pitch misalignments, as described by Yaras (1991),

Figure 3.4.1 Test Blade

it is difficult to get reliable measurements inside the tip gap with a three-hole probe.

A pressure probe was preferred over the laser doppler velocimeter and hot wire probes mainly due its ability to measure total pressure in addition to velocity. Simplicity and ruggedness were other factors that contributed to this decision. The seven-hole probe has the advantage over the more conventional five-hole probe in that it can be used over a wider range of flow incidence.

The seven-hole probe used in this study is made of seven steel hypodermic tubes, which are contained in a brass tube. The tip of the probe is tapered to a 30° semi-apex angle. The space between the seven tubes is filled with solder. The geometry of the probe is given in Figure 3.5.1. As shown, the head of the probe is 2.4 mm in diameter, which, for the examined tip clearances range ($\tau = 22.8$ to 52.1mm), enables measurements to be made with reasonably little interference to the flow through the tip gap.

As shown in Figure 3.5.1, the overall length of the seven-hole probe is only 94.5mm. Since the blade thickness is 78mm, when the probe is traversed near the pressure side the probe holder may be located very close to the probe end connecting the tubing. This, as will be discussed in Section 5.3, might result in some experimental uncertainty.

This seven-hole probe was calibrated up to 50° f.c.w misalignment with the probe axis using 5° steps by Yaras and Sjolander (1990). They suggested that in a reasonably uniform flow, the angles inferred from the probe measurements would, conservatively, be accurate to within $\pm 2^\circ$ and the total and dynamic pressures to within $\pm 5\%$ of the local dynamic pressure. The calibration was done at a local velocity of 14m/sec . It was also found that Reynolds number variation had relatively small effects on the probe calibration.

Figure 3.5.1 Seven-Hole Probe Geometry

3.6 Traversing Gear

For pitot tube and seven-hole probe measurements, a traversing gear which provides linear motion in the spanwise direction was used, as shown in Figure 3.6.1. It was installed on a 6.35mm thick aluminum plate, which was bolted onto the backplate for a seven-hole probe traverse or on the top wall for a pitot tube traverse.

A four phase stepping motor which can provide clockwise and counterclockwise rotation was used in a full stepping mode, yielding one revolution for each 200 steps. Driven by the signals from the parallel port from a microcomputer, a DMA 64 controller controls the rotation of the motor. As shown in Figure 3.6.1, the rotation output from the motor is converted into the linear motion through a 1/4 - 20 threaded rod. The gear can provide a linear motion with an estimated position accuracy of $\pm 0.25\text{mm}$ in either direction.

The stem used to carry the rotation arm for the seven-hole probe was supported by the traversing gear. The rotation arm allowed motion in the plane parallel to the endwall. By choosing appropriate lengths of the rotation arms, this gave access to any location in the tip clearance, as shown in Figure 3.3.2.

From the expected behaviour of the tip leakage flows, it was estimated that the stem which held the rotation arm might be located in the region of the tip leakage vortex, where the flow direction varies substantially. This, combined with the fact that the distance from the seven-hole probe to the traversing gear holder is large and that tip gap flow is quite a three-dimensional flow, might cause vibration of the seven-hole probe to occur. Therefore, a stabilizing bracket for the seven-hole probe was designed and fabricated, as shown in Figure 3.3.2. Experiments confirmed that without this stabilizing bracket, it was almost impossible to take the measurements inside the tip gap. It was observed that with this stabilizing bracket the vibration on the seven-hole probe tip was largely eliminated when the probe was traversed in the separation bubble.

Figure 3.6.1 Traversing Gear

3.7 Data Acquisition System and Data Sampling

The data acquisition system for measuring pressure consists of a Rosemount variable capacitance differential pressure transducer, a Baratron differential pressure transducer (Model 220CD-00100A2BS) manufactured by MKS Instruments Inc., a Model 901 CPU and Intelligent Serial Interface (M901), a Model 81 Electronics Measurement System (M 81) produced by Sciometric Instruments Inc. and an PC-XT micro computer.

Almost the same data acquisition system was used in several earlier cascade flow studies. A detailed description of the data acquisition system can be found in Yaras (1991).

The only difference in the present system is that the Rosemount transducer was used to measure the contraction pressure difference while the Baratron transducer was used to measure all other pressures. The Baratron transducer is temperature controlled at 45°C, and has an input range of ± 2 psi corresponding to a full scale output of ± 10 Volts DC with an accuracy of $\pm 0.15\%$ of the reading. The calibration of both transducers was checked regularly.

An important parameter in data acquisition is the length of the sampling period. It should be long enough to include a large number of samples, and to obtain good long term average values. The length of the sampling time depends on the type of the flow in which the measurements will be taken as well as the type of pressures to be measured. For the static pressure measurement and pitot tube traversing, a sampling period of 40 seconds was found to be sufficient. For the seven-hole probe measurement, a slightly longer sampling time of 50 seconds was found to be appropriate.

A 48 port Scanivalve controlled by a microcomputer was used to switch pressures onto the Baratron transducer. Because of the internal volume of the Scanivalve, the pressure transducer and the tubing which connect the instruments, a period of time is necessary for the pressure reading to stabilize. For seven-hole probe traverses in the tip gap, since the flow directions at two adjacent stations may vary by a large amount, a much longer waiting time is needed for the pressures sensed by the transducer to change from high positive to high negative values. In the present study, for seven-hole

probe measurements a 30 second waiting time was used prior to each sampling period after stepping the Scanivalve, and for static pressure and pitot tube measurements a 10 second waiting time was used.

The data acquisition system allows the input signal to be amplified by a factor which can be adjusted under computer control. Therefore, before any data taking is done, the signal is sampled for 10 seconds to establish its level and then an appropriate amplification factor is chosen.

Chapter 4

Description of Experimental Technique

In this chapter, a brief outline is given of the scope of the measurements, followed by more detailed explanation of specific aspects of the experimental technique. Finally, the test section inlet flow is documented.

4.1 Scope of the Measurements

The objectives of the present study were three-fold: firstly, by a combination of flow visualization and experimental measurements to develop a better understanding of the tip leakage flow; secondly, to make measurements which were sufficiently complete to test tip leakage flow models; and finally, to provide baseline data for future study on the effects of the blade geometry and tip injection on the tip leakage flow, and in particular on the tip leakage losses.

For the off-design operation, the same blade and the tip gap may experience a totally different driving pressure difference across the tip gap. Thus, it is very important to see the effects of the driving pressure difference. In the present study, two different driving pressure differences were employed, as described in Section 3.3. For these two cases, the pressure differences ($Cp_{max} - Cp_{min}$) on the endwall along $\alpha = 45^\circ$ at $\tau/t = 0.667$ were 0.6 and 4.6.

The measurements may be summarized as follows:

- (1) The pressure transducers were calibrated regularly.
- (2) The contraction pressure difference for the wind tunnel was calibrated against the test section inlet dynamic pressure for each tip clearance setting. The contraction pressure difference was subsequently used to control the wind tunnel operating point.
- (3) The calibration of the seven-hole probe was checked regularly.
- (4) The surface flow visualization, static pressure measurements and the seven-hole probe measurements were conducted for several tip clearance settings, which are listed in Table 4.1.1.

Table 4.1.1 Scope of the Measurements

τ/t	Flow Visualization	Static Pressure	Seven-Hole Probe
0.667	✓	✓	✓
0.667*	✓	✓	✓
0.462		✓	✓
0.354	✓	✓	✓
0.292	✓	✓	✓
0.111*	✓	✓	

*Low driving pressure difference case

- (5) At the test section inlet, the endwall boundary layer velocity distribution and the turbulence intensity were measured.
 - (6) Finally, Reynolds number effects on the tip gap flow were explored.
- A more detailed description of the measurements is given below.

4.2 Wind Tunnel Operating Point

The operating point of the wind tunnel is determined in terms of the Reynolds number. The Reynolds number is defined as follows:

$$Re = \frac{\rho V_d l}{\mu}$$

where

ρ : density

μ : viscosity

V_d : test section inlet centerline velocity

l : characteristic length.

Generally, the blade chord is used as the characteristic length for the Reynolds number. However, due to the ambiguous meaning of the blade chord for the present test blade, as one can see from Figures 3.3.1 and 3.3.2. the blade thickness is chosen to be the characteristic length for the Reynolds number.

The flow density and viscosity were determined from the stagnation temperature and the static pressure at the test section inlet. The inlet centerline velocity was calculated from the dynamic pressure.

The operating point of the wind tunnel was monitored from the static pressure difference at the contraction of the wind tunnel settling chamber. These two static pressure taps are located near the start and the end of the contraction. For each tip clearance setting, the contraction pressure difference ($P_{sc1} - P_{sc2}$) was calibrated against the dynamic pressure at the test section inlet centerline, 5.4 blade thicknesses upstream of the test blade.

Since the tip gap flow was of primary interest in the present study, it was important to simulate the gap Reynolds number in the tip gap. The gap Reynolds number is defined as follows:

$$Re_{gap} = \frac{\rho V_{gap} t}{\mu}$$

where

t : blade thickness;

$V_{gap} = V_d \sqrt{1 - C_{pexit}}$: nominal gap velocity;

C_{pexit} : static pressure coefficient on the endwall
at the gap exit, obtained from Section 5.2.

Sjolander and Amrud (1987) and Yaras et al. (1989, 1990, 1992) used $Re = 4.3 \times 10^5$ as their operating point, which was based on the blade chord and the test section inlet centerline velocity. This value corresponded to $Re_{gap} = 5.9 \sim 8.9 \times 10^4$ with

varying gap height and gap exit location. In the present study, it was initially planned to use the same gap Reynolds number. However, due to the large scale feature of the present test section, the velocity at the test section inlet was lower than 8m/sec if the gap Reynolds number was equal to 8.9×10^4 . With such a low inlet velocity, many factors, such as the change of the flow temperature or the operation of another wind tunnel in the same lab, might result in some experimental uncertainty. Thus, a higher Reynolds number had to be chosen.

In the present study, the wind tunnel operating point was chosen to be $Re = 7.8 \times 10^4$, which was based on the blade thickness and the test section inlet centerline velocity. This value resulted in the test section inlet velocity to be approximately equal to 15m/sec . Since the corresponding gap Reynolds number is clearly larger than would be expected in an actual turbine, measurements were also made to explore the effects of varying the Reynolds number, as described in Section 5.5. Table 4.2.1 lists the wind tunnel operating points used in the present study, and the corresponding gap Reynolds numbers along $\alpha = 45^\circ$ line.

