This research examines the use of passive and active blade-mounted flow control to reduce the unwanted losses associated with the blade tip clearance flow in a stationary, open-return, rectilinear turbine cascade at one atmosphere.

Traditional flow control techniques have focused on passive methods to improve the aerodynamics in the tip region. However, passive methods can create increased heat transfer coefficients on the blade tip and clearance endwall, leading to blade degradation. To improve on these methods, various active flow control methods were designed and tested. The active control was designed to improve the clearance flow aerodynamics without introducing negative heat transfer effects. The flow control methods implemented were single dielectric barrier discharge plasma actuators of various designs and a passive partial suction-side squealer design. The passive squealer was used to benchmark the active designs against a known favorable device.

The tip clearance flow was investigated over Reynolds numbers ranging from $5.3 \times 10^4$ to $1.04 \times 10^5$ at clearance heights between one and four percent of axial blade chord. The tip clearance flow was documented using flow visualization and pressure measurements on the blade and endwall surfaces, inlet endwall boundary layer surveys, and wake pressure measurements downstream of the blade. These were carried out in order to understand the receptivity of the tip clearance flow to
various types of flow control and the applicable range over which the flow control was effective.

The plasma actuator designs caused a reduction in the downstream total pressure loss coefficient ranging between 2% to 12%, depending on Reynolds number, while the passive squealer showed a change of approximately 40%. The results show that the plasma actuator was able to favorably mitigate the adverse effects of the tip clearance flow in a similar manner as the squealer tip, without the drawbacks of the passive method. Plasma actuation was demonstrated as a suitable as a means of reducing the tip clearance flow loss.
This dissertation is dedicated to my family and friends. Thank you for your love, encouragement, and support during my graduate studies.
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SYMBOLS

*English*

- \(c\) Blade Chord
- \(c_p\) Total Pressure Loss Coefficient
- \(\bar{c}_p\) Overall Mass-Averaged Total Pressure Loss Coefficient
- \(c_{ps}\) Blade Static Pressure Coefficient
- \(c_{p-blade}\) Endwall Static Pressure Coefficient Under the Blade Only
- \(c_{p-wall}\) Endwall Static Pressure Coefficient
- \(c_x\) Blade Axial Chord
- \(\vec{f}\) Plasma Volumetric Body Force on Fluid
- \(f(\alpha)\) Five-Hole-Pitot Probe Function Relating Probe Pressures to \(\alpha\)
- \(f(\beta)\) Five-Hole-Pitot Probe Function Relating Probe Pressures to \(\beta\)
- \(h\) Enthalpy
- \(j\) Electric Current
- \(k\) Boltzmann’s Constant
- \(n\) Normal Direction
- \(n_e, n_i\) Electron and Ion Densities Within Plasma
- \(n_o\) Background Plasma Density
- \(p\) Blade Pitchwise Spacing
- \(s\) Blade Span, Entropy
- \(t\) Blade Thickness
- \(u, v, w\) Cascade Streamwise, Transverse, Wall-normal Coordinate Velocities
- \(x, y, z\) Cascade Streamwise, Transverse, Wall-normal Directions
\( x/c_x \) Normalized Axial Position on Blade  
\( v_{ax} \) Local Streamwise Vector In Cascade Blade Wake  
\( v_{m2} \) Overall Mass-Averaged Resultant Velocity Vector In Cascade Blade Wake  
\( A_{ij} \) Local Area Surrounding Grid Point within the Five-Hole-Pitot Probe Blade Wake Measurement  
\( \vec{B} \) Magnetic Induction  
\( C_p \) Specific Heat At Constant Pressure  
\( \overline{C_{pt}} \) Endwall Area-Averaged Static Pressure Coefficient  
\( \overline{C_{pt,bl}} \) Endwall Area-Averaged Static Pressure Coefficient Under the Blade Only  
\( \vec{D} \) Electric Induction  
\( \vec{E} \) Electric Field  
\( H \) Normalized Spanwise Location (Hub: \( H=0 \), Endwall: \( H=1 \)), Magnetic Field Strength, Boundary Layer Shape Factor  
\( H_t \) Five-Hole-Pitot Probe Total Pressure During Calibration  
\( M \) Mach Number  
\( P \) Pressure  
\( P_m \) Five-Hole-Pitot Probe Mean Static Pressure  
\( Q_p \) Five-Hole-Pitot Probe Dynamic Pressure Flow Coefficient  
\( R \) Radius of a Streamline, Gas Constant  
\( Re \) Reynolds Number Based on \( c_x \) and \( U_i \)  
\( Re_c \) Reynolds Number Based on \( c \) and \( U_i \)  
\( S_p \) Five-Hole-Pitot Probe Static Pressure Flow Coefficient  
\( T \) Temperature  
\( TI \) Boundary Layer Turbulence Intensity  
\( U \) Velocity of a Streamline  
\( U_\infty \) Mean Freestream Inlet Velocity to the Cascade
\( \mathbf{U}_{\text{gap}} \) Mean Tip Clearance Gap Velocity

