The Effect of Gap Size and Reynolds Number on Turbine Blade Flat and Suction-Side Squealer Tip Flowfields

A Thesis

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by

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In order to further the understanding of turbine tip leakage and passage flow mechanisms for undesirable entropy production, an experiment was conducted in a linear cascade at the Hessert Laboratory. Blade surface and tip endwall static pressure, total pressure loss, and wake vorticity measurements were taken to document the effects of upstream axial Reynolds number, $1.0 \times 10^5 < Re_2 < 5.0 \times 10^5$, and tip gap height, $0.015 < g/c_x < 0.05$ for flat and partial, suction-side squealer tip geometries. Interaction of tip leakage and passage vortices proved critical, and an inverse relationship was observed between the two structures in terms of streamwise vorticity, core total pressure loss, and the vortex size denoted by the $-\lambda_2$ criteria. Highlighting a considerable limitation of the passive control strategy, the squealer tip more effectively reduced total pressure loss under a “thick” blade model, defined by a blade thickness to gap height ratio, $t/g = 3.5$, than at a slightly larger clearance.
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$p_m$ Averaged Static Pressure from Five-hole Probe Circumferential Ports [Pa]

$Q_p$ Dynamic Pressure Coefficient from Five-hole Probe Calibration

$Re_2$ Upstream Reynolds Number

$Re_3$ Cascade Exit Reynolds Number

$S$ Blade Span [m]

$S_p$ Static Pressure Coefficient from Five-hole Probe Calibration

$S_{ps}$ Blade Pressure Surface Arc Length [m]

$S_{ss}$ Blade Suction Surface Arc Length [m]

$t$ Maximum Blade Thickness [m]

$U$ Upstream Freestream Velocity [m/s]

$V$ Five-hole Probe Calibration Freestream Velocity [m/s]

$V_x$ Axial Velocity at Cascade Exit [m/s]

$v$ Pitchwise Component of Secondary Velocity [m/s]

$w$ Spanwise Component of Secondary Velocity [m/s]

$x$ Streamwise Distance [m]

$x^*$ Chordwise Position, $x/c_x$

$y$ Pitchwise Distance [m]

$z$ Spanwise Distance [m]
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CHAPTER 1

INTRODUCTION

1.1 Motivations

1.1.1 Turbomachine Performance and Efficiency

Turbomachines provide one of the most efficient means to harness energy and produce power. They are invaluable to aerospace, marine, and land propulsion, steam, nuclear, coal, hydroelectric, gas turbine, and wind power production, and petroleum, water, and artificial heart pumps.

With turbomachinery applications truly global in scale, repercussions of even slightly improved turbomachinery designs are highly desirable. As an example, commercial air travel constitutes the most efficient means of transportation to date, and since 1976 improved turbomachinery designs have reduced engine fuel consumption by over 50% [14]. However, while air travel stands today at an all time efficiency high, it faces ever-increasing use and potential for even more fuel savings. Figure 1.1 projects fuel use and cost savings in millions of dollars for just U.S. national and
regional commercial carriers with a 1% reduction in fuel consumption.

Using the conservative values of 2001 U.S. dollars and fuel prices, cost savings are projected to reach $240 million annually by 2025, equating to billions of dollars and gallons of fuel saved in the next 20 years. Improved mission payload and range capabilities come with the improved efficiency and would also benefit all turbofan and turbojet aircraft.

Further, the same mechanisms that contribute to the inefficiency are those which also detract from the safety and reliability of turbojet and turbofan engines. Efforts to improve the efficiency will yield diminished catastrophic engine failure, longer overall engine life, and reduced maintenance. Due to costs of near $1 million to overhaul commercial gas turbine engines, extended time-on-wing is of key importance [14].

Since the advent of gas turbine propulsion, efforts have focused on further refining
the machines and have reached several final areas of potential improvement—limiting the overall combustor size without sacrificing a complete combustion process, discovering materials or improved cooling methods for turbine blades to allow for higher burner temperatures, and limiting the loss in total pressure across compressor and turbine stages. The last largely determines the machine efficiency and is primarily attributed to the tip leakage flow in compressor and turbine rotors.

1.1.2 Tip Leakage Vortex Problem

The presence of a rotating compressor or turbine inside of a jet engine necessitates a small space between the tip of the rotating blade and the outer casing. Flow that leaks into this gap region presents a two-fold mechanism for inefficiency across an un-shrouded turbine rotor. First, flow that escapes into the gap does not undergo turning and thus does not participate in the conversion of flow freestream kinetic energy to rotor rotational energy, leading to significant reduction of turbine work extraction. Secondly, the tip leakage flow introduces a large degree of three-dimensionality to the flow field, which leaves the gap region and traverses down the span. The three-dimensional flow results in viscous dissipation to cause aerodynamic inefficiencies, which also detract from overall turbine energy conversion and overall turbomachine efficiency.

In addition to the efficiency penalty, tip leakage poses the following problems to turbomachines [13]:

1. Blade tip unloading, which reduces the flow turning and affects compressor stall and surge margin.
Fig. 1.2: Variation in Tip Clearance for a Typical Mission Profile [14]

2. Flow distortion, amplified through downstream stages, which serves to create noise.

3. Unsteadiness, amplified through downstream stages, which imposes structural challenges of blade vibration, flutter, and heightened stress.

4. Nontrivial heat transfer complications, which threaten blade life and increase maintenance hours.

5. Cavitation in liquid applications (i.e., pumps) due to the low pressure core of the tip vortex, which result in blade damage.

Made apparent by the above considerations, tip leakage flow creates serious complications for the aerodynamic, thermal, structural, and acoustic operation and design of turbine and compressor rotors, and application of flow control to the tip leakage problem holds multi-faceted potential. To complicate the problem and highlighting the potential for an active control strategy, the tip spacing may change drastically during a typical flight envelope due to various engine and external flight loads as shown in Figure 1.2.
Besides the tip leakage flow, other three-dimensional, secondary flow features are
inherent to the flow turning over a turbine blade, contribute to the overall loss, and
can be seen to interact closely with the tip leakage flow. In the complex flow field
over a turbine rotor tip, experimental investigation has concluded the presence of
multiple, interacting loss mechanisms that must all be considered in an evaluation of
tip performance.

1.2 Loss Mechanisms and Entropy Creation

Of particular interest to engine developers and users is the isentropic efficiency
across a turbine stage, defined as a ratio of the total work actually extracted by the
total work isentropically extracted (i.e., the hypothetical work to be extracted under
isentropic conditions) from the flow.

Turbomachinery losses are usually reported in terms of the total pressure ratio
across the component of interest because a stagnation pressure drop corresponds to
entropy production. In common turbomachines where flow can be approximated as
adiabatic, the isentropic condition is disrupted only by thermodynamic irreversibilities
to occur in boundary layers, in free shear layers, and across shock waves. Figure 1.3
demonstrates the thermodynamic process over a turbine rotor.

Presented by Denton [6], the approach of viewing the loss in terms of entropy
creation is useful due to the fact that entropy is a quantity independent of measure-
ment in a linear cascade or a rotating rig. Applying the first and second laws of
thermodynamics for a calorically perfect gas, entropy, temperature and pressure may
Fig. 1.3: Thermodynamic Process over One Turbine Stage [13]
be related by,
\[
s_3 - s_2 = c_p \log \frac{T_3}{T_2} - R \log \frac{P_3}{P_2},
\]
(1.1)
where \( s, T, P, R, c_p, 2, \) and \( 3 \) indicate the entropy, static or total temperature, static or total pressure, universal gas constant, specific heat at constant pressure, and axial locations before and after the rotor, respectively. For adiabatic flow through a linear cascade where stagnation temperature may be assumed constant, equation 1.1 reveals a direct relationship between entropy production and change in total pressure,
\[
\Delta s = -R \log \frac{P_{03}}{P_{02}},
\]
(1.2)
which for small changes in total pressure may be approximated as,
\[
\Delta s = -R \log (\frac{\Delta P_0}{P_{02}} + 1) = -R \frac{\Delta P_0}{P_{02}},
\]
(1.3)
where \( \Delta P_0 = P_{03} - P_{02} \). In Equation 1.3, a change in total pressure is linearly proportional to a change in entropy. Thus, through the adiabatic process of a linear cascade the drop in total pressure across the blade row is a sound indicator of the departure from isentropic flow and the corresponding turbine rotor efficiency penalty.

Turbomachinery loss has historically been segregated into three types of profile, endwall, and leakage loss, not one of which has been shown to be independent of the other two. Across a turbine rotor, the three components have been found to hold approximately equal contributions. However, profile, endwall, and leakage loss mechanisms all stem from unique flow physics.
Considering loss in terms of entropy creation, the potential processes to bring about loss include viscous friction in boundary and free shear layers, heat transfer, and high viscous stresses and heat conduction inside of a shock wave. Because the flow over a typical turbine rotor and in a linear cascade may be generally considered adiabatic and operation of the linear cascade will be at all times subsonic, the only mechanism to create loss is that of viscous mixing throughout the disturbed local flowfield around a turbine rotor.

Early work with turbomachinery losses adopted a control volume approach, meaning that surveys upstream and downstream of a turbine rotor, for example, were compared to determine loss coefficients from which loss correlations were constructed. Unfortunately, one cannot judge accurately what happens inside of a control volume merely by what enters and leaves it. A lack of insight into the physical flow mechanisms characteristic of the turbine rotor flowfield proved either inconsistent or misleading in the prediction of rotor performance and any flow control efforts. In the late 1970’s and early 1980’s, the research focus shifted to measurements inside of the blade passages and even inside of the tip gap. These efforts led to much enhanced understanding of the flow mechanics and any successful flow control efforts owe credit to detailed and painstaking studies by Yamamoto [24],[25],[26], and Bindon [2], among others.

Despite the aforementioned surge to provide detailed physical insight and due in part to extremely complicated flow physics, boundaries of the loss categories often remain blurred or inconsistent from author to author. Such a trend justifies a detailed
Fig. 1.4: Typical Wake Structures at 2.1% Clearance [26]

explanation of the viscous mixing loss categories and corresponding flow mechanics considered in this thesis.

1.2.1 Profile Loss

The first of the loss categories, profile loss is primarily a two-dimensional effect and includes the loss created by viscous and turbulent dissipation in boundary layers and wakes at spanwise locations far away from the hub and tip endwalls, similar to the aerodynamic loss attributed to “skin friction” on an infinite wing. Even though stagnation enthalpy remains constant in a generally adiabatic flow over a turbine rotor, some mechanical energy in the boundary layer is dissipated into heat, which produces
entropy. Heyes et. al. [9] took the profile loss measurement at a spanwise location of $0.414c_x$ from the tip endwall to quantify the boundary layer entropy production. Additionally, two trailing vortices in the blade wake and nonuniform velocity profiles eventually mix and dissipate to finalize the profile loss. Such mixing losses can take up to a full chord downstream of the rotor to fully mix out [13].

A great deal is known about profile loss as it can be modeled accurately by two-dimensional flow experiments and analysis. Magnitude of the boundary layer loss depends greatly on Reynolds number and blade surface roughness. Both factors are involved in the laminar, transitional, or turbulent state of the boundary layer. At Reynolds numbers below $1 \times 10^5$, dissipation in the boundary layer and boundary layer separation yield high profile losses. These losses rapidly fall with increase in Reynolds number until the transitional region in the range of $2 \times 10^5 < Re < 6 \times 10^5$, where the loss eventually increases with Reynolds number and as the transition point moves upstream. Above $Re = 6 \times 10^5$, profile loss increases only slightly with Reynolds number and is more dependent on surface roughness.

Giving the relative thickness of the trailing edge in comparison with thicknesses of the suction and pressure side boundary layers, it should be no surprise that approximately one third of subsonic profile loss arises just aft of the blade as the two boundary layers mix together. Denton [6] highlights the effect of Mach number on two-dimensional loss and demonstrates that the trailing edge mixing loss is the component of profile loss that increases rapidly as the sonic condition is neared. Intuitively, thin trailing edges have less trailing edge loss than thick edges due to a decrease in
the overall mixing and entropy change.

1.2.2 Endwall Loss

Endwall loss is also commonly referred to as secondary loss, which can often be confusing since many authors group the leakage flow physics or all unaccounted losses including some components of profile loss under the “secondary” loss heading. To a certain extent, the ambiguity is understandable as endwall loss may be difficult to quantitatively isolate from the other loss processes and may always be seen as influenced by its near neighbors of profile and leakage loss. The secondary flows considered here also contribute to a non-uniform passage and hence stage exit flow, which only increases the entropy generation in downstream stages of a real turbine.

1.2.2.1 Passage Vortex

The secondary flow feature with closest proximity to the leakage vortex occurs on both endwalls as a result of the no-slip condition and is referred to as the passage vortex. Shown in Figure 1.5, the passage mainstream flow and endwall boundary layer flows are subjected to the same pitchwise pressure gradient near the blade row. A lower velocity in the boundary layer region, as a result of the no-slip condition at the wall, couples with equilibrium of the centrifugal and pressure forces. The result is a smaller radius of curvature in the boundary layer than in the freestream.

As a consequence, the endwall flow is directed toward the suction surface while flow further from the wall more closely resembles the main passage streamline. The
Fig. 1.5: Mechanism for Reduced Endwall Boundary Layer Turning Radius and Passage Vortex Formation [7]
reversal of pitchwise velocity in the boundary layer leads to a vortical structure and the passage vortex is born. Yamamoto [25] demonstrated that the strength of the passage vortex is largely dependent on the overall flow turning angle through the blade row. His depictions of the wake near the trailing edge clearly demonstrate the dominance of one tip endwall and one hub endwall passage vortex in the blade wake.

1.2.2.2 Horseshoe Vortex

Also at both endwalls, incoming boundary layer fluid lifts off from the endwalls upon encountering a chordwise pressure gradient due to the blade leading edge presence. The fluid splits into a pressure and suction side leg of a “horshoe” vortex. An explanation of the growth of the vortex system is shown below in Figure 1.6. Due to the pressure gradient between pressure and suction surfaces, the pressure side leg migrates toward the suction surface of the adjacent blade and in doing so merges and grows with the passage vortex.

Meanwhile the suction side leg hugs the blade suction surface and upon being met by the passage vortex and pressure side leg of the horshoe vortex, it also separates from the endwall and proceeds to encircle the merged passage/horshoe conglomeration. Because the majority of boundary layer fluid has been ingested by the three-vortex combination clinging to the suction surface, a new boundary layer is formed at the liftoff point of the pressure side horshoe, suction side horshoe, and passage vortices. Again, it is emphasized that the entire horshoe and passage vortex interaction occurs at both endwalls.
1.2.3 Leakage Loss

Leakage and endwall losses combine to account for 50-70% of the total losses across a turbine or compressor rotor, but of the two the leakage flow has the greater impact [13]. Similar to the tip vortices on a finite wing, a pressure gradient over the wing or blade tip between the pressure and suction surfaces drives a spanwise mass flow over the tip. Typically, the resulting flow separates over the abrupt turn encountered at the tip and forms a vortical structure as it departs the blade tip before the trailing edge and propagates downstream.

However, Denton [6] demonstrated different tip flowfields for “thin” blades, defined by a thickness to gap ratio, $t/\tau$, of less than four, versus their “thick” counterparts. Tip flow over a thin blade does not reattach onto the tip surface, which presents an
Fig. 1.7: Qualitative Structure of the Tip Leakage Vortex [2]
altered mechanism for the tip discharge shown in Figure 1.8

As mentioned previously, the tip gap mass flow escapes the rotor frame and thus does not participate in the input of work into the rotor. Secondly, tip flow is laden with viscous stresses in the gap and in the endwall region yielding hot spots of entropy production. Intuitively, large gap sizes yield a higher leakage mass flow rate while viscous effects limit the mass flow in small gaps. Thus, the two most important factors affecting tip flow are the relative gap size and the blade loading, which creates the pressure gradient to force the tip leakage. The blade loading may be seen as a strong function of blade geometry, flow inlet angle, and axial chord Reynolds number.

1.2.4 Shortcomings of a Linear Cascade

Linear cascade experiments have been used for many years to quantitatively as well as qualitatively describe the flow physics across turbine rotors. However, several
key distinctions limit an exact representation of the turbine rotor. The majority of
the idealizations stem from a lack of endwall relative motion with respect to the blade
tip in a linear cascade. In a rotating machine, the opposite is true. Although the flow
pattern is not greatly altered, rotation in a turbine tends to reduce the leakage flow
by increasing the pressure on the suction side of the gap [16]. An additional artifact
of tip flow investigations in linear cascades that is not observed in rotating rigs is a
recirculation zone planted on the endwall. The effect is repeatedly mentioned in the
literature ([27], [11], [20], and [7]). Figure 1.9 demonstrates two observations of the
endwall recirculation region.

Rotation also yields a radial variation in blade loading, which results in a radial
pressure gradient of magnitude equal to that of the observed pitchwise pressure gra-
dient [15]. In addition, a scraping vortex on the leading surface of the blade can be
observed in rotating rigs and results in altered tip loading.
Most importantly, although relative endwall motion increases leakage flow over a compressor rotor, it reduces the leakage over a turbine rotor. Yaras and Sjolandar [28] found the discharge coefficient to be half of its linear cascade value at full tip speed in a rotating rig. On the other hand, they also found a much reduced casing boundary layer thickness in the rotating frame than shown in the tip endwall of a linear cascade. In a real turbine, multiple stages upstream will wildly distort the tip flowfield in particular and the relative motion of the casing contributes to a thin boundary layer, while a linear cascade allows for an undisturbed upstream condition and a thick endwall boundary layer. Thicker cascade boundary layers may reduce leakage flow to underrepresent tip loss or may result in a stronger passage vortex to detract from leakage loss but increase secondary loss. Regardless, a linear cascade presents an idealized condition. Denton [6] points out that a linear cascade can only truly simulate the flowfield of the first stage rotor. For these very reasons, a rotational turbine facility is currently under design in the Hessert Center, possibly to be used for an in-depth study of the current analysis.

Other idealizations include the absence of cooling flow common to turbine rotors, which may provide minimal thermal and aerodynamic alterations to the flow or the presence of complicated shock patterns due to the relative motion of the casing.
1.3 Flow Control of the Leakage Flow

1.3.1 Control Techniques

Upon the advent of the two most successful active flow control schemes, plasma actuators and synthetic jets, flow control was logically divided into two categories. Passive flow control entails a single-state actuation mechanism, such as fixed geometric modifications. Active control provides a means to alter the flow field in an unsteady manner and to disable the actuation altogether. Both types of control hold demonstrated potential to turbine tip flow fields.

