CONTROL OF THE TIP-GAP FLOW OF A LOW PRESSURE TURBINE BLADE IN A LINEAR CASCADE

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by

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Abstract

by

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This research documents passive and active flow control techniques for reducing
losses associated with the tip leakage vortex of a low pressure turbine blade cascade.
Experiments were conducted in a linear cascade of Pratt and Whitney Pack-B tur-
bine blades for an inlet Mach number of 0.2 and an exit Mach number of 0.3. The
flow was documented using blade-tip and end wall static pressure measurements and
downstream total pressure loss coefficients. Additionally, surface flow visualization
was performed on the end wall and blade tip for a greater understanding of the flow
behavior.

Blade mounted passive flow control investigated two gap-to-chord ratios for
which four tip thickness-to-gap ratios were simulated using pressure-side winglets.
The performance of a partial suction-side squealer tip was determined to depend on
both gap-to-chord and thickness-to-gap ratios. Active flow control utilized a SDBD
plasma actuator situated on the blade tip, with plasma initiated at an unsteady
frequency to amplify unsteady characteristics inherent in the flow. This resulted
in modest reductions in the downstream total pressure loss associated with the tip
leakage vortex.

Vortex generators on the end wall were designed to produce vorticity of opposite
sign of the tip leakage vortex. Two vortex generator heights and two placements were investigated at two gap-to-chord ratios. For both gap-to-chord ratios, a vortex generator roughly the height of the gap placed upstream of the trailing edge produced the best results. Active plasma vortex generators in similar locations were also investigated, and a plasma actuator across from the trailing edge was found to be roughly one third as effective as the passive vortex generators at reducing the losses associated with the tip leakage vortex.

Wall roughness was also investigated as a means of diffusing the tip leakage vortex. 3-dimensional roughness almost completely eliminated the tip leakage vortex as observed downstream while modestly increasing the blade tip loading. 2-dimensional roughness decreased total pressure losses associated with the tip leakage vortex downstream by 40% with little change to the blade loading. Simulated roughness using plasma actuators had essentially no effect on the tip gap flow.
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CHAPTER 1

BACKGROUND

This document details research performed in the Hessert Laboratory to study the behavior of the flow in the tip-gap region of a low pressure turbine blade using a linear cascade of Pratt and Whitney Pack-B low pressure turbine blades. The motivation for this research, a brief explanation of some of the previous studies in the field, and the objectives of this research are outlined in this chapter.

1.1 Motivation

Oil prices have seen a steady increase in recent years. As a result, aircraft fuel efficiency has gained increasing attention. In 2002, Lattime\cite{1} demonstrated that with the 2001 average fuel cost, a 1% increase in fuel efficiency would have saved the U.S. commercial aircraft industry alone over $160 million. As fuel prices and use have continued to increase since 2002, so have the savings available that can result from increases in efficiency.

Most modern jet aircraft are powered by axial turbine engines, also called turbo-machines. A number of flow control techniques have been suggested for all of the internal portions of a turbine engine\cite{12}. Much of the research has focused on the compressor and turbine stages. The turbine stages are of particular interest because that is where kinetic free stream energy is converted to work.
A small clearance is necessary between the rotating turbine blades and the turbine casing to prevent the blades from contacting the casing. When this occurs, it is referred to as "blade rub". If a blade rubs even a little, the blade and casing are damaged, and life of the engine is decreased. If the blade rub occurs frequently, catastrophic failure of the engine can occur. Additionally, as an aircraft engine goes through a flight profile, heating and shifting of the blades causes the gap to expand and contract. Typical clearances that occur throughout a flight are presented in Figure 11. Therefore, the turbine stage must be designed for the minimum clearance that occurs during flight. This may result in significant clearances during some portions of the flight.

The clearance between the blade and the casing, known as the tip-gap, is a major source of loss in gas turbine engines for two reasons. First, the flow in the gap is not turned by the blades and thus does not contribute to the energy extraction. Second, the flow exits the tip-gap as a vortical structure. This structure generates pressure losses as it mixes with the main-stream flow. Additionally, the hot, coherent vortex increases wear and unsteady loads on the turbine components.

Approaches to increase turbine efficiency by modifications to the tip-gap region also increase turbine life and decrease the risk of catastrophic failure. That is the motivation for this research. The approach of this research is to focus on understanding the physics of the tip-gap flow, and to seek methods for reducing the losses associated with the flow through the tip-gap and, in particular, with the tip leakage vortex.

1.2 Loss Measurement

In order to study the energy loss that occurs in a turbine stage, it is important to understand what this loss is and how it can be measured. Typically, loss in axial
Various clearances seen during a typical flight mission.

Turbo-machines is defined as a loss of isentropic efficiency. This loss is the difference between the actual work and the ideal work the system could generate if the flow were isentropic. If the flow is adiabatic, which is typically true for turbo-machines, entropy creation by flow irreversibility is the primary contributor to this loss.

Turbines extract work by turning flow stream which generates an aerodynamic load on the blades. The aerodynamic load results in a torque that rotates the blade row about the turbine axis. It is difficult to obtain measurements of a rotating flow. To simplify turbine studies, it is typical to unroll the blade row and study it as a linear cascade. The blades in a linear cascade are stationary and the flow turns through a corner bend in the tunnel. This experimental arrangement makes studying the properties of the flow, such as pressure and velocity, both upstream and downstream of the blade row much easier. This greatly simplifies the measurements and helps to gain insight into the loss mechanisms. For these measurements for quantifying loss to be meaningful, a method for determining how they relate to
losses in a rotating rig is necessary.

Entropy generation is always an indication of loss, and entropy production across a blade row does not depend on whether the blades are stationary or rotating. Therefore, it is of interest to relate measurable quantities, such as temperature and pressure, with entropy creation. Denton[8] used the second law of thermodynamics to show the relationship between pressure and entropy in terms that apply to both linear cascades and rotating blade rows. A similar derivation is repeated here, following the derivation of Jumper[13].

The first and second laws of thermodynamics provide the following equation that relates entropy \( s \), internal energy \( u \), pressure \( P \), and specific volume \( v \) which is,

\[
Tds = du + Pdv. \tag{1.1}
\]

Because we are considering a fixed mass system, the specific (per mass) values are considered. This equation is comprised entirely of state variables, and is thus path independent. Therefore, only the inlet and exit states are needed for integration.

This research is performed in air, so the perfect gas laws can be applied. We also use the definition of enthalpy \( h \), and the specific heat of air at constant pressure \( (c_p) \). That is,

\[
PV = RT, \tag{1.2}
\]

\[
h = u + Pv, \tag{1.3}
\]

and
\[ c_p = \left( \frac{\partial h}{\partial T} \right)_P. \]  

(1.4)

Additionally, it is noted that \( u \) is only a function of \( T \) so that \( h \) is also only a function of \( T \). Then the partial differential in Equation 1.4 becomes an exact differential and,

\[ dh = c_p dT. \]  

(1.5)

Now, returning to Equation 1.1 and using Equation 1.3 for \( u \) yields

\[ ds = \frac{d(h - P v)}{T} + \frac{P}{T} dv. \]  

(1.6)

Expanding results in

\[ ds = \frac{dh}{T} - \frac{P dv}{T} - \frac{v}{T} dP + \frac{P}{T} dv. \]  

(1.7)

This research involves a fixed specific volume, so \( dv = 0 \). The entropy equation then simplifies to

\[ ds = c_p \frac{dT}{T} - \frac{v}{T} dP. \]  

(1.8)

Using Equation 1.2, the second term can be rewritten as \( \frac{R}{P} dP \). Then Equation 1.8 can be integrated to give

\[ s_2 - s_1 = \int_1^2 \frac{c_p}{T} dT - \int_1^2 \frac{R}{P} dP. \]  

(1.9)

The constants \( R \) and \( c_p \) can be pulled out of the integrals. The change in entropy can then be written as

\[ \Delta s = s_2 - s_1 = \frac{c_p}{R} \ln \left( \frac{T_2}{T_1} \right) - R \ln \left( \frac{P_2}{P_1} \right). \]  

(1.10)
This says that the change in entropy is related to the ratios of temperature and pressure.

For this research, it is desirable to know the relation between the total properties. For a steady control volume, the production of entropy ($\dot{P}_s$) is zero. That is

$$\dot{P}_s = \dot{m}_{out}s_{out} - \dot{m}_{in}s_{in} - \frac{Q_{in}}{T_{in}} = 0,$$

where $\dot{m}$ is the mass flow. Given that there is no work on the system and the mass is fixed, $s_{out} = s_{in}$. Also by definition, the entropy in an isentropic flow does not change, so that $s_1 = s_{t1}$ and $s_2 = s_{t2}$. Based on this, Equation 1.10 can be written in terms of the total properties, namely:

$$\Delta s = \Delta s_t = c_p \ln \left( \frac{T_{t2}}{T_{t1}} \right) - R \ln \left( \frac{P_{t2}}{P_{t1}} \right).$$

Finally, given that the temperature change in the cascade is negligible,

$$\Delta s = -R \ln \left( \frac{P_{t2}}{P_{t1}} \right).$$

Equation 1.13 says that the ratio of the total pressures in the control volume measured upstream and downstream of a blade row is proportional to the change in entropy. This is the approach that will be used to estimate the losses associated with the tip-gap flow.

1.3 Loss Mechanisms in Turbines

Research on loss producing mechanisms in turbines dates back to the 1950s. These have been typically classified as “profile loss”, end wall or “secondary loss”, and leakage or “tip-gap loss”. However, the flow field in a turbine is highly complicated and these loss mechanisms interact a great deal. An appreciation for and
understanding of the complex flow fields associated with turbine efficiency is an important step towards studying the losses associated with the tip-gap flow.

Profile loss is a result of boundary layer growth on the blade surfaces. The flow field that is associated with profile loss is assumed to be two dimensional. It was the first type of loss to be calculated and is relatively simple to predict.

One of the earliest studies of “secondary” losses was by Rains[14] in 1954. This was performed in a rotating water compressor rig. Rains distinguished between secondary and tip-gap flow features, and noted the importance of the tip-gap flow and the losses associated with it. Since that time, “tip-gap” and “secondary” losses have been distinguished from one another. Research continues on the mechanisms behind these two flow features. This section focuses on these two flow features and their possible interaction.

1.3.1 Secondary Losses in Turbines

The secondary losses in turbine blade cascades arise from the boundary layers on the end walls. Much of the early investigations of secondary losses in turbo-machines, such as those by Rains[14], focused on compressors. In the late 1970’s researchers began paying more attention to the turbine stage. Despite years of research, the secondary losses are not fully understood. An excellent review of work in the area was provided by Langston[15].

There are two primary structures in the secondary flow. One is the horseshoe vortex, which is created by the stagnation point at the blade leading edge. The horseshoe vortex wraps around the blade near the blade tip. The pressure-side leg of the horseshoe vortex moves from the blade, across the passage, towards the suction side of the adjacent blade. The suction-side leg of the horseshoe vortex remains close to the blade.
Another secondary flow structure is the “passage” vortex. As flow travels through the passage between the blades, the boundary layer is subjected to a transverse pressure gradient caused by the flow turning across the blade row. This transverse pressure gradient causes the boundary layer to roll up into a vortex, termed the “passage” vortex.

The remainder of this section contains a discussion of the change in understanding of the formation and interaction of the secondary vortices over the last several decades. A basic understanding of these secondary flow vortices was presented by Langston, Nice, and Hopper\cite{16} in 1977. They used ink flow visualization on one end wall and blade, as well as pressure measurements on the end wall, and on the pressure- and suction-sides of one blade. Based on their findings, they proposed flow characteristics which were schematically presented by Langston\cite{2} in 1980. This is shown in Figure 1.2. This figure depicts the boundary layer on the end wall. The stagnation pressure on the leading edge of the blade forms a horseshoe vortex that wraps around each blade. The pressure-side leg of the horseshoe vortex is pushed by the transverse pressure gradient towards the suction-side of an adjoining blade. Langston referred to this as the “passage vortex.” The passage vortex generates a smaller secondary vortex with an opposite sense of circulation. This is noted as the “counter vortex” in the illustration in Figure 1.2. The counter vortex has the same sense of circulation as the tip leakage vortex, which is not shown in this illustration.

In 1983, Sieverding and Bosche\cite{17} used colored smoke to visualize the interaction between the different vortices in the tip-gap region. They found that the suction-side leg of the horseshoe vortex wrapped around the passage vortex when it reached the suction-side of the blade. Both of these structures have the same sense of circulation so they could combine their effects. In 1987, Sharma and Butler\cite{3} proposed a new model based on previous investigations by others as well as their
Figure 1.2. Behavior of secondary flows as proposed by Langston[2].

own flow visualization investigations in wind tunnel and water tunnel cascades. This led to the illustration of the flow in the tip-gap region that is shown in Figure 1.3. This shows the formation of the horseshoe vortex proposed by Langston modified now in that as the pressure-side leg moves across the passage to the suction-side, it gathers fluid from the passage end wall boundary layer and becomes the passage vortex, rather than simply merging with it. Additionally, it shows the merging of the suction-side leg of the horseshoe vortex with the passage vortex that was observed by Sieverding and Bosche.

The most recent model of the secondary flows in the tip-gap region known to the author was proposed by Wang et al. [4], who incorporated the models previously discussed, and used smoke wires to investigate the flow field. They also compared the visualizations with local mass transfer measurements. An illustration of this model for the tip-gap flow field is shown in Figure 1.4. This model includes additional counter-rotating secondary vortices formed by the horseshoe vortex. The counter-rotating secondary vortices form on both the pressure- and suction-side legs of the primary horseshoe vortex. In this model of the flow, the passage vortex contains
the pressure-side leg of the horseshoe vortex and its secondary vortex. These merge with the suction-side leg of the horseshoe vortex and its secondary vortex that wraps around the neighbor blade across the passage. The resulting interaction is a very complex interaction of large and small scale vortices. This is illustrated in the cross-section views in Figure 1.4.

1.3.2 Tip-Gap Flow

The third mechanism of loss in the turbine is the tip-gap loss. As already mentioned, the clearance between the blade tip and end wall provides a passage through which the air flow can leak from the pressure-side to the suction-side of the blade. An illustration of the flow structures that are associated with the tip-gap flow, as given by Bindon, is shown in Figure 1.5.

As shown by the figure, the flow that enters the gap separates at the pressure-side edge. The chordwise extent of this separation and the location of reattachment depend on the ratio of the blade thickness and the gap height. The flow that exits
from the gap to the suction side of the blade is turned by the flow in the passage. This causes it to roll up into a stream-wise vortex referred to as the “tip leakage vortex.”

This tip leakage vortex is known to be a major source of loss in turbines and should be easier to affect than the more complicated passage and horseshoe vortices. Hence, it has been the subject of many detailed investigations. Sjolander and Amrud\[18\] performed detailed flow visualization and pressure measurements on the tip-gap flow. They proposed that rather than a single tip leakage vortex, there are three different vortices that were observed to shed from the tip at various chord-wise locations. The corresponding pressure change on the end wall, however, only appeared to show one vortex. All of the observed vortices rotated in the same direction, and possibly merged downstream.

Tallman and Lakshminarayana\[6\] performed a numerical simulation of the tip-gap flow and also observed the flow to be exiting the gap at a variety of points. The
three distinct flows they proposed are illustrated in Figure 1.6. The “Type 1” flow crosses the gap in the leading half chord of the blade. This flow is rotational even before exiting the gap due to a mean flow shear on the blade tip. The Type 1 flow clings to the suction side of the blade. It was proposed that this formed the core of the tip leakage vortex.

The “Type 2” flow also exits from the gap in the leading half chord of the blade. This flow, however, is closer to the end wall side of the gap, and thus is rotated by the shear stress on the end wall. Therefore, the Type 2 flow has an opposite sense of rotation than the Type 1 flow. The Type 2 flow structure travels with the tip leakage vortex formed by the Type 1 flow, but does not roll up inside it.

The “Type 3” flow forms on the downstream half chord of the blade. This flow exits the gap and wraps around the Type 1 flow, joining the tip leakage vortex. The tip leakage vortex separates from the blade somewhere downstream of the half chord.
location. These numerical results appear to agree with Sjolander and Amrud’s\textsuperscript{[18]} observations.

The research by Tallman and Lakshminarayana\textsuperscript{[6]} seems to indicate that the shear stress on the flow by the blade tip as the flow moves through the gap is the primary source of rotation, rather than the historical belief that the flow angularity between the gap flow and main stream flow causes the rotation. It also brings into question the idea that the downstream 50\% of the blade is most critical to the tip leakage vortex formation.

Figure 1.6. Tip-gap flows from a numerical model by Tallman and Lakshminarayana\textsuperscript{[6]}.

Most of what is known of the downstream effect of the tip-gap flow is due to the detailed investigations of Yamamoto\textsuperscript{[19, 20, 21]}, who used a five hole Pitot probe in multiple planes located upstream of the leading edge, in the blade passage, and downstream of a linear cascade of turbine blades. The earliest of these works focused on the secondary flows. Later work provided details of the interaction between the secondary flows and the tip leakage vortex. In 1989, Yamamoto\textsuperscript{[21]} theorized based on secondary velocity vectors in a cascade passage and downstream of it, that the flow that exited the tip-gap was driven away from the end wall by the passage vortex, creating the leakage vortex. In the 1988 study, Yamamoto\textsuperscript{[20]} found that
the leakage vortex was formed by fluid taken from the passage vortex. Thus the larger the gap, the more fluid was available to the leakage vortex and the less to the passage. This caused the strength of the leakage vortex to grow and the strength of the passage vortex to decrease. Additionally, because the tip leakage vortex rotates opposite to the passage vortex, the tip leakage vortex pushed the passage vortex away from the suction-side of the blade and back toward the pressure-side of the passage.

In a study by McCarter[22] in a rotating turbine rotor rig, a similar interaction between the leakage and passage vortices was observed. Additionally, the study indicated that the tip leakage vortex dissipated more slowly than the passage vortex and the effect of the tip leakage vortex was critical to downstream loss.

Research continues in to the effects of the tip leakage vortex. In 2008 Palafox et al.[23] showed strong interactions between the tip leakage vortex, passage vortex, and scraping vortex using PIV in a linear cascade with a moving end wall. The present research was conducted with a stationary end wall. Therefore, there is no scraping vortex. However, comparisons between linear cascades and rotating rigs have shown that the absence of the scraping vortex did not significantly alter the dominant features of the tip-gap flow, in particular, the loss mechanism of the tip leakage vortex.

1.4 Flow Control Techniques

The effect of the tip-gap flow and related losses are now reasonably understood. While further efforts towards understanding these flow structures continue, efforts have begun to reduce the associated losses caused by the flow. A variety of flow control techniques have been examined. These can be described as passive and active techniques. Passive flow control techniques involve fixed modifications or additions
to the blade or end wall. Active flow control techniques make an active change in
the flow by the injection or removal of mass, momentum, or a body force. Active
flow control techniques include blowing\cite{24}, cooling\cite{25} \cite{26}, and plasma actuators
\cite{11} \cite{27} \cite{28} \cite{29}.