Table 4.2.1 Wind Tunnel Operating Point (Re) and Gap Reynolds Number ($Regap$) along $\alpha = 45^\circ$ Line

τ/t	Re	$Regap$
0.667	7.8×10^4	1.7×10^5
0.667*	7.8×10^4	1.1×10^5
0.667*	1.17×10^5	1.6×10^5
0.462	7.8×10^4	2.2×10^5
0.354	7.8×10^4	2.2×10^5
0.354	3.9×10^4	1.1×10^5
0.292	7.8×10^4	2.1×10^5
0.111*	7.8×10^4	1.5×10^5

*Low driving pressure difference case

During the experiments, the stagnation temperature of the flow entering the test section varied, which would result in the Reynolds number drifting. To compensate for the temperature change, the rotational speed of the fan was adjusted to increase or decrease the test section inlet velocity. By controlling the drifting of the operating point, the wind tunnel operating point Re was kept constant to within 0.1×10^4 .

4.3 Surface Flow Visualization

A mixture of light motor oil (10W30) and white titanium dioxide pigment was used in two forms for flow visualization: as dots and as a film. The traces left by the dots, which represent surface shear stress trajectories are most convenient for identifying converging and diverging shear stress vectors, and therefore three dimensional separation and reattachment lines. The oil film, on the other hand, is more useful for getting an overall picture of areas where the wall shear stress is either high or low, according to the degree of the oil is scoured away, unaffected or accumulated. The results of both methods are then used together to try to interpret the flow behaviour near the surface.

The mixture of the white pigment with the motor oil was a light yellow. For better contrast with the oil mixture, the backplate and the blade tip surface were covered with black self-adhesive vinyl film.

4.4 Static Pressure Measurement

The holes in the aluminum blade tip which were used for the static pressure measurements were examined under a 50 power microscope and found to be sharp-edged and free from burrs. The data of Shaw (1960) for the static hole error of 'small' holes should therefore apply. The high shear stress at the measurement stations inside the separation bubble in the gap increases the error but it should still be small.

Due to a technical mistake, the static tap holes on the endwall were drilled from the back side of the sheet, resulting in melted plexiglass accumulating near the edge of the holes in the flow side. Although the burrs near the edges of static holes were

carefully removed, the holes in the endwall were not as clean as the ones on the blade tips. These would, in general, be exposed to very thin boundary layers, thus leading to larger errors, but since it would be difficult to estimate the magnitude of the resulting errors, no attempt was made to apply corrections.

Due to the simplicity and convenience of static pressure measurement, it was used to estimate the Reynolds number effects on the tip leakage flow before the seven-hole probe was traversed in the tip gap.

4.5 Seven-hole Probe Traverses

Since the seven-hole probe has to be traversed inside the tip gaps and occasionally go through separation bubbles, it is expected that flow angles and velocity magnitudes sensed by the probe might change significantly during traversing. Thus, before making a detailed traverse in the tip gap, the probe was quickly traversed inside the tip gap with relatively larger steps and shorter waiting and sampling time, to obtain a rough picture of the flow. Then the probe was re-set in the middle of the flow angle range so as to keep the flow angles at the traverse stations as much as possible within the calibration range of the seven-hole probe. For the case where the probe was traversed through the separation bubble and it was impossible to keep all the flow angles within the calibration range, the traverse stations near the endwall were given priority. The information at those locations was of primary interest, since this is where most of the mass flow through the gap occurs.

Because in the present study this probe would be traversed in the separation bubble where the variation of flow angles and velocity magnitudes could be very substantial, the Reynolds number effects were checked in addition to the probe calibration. The results showed that the errors were within the range described by Yaras and Sjolander (1990). The seven-hole probe measurements were taken from June 1992 to August 1992. The seven-hole probe calibration was checked at the beginning and the end of the test. It was also checked about half way through the test. The calibration results were found to be the same.

As discussed earlier, static pressure measurement was used to explore Reynolds

number effects. Mild Reynolds number effects were found at $\tau/t = 0.354$ for high driving pressure difference, and at $\tau/t = 0.667$ for low driving pressure difference. Thus, seven-hole probe measurement was taken for these two cases at different Reynolds numbers.

4.6 Turbulence Intensity

The free stream turbulence intensity is one of the important parameters in wind tunnel testing. It is a measure of the fluctuations in velocity, and is defined as the root mean square of the fluctuating velocity component divided by the mean velocity. This value, along with the Reynolds number, influences the growth and transition of the boundary layer.

The free stream turbulence intensity (u'/Vcl) was measured at the midspan of the test section inlet centerline using a hot wire and a constant temperature anemometer. The probe is a DISA probe type 55P12 with a tungsten wire of 6 micrometers in diameter. The anemometer is a TSI constant temperature linearized anemometer Model 1054A with a TSI variable decade probe resistance Model 1056 and a TSI power supply Model 1051-2. The analog signals from the hot wire were measured with a Fluke Model 8050A digital multimeter. The mean velocity was measured from the linearized DC signal, and the fluctuating component of the velocity was measured from the root mean square of the AC signal.

The resulting turbulence intensity, u'/Vcl was determined to be 0.55% which is much lower than Yaras' (1987) results of 1.5%. The reason for the difference is not clear, but it might result from different operating points of the centrifugal fan of the wind tunnel for the two cases.

In this study, the free stream turbulence intensity was measured at $\tau/t = 0.667$ for high driving pressure difference when the Reynolds number was 7.8×10^4 .

4.7 Test Section Inlet Velocity Distribution

The endwall boundary layers were traversed at the test section inlet centerline. The traverse station was located 6.5 blade thickness upstream of the test blade. An L-shaped pitot tube was used to measure the total pressures, and a static pressure tap on the upper sidewall was used to measure the static pressure. For these measurements, the traversing gear described in Chapter 3 was used.

Because the flow in the tip gap region of the cascade was of primary interest in this study, the pitot tube was only traversed about half way through the span from the endwall to the backplate. The correct probe tip position at the wall was determined aerodynamically at the beginning of each traverse. For the four tip clearances of high driving pressure difference, the boundary layer velocity distributions, which are non-dimensionalized by the inlet centerline velocity, are given in Figure 4.7.1. The comparison of velocity distributions between the high and low driving pressure difference for $\tau/t = 0.667$ is shown in Figure 4.7.2. The corresponding boundary layer parameters (displacement thickness, momentum thickness and the shape factor) are presented in Table 4.7.1.

In order to determine the boundary layer parameters, the boundary layer thickness had to be estimated, which was ambiguous because of the non-uniformity of the test section inlet flow. Since a slight variation in the location of the outer edge of the boundary layer does not significantly affect the parameters, the boundary layer thickness was chosen to be the point where the velocity was approximately 99% of the test section inlet centerline velocity.

It can be seen that although the tip clearance varies significantly, the endwall boundary layers appear to be very similar. When the driving pressure difference is low, the velocity profile is fuller and the boundary layer thickness is relatively thin compared with the high driving pressure difference case.

Table 4.7.1 Inlet Flow Boundary Layer Parameters

τ/t	δ^*/h	θ/h	H
0.252	0.0060	0.0039	1.54
0.354	0.0061	0.0040	1.52
0.462	0.0060	0.0039	1.53
0.667	0.0069	0.0047	1.46
0.667*	0.0051	0.0033	1.55

*Low driving pressure difference case

Figure 4.7.1 Test Section Inlet Endwall Boundary Layer
($Re = 7.8 \times 10^4$)

Figure 4.7.2 Test Section Inlet Endwall Boundary Layers for Different Driving Pressure Difference ($\tau/t=0.667$, $Re=7.8 \times 10^4$)

Chapter 5

Results and Discussion

5.1 Flow Visualization

Extensive surface flow visualization on the endwall and blade tip was performed at $\tau/t = 0.292$, $\tau/t = 0.354$ and $\tau/t = 0.667$ for high driving pressure differences. A steady trend was observed in the flow patterns from the smallest to the largest tip gap height. Therefore, only the results from the extreme tip gap heights of $\tau/t = 0.292$ and $\tau/t = 0.667$ are presented.

The flow behaviour near the endwall, as interpreted from the surface flow visualization photographs, is outlined with the aid of Figure 5.1.1. Also included on the figure are the locations of static taps on the endwall along $\alpha = 45^\circ$ line which will be discussed in Section 5.2. On the endwall, it can be seen that in the tip gap region the shear stress vectors are almost aligned with the radius from centre of the curvature of the blade. Thus the $\alpha = 30^\circ$, $\alpha = 45^\circ$ and $\alpha = 60^\circ$ lines are approximately the paths of the flow entering the tip gap. Near the test section inlet, the shear stress vectors in the inner channel are almost in the radial direction. However, after that, the shear stress vectors in the inner channel begin turning back toward the test section inlet. This is apparently because the blockage at the exit of the inner channel forces the flow in the inner channel to reverse and pass through the tip gap.

For $\tau/t = 0.292$, the first 40% of the blade thickness on the endwall from the

Figure 5.1.1 Schematic of the Flow Pattern on the Endwall

gap entrance was observed to have very high wall shear stress, since the oil film and pigment in this area had been scoured away almost completely. After emerging from the gap, the flow remains attached until the endwall shear stress trajectories converge on a well defined three-dimensional separation line S1, as shown in Figure 5.1.1. Further out in the outer channel, another separation line S2 is very clear. It is interpreted as the separation line for the passage vortex, since the shear stress trajectories of the main flow in the outer channel converge on this line S2. Between the two separation lines is a region of very low wall shear stress, as indicated by the accumulation of oil. The tip leakage vortex and passage vortex do not appear to merge, presumably because of their opposite direction of rotation, as observed by Sjolander and Amrud (1987).

A surprising observation on the tip gap flow was that a separation line appeared a short distance inside the tip gap, as shown in Figure 5.1.6. This is evident from Figures 5.1.2 and 5.1.4 which show oil dots moving near the pressure side corner.

Based on the static pressure measurements and smoke flow visualization, Bindon (1987a, 1987b) speculated that the separation bubble did not appear to form at the pressure side corner of the blade tip but a short distance inside the tip gap. According to Bindon's interpretation, the line S0 in Figure 5.1.6 was the separation line of the separation bubble.

A more reasonable interpretation of the tip gap flow behaviour is shown in Figure 5.1.7. As can be seen, a secondary separation and dividing surface rolls up beneath the primary vortex and forms a secondary vortex. This is consistent with the existence of both primary and secondary separation lines observed from surface flow visualization. Peake and Tobak (1980) reported very similar behaviour of flow over a delta wing.

The combination of the primary and secondary vortex in section is similar to the so-called separation bubble, as referred to by Rains (1954), Sjolander and Amrud (1987), and Moore and Tilton (1988). The vortical feature of the separation bubble was reported by Yaras et al. (1989), and Shun Kang and Hirsch (1992). The present study also confirmed this (see Section 5.3). Due to its separation feature and the behaviour in the tip gap flow, in the subsequent text the tip separation vortex is still referred to as the separation bubble.