1.3.1.1 Passive Control

Earliest flow control techniques in general applications and also in turbines fell under the passive definition. Winglets, shrouds, steady cooling flow, rub strips, grooved endwalls, knife edges, and squealer tips are the most common of these techniques. Each device attempts to limit the mass flow through the gap, often positioning the geometry in close proximity to the outer casing. The devices must not damage the engine upon rubbing the endwall when the tip clearance rapidly shrinks as in Figure 1.2. Of greater concern are the massive thermal loads to be withstood by any actuation techniques to be applied in temperatures near throttle settings of $2200^\circ R$. Although effective in tip loss reduction, the squealer tip stands as a fence to recirculating and extremely hot gas, which poses challenging thermal constraints on any known material.
1.3.1.2 Active Control

Active control by nonintrusive plasma actuators is the overriding objective of the general project, which encompasses this preliminary passive investigation. Disadvantages of the most popular passive techniques to date have been discussed and all highlight the potential of a flush-mounted, lightweight, mechanically simple, and in-flight adjustable actuation scheme such as plasma actuation.

In order to logically proceed with active control, the underlying philosophy dictates a close documentation of flow physics under the promising squealer tip passive control technique. Knowledge of the physical changes brought about through the squealer tip provides a starting point and a benchmark from which to judge performance of plasma actuator active control. Further, Bindon and Morphis [3] mentioned difficulty in producing similar secondary flowfields over the same blade geometry but in different tunnels. Thus, of equally great importance is to undergo the current investigation in equivalent experimental conditions in which the plasma actuator tips will be tested.

1.3.2 History of the Squealer Tip

On 20 April, 1971, the U.S. Patent Office granted a patent to Fred Gross, Jr. for a new labyrinth seal, the first true tip modification to resemble a squealer tip. Several previous labyrinth seal designs involved cooling air ejection, shrouds, and interlocking structures on the blade tip and blade endwall to minimize the leakage flow. Gross sought to fit the pressure-side rim of the rotor with an airfoil oriented to pump leakage flow back towards the pressure side. Although Gross’ approach was simply to limit
the amount of fluid bypassing the rotor without performing any useful work and was not concerned with a tip vortex or any mechanism for entropy creation, the geometry shown in Figure 1.10 below predates any mention of the “squealer” tip.

Because preliminary investigation of the squealer concept was pursued in industrial settings before its inception as an interesting area of academic research, the first mention of the squealer geometry in turbomachinery literature dates to as recent as 1982 in the first of an exhaustive two part tip flow modeling and geometric design optimization by Booth, Dodge, and Hepworth [4]. In various water tunnel rigs, they
slightly varied local Reynolds number and gap size and experimented with 17 different tip geometry configurations, including thick and thin flat surfaces, a knife edge, grooved tips (i.e., a suction and pressure side squealer), a pressure side squealer, and variations of the first winglet implementation such as a straight, tilted and filleted designs. The authors classified the results based on the overall decrease in leakage mass flow, measured in terms of a non-dimensional discharge coefficient, under the rational that a decreased leakage flow rate decreases the overall entropy creation from the leakage vortex. Seen in Figure 1.11, the knife-edge was judged the most effective with a 25% reduction in discharge coefficient from the baseline flat tip case but generally disregarded due to what the authors deemed infeasible heat transfer and mechanical design constraints.

Interestingly, the knife-edge configuration is about as close as the authors came to the suction-side squealer. The authors further confirmed the superiority of squealer geometries to other passive techniques when they ranked the grooved tip, pressure-side squealer, and winglet as the three most successful reducers of leakage mass flow, respectively. Essentially, the grooved tip presented a full partial and suction side squealer design. As previously mentioned and readily observed in Figure 1.11, the design labeled a “squealer” by the authors was oriented on the pressure side only, explaining the preference of the grooved, or pressure and suction side squealer tip. Despite the most promising reductions in discharge coefficient, the groove and squealer designs were shown to be greatly influenced by Reynolds number and specific blade geometry. Such sensitivity was used by the authors to award the overall less effective
Fig. 1.11: Tip Geometries Investigated by Booth [4]
but more consistent winglet design as the superior passive tip geometry to reduce leakage loss.

Bindon and Morphis [3] closely examined effects of the separation bubble born on the sharp pressure side edge of a flat tip surface. A flat square tip, flat tip with radiused pressure edge, and contoured tip with radius pressure edge constituted the three geometries considered, shown in Figure 1.12. The contoured tip geometry was essentially the first suction-side squealer tip design.

Elimination of the bubble was found to dramatically reduce internal total pressure loss and entropy creation in the gap by 78% for both radiused tips, but also shown to increase the mixing loss of the leakage jet with the passage flow by 66% for the flat radiused tip and by 39% for the contoured radiused tip. Overall loss coefficients, mass-averaged over quarter span and one pitch were lowest for the contoured design, followed by the flat tip, and finally by the flat radiused tip.

It was hypothesized that the reduction in gap loss was due to the absence of the major contribution of separation bubble viscous effects. However, once the separation bubble was eliminated, Bindon and Morphis could not locate an area of high loss con-

Fig. 1.12: Tip Geometries Investigated by Bindon [3]
centration in the leakage jet that had previously been attributed to the bubble. Thus, the authors were lead to conclude that any activity associated with the separation bubble must not contribute directly to the mixing loss. Nevertheless, the authors commented on the tremendous potential of “contouring” and the work can be seen as a successful seminal demonstration of suction-side squealer tip flow control.

Results of Bindon and Morphis were evaluated by Heyes et. al. [9], who provided an interpretation of tip performance based on discharge coefficient. In two linear cascades, suction and pressure side squealer tips were compared with a plain tip baseline condition. Measurements included static blade pressure distributions, endwall static pressure distributions, hot wire gap surveys, and exit plane wake surveys at axial locations of 1% and 12.5% of axial chord downstream of the trailing edge. Gap size of 1% and 1.3% axial chord were investigated. Contours of stagnation pressure loss for the three geometries demonstrated varying distances of the leakage vortex loss core from the suction side, suggesting that leakage flow momentum was highest for the suction side squealer and lowest for the pressure side squealer.

In addition to removing the separation bubble, rounding of the pressure side edge increases the contraction undergone by the flow into the gap region. The result is reduced loss inside of the gap. However, an increase in contraction coefficient leads to an increase in discharge coefficient, which ultimately correlates to an increase in the overall loss. Subsequently, tip geometries should reduce the contraction coefficient even at the expense of higher loss production in the gap. The authors cite the failure of Bindon and Morphis [3] to reduce overall loss with a radiused pressure edge as
evidence of this conclusion. Slight reductions in discharge result with minimal—even one thousandth of an inch—radiusing of the pressure side edge. With the suction side squealer, the *vena contracta* moves to form on the suction side of the gap, and it is this jet contraction which obstructs the flow path and results in less discharge and loss. Authors observed, at least in some cases, less loss associated with the suction squealer and supported the design principle of limiting mass flow in the gap to limit tip loss.

After a bustle of activity in the late 1980’s and early 1990’s, experimental research activity pertaining to squealer tip geometries fell silent as computational techniques began to replace empirical correlations. Heat transfer problems with the squealer tip were pursued in experiment and simulation. However, recent experimental studies by Dey [7], Carter [15], Papa et. al. [17], Azad et. al. [1] and Key and Arts [12] highlight the limited understanding but loaded potential of tip leakage flow control as one of the last few arenas of turbomachinery efficiency improvement.

Dey experimented with three different suction side squealer configurations. He found that leakage vortex was weakened as the gap was reduced from 1.34% to 0.72% of the span for a flat tip blade. Squealer-tipped blades performed better than the flat tip baseline blades only when the clearance was very small. A pressure side squealer addition did not register any beneficial effect. Squealers extending from the trailing edge to about 4% chord performed best.

Focusing on heat transfer complexities in the gap region, Papa et. al. [17] and Azad et. al. [1] conducted detailed mass transfer measurements inside of multiple gaps
with squealer of heights 4.8% and 3.77% of span, respectively. Loss measurements were not conducted, but mass transfer through the gap was lower with the squealer tip and than a flat tip and intuitively decreased with gap size. An increase in gap size moved the starting point of leakage farther downstream on the blade tip. Lower gap sizes caused an early leakage from the blade tip and a recirculating, dead-flow zone near the trailing edge after the gap flow had departed.

Most recently, Key and Arts investigated pertinent parameter changes of cascade exit Mach and Reynolds number in terms of aerodynamic loss. Using blade static pressure taps and tip endwall pressure taps, they confirmed that a suction-side squealer tip was most effective in reducing loss but resulted in an undesirable recirculation zone. Variation in Reynolds number was not registered in the squealer or flat surface tip blade loading, and the presence of the squealer was only minimally reflected in tip blade loading. Endwall pressure ratios were reduced, and the tip leakage vortex was weaker, and thus larger, for the squealer tip condition. Lastly, pitchwise-averaged aerodynamic losses increased towards the tip, and a systematic shift in chordwise location of a local maximum aerodynamic loss due to secondary flows is observed with the squealer tip.

1.4 Objectives

After examining contributions in the literature pertaining to unshrouded turbine blades and squealer tip modifications, it is obvious that much progress has been made but also that the experimental data set is far from complete. The two most critical
parameters to the tip leakage flow appear to be the Reynolds number, responsible for loading the blade and feeding fluid over the pressure side edge and into the gap, and the gap height, which ultimately regulates the amount of mass flow through the gap and the fluid behavior over the blade tip.

The effects of these two critical parameters are sought in a more expansive data set for flat and partial suction-side squealer tip geometries. Further noting the sensitivity of the flowfield, results are desired across the parameters and tip geometries for a single blade and experimental apparatus. Little work has varied Reynolds number for either tip geometry, and for the squealer tip the available data set for Reynolds numbers near real operating conditions is severely lacking.

Aims of this research are to develop an experimental test condition, data acquisition tools, thorough analysis, and complete data set to document the effect of more realistic Reynolds numbers and gap heights for both flat and squealer tips. Advantages and limitations to the squealer tip modification are sought in terms of total pressure loss. Further, understanding of the flow mechanism is desired through multiple measurements taken on the blade, above the gap, and in the wake. Through efforts to closely examine these parameters and provide a more complete data set, foundations for active flow control by plasma actuators will be laid and motivations for active control clearly established.
CHAPTER 2

EXPERIMENTAL PROCEDURE

The ensuing experimental investigation was conducted in the main laboratory of the Hessert Laboratory for Aerospace Research. Construction of an entirely new inlet contraction and test section proved to be the first requisite task of the project. All elements of the design, construction, and procedure of data acquisition involved in the experiment are described in detail in the following chapter.

2.1 Wind Tunnel

2.1.1 Inlet

Due to space limitations and the necessary flow turning angle of 95°, the final wind tunnel took on an unusual appearance. As seen in Figure 2.1, air was drawn near the ceiling of the main laboratory, down through the section, and into the horizontal diffuser. To achieve an acceptably steady freestream flow condition, stagnate air was smoothly conditioned to enter the tunnel by two 18.5 and two 11-inch sections of halved 2-inch PVC pipe.

Incoming flow then met eight inches of coarse honeycomb immediately followed
Fig. 2.1: Wind Tunnel Design
by two consecutive fine mesh screens positioned eight inches apart. These sections were structurally supported by stained and braced plywood to prevent any warping effect and were sealed at the edges with wood glue.

Connected with a wood screw-mounted metal flange, the inlet was designed with a cross-sectional area contraction ratio of 8:1. The contraction contour was defined by a fifth-order polynomial defined according to the six boundary conditions of fixed initial and final positions, fixed initial and final slopes, and fixed initial and final concavities. The two curves to set the contour of the inlet under these constraints are represented by the polynomials,
\[ y = 8.4853 - 0.0112x^3 + 9.9199 \times 10^{-4}x^4 - 2.3381 \times 10^{-5}x^5, \]  
\( (2.1) \)

and,

\[ y = 5.6569 - 0.0075x^3 + 6.6132 \times 10^{-4}x^4 - 1.5588 \times 10^{-5}x^5. \]  
\( (2.2) \)

Coordinates were developed according to the curves and were then transposed into a reference frame suitable for construction. Specifically, arc lengths were computed at every coordinate so that the curves could be replicated on four sheet metal surfaces and then welded together.

### 2.1.2 Test Section

Prior to the beginning of this experiment, similar work on LPT PakB blades had been done with much greater aspect ratio in a linear cascade located in the lower level of the Hessert Center. With similar funding, motivations, and experimental approach, the current project was to progress in conjunction with the larger facility. As such, the design of the test section was constrained primarily by two issues: (a) matching the solidity of 1.13 set in the design of the previously-constructed cascade, and (b) matching the cross-sectional area at the entrance of the diffuser. The test section was then constructed with an initial cross-sectional area of 24 square inches and a final cross-sectional area of 20 square inches. Fabricated from Plexiglas for future flow visualization application, the section contained three cast turbine blades and an adjustable tailboard. The tailboard was incorporated to permit small adjustments
in the minimal cross-sectional area, which at a set back pressure would change the overall flow velocity through the cascade and thus the blade pressure profiles. Such a procedure was implemented in the large AR PakB facility and proved successful in slightly altering the blade profile to match an Eulerian computational solution. However, due to the nature of the highly three-dimensional flow and the limited aspect ratio of the high speed cascade, slight alterations in back pressure were inadequate to match the Eulerian solution. A baseline midspan distribution without any gap spacing was referenced instead.

Two of the cascade blades were positioned to intersect the walls at the leading and trailing edges. Thus, the middle blade was instrumented for close investigation of the flow physics. Methods of instrumentation to be discussed included blade surface midspan and tip static pressure taps, gap-side endwall static pressure taps, and a five-hole probe that was traversed in the wake of the subject blade.
When the five-hole probe was introduced to the tunnel, a slot was cut in the original section wall such that the probe arm could be inserted and moved in two directions by a traverse. Then, a volume encompassing the slot had to be sealed to prevent the induction of a wall jet through the slot under the high suction gradient. A box was designed to hermetically isolate the slot and a two-dimensional traverse mechanism from the ambient conditions. The slot was located 1.0 \( c_x \) downstream of the trailing edge.

In order to set freestream velocity, one of the blade static pressure taps was initially fixed at the stagnation point of the middle blade to record total pressure while an upstream static tap would record a reference static pressure. However, recognizing that the blade stagnation point could shift with Reynolds number, a Pitot-static tube was inserted as far upstream and as close to the wall without boundary layer interference as possible, to record both total and static pressures. The Pitot-static tube was reinforced with an additional \( \frac{11}{16} \)-inch block of Plexiglas mounted to the test section wall. A turn-knob allowed for adjustment and hold of the Pitot-static tube in the desired location. The minimally intrusive measurements of total and static pressure could be used to set the freestream Reynolds number using conservation of momentum under the assumption of incompressible flow (i.e., Bernoulli’s equation).

With the goal of achieving transonic axial flow Mach numbers through the cascade and thereby simulate the local relative blade tip Mach condition, the inlet and test sections were designed to mount on to a permanent diffuser. The diffuser leads to a system of three high powered vacuum pumps which may run independently or in
2.1.3 Diffuser & Vacuum Pumps

A linear diffuser follows the test section for approximately five feet until meeting the rotary valve of the vacuum pump system. The system has three such valves, which permit three different test sections to be interfaced with the vacuum pumps at any time. The three pumps are identical, Allis-Chalmers Model 1165 water-sealed, rotary pumps. At the design condition of 435 r.p.m, each pump can pull a flow rate of 3310 cubic feet per minute of standard air to create a back pressure of 18 inches of H$_2$O.

2.2 Linear Cascade

In the effort towards consistency, the cascade geometry held identical angular dimensions to the high AR, low-speed cascade. Incoming flow was turned 95 degrees through the cascade, meaning the cascade itself has an inlet angle of 55 degrees and an exit angle of 30 degrees, as shown in Figure 2.5. The flow turning and inlet angle define the cascade as a nozzle, accelerating flow through the passage. The solidity was set to the high AR, low-speed cascade value of $\frac{c}{P} = 1.13$. Consistent with the turbine blade literature, the chord used in the defining the solidity is not the actual chord, but a chord measurement down the axis of revolution of the turbine stage.

The Pratt & Whitney low pressure turbine PakB geometry holds a stagger angle, $\zeta = 26.16$ degrees. The span of the blade was set by the vacuum pump diffuser width.
The blades were cast by pouring a urethane mixture in an NC-machined Alcoa MIC-6 cast aluminum mold. A polyol resin, thickened with molecular sieves and glass microspheres, was mixed with an equal part isocyanate resin and microsphere mixture to compose the urethane casting material. Before coating the resin, the mold was then sealed with Plexiglas plates on both ends and bolted together. When static blade taps were desired, tygon tubing was wedged into shallow holes drilled into the mold surface. When not required, these holes were filled with clay, and the clay was then trimmed to be flush with the mold surface. Both inside surfaces of the mold were cleaned with alcohol and three coats of modeling wax were applied to aid
in separation of the blade from the mold upon setting. A cardboard ring was sealed with wax around a hole drilled in one of the Plexiglas surfaces, in which two small spillover holes were also drilled to indicate a uniform cast. The polyol and isocyanate resins were then combined for 90 seconds before they were poured into the hole until the mold was completely filled, as seen in Figure 2.6. After less than 24 hours, the mixture was completely set and the blade was easily removed from the mold. With several strips of capton tape placed on the mold before casting, the original cast of the instrumented blade can be seen to include a 5mm deep indentation to accommodate a preliminary, flush-to-surface plasma actuator.

The choice of material and the amount of material used in the mold constituted
a conservative mold design to ensure as straight a blade as possible as well as to aid in the milling process. In the production of MIC-6 aluminum, molten metal is continuously cast eliminating weaknesses and seams previously inherent to the casting process. Pictured in Figure 2.7, the mold was machined to high tolerance using a numerically controlled mill. All elements of the mold design focused on the casting of completely straight and identical blades. Blades were cast to a span of 4.5 inches and then cropped to a span of 4 inches, the depth of the test section. The middle blade was trimmed further to allow for a maximum gap setting of 5% axial chord. Numerous sheet metal spacers with thickness of 1% and 0.5% of axial chord were trimmed to the PakB airfoil to allow placement between the blade and the fixed wall. In this manner, gap settings from 5% to 0.5% of axial chord were achieved. Rubber spacers were also trimmed to the PakB shape and mounted to the gap side of the blade in several cases to eliminate the gap altogether and establish baseline conditions.