1.4.1 Passive Control: Blade Tip Geometry Modifications

The tip leakage vortex has shown extreme sensitivities to blade tip geometry.
Bindon and Morphis\cite{30} investigated three blade tip geometries: a flat square-edged
tip, a flat radiused pressure-edge tip, and a contoured tip radiused to create a radial
leakage flow. They found that radiusing the pressure edge significantly reduced the
tip-gap pressure loss by eliminating the separation bubble on the blade tip. However,
it increased the downstream mixing losses. Additionally, this study found that a
blade created to match a blade from a previous study produced very different loss
characteristics from the original. The conclusion was that creating two “identical”
blades was very difficult, and that the micro-flow phenomena was highly sensitive
to small changes in tip geometry.

Ameri and Bunker\cite{7} performed numerical and experimental heat transfer in-
vestigations on sharp and radiused edged blades. They found that simulations of
radiused edges agreed better with the experiments because of the absence of the
separation bubble on the blade tip. Streamlines for flat and radiused edge blades
from their simulations are compared in Figure 1.7. These illustrate the affect of
a pressure-side radius. The top illustrates a sharp-edged blade, and the bottom a
radiused-edge blade. The images show that the tip leakage vortex forms further
upstream for the sharp-edged blade. Additionally, their studies showed that heat
transfer on the radiused tip was higher than that on the flat tip.

Noting the impact blade tip geometry has on the tip-gap flow, Heyes, Hodson,
Figure 1.7. Comparison of flow behaviors for flat and radiused edge blades[7].
and Dailey\cite{31} investigated wake pressure losses for various geometries. These investigations included flat tip blades of various edge radius, partial suction-side squealer tips, and partial pressure-side squealer tips. Squealer tips are a cavity in the blade tip formed by recessing a portion of the blade. A full squealer tip recesses the entire middle of the blade tip except for a rim all the way around the perimeter. A partial squealer tip leaves a partial rim in specified locations. A schematic of different squealer geometries is shown in Figure 1.8. The approach of squealer tips provide an additional pressure drop in the clearance gap under the blade. It also may act as a seal to the flow, preventing it from escaping the gap. Heyes et al.\cite{31}, found that both partial squealer tip configurations reduced the wake loss compared to the flat tip, with the suction-side squealer being the most beneficial of the two configurations. Key and Arts\cite{32} performed an extensive investigation of flat tip and squealer tip geometries. These investigations also showed a reduction in wake loss when squealer tips were implemented. Their investigation was one of the very few performed at high speeds, so their results are particularly significant.

Douville\cite{9} performed cascade experiments over a range of Reynolds numbers that bridged the gap between those of other investigations. Douville’s research documented the effect of Reynolds number, gap size, and flat tip vs. squealer tip geometry. It was found by varying the gap size for the same blade, the squealer tip had a greater impact on the tip leakage vortex at smaller gaps. Specifically, a blade thickness-to-gap-height ratio greater than 3.5 resulted in more sensitivity to the addition of a squealer tip. This was thought to be related to Denton’s\cite{8} proposal that the separation bubble on the blade tip does not reattach for blade thickness-to-gap ratios less than 4. Denton coined the term of a blade being “thin” or “thick”. A blade is “thin” if the thickness-to-gap ratio was less than 4, and “thick” if it was greater. Denton theorized that if the blade was “thin”, the flow did not reattach
after separating from the pressure-side edge. Douville theorized that this made the flow in the tip-gap less susceptible to flow control techniques like the suction-side squealer tip. If, however, the blade was “thick”, the flow reattached to the blade tip before exiting the gap. Douville theorized that this made the flow more susceptible to the effects of a partial suction-side squealer. A schematic representation of a “thick” and a “thin” blade arrangement is shown in Figure 1.9.

Figure 1.8. Schematic of various blade mounted passive flow control techniques compared with a flat-tip geometry.

It is interesting to note in the work by Douville[11] that pressure measurements on the end wall across from the blade showed a large region of low pressure for flat-tip blades that was not present for the blade with a squealer tip. This indicated that fluid was trapped under the blade by the squealer tip. Ameri, Rigby, and Steinhorssen[33] performed a numerical investigation of heat transfer on turbine blades with a squealer tips and found that while the wake pressure loss was decreased, the heat transfer to the blade tip was increased. Azad, Han, and Boule[34] performed heat transfer experiments on blades with squealer tips and found their results to agree well with the simulations of Ameri et al. [33]. This increased heat transfer is possibly due to the air that was trapped and recirculated under the blade.
tip as observed by Douville. The added heat transfer could decrease the life of the blades, and therefore makes squealer tips less desirable as a flow control strategy.

Another blade tip geometry modification that has been investigated to reduce the effects of the tip leakage vortex is a “winglet”. This geometry was first proposed by Booth, Dodge, and Hepworth\cite{35}. It involves an extension on either the pressure- or suction-sides, or both, that protrudes into the blade passage. A schematic of a pressure-side winglet is shown on the far right of Figure\ref{fig:blade_tip_characterization}. Booth et al. investigated a variety of combinations of winglets and squealer tips. The investigations indicated that winglets performed better. Dey\cite{26} performed 5-hole probe and heat transfer measurements on blades with flat tips, suction-side winglets, pressure-side winglets, suction-side squealer tips, pressure-side squealer tips, and both suction- and pressure-side squealer tip geometries. This investigation also varied the gap height. The results showed that suction-side winglets had no effect on the wake pressure loss, except to move the tip leakage vortex further from the blade. The

\begin{figure}[h]
\centering
\includegraphics[width=0.8\textwidth]{blade_tip_characterization.png}
\caption{Schematic of “thick” and “thin” blade flow characterization as proposed by Denton\cite{8} and Douville\cite{9}.}
\end{figure}
pressure-side winglets, however, were found to be highly effective. The suction-side squealer tips performed the best among the squealer tip geometries. Their effectiveness was not improved by the addition of a pressure-side squealer tip. Additionally, they worked significantly better at the smallest gap heights. For the same gap height, the winglets outperformed the squealer tips.

Papa, Goldstein, and Gori [36] investigated two blade tip geometries in order to obtain more detailed measurements. Their investigations focused on a full squealer tip and a combination suction-side squealer tip with a pressure-side winglet. The results of those investigations showed that the second configuration had beneficial results compared to the first.

1.4.2 Casing Treatments

Casing treatments are an alternative to tip geometry modifications. Many casing treatments have been investigated for the compressor stage to affect stall. An example of this research is by Vo, Tan, and Greitzer [37]. The addition of slots or grooves has shown improvement in controlling stall [38, 39, 40]. These treatments have not been used in the turbine stage because of the high temperatures. However, non-axisymmetric end walls [41, 42] and end wall boundary layer removal [43] have been used to reduce the losses due to the horseshoe vortex. A recent study by Rao, Gümüşel, Kavurmacioglu and Camci [44] in a rotating, low temperature turbine found roughness on the casing wall to have a significant effect on the tip leakage vortex.

1.5 Plasma Actuators

In addition to passive control techniques, it is of interest to find active control techniques that are effective at reducing the effects of the tip leakage vortex. This research focuses on active flow control using plasma actuators.

The mechanisms for single dielectric barrier discharge (SDBD) plasma actua-
tors have been and continue to be investigated\textsuperscript{45, 46}. These actuators have been shown to work in a number of applications including separation control on airfoils at high angles of attack\textsuperscript{47}, and separation control on low pressure turbine blades\textsuperscript{48}. Recently a new plasma actuator configuration that produces stream-wise vortices has been developed and shown to be successful for separation control by He\textsuperscript{49, 50}. This actuator design ionizes the air on two sides of the exposed electrode. This induces the flow in two directions away from the exposed electrode. When the exposed electrode is aligned with the mean flow direction it produces two counter-rotating stream-wise vortices.

Thomas et al.\textsuperscript{10} have recently investigated plasma actuators with exposed electrodes that had sawtooth edges on the exposed electrode. These were shown to generate body forces that were larger than those with straight edges at the same voltage. A comparison of the generated body force versus the actuator input voltage for straight and sawtooth edged exposed electrodes is shown in Figure\textsuperscript{1.10}. This offers the possibility to obtain greater actuator authority at lower voltages. Research continues toward actuators that exhibit larger body forces.

The ac voltage supplied to plasma actuators can be manipulated to create a “steady” force or an unsteady force. A steady operation can produce a steady wall jet that can act as a flow blocking device like the squealer tip. Unsteady operation can be used to excite fluid instabilities that can enhance mixing.

A schematic of the tip-gap flow is shown in Figure\textsuperscript{1.11}. This indicates that it is possible that two flow instabilities are present in the tip-gap flow. One instability is related to the shear layer formed by the gap flow shearing with the free-stream flow, and the other is related to the exit jet core. For flow instabilities, the amplification rate is typically governed by a non-dimensional frequency corresponding to the Strouhal number, $\beta$, governed by Equation\textsuperscript{1.14}.
Figure 1.10. Body force as a function of voltage for straight and sawtooth edge exposed electrodes, by Thomas et al. [10].

Figure 1.11. Schematic of the tip-gap flow with possible sources of flow instabilities.
\[ \beta = \frac{f \cdot l}{U}, \] (1.14)

where \( f \) is the instability frequency, \( l \) is a characteristic length scale, and \( U \) is a characteristic velocity. For free shear layers, a commonly used length scale is the momentum thickness. A representative velocity is the value at the edge of the shear layer. Typical Strouhal numbers of the most amplified frequencies are on the order of \( 0.01 < \beta_s < 0.04 \) [51]. For a viscous jet flow, a representative length scale is the diameter of the jet. The characteristic velocity is the velocity of the center of the jet. For circular jets, the most amplified Strouhal numbers are typically \( 0.25 < \beta_j < 0.5 \), but most often \( \beta_j = 0.44 \) [51]. The instability of 2-dimensional slot flows at a wall that are meant to simulate a blade tip-gap flow were performed by Bae et al. [52]. They found what appeared to be a preferred instability frequency. The effect that these instabilities might have on the tip leakage vortex development and the loss coefficient is an open question.

Passive and active control on the blade or casing in the turbine stage have challenges due to the hot, turbulent gasses. Approaches that trap the hot gas and enhance heat transfer to metal parts is particularly undesirable. Therefore, any of the approaches have to minimize that effect and be robust enough to withstand the hot environment as well as rub events. The plasma actuator has that potential if sufficient flow authority can be reached.

1.6 Objectives

The aim of the research discussed in this document is to further document the flow physics associated with the blade tip clearance flows. Ultimately this is intended to lead to methods for reducing the losses associated with this flow. A particular emphasis will be placed on the tip leakage vortex.
As a first step, the focus will be on passive flow control approaches. This will involve investigations of the sensitivity of passive control devices on parameters such as gap-to-chord and thickness-to-gap ratio. Next, active flow control techniques will be sought which emulate the successful passive flow control approaches.

Given that blade mounted flow control devices have historically been the primary passive method used for reducing the tip-gap losses, these will be investigated. These investigations will include squealer tips and winglets. Some casing treatments will also be examined. These passive control strategies will be used to guide the development of plasma actuators. The effectiveness of the plasma actuators will be compared against both the baseline flow and the passive flow control devices. In all cases, the effect of the flow control device on the tip-gap parameters will be investigated. The metric of merit in all cases will be to minimize the downstream pressure loss coefficient associated with the tip leakage vortex measured in the wake of the blades.
CHAPTER 2

EXPERIMENTAL SET-UP

The experiments discussed in this document were performed in the main lab of the Hessert Laboratory at the University of Notre Dame. This chapter outlines the experimental set-up, detailing the test facility and blade construction, and outlining the modifications for different flow control strategies.

2.1 Experiment Facility

The Hessert Laboratory is equipped with a set of vacuum pumps that are used in draw-down wind tunnel experiments. At the start of the research the vacuum pumps consisted of three Allis-Chalmers model 1165 water-sealed pumps. Each pump could generate a vacuum pressure of 18 inches of mercury at a flow rate of 3310 cfm. During the last part of the research, the three pumps were replaced with two Dekker Magna-Flo 4790-1000-050 vacuum pumps. Each of these pumps can create a vacuum pressure of 18 inches of mercury at a flow rate of 4640 cfm. The Dekker pumps are driven by variable rpm motors which are used to control the flow rate of air.

A linear cascade of Pratt & Whitney Pack-B low pressure turbine (LPT) blades was designed and constructed to utilize the vacuum pump system. A photograph of the cascade is shown in Figure 2.1 A schematic of the cascade is presented in
Figure 2.2 For reasons of space, the cascade was oriented vertically with the inlet at the top near the ceiling.

Figure 2.1. Photo of the test section and diffusor in which the present research was performed.

A one-inch radius was used at the inlet to the cascade facility to smooth the inflow and prevent flow separations. Following the flow inlet, turbulence management started with 5 inches of honeycomb having 0.25 inch cells that was designed to remove any swirl from the flow. This was followed by 2 low solidity screens that were spaced 8 inches apart. This was followed by a 8 inch long settling chamber. The flow then passed through a contraction. The contraction had an 8:1 cross-
Figure 2.2. Schematic of transonic linear cascade facility.
Figure 2.3. Detail of cascade test section.
sectional area ratio and a length-to-diameter ratio of 1. The shape of the walls of the contraction was a fifth order polynomial. The exit dimension of the contraction was six inches by four inches, which matched the dimensions of the cascade test section.

A schematic of the cascade test section is shown in Figure 2.3. The test section was fabricated from 0.5 inch thick Plexiglas. It was held together by screws. The inlet and exit were Flanged for attachment. The cascade had a 95 degree turning angle based on an inlet angle of 55 degrees and an exit angle of 30 degrees. The test section area decreased through the cascade from the inlet area of 24 square inches to an exit area of 20 square inches. Following the test section, the flow passed through a 5 foot long diffusor that had an expansion angle of 3 degrees.

The cascade was comprised of three Pratt & Whitney Pack-B LPT blades that had a stagger angle of 26.16 degrees. The blades had an axial chord, $c_x$, of 4.14 inches, span of 4 inches. The aspect ratio was 0.96 and solidity was $(c_x/pitch)$ of 1.13.

With these specifications, the tunnel was capable of running at an exit Mach number of 0.6. All of the experiments discussed in this research were performed at an exit Mach number of 0.3 corresponding to an inlet Mach number of 0.2. The inlet Reynold’s number based on axial chord was $0.5 \times 10^6$. The flat plate Reynold’s number based on the distance from the cascade inlet to the leading edge of the blades was $1.2 \times 10^6$, indicating that the boundary layer was fully turbulent at the entrance to the blade row.

2.2 Cascade Blades

The blades were cast as a resin-glass bead mixture in a numerically machined mold. A photograph of the two mold halves is shown in Figure 2.4. The cast blades
had a 4.5 inch span. These were then cut to give the desired span for the experiment. The two blades at the inside and outside of the bend were cut to a 4 inch span. These two blades were attached to the test section at both the tip and the hub.

![Image](image.jpg)

Figure 2.4. Photograph of the numerically machined mold used to cast the Pack-B blades.

The center blade was machined so that blade tips of various geometries or plasma actuator configurations could easily be added to the blade tip. The span of the center blade was less than the width of the test section to produce a tip-gap clearance. The gap height could be varied without removing the blade.

The main blade segment was machined to have a span of 3.42 inches. A short piece of molded blade that had a span of 0.1875 inches was attached to the center blade at the tip using two plastic screws. Plastic shim stock of varying thickness that were numerically machined to the blade shape were placed between the removable tip piece and the main blade to vary the gap height. Clearance gaps of up to 0.08\( c_x \) were possible. The removable tip piece had a milled cavity and two plastic alignment posts that were 0.0625 inches in height. Tip-end pieces with the blade shape were numerically machined out of 0.0625 inch thick epoxy glass board. Holes were drilled in the tip-end pieces to align them on the plastic alignment posts. The tip-end pieces were secured to the blade with silicone.

Surface pressure taps were created near the blade tip by casting in 0.02 inch
TABLE 2.1

LOCATIONS OF BLADE STATIC PRESSURE TAPS

<table>
<thead>
<tr>
<th>Port</th>
<th>Pressure Side</th>
<th>Suction Side</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>x/c&lt;sub&gt;x&lt;/sub&gt;</td>
<td>s/S_p&lt;sub&gt;ps&lt;/sub&gt;</td>
</tr>
<tr>
<td>1</td>
<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>
<td>6</td>
<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>

diameter Tygon tubes. These tubes extended out of the hub of the blade and exited the test section. The pressure taps were located near the tip of the main blade segment. The distance of the pressure taps from the tip varied based on the number of shims that were placed on the blade tip to set the gap height. For example, when the gap height was 0.08c<sub>x</sub>, the taps were 0.06c<sub>x</sub> from the tip. When the gap was 0.05c<sub>x</sub>, the taps were 0.09c<sub>x</sub> from the tip. Table 2.1 lists the chord- and span-wise locations of the pressure taps. There were no pressure taps near the blade tip for the blade mounted active flow control study.
2.3 End Wall Design

Three end wall designs were used at various times during this research. The first was a solid end wall with 30 static pressure ports located under the blade tip. The locations of these static pressure taps with relation to the center blade are shown in Figure 2.5.

![Endwall Static Port Locations](image)

**Figure 2.5.** Locations of the static pressure ports on the end wall.

The second wall had a removable section that was 4 inches by 6 inches in size. This section was centered over the blade and could be filled with a variety of inserts. A clear plexiglass insert was used for flow visualizations. Teflon, Macor, and glass epoxy board inserts were used as dielectrics for wall mounted plasma actuators. The third wall used the same insert but the removable section was centered under the blade trailing edge. The locations of these removable wall sections are illustrated in Figure 2.6.
Figure 2.6. Locations of the removable end wall sections in which inserts were placed.
All of the inserts had a 0.75 inch wide by 0.25 inch thick lip by which they were secured to the test section. An illustration of the different end wall inserts is shown in Figure 2.7. The plexiglass insert was 0.75 inches thick. It mounted flush with the end wall on the inner surface. The Teflon insert was milled out to leave a wall thickness of 0.25 inches. The Teflon wall was used as the dielectric layer for a plasma actuator.

One of these inserts was machined to provide an angled lip to hold a counter-angled insert of either 0.125 inch thick Macor or 0.0625 inch thick glass-epoxy board. These were also used a dielectrics for plasma actuators. When the 0.0625 inch thick glass-epoxy board was used, hot glue was used to secure it in place, and wood braces were used to prevent it from deforming.

![Figure 2.7. Schematic of the different end wall inserts.](image-url)
2.4 Wake Survey with a Five-Hole Probe

A few additional modifications were made to the tunnel to allow investigation of the flow. A hole was drilled near the inlet to allow access of a Pitot probe to obtain upstream static and total pressure readings. In addition, a slot was milled in the hub wall one axial chord length downstream of the blades. This provided access for a United Sensor 0.125 inch diameter cobra head five-hole Pitot probe.