Figure 5.1.2 Blade Tip Flow Visualization at $\tau/t = 0.667$ (Oil Dots)

Figure 5.1.3 Blade Tip Flow Visualization at $\tau/t = 0.667$ (Oil Film)

Figure 5.1.4 Blade Tip Flow Visualization at $\tau/t = 0.292$ (Oil Dots)

Figure 5.1.5 Blade Tip Flow Visualization at $\tau/t = 0.292$ (Oil Film)

Further inside the gap, the separated shear layer was reattached as shown in Figure 5.1.6. The length of the separation bubble depends on the chordwise location and the tip gap height. For $\alpha = 45^\circ$, tip gap height effects on the length of the separated shear layer can be seen in Figure 5.1.7. As shown in Figures 5.1.7 (a) and 5.1.7 (b), the separated shear layer at $\tau/t = 0.667$ is larger than that at $\tau/t = 0.292$. This is not consistent with Storer and Barton's (1991) conclusion that the tip gap flow is unlikely to reattach if the blade thickness is less than about 1.5 times the gap height, since for $\tau/t = 0.667$ the blade thickness is 1.5 times the gap height but the flow is reattached. The presence of the separation bubble in the tip gap suggests that there is a convergent-divergent passage in the tip gap.

In gas turbines, it has been noticed that there is a 'burnout' region on the pressure side of turbine blade tips for some configurations, but no reasonable explanation has been found. The high shear stress region induced by the primary vortex near the pressure corner might be the reason.

Figures 5.1.3 and 5.1.5 show that on the blade tip, the traces of the oil dots also converge to the edge of the blade suction side. As a result, a vortex may form near the suction corner, as shown in Figure 5.1.7. Sjolander and Amrud (1987) showed this feature on the tentative interpretation of their flow visualization, and believed that this vortex resulted from flow towards the gap induced by tip leakage vortex. They indicated that because the two vortices have the same direction of rotation the tip leakage vortex may entrain this vortex. Shun Kang and Hirsch (1992) showed this vortex in their study on compressor blades and named it as 'secondary vortex'. Surprisingly, however, their conclusion was that the rotation of this vortex had an opposite direction to the tip leakage vortex.

In contrast to the results of flow visualization in a linear cascade reported by Sjolander and Amrud (1987), Figure 5.1.7 shows that for $\tau/t = 0.292$, the two separation lines S1 and S2 are further away from the tip gap exit, as compared with $\tau/t = 0.667$. This might be due to the fact that for the present test section, the driving pressure difference for $\tau/t = 0.292$ is higher than for $\tau/t = 0.667$, as will be discussed in Section 5.2. For the linear cascade mentioned above, however, Yaras (1987) found that the endwall driving pressure difference for different gap heights

Figure 5.1.6 Schematic of the Flow Pattern on the Blade Tip

(a) $\tau/t=0.667$ (b) $\tau/t=0.292$

Figure S.1.7 Tentative Interpretation of Gap Flow Structure
for $\tau/t=0.667$ and $\tau/t=0.292$

was quite similar to the blade pressure difference at midspan, which is essentially independent of the tip gap height.

For $\tau/t = 0.292$, there is a static tap located approximately half way between the gap exit and the first separation line S1. Due to the quasi-circular nature of the tip leakage vortex, as noticed by Yaras and Sjolander (1990), and Shun Kang and Hirsch (1992), this tap might be very close to the centre of the vortex. Therefore the static pressure sensed by this tap may be influenced by the tip leakage vortex, as will be discussed in the subsequent section.

It should be noted that since the flow visualization was taken on vertical surfaces, a less-accurate picture at separation lines may be obtained, where the oil accumulated and was acted upon by gravity. This gravitational effects can be seen in Figures 5.1.3 and 5.1.5. However, the effects were very small and did not affect the interpretation of the flow behaviour.

5.2 Static Pressure Measurements

5.2.1 Static Pressure Measurements for High Pressure Difference Cases

As discussed in Section 3.3, both the endwall and the blade tip are instrumented with static taps along the $\alpha = 30^\circ$, $\alpha = 45^\circ$ and $\alpha = 60^\circ$ lines. The effects of the tip clearance height on the static pressure distribution were investigated for four different tip clearances. The clearance height/blade thickness ratios (τ/t) are 0.667, 0.462, 0.354 and 0.292 which correspond to the tip clearances of 52.1mm, 36.0mm, 27.6mm and 22.8mm. For the last three tip clearances, the static pressure distributions were measured only along the $\alpha = 45^\circ$ lines. For the first one, i.e. the largest tip clearance, the static pressure distributions were taken along the $\alpha = 30^\circ$, $\alpha = 45^\circ$ and $\alpha = 60^\circ$ lines, which were approximately the paths for flow entering the tip clearances for this particular test section. Along each line, static taps were located from upstream of the gap inlet to downstream of the gap exit in order to trace the flow before, within, and after the tip gap. The coordinates of static taps on the endwall and blade tip are given in Appendix A and Appendix B.

Because the static pressure distribution for $\tau/t = 0.354$ is similar to that for $\tau/t = 0.292$, only the latter case is presented here. Plots of static pressure coefficients based on the test section inlet dynamic pressure and static pressure for $\tau/t = 0.292$, 0.462 and 0.667 are given in Figures 5.2.1.1, 5.2.1.2 and 5.2.1.3. Figure 5.2.1.4. shows the static pressure distributions on the endwall for different tip clearances. On the graphs, the measured points are connected by straight lines. The definition of x' used in the figures is $x' = R - 438.5(mm)$, where R is the radius from the curvature centre of the blade to the static taps and 438.5mm is the radius to the blade pressure side.

Figure 5.2.1.1 Static Pressure on the Endwall and Blade Tip ($\tau/t=0.667$)

Figure 5.2.1.2 Static Pressure on the Endwall and Blade Tip ($\tau/t=0.462$)

Figure 5.2.1.3 Static Pressure on the Endwall and Blade Tip ($\tau/t=0.292$)

Figure 5.2.1.4 Static Pressure Distributions on the Endwall for Different Gap Heights

Figure 5.2.1.1 shows the static pressure distribution both on the endwall and the blade tip for $\tau/t = 0.667$, which is the maximum tip clearance available for the present test section. It can be seen that the flow begins accelerating before entering the gap. At the entrance to the gap, the static pressure coefficient C_p on the endwall has fallen from -0.083 to -3.5. The flow continues to accelerate along the endwall until 0.72 times the tip clearance through the tip gap where the static pressure reaches the minimum of -5.0. After reaching the minimum, the static pressure starts to rise rapidly. The static pressure on the endwall shows the flow near the endwall experiences an acceleration and then deceleration in the tip gap, which to some extent confirms the conclusion from the flow visualization that there is a convergent-divergent passage in the tip gap.

On the blade, the static pressure is very low near the entrance of the tip gap, which means that very strong shear stresses exist in this region. Combined with the results of flow visualization, this again might explain the 'burnout' of the gas turbine blades. The static pressure begins to recover after reaching the minimum of -8.6 at 0.46 times the tip clearance through the tip gap.

As shown in Figure 5.2.1.1, there is a static pressure difference between the endwall and the blade tip near the gap entrance. This is probably due to the streamline curvature near the separation bubble and very strong shear stress in that region.

Figure 5.2.1.2 shows that for $\tau/t = 0.462$ as the tip gap is approached, the flow is accelerated more quickly than for $\tau/t = 0.667$, and the pressure difference across the gap on the endwall is obviously larger than that for $\tau/t = 0.667$. This trend is expected because the pressure difference sensed by the fluid near the endwall will eventually disappear at a very large tip clearance. On the blade tip, after recovering to the same pressure as that on the endwall, the static pressure lines cross. This 'overshooting' of static pressure on the blade tip may be because the flow is three-dimensional and some pressure differences could result from complicated streamline curvatures. After this 'overshooting', the static pressure on the blade tip should return to be roughly equal to the static pressure on the endwall.

Generally, after the flow leaves the gap exit and merges with the main flow, there is a steady rise in static pressure, as shown in Figures 5.2.1.1 and 5.2.1.2. However, for $\tau/t = 0.292$ and $\tau/t = 0.354$, there is an unusual static pressure drop on the

endwall after the flow leaves the gap, as shown in Figures 5.2.1.3 and 5.2.1.4. From the surface flow visualization, it is evident that after the flow emerges from the gap, tip leakage vortices are formed for all tip clearance settings. Although the circulation of the tip leakage vortex is reduced as the tip clearance is decreased, as observed by Lakshminarayana and Horlock (1962), Lewis and Yeung (1977), Inoue et al. (1986) and Yaras et al. (1989), for small tip clearances the tip leakage vortex is located relatively close to the endwall. From the results of the surface flow visualization, for $\tau/t = 0.292$ the static tap on the endwall, which sensed a drop of static pressure is located approximately halfway between the tip leakage separation line and the suction side of the blade. Due to the quasi-circular nature of the tip leakage vortex, its center might be very close to that static tap. Thus, for $\tau/t = 0.292$ and $\tau/t = 0.354$, the tip leakage vortex may affect the static pressures on the endwall, which is, as discussed above, the case. The effects of the tip leakage vortex on the static pressure on the endwall are reduced with increasing the tip clearance, which is an expected trend since the vortex will be further away from the endwall at larger tip clearances. Beyond the region of the tip leakage vortex, the static pressure close to the bottom wall recovers to normal values.

As described by Storer and Barton (1991), full reattachment of the separation bubble inside the tip gap is signified by a return to a nearly constant static pressure near the exit of the gap, following a region of pressure recovery. Due to the relatively coarse distribution of static taps on the blade tip and the lack of static taps near the tip gap exit, the constant static pressure region was not observed. However, based on the results of the surface flow visualization, it appears that the separated flow is reattached at almost the same region where both the endwall and blade tip static pressures reach the same values. Consequently, static pressure measurements on the endwall and blade tip can be used to investigate the length of the separated shear layer with varying tip clearances. The results show that the smaller the tip clearance, the shorter the length of the separated shear layer.

Figure 5.2.1.4 shows the static pressure distribution on the endwall for the four tip clearances. It can be seen that although there is a significant gap height difference between $\tau/t = 0.462$ and $\tau/t = 0.292$, the static pressure distribution on the endwall

up to the pressure minimum is surprisingly similar. This means that the acceleration of the gap flow is essentially independent of tip gap heights, and it might be possible to construct a unique model to describe the acceleration of the tip gap flow. For $\tau/t = 0.667$, although the acceleration in the gap and the static pressure recovery are quite different from the other cases, the static pressure distribution before the gap entrance is still similar to those of the other tip clearance settings.

Figure 5.2.1.4 also shows that after reaching the pressure minima, the flows are no longer similar. In this region, depending on the gap height, inertia force and pressure gradient, the separated flow is reattached at different locations, as shown in Section 5.1.