Two standard 10-24 $\frac{3}{4}$-inch screws secured each end of the two outer blades while
the middle blade was secured by two such screws only on the cantilevered end. Sixty-one lines of tygon tubing as well as four electrical contact leads for the future plasma actuator configuration were passed from underneath the middle blade through holes drilled in the Plexiglas endwall. Holes were drilled in all sheet metal spacers to allow for the two securing screws, tygon tubes, and plasma leads to pass through to the blade. The plasma leads were positioned to sit flush with the blade surface and cut to just extend to the blade hub in a minimally intrusive manner. Further, the leads were smoothed with a modeling wax every time the blade was moved or altered to minimize their effect on the flow field.

As the preliminary flow control effort in the facility, the current research concentrated on documenting an interesting passive flow control technique known as a partial, suction-side squealer tip. Only after a fuller understanding of the altered flow physics with a successful passive geometry can an ambitious application of active
Fig. 2.8: Partial, Suction-side Squealer Tip

flow control such as plasma actuation be undertaken. Specifically, the investigation
detailed the effects of a partial suction-side squealer tip mounted to the end of the
blade in the gap region. To construct the squealer geometry, a sheet metal strip with
length approximately equal to 75% of the suction surface and height of 0.025 $c_x$ was
chosen to be consistent with previous studies (Figure 2.8).

2.3 Traverse Mechanism

In order to perform many surveys of a planar area in the wake of the blade, a fully
automated, two-dimensional traverse was assembled. The two directions of interest
were the spanwise direction (into the test section) and the pitchwise direction, situated
along the 30 degree inlet angle of the cascade shown in Figure 2.5 on page 37.

A Velmex MA2509-P10 9-inch travel traverse was coupled with a Applied Motion
Products HT23-396 stepper motor and PD 2035 step motor controller to traverse in
the pitchwise direction. Mounted to the 9-inch traverse was a fabricated miniature slide assembly. The assembly is pictured in Figure 2.9 and consists of a Micro MO AM1524 stepper motor, NB 115 mm slide, and $3\frac{3}{16}$-inch pitch screw, and machined parts to which the survey probe was mounted. A motor controller was assembled by Joel Preston in the Hessert Center machine shop and communicated to the motor information for appropriate number of steps, direction, and voltage level. Without an automatic voltage down setting when idle, the small stepper motor was in danger of overheating with extended experimentation. To rectify the problem, an additional bit was sent to the controller to lower the voltage send to the motor when it was not being used. As designed, the entire traverse assembly effectively gathered data over the course of extensive data sampling often eclipsing 90 minutes.

To prevent the induction of a wall jet due to the high pressure gradient applied across the milled slot, a Lexan box was constructed to hermetically seal the immediate
surroundings of the slot and thus eliminate the pressure gradient across it. Further, blade static pressure taps and electrical connections to control the traverse had to be taken outside of the box. An ITT Cannon MIL-C-26482 box-mounting miniature electrical connector linked the motor controllers to the traverse motors located inside of the box, and was specified to leak less than 0.001 cubic feet per hour under a 100 psi differential pressure gradient. Steel tubing was wedged through small, drilled holes through the Lexan walls to bring all of the tygon tubing lines outside of the box. Through the use Lexan walls, silicon sealant, and reinforcing Lexan blocks welded to the original box walls, the entire box assembly was structurally supported and ensured air tight to guarantee an equalized pressure gradient on both sides of the slot. The pressure seal was tested visually by burning an incense rope at all of the edges of the box and ensuring that none of the smoke leaked in while the suction gradient was active. Lexan proved to be an ideal material for modifications to the box as it was resistant to stress fracture under repeated drilling. It was also resilient to the suction loads applied as the walls would elastically flex where Plexiglas may have cracked or shattered.

2.4 Five-hole Pitot Probe

Planar pressure contour plots are useful to distinguish the flow structures in a given wake. Such pressure plots are most often constructed through the use of a Pitot probe or a rake of total pressure ports, which can quantify the magnitude of the velocity vector. However, if the true flow vector is not approximately parallel to the
body of the Pitot probe, the magnitude reading can be incorrect. By obtaining the
direction and hence true magnitude of the velocity vector, five and seven-hole Pitot
probes can shed this assumption of one-dimensional flow and paint a more accurate
picture of the flow structures.

Noting these motivations, a United Sensor 1/8” diameter five-hole Pitot probe was
used in the wake surveys. A schematic of the probe body and detail of the chamfered
head design can be seen in Figures 2.10(a) and 2.10(b). The probe featured a length
of 9 inches, a recessed head configuration that positioned the actual head of the
probe in the same streamwise location as the sting to which it was mounted, and
a hemispherical take-off tube transition which permitted the use of common tygon
tubing to connect the five tubes to a Scanivalve system.
Fig. 2.11: Calibration Figure to Determine Pitch and Yaw Angles from Measured Port Pressures

A tried and true calibration technique detailed by Bryer & Pankhurst [5] was utilized to interpret measured port pressures in terms of flow properties such as dynamic, total, and static pressures. The calibration technique involved constructing a grid of pitch and yaw angles, and determining a static and dynamic flow coefficient at each point on the grid. Figure 2.11 allows for the interpolation of pitch angle, \( \alpha \), and yaw angle, \( \beta \), by plotting the quantities,

\[
f(\alpha) = \frac{P_3 - P_1}{P_5 - P_m}, \tag{2.3}
\]

and,

\[
f(\beta) = \frac{P_2 - P_4}{P_5 - P_m} \tag{2.4}
\]
where $p_m$ is defined in Pascals as,

$$p_m = \frac{p_1 + p_2 + p_3 + p_4}{4}. \quad (2.5)$$

At every $\alpha-\beta$ calibration point in Figure 2.11, static and dynamic pressure coefficients were defined as,

$$S_p = \frac{H - p_5}{p_5 - p_m}, \quad (2.6)$$

and,

$$Q_p = \frac{p_5 - p_m}{\frac{1}{2} \rho V^2}. \quad (2.7)$$

where $H$ was set as the total pressure, in Pascals, measured during calibration from an adjacent, aligned Pitot tube. Values of the two coefficients were plotted with pitch and yaw angles in Figures 2.12 and 2.13. Armed with pressure data from all five of the probe’s pressure ports, the actual pitch and yaw angles were found from Figure 2.11. Knowing the values of pitch and yaw, corresponding values of static pressure coefficient and dynamic pressure coefficient were read from Figures 2.12 and 2.13, respectively. Once values of the coefficients were determined, dynamic pressure was defined by inverting Equation 2.7. Similarly, total pressure, $H$, was found from Equation 2.6, and knowing total and dynamic pressures, Bernoulli’s relationship was used to give the static pressure.

The technique demonstrates results accurate to $\pm 3\%$ of the freestream measurement when yaw and pitch angles were limited to $\pm 25^\circ$, which proved sufficient. For
velocities entering the compressible flow regime, uncertainty in the technique becomes inescapable and new calibrations must be undergone.

2.5 Pressure Measurements

2.5.1 Blade Surface Taps

By drilling shallow holes in the aluminum mold and inserting threads of 0.020-inch O.D. tygon tubing, the middle blade was cast with 15 static taps flush with both the pressure and suction sides of the blade, at spanwise locations of 0.5 S (midspan) and 0.995 S. To place the taps, the pressure and suction arc lengths were cut uniformly into 15 segments. Tap locations are displayed in Table 2.1. One additional tap was placed on the leading edge at the midspan location, to total sixty-one static taps.
## TABLE 2.1

**PAKB SURFACE STATIC PRESSURE PORT LOCATIONS**

<table>
<thead>
<tr>
<th>Port</th>
<th>Pressure</th>
<th>Suction</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$x/c_x$</td>
<td>$s/S_{ps}$</td>
</tr>
<tr>
<td>stagnation</td>
<td>0.0048</td>
<td>0.0043</td>
</tr>
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<td>0.0672</td>
<td>0.0605</td>
</tr>
<tr>
<td>2</td>
<td>0.1389</td>
<td>0.1209</td>
</tr>
<tr>
<td>3</td>
<td>0.2145</td>
<td>0.1814</td>
</tr>
<tr>
<td>4</td>
<td>0.2926</td>
<td>0.2418</td>
</tr>
<tr>
<td>5</td>
<td>0.3706</td>
<td>0.3023</td>
</tr>
<tr>
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<td>0.4463</td>
<td>0.3628</td>
</tr>
<tr>
<td>7</td>
<td>0.5181</td>
<td>0.4232</td>
</tr>
<tr>
<td>8</td>
<td>0.5854</td>
<td>0.4837</td>
</tr>
<tr>
<td>9</td>
<td>0.6481</td>
<td>0.5442</td>
</tr>
<tr>
<td>10</td>
<td>0.7067</td>
<td>0.6046</td>
</tr>
<tr>
<td>11</td>
<td>0.7616</td>
<td>0.6651</td>
</tr>
<tr>
<td>12</td>
<td>0.8132</td>
<td>0.7255</td>
</tr>
<tr>
<td>13</td>
<td>0.8619</td>
<td>0.7860</td>
</tr>
<tr>
<td>14</td>
<td>0.9081</td>
<td>0.8465</td>
</tr>
<tr>
<td>15</td>
<td>0.9521</td>
<td>0.9069</td>
</tr>
</tbody>
</table>
Six taps on the midspan distribution and seven on the tip distribution were damaged either in blade casting or during the process of squeezing the tygon tubing through the test section endwall to place the blade. However, even with several lost taps, the distributions proved invaluable in this study.

2.5.2 Endwall Static Taps

Of interest in similar investigations, thirty wall static taps were superimposed in ten rows of three over the PakB geometry on the gap-side test section endwall, as depicted in Figure 2.14. Major differences in flow structure passing “under” the blade through the gap region could be observed with these static pressure measurements. Interesting correlations could also be drawn between endwall static pressure contour plots and the wake pressure coefficients gathered by the five-hole Pitot probe.
Fig. 2.14: Endwall Static Tap Locations Imposed Over PakB Blade
**TABLE 2.2**

STRUCTURE OF THE FINAL DATA SET, $g/c_x = 0.04$

<table>
<thead>
<tr>
<th>$Re \times 10^3$</th>
<th>Squealer Tip</th>
<th>Midspan $c_p$</th>
<th>Tip $c_p$</th>
<th>Endwall $c_p$</th>
<th>Wake Survey</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>√</td>
<td>√</td>
<td>√</td>
<td>√</td>
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</tr>
<tr>
<td>600</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>√</td>
</tr>
</tbody>
</table>

2.5.3 Five-hole Probe Wake Survey

The five-hole probe was positioned at 100 points in a 2" by 1.25" (10 by 10) grid by the two-dimensional traverse system. At each grid location, a total pressure upstream of the blade, the five pressure readings from the five-hole probe, and one downstream static pressure reading taken from the endwall static taps were referenced against upstream static pressure and recorded. The total of 700 readings took roughly 90 minutes to complete in the laboratory, for each wake survey undertaken.

2.5.4 Scanivalve and Pressure Transducer

Due to the large number of sampled pressures from the blade and endwall static taps, and the five-hole probe surveys, a 4839-2145 Scanivalve port selector was a vital
piece of equipment to the project. The Scanivalve system enables a maximum of 40 pressure port readings with the use of only a single pressure transducer by rotating a ring of attached pressure tubes.

The Scanivalve was connected to a Validyne DP-103 low pressure differential pressure transducer. The DP-103 was advantageous because it could take many different diaphragms, which in turn adjusted the range and thus the sensitivity of the pressure measurement. An 8-22 diaphragm was used for the duration of the testing, with a sufficient pressure range of 1.6 in. Hg. The transducer was further linked to a Validyne CD-23 voltage readout, which in turn supplied voltage values to the interfaced Powerdaq analog-to-digital board. The board could take a maximum voltage of 5 volts. In switching between high and low Reynolds number cases, the CD-23 span was adjusted to achieve an optimal sensitivity—as close to 5 volts without exceeding five volts—to the sensed pressures. The data acquisition was then carried through by means of lin64b, the acquisition computer.

2.6 Data Acquisition

A Linux-equipped computer with a AMD 3500 939-pin 64-bit processor and UEI PD2-MF-16-400/14H acquisition card was utilized for the data acquisition. All acquisition codes were written and compiled in the C language to communicate with the UEI card and Powerdaq AD board. The board offset was typically on the order of 0.0002 volts.

The board was configured to send 7 bits of output data to communicate the
TABLE 2.3

DIGITAL OUT CONFIGURATION

<table>
<thead>
<tr>
<th>Channel</th>
<th>Designation</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Scanivalve Home</td>
</tr>
<tr>
<td>1</td>
<td>Scanivalve Step</td>
</tr>
<tr>
<td>2</td>
<td>Motor 1 Step</td>
</tr>
<tr>
<td>3</td>
<td>Motor 1 Direction</td>
</tr>
<tr>
<td>4</td>
<td>Motor 2 Step</td>
</tr>
<tr>
<td>5</td>
<td>Motor 2 Direction</td>
</tr>
<tr>
<td>6</td>
<td>Motor 2 Voltage Down</td>
</tr>
</tbody>
</table>

information depicted in Table 2.3. Because the only signal digitally acquired was through the pressure transducer display, only one channel of input data was recorded.

Two computer codes constituted the basis of all codes used in blade and endwall static tap, five-hole probe wake survey, and five-hole probe calibration pressure data acquisition. TestPressure.c, originally composed by graduate student Junhui Huang, was modified and implemented to acquire blade profile and static endwall pressure data that did not involve commands to move a traverse. The author extended this code to send commands to a stepper motor to rotate the five-hole probe for numerous angularity calibrations in calibrate.c.

Extensive modification of TestPressure.c permitted the capability to move a traverse in an absolute reference frame, track the current position in a file, and then acquire and append seven pressure readings with corresponding position coordinates to a different file before moving to a new position and repeating the process. Devel-
Fig. 2.15: Structure of the Primary Pressure Acquisition Code TestPressure.c
Fig. 2.16: Code Structure of Traverse.c, Used to Survey a Planar Wake Region with a Five Hole Pitot Probe
opment of this code, *traverse.c*, is attributed primarily to graduate student Daniel VanNess, although significant modifications were made to the original code as well. Flowcharts demonstrating the processes employed by both programs are visible in Figures 2.15 and 2.16.
CHAPTER 3

RESULTS I: EFFECT OF GAP SIZE

Data acquired from the blade static pressure ports, endwall static pressure ports, and a five-hole Pitot probe aft of the cascade are analyzed in the following section. In conjunction with the objectives of the project, the data set is both qualitatively reviewed as well as quantified in plots of static pressure coefficients, blade loading, and overall, pitchwise, and spanwise mass-averaged coefficients of total pressure loss. Throughout the data set, the three parameters of axial Reynolds number, gap height, and tip geometry (i.e. flat or squealer tip) were varied. Results then are logically presented by fixing two of the three parameters and investigating the effect of a change in the one remaining parameter. To minimize overlap in the analysis, the effect of gap size and Reynolds number are presented first for the flat tip. During the investigation of the effect of the squealer tip, the effect of gap size and Reynolds number for a squealer tip is inspected.

To isolate the parameter of gap size, the Reynolds number was held constant through the following analysis. All plots presented below correspond to the flat tip
case, because the final section will more closely investigate the performance of the squealer tip.

3.1 Blade Surface Static Pressure Distributions

Using the acquisition code TestPressure.c, pressure coefficients were calculated across the pressure and suction surfaces of the blade at both 50% and 99.5% spanwise locations. Pressure coefficients were defined according to,

\[ c_p = \frac{P_s - P_i}{P_{ti} - P_i} \]  (3.1)

where \( P_s \), \( P_{ti} \), and \( P_i \) are defined as the local static pressure, upstream total pressure, and upstream static pressure, in volts, respectively. Pressures were recorded in volts only because a differential pressure transducer was used. Pressure coefficients could be calculated by referencing upstream static pressure, \( P_i \), on the transducer and simply dividing voltages of the numerator and denominator. Calculation of the pressure coefficients in this manner was straightforward and eliminated many of the systematic uncertainties associated with the transducer.

Before blade distributions may be presented and considered, a limitation must be mentioned. The test section was designed with a tailboard to adjust the downstream pressure gradient and thus match an inviscid and two-dimensional Euler solution for the pressure distribution. This approach worked well for the low Reynolds cascade, with blades on the order of a meter in span. As shown in Figure 3.1 however, the two-dimensional solution could not be met in the current cascade due to a small span
Fig. 3.1: Blade Midspan Pressure Distribution for $Re_2 = 1.0 \times 10^5$ with 2D Euler Solution

and the highly three-dimensional nature of the flowfield.

Instead, the gap was filled with rubber spacers and a distribution was taken at midspan to provide a benchmark from which to qualitatively evaluate effects of gap height, Reynolds number, and tip geometry on the blade pressure distributions. Both midspan and tip pressure distributions varied substantially throughout the data set, so the midspan pressure distribution without gap shown in Figure 3.1 is the least three-dimensional view of the loading on the PakB blade. Further, because midspan pressure distributions showed similar sensitivity to the change in the parameters and given the proximity of the tip static taps to the gap, distributions presented below
are restricted to the more interesting tip cases.

During blade casting and insertion of the instrumented blade into the test section, several of the tygon tubing lines were ruptured. Thus, not all of the ports shown in Table 2.1 were available for data acquisition but the remaining ports sufficiently describe the pressure distribution.

Figure 3.2 demonstrates blade static pressure distributions for upstream Reynolds numbers of $1.0 \times 10^5$ and $5.0 \times 10^5$ at varying gap sizes. Immediately, blade unloading near the leading edge and enhanced loading near the trailing edge may be observed as differences between the midspan and tip static pressure distributions. Near the leading edge, the pressure difference between pressure and suction surfaces is much reduced. At about $x/c_x = 0.75$, the pressure distribution reaches a maximum value where gap inlet flow velocities peak and the vortex is thought to depart from the blade.
tip. Pressure values then trail off toward the trailing edge with behavior similar to the midspan distribution.

At a smaller gap size, these trends are less noticeable. The loading is stronger near the trailing edge and coefficients are more positive for the smallest gap case $g/c_x = 1.5\%$ than for $g/c_x = 4\%$ and $g/c_x = 5\%$. Blade unloading at the leading edge, blade loading near $x/c_x = 0.75$, and reduction in static pressure along the pressure surface are all reduced in the smaller gap cases for both Reynolds numbers. At larger gap for the high Reynolds number case, the blade unloading is particularly obvious upstream of $x/c_x = 0.4$, and the sudden changes and erratic behavior of the
static pressure coefficients point to a much more volatile and more three-dimensional flow scenario.