The five-hole Pitot probe was traversed in a 2-dimensional area that was 1.25 inches in the span direction (approximately one-third of the blade span) and 2 inches in the pitch direction (approximately half of the blade pitch). The measurement window is shown in Figure 2.8.

Spatial samples in the measurement window were taken in a ten by ten point grid. In the pitch direction the motion was controlled by a Velmex MA2509-P10 linear traversing mechanism that was driven by an Applied Motion Products HT23-396 stepper motor and PD 2035 step motor controller. Movement in the span direction used a smaller traversing mechanism that was driven by a Micro MO AM1524 stepper motor. The probe was mounted to the smaller traverse which was attached on the larger traverse. The entire traverse assembly was enclosed in a Lexan box to prevent air leakage through the wall slot. Sealed connectors were used for the electrical and pressure tubing connections.

2.5 Flow Control Strategies

2.5.1 Blade Tip Geometries

Squealer tips have historically been formed on blade tips by milling out a cavity to leave a wall around the perimeter of the blade tip. A full squealer tip formed a perimeter around the full perimeter of the blade tip. A partial squealer tip had a perimeter over only a portion of the blade tip. Partial squealer tips have been either
Figure 2.8. Five-Hole Probe wake survey window.

For this research, a partial suction-side squealer was created by attaching a 0.031 inch thick metal strip to the blade tip. The height of the strip was 0.1035 inches, or 0.25\(c_x\). The partial squealer tip followed the edge contour of the blade. It extended over the center 75% of the blade chord. A photograph of the partial suction-side squealer tip on the blade is shown in Figure 2.9.

Winglets of various sizes were created using a numerically controlled LPKF milling machine. The material for these consisted of copper clad, 0.0625 inch thick glass-epoxy boards. The milling machine removed unwanted portions of the copper cladding, with the remaining copper cladding acting as electrodes for the plasma actuators. The winglet sizes were chosen to simulate different blade thickness-to-gap ratios. A photograph of the winglets is shown in Figure 2.10. Their sizes and corresponding thickness-to-gap ratios are listed in Table 2.2.
Figure 2.9. Photograph of Pack-B blade with a partial suction-side squealer tip.

TABLE 2.2

THICKNESS-TO-GAP RATIOS AND CORRESPONDING WINGLET SIZES
AT TWO GAP-TO-CHORD RATIOS

<table>
<thead>
<tr>
<th>$t/g$</th>
<th>winglet size (in) at 0.05 $g/c_x$</th>
<th>winglet size (in) at 0.08 $g/c_x$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.75</td>
<td>N/A</td>
<td>0</td>
</tr>
<tr>
<td>2.83</td>
<td>0</td>
<td>0.350</td>
</tr>
<tr>
<td>3.3</td>
<td>0.100</td>
<td>0.515</td>
</tr>
<tr>
<td>3.7</td>
<td>0.180</td>
<td>0.640</td>
</tr>
<tr>
<td>4.3</td>
<td>0.300</td>
<td>N/A</td>
</tr>
</tbody>
</table>
Figure 2.10. Winglets used to create various thickness-to-gap ratios.

2.5.2 Casing Treatments

Blade mounted techniques, such as squealer tips, are designed to reduce the flow that passes through the tip-gap region under the blade. Casing treatments are based on a different approach for reducing the effects of the tip leakage vortex. The casing treatment methods were investigated to cause the tip leakage vortex to dissipate more quickly. The supposition was that if the vortex dissipated quickly, it would mix with and transfer its momentum to the surrounding flow.

Two basic casing treatment strategies were utilized. The first used vortex generators which were designed to produce vorticity of the opposite sign of the tip leakage vortex. This was done by placing vortex generators on the casing wall on the suction side of the center blade. The vortex generators consisted of rectangular pieces of 0.005 inch thick shim material that was bent to form an “L” shape. The
vortex generators were 0.24 inches wide. Two tab heights of 0.12 and 0.20 inches were used. A schematic of the vortex generators showing their basic shape is shown in Figure 2.11.

![Figure 2.11. Schematic of the passive vortex generators.](image)

Two arrangements of the vortex generators were used. In one arrangement, the vortex generator was placed on the end wall across from the trailing edge of the blade, and oriented to be at a 57° angle with respect to the blade edge. This orientation placed it at a 30° angle with respect to the direction of the flow through the tip-gap. The other configuration placed the vortex generator 0.75 inches upstream of the blade trailing edge. The orientation was the same as the tab at the trailing edge. An illustration of the two vortex generator placements is shown in Figure 2.12.

The second casing treatment investigated was wall roughness. This involved bonding sandpaper to a wall insert. The insert was then placed in the opening directly under the center blade. Roughness grits of 120 and 280 were used. The gap heights were kept the same as the smooth wall cases.
2.5.3 Plasma Actuators

Active flow control was implemented using single dielectric barrier discharge (SDBD) plasma actuators. As described in Chapter 1, the actuator consisted of two electrodes separated by a dielectric material. An illustration is shown in Figure 2.13. One electrode was exposed to air, while the second was covered by the dielectric layer. A high voltage a.c. was supplied to the electrodes. When the amplitude was large enough, the air over covered electrode ionized. The ionized air in the presence of the electric field produced by the electrode geometry results in a body force vector that acts on the neutral air. The body force vector direction depends on the electrode geometry. In the arrangement in Figure 2.13 the body force vector is towards the covered electrode.

The LPKF milling machine that was used to fabricate the winglets was also used to fabricate the blade mounted plasma actuators. The copper cladding was milled off of both sides, leaving the two electrodes. The exposed electrode was located
Figure 2.13. General schematic of a SDBD plasma actuator with asymmetric electrodes.

at approximately the same location as the suction-side squealer tip. The covered electrode extended from that location toward the pressure-side of the blade tip. This is shown in the photograph in Figure 2.14. In the arrangement, the body force vector opposes the flow that passes under the blade through the tip-gap.

Figure 2.14. Electrode configuration used for the blade mounted plasma actuator.

The casing treatment plasma actuators were fabricated using the Teflon, Macor,
or glass-epoxy substrates. The plasma actuators using the Teflon and Macor substrates used copper foil tape for the electrodes, while the plasma actuators made with the glass-epoxy substrates were copper clad and fabricated using the LPKF milling machine with the above described technique. The electrode arrangements used for these plasma actuators are discussed in detail in Chapter 5. Each was designed to mimic a passive control technique discussed in that chapter.

Two electronics set-ups were used to power the plasma actuators. The electronics which supplied the a.c. amplitude to the “active plasma suction-side squealer” actuator are diagramed in Figure 2.15. Two Stanford Research Systems DS335 function generators provided a timing circuit with a triangle waveform at 16 kHz and a square waveform at a lower frequency. The timing circuit was powered by an Agilent E3630A d.c. power supply that was set at 5.27 V d.c. The signals went to a low power amplifier built by Post which had a variable gain that was used to set the duty cycle of the signal. The signal was then sent to a high power amplifier, also built by Post, which increased the output power to drive a step-up transformer. This amplifier was powered by a ±48 volt d.c. power supply. From here, the signal was viewed via a LeCroy Waverunner CT342 oscilloscope and sent through 2 resistors to the step-up transformer. The transformer had a 120:1 winding ratio. This signal was supplied to the exposed electrode while the covered electrode was grounded.

The electronics which supplied the a.c. amplitude to the plasma actuator casing treatments are diagramed in Figure 2.16. A triangle waveform was supplied by an Agilent 33220 function generator. The signal was monitored by a LeCroy Waverunner LT342 oscilloscope and input into two Crown Xti400 100W audio amplifiers. These were used to power two step up transformers with winding ratios of 180:1. The transformers delivered two signals 180 degrees out of phase. One went to the covered electrode and the other to the exposed electrode. This doubled the peak-to-
Figure 2.15. Plasma actuator electronics used for blade mounted flow control.
peak voltage across the actuator. The signal from the transformers was monitored via a high voltage probe connected to the LeCroy oscilloscope.

Figure 2.16. Plasma actuator electronics configuration used for wall mounted flow control.
A variety of measurement techniques were utilized to investigate the tip-gap flow for this research. Wall pressure taps and end wall flow visualization provided an understanding of the flow on the end wall. Blade static pressure taps and blade tip flow visualization provided insight into the flow under the blade tip. Five-hole Pitot probe measurements provided an understanding of the tip leakage vortex behavior in the blade row wake. The techniques used to acquire and analyze the data presented in this document are detailed in this chapter.

3.1 Five-Hole Pitot Probe Calibration

The majority of quantitative data was discerned from the downstream flow surveys obtained using a five-hole Pitot probe. The five-hole probe has a central total pressure port and four surrounding static pressure ports that are on a 45 degree bend that is angled away from the center port, as shown in Figure 3.1. When calibrated properly, measurements from the five ports provide an indication not only of the local flow speed, but also the local flow direction, providing the velocity vector at the measurement point. The method used to calibrate the five-hole probe came from Bryer and Pankhurst [54]. The technique is briefly summarized here.
The probe was placed in a steady, straight flow at a known velocity. Pressure values were then obtained from all five ports. A separate Pitot-static probe was placed in the air stream and used as a calibration reference. The five-hole probe was rotated about the measurement tip in both directions. The pressure from the five holes were then recorded for each of the pitch (\( \alpha_1 \)) angles and yaw (\( \alpha_2 \)) angles. The pressure data was then represented by the two angle functions, \( f(\alpha_1) \) and \( f(\alpha_2) \), that are defined as

\[
f(\alpha_1) = \frac{p_3 - p_1}{p_5 - p_m}, \tag{3.1}
\]

and,

\[
f(\alpha_2) = \frac{p_2 - p_4}{p_5 - p_m}. \tag{3.2}
\]

The two angle functions represent a 2-dimensional calibration region. An example
Figure 3.2. Calibration space used to determine pitch and yaw angles from the five-hole Pitot probe pressure measurements.

is shown in Figure 3.2. This is used to determine the local flow angle from the five-hole probe measurement. The static pressures are averaged to proved $p_m$, which is defined as

$$p_m = \frac{p_1 + p_2 + p_3 + p_4}{4}. \quad (3.3)$$

Based on the angular calibration, static and dynamic flow coefficients, $S_p$ and $Q_p$, were calculated. The static coefficient is given by

$$S_p = \frac{H - p_5}{p_5 - p_m}, \quad (3.4)$$

where

$$H = \frac{1}{2} \rho U^2, \quad (3.5)$$
Figure 3.3. Contours of the static pressure coefficient, $S_p$, as a function of 2-dimensional flow angle.

where $U$ is the velocity obtained from the calibration reference Pitot-static probe. The dynamic flow coefficient, $Q_p$ is defined as

$$Q_p = \frac{p_5 - p_m}{H}. \quad (3.6)$$

Contours of these two coefficients are shown in Figures 3.3 and 3.4.

Following the calibration, the dynamic pressure and flow angle were found from the five-hole probe measurements by solving for $H$ from Equations 3.4 and 3.6. The uncertainty in the calibration was $\pm 3\%$ for flow angles within $\pm 25$ degrees. This was determined to be suitable for these studies.

3.2 Pressure Data Instrumentation and Acquisition

All of the blade tip and end wall static pressure taps were connected to a forty port 4839-2145 Scanivalve port selector via Tygon tubing. The Scanivalve was con-
Figure 3.4. Contours of dynamic pressure coefficient, $Q_p$, as a function of 2-dimensional flow angle.

trolled by a SCANCO CLR2152-56 solenoid controller which received input from the data acquisition computer. The pressure output from the Scanivalve was connected to a Validyne DP-103 pressure transducer equipped with an 8-28 diaphragm. This diaphragm had a maximum pressure range of 1.6 in Hg. The pressure transducer was proceeded by a Validyne CD-23 carrier demodulator. This provided a voltage output that was linearly proportional to the pressure differential across the diaphragm. The output from the pressure transducer was low pass filtered at 100 Hz using a Stanford Research Systems SR650 Dual channel filter. This was monitored using a LeCroy Waverunner LT342 Oscilloscope and acquired and stored with the data acquisition computer. The range of voltages from the pressure transducer was between $\pm$ 5 volts. This was within the A/D maximum voltage limits to minimize quantization error. A diagram of this instrumental set-up is shown in Figure 3.5.

For the casing treatment studies, the five-hole Pitot probe used five separate Vali-
dyne DP-103 pressure transducers so that pressure measurements could be acquired in a shorter time. For these measurements, the total pressure port was connected to a transducer with an 8-32 diaphragm. The other four ports were connected to transducers with 8-28 diaphragms. These transducers were connected to a Validyne CD280-dual carrier demodulator. The output was acquired by the data acquisition computer. The low pass filter was not used for these measurements.

The data acquisition computer had an AMD 3500 939-pin 64 bit processor, PD2-MF-16-400/14H acquisition card and a Powerdaq 16 bit A/D converter card. The sensitivity of the A/D input was 0.2 mv. The operating system was Linux. All of the time series acquisition codes were written in C.

The acquisition codes were all modified from the code “TestPressure.c” that was originally written by Junhui Huang[55]. It was later modified by Travis Douville[9] and again by the present author for the current experiment. One major software modification added A/D gain control to achieve more accurate voltage measurements. This modification was titled “TestPressuregain.c”. Extensive software modifications were also made to incorporate control of the traversing mechanisms. This code was titled “traverse.c”.

3.3 Pressure Data Analysis

The blade static tap measurements were all referenced to the upstream static pressure, $P_{su}$, which was connected to the reference side of the pressure transducer so that the transducer voltage at the $i^{th}$ pressure port corresponded to $P_i - P_{su}$. With the pressure data recorded in this way, it was very easy to obtain the pressure coefficients defined as

$$c_p(i) = \frac{P_i - P_{su}}{P_{tu} - P_{su}},$$

(3.7)
Figure 3.5. Static pressure data acquisition instrumentation.
where $P_{tu}$ is the upstream total pressure.

The easiest way to view the static pressure on the blade was in plots of pressure coefficient as a function of the non-dimensional location with respect to axial chord, $x/c_z$. Integrating the static pressure coefficients around the blade then gave provided an indication of blade tip loading. The static pressure taps on the end wall were also non-dimensionalized using Equation 3.7.

As mentioned in the Chapter 1, the entropy production in the cascade wake is a good indication of loss, and the total pressure loss coefficient is linearly proportional to the entropy production for low Mach number flows. The total pressure loss is defined as

$$c_{pt} = \frac{P_{tu} - P_{te}}{P_{te} - P_{se}},$$

where $P_{tu}$ is the total pressure measured upstream of the cascade, $P_{te}$ is the total pressure measured downstream of the cascade, and $P_{se}$ is a static exit pressure calculated from the other four ports of the five-hole probe.

A mass-average for the total pressure loss coefficient was calculated using the method of Yamamoto[56, 57]. The definition of the mass averaged total pressure loss coefficient is

$$C_{PT} = \frac{\sum_k \sum_j (c_{pt} A U)_{k,j}}{\sum_k \sum_j (A U)_{k,j}},$$

where $k$ and $j$ are pitchwise and spanwise counters, $A$ is the area, and $U$ is the velocity. In initial investigations the entire surveyed window was mass-averaged. However, after a few different cases were investigated, it was noted that as the size and strength of the tip-leakage vortex varied in different cases, the amount of the passage vortex that was observed in the measurement region also varied. As a result, an increase in loss might be observed not because the loss increased, but because...
more of the loss due to the passage vortex was observed. To resolve this issue, three
different techniques were used to restrict the mass-averaging to similar windows. A
plot of the $C_{PT}$ value obtained with each technique for 8 cases is shown in Figure
3.6. The first technique mass-averaged over areas of the same mass flow rate. To
obtain such areas, a desired mass flow rate was chosen, then the area around the
location of highest pressure loss inside the tip leakage vortex was expanded until the
mass flow rate inside the area was the desired value. The pressure loss inside that
area was then mass-averaged. The second technique set a square area based on the
size of the smallest tip-leakage vortex. This area was centered on each tip-leakage
vortex based on the location of maximum pressure loss and mass-averaged values
of the same area were compared. Unfortunately, with both of these techniques,
occasionally part of the passage vortex is included, and differing amounts of the free
stream flow are averaged in. Finally, the pressure coefficients inside the area defined
by the $-\lambda_2$ contour were mass-averaged and compared. This allowed an averaging
of the pressure loss associated with the tip-leakage vortex without consideration
of other structures. The final technique is what was used to evaluate every case
presented in this document.

Streamwise vorticity, $\omega_x$, was calculated based on the velocity vectors obtained
with the five-hole Pitot probe. This was calculated in discrete form as

$$\omega_x = \frac{\partial w}{\partial y} - \frac{\partial v}{\partial z},$$

(3.10)

where $w$ is the pitchwise component of velocity and $v$ is the spanwise component of
velocity. Second order central differencing, with Taylor series approximations at the
boundaries, was used to approximate the derivatives at each measurement point.
The stream-wise vorticity was then normalized by the upstream freestream velocity,
$U$ and the axial chord, $c_x$. 53
Figure 3.6. Results of mass averaging over different areas.

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

(3.11)

The \(-\lambda_2\) method of Jeong et al.\cite{58} was used to identify coherent vortices in the measurement region. This method identified vortices as regions bounded by contours of negative values of the second eigenvalue of the velocity gradient tensor. A closed constant level contour of a slightly negative eigenvalue is expected to enclose the outer boundary of a vortex.

A combination of velocity vectors, total pressure loss contours, vorticity magnitude contours, and the \(-\lambda_2\) contours were used to provide an indication of the locations and strengths of vortical structures that formed in the wake of the blades in the cascade. A particular metric of merit used in evaluating the flow control was the loss coefficient associated with the tip leakage vortex.
3.4 Flow Visualization Techniques

Flow visualization was performed to gain a general understanding of the flow around the blades in the cascade near the end wall. Two methods were used. Each was chosen because they were expected to work well at the high tunnel speeds (about 250 ft/s at Re = 500K).

3.4.1 Ink Dot Technique

The first flow visualization technique was the ink dot technique of Langston[59] and Aunapu[60]. This technique was used to visualize the flow on the underside of the blade tip. An outline of the method is as follows:

1. White contact paper was mounted on the blade tip.
2. An extra-fine point permanent ink marker was used to make a pattern of dots on the contact paper.
3. When the permanent ink dots dried, an air brush was used to spray a very thin coat of methyl salicylate (also known as oil of wintergreen) over the dots.
4. The test section was then sealed and the tunnel air flow was quickly brought to speed.
5. The ink dots were allowed to move along the surface.
6. The tunnel was turned off, and the contact paper was carefully removed to preserve the ink traces.
7. For each case, the flow visualization was performed three times. The scanned patterns were then digitally averaged.