Figure 5.2.1.5 shows the static pressure distribution on the endwall along the $\alpha = 30^\circ$, $\alpha = 45^\circ$ and $\alpha = 60^\circ$ lines. It can be observed that there is a pressure difference between each line, which means that there is a pressure difference tangential to the blade surfaces. Figure 5.2.1.5 also shows that the driving pressure difference rises with the decrease of α . This is because, as the fluid leaks through the tip gap, the mass flow decreases in the pressure side passage and increases in the suction side passage. This then results in an increase in the pressure difference across the tip gap.

5.2.2 Static Pressure Measurements for Low Pressure Difference Cases

As mentioned earlier, the effects of the driving pressure difference were investigated. The static pressure of $\tau/t = 0.667$ for the low driving pressure difference, together with that obtained for the high pressure difference, is shown in Figure 5.2.2.1. Figure 5.2.2.2 shows the static pressure distribution of $\tau/t = 0.111$ for low driving pressure difference.

It is clear that the trend of the static pressure change for low pressure difference is similar to that for high pressure difference, except that the driving pressure difference is much smaller. Thus, there will be a relatively stronger tangential velocity component for the low pressure difference cases.

Figure 5.2.1.5 Static Pressure on the Endwall at different Locations ($\tau/t=0.667$)

Figure 5.2.2.1 Static Pressure on the Endwall for High and Low Pressure Difference Cases ($\tau/t=0.667$)

Figure 5.2.2.2 Static Pressure on the Endwall and Blade Tip for Low Pressure Difference Case ($\tau/t = 0.111$)

Figure 5.2.2.2 shows the static pressure of $\tau/t = 0.111$ for the low driving pressure difference, which corresponds to $\tau = 8.6mm$. This is the smallest tip gap height in the present study. With so small a gap height, only a very small amount of flow will go through the gap. Therefore, the pressure differences at different radial lines should be very similar. In fact, Figure 5.2.2.2 shows that the static pressure distributions essentially collapse onto one curve. It is very interesting to notice that on the $\alpha = 45^\circ$ line, one static tap sensed a pressure drop after the flow left the gap exit. The reason for this was discussed in Section 5.2.1.

5.3 Seven-Hole Probe Measurements In the Tip Gap

Seven-hole probe measurements were performed within the tip gaps in order to get a detailed picture of velocity magnitudes, flow angles, dynamic pressures and total pressures.

Due to limitations of time, for all tip clearance settings the seven-hole probe was traversed from the blade tip to the endwall at mainly two planes: an interior plane and an exit plane. In the cases of $\tau/t = 0.667$ and $\tau/t = 0.292$, another plane, which is called the middle plane in this thesis, was traversed. For $\tau/t = 0.667$ the seven-hole probe was also traversed near the inlet of the tip gap. All the above traverses were done along the $\alpha = 45^\circ$ line, which as discussed earlier, is approximately the path of the gap flow.

The interior plane is the plane where the static pressure on the endwall reaches the minimum. As discussed earlier, there is a convergent-divergent passage in the tip gap. Thus, it is reasonable to expect that the interior plane is located very close to the peak of the separation bubble.

The exit plane was not located exactly at the suction edge of the blade. Instead, it was located 2mm inside the gap from the suction side. The purpose of this was to make the seven-hole probe measurements as efficient as possible. If the seven-hole probe were located just at the exit of the tip gap, measurements close to the blade tip might be out of the calibration range due to the substantial change of flow angles in that region. For the same reason, the inlet plane was located 1mm inside the tip gap.

Figure 5.3.1 shows the coordination system used in the present tip gap flow study. The origin is at the tip gap inlet on the endwall along $\alpha = 45^\circ$ line. The components of the fluid velocity in the x , y , and z directions are designated by u , v , and w , which sometimes are called the normal velocity, the tangential velocity and the velocity in spanwise direction.

Figure 5.3.1 Coordination System for the Tip Gap Flow

2 of/de 2

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS
STANDARD REFERENCE MATERIAL 1010a
(ANSI and ISO TEST CHART No. 2)

In order to explore the effects of the driving pressure difference, seven-hole probe measurements were conducted at $\tau/t = 0.667$ for both high and low driving pressure differences. The location of all the traverse planes in the present study are listed in Table 5.3.1.1.

Table 5.3.1.1 Locations of the Seven-Hole Probe Traverse

τ/t	$r(mm)$			
	Inlet Plane	Interior Plane	Middle Plane	Exit Plane
0.292		28	34	76
0.351		28		76
0.462		28		76
0.667	1	39	28	76
0.667*		28	34	76

*Low driving pressure difference case

As to the uncertainty of seven-hole probe measurements, from Section 3.5, it is known that there is a significant distance between the probe holder and the probe tip. Since there is only one screw to fix the probe into the slot of the holder, with the aerodynamic force resulting from w , the probe tip might be pushed a little towards the blade tip or the endwall. Therefore, for the seven-hole probe measurements at the interior plane, the errors introduced by this might cause the flow to appear to be towards the blade tip or the endwall. Close to the endwall and the blade tip, the wall effects on the seven-hole probe measurements might also become more important. Thus, the measurements close to the endwall and the blade tip should be treated with caution.

5.3.1 Velocity in the Tip Gap

The local magnitude and the direction of the flow are two of the quantities obtained from seven-hole probe measurements. For convenience they are presented together in

the form of a velocity vector. The magnitude of the velocity is non-dimensionalized by the magnitude of the test section inlet centerline velocity. In order to achieve a good description of velocity distributions in the tip gap, the velocity vector is presented in both xyz and xz coordination systems, as shown in Figures 5.3.1.1 to 5.3.1.6.

At the interior plane and middle plane, several measurements at stations close to the blade are not shown because they are out of the seven-hole probe calibration range. In such cases, the lack of data is not important since those traverse stations are located inside the separation bubble. Only the data outside the separation bubble will be used to determine the contraction ratio of the separation bubble.

From the figures, it is clear that close to the blade tip the velocity vector varies rapidly in both magnitude and direction particularly at interior planes and middle planes. These variations are undoubtedly due to the separation bubbles that form on the blade tip near the pressure side of the blade, as discussed earlier in Section 5.1. Close to the blade tip, the behaviour of the velocity vector at the interior plane is at first sight unexpected in that it is directed from the suction to the pressure surface and has a very strong tangential component. However, this is consistent with the conclusion obtained from the surface flow visualization that the separation bubble has a vortical structure. Outside of the separation bubble, the velocity varies smoothly towards the endwall. For all clearances, at the interior plane and the middle plane there is no evidence of a boundary layer on the endwall, even at the outermost station which is less than 2mm from the endwall. This might be due to the fact that the boundary layer from the test section inlet is swept across the tip gap near the leading edge of the blade, as observed by Sjolander and Amrud (1987), and the newly developed boundary layer on the endwall should be very thin. At these two planes, the velocity magnitude above the separation bubble is apparently larger than those at the inlet and exit planes. This is because the stream tube passing through the gap converges over the separation bubble, which from continuity causes the flow to be accelerated.

Figure 5.3.1.1 Velocity Vectors in the Tip Gap ($\tau/t = 0.667$)

Figure 5.3.1.2 Velocity Vectors in the Tip Gap ($\tau/t = 0.462$)

Figure 5.3.1.3 Velocity Vectors in the Tip Gap ($\tau/t = 0.354$)

Figure 5.3.1.4 Velocity Vectors in the Tip Gap ($\tau/t = 0.292$)

(a) $\tau/t = 0.667$

(b) $\tau/t = 0.462$

Figure 5.3.1.5 Velocity Vectors Within the Tip Gap in xz Plane

($\tau/t = 0.667$, $\tau/t = 0.462$)

Figure 5.3.1.6 Velocity Vectors Within the Tip Gap in xz Plane
 $(\tau/t = 0.354, \tau/t = 0.292)$

As implied by the flow visualization, for $\tau/t = 0.667$ the separation bubble is reattached very close to the exit. This is, to some extent, confirmed by the seven-hole probe measurements, since the streamline curvature is evident at the gap exit for $\tau/t = 0.667$, as shown in Figure 5.3.1.5. At the interior plane and middle plane, very similar flow behaviour can be observed, which means that near the peak point of the separation bubble the flow does not change too much. This to some extent also shows the good repeatability of the seven-hole probe measurements in the present study.

Yaras et al. (1989) observed that the direction and magnitude of the velocity vectors at the gap inlet are approximately the same as at the gap exit. However, Figure 5.3.1.1 shows that the velocity close to the endwall at the inlet is almost in the xz plane and obviously is not parallel to the velocity at the exit.

For the three smaller tip clearances, as shown in Figures 5.3.1.2 to 5.3.1.4, the flows measured above the separation bubbles at the interior and exit planes appear to be approximately from the same stream tube. However, for $\tau/t = 0.667$, the flows at the four seven-hole probe traverse planes are clearly no longer from the same stream tube, as one can see from Figure 5.3.1.1.

Plots of velocity components u and v which are non-dimensionalized by the test section inlet centerline velocity magnitude are shown in Figures 5.3.1.7 and 5.3.1.8. In those plots, $(\tau - z)/\tau = 0$ is the blade tip and $(\tau - z)/\tau = 1.0$ is the endwall. Figure 5.3.1.7 shows the u and v distributions at the interior plane, and Figure 5.3.1.8 shows those at the exit plane for all tip gap settings.

At the interior plane, it can be seen that the velocity component u increases with the decrease in tip gap heights. This is consistent with the results of the static pressure measurements which showed that the driving pressure difference was larger for the smaller tip clearances.

From the profile of u , it can be observed that at the exit approximately in the first 40% of the gap height from the blade tip the gap flow moves more slowly. The flow in this region is, in fact, not fully mixed for all tip gap settings.

At the interior plane, there is no evidence of a boundary layer. However, at the exit a boundary layer can be observed close to the endwall.

Figure 5.3.1.7 Velocity Components u and v at Interior Plane

Figure 5.3.1.8 Velocity Components u and v at Exit Plane

Figures 5.3.1.7 (b) and 5.3.1.8 (b) show the tangential velocity v distributions at the interior and exit planes. These data provide a basis for testing tip leakage flow models which consider the effects of the tangential velocity. Booth et al. (1982), Yaras (1987), and Heyes and Hodson (1991) have all proposed such tip gap flow models. Although each model has its own way to determine and consider the normal velocity, the common point is that the tangential velocity was assumed to be constant across the gap. The results show that this assumption is consistent with the experimental results, since the tangential velocity is essentially constant at the gap exit. Although at the interior plane the tangential velocity near the blade tip is larger than that close to the endwall, outside the separation bubble the tangential velocity distribution at the interior plane is approximately the same as at the exit plane.

Although there is significant difference in tip gap height for the three small tip clearances, very similar velocity distributions are observed. This is consistent with the results of the static pressure measurements, as shown in Figure 5.2.1.4. The uniformity of the flow in the tip gap is obviously encouraging from the point of view of devising leakage flow prediction methods. One of the existing models will be examined in Section 5.4.