Integrating the non-dimensional pressure coefficients yields a measure of overall blade loading,

\[
c_{p,\text{blade}} = \int_0^{c_x} c_p \delta x = \int_{c_x=0}^{c_x=1} c_p \delta \frac{x}{c_x} = \int_0^1 c_p \delta x^*,
\]

where \( c_p \) denotes the static pressure coefficient given in Equation 3.1. Points are then splined together for the midspan distributions and for the tip distributions in Figures 3.3 and 3.4, respectively. Both figures demonstrate a significant drop in blade loading with an increase in gap size, which can be explained by an increased discharged into the tip gap with larger clearance. The drop in blade lift is nearly linear at the tip and for the high Reynolds number case at midspan. At midspan, increasing the gap size by 70% yields a 24% drop in blade loading at the high Reynolds number and a 26% drop for the low speed case.

Tip loading is generally lower than that at the midspan due to high spanwise velocities into the gap on the pressure surface. Low speed blade loading also drops roughly 25% over the gaps surveyed, but the high speed loading drops slightly less at 18.5%. However, the magnitude of the loss in blade loading is nearly equivalent in both cases. That blade loading drops equivalently for both spanwise locations speaks to the highly three dimensional flow at the midspan, and such is the case for actual turbine blades. Reduced blade loading over at least half of the span highlights the loss of work input to the turbine due to flow escaping the blade through the tip gap.
Fig. 3.4: Tip Blade Loading, $c_{p,\text{blade}}$ Across Gap Height
Fig. 3.5: Tip Endwall Contours of Static Pressure Coefficients

as one of the undesirable tip leakage contributions.

3.2 Endwall Static Pressure Distributions

Rows of static pressure ports in the tip side endwall at locations depicted in Figure 2.14 were used to construct contour plots of static pressure coefficient, $c_p$, over the blade and above the gap. Results across gap size are shown in Figure 3.5 and may be cross-referenced with the corresponding blade pressure profiles of Figure 3.2.

Due to the area contraction incorporated into the design of the PakB geometry, the linear cascade acts as a nozzle and incoming flow accelerates through the pas-
sage. This is evident in the gap as well and corresponds to the smooth transition from near zero static pressures at the leading edge to negative values at the trailing edge. Bernoulli’s relation correlates this drop of static pressure to a rise in local fluid velocity. As the plots have been normalized according to Equation 3.1, the static pressure distribution over much of the blade is similar even between the low and high Reynolds number cases.

One interesting feature of the endwall contour plots is an isolated region of negative static pressures occurring directly over the blade. It is hypothesized that such a region of isolated static pressure indicates a high streamwise velocity departure of the tip leakage vortex from the blade tip. The point of departure moves forward with an increase in gap size. For $Re_2 = 1.0 \times 10^5$, the region of negative pressures begin at approximately $0.75c_x$ for the smallest gap size but move to $0.6c_x$ for the larger gap cases. The same does not hold true at $Re_2 = 5.0 \times 10^5$, where the earliest departure is before $0.5c_x$ for the case of $g/c_x = 1.5\%$. In fact, at the higher Reynolds number, the departure zone of the tip leakage flow grows tighter as the gap gets larger. Contour plots of $g/c_x = 1.5\%$ and $g/c_x = 4\%$ represent the cases with the greatest tip blade loading (Figure 3.4), and consequently contain the largest zone of the tip leakage flow departure. Perhaps a larger pressure gradient over the blade tip drives more mass flow into the gap or creates a larger tip vortex to depart from the pressure side edge and traverse the tip region. A strong streamwise velocity is also seen near the mean camber line in the departure zone and shortly upstream, indicating a dynamic streamwise motion tied to the tip leakage flow. An interesting result from the countour
plots is the direct correlation between the area of the departure zone of negative static pressures over the gap and the overall blade loading. The greater the tip blade loading becomes, the larger the departure zone grows.

3.3 Wake Surveys

A five hole probe traversed the wake in the pitchwise and spanwise directions at a downstream location of $1.0c_x$. Slightly more than one third of a span and one half of a pitch were traversed and data acquired in a time-resolved manner. Figure 3.6 below denotes the origin, coordinate system, and region of interrogation.

3.3.1 Vorticity

Using secondary velocity vectors gleaned from the five hole Pitot probe, streamwise vorticity was calculated as,

$$\omega_x = \frac{\delta w}{\delta y} - \frac{\delta v}{\delta z},$$  \hfill (3.3)

in units of $s^{-1}$. Partial derivatives were approximated via second order central differencing at node $(i, j)$,

$$\frac{\delta w}{\delta y} = \frac{w(i + 1, j) - w(i - 1, j)}{2\Delta y},$$  \hfill (3.4)

and second order Taylor series approximations at the boundaries,

$$\frac{\delta w}{\delta y} = \frac{4w(i + 1, j) - w(i + 2, j) - 3w(i, j)}{2\Delta y},$$  \hfill (3.5)
Fig. 3.6: Coordinate System and Region of Interrogation in the Wake
Equation 3.3 was then normalized by the upstream freestream velocity, \( U \) and the axial chord, \( c_x \),

\[
\Omega_x = \frac{\omega_x c_x}{U}. \tag{3.6}
\]

Contour plots of the normalized vorticity, \( \Omega_x \) are provided below with superimposed secondary velocity vectors and \(-\lambda_2\) criterion to outline the vortical structures. The reader is referred to Jeong et. al. [10] for details of the \(-\lambda_2\) technique.

On the left side of the windows of Figure 3.7, a positive vorticity and the zero contour of \(-\lambda_2\) clearly denotes the passage vortex. A region of negative vorticity corresponds to the tip leakage vortex which enters into the top of the picture and is also outlined by the zero contour of the second eigenvalue of the velocity gradient.
Fig. 3.8: Normalized Vorticity Contours for $g/c_x = 1.5, 4, \text{ and } 5\%$ at $Re_2 = 5.0 \times 10^5$

tensor. Rapid changes in vorticity in this small, 1.5 in. $\times$ 2.0 in. window indicate a great deal of mixing and three-dimensionality. Intuitively, the passage vortex is pushed to the left side of the window as the gap grows on the right side of the window. However, the passage vortex is also forced downward and the opposite vorticity associated with the tip vortex moves slightly lower and much closer to the endwall at larger gap sizes. Thus, the interaction between the passage vortex and tip vortex is heightened at small gap sizes due to their proximity to one another.

Secondary velocity vectors demonstrate a strong cross-passage fluid motion of the suction, low-pressure side between the two vortices in Figure 3.8. Extremes in vorticity attributed to the tip side passage vortex and tip leakage vortex are more clearly visible at the higher Reynolds number. Although the tip vortex is tighter, stronger, and closer to the endwall at the lowest gap size, the separation between outlines of
the leakage and passage vortices remains relatively constant at approximately 0.05S. The passage vortex also grows in size but shows higher values of positive vorticity at \( g/c_x = 4\% \) and \( g/c_x = 5\% \). A region of high vorticity also appears on the endwall near the lower right corner of the window at a gap size of 4\%, which may be attributed to linear cascade endwall circulation seen in the literature. It is interesting that a similar region is not present in the 5\% case. The tip vortex may be seen to move lower at each increase in the gap size. Lastly, strong secondary flow vectors are seen on the endwall at the two larger gap sizes while the fluid has very little secondary velocity component at the endwall in the 1.5\% gap case. Undoubtedly the reason lies with the proximity of the tip leakage vortex to the endwall and the compactness of the vortex.

3.3.2 Total Pressure Loss

Corresponding to motivations of the research, great interest lies in total pressure loss downstream of the blade row. In an effort towards consistency with corresponding efforts in the literature, total pressure loss was defined as,

\[
c_{Pt} = \frac{P_{ti} - P_{te}}{P_{te} - P_e},
\]

(3.7)

where \( P_{ti}, P_{te}, P_e \) are the upstream total pressure, downstream total pressure, and downstream static pressure measured in Pascals, respectively. Again, countours of \( c_{Pt} \) are constructed with superimposed secondary velocity vectors and zero contours of the \(-\lambda_2\) criterion. High values of \( c_{Pt} \) indicate high regions of loss, and it is not
Fig. 3.9: Total Pressure Loss Contours, $c_{Pt}$ for $g/c_x = 1.5, 4,$ and $5\%$ at $Re_2 = 1.0 \times 10^5$

surprising that such regions are found enclosed by or in close proximity to the passage or leakage vortices.

Although difficult to judge in the 4\% gap case of Figure 3.9, the passage vortex certainly contains a higher region of loss at the largest gap setting than at 1.5\%. As the vortical structures move farther apart, the region of loss between them grows, which may discredit the idea that two close regions of vorticity generate greater loss due to the enhanced mixing. More loss is generated between the tip vortex and passage vortex in the two cases when they are farther apart than when they are in close proximity. Although peaks in the loss can be seen in the passage vortex for these cases, attention should be called to the region between the two counterrotating structures.
Re = 500K

Fig. 3.10: Total Pressure Loss Contours, $c_{Pt}$ for $g/c_x = 1.5$, 4, and 5% at $Re_2 = 1.0 \times 10^5$

At the high Reynolds number, peaks in loss again are seen inside of vortices, but this time can be found in a clearly defined tip leakage vortex. The fringes of the passage and tip leakage vortex remain close to one another, but the centers of the two vortices appear to be closer in the $g/c_x = 1.5\%$ case. For this reason, the loss in the tip vortex does not peak as high and is instead shared with the passage vortex.

At the lowest gap setting, the two vortices are clearly distinct and close together, and as a consequence the loss is concentrated inside the vortices themselves instead of distributed between them. As the vortices spread out in the wake, a region of low loss connects them and it is this region that must not be overlooked when considering the overall loss. A key distinction between the $g/c_x = 1.5\%$ and $g/c_x = 4.5\%$ cases
is the differing models for the flowfield over the gap (Fig. 1.8). At $g/c_x = 1.5\%$, the blade thickness is large compared to the gap size and the flow reattaches after separating over the pressure side edge. This behavior may lead to a crisper and tighter tip leakage-passage vortex pair with distinct regions of loss separated by regions of little to no loss. At the two larger gap settings, the thin blade model dictates a complete separation over the pressure side edge and a lack of reattachment. Possibly due to enhanced mixing through the gap following such a violent separation, this type of departure from the blade tip causes a larger vortex and greater spacing between the tip and leakage vortices. Thus, not only is the loss attributed to the tip vortex greater, but it extends in a region connecting to other structures in the wake such as the passage vortex and even a region near the endwall.

In order to isolate trends in the loss mechanisms, the total pressure loss was mass-averaged over pitch, span, and the entire window, similar to the technique of Yamamoto [26]. First, the value of interest denoted by $\phi$ but commonly substituted as resultant velocity, axial velocity, or total pressure loss, is arithmetically averaged by the four nearest neighboring grid points in an effort to spatially smooth the values. To arithmetically average values for the grid borders, points outside the grid are linearly approximated based on the values inside of the grid. Yamamoto then defines overall mass-averaged quantities, $\bar{\phi}$ as,

$$\bar{\phi} = \frac{\sum_j \sum_i \phi V_x A_{ij}}{\sum_j \sum_i V_x A_{ij}},$$

(3.8)
where $i$, $j$, $V_x$, and $A_{ij}$ are the spanwise counter, pitchwise counter, axial velocity, and an elemental area defined as a diamond by the four nearest neighbors, $(i - 1, j)$, $(i + 1, j)$, $(i, j - 1)$, and $(i, j + 1)$, to grid point $(i, j)$. Total pressure loss coefficients are given,

$$\frac{C_{Pt}}{\rho V_x^2} = \frac{P_{ti} - P_{te}}{\frac{1}{2} \rho V_x^2}, \quad (3.9)$$

where $\rho$ and $\overline{V_x^2}$ are the density in $\frac{kg}{m^3}$ and mass-averaged axial velocity in $\frac{m}{s}$, respectively. Equation 3.9 demonstrates a departure from Yamamoto’s technique in that the mass-averaged axial velocity is substituted for the mass-averaged resultant velocity. Drastic changes between test cases in the data set caused completely different flow structures to be considered in the interrogation window and as a consequence, the averaged resultant velocity fluctuated between cases. In searching for a more consistent normalization, the axial, streamwise velocity component was chosen. Yamamoto avoided this complication with full spanwise and pitchwise wake surveys under only one Reynolds number and gap size condition.

Overall mass-averaged loss coefficients are then plugged back into Equation 3.8 to yield,

$$\overline{C_{Pt}} = \frac{\sum_j \sum_i C_{Pt} V_x A_{ij}}{\sum_j \sum_i V_x A_{ij}}. \quad (3.10)$$

Intuitively following from Equation 3.8, averaging over one dimension yields a pitch-wise or spanwise mass-averaged quantity. Pitchwise mass-averaged loss coefficients
are defined as,

\[ C_{PL,PITCH} = \frac{\sum_j C_{Pl} V_x A_{ij}}{\sum_j V_x A_{ij}}, \]  

(3.11)

while spanwise loss coefficients were defined by,

\[ C_{PL,SPAN} = \frac{\sum_i C_{Pl} V_x A_{ij}}{\sum_i V_x A_{ij}}. \]  

(3.12)

Figure 3.11(a) shows consistent pitchwise distribution of the loss across the three different gap sizes. A peak near \( y/P = 0.3 \), for the smallest gap, corresponds to the pitchwise location of the passage vortex core in Figure 3.9. Movement of this peak down in pitch farther from the trailing edge is representative of the movement of the passage vortex as the gap size is increased.

Applying the same technique to the high Reynolds number case in Figure 3.11(b) reflects the removal of the passage vortex and tip leakage vortex pair and replacement by a much stronger lone tip leakage vortex as the gap steps from 1.5% to 4% to 5% of axial chord. The distributions more clearly demonstrate the strengthening of the loss with tip clearance and imply that the change in flow model based on a “thick” or “thin” categorization is more drastic at a higher Reynolds number. Again, the cases of 4% and 5% gap are qualitatively similar but differ from the 1.5% gap data. Loss at 5% tapers off slower toward the endwall than the slightly smaller gap condition. Peak in the loss for both “thin” blade cases occurs farther from the trailing edge near \( y/P = 0.25 \) while the 1.5% gap case is maximum near \( y/P = 0.1 \).
Fig. 3.11: Pitchwise Distribution of Spanwise Mass-averaged Total Pressure Loss Across Gap Height

(a) $Re_2 = 1.0 \times 10^5$

(b) $Re_2 = 5.0 \times 10^5$

Fig. 3.12: Spanwise Distribution of Pitchwise Mass-averaged Total Pressure Loss Across Gap Height

(a) $Re_2 = 1.0 \times 10^5$

(b) $Re_2 = 5.0 \times 10^5$
For both Reynolds numbers, the spanwise distributions for all three gap sizes are qualitatively similar. Results are plotted such that the end of the blade is always located at $z/S = 1.0$ and measurements farther down the span correspond to spanwise locations inside the gap. Loss reaches a maximum far down the span and a minimum at the end of the blade in Figure 3.12(a). Shortly after reaching the minimum at the end of the blade, losses slightly grow in the gap region, the effect increasing with gap size. With similar structures located in the wake windows, behavior of the loss can be judged. At nearly every spanwise location of survey, the loss incrementally increases with gap size. In fact, the trend is probably underestimated as less and less of the complete passage vortex is included as the gap size increases, as seen in Figure 3.9. Losses near the blade tip show much less variation with gap height, and loss locations near $0.75S$ show greater sensitivity due to movement of the passage vortex.

At $Re_2 = 5.0 \times 10^5$ the spanwise loss distribution reinforces the similarities between 4% and 5% cases as well as the drastic differences between the thick and thin gap settings as observed in the wake images of Figure 3.10. The smallest gap size in Figure 3.12(b) shows two local maxima at $z/S = 0.78$ and $z/S = 0.98$ corresponding to the passage and tip leakage vortices, respectively. Loss peaks associated with the tip leakage vortex move to $z/S = 0.95$ and a loss coefficient of 0.3 and 0.4 for $g/c_x = 4\%$ and $g/c_x = 5\%$, respectively. Major differences in the tip leakage vortex loss as well as throughout the other spanwise locations are observed even between the two largest gap sizes.

Plotting the overall mass-averaged total pressure loss against gap height gives
Fig. 3.13: Overall Mass-averaged Total Pressure Loss Coefficients across Gap Height
Figure 3.13. As the gap size grows, the total pressure loss in the wake increases in a nonlinear fashion. If the gap height is increased by a factor of 3.3, a 68% rise in total pressure loss is observed for an upstream axial Reynolds number of $5.0 \times 10^5$. Over the same gap increase at a Reynolds number of $1.0 \times 10^5$, total pressure loss measured in the wake survey increases by 23%. An approximately equal change in total pressure loss coefficient is seen from one gap spacing to the next, which means that the effect is magnified at larger gap sizes. Obviously, sound design of turbine blades seeks to minimize the gap height and thus the discharge into the gap. However, the penalty in terms of total pressure loss increases disproportionally at large tip clearances.
CHAPTER 4

RESULTS II: EFFECT OF REYNOLDS NUMBER

4.1 Blade Surface Static Pressure Distributions

Blade surface static pressure coefficients are now analyzed across upstream Reynolds numbers of $1.0 \times 10^5$ and $5.0 \times 10^5$, corresponding to downstream Reynolds numbers of $1.5 \times 10^5$ and $7.5 \times 10^5$. For the smallest gap size, the influence of Reynolds number is seen on the suction surface. At a higher Reynolds number, the suction surface static pressure coefficients are more negative while the pressure surface values remain similar to those of the low Reynolds number case. Negative values of pressure coefficient not seen in Figure 3.2(a) until $x/c_x = 0.7$ are found at the higher Reynolds number cases near mid-chord. The trend of similar pressure side static pressures and differing suction side values is consistent through the two larger gap sizes as well, shown in Figures 4.1(b) and 4.1(c).