3.4.2 Surface Oil Visualization

A second flow visualization technique was used to visualize the flow on the end wall. This used a mixture of silicone oil with viscosity 10,000 Cs, Oleic acid and Titanium Dioxide in a volume ratio of 15:5:1, respectively. The titanium dioxide particles reflect light very well, and made the streak lines on the wall very visible when illuminated by an incandescent light source. Lower viscosity silicone oil was also investigated but was found to flow off of the wall too quickly.
The approach was to brush a thin, uniform layer of the oil mixture on the end wall. The tunnel was then brought up to speed. The oil mixture on the wall was then transported by the local shear stress. This took the form of wall streak lines. Because the end wall is clear, the images could be recorded while the tunnel was running. This provided a clear view of the flow patterns on the end wall in relation to the blades.
CHAPTER 4

BLADE MOUNTED FLOW CONTROL

As stated in the objectives in Chapter 1, the aim of this research is to further document the tip leakage vortex behavior, and to use this insight to examine active flow control strategies to reduce losses associated with the tip-gap flow. This chapter presents results for blade-mounted passive flow control that involved pressure-side winglets and suction-side squealer tips. In addition, it presents results for active flow control using plasma actuators positioned in a manner similar to the suction-side squealer tips.

4.1 “Thick” vs. “Thin” Blade Regimes

As described in the Chapter 1 Denton [8] proposed that the behavior of the flow in the tip-gap for both compressor and turbine blades changed when the thickness-to-gap ratio was large. Denton found that if the blade thickness was more than about four times the tip-gap height, the flow behaved as if the blade were “thick”. The flow topology of a “thick” blade consisted of a separation and reattachment line that formed on the pressure-side corner of the blade tip, as shown in Figure 1.9 in Chapter 1. According to Denton, if the blade were “thin” the flow that separated from the pressure-side blade edge never reattached on the blade tip. In Douville’s initial investigations [9] into the effect of Reynolds number and gap height on the tip-
gap flow, it was found that the mass averaged total pressure loss coefficient of a blade with a 4% gap-to-chord ratio was greatly reduced by the addition of a squealer tip. However, when the gap-to-chord ratio was increased to 5% by increasing the tip-gap height, the squealer tip had essentially no effect on the total pressure loss coefficient. This is shown in Figure 4.2. Douville et al.\cite{11} proposed that the “thick” or “thin” blade behavior defined by Denton determined the effectiveness of the squealer tip. When the blade was “thin,” the flow separated from the blade without reattaching and the squealer tip was ineffective. However, when the blade was “thick,” the flow reattached to the blade tip and the squealer tip was effective.

Figure 4.1. Flow under a blade tip determining the effectiveness of a squealer tip based on “thick” or “thin” blade behavior as proposed by Douville et al.\cite{11}.

A modification of Figure 1.9 is shown in Figure 4.1 to illustrate the effect of a
squealer tip on “thick” and “thin” blades. Douville proposed that the threshold between “thick” and “thin” blade behavior at a thickness-to-gap ratio of 3.5, which is slightly less than the value of 4 that Denton proposed.

Figure 4.2. Effect of gap-to-chord ratio on mass-averaged total pressure loss coefficients for blades with flat tips and squealer tips at Reynolds numbers of 100,000 and 500,000[9].

The potential dependence of passive flow devices on turbine blades[35, 26, 36] motivated the research of independently varying blade thickness and tip-gap height discussed in this chapter. The blade thickness was effectively varied by the addition of pressure-side winglet. Two gap-to-chord ratios of 5% and 8% were investigated. For the 5% gap-to-chord ratio, winglets were added to provide thickness-to-gap ratios were 2.83, 3.3, 3.7, and 4.3. For the 8% gap-to-chord ratio, thickness-to-gap ratios of 1.75, 2.83, 3.3, and 3.7 were investigated. Details of this experimental setup were described in Chapter [2].
4.1.1 Flow Documentation

As mentioned in Chapters 2 and 3, the flow was documented using a five-hole Pitot probe that was located one axial chord downstream of the blade row. The pressure measurements obtained by the five-hole probe were processed to yield contours of vorticity magnitude and pressure loss coefficients. Vorticity contours for the flat tip blade with no winglet at a gap-to-chord ratio of 5% are shown in Figure 4.3. The red regions correspond to negative (counter-clockwise) vorticity and the blue regions correspond to positive (clockwise) vorticity. The dashed lines are the zero-level contours of the $-\lambda_2$ criterion discussed in Chapter 3, which are expected to enclose coherent vortices. Two vortices of opposite rotation are identified by this criteria. The vortex with positive vorticity on the right side of the measurement region is the tip leakage vortex, of primary interest to these studies. The vortex with negative vorticity on the left side of the measurement region is the passage vortex. The arrows are the secondary velocity vectors.

The $-\lambda_2$ contours in Figure 4.3 are observed to mark the locations of highest vorticity and magnitude. It is therefore presumed that they do enclose vortex cores. The remaining vorticity plots are not included in this chapter, but are provided for reference in Appendix A. Subsequently, the zero-level $-\lambda_2$ contours will be used to mark the locations of the vortical structures in the analysis of pressure loss coefficient contours.

Figure 4.4 shows total pressure loss coefficient contours for the flat tip blade with no winglet at a gap-to-chord ratio of 5%. The red regions correspond to regions of high pressure loss. These regions fall within the zero-level $-\lambda_2$ contours, indicated by the black lines. This indicates the large pressure loss is associated with the tip leakage and passage vortices.

The pressure loss coefficient contours inside the $-\lambda_2$ contour from Figure 4.4
Figure 4.3. Vorticity contours in the wake of the blade row measured one axial chord downstream. The gap-to-chord ratio is 5% and thickness-to-gap ratio is 2.83 (no winglet).
are shown in Figure 4.5. These are the pressure loss coefficients associated with
the tip leakage vortex. These values were mass averaged using Equation 3.9 for a
quantitative comparison of the different cases.

Figure 4.4. Total pressure loss coefficient contours in the wake one axial chord
downstream of the blade row for a flat tip blade at a gap-to-chord ratio of 5% and
thickness-to-gap ratio of 2.83 (no winglet).

Pressure distributions measured at the blade tip that correspond to the condi-
tions in Figure 4.4 are shown in Figure 4.6. For reference, the red curve is shown
which corresponds to a blade with no gap. The 5% gap-to-chord case primarily
deviates from the no-gap case on the suction side of the blade. The portion of the
blade for \( x/c_x \leq 0.6 \) shows the influence of the flow passing through the gap under
the blade. The portion for \( x/c_x > 0.6 \) is due to the tip leakage vortex that separates
from the blade suction-side near the trailing edge. The tip leakage vortex produced
a lower pressure in its core that increased the blade tip loading in the region.
Figure 4.5. Total pressure loss coefficient contours inside the zero-level -$\lambda_2$ contour corresponding to the tip leakage vortex. Measurements were made one axial chord downstream of the blade row for a flat tip blade at a gap-to-chord ratio of 5% and thickness-to-gap ratio of 2.83 (no winglet).
Figure 4.6. Blade tip pressure distribution for a flat tip blade at a gap-to-chord ratio of 5% and thickness-to-gap ratio of 2.83. The red curve corresponds to a no-gap blade and is included for reference.

Pressure coefficient contours provided by pressure taps on the end wall across from the blade with the same blade thickness and gap height conditions as Figure 4.4 and 4.6 are shown in Figure 4.7. The left plot shows the pressure contours obtained using all 30 wall pressure taps. The right plot shows the pressure contours just inside the region directly across from the blade. The negative level contours indicate where the flow is passing under the blade at velocities that are larger than the reference at the cascade inlet. The largest velocities occur in the region of $x/c_x \geq 0.6$. This corresponds to the region where the majority of the flow exits the gap.

4.1.2 Results: Wake Pressure Contours

Contour plots of the total pressure loss coefficient in the blade wake for all of the thickness-to-gap ratios at a 5% gap-to-chord ratio are shown in Figure 4.8. The flat tip cases are shown in the top row and the squealer tip cases are shown in the bottom row. The effective thickness of the blade increases from left to right. Note,
Figure 4.7. End wall static pressure coefficient contours for a flat tip blade at a gap-to-chord ratio of 5% and a thickness-to-gap ratio of 2.83 (no winglet).

as depicted in Figure 2.8, the trailing edge of the blade is along the top of each plot and the end wall corresponds to the right edge.

The highest loss coefficient contours in Figure 4.8 (regions of yellow or red) are the tip leakage and passage vortices. Varying amounts of the passage vortex are seen on the left side of the plots. The tip leakage vortex is the primary vortical structure located to the right on each plot, nearest the location of the gap. Changes in the magnitudes and center locations of the maximum loss coefficient are observed to occur with changes in the thickness-to-gap ratio for both flat tip blades and squealer tip blades.

The mass-averaged total pressure loss coefficient associated with the tip leakage vortex was determined for each case by mass averaging the total pressure loss coefficients inside the $-\lambda_2$ contour. A plot of these values for the eight cases shown in Figure 4.8 are shown in Figure 4.9. The star symbols correspond to the flat tip blades and the circle symbols correspond to the blades with squealer tips. The curves between the symbols were added to aid visualization of the trends of the
Figure 4.8. Total pressure loss coefficient contours in the wake measured one axial chord downstream of the blade row for different winglets producing different thickness-to-gap ratios at a gap-to-chord ratio of 5%.
data. The bars indicate the repeatability of the results. For the flat tip blades, increasing the blade thickness-to-gap ratio reduced the loss coefficient. Compared to the smallest thickness-to-gap ratio, the larger ratios of 3.3, 3.7, and 4.3 reduced the loss coefficient by 6.2%, 7.8% and 16.1% respectively.

The addition of the squealer tips on the blades further reduced the loss coefficients. For the thickness-to-gap ratio of 2.83, the improvement over the flat-tip blade was 17.2%. Similarly, improvements of 18.5%, 15%, and 6% were obtained by adding the squealer tip to blades with thickness-to-gap ratios of 3.3, 3.7, and 4.3. The results obtained with the squealer tip suggest that there is an optimum thickness-to-gap ratio for squealer tip effectiveness. This occurs at a thickness-to-gap ratio of about 3.3.

![Graph](image)

Figure 4.9. Mass averaged total pressure loss coefficients associated with the tip leakage vortex at a gap-to-chord ratio of 5%.

Contour plots of the total pressure loss in the wake for all thickness-to-gap ratios investigated at a gap-to-chord ratio of 8% are shown in Figure 4.10. The flat tip
cases are again shown in the top row, and the squealer tip cases are shown in the bottom row. As before, thickness-to-gap ratio increases from left to right.

Figure 4.10. Total pressure loss coefficient contours in the wake measured one axial chord downstream of the blade row for different winglets producing different thickness-to-gap ratios at a gap-to-chord ratio of 8%.

The highest loss coefficients (red regions) are again associated with the tip leakage vortex. The passage vortex was either outside of the measurement region or not present for these cases. The magnitude and center location of the maximum loss coefficient are again observed to vary with the thickness-to-gap ratio, and the addition of the squealer tip.

The mass-averaged total pressure loss coefficients associated with the tip leakage vortex are plotted in Figure 4.11 for the cases shown in Figure 4.10. The square symbols correspond to the flat tip blades and the triangle symbols correspond to the blades with squealer tips. The flat tip blades appear to have a minimum loss
coefficient at a thickness-to-gap ratio of approximately 3 at this larger gap-to-chord ratio. The squealer tip, on the other hand, improves with increasing thickness-to-gap ratio. As a result, there was virtually no improvement with the squealer tip until a thickness-to-gap ratio of approximately 3, where the loss coefficient of the flat tip blade began to increase.

![Graph](image)

Figure 4.11. Mass averaged total pressure loss coefficients associated with the tip leakage vortex at a gap-to-chord ratio of 8%.

The mass averaged total pressure loss associated with the tip leakage vortex for both gap-to-chord ratios at all of the thickness-to-gap ratios for both the flat and squealer tip cases is plotted in Figure 4.12. This indicates that the trend for the 8% gap-to-chord ratio blade with the squealer tip is comparable to that of the 5% gap-to-chord ratio flat tip blade. That is, for the flat tip blade, there is a blade behavior that is dependent on the thickness-to-gap ratio and independent of the gap-to-chord ratio. The squealer tip is most effective with the 5% gap-to-chord ratio blade, and less sensitive to changes in thickness-to-gap ratio than to changes
in gap-to-chord ratio. This indicates that the original interpretation by Douville et al. [11] that the effectiveness of the squealer tip depended on the thickness-to-gap ratio was not correct.

4.1.3 Results: Blade Tip Pressure Distributions

Pressure distributions at the blade tip at all of the thickness-to-gap ratios investigated at a gap-to-chord ratio of 5% are plotted in Figure 4.13. Again for reference, the solid red curve corresponds to the pressure distribution at the mid-span of the blade with the tip-gap closed. The measured distribution again agrees well with the ideal distribution on the pressure-side of the blade. The differences appear on the suction-side. The distributions for the different thickness-to-gap ratios roughly overlay each other except on the trailing edge for \( x/c_x > 0.6 \). This is the region where the tip leakage vortex influences the pressure distributions. The magnitude of the
pressure coefficient values should correlate with the strength of the tip leakage vortex. The results indicate that the strength of the vortex is largest with the smallest thickness-to-gap ratio. The strength of the tip leakage vortex then decreases with increasing thickness-to-gap ratio.

Comparisons of the blade tip pressure distributions with flat and squealer tip blades at all thickness-to-gap ratios for a gap-to-chord ratio of 5% are shown in Figure 4.14. The flat tip cases are represented by circles and the squealer tip cases are represented by squares. For each thickness-to-gap ratio, the squealer tip has two effects. The first is to decrease the suction-side pressure at $x/c \leq 0.6$. This indicates that there is less flow under the blade when the squealer tip is installed. The second effect is an increase in the suction-side pressure for $x/c > 0.6$. This indicates that the squealer tip reduced the strength of the tip leakage vortex. These results were consistent for all of the thickness-to-gap ratios investigated.

Figure 4.13. Pressure distributions at the blade tip for all flat tip blades at all of the thickness-to-gap ratios investigated at a gap-to-chord ratio of 5%.
Figure 4.14. Pressure distributions at the blade tip for all of the thickness-to-gap ratios investigated at a gap-to-chord ratio of 5%.
Figure 4.15. Blade tip loading as a function of thickness-to-gap ratio for flat and squealer tip blades at a gap-to-chord ratio of 5%.
A measure of blade tip loading was obtained by integrating the tip $c_p$ distributions around the blade. The tip loading for all of the cases shown in Figure 4.14 are shown in Figure 4.15. For both the flat blade and the blade with the squealer tip, the tip loading has a minimum at a thickness-to-gap ratio of approximately 3.5. This thickness-to-gap ratio corresponds to the “thick”/“thin” blade regime change proposed by Denton[8] and Douville et al.[11].

Pressure distributions at the blade tip for the 8% gap-to-chord ratio for all of the thickness-to-gap ratios are plotted in Figure 4.16. In contrast to the 5% gap-to-chord ratio, changes in the pressure distribution now occur on the pressure-side of the blade for all of the different thickness-to-gap ratios. There is less variation on the suction-side between the flat tip cases, except near the trailing edge. At the trailing edge, larger suction pressures are seen with increasing thickness-to-gap ratio. This indicates a stronger tip leakage vortex with larger thickness-to-gap ratios.

Comparisons of the pressure distributions for flat and squealer tip blades at each thickness-to-gap ratio investigated at a gap-to-chord ratio of 8% are shown in Figure 4.17. The flat tip cases are represented by circles and the squealer tip cases are represented by squares. For each thickness-to-gap ratio, the flat tip and squealer tip cases roughly overlay each other except near the trailing edge on the suction side ($x/c_x > 0.6$). This is different from the 5% gap-to-chord ratio case where the squealer tip changed the entire distribution on the suction-side of the blade tip. Near the trailing edge, the differences between the flat tip and squealer tip cases is largest for the larger thickness-to-gap ratios. At the smaller thickness-to-gap ratios, less than 3, the squealer tip has no apparent effect on the blade tip loading.

The blade tip loading for the cases in Figure 4.17 are shown in Figure 4.18. The effect of thickness-to-gap ratio on the blade tip loading is substantially different for
Figure 4.16. Pressure distributions at the blade tip for flat tip blades at all of the thickness-to-gap ratios investigated at a gap-to-chord ratio of 8%.

Figure 4.17. Pressure distributions at the blade tip for all of the thickness-to-gap ratios investigated at a gap-to-chord ratio of 8%.
Figure 4.18. Blade tip loading as a function of thickness-to-gap ratio for both flat tip and squealer tip blades at a gap-to-chord ratio of 8%.
the 8% gap-to-chord ratio compared to the 5% gap-to-chord ratio results. Although there is a good deal of scatter in the results, the trend for the flat tip blade is approximately linear increase with increasing thickness-to-gap.

The addition of the squealer tip increased the blade tip loading for the lowest thickness-to-gap ratios, and decreased the blade tip loading for the larger thickness-to-gap ratios. This was observed in the pressure distributions in Figure 4.17 and entirely relates to the strength of the tip leakage vortex. For the smaller thickness-to-gap ratios, the squealer tip was not effective in reducing the tip leakage vortex strength.

4.1.4 Results: End Wall Pressure Contours

Pressure coefficient contours on the end wall across from the blade tip for a thickness-to-gap ratio of 2.83 and a gap-to-chord ratio of 5% are shown in Figure 4.19. The contours for the flat tip blade are shown on the left and for the squealer tip blade on the right. The pressure contours inside the area directly under the blade are shown below the contours for the entire area surveyed. For the flat tip, an isolated region of negative static pressures is observed directly under the blade in the region of $x/c_x \geq 0.6$. This corresponds to the region where high velocity air passes under the blade and feed the tip leakage vortex. The squealer tip at this condition effectively suppresses the air from passing under the blade. This is evident from the significantly lower suction pressures under the blade with the squealer tip.

The end wall pressure distributions for thickness-to-gap ratios of 3.3, 3.7 and 4.3 for a gap-to-chord ratio of 5% are presented in Figures 4.20, 4.21 and 4.22. The negative pressure region under the blade was observed for all of the flat tip cases. The extent of this region decreased slightly with increasing thickness-to-gap ratio. As before, the suction pressure was significantly suppressed by the addition of the
squealer tip for all of the thickness-to-gap ratios.

The averaged pressure coefficients for the cases in Figures 4.19 - 4.22 are presented in Figure 4.23. The flat tip cases are represented by stars and the squealer tip cases are represented by circles. Curves were added only to aid in visualizing the trends. The top plot corresponds to the average over the entire measurement area, while the bottom plot corresponds to the average inside the region directly under the blade. Both of these averages show similar trends. This apparently indicates a maximum suction pressure under the blade at a thickness-to-gap of approximately 3.5. This would correspond to where the velocity of the flow under the blade is the highest. The addition of the squealer tip reduced the suction pressure. The
Figure 4.20. End wall static pressure coefficient contours for flat and squealer tip blades at a thickness-to-gap ratio of 3.3 and a gap-to-chord ratio of 5%. 
Figure 4.21. End wall static pressure coefficient contours for flat and squealer tip blades at a thickness-to-gap ratio of 3.7 and a gap-to-chord ratio of 5%.
Figure 4.22. End wall static pressure coefficient contours for flat and squealer tip blades at a thickness-to-gap ratio of 4.3 and a gap-to-chord ratio of 5%.
Figure 4.23. Area averaged pressure coefficient on the end wall as a function of thickness-to-gap ratio at a gap-to-chord ratio of 5%. 