\( \mathbf{U}_a \) Mean Plasma Actuator Velocity

\( V \) Local Resultant Velocity Vector In Cascade Blade Wake

\( X, Y, Z \) Cascade Axial, Pitchwise, Spanwise Coordinate Directions

\( Zw \) Zweifel Coefficient

**Greek**

\( \alpha, \beta \) Pitch, Yaw Angles of the Downstream Velocity Vector

\( \delta \) Boundary Layer Thickness

\( \delta^* \) Boundary Layer Displacement Thickness

\( \gamma \) Ratio of Specific Heats

\( \epsilon \) Dielectric Coefficient

\( \epsilon_0 \) Permittivity of Free Space

\( \zeta \) Energy Loss Coefficient

\( \zeta_s \) Turbine Entropy Loss Coefficient

\( \eta \) Efficiency

\( \eta_t \) Total-to-Total Efficiency

\( \theta \) Stagger Angle, Boundary Layer Momentum Thickness

\( \kappa \) Relative Static Permittivity

\( \lambda_D \) Debye Length

\( \nu \) Kinematic Viscosity

\( \pi \) Stage Pressure Ratio

\( \rho \) Density of a Fluid Particle

\( \rho_C \) Plasma Net Charge Density

\( \sigma \) Cascade Blade Solidity

\( \tau \) Tip Clearance, Stage Temperature Ratio

\( \phi \) Electric Potential
\[\Delta s\] Change in Entropy
\[\Omega_x\] Streamwise Vorticity
\[\hat{\Omega}_x\] Nondimensionalized Streamwise Vorticity

**Subscripts**

- \(a, b\) Actuated and Baseline properties
- \(blade\) Local Property on the Blade Surface
- \(e\) Quantity Downstream of the Test Blade
- \(i\) Quantity Upstream of the Test Blade
- \(isen\) Isentropic State
- \(max\) Maximum Quantity
- \(ref\) Reference Quantity
- \(s\) Static Property, Isentropic Rotor-Stator Turbine Stage
- \(t\) Total Property
- \(wall\) Cascade Endwall Surface property
- \(A\) Fluid particle A Outside of a Boundary Layer
- \(B\) Fluid particle B Inside of a Boundary Layer
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1.1 Motivation

The efficiency of gas-turbine engines is an ongoing concern for both the designers and operators of these devices. High fuel costs and the broad awareness of finite oil resources have prompted a greater interest towards reducing the fuel usage, and thus, the aerodynamic inefficiencies that are present within turbomachines.

Inefficiency within a turbine stage accumulates due to aerodynamic losses from a variety of sources. Overall, the largest single source of loss is caused by the tip leakage flow [1]. In both the compressor and turbine stages of unshrouded gas-turbine engines, the rotation of the blade rows with respect to the stationary casing, or endwall, necessitates having a small tip clearance from the casing. The leakage of flow through this tip clearance causes a number of adverse effects that result in a reduction in the overall efficiency of a turbine stage. It has been stated that the loss in turbine stage efficiency due solely to the tip clearance flow varies between 2% to 6% for a realistic clearance of 1% of blade span, and that this loss accounts for 20% to 40% of the overall loss within the turbine [2, 3].