5.3.2 Total and Dynamic Pressure Distribution in the Tip Gap

Total pressure and dynamic pressure are two of the parameters obtained from seven-hole probe measurements. The total pressure is presented in the form of a total pressure loss coefficient:

$$Cp_o = \frac{(P_o - P_{oin})}{q_{gap}} ,$$

and

$$q_{gap} = \frac{1}{2} \rho V_{cl}^2 (1 - Cp_{exit}) ,$$

where

P_{oin} : the test section inlet total pressure,

$C_{p_{exit}}$: the static pressure coefficient on the endwall at the tip gap exit.

Thus, q_{gap} is the effective gap dynamic pressure in the gap, neglecting losses.

It should be noted that using the test section inlet total pressure as the reference total pressure in the loss coefficient means that the loss also includes losses from the test section inlet to the gap inlet. However, it is expected that there will be negligible losses up to the gap inlet. The gap effective dynamic pressure is used as the reference pressure, since losses would be expected to scale on this dynamic pressure. On the other hand, there are many ways to describe the gap dynamic pressure, such as the mass averaged value of the measured dynamic pressure across the gap exit, *etc.* Among them, $q_{gap} = 1/2\rho V_d^2(1 - C_{p_{exit}})$ is the simplest, most clearly defined one. For the above reasons, the dynamic pressure coefficient is defined as:

$$C_q = Q/q_{gap} ,$$

where

Q : dynamic pressure in the gap;

q_{gap} : effective gap dynamic pressure.

The development of the total pressure loss from the gap inlet to exit for $\tau/t = 0.667$ is shown in Figure 5.3.2.1. The results are, at first sight, quite unexpected in that outside the separation bubble the total pressure increases from the gap inlet to the exit. However, as shown in Figure 5.3.1.1, the three traverse planes are, in fact, not for the same stream tube due to the presence of the significant tangential velocity.

The total pressure loss occurring near the endwall at the inlet plane might have resulted from the specific test section set-up in the present study. As implied in the flow visualization, shown in Figure 5.1.1, because of the blockage at the exit of the test section inner channel, some fluid in the inner channel has to reverse from the test section exit and go through the tip gap. Thus a boundary layer might be developed from the test section outlet and swept into the rear part of the tip gap, resulting in low momentum fluid appearing at the gap inlet near the endwall.

At the gap inlet, due to the presence of the separation in the tip gap, the flow near the blade tip converges, as shown in Figures 5.3.1.1 and 5.3.1.5 (a). The very high pitch angles in this region may have introduced some measurement uncertainty,

Figure 5.3.2.1 Total Pressure Distribution ($\tau/t=0.667$)

Figure 5.3.2.2 Total Pressure Distribution ($\tau/t=0.292$)

which is, as shown in Figure 5.3.2.1 the case, since the total pressure loss is larger than 0.0 near the blade tip.

The behaviour of total pressure and dynamic pressure near the blade tip looks also unexpected. The highest total pressure loss appears at the interior plane in the first 40% of the gap height from the blade tip. This is because, at the interior plane the flow close to the blade tip is inside the separation bubble, where large total pressure loss is expected. The fluid in the separation bubble is, in fact, not from the same stream tube as the fluid at the gap exit.

Since the total pressure developments in the gap for $\tau/t = 0.462$, $\tau/t = 0.354$ and $\tau/t = 0.292$ are very similar, only the results for $\tau/t = 0.292$ are shown in Figure 5.3.2.2. Unlike $\tau/t = 0.667$, for $\tau/t = 0.292$ the total pressure decreases from the interior plane to the exit plane. The flow measured is now expected to be from essentially the same stream tube.

Figures 5.3.2.3 and 5.3.2.4 show the total pressure distributions at the interior and the exit planes for all the tip clearance settings. It can be observed that above the peak of the separation bubble, the flow for the three small tip clearances is almost free stream flow, which is consistent with Booth's (1982) and Yaras' (1987) results. However, at the exit the flow near the endwall has experienced some loss as the boundary layer develops on the endwall. When the gap height is small, the flow at the exit becomes more non-uniform and more loss appears near the blade tip. This is an expected trend, because when the tip gap height is decreased, the flow has more relative length to mix with the low momentum flow near the blade tip.

At the interior plane, the large magnitude of the velocity in the gap causes high velocity gradients within the separation bubble and thus results in high shear stress. From earlier discussion, it is known that the smaller the gap height, the larger the velocity across the gap. Therefore, for the smaller gap height, high total pressure loss should be expected within the separation bubble. This is what Figure 5.3.2.3 shows. It also shows that the size of the separation bubble relative to the tip gap height increases with the reduction of the gap height.

Figure 5.3.2.3 Total Pressure Distribution at Interior Plane

Figure 5.3.2.4 Total Pressure Distribution at Exit Plane

Figures 5.3.2.5 and 5.3.2.6 show the dynamic pressure distributions for $\tau/t = 0.667$ and $\tau/t = 0.292$. The dynamic pressure distributions at the interior and exit plane are shown in Figures 5.3.2.7 and 5.3.2.8.

5.3.3 Seven-Hole Probe Measurements for Low Pressure Difference Case

As mentioned earlier, seven-hole probe measurements were performed for $\tau/t = 0.667$ when the driving pressure difference was low. The results compared with those obtained for high driving pressure difference are shown in Figures 5.3.3.1 to 5.3.3.7.

As shown in Figure 5.3.3.1, the flows for the high and the low driving pressure difference are quite different. This is because a low driving pressure difference causes a relatively small normal velocity across the gap. Although the separation vortex, *i.e.* the separation bubble, is clearly evident near the blade tip, as shown in Figure 5.3.3.1, the strength of the vortex and the total pressure loss in the separation bubble are much smaller compared with those obtained for high driving pressure difference.

As to the tangential velocity, it is quite uniform across the tip gap. At the gap exit, no separation wake can be observed. This is consistent with the observation from the flow visualization that the flow is reattached earlier when the driving pressure difference is low.

5.3.4 Contraction Ratio, Mass Flow Rate and Mass-Averaged Total Pressure Loss in the Gap

Because in the present study, seven-hole probe measurements were conducted along the $\alpha = 45^\circ$ line only, it is possible only to determine the contraction ratio, mass flow

Figure 5.3.2.5 Dynamic Pressure Distribution ($\tau/t=0.667$)

Figure 5.3.2.6 Dynamic Pressure Distribution ($\tau/t=0.292$)

Figure 5.3.2.7 Dynamic Pressure Distribution at Interior Plane

Figure 5.3.2.8 Dynamic Pressure Distribution at Exit Plane

(a) Low Pressure Difference

(b) High Pressure Difference

Figure 5.3.3.1 Velocity Vectors in the Tip Gap for Different Pressure Difference
 $(\tau/t = 0.667)$

Figure 5.3.3.2 Components u and v at Interior Plane for High and Low Pressure Difference Cases ($\tau=0.667$)

Figure 5.3.3.3 Velocity Components u and v at Exit Plane for High and Low Pressure Difference Cases ($\tau/t=0.667$)

Figure 5.3.3.4 Total Pressure Distributions at Interior Plane for High and Low Pressure Difference ($\tau/t=0.667$)

Figure 5.3.3.5 Total Pressure Distributions at Exit Plane for High and Low Pressure Difference ($\tau/t=0.667$)

Figure 5.3.3.6 Dynamic Pressure Distributions at Interior Plane for High and Low Pressure Difference ($\tau/t=0.667$)

Figure 5.3.3.7 Dynamic Pressure Distributions at Exit Plane for High and Low Pressure Difference ($\tau/t=0.667$)

rate and the total pressure through the gap at this specific location.

The contraction ratio σ was introduced from the orifice flow to describe the relative size of the stagnant region of fluid, as shown in Figure 5.3.4.1. For the idealized orifice flow, from mass conservation it can be expressed as follows:

$$\sigma = \frac{\bar{u}_e}{\bar{u}_{max}} . \quad (5.1)$$

Using the results of the seven-hole probe measurements at the exit plane and the interior plane, the contraction ratio σ can be obtained by solving the following equation:

$$\int_0^\tau \left(\frac{u}{V_{cl}} \right)_{exit} d\left(\frac{z}{\tau}\right) = \int_0^{\sigma\tau} \left(\frac{u}{V_{cl}} \right)_{interior} d\left(\frac{z}{\tau}\right) . \quad (5.2)$$

Obviously, the contraction ratio is the location of the dividing streamline of the separation bubble at the interior plane in the tip gap.

In order to compare the mass flow through the gap for different tip clearances, the mass flow rate through the gap is non-dimensionalized as follows:

$$\dot{M}_{gap} = \frac{(\dot{m}/A)_{gap}}{(\dot{m}/A)_{cl}} . \quad (5.3)$$

Neglecting the density changes, this becomes:

$$\dot{M}_{gap} = \int_0^\tau \frac{u}{V_{cl}} d\left(\frac{z}{\tau}\right) = \frac{\bar{u}_e}{V_{cl}} . \quad (5.4)$$

It is clear that $\dot{M}_{gap}$ is, in fact, the ratio of the average velocity normal to the chord line to the undisturbed test section inlet velocity.

The mass-averaged total pressure loss coefficient is defined as follows:

$$C'p_o = \frac{\int C p_o \rho u dz}{\int \rho u dz} . \quad (5.5)$$

If the density changes are neglected, it can be expressed as:

$$C'p_o = \left(\int_0^\tau C p_o \frac{u}{V_{cl}} d\left(\frac{z}{\tau}\right) / \int_0^\tau \frac{u}{V_{cl}} d\left(\frac{z}{\tau}\right) \right)_{exit} . \quad (5.6)$$

Figure 5.3.4.1 Contraction Ratio σ

The contraction ratio σ , total pressure loss coefficient C'_{p0} and non-dimensionalized mass flow rate through the gap $\dot{M}_{gap}$ were calculated for all tip clearances, and the results are listed in Table 5.3.4.1. Comparing the location of the dividing streamline at the interior plane with that at the middle plane, it is found that the size of the separation bubble is bigger at the interior plane, which confirms that the peak of the separation bubble is near the point of minimum static pressure on the endwall, since this was how the interior plane was chosen.

From the change of the contraction ratios, the separation bubble decreases its relative size with increase in gap heights. This should bring the mass-averaged loss down, which is, in fact, the case.

It should be noted that the seven-hole probe measurements were taken along the $\alpha = 45^\circ$ line only. For $\tau/t = 0.667$, since the fluid at the interior plane and exit plane is no longer from the same stream tube, the contraction ratio should be treated with caution.