For a gap of 1.5% axial chord, the higher Reynolds number values are substantially decreased over the entire suction surface. For the case of 4% gap, both Reynolds
Blade Tip Pressure Distribution

(a) $g/e_x = 1.5\%$

(b) $g/e_x = 4\%$

(c) $g/e_x = 5\%$

Fig. 4.1: Blade Surface Static Pressure Coefficients
numbers hold similar suction side static pressure coefficients near the leading edge but the high Reynolds number cases has reduced values over the range $0.3 < x/c_x < 0.95$. And for a gap height of $0.05c_x$, static pressure coefficients for the high Reynolds case drop noticeably in the suction side region from roughly $0.4 < x/c_x < 0.8$. Such differences are important because a stronger pressure gradient between pressure and suction surfaces pulls more fluid through the gap, the chordwise location of maximum pressure gradient indicates the position of maximum velocity through the gap, and the differences occur over chordwise locations where one expects departure of the tip leakage vortex from the blade tip. In general, the stronger pressure gradient seen in the highlighted regions for the high Reynolds number case means greater mass pulled over the pressure side edge and into the gap and increased mixing and entropy production in the wake.

Although the effect is not uniform over the pressure surface, it is hypothesized that the tip pressure side coefficients are lower than those of the midspan coefficients due to the strong spanwise flow pushing over the tip pressure side static taps and into the gap. However, it is not expected that the coefficients for the two drastically different Reynolds numbers would collapse upon each other, because they have been normalized by the upstream dynamic pressure. The observation indicates that the local spanwise velocity of fluid from the pressure side and into the gap scales proportionally across Reynolds number.

Qualitative results of Figures 4.2 and 4.3 are quantified by integrating the pressure coefficients over the chord, via Equation 3.1. The integrated value, $c_{p,\text{blade}}$, is then
Fig. 4.2: Midspan Blade Loading, $c_{p,\text{blade}}$ Across Exit Reynolds Number, $Re_3$
Fig. 4.3: Tip Blade Loading, $c_{p, blade}$ Across Exit Reynolds Number, $Re_3$

plotted for separate gap sizes across the cascade exit Reynolds number. In all gap cases, the loading is significantly increased from the low to the high Reynolds number case, although the rise is much less dramatic at midspan. The change in blade loading with Reynolds number is consistent for all three gap sizes. Figure 4.2 demonstrates an increased midspan loading at $g/c_x = 1.5\%$ than for the other two cases. However, the corresponding tip loading is not as drastic an increase over the two other tip cases, meaning that gap size has a more drastic effect on the midspan blade loading while Reynolds number is a more critical parameter at the tip.

Shown in Figure 4.3, tip loading drops progressively more across Reynolds number.
as the gap size increases. At $g/c_x = 1.5\%$, $g/c_x = 4\%$, and $g/c_x = 5\%$, tip loading drops 21%, 24%, and 32% from the highest Reynolds number to the lowest, respectively. Therefore, Reynolds number is a more critical parameter at larger gap sizes. High Reynolds number and large gap sizes together result in maximum blade loading, flow driven through the gap, and as a consequence, heightened loss.

4.2 Endwall Static Pressure Distributions

Major differences in the endwall static pressure distributions have to do with the tip leakage vortex roll-up off the flat tip. The smooth acceleration is again observed through the passage except for in a small region where the separation is expected just aft of mid-chord. For $Re_2 = 1.0 \times 10^5$, the departure occurs in the region $0.6 < x/c_x < 0.75$, but at the higher Reynolds number the zone is much larger and begins sooner. The larger area of departure indicates greater mass flow driven through the gap by the heightened tip blade loading.

Increased losses are expected at the greater Reynolds number as the tip blade loading increases and a greater mass flow is thrown into the gap and thus contributes to the mixing loss mechanisms through the passage and in the wake.

4.3 Wake Surveys

4.3.1 Vorticity

At a small gap size and low Reynolds number where the discharge into the gap is low tip leakage vortex is undoubtedly weaker. With less fluid entering the tip clearance
region and entraining in the vortex, more is free to populate the passage vortex, which becomes tighter and shows strong positive vorticity. At a greater Reynolds number, the increase in gap discharge corresponds to a weakening of the passage vortex rotation and generated total pressure loss. Secondary velocity vectors indicate less rotation in the passage vortex, which is confirmed by lower values of positive vorticity. In the same window one notes the tip vortex, which, in addition to moving between the passage vortex and the endwall, has been fed by a greater mass flow into the gap to become tight and strong. The passage and tip leakage vortices share an inverse relationship. Placement of the two structures depends on their relative strengths. A strong passage vortex pushes a weak passage vortex away from the endwall and toward the pressure side in the pitchwise direction. Conversely, a strong leakage
Fig. 4.5: Normalized Vorticity Contours for $Re_2 = 1.0 \times 10^5, 5.0 \times 10^5$, $g/c_x = 1.5\%$

vortex positions itself between the passage vortex and the tip endwall while reducing the vorticity of the passage vortex.

Figure 4.6 provides the best demonstration of the placement of the vortices in the window and is the only case in which an upstream Reynolds number of $6.0 \times 10^5$ was run. At a tip clearance of $0.04c_x$, upstream Reynolds numbers of $1.0 \times 10^5, 2.0 \times 10^5, 3.0 \times 10^5, 5.0 \times 10^5$, and $6.0 \times 10^5$ were checked. At the lowest Reynolds number, when the blade has been loaded least and as a consequence less fluid has entered the tip gap, the tip leakage vortex is weaker and is pushed away from the passage vortex. With slightly more fluid entering the gap in the case of $Re_2 = 2.0 \times 10^5$, the tip vortex enters the pictures with opposing vorticity and in close proximity to the passage structure.
The vorticity of the tip leakage vortex grows with Reynolds number while the passage vorticity remains relatively constant. A strong channel of secondary flow between the two vortices develops and shoots flow across the pitch toward the suction side. At the three highest Reynolds numbers, the part of the passage vortex pictured near the tip leakage vortex appears to elongate and weaken as the leakage vortex and magnitude of its vorticity grows.

Similar behavior to that seen in Figure 4.5 is seen at the two Reynolds numbers of Figure 4.7. Again the tip leakage vortex is originally dominated by the passage vortex. As the mass flow into the gap increase with Reynolds number, the tip vortex gets much stronger and moves between the passage vortex and the tip endwall. Once again, greater gap discharge due to Reynolds number effects creates a drastically more
three-dimensional wake behind the cascade.

4.3.2 Total Pressure Loss

Due to movement of the wake characteristics into and out of the window as Reynolds number is varied, useful qualitative conclusions may be drawn from the data, but limitations to quantifying the loss must be clearly understood.

Comparing the extreme Reynolds number cases in Figure 4.8, the passage vortex loss region becomes tighter and moves pitchwise toward the trailing edge as the tip leakage vortex becomes much stronger and moves between the passage vortex and the endwall. Pitchwise movement of the passage vortex can be attributed to the shift of the tip vortex down in pitch and toward the endwall. Even a weak tip leakage vortex
Fig. 4.8: Total Pressure Loss Contours, $c_p t$ for $Re = 1.0 \times 10^5$ and $5.0 \times 10^5$, $g/c_x = 1.5\%$

has an effect on the pitchwise location of the passage vortex.

For each case, all of the peak loss regions are located in or near the core of a passage or tip vortex contour. Dominance of the tip leakage loss is obvious at the high Reynolds condition. As noted previously, the regions of loss associated with the tip and passage vortex are much more distinct as the vortices grow in strength as indicated by the vorticity magnitude and tightening of the vortex core.

In the case of a larger gap setting of 4%, low Reynolds number locations of the tip leakage and passage vortex locations differ from the corresponding locations at the 1.5% gap under the thick blade flow model. The two structures start in opposite corners of the window, but even a slight increase to an upstream Reynolds number of $2.0 \times 10^5$ causes the pair to interact much more closely. The vortices get closest
Fig. 4.9: Total Pressure Loss Contours, $c_{pt}$ for $Re_2 = 1.0 \times 10^5$ and $5.0 \times 10^5$, $g/c_x = 4\%$

together in the case of $Re_2 = 3.0 \times 10^5$ and strong secondary flow vectors through the channel between the two structures and a large area of low loss hints at the heightened interaction. At the two highest Reynolds numbers, the tip vortex becomes much more distinct and the loss becomes concentrated in the tip leakage vortex core. Strength of the tip vortex as demonstrated by magnitude of the secondary velocities, tightness of the $-\lambda_2$ contours, and magnitude of the loss and as well as vorticity (Fig. 4.6) located inside of the contours, can be seen to drastically increase between upstream Reynolds numbers of $2.0 \times 10^5$ to $6.0 \times 10^5$. Figure 4.9 shows that only at the high Reynolds number cases is the loss truly attributed to a distinct tip leakage vortex, which indicates the importance of high Reynolds number investigations. Further, the
case of $Re_2 = 2.0 \times 10^5$ shows the tip vortex clearly outlined by the $-\lambda_2$ contour but corresponding to very slight total pressure loss. At $Re_2 = 1.0 \times 10^5$, loss of the tip leakage flow fails to register although secondary velocity vectors point to a vortex in the upper right corner of the window. Because the top of the window is the trailing edge and the right side is the endwall, movement of the vortex into the interrogation zone at 4% gap is improbable. As indicated by the $-\lambda_2$ contour, it is more likely that the vortex is present throughout each view in Figure 4.9, but that the tip blade loading is insufficient to supply enough mass flow to the leakage vortex to create a sufficiently strong loss mechanism.

Unseen in vorticity plots of Figure 4.6, a small zone of loss positioned on the endwall can be see in each window of total pressure loss at the 4% gap size. This structure is puzzling but seems to correlate with previous sightings of an endwall recirculation zone common to linear cascade environments in the absence of tip endwall relative motion as portrayed in Figure 1.9.

Figure 4.10 demonstrates similar behavior to the 1.5% case, although now the window captures a larger gap region and less down the blade span. Again a stronger tip leakage vortex drops down in the pitchwise direction and moves off of the endwall as the passage vortex is moved spanwise toward the hub, elongates as a weakened vortical structure, and moves up in the pitchwise direction as well. In addition to the tip leakage flow growing stronger with Reynolds number, the structures become more distinct and the loss is derived more from the structures themselves than from the interaction between the two. Thus, controlling the tip leakage flow cuts down the
losses incurred in the tip leakage vortex at high Reynolds numbers, but also limits the loss contribution from the interaction of a strong passage and weak tip vortex at low Reynolds numbers.

Pitchwise mass-averaged loss coefficients shown in Figure 4.11 correspond to the 1.5%, 4%, and 5% gap sizes. The case of 4% is more crowded as it was the gap setting in which the effect of Reynolds number was most closely examined. The effect of Reynolds number at the smallest gap setting is to strengthen the tip leakage vortex as well as bring it completely into the window and increase the loss closer to the trailing edge in the pitchwise direction. An expansive passage vortex alone causes the loss peak for $Re_2 = 1.0 \times 10^5$ at $y/P = 0.3$. Also shown in 4.11(a), the peak losses at the high Reynolds number have only deviated roughly $0.25P$ at $1.0c_x$ downstream.
of the trailing edge. The fact that the tip leakage vortex tracks so closely to the trailing edge after completing a 95° turn may indicate a belated departure from the tip surface.

As the blade loading increases with Reynolds number, the maximum pressure gradient over the tip grows and higher pressure gradients are encountered further upstream on the blade tip. Peak tip leakage vortex losses shift down in pitch, away from the trailing edge as Reynolds number is incrementally increased in Figure 4.11(b) to substantiate the claim. For the tip losses to shift away from the trailing edge, the logical conclusion is that the vortex is departing the blade tip at a slightly earlier fraction of the blade axial chord.

In Figure 4.11(c), a peak near a pitchwise location of 0.25 for the high Reynolds case corresponds to the tip vortex, perhaps also indicating the role a larger gap may play in permitting earlier tip leakage vortex departure from the tip. The low Reynolds number case does not return to near zero loss at the pitch extreme because a large piece of the passage vortex resides there.

Low Reynolds number total pressure loss for $g/c_x = 1.5\%$ derives from a large passage vortex near the spanwise limit in Figure 4.8. The general shape is preserved at $Re_2 = 5.0 \times 10^5$ but the presence of the tip vortex creates a new peak at 0.975S. Contribution from the passage vortex centered at 0.775S is much reduced in the case of a nearby tip leakage vortex. Even at the small gap size, the dominance of the tip leakage vortex at an applicable Reynolds number is undeniable.

Figure 4.12(b) demonstrates a tip vortex free wake at the two lowest Reynolds
Fig. 4.11: Spanwise Mass-averaged Total Pressure Loss Across Reynolds Number
(a) $g/c_x = 1.5\%$

(b) $g/c_x = 4\%$

(c) $g/c_x = 5\%$

Fig. 4.12: Pitchwise Mass-averaged Total Pressure Loss Across Reynolds Number
numbers, and then an introduction of a progressively strong tip vortex and the subsequent spanwise movement of the passage vortex. At Reynolds numbers of \(1.0 \times 10^5\) and \(2.0 \times 10^5\) the tip leakage vortex is at least partially present but too weak to contribute to slope of the spanwise distribution. As expected, the tip vortex is strongest and creates loss coefficients of roughly 0.3 at the highest Reynolds number conditions.

Contribution of the passage vortex is strongest at the low Reynolds numbers in the absence of strong tip leakage flow. But a substantially stronger leakage vortex at \(Re_2 = 3.0 \times 10^5\) does little to impact the passage vortex loss or the loss due to the interaction between them. This trend is not expected and highlights a key limitation to quantifying the loss from drastically different wake structures shown in Figure 4.9. Only with complete wake survey over a full pitch and span can the effort to quantify loss coefficients truly be valid, and even then one must ensure the same structures in all of the wakes. As the effect of Reynolds number is to reorder the locations of the important passage and tip leakage vortices, it should be stated that not all loss cases can be compared to one another. Nevertheless, efforts proceed to quantify the results to flush out qualitative trends in the flow physics.

Figure 4.12(c) provides an excellent example of this limitation, as the trends in the spanwise loss distribution are just about opposite. The Reynolds number extremes of the 4% cases are similar. Increasing the Reynolds number strengthens the tip leakage vortex and the subsequent loss, seen as a peak at \(z/S = 0.95\). Because the tip leakage vortex is effectively discarded at the low Reynolds number but very present in the window at the high Reynolds number, the figure demonstrates the drastic
Fig. 4.13: Overall Mass-averaged Pressure Loss vs. Exit Reynolds Number, $Re_3$

effect in total pressure loss with a strong tip leakage vortex and also how much loss ($C_{P_{l,PITCH}} > 0.35$) can be generated by the passage vortex without the tip leakage vortex to hold it in check.

Overall mass-averaged total pressure coefficients are calculated via Equation 3.10 and are presented in Figure 4.13. The movement of different wake structures into and out of the window of interrogation causes inconsistent total pressure loss averaging through the range of Reynolds numbers.

For example, the case of $Re_2 = 3.0 \times 10^5$ or $Re_3 = 4.5 \times 10^5$ contains a large low-loss region between the counter-rotating passage and tip leakage vortices. A similar
region is not captured in other wake surveys and contributes to a high loss coefficient seen as a spike in Figure 4.13. In fact, there is so much of a change across Reynolds number that only the wake structures between $Re_2 = 5.0 \times 10^5$ and $Re_2 = 6.0 \times 10^5$ ($Re_3 = 7.5 \times 10^5$ and $Re_3 = 9.5 \times 10^5$) at gap size of $g/c_x = 4\%$ are similar enough to justify a comparison of the effect of Reynolds number on the overall loss. This effect is seen as quite mild compared to some fluctuations in the figure.
CHAPTER 5

RESULTS III: EFFECT OF TIP GEOMETRY

Documenting the squealer tip was first considered as a flow control strategy, where the gap size is judged in all cases to be the distance from the flat tip to the endwall, \( g \). However, squealer tips are fabricated by milling out unnecessary blade tip material to leave a short ridge, which extends beyond the recessed flat tip surface. For this reason, the gap height is typically defined from the tip of the squealer to the tip endwall, deemed in this thesis a “true” gap, \( \tau \), and portrayed in Figure ref: gapdef.

Seeking to match some performance characteristics of the squealer tip with a future plasma actuator, the effect of the tip geometry modification was documented first at an equivalent effective gap \( g/c_e \). In addition to its aerodynamic performance, popularity of the squealer tip derives from its ability to limit gap discharge by extending the blade while posing less risk of a catastrophic blade rub than a full tip extension. The definition of the squealer tip as a blade extension is consistent with such a performance characteristic. Further, from the perspective of a flow control application, a flat tip was considered the “actuation off” case and a squealer tip was
approached as the “actuation on” strategy. Interest lies in documenting the passive actuation effect of the squealer tip over a flat blade at similar effective gap sizes. Consistent with these definitions, the squealer tip with a height of $0.025c_x$ was fixed to flat tip blades at $g/c_x = 4\%$ and $g/c_x = 5\%$. Analysis first focused on comparison of tip configuration with equivalent effective gap sizes. A second analysis follows and compares the 4\% gap size with squealer tip, $g/c_x = 4\%$, $\tau/c_x = 1.5\%$, with the flat tip at 1.5\% gap, $g/c_x = \tau/c_x = 1.5\%$.

5.1 Blade Surface Static Pressure Distributions

Effective gap comparisons of $g/c_x = 4\%$ and $g/c_x = 5\%$ are presented in Figures 5.2 and 5.3, respectively. The results are surprising. At both Reynolds numbers and gap sizes, a systematic increase in pressure side static pressure is observed over much
Fig. 5.2: Blade Surface Static Pressure Distribution for Flat and Squealer Tips, $g/c_x = 0.04$

of the pressure side surface. A static pressure rise is equated to a local velocity drop over the static ports on the pressure side rim, which indicates a drop in the mass flow over the edge and into the gap zone.

On the suction surface, some results are also consistent across every case. The characteristic leading edge unloading is reduced with the squealer tip when compared to the flat tip case. Trailing edge loading is consistent with a flat or squealer tip for the most part, with the exception of $Re_2 = 5.0 \times 10^5$ at a gap size of $g/c_x = 4\%$. Figure 5.2(b) demonstrates slightly reduced loading at the trailing edge. Due to increased pressure side static pressures, the pressure gradient established between pressure and suction surfaces over the blade tip is strengthened with the addition of the squealer tip, in a systematic fashion.
(a) $Re_2 = 1.0 \times 10^5$, $g/c_x = 5\%$

(b) $Re_2 = 5.0 \times 10^5$, $g/c_x = 5\%$

Fig. 5.3: Blade Pressure Distribution for Flat and Squealer Tips, $g/c_x = 0.04$

(a) $Re_2 = 1.0 \times 10^5$

(b) $Re_2 = 5.0 \times 10^5$

Fig. 5.4: Blade Pressure Distribution for Squealer Tip Across Gap Height
Fig. 5.5: Blade Pressure Distribution for Squealer Tip Across Reynolds Number

Effect of gap height for the squealer tip is equivalent for the flat tip. As only a one percent change in the gap height was considered with the squealer tip, the result is not surprising. Minimal changes are seen only in regions of large transitions in static pressure coefficient.