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(a) entire area

(b) blade area
effectiveness of the squealer tip was not effected by the thickness-to-gap ratio.

Pressure coefficient contours on the end wall for the 8% gap-to-chord ratio cases are shown in Figure 4.24 - 4.27. Again, the pressure contours for the flat tip blade are shown on the left and the pressure contours for the squealer tip blade are shown on the right. Contours inside the area directly under the blade are shown below contours from the entire measurement area. At this gap-to-chord ratio, the isolated region of negative pressure coefficient that occurs for the flat tip cases is greatly reduced in size and strength compared to the smaller gap-to-chord cases. This is attributed to the extreme gap size which results in less acceleration of the flow through the tip-gap.

Figure 4.24. End wall static pressure coefficient contours for flat and squealer tip blades at a thickness-to-gap ratio of 1.75 and a gap-to-chord ratio of 8%.
Figure 4.25. End wall static pressure coefficient contours for flat and squealer tip blades at a thickness-to-gap ratio of 2.83 and a gap-to-chord ratio of 8%.
Figure 4.26. End wall static pressure coefficient contours for flat and squealer tip blades at a thickness-to-gap ratio of 3.3 and a gap-to-chord ratio of 8%.
Figure 4.27. End wall static pressure coefficient contours for flat and squealer tip blades at a thickness-to-gap ratio of 3.7 and a gap-to-chord ratio of 8%.
Figure 4.28. Area averaged pressure coefficient on the end wall as a function of thickness-to-gap ratio at a gap-to-chord ratio of 8%.
The averaged wall pressure coefficients for these cases are plotted in Figure 4.28. Again, the flat tip cases are represented by squares and the squealer tip cases are represented by triangles. Lines were again added to aid in visualizing the trends. The top plot is averaged over the entire measurement area and the bottom plot is the average of the area directly under the blade. Unlike the smaller (5%) gap-to-chord case, averaging over the entire area produced different trends compared to averaging the area directly under the blade. The difference appears to be due to a more dominant effect of the tip leakage vortex with a larger gap-to-chord ratio. The pressure field produced by the tip leakage vortex is felt on the wall on the suction-side of the blade and further downstream.

4.1.5 Results: Blade Tip Surface Flow Visualization

The ink dot surface flow visualization on the blade tips provided further insight into the tip-gap flow features. The visualization technique was described in detail in Chapter 3. The tip visualization records for the 5% gap-to-chord ratio are shown in Figure 4.29. The visualizations for the 8% gap-to-chord ratio are shown in Figure 4.30. The original images are shown above images to which an interpretation of the flow vectors have been added. The tracks of ink dots are assumed to follow the local shear stress on the end of blade as a result of the flow moving through the tip-gap. Three independent records of ink dot visualization were taken for each case. These were scanned digitally and averaged.

Viewing the surface visualization records, there are a number of obvious features. First, there is a well defined reattachment line at which the shear direction bifurcates, with some vectors pointed towards the suction-side, and others pointed towards the pressure-side of the blade. The arrows pointing towards the pressure-side suggest a flow recirculation from the reattachment point to the edge of the
pressure-side of the blade. In addition, there is a well delineated portion of the blade at the leading edge where the shear vectors indicate that the passage flow traverses through the gap without separating from the blade.

Given these characteristic features of the surface visualization at both gap-to-chord ratios, two conclusions can be drawn. First, the region where the passage flow traverses through the gap without separating from the blade moves further back along the chord of the blade as the thickness-to-gap ratio increases. Therefore, the chordwise extent of the reattachment line shortens as the thickness-to-gap ratio increases. Second, the chordwise extent of the reattachment line shortens as the gap-to-chord ratio increases, even for the same thickness-to-gap ratio.

These conclusions disagree with Denton’s proposal\[8\] that the difference between “thin” and “thick” blade behavior is due to a flow separation from pressure-side edge of the blade tip that reattaches to “thick” blades but does not reattach to “thin” blades. The present flow visualization indicates that if the flow under the blade separates from the pressure-side edge, it reattaches. The chordwise location of the flow separation and reattachment, however, does vary with thickness-to-gap ratio.

The length of the region where the flow does not separate was measured as a percentage of the blade chord. This is plotted in Figure 4.31. This indicates that the percentage of the chord where the flow does not separate from the blade tip pressure-side edge increases linearly with the thickness-to-gap ratio. There are no abrupt changes in this behavior or any indication of behavior that might define a “thick” or “thin” blade regime.

The proposal by Denton indicated that there was a sharp transition in the tip-gap flow behavior that differentiated between the “thin” and “thick” blade regimes. The ink dot visualization results indicate that there was a relatively smooth transition in the behavior of the flow with no clear indication of flow regimes. Aside from the
Figure 4.29. Ink dot flow visualization at a gap-to-chord ratio of 5% on the flat blade tip for thickness-to-gap ratios of (a) 2.83 - no winglet, (b) 3.3, (c) 3.7, and (d) 4.3.
Figure 4.30. Ink dot flow visualizations at a gap-to-chord ratio of 8% on the flat blade tip for thickness-to-gap ratios of (e) 1.75 - no winglet, (f) 2.83, (g) 3.3, and (h) 3.7.
Figure 4.31. Percent of axial chord on the blade tip for which the flow does not separate from the pressure-side edge as a function of the thickness-to-gap ratio for gap-to-chord ratios of 5% and 8%.

designation of such regimes, the existence and extent of the reattachment line is important because it represents a site in the flow that could be highly receptive to unsteady forcing.

The ink dot flow visualization was also used to examine the flow under the blade for a number of the squealer tip blades. These are shown for the 5% gap-to-chord ratio in Figures 4.32 and 4.33. For these, the flat tip cases are shown again for reference. The location of the squealer tip is shown as a solid black curve on the images.

The addition of the squealer tip had a dramatic effect on the flow under the blade. The ink dot flow visualization indicates that a majority of the flow under the blade was turned by the squealer tip. At the larger thickness-to-gap ratio, the
Figure 4.32. Ink dot flow visualization on the blade tip at a gap-to-chord ratio of 5% and thickness-to-gap ratio of 4.3 for (a) a flat blade tip and (b) a blade with a squealer tip. The solid black curve indicates the location of the squealer tip.

Flow appears to turn relatively smoothly. However, at the smaller thickness-to-gap ratio, there is an indication of a flow re-circulation bubble under the blade. This is a major drawback as it could enhance heat transfer that could lead to erosion of the blade tip.

The effect of the squealer tip at a gap-to-chord ratio of 8% is shown in Figures 4.34 and 4.35. Based on the ink dot visualization it does not appear that the squealer tip is as effective in turning the flow at the larger gap-to-chord ratio. The flow appears to be directed toward the pressure side at the trailing edge, possibly indicating a recirculation bubble under the blade at both thickness-to-gap ratios.
Figure 4.33. Ink dot flow visualization on the blade tip at a gap-to-chord ratio of 5% and thickness-to-gap ratio of 3.3 for (a) a flat blade tip and (b) a blade with a squealer tip. The solid black curve indicates the location of the squealer tip.
Figure 4.34. Ink dot flow visualization on the blade tip at a gap-to-chord ratio of 8% and thickness-to-gap ratio of 2.83 for (a) a flat blade tip and (b) a blade with a squealer tip. The solid black curve indicates the location of the squealer tip.
Figure 4.35. Surface flow visualization on the blade tip at a gap-to-chord ratio of 8% and thickness-to-gap ratio of 3.7 for (a) a flat blade tip and (b) a blade with a squealer tip. The solid black curve indicates the location of the squealer tip.
4.2 Active Flow Control

Plasma actuator flow control investigations were performed with a plasma actuator that was designed to simulate a suction-side squealer tip. These investigations were performed at a gap-to-chord ratio of 4%, corresponding to a thickness-to-gap ratio of 3.5. The inlet Reynolds number was 500,000. The design of the plasma actuator was discussed in detail in Chapter 2. A schematic of the actuator is reproduced here in Figure 4.36. The base material was 0.0625 inch thick glass-epoxy board that was copper clad on both sides. Milling copper off of the board. The geometry of the electrodes was designed to produce a body force in the direction towards the pressure-side of the blade tip. This is indicated by the arrows shown in the picture of the actuator shown in Figure 4.36.

![Figure 4.36. Actuator configuration used in blade mounted active flow control investigations.](image)

Rather than attempting to produce a steady body force to block the flow through the gap like a passive squealer tip, the plasma actuator was operated at an unsteady frequency producing a body force that would drive instabilities inherent to the tip-
gap flow. Two sensitive locations where the effect of the flow control could be assessed were identified. One of these was inside the tip leakage vortex. The other was on the edge of the passage vortex. The five-hole Pitot probe was separately placed at each of these two locations. A range of unsteady actuator frequencies were examined to determine if any of the frequencies had a strong effect on the pressure loss measured at the specified spatial locations. The frequencies investigated ranged from 100 Hz to 2100 Hz. For comparison at different speeds, the non-dimensionalized frequency was defined as

\[ \beta = \frac{f \ast g}{U_i}, \]  

(4.1)

where \( f \) is the frequency, \( U_i \) is the upstream velocity at the entrance to the cascade, and \( g \) is the gap height.

The two spatial locations that were considered are shown relative to the measurement region in the left plot in Figure 4.37. The change in the measured pressure coefficient at the two locations is shown on the right for the range of excitation frequencies. Circles have been added to indicate the non-dimensional frequencies where the largest change occurred. These are at \( \beta = 0.03 \), corresponding to a frequency of 500 Hz at one spatial location, and at \( \beta = 0.07 \) corresponding to 1250 Hz at the second spatial location.

To confirm that observed peaks were due to instabilities which could be non-dimensionalized, investigations were also performed at a Reynolds number of \( 0.2 \times 10^6 \). The results are shown in Figure 4.38. A clear peak is seen at \( \beta = 0.03 \) and a broader peak is seen between \( 0.07 < \beta < 0.1 \). Also, a peak is seen at about \( \beta = 0.14 \) which could be a harmonic of the observed lower frequency peaks. The observed non-dimensional frequencies and the possible presence of harmonics indicate that the observed instability is most likely a shear instability.
Figure 4.37. Effect of varying the excitation frequency on downstream total pressure at two specified spacial locations in the wake of the blade row at $\text{Re} = 0.5 \times 10^6$. 
Figure 4.38. Effect of varying the excitation frequency on downstream total pressure at two specified spacial locations in the wake of the blade row at $Re = 0.2 \times 10^6$.

Full spatial surveys were performed with plasma actuation at these frequencies. Pressure loss coefficient contours for these as well as a baseline (actuator off) case are shown in Figure 4.39. Note again that the trailing edge of the blade is at the top of each plot and the end wall is along the right edge. While some change is observed in the shape, size, and strength of the tip leakage vortex with unsteady actuation, almost no insight into the effectiveness of the blade mounted plasma actuator can be made by qualitatively comparing these plots.

Mass-averaged total pressure loss coefficients associated with the tip leakage vortex were obtained by mass averaging the pressure coefficients inside the zero-level $-\lambda_2$ contour associated with the tip leakage vortex. These coefficients are shown in Figure 4.40. The bars represent the repeatability of multiple tests. Overall, a decrease in the total pressure loss coefficient associated with the tip leakage vortex
Figure 4.39. Total pressure loss coefficient contours in the wake one axial chord downstream of the blade row at a gap-to-chord ratio of 4% for two periodic excitation frequencies produced by a plasma actuator on the blade tip.

was observed with increasing actuation frequency. This corresponded to 10% and 12% improvements in the loss at the two unsteady frequencies, $\beta = 0.03$ and $\beta = 0.07$ respectively.

The corresponding end wall pressure distributions for unsteady plasma actuation are shown in Figure 4.41. These indicate virtually no change in the pressure distribution under the blade. This is in direct contrast to the passive squealer tip, which restricted the flow from passing under the blade and significantly raised the static pressure. The plasma actuator on the tip produced a minimal obstruction so that the flow was allowed to pass through the gap as it did for the baseline flat blade tip. This is an improvement over passive squealer tips which cause a secondary flow to form under the blade tip that can trap hot gases.

The area-averaged end wall pressure coefficients for the baseline and two unsteady excitation frequencies are shown in Figure 4.41. Averaging over the entire sampled area, the excitation at $\beta = 0.03$ increased the end wall averaged $c_p$ by 2.2%.
Figure 4.40. Mass averaged total pressure loss coefficients associated with the tip leakage vortex at a gap-to-chord ratio of 4% with two periodic excitation frequencies provided by a plasma actuator on the blade tip.

The excitation at $\beta = 0.07$ increased it by only 1.1%. When only the area under the blade was considered, the excitation at $\beta = 0.03$ increased the average $c_p$ by 3.5%, and the excitation at $\beta = 0.07$ increased the average $c_p$ by 4.9% compared to the baseline blade tip. The pressure magnitude under the blade is indicative of the velocity of the flow through the gap and the strength of the tip leakage vortex. The increase in the static pressure with excitation is an indication that both were reduced.
Figure 4.41. End wall static pressure coefficient contours at a gap-to-chord ratio of 4% for two periodic excitation frequencies produced by a plasma actuator on the blade tip.
Figure 4.42. Area-average pressure coefficients on the end wall under the blade for a gap-to-chord ratio of 4% at two periodic excitation frequencies produced by a plasma actuator on the blade tip.
As mentioned in Chapter 1, most flow control strategies utilized to date to reduce the losses associated with the tip leakage vortex have focused on modifications to or on the blade tip. This was the approach that was investigated in Chapter 4. In compressors, casing treatments have been found to be effective in greatly reducing the effects of the tip leakage vortex \cite{38, 39, 40}. Casing treatments have not been implemented in turbines because the heat transfer effects erode them so fast that maintenance costs quickly outpace cost savings from the efficiency gain. Despite this, Rao et al.\cite{44} showed that the tip leakage vortex is sensitive to small changes in the casing by implementing wall roughness. That study saw an influence of wall roughness on the tip leakage vortex.

Blade mounted techniques, such as squealer tips and winglets are designed to reduce the flow that passes through the tip-gap region under the blade. These are extremely effective in reducing the tip leakage vortex and the associated loss coefficient. However, when used in hot-gas sections, the squealer tips can trap hot gas under the blade tip, reducing the blade life. This chapter discusses casing mounted techniques which avoid these problems. The methods discussed in this chapter leave the tip-gap flow unaffected and allow the tip vortex to form but cause the tip leakage vortex to dissipate more quickly. The supposition was that if the
vortex dissipated quickly, it would mix with and transfer its momentum to the surrounding flow, and the overall effect would be an increase in efficiency.

Two methods were used to dissipate the tip leakage vortex: 1) introduce vorticity of opposite sign by placing vortex generators on the casing wall, and 2) increase the boundary layer thickness on the casing wall using wall roughness. For each of these passive techniques, a correlating active control technique was also employed. The effect on the flow was measured by downstream pressure surveys and blade-tip static pressure measurements. The specifics of the measurement techniques were discussed in Chapter 3. The effect of these flow control strategies on the tip-gap flow are discussed in this chapter.

5.1 Passive Vortex Generators

Passive vortex generators were placed on casing wall on the suction-side of the blade. The objective was to dissipate the tip leakage vortex by introducing vorticity of opposite sign to that of the tip leakage vortex. Two gap-to-chord ratios of 3.5% and 5% were investigated, corresponding to thickness-to-gap ratios of 4.1 and 2.83 respectively.

5.1.1 Baseline Flow

To begin these investigations, the baseline flow was examined to gain insight into areas of the flow that might be receptive to control. A first step in this involved surface oil flow visualization on the end wall. The technique was described in detail in Chapter 3. Initially, a uniform coating of the oil mixture was applied on the cascade end wall. The tunnel was then started and the oil allowed to flow. High viscosity silicone oil mixed with oleic acid and titanium dioxide was used. The cascade end wall is clear plexi-glass so that the oil pattern can be viewed from the outside. The oil pattern was illuminated with a simple incandescent light source.
Figure 5.1. Baseline flow visualization at a gap-to-chord ratio of 5% with the following features indicated: 1) the horseshoe vortex, 2) the flow traversing across the passage from the pressure side of one blade to the suction side of the adjoining blade, 3) the flow across the tip-gap, 4) an interaction region between the tip-gap and passage flows.
Figure 5.2. Baseline flow visualization at a gap-to-chord ratio of 3.5% with the following features indicated: 1) the horseshoe vortex, 2) the flow traversing across the passage from the pressure side of one blade to the suction side of the adjoining blade, 3) the flow across the tip-gap, 4) an interaction region between the tip-gap and passage flows.
The assumption is that the oil will follow the local shear stress distribution that exists on the end wall, and that that shear stress distribution is the result of the coherent motions of the flow. A photographic record of the oil pattern was taken after the pattern stopped changing, but while the flow was still moving through the tunnel at the same speed. This usually would take about two minutes to occur. Therefore the image obtained represents a time-averaged representation of the wall shear stress pattern.

The baseline (uncontrolled) case at a gap-to-chord ratio of 5% is shown in Figure 5.1. The flow direction is from the top, through the cascade, and out to the left. A number of characteristic features are observed by viewing this surface flow visualization. These features have been labelled 1-4 in the image. They correspond to

1. the horseshoe vortex that forms at the leading edge of the blade,
2. the passage vortex (two locations) that traverses the passage on either side of the center blade,
3. the tip-gap flow moving under the center blade, and
4. the tip-leakage vortex.

Just prior to the leading edge of the blade, the flow stagnates and forms pressure- and suction-side legs of the horseshoe vortex. The pressure-side leg traverses across the passage (top portion of the image) and entrains fluid from the boundary layer, forming the passage vortex. The passage vortex then impacts the suction-side of the neighbor blade at the inside of the turn. A similar effect happens on the suction-side of the center blade, where the passage vortex emanates from the pressure-side of the blade on the outside of the turn. The passage vortex that impacts the suction-side of the center blade then interacts with the tip leakage vortex (4) that forms at about the mid-chord location. The tip leakage vortex is fed by the flow that comes through the gap under the blade. The surface streak lines indicate this to
occur from approximately 30% of the chord (measured from the leading edge) to the trailing edge. The direction of the tip-gap flow is approximately 70° to the mean flow direction in the passage between the blades.

Figure 5.2 shows the baseline oil flow visualization for the baseline case at a gap-to-chord ratio of 3.5%. The same four features are identified. For this smaller gap case, the interaction between the suction-side leg of the horseshoe vortex and the passage vortex is much more defined, indicating a strengthening of these two structures compared to the 5% gap-to-chord ratio case. Additionally, the interaction between the tip-gap and passage vortices is pushed closer to the blade, indicating a stronger passage vortex in relation to the tip leakage vortex. This makes intuitive sense, as less fluid is pulled through the smaller gap and more is entrained in the horseshoe and passage vortices.