Table 5.3.4.1 Contraction Ratio, Mass Flow Rate and Total Pressure Loss from Measured Data

τ/t	σ	$\dot{M}_{gap}$	C'_{p0}
0.292	0.566	1.86	-0.400
0.354	0.617	1.99	-0.322
0.462	0.683	2.12	-0.232
0.667	0.710	1.59	-0.113
0.667*	0.648	0.62	-0.132

*Low driving pressure difference case

5.4 Comparison of the Experimental Results with Moore and Tilton's (1988) Prediction

5.4.1 Moore and Tilton's Model

Based on Rains' model (1954) of flow at rest entering a slot, Moore and Tilton (1988) proposed two theories to describe the tip gap flow. The agreement between their experimental results and the prediction was quite good. However, due to the shortage of the experimental data in the tip gap, the validity of this model has not been thoroughly tested.

With the detailed seven-hole probe and static pressure measurements in the present study, it is possible to test Moore and Tilton's model more completely. As a starting point, the model can be summarized and analyzed as follows:

(a) Up to the tip clearance entrance, the flow is assumed to resemble a two-dimensional flow at rest entering a sharp square-edged orifice. Based on the symmetry, the centerline of the orifice is the endwall of the tip gap.

The main assumption in this theory is that the pressure gradients across the tip gap are much larger than those tangential to the camber line of the blade. It is also assumed that the flow is incompressible and the Reynolds number is sufficiently high for viscous effects to be neglected.

The first assumption was originally proposed by Rains (1954) based on the fact that the pressure gradient across the gap is much larger than the one parallel to the blade. The results of Booth et al. (1982) confirmed this assumption to some extent, and Yaras et al. (1989) and the present study showed that the flow essentially completes its acceleration before entering the gap. This implies that the normal velocity remains unchanged through the gap under ideal conditions. The second assumption concerning viscous effects was supported by Yaras' (1987) results.

The solution for the orifice flow, derived from an analysis by Milne-Thompson (1968) is:

$$x + iz = \frac{2\sigma\tau}{\pi} \left(\ln \frac{u_{max} + \xi}{u_{max} - \xi} - \frac{u_{max}}{\xi} \right), \quad (5.7)$$

where

τ : the tip gap height;

σ : the contraction ratio;

$\xi = u - iw$: the complex velocity;

u_{max} : the velocity above the no flow region.

The contraction ratio σ was shown to be equal to $\pi/(\pi + 2) = 0.611$.

The velocity field described by this equation is shown in Figure 5.4.1.1. The velocity vectors cover only part of the tip gap flow region with the remainder being a region of no flow. This no flow region is similar to the separation bubble in the tip gap, with the exception that the potential flow never reattaches.

(b) From the tip gap entrance to the point where the stream tube inside the gap reaches its minimum size, *i.e.* the interior plane, the flow is assumed to be a potential flow. In the remainder of the gap, it is assumed that mixing occurs till the gap exit. The momentum equation and continuity equation are applied in this part of the flow.

The potential flow assumption has been confirmed, as discussed earlier. The other assumptions made for this part of the flow are that the flow is steady and the shear stress is neglected. The velocity and the static pressure at the interior plane and the exit plane of the tip gap are assumed to be constant. As can be seen from previous discussions, the assumption of the static pressure and velocity is not well supported by the measurements.

5.4.2 Contraction Ratio

Since in the following sections contraction ratios will be used in the prediction of the static pressure on the endwall, the total pressure loss and the mass flow rate through the gap, the contraction ratio will be examined first.

For the potential flow model described by Equation 5.7, Rayleigh in 1867 showed that the theoretical contraction ratio is $\pi/(\pi + 2) = 0.611$.

Due to the lack of experimental data for the actual flow in the tip gap, Moore and Tilton, on the basis of the potential flow hypothesis used the following approximation to determine the contraction ratio:

Figure 5.4.1.1 Potential Flow Velocity Vectors
(reproduced from Moore and Tilton (1988))

$$\sigma = \frac{\bar{u}_e}{\bar{u}_{max}} \approx \frac{\bar{u}_e}{V_{cl}\sqrt{1 - C'p_{min}}} \quad (5.8)$$

Moore and Tilton used their pitot tube measurements at gap exit to get the mean exit velocity $\bar{u}_e$. For the present data, $\bar{u}_e/V_{cl}$ can be obtained from the results of seven-hole probe measurements, as shown in Equation 5.4, and $C'p_{min}$ is obtained from the static pressure measurements (Section 5.2).

The contraction ratios obtained from Equation 5.8, together with the contraction ratios from the seven-hole probe measurements, are listed in Table 5.4.2.1. In order to compare with the theoretical value of 0.611, a parameter ε was calculated and is also listed in Table 5.4.2.1. The definition of ε is as follows:

$$\varepsilon = \frac{\min(\sigma, 0.611)}{\max(\sigma, 0.611)} \times 100\% \quad (5.9)$$

As shown in Table 5.4.2.1, the experimental results are slightly larger than those of the model's prediction. This is expected because in the potential flow model the velocity above the interior plane is assumed to be uniform. However, in the real gap flow the velocity magnitude close to the dividing streamline is smaller than that close to the endwall, which results in a larger contraction ratio.

For $\tau/t = 0.667$, since the flow at the interior plane and the flow at the exit plane are no longer from the same stream tube (as discussed in Section 5.3), the contraction ratio should be treated with caution.

In the potential orifice flow model described by Equation 5.7, the contraction ratio σ is equal to 0.611, which means that the relative size of the separation bubble is independent of the gap height. This is, to some extent, confirmed by the results in the present study since the ε of the experimental results is within 10% except for $\tau/t = 0.667$.

However, it should also be noted that the size of the separation bubble slightly changes with the tip clearance, as shown in Table 5.4.2.1. This may result in the variation in mass flow rate and total pressure loss through the gap, as will be discussed in Section 5.4.4.

Table 5.4.2.1 Contraction Ratio σ

τ/t	Moore & Tilton's Model Prediction (Equation 5.8)		Experimental Results (Equation 5.2)	
	σ	ϵ %	σ	ϵ %
0.292	0.562	92	0.566	93
0.354	0.614	99	0.617	99
0.462	0.660	93	0.683	90
0.667	0.647	94	0.710	86
theoretical $\sigma = 0.611$				

5.4.3 Static Pressures on the Endwall

From Equation 5.7, for the solution on the endwall, $z = 0$, $w = 0$

Thus,

$$x = \frac{2\sigma\tau}{\pi} \left(\ln \frac{u_{max} + u}{u_{max} - u} - \frac{u_{max}}{u} \right) . \quad (5.10)$$

By introducing a non-dimensional parameter,

$$\zeta = \frac{u_{max}^2 - u^2}{u_{max}^2} = \frac{Cp - Cp_{min}}{1 - Cp_{min}} , \quad (5.11)$$

Equation 5.10 becomes

$$\frac{x}{\tau} = \frac{2\sigma}{\pi} \left(\ln \frac{2 - \zeta + 2 + 2\sqrt{1 - \zeta}}{\zeta} - \frac{1}{\sqrt{1 - \zeta}} \right) . \quad (5.12)$$

Using ζ in the above equation establishes the similarity between the orifice flow and the tip gap flow.

The comparison between Moore and Tilton's model prediction and the measured static pressure distribution on the endwall is shown in Figure 5.4.3.1. In the calculation, Cp_{min} in Equation 5.11 is chosen to be the minimum static pressure on the endwall, which was obtained from experimental results (Section 5.2). The contraction ratio σ in Equation 5.12 is chosen to be 0.611.

Since in the potential flow model the flow is accelerated from rest, the static pressure coefficient far away from the entrance is equal to 1.0. In the tip gap flow, however, there is no stagnation region in the passage between blades. Therefore, the static pressure always starts from a lower value than the stagnation pressure. The results show that before the static pressure reaches the minimum, the predicted trend of static pressure is quite similar to that of the real tip gap flow. It is evident that the agreement between the experimental results and the model's prediction is better for the smaller tip clearances. This is because in the potential model the flow in the gap is assumed to be a one-dimensional flow. However, as discussed in Section 5.3.1, the static pressure taps on the endwall do not lie along the same streamline for larger tip clearances. When the tip gap is smaller, the static taps are essentially located along the same streamline.

Figure 5.4.3.1 Static Pressure on the Endwall Compared with Potential Flow Theory

5.4.4 Discharge Coefficient

The discharge coefficient for a tip gap flow can be estimated using the measured exit flow rate and the static pressure at the gap exit, P_{exit} . Thus,

$$Cd = \frac{\bar{u}_r}{V_{cl}\sqrt{1 - C_{p_{exit}}}} = \frac{\dot{M}_{gap}}{\sqrt{1 - C_{p_{exit}}}} \quad (5.13)$$

Moore and Tilton's model gave a discharge coefficient,

$$Cd = \frac{\sigma}{\sqrt{1 - 2(\sigma - \sigma^2)}} \quad (5.14)$$

and from potential theory $\sigma = 0.611$; thus

$$Cd = 0.8438 \quad .$$

Figure 5.4.4.1 summaries the discharge coefficients for all four clearance cases. Also included are the discharge coefficients used in Yaras et al.'s (1989) model. These discharge coefficients were the ones needed to give good agreement with Yaras et al.'s (1989) data. It can be noted that Cd varies with the tip gap height, which is consistent with the observed behaviour of the separation bubbles. The discharge coefficients obtained from Yaras et al.'s method are lower than both the prediction and the ones obtained from Equation 5.13. It is noticeable that all the measured values of discharge coefficient were lower than the value of 0.8438 predicted by this potential model.

In order to investigate the analogy between the tip gap flow and the orifice flow, the discharge coefficients obtained from Equation 5.13 and the variation of orifice discharge coefficients with orifice length documented by Lichtarowicz (1965) are shown in Figure 5.4.4.2. While the magnitudes are similar, the two sets of data appear to have different trends with increasing orifice length.

Figure 5.4.4.1 Variation of Discharge Coefficient with Gap Height

**Figure 5.4.4.2 Comparison of Discharge Coefficient
Between the Tip Gap Flow and the Orifice Flow**

5.4.5 Minimum Static Pressure on the Endwall, Mass Flow Rate and Total Pressure Loss

Based on Moore and Tilton's idea, by applying the energy equation to the fluid between the interior and exit plane, the expressions for the minimum static pressure coefficient on the endwall, the mass flow rate through the gap and the mass-averaged total pressure loss coefficient can be obtained:

$$Cp_{min} = \frac{Cp_{exit} - 2(\sigma - \sigma^2)}{1 - 2(\sigma - \sigma^2)} ; \quad (5.15)$$

$$\dot{M}_{gap} = \frac{\bar{u}_e}{V_{cl}} = \sqrt{\frac{1 - Cp_{exit}}{1 + \left(\frac{1}{\sigma} - 1\right)^2}} ; \quad (5.16)$$

$$C'p_o = -\frac{\sigma(1 - \sigma)}{\sigma^2 + (1 - \sigma)^2} . \quad (5.17)$$

A detailed derivation of the above expressions can be found in Appendix C.