The potential savings of reducing this unwanted loss is great. Lattime and Steinetz [4] reported that for only a 1% reduction in specific fuel consumption, the potential fuel savings for commercial U.S. airline carriers alone would be approximately 53 billion liters (14 billion gallons) annually. In Appendix B, a parametric
cycle analysis was completed, showing the effect that changes in component efficiency has on the thrust specific fuel consumption. Here, the effect of the turbine and compressor performance on the fuel consumption is detailed. From the results of Lattime and Steinetz, as well as this qualitative analysis, even a small reduction in the flow penalty would yield a large improvement in emissions and cost savings.

1.2 Introduction

This research is an experimental investigation of the feasibility of using flow control to mitigate the tip leakage flow loss. Reduction or elimination of this flow loss is problematic, for three reasons. First, the tip clearance flow is a highly three-dimensional, complex flowfield which requires detailed, concentrated flow control methods to counteract. Secondly, the tip clearance, and therefore the loss incurred, varies during the operating cycle of the engine. So it may be necessary to tailor flow control over a range of Reynolds numbers and tip clearance values to cover the various operating points within the engine. Finally, the loss will increase over the life of the engine due to deterioration of the blades [5]. Degradation has been observed on the pressure side corner of the turbine blade tip, which leads to ‘burnout’ of the tip surface [6, 7, 8].

In an attempt to understand the tip clearance flow and other ‘secondary losses’, researchers have tried to characterize losses that occur in a turbine stage in hopes that flow control can be favorably applied. Here, ‘secondary losses’ and ‘secondary flow’ refer to any flow other than the two dimensional inviscid main passage flow. The secondary flow region extends inward approximately from 10% to 30% of the span from the tip [3].
1.3 Loss Generation Mechanisms

1.3.1 Sources of Loss

Aerodynamic losses are generated from a number of sources. Denton states that losses are typically broken down into profile loss, endwall loss, and leakage loss [9]. Profile loss is loss generated in the blade boundary layers and blade wakes. Endwall loss refers to loss that is a result of the endwall boundary layer passing through the tip clearance. Leakage loss pertains to losses incurred by the flow that passes through the blade tip clearance. The leakage flow through the clearance causes both endwall loss and leakage loss. Lakshminarayana stated that the most important loss generation mechanisms are [3]:

1. the blade-tip-endwall clearance flow,
2. the interaction of blade and endwall boundary layers with the flow in the blade-tip region, and
3. the diffusion and mixing of flows.

It is the loss of flow through the tip clearance from the main passage and the subsequent irreversible mixing and diffusion that are the sources of inefficiency. The presence of the blade tip clearance will cause less-than-ideal conversion of the available total enthalpy into rotational motion. The turbine tip clearance allows air to leak over the blade tip and no work can be extracted from this fluid. Because of the large extent to which the different loss mechanisms interact, they are not easily separable from each other.

From the list of loss generation mechanisms, the first two items create three distinct vortices: the tip leakage vortex, the horseshoe vortex, and the passage (or crossflow) vortex. Figure 1.1 shows these near-tip flow phenomena that are traditionally found to occur in a turbomachine. Figure 1.1(a) shows a qualitative drawing of the tip leakage vortex that is created by the flow that leaks through the tip clearance. Figure 1.1(b) illustrates the stagnation point, or horseshoe vortex, as well
as the passage vortex on one side of the blade. In Figure 1.1(c), a different view of the passage vortex is given. It is known that the leakage vortex is in close proximity to the passage and horseshoe vortices, allowing for interaction and mixing between all three structures. Therefore the leakage flow and tip leakage vortex cannot be studied in isolation. Other nearby vortical flows, in addition to the leakage vortex, must be examined, understood, and included in the analysis, as the leakage flow interacts highly with them. Each vortex will be discussed in more detail in the next sections.
1.3.2 Tip Leakage Vortex