Vorticity contours in the wake of the baseline blade at a gap-to-chord ratio of 5% were obtained using the five-hole pitot probe described in Chapters 2 and 3. These contours are shown in Figure 5.3. Recall that the trailing edge of the blade is along the top edge of the plot, and the cascade end wall is along the right edge. The colors represent the different vorticity contours. The arrows are the secondary velocity vectors indicating local flow speed and direction. The dashed black lines are zero contours of the $-\lambda_2$ criteria, identifying coherent vortices. By these criteria, two vortical structures are apparent. A region of positive vorticity (red) to the left of the image is the passage vortex, and a region of negative vorticity (blue), closer to the gap, is the tip leakage vortex. The vorticity contours for the baseline case at a gap-to-chord ratio of 3.5% are shown in Figure 5.4. It is evident from these figures that the tip leakage vortex is smaller and weaker for the 3.5% gap-to-chord blade than for the 5%.

Pressure loss coefficient contours in the wake of the baseline blade at a gap-to-
Figure 5.3. Vorticity contours one axial chord downstream of the blade row for the baseline case at a gap-to-chord ratio of 5%.
Figure 5.4. Vorticity contours one axial chord downstream of the blade row for the baseline case at a gap-to-chord ratio of 3.5%.
chord ratio of 5% are shown in Figure 5.5. The pressure loss coefficient contours are shown as filled color regions. The colors associated with the highest values (red and yellow) indicate the highest pressure loss regions. The velocity vectors and $-\lambda_2$ contours overlay the pressure loss contours. The $-\lambda_2$ contours were confirmed to circle the vortex cores in the vorticity contour plots in Figures 5.3 and 5.4. It was observed in Figure 5.5 that the $-\lambda_2$ contours generally enclose regions with highest pressure loss. This is consistent with the supposition that the largest losses would be associated with the coherent vortical motions in the tip-gap flow field. This was confirmed by the pressure coefficient surveys that show that the largest losses overlay the cores of the tip leakage and passage vortices.

Figure 5.5. Pressure loss coefficient contours one axial chord downstream of the blade row for the baseline case at a gap-to-chord ratio of 5%. 

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Figure 5.6. Pressure loss coefficient contours one axial chord downstream of the blade row for the baseline case at a gap-to-chord ratio of 3.5%.
Total pressure loss coefficient contours for the baseline case at a gap-to-chord ratio of 3.5% are shown in Figure 5.6. The region enclosed by the zero contours of the $-\lambda_2$ criteria that is associated with the tip leakage vortex is significantly smaller than for the larger gap case. There is also significantly less high pressure loss (red) associated with the vortex. It is also noted that more of the passage vortex is contained in the measurement window. This is likely the result of the smaller tip leakage vortex that does not displace the passage vortex as much.

As evidenced by this discussion of the vorticity and pressure loss coefficient contours, more information is obtained from the pressure loss coefficient contour plots. For this reason, only pressure loss coefficient contour plots will be shown for the remainder of this chapter. Vorticity plots are included in Appendix A for interested readers.

To a large extent, the flow that passes through the tip-gap is driven by the pressure difference between the pressure and suction sides of the blade. The pressure taps near the tip of the blade provide an indication of the pressure distribution around the blade near the tip-gap. These are presented as pressure coefficients for the baseline condition in Figure 5.7. In the figure, the circle symbols are the pressure coefficients measured at different chord locations at the tip of the blade with a gap-to-chord ratio of 3.5%, and the squares are the pressure coefficients measured on the blade with a gap-to-chord ratio of 5%. Pressure coefficients measured at the mid-span of the blade with the tip-gap closed are shown by the red curve. This curve is the ideal, obtained in the absence of tip-gap flow.

There is little difference between the pressure coefficients on the pressure-side of the blade. However, there is significant difference between them on the suction-side of the blade. In particular, the pressure coefficients are not as negative for the 3.5% and 5% gap-to-chord cases for $x/c_x \leq 0.6$. This indicates that less flow is
coming through the gap under the blade when the gap is smaller. At the larger chord locations, near the trailing edge, the pressure coefficients are indicative of the tip leakage vortex. The more negative pressures in this region for the larger, 5% gap-to-chord ratio indicate that the tip leakage vortex in this case is stronger.

![Graph showing pressure coefficients near the tip of the blade](image)

**Figure 5.7.** Pressure coefficients near the tip of the blade for the baseline case at gap-to-chord ratios of 3.5% and 5%.

5.1.2 Passive Vortex Generator Design

Vortex generators were placed on the casing wall on the suction side of the center blade. These were designed to introduce vorticity into the flow that was of the opposite sign compared to the tip leakage vortex. As discussed in Chapter 2, these vortex generators consisted simply of rectangular pieces of thin shim material that was bent to form an “L” shape. A schematic of the vortex generator tabs,
and their placement with respect to the blade was shown in Figure 2.12. This is reproduced here, for reference, in Figure 5.8. Two vortex generator designs were used. In each design, the tabs were 6 mm (0.24 in) wide. The difference between the two tabs was their heights, which were 3 mm (0.12 in) and 5 mm (0.20 in). These vortex generator heights corresponded to height-to-gap ratios of 0.58 and 0.96 at a gap-to-chord ratio of 5%. They corresponded to 0.81 and 1.35 at a gap-to-chord ratio of 3.5%.

Two configuration of the vortex generators were used. In one configuration, the vortex generator was placed at the trailing edge of the blade and oriented to be at a $57^\circ$ angle with respect to the blade surface. This orientation placed it at a $30^\circ$ angle with respect to the direction of the flow through the tip-gap, as evidenced by surface flow visualization that was shown in Figures 5.1 and 5.2. The other configuration placed the vortex generator 19 mm (0.75 in), or $0.2c_x$, upstream of the blade trailing edge. The orientation with respect to the flow was the same for this case as for the vortex generator placed at the trailing edge. A schematic of these vortex generators and their placements was provided in Chapter 2, and is reproduced here in Figure 5.8.

5.1.3 End Wall Oil Flow Visualization

Surface flow visualization records of the blade with the different vortex generator configurations at a gap-to-chord ratio of 5% are shown in Figures 5.9 through 5.12. In each visualization the location of the vortex generator has been circled. Investigations using the smaller height vortex generator (height-to-gap ratio of 0.58) are shown in Figure 5.9 for the vortex generator at the trailing edge, and in Figure 5.10 for the vortex generator located $0.2c_x$ upstream of the trailing edge. Similarly, investigations using the larger height vortex generator (height-to-gap ratio of
Figure 5.8. Schematic of the passive vortex generators.
0.96) are shown in Figure 5.11 for the vortex generator at the trailing edge, and in Figure 5.12 for the vortex generator located $0.2c_x$ upstream of the trailing edge. Comparing these images to the image for the baseline flow shown in Figure 5.1, it appears that the width of the tip leakage vortex is substantially reduced with all of the vortex generators in place. The vortex generators at the upstream location appear to be more effective than those across from the trailing edge. With the larger height vortex generator placed at the upstream location (Figure 5.12), the surface flow visualization suggests that the tip leakage vortex is quite dissipated by one chord length downstream of the blade. Confirmation of this will come from the wake measurements.

![Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 5% and a vortex generator with a height-to-gap ratio of 0.58 placed on the end wall across from the trailing edge.](image)

Figure 5.9. Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 5% and a vortex generator with a height-to-gap ratio of 0.58 placed on the end wall across from the trailing edge.
Figure 5.10. Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 5\% and a vortex generator with a height-to-gap ratio of 0.58 placed on the end wall 0.2c_x upstream of the trailing edge.
Figure 5.11. Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 5% and a vortex generator with a height-to-gap ratio of 0.96 placed on the end wall across from the trailing edge.
Figure 5.12. Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 5% and a vortex generator with a height-to-gap ratio of 0.96 placed on the end wall $0.2c_x$ upstream of the trailing edge.
with the different vortex generator configurations are shown in Figures 5.13 through 5.16. Investigations using the smaller height vortex generator (height-to-gap ratio of 0.81) are shown in Figure 5.13 for the vortex generator at the trailing edge, and in Figure 5.14 for the vortex generator located 0.2cₓ upstream of the trailing edge. Similarly, cases investigating the larger height vortex generator (height-to-gap ratio of 1.35) are shown in Figure 5.15 for the vortex generator at the trailing edge, and in Figure 5.16 for the vortex generator located 0.2cₓ upstream of the trailing edge. Comparing these images to the image for the baseline flow shown in Figure 5.2, it appears that the width of the tip leakage vortex is again substantially reduced with all of the vortex generators in place. Also, the vortex generators at the upstream location appear to be more effective than those across from the trailing edge. However, the shorter (height-to-gap ratio of 0.81) vortex generator seems to be more effective at dissipating the tip leakage vortex than the taller (height-to-gap ratio of 1.35) vortex generator. This, in combination with the flow visualization performed with a 5% gap-to-chord ratio blade, implies that a height-to-gap ratio of around 1 is optimum for vortex generators on the end wall.

5.1.4 Results: Pressure Loss Coefficients

Surveys with the 5-hole Pitot probe were conducted in the wake of the center blade with the vortex generators to quantify their effect on the pressure loss coefficient. The surveys were taken one axial chord length downstream of the blade row, as before.

The pressure loss coefficient contours for the four vortex generators at a gap-to-chord ratio of 5% are shown in Figure 5.17. Also included in the figure is the baseline case discussed previously. The top row of pressure loss contours correspond to the shorter vortex generator, with a height-to-gap ratio of 0.58. The bottom
Figure 5.13. Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 3.5% and a vortex generator with a height-to-gap ratio of 0.81 placed on the end wall across from the trailing edge.
Figure 5.14. Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 3.5% and a vortex generator with a height-to-gap ratio of 0.81 placed on the end wall $0.2c_x$ upstream of the trailing edge.
Figure 5.15. Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 3.5% and a vortex generator with a height-to-gap ratio of 1.35 placed on the end wall across from the trailing edge.
Figure 5.16. Surface flow visualization on the end wall for a blade with a gap-to-chord ratio of 3.5% and a vortex generator with a height-to-gap ratio of 1.35 placed on the end wall $0.2c_x$ upstream of the trailing edge.
row corresponds to the taller vortex generator, with a height-to-gap ratio of 0.96. The left column corresponds to the vortex generator placed at the trailing edge, while the right column corresponds to when the vortex generator was located at the upstream location. With the baseline flow, the tip leakage vortex had the largest loss coefficients (red contours). Focusing on the tip leakage vortex, both the 0.58 and 0.96 height-to-gap ratio vortex generators reduced the pressure loss coefficient. In addition, the upstream vortex generators were more effective than those located at the trailing edge. This is consistent with the flow visualization. In general, the taller vortex generator placed at the upstream location was the most effective in reducing the pressure loss associated with the tip leakage vortex. The $-\lambda_2$ contour in this case indicated that the vortex size was also reduced by the addition of a vortex generator.

The pressure loss contours for the four vortex generators at a gap-to-chord ratio of 3.5% are shown in Figure 5.18. Also included in the figure is the baseline case. The top row of pressure loss contours correspond to the shorter vortex generator, with a height-to-gap ratio of 0.81. The bottom row corresponds to the taller vortex generator, with a height-to-gap ratio of 1.35. The left column shows the results with the vortex generator placed at the trailing edge, while the right column corresponds to when the vortex generator was placed at the upstream location.

With the baseline flow, the tip leakage vortex had the largest loss coefficient (red/yellow contours), although it was significantly less than at a gap-to-chord ratio of 5%. Focusing on the tip leakage vortex, both the 0.81 and 1.35 height-to-gap ratio vortex generators placed at both locations reduced the pressure loss coefficient when compared to the baseline case. While the size of the tip leakage vortex is smaller, as evidenced by the $-\lambda_2$ contours, the loss coefficients associated with the vortex are higher (more yellow) for the taller vortex generators. It is difficult to
Figure 5.17. Pressure loss coefficient contours one axial chord downstream of the blade row for all passive vortex generator cases at a gap-to-chord ratio of 5%.
Figure 5.18. Pressure loss coefficient contours one axial chord downstream of the blade row for all passive vortex generator cases at a gap-to-chord ratio of 3.5%.
determine from the contours whether the vortex generators are more effective when placed at the upstream or trailing edge location. The mass-averaged total pressure loss coefficient will be used to determine this.

The overall effect of the vortex generating tabs on the pressure loss due to the tip leakage vortex is summarized in Figures 5.19 and 5.20. These figures show the mass-averaged total pressure loss coefficients inside the region defined by the $-\lambda_2$ contour for the 5% and 3.5% gap-to-chord ratios, respectively. Five cases are presented in each figure. These include the baseline case and two heights of vortex generators placed at two locations. The cases are labelled on the horizontal axis. Error bars representing the repeatability of the results are also indicated.

As expected, at both gap-to-chord ratios, the baseline case has the highest pressure loss associated with the tip leakage vortex. This is substantially reduced with the addition of the vortex generators. For the 5% gap-to-chord ratio, the taller vortex generator with a height-to-gap ratio of 0.96 was more effective than the shorter vortex generator with a height-to-gap ratio of 0.58 when placed at the blade trailing edge. Also, the vortex generators of either height were more effective when placed at the upstream location. Overall at the upstream location, both vortex generators reduced the pressure loss coefficient by 25%. At the trailing-edge location, the taller vortex generator was more effective than the shorter one, with 20% and 15% reductions, respectively.

For the 3.5% gap-to-chord ratio, the shorter vortex generator with a height-to-gap ratio of 0.81 was more effective than the taller vortex generator with a height-to-gap ratio of 1.35 when placed at either location. In fact, the shorter vortex generator produced a 40% decrease in total pressure loss associated with the tip leakage vortex when placed at the trailing edge, and a 43% decrease when placed $0.2c_x$ upstream of the trailing edge. The taller vortex generator produced reductions of 26% and 21%
Figure 5.19. Mass-averaged total pressure loss coefficients associated with the tip leakage vortex for all of the passive vortex generator cases at a gap-to-chord ratio of 5%.
when placed at the same locations. This agrees with the observation from the flow visualization that a vortex generator with a height-to-gap ratio of about 1 appears to be optimum.

Figure 5.20. Mass-averaged total pressure loss coefficients associated with the tip leakage vortex for all of the passive vortex generator cases at a gap-to-chord ratio of 3.5%.

5.1.5 Results: Tip Pressure Distributions

The effect of the passive vortex generators on the blade tip loading was also examined. For this, pressure coefficient distributions near the tip of the blade were obtained for each of the vortex generator cases. The tip pressure distributions are shown in Figures 5.21 and 5.22. Again for reference, the red curve indicates measurements obtained at the mid span of the blade when tip-gap was closed. The pressure distribution for the baseline conditions are shown by the circle symbols.

For both gap-to-chord ratios, the pressure distributions for the four vortex gen-
erator cases generally lay upon that of the baseline with the exception of slight
differences on the suction side of the blade for \( x/c_x \geq 0.6 \). This region is in the
vicinity of the vortex generators and where the tip leakage vortex generally sepa-
rates from the blade suction-side.

The blade tip pressure coefficient distributions were integrated to determine the
blade tip loading. These are shown in Figures 5.23 and 5.24 for the blade at gap-
to-chord ratios of 5% and 3.5%, respectively. The trends for the two gap-to-chord
ratios are very different. For the 5% gap-to-chord ratio, the vortex generators result
in a slight decrease in blade loading compared to the baseline condition. The largest

Figure 5.21. Blade tip pressure coefficient distributions for all of the passive vortex
generator cases at a gap-to-chord ratio of 5%.
Figure 5.22. Blade tip pressure coefficient distributions for all of the passive vortex generator cases at a gap-to-chord ratio of 3.5%.
reduction in the blade loading (12%) occurred for the 0.58 height-to-gap ratio vortex generator placed at the upstream location. The 0.96 height-to-gap ratio vortex generator at the same upstream location only reduced the blade loading by 6%. At the trailing-edge location, the shorter vortex generator reduced the blade loading by the least amount (5%), and the taller vortex generator reduced the blade loading by 9%. The objective should be to reduce the pressure loss coefficient in the wake due to the tip leakage vortex, while maintaining high blade loading. Based in this, the 0.96 height-to-gap ratio vortex generator placed at the upstream location would be the best at the gap-to-chord ratio of 5%.

At the smaller gap, adding the vortex generator generally increased the blade loading near the tip. At the trailing edge, the shorter vortex generator (height-to-gap ratio of 0.81) increased the blade loading less than 1% while the taller vortex generator (height-to-gap ratio of 1.35) decreased the blade loading slightly more than 1%. When placed 0.2c_x upstream, the shorter vortex generator increased blade loading by 5% while the taller vortex generator increased blade loading by 4%. Based on this, the upstream location is a better choice for the vortex generator placement at the smaller gap-to-chord ratio. These measurements agree with the wake pressure loss coefficient measurements and surface flow visualization records that a vortex generator with a height-to-gap ratio of about 1 appears to be optimum.

5.2 Plasma Vortex Generators

The active flow control discussed in this section builds off of the passive flow control discussed in Section 5.1. In this case, the passive vortex generators were replaced by plasma vortex generators (PVGs). The advantages of these are that they can be turned on or off, and when off they do not add any parasitic drag. These studies were only conducted at a gap-to-chord ratio of 5%, corresponding to
Figure 5.23. Blade tip loading for all of the passive vortex generators at a gap-to-chord ratio of 5%.

Figure 5.24. Blade tip loading for all of the passive vortex generators at a gap-to-chord ratio of 3.5%.
5.2.1 Actuator Design

The basic design and operation of single dielectric barrier discharge plasma actuators were discussed in Chapter 2. As mentioned in Chapter 1, Thomas et al. [10] recently investigated actuators with exposed electrodes having sawtooth edges. These were shown to have body forces that were larger than those with straight edge electrodes at the same voltage. This offers the possibility to obtain greater actuator authority at lower voltages. Noting this, the actuators used in this study all had sawtooth edges.

For this investigation, plasma actuators were placed on the wall across from the gap from the blade. Three configurations were investigated. These are shown in Figures 5.25 through 5.27. In showing the actuator configurations, an image of the actuators has been overlayed onto a scaled image of the baseline surface flow visualization. This is intended to show the relative placement of the plasma actuators with respect to the flow structures of interest. Surface flow visualization with the plasma actuators was not possible because the silicone oil mixture would affect the actuator performance.

The first actuator, shown in Figure 5.25 was a single saw-toothed electrode facing in the upstream direction to act against the local mean flow. The exposed electrode edge was located across from the blade trailing edge. The exposed electrode was 44.5 mm (1.75 in) wide, and 19 mm (0.75 in) long (including the saw-toothed edge). The saw-toothed portion of the exposed electrode was made up of triangles that were 3.2 mm (0.125 in) wide and 6.35 mm (0.25 in) long. The covered electrode was 37 mm (1.25 in) long and extends out 44 mm (1.625 in), 9.5 mm (0.375 in) of which are overlapped with the exposed electrode.
The second actuator, shown in Figure 5.26, was also a single saw-toothed electrode of the same dimensions and facing upstream, but placed $0.2c_x$ upstream of the trailing edge. This was the upstream location used with the passive vortex generators.