The effect of Reynolds number reflected in blade loading for a squealer-tipped blade is also the same as for a flat tip. Pressure side static pressure coefficients remain consistent while suction surface values reflect the increase in blade loading with Reynolds number by becoming more negative. Suction side pressure coefficient values are the same near the trailing edge, regardless of Reynolds number, as seen in Figure 4.1(c).

To quantitatively summarize findings, Figure 5.6 shows midspan loading in the case of a squealer tip to follow the same trends as with a flat tip. As with the flat
Fig. 5.6: Midspan Blade Loading Across Gap Size for Flat and Squealer Tip
Fig. 5.7: Tip Blade Loading Across Gap Size for Flat and Squealer Tips
tip cases, a drop in blade loading corresponds to a change in gap size from 4% to 5%. The loss of blade loading across gap for the squealer tip is proportional to loss with the flat tip. Midspan cases with squealer tip are substantially more loaded than their flat tip counterparts. At $Re_2 = 5.0 \times 10^5$ and for both gap sizes, the drop in midspan loading due to flow through the gap once the squealer is removed is 15%. Similarly, for $Re_2 = 5.0 \times 10^5$ and $g/c_x = 4\%$, the loss of loading is 14% once the squealer tip is removed. The drop is 11% for $Re_2 = 5.0 \times 10^5$ and $g/c_x = 5\%$.

Blade loading trends at the 0.995$S$ tip static pressure ports are not consistent between the two different tip configurations. While flat tip loading drops almost linearly as the gap size grows for both Reynolds numbers, once the squealer tip is added, the loading shows a slight drop for $Re_2 = 5.0 \times 10^5$ and a minor rise for $Re_2 = 1.0 \times 10^5$ in Figure 5.7. That the tip loading for a squealer configuration undergoes an increase with tip clearance at the lower Reynolds number is unexpected, but may indicate that the squealer is more effective in cutting the low speed mass flow into a slightly larger gap.

As the Reynolds number is stepped from low to high, the midspan distributions show an increase in the blade loading by approximately 10% over the Reynolds number range for every case. Intuitively, the smallest gap size represents the case of least over-the-tip flow and greatest midspan blade loading. As gap increases, blade loading at a given Reynolds number decreases. Cases of 1.5% tip gap and 4% gap with a squealer tip ($\tau/c_x = 1.5\%$) do not collapse onto one another, because blade loading remains reduced in the 4% effective gap case.
Fig. 5.8: Midspan Blade Loading Across Reynolds Number for Flat and Squealer Tip
Fig. 5.9: Tip Blade Loading Across Reynolds Number for Flat and Squealer Tip
Tip loading dependence on Reynolds number, shown in Figure 5.9, distinguishes flat and squealer tip cases by the slope of the increase from low to high speed. All three flat tip cases share a similar slope while the squealer tip data show lower tip loading at the low Reynolds number and a greater loading at the high Reynolds number case, resulting in a steeper slope. Blade tip loading for the squealer configuration cases is very similar at both Reynolds numbers, hinting that the squealer tip blade loading is less sensitive to a change in gap size than the corresponding flat tip. A 1% change in gap size for the flat tip yields more than $0.05c_{p,\text{blade}}$ difference, greater than that seen for a similar squealer tip gap change. This is especially true at the high speed squealer tip configuration where the blade loading appears to be the least sensitive to gap change.

5.2 Endwall Static Pressure Distributions

A drastic difference between the endwall static pressure behavior is observed in all cases in which the squealer tip has been attached. Comparing across gap size for an upstream Reynolds number of $1.0 \times 10^{5}$, a generally smooth acceleration around the blade and through the passage is seen for all cases. For the flat tip case, the smooth acceleration was found everywhere except in a confined area directly over the blade shortly aft of mid-chord. The region of high velocity rolling from the pressure to suction side over the blade tip is thought to indicate a strong leakage jet forcing the tip vortex to roll off of the blade tip. Top rows of plots for Figures 5.10 and 5.11 correspond to these flat tip cases at $g/c_x = 4\%$ and 5\%. 109
Effect of increasing the gap size on the endwall pressures was to decrease the velocity (e.g. static pressure drop) and the size of the departure zone. The $g/c_x = 4\%$ case of Figure 5.10 demonstrated a departure zone that began at a similar location but extended farther toward the trailing edge. The addition of the squealer tip, seen in the bottom row of the same two figures, completely removes the spike in static pressure loss to leave only the through-passage acceleration due to the nozzle effect.

The major alteration to the endwall pressure distributions is also seen in the high Reynolds number condition. Due to a larger departure zone at a higher Reynolds number, the squealer effect in the gap is clearly shown in Figure 5.11. With a squealer tip configuration, the high velocity leakage jet to force and/or indicate the departure from the tip gap is completely removed and a smooth acceleration through the passage.
is once again revealed. Undoubtedly, the squealer tip reduces losses in the gap region due to a postponed and less violent separation, as proposed in the literature. Viewing the effect of the squealer tip as shown by the endwall static pressure distributions, one would expect a departure of the tip leakage vortex fluid far downstream, perhaps not until the trailing edge. Another mechanism for tip leakage departure would be a more gentle spilling of gap fluid over the suction-side squealer tip in contrast to a strong leakage jet.

For 4% gap size, the Reynolds number parameter, which is seen to carry large repercussions at the endwall for the flat tip, is largely ineffective in altering the endwall pressure profiles for the squealer tip. Even as the blade becomes much more
loaded over the squealer tip, an observation which correlates with an increase in the departure activity for the flat tip, there is a lack of any noticeable difference for the squealer configuration. Existence of an enhanced pressure gradient but absence of the departure region means that the squealer tip stands as a fence to over-the-tip flow, through the entire blade chord. It is quite possible then that the tip leakage flow remains isolated over the blade throughout the entire tip gap and its losses are not fully quantified until far downstream. Losses in the gap should be reduced with the violent departure zone removed.

Again at 5% gap in Figure 5.13, where Reynolds number played an important role previously for the flat tip, the effect is non-existent for the squealer configuration as told by endwall static pressure ports. The distributions in the bottom row corre-
Fig. 5.13: Effect of Reynolds Number for Flat and Squealer Tip, $g/c_x = 5\%$

sponding to the squealer geometry almost replicate the flat tip distributions with the departure zone removed.

The previous observations justify the heat transfer challenges in the application of squealer tips. Already positioned in an extreme thermal environment, the squealer tip physically blocks high-velocity leakage of hot burner exit fluid, which results in an undesirable recirculation and a more even distribution of hot air over much of the blade tip surface. Undoubtedly, this behavior is a concern for any type of actuation.
5.3 Wake Surveys

5.3.1 Vorticity

Change of the tip geometry to include a squealer tip brings a more negative vorticity to the tip leakage vortex. As the tip leakage vortex grows stronger and moves away from the trailing edge in pitch, the passage vortex does not always decrease in size but shows considerable drop in vorticity. Such an observation cannot be made for the flat tip case due to the distance of the tip leakage flow from the endwall, which pushes the passage vortex down the span and mostly out of the window. With the squealer tip, the tip leakage vortex hugs the endwall. As a result, the passage vortex is more completely shown in the interrogation zone.

At the largest gap setting, vorticity of the passage vortex is very strong at low
speed, but the previous trend is consistent here and applies as the Reynolds number is increased. At $Re_2 = 5.0 \times 10^5$, the tip leakage vorticity is strong and the vortex is pushed to the endwall. As a result, a more complete view of the full passage vortex is provided and indicates a much reduced vorticity. Again, as hypothesized before, the passage and tip leakage vortices seem to share an inverse interaction. Because the squealer tip positions the tip leakage vortex closer to the endwall, a more complete view of the vortex pair relationship is available through the normalized vorticity plots of Figure 5.15. Probable causes for closer proximity of the tip leakage vortex to the endwall lie in previous observations that the vortex departs from the tip surface at a more downstream position and remains isolated through more of the passage.
Fig. 5.16: Total Pressure Loss with Reynolds Number and Tip Geometry, $g/c_x = 0.04$

5.3.2 Total Pressure Loss

Tip leakage and passage vortices start far apart in Figure 5.16, move closest together at $Re_2 = 3.0 \times 10^5$, and then move apart again at higher Reynolds numbers, similar to the trend in the flat tip case. Immediately noticed at low speeds is a much more coherent tip leakage vortex, possibly because it has traveled farther over the blade tip. Thus, the squealer tip leakage vortex has experienced less mixing than leakage flow over a flat tip, which departs the tip gap near 70% axial chord and undergoes intensified mixing through the remainder of the passage.

The tip leakage vortex is more distinct and smaller in size with the squealer configuration for $2.0 \times 10^5 < Re_2 < 5.0 \times 10^5$, which can be explained by less mass flow into the tip gap feeding the tip leakage vortex and also a stalled departure from
Fig. 5.17: Total Pressure Loss with Reynolds Number and Tip Geometry, $g/c_x = 0.05$

the tip gap and thus a less mixed leakage flow. Undoubtedly, the effect of less mixing of the leakage vortex with the freestream and also the passage vortex with a squealer tip configuration is more extreme closer to the trailing edge of the blades. Because many studies in the literature consist of surveys taken at $0.5c_x$ or less downstream of the cascade, the reduced tip leakage mixing by the position of survey may have overindicated the performance of the squealer tip. Even at a full chord downstream, a tighter, less-mixed vortex is observed, and similar loss measurements taken upstream would likely show more improved squealer tip performance.

At a slightly larger gap, similar results are expected and observed. Losses are higher due to greater flow through the gap, which as expected strengthens both loss
mechanisms. Enhanced loss is particularly evident in the core of the tip leakage vortex for an upstream Reynolds number of $Re_2 = 5.0 \times 10^5$. The tip vortex is immediately more organized at the low speed case of Figure 5.17 and further from the endwall. At high speed, the tip vortex hugs the endwall as before in Figure 5.16 and brings more of the passage vortex into the window.

Clear qualitative differences are observed in vorticity and total pressure loss, but the results must be quantified to draw definitive conclusions from the data. For increasing Reynolds number, Figure 5.18 shows the pitchwise distribution of loss in the case of a squealer tip. At the lowest Reynolds number, the tip leakage vortex is seen at $y/P = 0.0$, but for greater speeds, a spike attributed to the tip leakage is not seen due to its movement to a similar pitchwise direction as the passage vortex contribution. As loss mechanisms become much more well-defined in the wake with an increase in Reynolds number, maximum loss values at $y/P = 0.25$ can be seen. Although plots for $Re_2 = 2.0 \times 10^5$ and $Re_2 = 3.0 \times 10^5$ look qualitatively similar, spanwise mass-averaged loss coefficients are very different. Then a transition from $Re_2 = 3.0 \times 10^5$ to $Re_2 = 5.0 \times 10^5$ shows very little difference in the pitchwise distribution. Thus, the range $2.0 \times 10^5 < Re_2 < 3.0 \times 10^5$ seems to be a critical stage of development of the total pressure loss. After some initial pitchwise movement of the loss mechanisms while they mature, positions of the tip leakage and passage vortices settle at $y/P = 0.25$.

In Figure 5.19(a), losses for $Re_2 = 1.0 \times 10^5$ demonstrate reduced squealer tip values at all pitchwise locations except very near the trailing edge where the tip vortex
Fig. 5.18: Pitchwise Distribution of Total Pressure Loss at 4% Gap with Squealer Tip has become more dominant. Results for the high Reynolds number are provided in Figure 5.19(b). A peak in the squealer values denotes the heightened organization of the wake with passage and tip leakage vortices in the same pitchwise position, which causes a higher loss than the flat tip at $y/P = 0.25$. However, the peak is very distinct and loss values at extremes in the pitchwise direction are significantly reduced from those of the flat tip due to less mixing and interaction of these two structures despite their close proximity.

Vortices also settle at a similar pitchwise location for the squealer tip with $g/c_x = 5\%$. Addition of the squealer tip causes 25\% greater loss coefficient at the peak region, due to the addition of a weak passage vortex but a very strong tip leakage loss zone.
that was not seen before. At the 4% gap, the core of the tip leakage also produced more loss and more of a passage vortex was captured in the window, but the tight organization of the structures resulted in less pitchwise loss. At 5% gap, the effect of the squealer tip is less pronounced. The tip leakage and passage vortices are still large and now the peak tip leakage loss is stronger than for the flat tip.

The effect of increasing gap size for the flat tip at low Reynolds number was to spread out the loss over more of the pitchwise survey zone (Figure 5.20(a)). Addition of a squealer tip reinforces the tip leakage vortex, which is represented by a higher loss at the trailing edge. Pitchwise losses are reduced for the most part when adding a squealer at 4% but the same cannot be said for a gap size of 5%. Figure 5.20(b) confirms the finding at the high Reynolds condition. Here the loss is greatest in the

Fig. 5.19: Squealer and Flat Tip Spanwise Mass-averaged Loss Coefficients at 4% and 5% Gap Height
range $0.0 < y/P < 0.35$ for the 5% gap case with squealer tip. Meanwhile the 4% gap with squealer tip demonstrates less loss than even the 1.5% case from the trailing edge until the local maximum at $y/P = 0.25$, after which it follows the same trend at greater values of total pressure loss.

Shifting the focus to the spanwise distributions of pitchwise mass-averaged loss allows one to clearly distinguish between tip vortex and passage vortex contributions. Spanwise distributions highlighting the effect of Reynolds number with the squealer tip in Figure 5.21 support the previous hypothesis of a balance between loss generated by the tip leakage and passage vortices.

As upstream Reynolds number is increased, loss due to the passage vortex near $z/S = 0.725$ sequentially decreases, although most notably so in the transition from
Fig. 5.21: Spanwise Distribution of Total Pressure Loss at 4% Gap with Squealer Tip

$Re_2 = 1.0 \times 10^5$ to $Re_2 = 2.0 \times 10^5$. A minimum value of loss is seen near 90% span, and then the tip leakage loss contribution begins. In contrast to the passage vortex trend, tip leakage loss increases sequentially with Reynolds number. The two opposing trends point to a give-and-take interaction between competing passage and tip leakage vortices.

Figure 5.22 portrays spanwise loss distributions for both Reynolds numbers, 4% effective gap size, and flat and squealer tip configurations. Loss derived from the passage vortex for $Re_2 = 1.0 \times 10^5$ drops off faster with the squealer. At $Re_2 = 5.0 \times 10^5$, the squealer reduces passage vortex loss and shifts many of the loss coefficients leading up to and including an equal tip leakage vortex loss coefficient. The spanwise
shift is toward the endwall by 0.02S.

From Figure 5.22(b), the shift of tip leakage loss toward the endwall is consistent at a gap size of 5% axial chord. There is a reduction of passage vortex loss with the squealer for the high speed case. Low speed loss distributions demonstrate very similar behavior with and without squealer tip. The squealer tip provides an advantageous loss behavior above 92% of span compared to the flat tip.

All the data sets at the low Reynolds number collapse to the same trend in Figure 5.23. The passage vortex is the dominating mechanism while the tip leakage vortex is positioned near the endwall. As the flat tip gap grows, so does the loss, as expected due to the increased mass flow into the gap and subsequent loss mechanisms. Here it is quite clear that the squealer tip is less effective at the larger gap of 5%, an
Fig. 5.23: Squealer and Flat Tip Pitchwise Mass-averaged Loss Coefficients at Low and High Reynolds Number

interesting result noting the success of the squealer tip for 4% gap.

At the high speed condition, the systematic shift of the tip leakage vortex toward the endwall with a squealer tip is demonstrated for both gap sizes. At the 4% gap, the shift is not accompanied by a increase in the peak loss coefficient, while this is the case for 5% gap. For a flat tip, a larger loss contribution from passage vortex is correlated with a larger gap size.

Limitations of comparing overall mass-averaged loss coefficients across Reynolds number have already been mentioned. When the flow structures in the window are consistent (e.g. not across most Reynolds numbers) their corresponding loss coefficients can be compared. In this manner, comparisons of 4% flat and squealer tip cases are most justified. The squealer tip demonstrates a substantial improvement
at every Reynolds number. Also, the lowest and highest Reynolds number cases may be compared between 1.5% gap and 4% gap, where the squealer tip is proven more effective at $Re_2 = 1.0 \times 10^5$ but less so at the high Reynolds number. Reynolds numbers of $Re_2 = 5.0 \times 10^5$, $Re_2 = 6.0 \times 10^5$ share qualitatively similar wake structures and point to a much more minimal increase.

Despite difficulties in quantifying and comparing results across Reynolds number, comparing the wake surveys across gap size, where there was minimal change, is justified. Results of the loss studies are effectively summarized with Figure 5.25. For all cases, increasing the gap size increases the loss nonlinearly. For both Reynolds
Fig. 5.25: Overall Mass-averaged Total Pressure Loss Across Gap Size

numbers, the squealer is effective at the 4% gap but largely ineffective at 5%. At 4% gap, even though peaks in the loss for the squealer tip exceed those for the flat tip, the absence of the extensive low loss regions of the flat tip case bring about a lower overall loss for the squealer configuration. At 5%, the structures are less extinct and mixing between the major vortex actors creates large, undesirable low-loss regions, which eventually ruin the case for the squealer at 5%. Figure 5.26 offers an explanation for the result.

Noting a consistent change in squealer tip performance in the transition from $g/c_x = 4\%$ to $g/c_x = 5\%$, a change in the flowfield was suspected and the criteria
defining a “thick” or “thin” blade, \( t/g = 4 \), was revisited. The distinction proposed by Denton [6] for a flat tip is applied to the squealer tip configuration in Figure 5.26. In the case of a thick blade, flow separates on the pressure side edge, forms a separation bubble and reattaches, and then is restricted from flowing across the blade tip by the presence of a squealer tip. Recirculating flow ensues directly above the blade tip and effectively extends the blade. Flow forms a *vena contracta* above this squealer tip to further constrict the mass flow over the suction side.

A thin blade shown in Figure 5.26(b) features a complete separation of the flow from the pressure side corner and the forming of a *vena contracta* again above the squealer tip. This time however, the presence of the squealer tip is not felt by tip gap flow and any available improvement is largely reduced. The distinction between thick
and thin blades in this manner is found at a boundary of $t/g = 3.5$, which is slightly different than the value of 4.0 proposed by Denton [6]. To find a more accurate boundary between the two flow types, more gap sizes will have to be explored. Table 5.1 lists the cases of the data set and the resulting flow model.