Assuming Cp_{exit} is fixed and equal to the experimental results and choosing σ to be 0.611, the $\dot{M}_{gap}$, $C'p_o$ and Cp_{min} can be predicted. Figures 5.4.5.1 to 5.4.5.3 show the comparison between the prediction and the test results. Also included in Figures 5.4.5.1 to 5.4.5.3 are the values obtained from Equations 5.15, 5.16 and 5.17 when the contraction ratio σ is equal to the experimental values. These values show the variation of $\dot{M}_{gap}$, $C'p_o$ and Cp_{min} with change in the separation bubble size.

From Figure 5.4.5.1, it can be seen that this model underestimates the static pressure at the interior plane. The expression for Cp_{min} is derived from the momentum equation, in which no shear stress in the gap is considered. In the tip gap, however, due to the high pressure gradient in the separation bubble, the wall shear stress might not be small enough to be neglected. Neglecting shear stress on the wall may have caused the minimum static pressure to be underestimated.

The main success of the model is that it provides a good prediction of the mass flow rate through the tip gap, as shown in Figure 5.4.5.2.

Figure 5.4.5.1 Minimum Static Pressure Coefficient on the Endwall

Figure 5.4.5.2 Mass Flow Rate through the Gap

Figure 5.4.5.3 Mass-Averaged Total Pressure Loss in the Gap

This flow model gives a constant value of -0.45 for the total pressure loss, which obviously is not consistent with the observed flow behaviour in the tip gap. It should be kept in mind that the model assumes that the flow is reattached in the tip gap. However, when the tip gap varies, depending on the gap aspect ratio, the flow may or may not reattach before leaving the gap. These two different tip gap flows have different total pressure loss at the gap exit. The model also assumes that the flow is fully mixed out at the gap exit, because the velocity at the exit is assumed to be uniform. However, as discussed in Section 5.3, this is not the case. Therefore, the model should overestimate the mixing losses, as shown in Figure 5.4.5.3. Figure 5.4.5.3 also shows that the total pressure loss varies with the size of the separation bubble.

Figures 5.4.5.4 to 5.4.5.6 show the total pressure loss distribution of Yaras (1987), together with this model's prediction. Surprisingly, this model provides a good prediction along most of the chord length, except a high loss region about 30% of the blade chord from the leading edge. As discussed in Chapter 2, the presence of this high total pressure loss region was because the blade was strongly forward loaded.

Figure 5.4.5.4 Yaras (1987) Total Pressure Loss Distribution ($\tau/\text{chord}=0.0198$)

Figure 5.4.5.5 Yaras (1987) Total Pressure Loss Distribution ($\tau/\text{chord}=0.0285$)

Figure 5.4.5.6 Yaras (1987) Total Pressure Loss Distribution ($\tau/\text{chord}=0.0318$)

5.5 Reynolds Number Effects

As discussed in Section 4.2, due to limitations of the apparatus the gap Reynolds number used was higher than would be expected in an actual turbine. Thus, it is very important to explore the effects of varying the Reynolds number on the tip gap flow. In order to find out whether the flow is altered with changes in the Reynolds number, static pressure measurements on the endwall and the blade tip were performed for a range of Reynolds numbers. Some mild Reynolds number effects were observed for two cases. One is $\tau/t = 0.354$ ($\tau=27.6\text{mm}$) when the driving pressure difference is high. The another is $\tau/t = 0.667$ ($\tau=52.1\text{mm}$) when the driving pressure difference is low.

While varying the Reynolds number, the characteristics of the test section inlet boundary layer on the endwall might change. Therefore, the boundary layers for the above two cases were traversed 6.5 blade thickness upstream of the test blade for a number of selected Reynolds numbers. The boundary layer velocity distributions are shown in Figures 5.5.1 and 5.5.2, and the boundary layer parameters are given in Tables 5.5.1 and 5.5.2. It is evident that at $\tau/t = 0.667$ for the low driving pressure difference, the boundary layer velocity distributions are quite similar except that the velocity profile at $Re = 7.8 \times 10^4$ is a little fuller than at $Re = 1.2 \times 10^5$. However, at $\tau/t = 0.354$ for the high driving pressure difference, the boundary layers at the test section inlet are quite different.

Figure 5.5.3 shows the static pressure distributions of $\tau/t = 0.354$ for high driving pressure difference at Reynolds numbers of 3.9×10^4 and 7.8×10^4 . For $\tau/t = 0.667$ and low driving pressure difference, a small discontinuous change was observed in the pressure when the Reynolds number varied from 3.9×10^4 to 1.5×10^5 as shown in Figure 5.5.4. Selected vector plots, velocity profiles, total pressure and dynamic pressure distributions for the gap flows are shown in Figures 5.5.5 to 5.5.9. The contraction ratios, mass flow rates through the gap and the total pressure losses are listed in Tables 5.5.3 and 5.5.4.

Table 5.5.1 Reynolds Number Effects on the Inlet Flow Boundary Layer ($\tau/t = 0.354$, High Pressure Difference)

<i>Reynolds Number</i>	δ^*/h	θ/h	H
7.8×10^4	0.0061	0.0040	1.52
3.9×10^4	0.0094	0.0060	1.63

Table 5.5.2 Reynolds Number Effects on the Inlet Flow Boundary Layer ($\tau/t = 0.667$, Low Pressure Difference)

<i>Reynolds Number</i>	δ^*/h	θ/h	H
7.8×10^4	0.0051	0.0033	1.55
1.2×10^5	0.0059	0.0042	1.42

Table 5.5.3 Reynolds Number Effects
($\tau/t = 0.354$, High Pressure Difference)

	Reynolds Number	
	$Re = 3.9 \times 10^4$	$Re = 7.8 \times 10^4$
σ	0.618	0.617
$\dot{m}_{gap}$	1.91	1.99
C'_{po}	-0.342	-0.322

Table 5.5.4 Reynolds Number Effects
($\tau/t = 0.667$, Low Pressure Difference)

	Reynolds Number	
	$Re = 7.8 \times 10^4$	$Re = 1.2 \times 10^5$
σ	0.648	0.685
$\dot{m}_{gap}$	0.616	0.609
C'_{po}	-0.132	-0.126

For $\tau/t = 0.354$ and high driving pressure difference, the Reynolds number effect is very localized. The static pressure change appears only in the separation bubble region. As shown in Figure 5.5.6, it is seen that there is very little effect on the velocity vectors. There is also little effect on the contraction ratio and the total pressure loss coefficient. The mass flow rate is a little larger at Reynolds number of 7.8×10^4 , since the driving pressure difference is higher.

For $\tau/t = 0.667$ and low driving pressure difference, however, it appears that the Reynolds number effects are relatively significant, since the Reynolds number effects are no longer localized and the static pressure distributions essentially collapse to two curves for a number of Reynolds numbers. This unusual behaviour is probably because the characteristics of the boundary layer in the tip gap was changed from laminar to turbulent. At Reynolds number of 1.2×10^5 due to the higher levels of energy for the fluid in a turbulent boundary, a smaller total pressure loss is expected near the blade tip. This is, as shown in Figure 5.5.8, the case. Table 5.5.4 shows that at Reynolds number of 1.2×10^5 , the size of the separation bubble is smaller. This causes the variation of static pressure distribution, the reduction of the driving pressure difference and then the reduction of the mass flow rate through the gap, as shown in Table 5.5.4. However, it can also be found that those variations are, in fact, not significant.

Figure 5.5.1 Reynolds Number Effects on the Inlet Endwall Boundary Layer
($\tau/t=0.354$, High Pressure Difference)

Figure 5.5.2 Reynolds Number Effects on the Inlet Endwall Boundary Layer
($\tau/t=0.677$, Low Pressure Difference)

Figure 5.5.3 Reynolds Number Effects on the Static Pressure
 $(\tau/t = 0.354, \text{ High Pressure Difference})$

Figure 5.5.4 Reynolds Number Effects on the Static Pressure
($\tau/t=0.667$, Low Pressure Difference)

(a) $Re = 1.2 \times 10^5$ (b) $Re = 7.8 \times 10^4$

Figure 5.5.5 Reynolds Number Effects on the Velocity Vectors
 ($\tau/t = 0.667$, Low Pressure Difference)

(a) $Re = 3.9 \times 10^4$

(b) $Re = 7.8 \times 10^4$

Figure 5.5.6 Reynolds Number Effects on the Velocity Vectors
 ($\tau/t = 0.354$, High Pressure Difference)

Figure 5.6.7 Reynolds Number Effects on the Velocity Components u and v
 $(\tau/t=0.667, \text{ Low Pressure Difference})$

Figure 5.6.8 Reynolds Number Effects on the Total Pressure ($\tau/t=0.667$, Low Pressure Difference)

Figure 5.5.9 Reynolds Number Effects on the Dynamic Pressure ($\tau/t=0.667$, Low Pressure Difference)

Chapter 6

Conclusions and Recommendations

6.1 Conclusions

Simplicity and versatility were prime considerations in the design of the test section. The performance of the test section was found to be satisfactory.

Surface flow visualization on the endwall and blade tip were used to understand the behaviour of the tip gap flow. The recent results essentially supported the observations from earlier studies, such as the formation of the tip leakage vortex, the passage vortex, the secondary vortex, and the separation vortex on the blade tip.

A new feature observed in the present study is that the existence of two vortices inside the separation bubble on the blade tip. The primary vortex could induce very high shear stress in the region where the shear layer separated. This may be the possible reason why there is a 'burnout' region on the pressure side of turbine blade tips for some configurations in gas turbines.

Static pressure measurements were performed both on the endwall and the blade tip, with particular attention given to the $\alpha = 45^\circ$ line. The results showed that the pressure minima are inside the gap while the pressure maxima are located far away from the gap entrance. This means that the tip gap flow begins to accelerate before it enters the gap. The acceleration was found to be determined by the driving pressure difference across the gap which varied along the blade chord.

It was also found that for all the tip clearance settings in this study, up to the pressure minima, the static pressure distributions on the endwall along the $\alpha = 45^\circ$ are very similar except for the largest gap height case. This indicates that for the clearance sizes in practice the driving pressure difference applied to the gap flow is essentially independent of tip gap heights, which supports the observation made by Yaras et al. (1989).

Combining the results of surface flow visualization and the static pressure measurements, it was found that when the tip gap was small, the centre of the tip leakage vortex was located very close to the endwall and could affect the static pressure on the endwall.

The seven-hole probe measurements within the tip gap enabled a very detailed examination of the tip gap flow. From these measurements, the separation bubble on the blade tip was clearly observed. The presence of the separation bubble was also evident from the results of the surface flow visualization and the static pressure measurements. This separation bubble is, in fact, a separation vortex, which results from the separation fluid being driven chordwise by a mild pressure gradient in that direction. As far as the amount of the tip leakage is concerned, the separation vortex behaved like a separation bubble, since it blocked the flow and reduces the amount of fluid that passes through the tip gap. The bubble appeared to reach its peak height at the location where the endwall static pressure reached its minimum, which was called the 'interior plane' in this study. The separation bubble size was found to be approximately 30% of the tip gap height, and its size relative to the gap appeared to drop slightly with increasing tip gap height.