The third actuator, shown in Figure 5.27, had two exposed sawtooth electrodes of the same dimensions that were facing each other. The electrodes were 25.4 mm (1 in) apart, centered on a line that was $0.2c_x$ upstream of the trailing edge. For this arrangement, plasma formed over the covered electrode between the two exposed electrodes. This produced equal-opposite body force vectors that produced a flow stagnation in the space between the two electrodes. The stagnation flow in the presence of the wall caused the air to jet in the wall-normal direction. This was then equivalent to a slotted wall jet.

![Figure 5.25. Schematic of the single exposed electrode plasma actuator placed facing upstream and located across from the trailing edge of the center blade.](image-url)

Figure 5.25. Schematic of the single exposed electrode plasma actuator placed facing upstream and located across from the trailing edge of the center blade.
Figure 5.26. Schematic of the single exposed electrode plasma actuator placed facing upstream and located $0.2c_x$ upstream of the trailing edge of the center blade.

Figure 5.27. Schematic of the two electrode plasma jet actuator with the centerline between the exposed electrodes located $0.2c_x$ upstream of the trailing edge of the center blade.
5.2.2 Results: Pressure Loss Coefficient Contours

Pressure loss coefficient contours measured on axial chord downstream of the blade row for the three plasma actuator cases are shown in Figure 5.28. It is important to note that the presence of the actuator electrodes had a passive effect on the flow. Therefore new baseline cases were measured that included the plasma actuators installed but with no voltage supplied.

The baseline cases for each plasma actuator configuration are shown in the top row in Figure 5.28. The actuated cases are shown in the lower row. The plasma actuator voltage was the same (40 kV) for all three cases. The single exposed electrode located at the trailing edge (Figure 5.25) is shown in the far left column. The single exposed electrode located at the upstream position (Figure 5.26) is shown in the middle column. Finally, the two electrode jet configuration (Figure 5.27) is shown in the far right column.

The focus is on the tip leakage vortex, which is the primary vortex appearing in each image. To evaluate the effect that the plasma actuators have on the vortex, one needs to compare the loss coefficient contours for the representative baseline with the actuated case, which appear in the same columns. In general the effect appears to be small.

To gain a better understanding of the results, the total pressure loss inside the \(-\lambda_2\) contour was mass-averaged. The results are shown in Figure 5.29. The circles represent the baseline cases and the x’s represent the actuated cases. The bars represent the repeatability of the results.

For the single exposed electrode actuator at the upstream location (Figure 5.26), the change between the baseline and actuated cases is inside the experimental uncertainty, and was judged to have no effect. The two-electrode plasma jet configuration (Figure 5.27) also showed an insignificant change in the loss coefficient that
Figure 5.28. Pressure loss coefficient contours measured one axial chord downstream of the blade row for plasma vortex generators located at the same locations as the passive vortex generators. All of the cases had a gap-to-chord ratio of 5%.
was inside the experimental uncertainty, and was also judged this to have no effect. However, for the single exposed electrode actuator at the trailing edge (Figure 5.25), there was a noticeable, 7%, reduction in the loss coefficient. This is approximately half of the smallest change that was observed with the passive vortex generators, but it was still significant.

![Graph showing total pressure loss coefficients](image)

Figure 5.29. Mass-averaged total pressure loss coefficients associated with the tip leakage vortex for plasma vortex generators located at the same locations as the passive vortex generators. All of the cases were at a gap-to-chord ratio of 5%.

5.2.3 Results: Blade Tip Pressure Distributions

Pressure distributions near the blade tip for the three plasma actuator cases were also measured. Baseline and actuated cases are shown in Figure 5.30. Once again, the distributions overlay each other except on the suction side of the blade near the trailing edge \((x/c_x \geq 0.6)\).

Integrating the pressure distributions around the blade provides an indication of blade loading. These are presented for the three actuator arrangements in Figure
The circles represent the baseline cases and the x’s represent the actuated cases. In each case, operating the actuator reduced the blade loading. A 7% reduction in the blade tip loading was produced by the single exposed electrode actuator at the trailing edge (Figure 5.25). A 10% reduction was produced for the single exposed electrode actuator at the upstream location (Figure 5.26). The two-electrode configuration (Figure 5.27) reduced the blade tip loading by 6%. The fact that the plasma vortex generators lowered the blade loading in all three cases indicates that the tip leakage vortex was being affected. However, only the actuator at the trailing edge location produced a change significant enough to also see the effect in the loss coefficient.

5.3 Wall Roughness

As previously mentioned, wall roughness has been shown to reduce losses associated with the tip leakage vortex [44]. This section presents results on the effects of 2- and 3-dimensional wall roughness on the tip-gap flow. Passive 3-dimensional roughness consisted of 120 and 280 grit sandpaper. The sandpaper was affixed to an insert in the end wall under the blade. The wall insert was adjusted so that the tip-gap height was the same as with a smooth wall. A passive 2-dimensional roughness consisted of copper-clad glass-epoxy board that was milled to leave parallel strips of copper that were 1.6 mm (0.0625 inches) wide and 73 mm (2.875 inches) long. The spacing between the copper strips was 17.5 mm (0.6875 inches). The height of the copper strips was 0.3 mm (0.012 inches). A photograph of this 2-dimensional roughness insert is shown in Figure 5.32. In all of the roughness cases, the gap-to-chord ratio was 5%, which corresponded to a thickness-to-gap ratio of 2.83.
Figure 5.30. Blade tip pressure coefficient distributions for plasma vortex generators located at the same locations used for the passive vortex generators. All of the cases were at a gap-to-chord ratio of 5%
Figure 5.31. Blade tip loading for plasma vortex generators located at the same locations as the passive vortex generators. All of the cases were at a gap-to-chord ratio of 5%.

Figure 5.32. Photograph of the 2-dimensional roughness insert.
5.3.1 Results: Pressure Loss Coefficient Contours

Pressure loss coefficient contours in the wake measured one axial chord downstream of the blade row for the baseline and three passive roughness cases are shown in Figure 5.33. The baseline case is a smooth teflon end wall insert. Recall that the end wall is along the right edge of the plot and the trailing edge of the center blade is along the top of the plot.

For the baseline case, the tip leakage vortex, seen on the right side of the measurement area, is relatively large and strong, as indicated by high pressure loss coefficients (red region) and the size of the zero-level $-\lambda_2$ contour. When either of the grit sandpaper were added to the end wall, the tip leakage vortex was almost completely eliminated. The 120 grit sandpaper seems to have had a greater effect towards reducing the losses associated with the tip leakage vortex. The 2-dimensional roughness also greatly reduced the size and strength of the tip leakage vortex. The mass-averaged total pressure loss inside the $-\lambda_2$ contour was reduced by 40% in this case.

Figure 5.33. Pressure loss coefficient contours measured one axial chord downstream for different passive roughness strategies at a gap-to-chord ratio of 5%.
5.3.2 Results: Blade Tip Pressure Distributions

Blade tip pressure coefficient distributions for the baseline case and three passive roughness cases are shown in Figure 5.34. The distributions overlay each other except on the suction-side of the blade near the trailing edge at $x/c_x \geq 0.6$.

Integrating the pressure distributions around the blade provides an indication of the blade tip loading. This is presented in Figure 5.35. Each of the passive roughness cases increased the blade loading. The 120 grit sandpaper increased the blade tip loading by 14% over the baseline. The 280 grit sandpaper increased the blade tip loading by 20%. Finally, the 2-dimensional roughness increased the blade tip loading by 3%.

Figure 5.34. Blade tip pressure distributions for the different passive roughness cases at a gap-to-chord ratio of 5%.
5.4 Active Wall Roughness

Based on the clear effect of wall passive wall roughness on the tip leakage vortex, two techniques were investigated using plasma actuators to simulate wall roughness. The first involved an actuator design that had 0.25 inch diameter circles milled from the copper-clad glass-epoxy board. When operated, plasma formed around the circumference of the circles and created wall-normal jets that emanated from the centers of the circles simulating 3-dimensional roughness. The second plasma roughness actuator used the copper strips that made up the passive 2-dimensional roughness. The parallel copper strips were the exposed electrodes. These were oriented to produce a body force that was created to oppose the mean flow in the passage between the blades. To prevent the exposed electrodes from producing passive roughness effects, the surface of each was covered by a layer of boron nitride and sanded smooth. Photographs of the plasma created by these two actuators are
5.4.1 Results: Pressure Loss Coefficient Contours

The effect of the 3-dimensional plasma roughness was investigated at two voltages of 11 and 15 kV. The pressure loss coefficient contours measured one axial chord downstream of the blade row are shown for these cases in Figure 5.37. With the plasma not operating, the loss coefficient was almost identical to the passive baseline case obtained with the smooth wall, previously shown in Figure 5.33. With the plasma roughness operating, what appears to be a minimal effect on the tip leakage vortex is observed.

To gain better insight into the effect of this plasma roughness, the mass-averaged total pressure loss coefficients associated with the tip leakage vortex were computed. The results are shown in Figure 5.38. The bars on the data represent the repeatability of multiple experiments. The change in the mass-averaged loss coefficient between cases is inside the repeatability uncertainty for these three cases. As a result, we conclude that the plasma roughness with the circular electrode had no measurable effect.

The effect of the 2-dimensional plasma roughness on the pressure loss coefficient measured one axial chord downstream of the blade row is shown in Figure 5.39. To contrast the effect, the left column corresponds to the 3-dimensional plasma roughness. The contours with the plasma roughness not activated are shown in the top row. The contours with the plasma roughness operating are shown in the bottom row. Comparing the 2-dimensional roughness against its baseline or against the 3-dimensional plasma roughness, it is difficult to ascertain any effect on the tip leakage vortex.

In an attempt to see any effect of the 2-dimensional plasma roughness, the total
Figure 5.36. Photographs of operating plasma roughness wall inserts.
Figure 5.37. Pressure loss coefficient contours for different actuation voltages of the 3-dimensional plasma roughness actuator at a gap-to-chord ratio of 5%.

Figure 5.38. Mass-averaged total pressure loss coefficient associated with the tip leakage vortex for different actuation voltages for the 3-dimensional plasma roughness comprised of 0.25 inch diameter circular electrodes at a gap-to-chord ratio of 5%.
Figure 5.39. Pressure loss coefficient contours of the 3-dimensional and 2-dimensional plasma roughness actuators at a gap-to-chord ratio of 5%.
pressure loss associated with the tip leakage vortex was computed. The result is shown in Figure 5.40. As with the 3-dimensional plasma roughness, the change that occurred with the 2-dimensional roughness was within the repeatability of the experiment.

Figure 5.40. Mass-averaged total pressure inside the $-\lambda_2$ criteria for the 3-dimensional plasma roughness and the 2-dimensional plasma roughness actuators compared to their baseline cases at a gap-to-chord ratio of 5%.

5.4.2 Results: Blade Tip Pressure Distributions

Blade tip pressure coefficient distributions were examined to document any effect of the plasma roughness on the blade tip loading. The blade tip pressure distributions for the 3-dimensional plasma roughness at the two voltages are shown in Figure 5.41. The distribution for the 2-dimensional plasma roughness is shown in Figure 5.42 with a 3-dimensional plasma roughness case shown for comparison. Comparing these, any difference is difficult to discern.

Integrating the pressure distributions around the blade tip provides an indication of blade tip loading. This is presented in Figure 5.43 for the 3-dimensional
Figure 5.41. Blade tip pressure distributions for different actuation voltages for 3-dimensional plasma roughness actuator at a gap-to-chord ratio of 5%.
Figure 5.42. Blade tip pressure distributions for the 3-dimensional and the 2-dimensional roughness actuators compared to their baseline cases at a gap-to-chord ratio of 5%.
plasma roughness, and in Figure 5.44 comparing the 3-dimensional roughness and 2-dimensional roughness. For the 3-dimensional plasma roughness, the blade loading decreased by 5.6% at 11 kV, and 4.2% at 15 kV. The blade loading with the 2-dimensional plasma roughness increased by 4.8%.

Figure 5.43. Blade tip loading for different actuation voltages for the 3-dimensional plasma roughness actuator at a gap-to-chord ratio of 5%.
Figure 5.44. Blade tip loading for 3-dimensional plasma roughness actuator and the 2-dimensional plasma roughness actuator compared to their baseline cases at a gap-to-chord ratio of 5%.
CONCLUSIONS

As stated in the objectives in Chapter 1, the aim of current research was to further document the tip leakage vortex behavior and reduction strategies. This research investigated a number of steps towards this aim. A two-passage linear cascade of low pressure, Pack-B turbine blades was designed and built for these investigations. The inlet Reynolds number for all investigations was 500,000 corresponding to an inlet Mach number of 0.2 and an exit Mach number of 0.3. The gap was variable up to 8% of axial chord. The flow was documented by total pressure loss coefficient contours obtained using a 5-hole Pitot probe in the wake of the blades, blade tip static pressure measurements, end wall static pressure measurements, and surface flow visualization on the blade tip and end wall.

Several flow control techniques were applied in efforts to reduce the losses associated with the tip leakage vortex. Passive blade mounted techniques have historically shown a great impact on this vortex. As discussed in Chapter 1, squealer tips and winglets have especially been shown beneficial. Therefore, initial investigations focused on how effective these techniques are and why they are so effective. Following this study, active flow control using a single dielectric barrier discharge (SDBD) plasma actuator on the blade tip was investigated. These studies placed a plasma actuator in a position similar to the suction-side squealer. Next, the potential of
flow control on the casing was investigated. First, vortex generators, both passive and active, were placed strategically on the end wall. Then, wall roughness and simulated active wall roughness using plasma actuators were investigated. This chapter summarizes the findings in each of these studies, and then provides a list of conclusions drawn from the work.

6.1 Blade Mounted Passive Flow Control

The results of the blade mounted passive flow control study, presented Chapter 4, indicated that the behavior of the flow in the tip-gap region depends both on thickness-to-gap and the gap-to-chord ratios. In all of the flat tip cases, regardless of the thickness-to-gap ratio, the ink dot flow visualization on the blade tip revealed a well defined reattachment line. This reattachment was associated with a flow separation from the pressure-side edge of the blade tip. The reattachment line never reached to the leading edge of the blade. Rather there was a well delineated portion of the blade at the leading edge at which the shear vectors indicate that the flow traversed through the gap without separating.

Given these characteristic features of the ink dot visualization, the region where the passage flow traversed through the gap without separating moved further back along the chord of the blade as the thickness-to-gap ratio increased. The chordwise extent of the reattachment line shortened as the gap-to-chord increased. This contradicts previous suggestions that “thin” blade behavior lacks a flow reattachment to the blade tip.

Even though a “thin/thick” regime could not be correlated with the presence or lack of a separation/reattachment line on the blade tip, there was evidence in other features of the tip-gap flow that indicated “thin” and “thick” blade behavior. This appeared in the mass averaged total pressure loss coefficient associated with the tip
leakage vortex, and with the blade tip loading. For flat tip blades at a gap-to-chord ratio of 5%, the total pressure loss associated with the tip leakage vortex was a minimum at a thickness-to-gap ratio of about 3.5. A similar behavior was found for blade tip loading. Based on these observations, a change in regime appeared to occur at a thickness-to-gap ratio of approximately 3.5.

The effect of squealer tips was found to primarily depend on the gap-to-chord ratio. At a gap-to-chord ratio of 5%, the squealer tip increased the blade tip loading for the full range of the thickness-to-gap ratios, encompassing both “thin” and “thick” blade regimes. At the higher 8% gap-to-chord ratio, the squealer tip had only a minor effect on the blade tip loading at smaller thickness-to-gap ratios that fell within the “thin” regime. The squealer tip had more impact at the larger thickness-to-gap ratios that fell in the lower edge of the “thick” regime. This indicates that the results observed by Douville et al. [11] were a result of the gap-to-chord ratio, not the thickness-to-gap ratio. The ink dot flow visualization indicated that the flow under the blade was turned by the squealer tip for every case investigated. In some cases a secondary flow recirculation was observed under the blade tip with the squealer tip. This secondary flow could lead to enhanced heat transfer at the tip that could cause it to erode more quickly than a flat blade tip.

6.2 Blade Mounted Active Flow Control

Blade mounted active flow control was also discussed in Chapter [4]. A plasma actuator was mounted on the tip of the blade that was designed to produce unsteady disturbances that acted with instabilities in the tip-gap flow. This method was found to reduce the loss coefficient of the tip leakage vortex by as much as 12% over the baseline condition. The optimum frequency occurred at a dimensionless value of $\beta = 0.07$, which corresponded to 1250 Hz. The actuator appeared to cause
reductions in both the size and strength of the tip leakage vortex. The leakage flow determined by end wall pressure readings under the blade were minimally changed by the actuator, indicating that the flow was not trapped in the manner of a passive squealer tip.

6.3 Wall Mounted Passive Vortex Generators

Passive flow control using vortex generators, discussed in Chapter 5, was highly effective at reducing the mass-averaged total pressure loss coefficient associated with the tip leakage vortex downstream of the blade row. At a gap-to-chord ratio of 5%, a passive vortex generator with a height-to-gap ratio of 0.58 placed across from the trailing edge resulted in a 15% reduction in total pressure loss associated with the tip leakage vortex. Moving this vortex generator 0.2\(c_x\) upstream of the trailing edge further reduced the total pressure loss by 25% over the baseline. Increasing the height of the passive vortex generator to a height-to-gap ratio of 0.96 resulted in a 20% and a 25% reduction in total pressure loss over the baseline when placed at the same trailing-edge and upstream locations. At a gap-to-chord ratio of 3.5%, a passive vortex generator with a height-to-gap ratio of 0.81 placed across from the trailing edge produced a 40% reduction in total pressure loss associated with the tip leakage vortex. Placing the same vortex generator 0.2\(c_x\) upstream resulted in a 43% decrease over the baseline. Placing a vortex generator with a height-to-gap ratio of 1.35 at the same locations resulted in 26% and 21% reductions, respectively.

At a gap-to-chord ratio of 5%, the passive vortex generators lowered the blade loading by different amounts depending on the configuration. Based on our objective to reduce the pressure loss coefficient due to the tip leakage vortex while also maintaining a high blade loading, it was concluded that the height-to-gap ratio of 0.96 vortex generator placed at the upstream location would be the best as it reduced
the blade loading the least. Conversely, at a gap-to-chord ratio of 3.5%, adding the vortex generators generally increased blade loading. Because placing the vortex generator with a height-to-gap ratio of 0.81 at the upstream location resulted in the greatest increase in blade tip loading, it was deemed the best height and location combination investigated at this gap-to-chord ratio.

Combining the above results, it was observed that a vortex generator with a height-to-gap ratio of about 1 placed at the upstream location seems to be optimum. However, a more refined grid of height and locations is needed to determine an exact optimum combination of these two variables.