5.4 Comparison Across True Gap Size, $\tau/c_x = 1.5\%$

Consistent with the manufacturing technique behind most squealer tips, previous work has frequently compared gap sizes defined from the end of the turbine blade or squealer tip to the endwall. In contrast, this thesis has investigated the effect of the squealer tip as a flow control strategy and has evaluated gap size, even in the case of the squealer tip, as the distance from the flat blade tip to the endwall. Results
Fig. 5.27: Blade Static Pressure Coefficients for at Low and High Reynolds Numbers,

\[ \tau/c_x = 0.015 \]

have justified the terminology at 5% tip clearance, where over-the-tip flow completely separates from the pressure side edge, forms a high *vena contracta*, and misses the squealer tip entirely. Wake surveys and endwall static pressures indicate that the squealer tip at 5% gap does little to limit losses anywhere but just over the blade tip, and the tip leakage performs as a blade at 5% effective gap.

The case of \( g/c_x = 4\% \) is a different story. Here the blade is deemed *thick*, where the thickness of the blade is sufficient to allow for a reattachment to the blade tip surface after the initial pressure side separation. This reattached flow directly encounters the presence of the squealer tip and recirculates. Recirculating flow effectively fills the milled out blade tip surface, creating the behavior similar to the blade at \( g/c_x = 1.5\% \). The squealer tip performs even better than its true gap counterpart
because it triggers a more constricting vena contracta between the squealer tip and the endwall. However, Figure 5.25 denotes a squealer tip advantage over the flat tip blade at same true gap setting for only the low Reynolds number. The issue of why the squealer tip does not pose the same advantage at $Re_2 = 5.0 \times 10^5$ is examined next.

Although major differences in tip blade static pressures were observed when gap size was varied with a flat tip configuration in Figure 5.7 on page 105, pressure distributions for the squealer tip do not demonstrate the same dependence on gap size. Addition of a suction-side squealer to the flat tip forced a considerable change for a 4% gap, enough to make the 4% gap with a squealer tip and a 1.5% flat tip blade share only minimal differences in Figure 5.27. Pressure side coefficients are comparable over the entire pressure side surface. Blade unloading and pressure gradient over the tip at $0.8c_x$ is greater for the 4% gap case at the low Reynolds number. At the high Reynolds number, the blade loading is still enhanced, but beginning at $0.5c_x$, 4% suction side pressure coefficients become more negative and remain so until $0.9c_x$. Cross-referencing Figure 5.27(a) for the low speed results, more negative values for the squealer case were not seen until $0.75c_x$ and lasted until the trailing edge. It is possible that gap flow is leaking from the tip gap sooner and mixing more at the higher Reynolds number.

As before, endwall static pressure coefficients show a complete removal of the departure zone at both Reynolds numbers once the squealer tip is added. The result is more conclusive now that both blades can be considered an equal distance from
the endwall. In both Reynolds number conditions, the squealer has stopped the rapid leakage jet induced departure over the tip surface. It is probable that leakage flow continues to leak over the squealer tip and depart the blade before the trailing edge, but in a less violent and sudden manner. This trickle of leakage flow is less noticeable when the blade loading is low but becomes more important at the high Reynolds number. Now, although the squealer tip has removed any leakage jet and strong departure of the leakage within a confined chordwise range, the trickle of leakage flow over the squealer tip constantly feeds the tip leakage mechanism over more of the axial chord distance. The effect of this leakage trickle grows with Reynolds number, and so less loading in the low Reynolds number case allows for a more effective squealer tip performance.
The squealer tip transforms an almost imperceptible tip leakage vortex into a region of strong negative vorticity, and similarly increases the vorticity of the passage vortex at $Re_2 = 1.0 \times 10^5$. At the high Reynolds number, vorticity of the tip vortex is similar while the passage vortex is smaller and more removed from the tip leakage flow.

Loss shown in Figure 5.30 indicates a weaker and thus more mixed out vortex pair for the flat tip. The greater mixing is indicated by an expansive area of low loss in light blue. With the squealer tip, the vortices are more distinct and separated and the loss between them is much lower, resulting in less overall loss. At $Re_2 = 5.0 \times 10^5$, the tip leakage vortex is considerably stronger, the passage vortex is considerably weaker and smaller, and the mixing in between the two is similar once the squealer has been
Fig. 5.30: Total Pressure Loss, $\tau/c_x = 0.015$
added to a flat tip. Although high loss in the tip vortex core is quite noticeable, the
real culprit to the reduced advantage of the squealer tip at high Reynolds number lies
with the low loss regions, which permeate the window. The increased mixing may be
due to the constant leakage trickle over a large portion of the suction-side squealer
tip.

Trends in pitchwise and spanwise distributions of total pressure loss are consistent
between the $g/c_x = 1.5\%$ and $g/c_x = 4\%$ squealer cases. In Figure 5.31(a), the
presence of a less mixed, more organized, and stronger tip leakage flow is seen at the
trailing edge, $y/P = 0.0$. The contribution from a stronger but less complete passage
vortex is reduced, and the mixing loss due to the interaction between the two is less
at all other pitchwise locations. For the high speed condition in Figure 5.31(b), the
loss peak at $y/P = 0.25$, which now corresponds to contributions from the pitchwise-
aligned passage and tip leakage vortices, is virtually the same despite a much stronger tip leakage loss in the core. The distribution serves as further evidence to the conclusion of an inverse relationship between the passage and tip leakage vortices. Addition of the squealer tip brings a stronger tip leakage vortex but also a much weaker passage vortex, and the resulting peak in the pitchwise loss contribution is similar. However, the difference lies in the regions in between the structures. For all other pitchwise locations, this mixing effect is heightened in the squealer tip over the flat tip case.

Spanwise distributions are convenient for separating tip leakage and passage vortex combinations. In Figure 5.32(a), the passage vortex generates a greater peak in the loss at $z/S = 0.725$. Tip leakage loss is not a strong contributor to be seen in loss peak at either low speed case but undoubtedly contributes to the heightened mixing loss at spanwise locations outside of the passage vortex. Spanwise distributing the
loss for the high speed condition in Figure 5.32(b) helps to display the dominance of the peak and mixing associate with the tip leakage mechanism in the squealer tip case.

Here the squealer tip is again effective in separating the two vortices and thus reducing the loss attributed to their interaction as seen in lower loss coefficients in the range $0.725 < \frac{z}{S} < 0.875$. However, the tip leakage vortex is so large and so dominant that it generates high loss not only in its core but with the summed contributions of mixing of its outer layer with neighboring streamwise fluid as well.
Although complications of the tip leakage and passage vortices do not pose an entirely new problem, data sets in the literature to date have failed to fully explore and compare the effects of Reynolds number and gap size for flat and suction-side squealer tip geometries. Variations in gap size and Reynolds number have been undergone for flat tips as separate undertakings but rarely on the same project. Further, most cascade work to date has dealt with low Reynolds numbers, which is shown to be a limiting condition. For these reasons and in conjunction with the objectives of the research, means of instrumentation presented in this thesis provided unique insights to the flow physics surrounding turbine blade tips.

A proven indicator of the mass flow through the gap, blade loading was examined first for the flat tip and confirmed to drop in linear fashion at tip and midspan locations as the gap height, $g$, was increased. Interestingly, the loss in blade loading was more severe at midspan. Blade loading increases linearly with Reynolds number, but more drastically at the tip location. Tip loading at the largest gap setting was
most sensitive to Reynolds number variation.

With the squealer tip, both midspan and tip blade loading are increased. Adding a squealer increases midspan loading by 10% for all cases, and the effect of Reynolds number on midspan blade loading for a squealer tip is the same as for a flat tip. Loading is less sensitive to a gap change at the tip than at midspan.

Endwall static pressure distributions indicate a smooth fluid acceleration through the passage, but also uncover a thin path of streamwise acceleration above the mean camber line of the blade. The local acceleration line could be an indication of the reattachment point of the separation bubble or the core of the tip leakage vortex, which is known to consist primarily of strong streamwise and weak secondary velocity both above the blade and far downstream in the wake. A region of local acceleration from the pressure to the suction side indicates a region where the blade tip pressure gradient drives a jet of fluid including the tip vortex off of the blade tip surface, over the suction side, and into the passage. Although independent of gap size, the size of the mixing zone is strongly dependent on Reynolds number and grows to occupy the entire last half of the chord for an exit Reynolds number of $Re_3 = 7.5 \times 10^5$.

Once the squealer tip has been added, endwall pressure distributions indicate a complete elimination of the departure zone, leaving only a smooth acceleration through the passage. It is hypothesized that the squealer tip stands as a fence to the leakage jet and permits the vortex to either slowly trickle over the suction edge or to remain over the tip surface until the trailing edge. The line of local acceleration above the camber line of the blade has also been removed for all cases with the squealer tip.
geometry as a further indication of a completely altered flowfield.

Surveys in the wake of a flat-tipped blade portray a more organized tip leakage vortex with stronger vorticity at a smaller gap. Interaction between the tip leakage and passage vortices contributes heavily to the entropy production and increases with gap height, and as a result the total pressure loss increases in a nonlinear fashion with the gap size. At low gap sizes, loss regions are found to be concentrated in regions of high vorticity, while larger gap sizes promote large low-loss regions in between the vortices. Also, the tip leakage vortex moves closer to the trailing edge and the tip endwall as the gap shrinks.

Variation in Reynolds number highlights an inverse relationship between the vorticity, size, and loss of the passage and tip leakage vortices in the case of a squealer tip. The tip leakage vortex grows stronger while the passage vortex weakens in terms of vorticity and loss as the Reynolds number is increased. Reynolds number reorders the wake. The tip leakage vortex and passage vortex systematically drop in pitch and eventually align at a pithwise location of $y/P = 0.25$ for the high Reynolds number condition. Due to greater mass flow through the gap, the overall loss grows substantially with Reynolds number as well.

Qualitatively, adding a squealer tip yields changes similar to those seen when reducing gap size. Structures become more clearly defined with strong vorticity and local loss regions and their interaction is downplayed. With a squealer tip, the loss is derived predominantly inside the two vortices. The squealer tip is notably more effective at $g/c_x = 0.04$ and fails to yield any improvement at $g/c_x = 0.05$, where
a peak in tip leakage loss overrides gains from the organization of the wake. It is hypothesized that a different flow model corresponds to this larger gap scenario, where leakage flow separates from the pressure side edge and bypasses the squealer tip entirely. Revisiting Denton’s [6] criteria to divide a thick and thin blade flow condition, a reduced value of $t/\tau = 3.5$ was found to match results for this data set. Distinctive observations from flat and squealer tip comparisons across effective and true gap height definitions justify the two-pronged evaluation of the passive squealer tip flow control performance.

Future work calls for investigations across even greater Reynolds numbers and wake surveys over at least half a pitch and half a span in order to fully and consistently appreciate the tip side passage vortex. An alternate approach is to isolate the loss attributed to the tip leakage vortex by a dividing contour line of zero vorticity between the tip leakage and passage flow structures in order to consistently evaluate performance for different gap sizes and Reynolds numbers. Future gap sizes of interest are those in the thick blade category, since they are more common and pose greater potential for flow control in turbines. However, variation in tip clearance over a typical mission profile can lead to thin blade conditions, and a method of flow control for that model is still useful and interesting. An investigation into an optimized height of a suction-side squealer tip is not seen in the literature but appears to be warranted as well, particularly after noting inconsistent actuation of the passive squealer tip as the gap changes. The experimental or computational optimization will undoubtedly enlist effective gap height as a design variable, and thus will point to an adaptable
flow control strategy for consistent actuation over the mission profile. Therefore, although progress remains to be realized by passive flow control mechanisms such as the squealer tip, the problem is also ready for the implementation of active control by plasma actuators.
MATLAB code to load and process wake survey pressure data acquired with a Five-hole Pitot Probe.

GRID5.M Runs measured data through experimental calibration to give pitch and yaw angles, velocity magnitude and total and dynamic pressure corresponding to a non-streamwise velocity vector.

FIGURE 1 Contour plot of Total Pressure Loss over interrogation window with superimposed secondary velocity vectors and the zero contour of the second eigenvalue of the velocity gradient tensor.

FIGURE 2 Contour plot of Nondimensional Streamwise Vorticity in the interrogation window
FIGURE 3 Contour plot of the second largest eigenvalue of the velocity gradient tensor.

CpPITCH Arithmetically & pitchwise mass-averaged total pressure loss coefficient.
CpSPAN Arithmetically & spanwise mass-averaged total pressure loss coefficient.
Cpt2 Arithmetically & overall mass-averaged total pressure loss coefficient.

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
clc
clear all
close all
format short g

To run:
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
LOAD FILE:
Re = 500; % Reynolds number
sp = 4; % which subplot to use
gc = 5; % gap to chord ratio (%)
s = 1; % 1 = squealer, 0 = no squealer

PLOT FORMATTING
on = 1; % print colorbar? 1 = yes
xon = 0; % print axes labels? 1 = yes
nr = 2; % number of rows
np = 2; % number of subplots
titleRE = 1; % Re title = 1, GAP title = 0
minom = -0.29; % minimum vorticity for colorbar
maxom = 0.30; % max. vorticity
mincp = -0.03; % minimum cp for colorbar
maxcp = 0.95; % maximum cp
nl2 = 0; % value of plotted lambda 2 contour
f1 = 1; % number of figure 1
f2 = 2; % number of figure 2
f3 = 3; % number of figure 3
ang = 1; % 1 = angularity cp, 0 = discount angularity
% 1. Load file:

if s == 1
  if gc == 1.5
    disp('Error, did not run this case.
  break
  else
    B = load(['r',num2str(Re),',sg',num2str(gc),',wake.dat']);
  end
else
  if gc == 1.5
    B = load(['r',num2str(Re),',g',num2str(gc),',wake.dat']);
  else
    B = load(['r',num2str(Re),',g1.5wake.dat']);
  end
end

% 2. Initializations:

T = 23.5 + 273.15;  % K
p = 29.08*3386.388*(1-0.0037426);  % Pa
R = 287;  % J/(kg K)
rho = p/(R*T);  % kg/m^3

if Re == 500 & gc == 5 & s == 1
  zoffset = -0.15;
end

% 3. load data
Ptrue = B;
pos = Ptrue(:,[1:2]);  % (y,z) coordinates written to file

if pos(1,1) == 0  % loop to load the corresponding z
\% and \( y \) vectors depending on probe acquisition path.

\[
z = \text{load('ztestlocation.dat')} + \\text{zoffset}; \quad \text{\%inches}
\]

\[
y = \text{load('ytestlocation.dat')}; \quad \text{\%inches}
\]

else

\[
z = \text{load('ztestlocationb.dat')} + \\text{zoffset}; \quad \text{\%inches}
\]

\[
y = \text{load('ytestlocationb.dat')}; \quad \text{\%inches}
\]

end

\[
z = z./\text{span}; \quad \% \text{non-dimensional span coordinates}
\]

\[
y = y./\text{pitch}; \quad \% \text{non-dimensional pitch coordinates}
\]

\[
\begin{bmatrix} Z \, Y \end{bmatrix} = \text{meshgrid}(z,y); \quad \% \text{for contour plotting}
\]

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

\% 4. Convert pressure readings to units of pressure \([\text{Pa}]\):

\% Transducer span settings:

\% 0.5 --> all \textit{r500s}, \textit{r300g4wake.dat}

\% 4.0 --> all \textit{r100’s}, \textit{r200}

\% loop to ID file and convert Volts to Pa

\[
\text{if} \quad \text{Re} == 500 \| \text{Re} == 300 \| \text{Re} == 600
\]

\[
\text{Ptrue} = 3977.5*\text{Ptrue}(,\,[3:9]); \quad \% \text{Pa, trans. span} = 0.5
\]

\[
\text{if} \quad s==1
\]

\[
\text{sprintf('span 0.5, Re = \%d with squealer tip...\n',Re)}
\]

\[
\text{else}
\]

\[
\text{sprintf('span 0.5, Re = \%d...\n',Re)}
\]

\[
\text{end}
\]

\[
\text{end}
\]

\[
\text{if} \quad s == 1
\]

\[
\text{sprintf('span 4.0, set Re = \%d with squealer tip...\n',Re)}
\]

\[
\text{else}
\]

\[
\text{sprintf('span 4.0, set Re = \%d...\n',Re)}
\]

\[
\text{end}
\]

\[
\text{end}
\]

\[
\text{Pidyn} = \text{Ptrue}(,1); \quad \% \text{Upstream dynamic pressure}
\]

\[
\text{Ptrue} = \text{Ptrue}(,\,[2:7]); \quad \% 7 \text{ is downstream static}
\]

\[
\text{Pestat} = \text{Ptrue}(,6); \quad \% \text{ref’d against upstream stat.}
\]

\[
\text{mu} = 1.81e-5; \quad \% \text{kg/(ms)}
\]
\[ ca = 0.105156; \] % m, axial chord
\[ U = \sqrt{\frac{2*P_{\text{dyn}}}{\rho}}; \] % m/s
\[ \text{Rey} = U \cdot ca \cdot \rho / \mu; \]
\[ M = U / \sqrt{1.4 \cdot 287 \cdot 298.15}; \]
\[ [U, \text{Rey}]; \]

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% 5. Redefine 5 hole probe ports:
%  
% match port number technique of Bryer and Pankhurst for grid5
% United Sensor [2005]: Bryer & Pankhurst [1971]:
%  
% 5 1
% 3 1 2
% 4 3
% 4 3
% 5 1
% 4 3
%  
% correct for scanivalve skips:
\prompt{for} i = 1:length(Pestat)
\prompt{if} Pestat(i) > 0
Pestat(i) = mean(Pestat);
\prompt{i}
\prompt{disp('scanivalve skip correction')}
\prompt{end}
\prompt{end}

% Verify downstream axial velocity & Reynolds #:
\[ \text{dpax} = P_{\text{true}}(:,5) - \text{Pestat}; \]
\[ \text{vax} = \sqrt{2 \cdot \text{dpax} / \rho}; \]
\[ \text{Max} = \frac{\text{vax}}{\sqrt{1.4 \cdot 287 \cdot 298.15}}; \]
\[ \text{Reyax} = \frac{\text{vax} \cdot ca \cdot \rho}{\mu}; \]

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% 6. Send pressures to calibration:
for i = 1:size(Ptrue,1)
    [alpha(i),beta(i),V(i),Pte(i),qe(i)] = grid5(Ptrue(i,:),rho);
end

% 7. Total Pressure loss:
% Define total pressure loss as a matrix corresponding to interrogation zone.
count = 0;
for i = 1:length(z) % spanwise
    for j = 1:length(y) % pitchwise
        count = count+1;
        if ang == 0
            cp(i,j) = (Pidyn(count)-Ptrue(count,5))/((Ptrue(count,5)-Pestat(count)));
        else
            cp(i,j) = (Pidyn(count)-(Pte(count)))/(qe(count));
            % (Pti-Psi)-(Pte-Psi)/qedyn
            % Pedyn is P_exit referenced to Psinlet
        end
    end
end
cp = cp'; % CRITICAL, to make cp y by z for plotting,
% read into the previous loop as z by y (look at j counter),
% want length(y) rows by length(z) columns
% see help for 'surf' plotting.