Outside the separation bubble the total pressure losses were very small at the interior plane, indicating that the tip leakage flow essentially consisted of free stream fluid. Note that this is an important observation as far as modelling the tip leakage flow is concerned. After the interior plane, the total pressure losses increased because of the mixing and the flow was no longer inviscid.

The velocity vector distributions within the tip gap showed that outside the separation bubble the velocity magnitude and directions were very similar for the three smaller tip gap heights examined. This is consistent with the observations concerning

the driving pressure difference across the gap. It was found that the normal velocity is determined by the driving pressure difference across the gap while the tangential velocity is essentially constant and independent of the tip gap height.

The effects of the driving pressure difference at $\tau/t = 0.667$ were explored. It was found that the driving pressure difference is the dominant factor in the tip gap flow. Finally, the effects of varying the Reynolds number appeared to be insignificant.

Due to the lack of detailed and accurate measurements inside the tip gap, few studies have been made on modelling the flow within the tip gap. For the same reason, the validity of the models could not be verified completely. The present experimental results were used to check the success of the leakage model of Moore and Tilton (1988), which is apparently one of the most detailed models available in the literature.

Based on Rains' (1954) idea, Moore and Tilton proposed a model to describe the tip gap flow. This model consisted of two parts: a potential two-dimensional orifice flow before the gap entrance; and a potential and mixing flow in the remainder of the tip gap. The potential orifice flow model was found to provide a good approximation to the real tip gap flow up to the interior plane. The predicted static pressure distribution on the endwall was quite close to the experimental results for the three smaller tip gap cases. This is consistent with the two-dimensional flow assumption made in the model. However, in order to predict the static pressure distribution on the endwall some information such as the minimum static pressure coefficient on the endwall, has to be known first. Although those values can be obtained by experiments or by computation, this requirement definitely limits the use of the model in practice. Moore and Tilton's orifice flow model gave the separation bubble size to be 29.9% of the tip gap size, which was surprisingly close to the experimental results.

The other part of the flow model, i.e. the modelling of the mixing process from the separation bubble to the gap outlet was tested using the data obtained in the present study. The discharge coefficients obtained from the present data were smaller than the model's prediction of 0.8438. The analogy between the tip gap flow and the orifice flow was investigated by comparing the discharge coefficients for these two types of flows. It was found that the discharge coefficients are similar, but the trend

with varying the orifice length is different.

Based on Moore and Tilton's ideas, the expressions of the total pressure losses and mass flow rate through the gap were derived. The experimental results confirmed that these expressions can give good prediction on the tip gap flow.

6.2 Recommendations

Although the performance of the test section was found to be satisfactory, improvements can be made.

Metal inserts can be put into the holes for holding the test blade and the sidewalls in order to avoid the wearing of the particle board.

As discussed in Chapter 2, although extensive research has been conducted to investigate the tip leakage flow and its effects on stage performance, detailed information within the tip gap has not been obtained. The present study provides progress toward this goal through experimental research. However, there are many aspects of the flow within the tip gap that could be investigated. The following recommendations are proposed for future studies in the area of reducing the detrimental effects of the tip leakage:

- (1) modifications to the blade tip geometry, e.g. use of convergence or divergence of the gap; use of cavities or grooves on the blade tip;
- (2) investigation of injection of cooling air from the blade tip;
- (3) investigation of effects of radii at the inlet and the outlet corner on the gap flow.

The above modifications to the blade could be achieved very easily on the present test section, since the geometry of the blade is very simple, the blade tip is replaceable, and the blade is hollow and with an air pipe. On the other hand, the tip gap flow measurements in the present study will provide a basis for the above future studies.

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Appendix A

Coordinates of Static Taps on the Endwall

Figure A.1 Static Pressure Taps on the Endwall

As shown in the Figure A.1, the following table lists the coordinates of static taps on the endwall.

R (mm)	r' (mm)		
	$\alpha = 30^\circ$	$\alpha = 45^\circ$	$\alpha = 60^\circ$
330.8	-107.7	-107.7	-107.7
365.8	*	-72.7	*
380.8	-72.7	-72.7	-72.7
390.8	-47.7	-47.7	-47.7
400.8	-37.7	-37.7	-37.7
410.8	-27.7	-27.7	-27.7
420.8	-17.7	-17.7	-17.7
430.8	-7.7	-7.7	-7.7
438.5	0.0	0.0	0.0
446.2	7.7	7.7	7.7
456.2	17.7	17.7	17.7
466.2	27.7	27.7	27.7
476.2	*	37.7	*
496.2	57.7	57.7	57.7
516.2	77.7	77.7	77.7
526.2	87.7	87.7	87.7
556.2	*	117.7	*
596.2	157.7	157.7	157.7

* No static tap at this location.

Appendix B

Coordinates of Static Taps on the Blade Tip

Figure B.1 Static Pressure Taps on the Blade Tips

As shown in the Figure B.1, the following table lists the coordinates of static taps on the blade tip.

R (mm)	x' (mm)		
	$\alpha = 30^\circ$	$\alpha = 45^\circ$	$\alpha = 60^\circ$
442.5	4	4	4
447.5	9	9	9
452.5	14	14	14
457.5	19	19	19
462.5	24	24	24
467.5	29	29	29
472.5	34	34	34
477.5	39	39	39
482.5	44	44	44
487.5	49	49	49
492.5	54	54	54
497.5	59	59	59
502.5	64	64	64
507.5	59	59	59
512.5	74	74	74

Appendix C

Derivation of Expressions for $\dot{M}_{gap}$, C_{pmin} and C'_{po}

The derivation is based on the following assumptions:

- (1) The flow is steady;
- (2) At each cross section plane, the pressure in the separation bubble is equal to that in the jet, so that no normal forces are exerted by the fluid in the separation bubble on the jet;
- (3) The fluid in the separation bubble cannot transmit shear forces from the wall to the jet, and the shear stress on the wall is neglected;
- (4) The velocity and static pressure at the interior plane and the exit plane of the tip gap are assumed to be uniform.
- (4) The total pressure loss up to the interior is negligible.

As shown in Figure C.1, by applying the momentum equation and the energy equation between cross section i and e , then

$$(\bar{p}_i - \bar{p}_e)A_e = \rho Q(\bar{u}_e - \bar{u}_{max}) = \rho A_e \bar{u}_e(\bar{u}_e - \bar{u}_{max}) . \quad (C.1)$$

$$\frac{(\bar{p}_i - \bar{p}_e)}{\rho g} = \frac{\bar{u}_e^2 - \bar{u}_{max}^2}{2g} + h'_f . \quad (C.2)$$

where h'_f is the energy loss head between cross section i and e .

When Equation C.1 and C.2 are combined, the expression of h'_f can be obtained as follows,

Figure C.1 Flow through a Tip Gap

$$h'_f = \frac{\bar{u}_e^2}{2g} \left(\frac{\bar{u}_{max}}{\bar{u}_e} - 1 \right)^2 . \quad (C.3)$$

From continuity

$$\bar{u}_e A_e = \bar{u}_{max} A_i ,$$

then,

$$\frac{\bar{u}_e}{\bar{u}_{max}} = \frac{A_i}{A_e} = \sigma . \quad (C.4)$$

Thus,

$$h'_f = \frac{\bar{u}_e^2}{2g} \left(\frac{1}{\sigma} - 1 \right)^2 . \quad (C.5)$$

C.1 Non-Dimensionalized Mass Flow Rate $\dot{M}_{gap}$

Applying energy equation to a point at test section inlet and a point at the gap exit jet, then

$$\frac{P_{cl}}{\rho g} + \frac{V_{cl}^2}{2g} = \frac{\bar{u}_e^2}{2g} + \frac{\bar{p}_e}{\rho g} + h'_f . \quad (C.6)$$

Substitute the expression of h'_f (Equation C.5) into Equation C.6, then

$$\frac{\bar{u}_e^2}{2g} + \frac{\bar{u}_e^2}{2g} \left(\frac{1}{\sigma} - 1 \right)^2 = \frac{P_{cl} - \bar{p}_e}{\rho g} + \frac{V_{cl}^2}{2g} . \quad (C.7)$$

If

$$C_{p_{exit}} = \frac{\bar{p}_e - P_{cl}}{\frac{1}{2}\rho V_{cl}^2} , \quad (C.8)$$

then, the expression for $\dot{M}_{gap}$ can be obtained

$$\dot{M}_{gap} = \frac{\bar{u}_e}{V_{cl}} = \sqrt{\frac{1 - C_{p_{exit}}}{1 + \left(\frac{1}{\sigma} - 1 \right)^2}} ; \quad (C.9)$$

C.2 Minimum Static Pressure Coefficient on the Endwall C_{pmin}

From Equation C.1, and Equation C.4,

$$\frac{\bar{p}_i - \bar{p}_e}{\rho \bar{u}_{max}^2} = \frac{\bar{u}_e}{\bar{u}_{max}} \left(\frac{\bar{u}_e}{\bar{u}_{max}} - 1 \right) = \sigma(1 - \sigma) , \quad (C.10)$$

then,

$$C_{pexit} - C_{pmin} = 2\sigma(1 - \sigma)\bar{u}_{max}^2/V_{cl}^2 . \quad (C.11)$$

Since,

$$C_{pmin} = 1 - \bar{u}_{max}^2/V_{cl}^2 , \quad (C.12)$$

thus, the expression of C_{pmin} can be written as follows,

$$C_{pmin} = \frac{C_{pexit} - 2(\sigma - \sigma^2)}{1 - 2(\sigma - \sigma^2)} . \quad (C.13)$$

C.3 Mass-Averaged Total Pressure Loss Coefficient C'_{po}

$$C_{po} = \frac{(P_o - P_{oin})}{q_{gap}} = -\frac{\rho g h'_f}{q_{gap}} , \quad (C.14)$$

where

$$q_{gap} = \frac{1}{2}\rho V_{cl}^2(1 - C_{pexit}) . \quad (C.15)$$

Substitute Equation C.5, C.9 and C.19 into C.18, thus

$$C_{po} = -\frac{\sigma(1 - \sigma)}{\sigma^2 + (1 - \sigma)^2} . \quad (C.16)$$

Since the velocity distribution at the gap exit is assumed to be uniform, then the mass-averaged total pressure loss coefficient C'_{po} can be expressed as follows:

$$C'_{po} = C_{po} = -\frac{\sigma(1 - \sigma)}{\sigma^2 + (1 - \sigma)^2} . \quad (C.17)$$

END

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FIN