6.4 Wall Mounted Plasma Vortex Generators

Based on the success of passive vortex generators, active plasma vortex generators were investigated. Three configurations were tested: one forcing upstream from the trailing edge, one forcing upstream and located 0.2\(c_x\) upstream of the trailing edge, and one that was designed to emulate a wall slotted jet at the upstream location. All of the results are discussed in Chapter 5. Of these plasma actuator configurations, only the plasma actuator at the trailing edge produced a notable 7% reduction in the pressure loss coefficient. However all of the plasma actuator configurations affected the tip leakage vortex, as was evident by reductions in the blade tip loading in all cases when the plasma vortex generators were operated.

6.5 Wall Roughness

The effect of wall roughness was investigated in Chapter 5. The effects of both 2- and 3-dimensional wall roughness on the tip leakage vortex were investigated by placing sandpaper and parallel steps on the end wall. Sandpaper of 120 and 280 grit almost completely eliminated the total pressure loss associated with the tip leakage
vortex measured one axial chord downstream. The 2-dimensional roughness reduced the pressure loss associated with the tip leakage vortex by 40% over the baseline case. Additionally, each method increased the blade loading, with the sandpaper by 14% and 20% and the 2-dimensional roughness by 3%.

6.6 Active Simulated Wall Roughness

Active flow control using plasma actuators as active roughness on the end wall was investigated in Chapter 5. Both 2-dimensional and 3-dimensional wall roughness were simulated. The present designs of plasma roughness were ineffective at reducing the pressure losses associated with the tip leakage vortex in the wake one axial chord downstream.

6.7 Overall Summary of Results

Table 6.1 summarizes the affectiveness of the various flow control techniques discussed in this document. The technique, gap-to-chord ratio, and thickness-to-gap ratio of each case is provided along with the percent change in mass-averaged total pressure loss associated with the tip leakage vortex, and percent change in blade tip loading when compared to a baseline case.

6.8 Conclusions

- Increasing the blade thickness-to-gap ratio increased the percentage of axial chord for which the flow remained attached to the blade as evidenced by blade tip surface flow visualization. There is an almost linear relationship between the blade gap-to-chord ratio and the percent of axial chord for which the flow remained attached to the blade tip.

- Adding a squealer tip to the blade redirected the flow on the blade tip away from the suction-side and back toward the pressure side. This encouraged recirculation on the blade tip.

- The sensitivity of the flow to the squealer tip was dependent on both gap-to-chord and thickness-to-gap ratios.
TABLE 6.1

EFFECT OF VARIOUS FLOW CONTROL TECHNIQUES

<table>
<thead>
<tr>
<th>Technique</th>
<th>g/c</th>
<th>t/g</th>
<th>%Δ</th>
<th>%Δ</th>
</tr>
</thead>
<tbody>
<tr>
<td>suction-side squealer tip</td>
<td>5%</td>
<td>2.83</td>
<td>-17.2</td>
<td>30</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>5%</td>
<td>3.3</td>
<td>-18.5</td>
<td>32</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>5%</td>
<td>3.7</td>
<td>-15</td>
<td>24</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>5%</td>
<td>4.3</td>
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<td>40</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>8%</td>
<td>1.75</td>
<td>0</td>
<td>43</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>8%</td>
<td>2.83</td>
<td>0</td>
<td>9.3</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>8%</td>
<td>3.3</td>
<td>-18.6</td>
<td>-49</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>8%</td>
<td>3.7</td>
<td>-26.3</td>
<td>-26</td>
</tr>
<tr>
<td>blade mounted plasma actuator ($\beta = 0.03$)</td>
<td>4%</td>
<td>3.51</td>
<td>-10</td>
<td>-</td>
</tr>
<tr>
<td>blade mounted plasma actuator ($\beta = 0.07$)</td>
<td>4%</td>
<td>3.51</td>
<td>-12</td>
<td>-</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.81 @ trailing edge</td>
<td>3.5%</td>
<td>4.1</td>
<td>-40</td>
<td>0.7</td>
</tr>
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<td>Vortex Generator, h/g = 0.81 upstream</td>
<td>3.5%</td>
<td>4.1</td>
<td>-42.6</td>
<td>5.1</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 1.35 @ trailing edge</td>
<td>3.5%</td>
<td>4.1</td>
<td>-26.2</td>
<td>-1.4</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 1.35 upstream</td>
<td>3.5%</td>
<td>4.1</td>
<td>-21.4</td>
<td>4</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.58 @ trailing edge</td>
<td>5%</td>
<td>2.83</td>
<td>-15</td>
<td>-5</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.58 upstream</td>
<td>5%</td>
<td>2.83</td>
<td>-25</td>
<td>-12</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.96 @ trailing edge</td>
<td>5%</td>
<td>2.83</td>
<td>-20</td>
<td>-9</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.96 upstream</td>
<td>5%</td>
<td>2.83</td>
<td>-25</td>
<td>-6</td>
</tr>
<tr>
<td>Plasma Vortex Generator, @ trailing edge</td>
<td>5%</td>
<td>2.83</td>
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<td>-7</td>
</tr>
<tr>
<td>Plasma Vortex Generator, upstream</td>
<td>5%</td>
<td>2.83</td>
<td>0</td>
<td>-10</td>
</tr>
<tr>
<td>Plasma Vortex Generator Jet, upstream</td>
<td>5%</td>
<td>2.83</td>
<td>0</td>
<td>-6</td>
</tr>
<tr>
<td>Wall Roughness, 120 grit sandpaper</td>
<td>5%</td>
<td>2.83</td>
<td>-100</td>
<td>14</td>
</tr>
<tr>
<td>Wall Roughness, 280 grit sandpaper</td>
<td>5%</td>
<td>2.83</td>
<td>-100</td>
<td>20</td>
</tr>
<tr>
<td>2-Dimensional Wall Roughness</td>
<td>5%</td>
<td>2.83</td>
<td>-40</td>
<td>3</td>
</tr>
<tr>
<td>3-Dimensional Plasma Roughness</td>
<td>5%</td>
<td>2.83</td>
<td>-1.7</td>
<td>-4.2</td>
</tr>
<tr>
<td>2-Dimensional Plasma Roughness</td>
<td>5%</td>
<td>2.83</td>
<td>1.7</td>
<td>4.8</td>
</tr>
</tbody>
</table>
• The transition between “thick” and “thin” blade behavior appears to occur at a thickness-to-gap ratio of about 3.5. However, this is not attributable to whether or not the flow separates, as it separates for all cases investigated. Rather, the flow remains attached over a larger portion of the blade.

• Using active flow control with plasma actuators to force the flow on the blade tip back toward the pressure-side at an unsteady frequency was moderately effective at reducing the losses associated with the tip leakage vortex.

• Casing mounted vortex generators were highly effective at reducing the losses associated with the tip leakage vortex. An optimum height-to-gap ratio of about 1 was found at two gap-to-chord ratios.

• Active vortex generators at the trailing edge were moderately effective at reducing losses associated with the tip leakage vortex.

• 3-dimensional wall roughness essentially eliminated downstream losses associated with the tip leakage vortex while increasing blade loading. 2-dimensional roughness also greatly reduced the downstream losses (40%).

• Simulated wall roughness using plasma actuators showed no effect for the actuator configurations investigated.
APPENDIX A

VORTICITY PLOTS

The studies presented in this documented used several measurement techniques to document the tip gap flow of a linear cascade of Pack-B turbine blades. One technique was the use of a five-hole Pitot probe in the wake one axial chord length downstream of the blade row. The pressure measurements provided total pressure loss coefficient contours, vorticity contours, secondary velocity vectors, and $-\lambda_2$ contours. The techniques used to obtain these from the acquired data were discussed in Chapter 3. The pressure loss coefficient contours are shown and described in detail in Chapters 4 and 5. The vorticity contours are shown in this appendix.

The measurement window is bounded on the right side by the end wall. The top edge of the measurement window is even with the trailing edge of the center blade. The filled color regions of every plot are the vorticity contours. The blue regions are regions of negative vorticity, and the red regions are regions of positive vorticity. The secondary velocity vectors, showing local flow velocity magnitude and direction are overlayed. Additionally, dashed black lines indicate the location of the zero-level $-\lambda_2$ contour, which encloses coherent vortices. In most images two vortices are observed. The primary vortex, with negative vorticity and located to the right edge of the plot near the tip gap, is the tip leakage vortex. The other, red vortex, of which varying amounts are seen is the passage vortex.
Figure A.1. Vorticity contours in the wake one axial chord downstream of the blade row for flat and squealer tip blades at various thickness-to-gap ratios at a gap-to-chord ratio of 5%.
Figure A.2. Vorticity contours in the wake one axial chord downstream of the blade row for flat and squealer tip blades at various thickness-to-gap ratios at a gap-to-chord ratio of 8%.

Figure A.3. Vorticity contours in the wake one axial chord downstream of the blade row for two periodic excitation frequencies produced by a plasma actuator on the blade tip at a gap-to-chord ratio of 4%.
Figure A.4. Vorticity contours one axial chord downstream of the blade row for all passive vortex generator cases at a gap-to-chord ratio of 5%.
Figure A.5. Vorticity contours one axial chord downstream of the blade row for all passive vortex generator cases at a gap-to-chord ratio of 3.5%.
Figure A.6. Vorticity contours one axial chord downstream of the blade row for plasma vortex generators located at the same locations as the passive vortex generators. All cases at a gap-to-chord ratio of 5%.

Figure A.7. Vorticity loss contours measured one axial chord downstream of the blade row for different passive roughness strategies at a gap-to-chord ratio of 5%.
Figure A.8. Vorticity contours for different actuation voltages of the 3-dimensional plasma roughness actuator at a gap-to-chord ratio of 5%.
Figure A.9. Vorticity contours for 3-dimensional plasma roughness and the 2-dimensional plasma roughness cases compared to their baseline cases at a gap-to-chord ratio of 5%.
This research investigated the effects of various flow control techniques on turbine efficiency. In Chapter I it was stated that entropy production is a good indicator of loss in efficiency. This appendix discusses the relationships between efficiency and entropy production.

Efficiency, $\eta$, is defined as

$$\eta = \frac{\text{actual work}}{\text{isentropic work}}.$$  \hspace{1cm} (B.1)

Recall the Gibb’s Equation from Chapter II

$$s_2 - s_1 = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}.$$  \hspace{1cm} (B.2)

Equation B.2 can be rearranged through a series of steps as,

$$\frac{1}{R} (s_2 - s_1) = \frac{c_p}{R} \ln \frac{T_2}{T_1} - \ln \frac{P_2}{P_1},$$  \hspace{1cm} (B.3)

$$\frac{c_p}{R} \ln \frac{T_2}{T_1} = \frac{1}{R} (s_2 - s_1) + \ln \frac{P_2}{P_1},$$  \hspace{1cm} (B.4)

$$\left( \frac{T_2}{T_1} \right)^{\frac{\gamma - 1}{\gamma}} = \exp \left( \frac{s_2 - s_1}{R} \right) \left( \frac{P_2}{P_1} \right),$$  \hspace{1cm} (B.5)
and finally

$$\frac{T_2}{T_1} = \left[ \exp \left( \frac{s_2 - s_1}{R} \right) \left( \frac{P_2}{P_1} \right) \right]^{\frac{\gamma - 1}{\gamma}}. \quad (B.6)$$

If the flow is isentropic, then

$$\frac{T_2}{T_1} = \left( \frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}}. \quad (B.7)$$

Returning to the definition of efficiency in Equation (B.1)

$$\eta = \frac{c_p(T_1 - T_2)_{\text{actual}}}{c_p(T_1 - T_2)_{\text{ideal}}} = \frac{1 - \frac{T_2}{T_1}_{\text{actual}}}{1 - \frac{T_2}{T_1}_{\text{ideal}}}, \quad (B.8)$$

or simply

$$\eta = \frac{1 - \left[ \exp \left( \frac{s_2 - s_1}{R} \right) \frac{P_2}{P_1} \right]^{\frac{\gamma - 1}{\gamma}}}{1 - \left( \frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}}}. \quad (B.9)$$

Therefore, the efficiency, $\eta$, is related to the change in entropy, $s_2 - s_1 = \Delta s$.

To visualize the affect of this entropy production on efficiency, consider the temperature-entropy diagram with constant pressure lines shown in Figure B.1.

Taking the system from $P_1$ to $P_2$ isentropically means $s_2 = s_1$, and the work is $c_p(T_{2i} - T_1)$. The actual, irreversible process going from $P_1$ to $P_2$ produces entropy, $\Delta s$. Also, $T_{2a}$ is clearly higher than $T_{2i}$, so the actual work is less than the ideal ($c_p(T_{2a} - T_1) < c_p(T_{2i} - T_1)$). This shows us that increases in entropy are indicative of a process in which the maximum work is not obtained. This difference between the actual work and the ideal work is the “loss” in the system.

There is no way to quantify exactly how much gain in efficiency is obtained with these techniques. However, if we assume 100% efficiency everywhere else and consider just the efficiency associated with the tip leakage vortex we can approximate it using the following derivation.
Figure B.1. Temperature-Entropy diagram.
The data discussed in this document is all referenced to the upstream static so that

\[ P_{te} \equiv P_{te} - P_{si}, \quad (B.10) \]

\[ P_{ti} \equiv P_{ti} - P_{si}, \quad (B.11) \]

and

\[ P_{se} \equiv P_{se} - P_{si}, \quad (B.12) \]

where \( P_{si} \) is the static pressure at the inlet, \( P_{te} \) is the total pressure at the exit, \( P_{ti} \) is the total pressure at the inlet, and \( P_{se} \) is the static pressure at the exit. These measurements allow simple calculation of the total pressure coefficient

\[ c_p = \frac{P_{ti} - P_{te}}{P_{te} - P_{se}} = \frac{P_{ti} - P_{te}}{P_{te} - P_{se}} = \frac{P_{ti} - P_{te}}{\frac{1}{2} \rho V_e^2}. \quad (B.13) \]

The pressure coefficient, \( c_p \), is a non-dimensional measure of the pressure loss across the cascade. In the background it was shown that

\[ \Delta s = -R \ln \frac{P_{te}}{P_{ti}}, \quad (B.14) \]

which tells us that if \( P_{te} < P_{ti} \) then \( \Delta s > 0 \). As demonstrated above, increases in entropy indicate loss. If \( c_p \) is positive, then \( P_{te} < P_{ti} \), so \( c_p \) is a good indication of \( \Delta s \), but it is not a true measure. To get a true measure of the increase in entropy, we need to know \( \frac{P_{se}}{P_{ti}} \), which is not measured.

Mattingly [61] provides the relation

\[ \frac{P_{te}}{P_{ti}} = \frac{1}{1 + c_p(1 - \frac{P_{se}}{P_{te}})}. \quad (B.15) \]
Unfortunately, $\frac{P_{te}}{P_{se}}$ is also based on referenced data. However, we know the inlet velocity and contraction ratio, so we know the exit Mach number ($M_e = 0.3$) and

$$\frac{P_{te}}{P_{se}} = \left[ 1 + \frac{1}{2} (\gamma - 1)M_e^2 \right]^\frac{\gamma}{\gamma - 1} \quad (B.16)$$

is an isentropic relation. So we can use Equation [B.16] in Equation [B.15] which can be used in Equation [B.14] to obtain the change in entropy.

To truly understand the efficiency, a mass-averaged total pressure loss ($c_{pt}$) from the entire exit area would be needed. Because we do not have this, we will consider entropy production based on the mass-averaged total pressure loss coefficients of the tip leakage vortex. We can only compare cases with their baseline cases.

The mass-averaged total pressure loss coefficients associated with the tip leakage vortex as evidenced by the $-\lambda_2$ contours are known. The exit Mach number, $M_e$, is about 0.31, $\gamma = 1.4$, and $R = 286.6 \frac{m^2}{s^2K}$. Then Equation [B.14] provides the $\Delta s$ values. Then if we assume a pressure ratio, $\frac{P_2}{P_1}$, of $\frac{1}{179}$, which is typical for low pressure turbines, then Equation [B.9] can be used to calculate the efficiency of the turbine if the only loss was that produced by the tip leakage vortex. The change in efficiencies of every controlled case examined in this document are shown in Table B.1.
### TABLE B.1

EFFICIENCY OF EACH CASE IF LOSS IS DUE ONLY TO ENTROPY PRODUCED BY THE TIP GAP VORTEX

<table>
<thead>
<tr>
<th>Technique</th>
<th>g/c</th>
<th>t/g</th>
<th>%Δη</th>
</tr>
</thead>
<tbody>
<tr>
<td>suction-side squealer tip</td>
<td>5%</td>
<td>2.83</td>
<td>0.5%</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>5%</td>
<td>3.3</td>
<td>0.5%</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>5%</td>
<td>3.7</td>
<td>0.5%</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>5%</td>
<td>4.3</td>
<td>0.1%</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>8%</td>
<td>1.75</td>
<td>0%</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>8%</td>
<td>2.83</td>
<td>0%</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>8%</td>
<td>3.3</td>
<td>0.6%</td>
</tr>
<tr>
<td>suction-side squealer tip</td>
<td>8%</td>
<td>3.7</td>
<td>0.9%</td>
</tr>
<tr>
<td>blade mounted plasma actuator (β = 0.03)</td>
<td>4%</td>
<td>2.83</td>
<td>0.2%</td>
</tr>
<tr>
<td>blade mounted plasma actuator (β = 0.07)</td>
<td>4%</td>
<td>2.83</td>
<td>0.3%</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.81 @ trailing edge</td>
<td>3.5%</td>
<td>4.1</td>
<td>1.2%</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.81 0.2cₓ upstream</td>
<td>3.5%</td>
<td>4.1</td>
<td>1.2%</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 1.35 @ trailing edge</td>
<td>3.5%</td>
<td>4.1</td>
<td>0.8%</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 1.35 0.2cₓ upstream</td>
<td>3.5%</td>
<td>4.1</td>
<td>0.6%</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.58 @ trailing edge</td>
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<td>2.83</td>
<td>0.4%</td>
</tr>
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<td>0.7%</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.96 @ trailing edge</td>
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<td>2.83</td>
<td>0.6%</td>
</tr>
<tr>
<td>Vortex Generator, h/g = 0.96 0.2cₓ upstream</td>
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<td>2.83</td>
<td>0.7%</td>
</tr>
<tr>
<td>Active Vortex Generator, @ trailing edge</td>
<td>5%</td>
<td>2.83</td>
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</tr>
<tr>
<td>Active Vortex Generator, 0.2cₓ upstream</td>
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<td>0%</td>
</tr>
<tr>
<td>Active Vortex Generator Jet, 0.2cₓ upstream</td>
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<td>Wall Roughness, 120 grit sandpaper</td>
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<td>3%</td>
</tr>
<tr>
<td>Wall Roughness, 280 grit sandpaper</td>
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<td>3%</td>
</tr>
<tr>
<td>2-Dimensional Wall Roughness</td>
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<td>2.83</td>
<td>1.1%</td>
</tr>
<tr>
<td>Active Wall Roughness</td>
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<td>2.83</td>
<td>0%</td>
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<tr>
<td>Active 2-Dimensional Wall Roughness</td>
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<td>2.83</td>
<td>0%</td>
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</table>
BIBLIOGRAPHY