% 8. Secondary Velocity:
% Compute relative velocity components,
% consistent with coordinate system:
for i = 1:length(alpha)
    u(i) = V(i).*cos(alpha(i)).*cos(beta(i)); % CRITICAL
    w(i) = V(i).*sin(alpha(i)); % CRITICAL
v(i) = V(i).*cos(alpha(i)).*sin(beta(i));  \% CRITICAL
end

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% 9. Vorticity Calculations:

% make w, v matrices first
count1 = 0;
for i = 1:length(z) \% spanwise
    for j = 1:length(y) \% pitchwise
        count1 = count1+1;
        wmat(i,j) = w(count1);
        vmat(i,j) = v(count1);
    end
end

% if these are not y rows by z cols, dwdz = dwdy & dvdy = dvdz
wmat = wmat';  \% CRITICAL
vmat = vmat';  \% CRITICAL

% Compute vorticity & -lambda 2 criteria via finite differences
for i = 1:length(y) \% spanwise, horizontal axis
    for j = 1:length(z) \% pitchwise, vertical axis
        if i == 1
            dwdy(i,j) = (4*wmat(i+1,j)-wmat(i+2,j)-3*wmat(i,j))/(Y(i+2,j)-Y(i,j))/pitch;
        elseif i == length(y)
            dwdy(i,j) = (-4*wmat(i-1,j)+wmat(i-2,j)+3*wmat(i,j))/(Y(i,j)-Y(i-2,j))/pitch;
        else
            dwdy(i,j) = (wmat(i+1,j)-wmat(i-1,j))/(Y(i+1,j)-Y(i-1,j))/pitch;
        end
        if j == 1
            dvdz(i,j) = (4*vmat(i,j+1)-vmat(i,j+2)-3*vmat(i,j))/(Z(i,j+2)-Z(i,j))/span;
        elseif j == length(z)
            dvdz(i,j) = (-4*vmat(i,j-1)+vmat(i,j-2)+3*vmat(i,j))/(Z(i,j)-Z(i-2,j))/pitch;
        else
            dvdz(i,j) = (vmat(i+1,j)-vmat(i-1,j))/(Y(i+1,j)-Y(i-1,j))/pitch;
        end
    end
end

if j == 1
    dvdz(i,j) = (4*vmat(i,j+1)-vmat(i,j+2)-3*vmat(i,j))/(Z(i,j+2)-Z(i,j))/span;
    dwdz(i,j) = (4*wmat(i,j+1)-wmat(i,j+2)-3*wmat(i,j))/(Z(i+2,j)-Z(i,j))/span;
end
elseif \( j == \text{length}(z) \)

\[
dvdz(i,j) = \frac{-4vmat(i,j-1)+vmat(i,j-2)+...}{Z(i,j+2)-Z(i,j)/\text{span}};
\]

\[
dwdz(i,j) = \frac{-4wmat(i,j-1)+wmat(i,j-2)+...}{Z(i,j+2)-Z(i,j)/\text{span}};
\]

else

\[
dvdz(i,j) = \frac{vmat(i,j+1)-vmat(i,j-1)}{Z(i,j+1)-Z(i,j-1)/\text{span}};
\]

\[
dwdz(i,j) = \frac{wmat(i,j+1)-wmat(i,j-1)}{Z(i,j+1)-Z(i,j-1)/\text{span}};
\]

end

\[
or{\text{om}}_x(i,j) = dwdy(i,j)-dvdz(i,j);
\]

\[
S = \begin{bmatrix} dvdz(i,j) & 0.5*(dwdy(i,j)+dvdz(i,j)) \\ 0.5*(dwdy(i,j)+dvdz(i,j)) & dvdy(i,j) \end{bmatrix};
\]

\[
O = \begin{bmatrix} 0 & 0.5*(dwdy(i,j)-dvdz(i,j)) \\ 0.5*(dvdz(i,j)-dwdy(i,j)) & 0 \end{bmatrix};
\]

\[
\lambda = \text{eig}(S^2+O^2);
\]

\[
\lambda_2(i,j) = \text{max}(\lambda);
\]

if \( \lambda_2(i,j) < 0 \)

\[
disp(\text{'Negative } \lambda_2 \text{'});
\]

end

end

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% 10. Cp & Vorticity Plotting:
% 
% define bounds for colorbar:
% cpmin = min(cp);
% cpmax = max(cp);
% scalemin = min(cpmin);
% scalemax = max(cpmax);
% scale = linspace(scalemin,scalemax,30);
% mincpdisp = min(min(cp))
% maxcpdisp = max(max(cp))
% minomdisp = min(min(om_x.*(cx*0.0254)/mean(U)))
% maxomdisp = max(max(om_x.*(cx*0.0254)/mean(U)))
% figure(f1)
% subplot(nr,np,sp)
% contourf(Z,Y,cp,scale), hold on, shading flat
% caxis([mincp,maxcp]) % to put all plots on same scale
0333 hf1 = colorbar;  % handle for colorbar
0334 axis equal, axis ij  % puts the origin in the upper left
0335 hold on
0336 quiver(ztemp,ytemp,w,v,'k')  % plots secondary velocity vectors
0337 [cl,hl] = contour(Z,Y,lambda2,[nl2 nl2],'r-.');
0338 set(hl,'LineWidth',3.5)
0339 set(gcf,'color','white')
0340 if on ~= 1
0341 set(hf1,'Visible','off')
0342 end
0343 if xon == 1
0344 xlabel('z/S'), ylabel('y/P')
0345 end
0346 if titleRE == 1
0347 title(['Re = ',num2str(Re),'K'])
0348 else
0349 title(['g/c_x = ',num2str(gc), '%'])
0350 end
0351 axis([min(ztemp)-0.015 max(ztemp)+0.015 ... 
0352 min(ytemp)-0.015 max(ytemp)+0.015])
0353
0354 scalemin = min(min(om_x.*(cx*0.0254)/mean(U)));
0355 scalemax = max(max(om_x.*(cx*0.0254)/mean(U)));
0356 scaleomx = linspace(scalemin,scalemax,80);
0357
0358 figure(f2)
0359 subplot(nr,np,sp)
0360 contourf(Z,Y,om_x.*(cx*0.0254)/mean(U),scaleomx)
0361 hold on, shading flat
0362 quiver(ztemp,ytemp,w,v,'k')
0363 caxis([minom,maxom])  % to put all plots on same
0364 hf2 = colorbar; axis equal, axis ij
0365 [cl,hl] = contour(Z,Y,lambda2,[nl2 nl2],'r-.');
0366 set(hl,'LineWidth',3.5)
0367 set(gcf,'color','white')
0368 if on ~= 1
0369 set(hf2,'Visible','off')
0370 end
0371 if xon == 1
0372 xlabel('z/S'), ylabel('y/P')
0373 end
0374 if titleRE == 1
0375 title(['Re = ',num2str(Re),'K'])

150
else
    title(['g/c_x = ',num2str(gc), ' %'])
end
axis([min(ztemp)-0.015 max(ztemp)+0.015 ...
min(ytemp)-0.015 max(ytemp)+0.015])
figure(f3)
contourf(Z,Y,lambda2,30), hold on, shading flat
colorbar, axis equal, axis ij
axis([min(ztemp)-0.1 max(ztemp)+0.1 ...
min(ytemp)-0.1 max(ytemp)+0.1])

%11. POST PROCESSING:

adjust z to spanwise locations (dep’t on gap sizes):
if gc == 15
    z = 2.75+0.015*cx+z;
extelse
    z = 2.75+gc/100*cx+z;
end

% Technique from: Yamamoto, A., 1987, "Production &
Devlp’t of 2ndy Flows and Losses in 2 Types of
Straight Turbine Cascades: Part 1--A Stator
Differences in this application:
1. Use of a mass-averaged AXIAL velocity term.
2. Unconditioned AND spatially smooth calculations.

I. WITHOUT ARITHMETICALLY AVERAGING:

Aij = 4*0.5*0.2222*0.1389*in2m^2;
vmass2 = sum((V.*vax').*Aij)/sum(Aij.*vax);
% overall mass-averaged resultant velocity
just sum once because V, vax are vectors
vz = sum((vax.^2).*Aij)/sum(Aij.*vax);
% using vmass2 can be inconsistent between cases
0419  cptnoav = (Pidyn-(Pte'))/(0.5*rho*vz^2);
0420  % want to normalize w/out vmass2
0421  Cpt = sum((vax'.*cptnoav').*Aij)/sum(Aij.*vax);
0422  % overall mass averaged Cp, without
0423  % averaging V & vax
0424
0425
0426  % II. With arithmetically averaging V, vax,
0427  % cpt over four neighboring grid points:
0428  %
0429  % a. Compute new cpt from smmoothed quantitites:
0430  % rearrange as 10x10 matrices
0431  % (spanwise rows X pitchwise cols-->CRITICAL, like cp):
0432  count = 0;
0433  for i = 1:length(z)  % pitch. cols, span. rows (for summation)
0434      for j = 1:length(y)  % y for pitchwise
0435          count = count+1;
0436          V1(i,j) = V(count);
0437          vax1(i,j) = vax(count);
0438          cptnoav1(i,j) = cptnoav(count);
0439          Pidyn1(i,j) = Pidyn(count);
0440          Pte1(i,j) = Pte(count);
0441      end
0442  end
0443
0444  % WITHOUT arithmetically averaging quantitites:
0445
0446  % PITCHwise mass-averaged Cp:
0447  Cptpitch = sum((vax1.*cptnoav1).*Aij,2)./sum(Aij.*vax1,2);
0448  % SPANwise mass-averaged Cp:
0449  Cptspan = sum((vax1.*cptnoav1).*Aij,1)./sum(Aij.*vax1,1);
0450
0451  % arithmetically average the neighbors:
0452  vaxav = vax1;  % boundaries stay the same (linearly extrapolated)
0453  Vav = V1;     % boundaries stay the same (linearly extrapolated)
0454
0455  for i = 2:length(z)-1  % average with four nearest neighbors
0456      for j = 2:length(y)-1
0457          Vav(j,i) = (V1((j-1),i)+V1((j+1),i)+V1(j,(i-1))...  
0458              +V1(j,(i+1)))/4;
0459          vaxav(j,i) = (vax1((j-1),i)+vax1((j+1),i)+...  
0460              vax1(j,(i-1))+vax1(j,(i+1)))/4;
0461      end
end
% using vaxav,
z2 = sum(sum((vaxav.^2).*Aij))/sum(sum(Aij.*vaxav));
% 10x10 matrix overall mass-avg
cpt = (Pidyn1-(Pte1))/(0.5*rho*vz2^2);

% b. Average cpt:
cptav = cpt;
for i = 2:7
    for j = 2:7
        cptav(j,i) = (cpt((j-1),i)+cpt((j+1),i)+...
cpt(j,(i-1))+cpt(j,(i+1)))/4;
    end
end
% pitchwise mass-averaged Cp:
CptPITCH = sum((vaxav.*cptav).*Aij,2)./sum(Aij.*vaxav,2);
% spanwise mass-averaged Cp:
CptSPAN = sum((vaxav.*cptav).*Aij,1)./sum(Aij.*vaxav,1);
% overall mass-averaged Cp:
Cpt2 = sum(sum((vaxav.*cptav).*Aij))./sum(sum(Aij.*vaxav));

% III. Uncertainty in Cpt:
% hysteresis, linearity, repeatability = 0.25% FSO
udp = (Pidyn-(Pte'))*0.0025;
% Pa, barometer resolution
uP = 16.93;
% K, temperature resolution
uT = 0.05;
% kg/m^3, density uncertainty via I.S.O. method
ur = sqrt((uP/(R*T))^2+(uT*p/(R*T^2))^2);
% Pa, uncertainty in dynamic pressure, uv = 3% (Bryer)
uq = sqrt((ur/2.*V.^2).^2+(0.03*rho/2)^2);
% uncertainty in total pressure loss coefficient
ucp = sqrt((udp'./qe).^2+(-uq*(Pidyn-(Pte'))./qe.^2).^2);
% overall mass-averaged uncertainty in total pressure loss
uOpt = sum((vax'.*ucp).*Aij)/sum(Aij.*vax);

% IV. Index of cp terms:
% Cpt2 Arithmetically & overall mass-averaged total pressure loss coefficient.
% CpPITCH  Arithmetically & pitchwise mass-averaged total pressure loss coefficient.
% CpSPAN  Arithmetically & spanwise mass-averaged total pressure loss coefficient.
% Cpt     Overall mass averaged w/out arithmetically avg'g v’s
% Cptpitch Pitchwise mass averaged w/out arithmetically avg’g
% Cptspan  Spanwise mass averaged w/out arithmetically avg’g

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% GRID5.m: Function which puts measured pressures through a calibration to give secondary velocity components.
% INPUTS: Ptrue 6-element vector of measured pressures.
% rho    lab ambient density.
% OUTPUTS: alpha Pitch angle.
% beta   Yaw angle.
% V      Angular Velocity.
% Pt     Total Pressure.
% Pdyn   Dynamic Pressure.

function [alpha,beta,V,Pt,Pdyn] = grid5(Ptrue,rho)

% 1. Load calibration files:
P0 = load(‘calp0cc.dat’);
P0 = P0([3:end-2],:);
P1 = load(‘calpneg25cc.dat’);
P1 = P1([3:end-2],:);
P2 = load(‘calppos25cc.dat’);
P2 = P2([3:end-2],:);
P3 = load(‘calpp125cc.dat’);
P3 = P3([3:end-2],:);
P4 = load(‘calpn125cc.dat’);
P4 = P4([3:end-2],:);
P5 = load('calpn20cc.dat');
P5 = P5([3:end-2],:);
P6 = load('calpp20cc.dat');
P6 = P6([3:end-2],:);
P7 = load('calpn15cc.dat');
P7 = P7([3:end-2],:);
P8 = load('calpp15cc.dat');
P8 = P8([3:end-2],:);
P9 = load('calpn5cc.dat');
P9 = P9([3:end-2],:);
P10 = load('calpp5cc.dat');
P10 = P10([3:end-2],:);
P = [P1;P5;P7;P9;P0;P10;P8;P6;P2];
a = [-25;-20;-15;-5;0;5;15;20;25]; % deg, pitch angles
b = [27 18 9 0 -9 -18 -27]'; % deg, yaw angles
[A,B] = meshgrid(a,b); % for plotting
P = (P(:,[2:7]).*1.1031+0.0028)*249.09; % V-->Pa
Pn(:,4) = P(:,3); % match port numbering (switch 3&4)
Pn(:,3) = P(:,4);
Pn(:,5) = P(:,1); % match port numbering (switch 5&1)
Pn(:,1) = P(:,5);
P(:,3) = Pn(:,3);
P(:,4) = Pn(:,4);
P(:,1) = Pn(:,1);
P(:,5) = Pn(:,5);
k =0;
clear P1 P2 P3 P4 P5 P6;
% matrices (length(alpha)xlength(beta), rows of const. alpha)
for i = 1:length(a)
    for j = 1:length(b)
        k = k+1;
P1(i,j) = P(k,1);
P2(i,j) = P(k,2);
P3(i,j) = P(k,3);
P4(i,j) = P(k,4);
P5(i,j) = P(k,5);
P6(i,j) = P(k,6);
end
end

% calibration properties:
pm = (P1+P2+P3+P4)./4;
fb = (P2-P4)./(P5-pm);
fa = (P3-P1)./(P5-pm);

H = P6; % ref'd against upstream Ps
S = (H-P5)./(P5-pm); % Static Pressure Coefficient
Q = (P5-pm)./P6; % Dynamic Pressure Coefficient, dynamic

% measured properties
pmtrue = (Ptrue(1)+Ptrue(2)+Ptrue(3)+Ptrue(4))./4;
fbtrue = (Ptrue(2)-Ptrue(4))./(Ptrue(5)-pmtrue);
fatrue = (Ptrue(3)-Ptrue(1))./(Ptrue(5)-pmtrue);

% INTERPOLATION:
alpha = griddata(fa,fb,A',fatrue,fbtrue); % interpolate for alpha
beta = griddata(fa,fb,B',fatrue,fbtrue); % interpolate for beta

Sp = griddata(A,B,S',alpha,beta); % interpolate for Static Pressure Coefficient:
Qp = griddata(A,B,Q',alpha,beta); % interpolate for Dynamic Pressure Coefficient:

Pdyn = (Ptrue(5)-pmtrue)./Qp; % Dynamic Pressure
V = sqrt(2*Pdyn/rho); % |V|
Pt = Sp*(Ptrue(5)-pmtrue) + Ptrue(5); % ref'd to upstream Ps

figure(1) % f(a) vs. f(b), to give angles
hold on
plot(fa,fb,'r:.
plot(fa',fb', 'r:.
ylabel('f(\beta) = (p_2-p_4)/(p_5-p_m)')
xlabel('f(\alpha) = (p_{3-p_{1}})/(p_{5-p_{m}})\)')
set(gcf,'color','white')

v = [0:0.025:0.125,0.125:0.075:0.5,0.5:0.2:2]
figure(2)  
  \text{Contour plot of Static Pressure}
  \% Coefficient, S, to give Sp
[c,h] = contour(A',B',S,v);
clabel(c,h);
axis([-25 25 -27 27])
xlabel('\alpha')
ylabel('\beta')
set(gcf,'color','white')
title('S')

v1 = [0:0.05:0.95,0.95:0.025:1.2];
figure(3)  
  \text{Contour plot of Dynamic Pressure}
  \% Coefficient, Q, to give Qp
[cq,hq] = contour(A',B',Q,v1);
clabel(cq,hq);
axis([-30 30 -30 30])
xlabel('\alpha')
ylabel('\beta')
set(gcf,'color','white')
title('Qp')
REFERENCES