Tip leakage flow occurs because of the favorable pressure differential across the blade tip between the pressure and suction surfaces [3, 10]. Because of the presence of the tip clearance, the flow leaks through the tip clearance under the blade tip without being turned through the blade turning angle like the main passage flow is. The flow leaks under the tip at various angles with respect to the local normal to the blade camber line, depending on the axial chord location. At the exit of the tip clearance, a discontinuity exists where the tip clearance flow meets the main passage flow. This causes the clearance flow to roll up into a coherent streamwise vortex, called the tip leakage vortex, on the suction-side corner of the blade tip. This vortex eventually detaches from the suction-side corner and moves downstream. It is the inseparable combination of the tip leakage flow and the tip leakage vortex, that causes losses in this region. Lakshminarayana [3] states that leakage flow itself is an inviscid effect. However, vortex roll-up and downstream diffusion are controlled by viscous effects. Taken together the leakage flow and the resulting loss is a combination of both, namely the inviscid fluid leaking through clearance, and the fluid scraping against the blade tip and the endwall.

Lakshminarayana [3] lists the major unwanted effects of the tip leakage flow:

1. Three-dimensionality of the leakage flow and vortex due to: clearance flow diffusion, vortex roll-up and entrainment, downstream convection, and mixing with the main passage flow.

2. Dissipation and mixing of the leakage flow and vortex.

3. Blade unloading at the tip and near tip region.

4. Inlet distortion to a downstream blade rows, with possible generation of noise, increased downstream blade stresses, and forced vibration and/or flutter.
5. Increased blade tip heat transfer coefficients, which affect the cooling requirements in the tip region.

1.3.3 Passage Vortex

Passage vortex formation within a cascade environment can be understood using a radial momentum balance [2, 3, 11]. It is assumed that the flow is incompressible, steady, and inviscid, and that there is no variation in the pitchwise velocity. Through a full rotor-stator stage, as the flow is turned through the blade turning angle, the centripetal acceleration must balance the pitchwise pressure gradient. As shown in Figure 1.2, at the endwall the slower moving boundary layer fluid, represented by point B, is still subjected to the same pitchwise pressure gradient as flow outside the boundary layer, represented by point A. In contrast to the fluid in point A, the boundary layer fluid of point B will have a smaller radius of curvature, and be deflected away from the pressure side of one blade toward the suction side of the adjacent blade, moving to point B’. The momentum balance is given in Equation 1.1.

![Figure 1.2. Passage Vortex Formation, taken from Lakshminarayana, 1996 [3]](image)
Here, the radius of curvature of fluid at point A is denoted as \( R_A \), while the radius of curvature for fluid at point B is \( R_B \). The variable ‘n’ is the normal direction, as shown in the figure. Because of continuity, the movement of flow near the endwall means that flow elsewhere must be directed back toward the pressure side of the original blade. This leads to a vortical structure called the passage vortex, which is present in both cascade flows and axial machines. The strength of the passage vortex is a function of the main flow turning angle, the momentum defect of the incoming casing boundary layer, and the pitchwise velocity gradient. Overturning occurs near the casing wall and underturning occurs farther away from the casing wall [10].

1.3.4 Horseshoe Vortex

The formation of a horseshoe vortex can be understood by examining Figure 1.1(b) [3, 10, 12]. Upstream of the blade, the casing boundary layer experiences large flow turning in the pitchwise direction because of the thick leading edge of the blade. The flow decelerates as it approaches the leading edge, setting up an adverse pressure gradient. The flow cannot withstand the adverse gradient, and so moves away from the blade pressure and suction surfaces in the pitchwise direction (toward the adjacent blades) and toward the casing in the spanwise direction, rolling up into a coherent vortex at the blade leading edge. The flow on each side of the blade forms one leg of the horseshoe vortex. The vortex leg on the blade suction-side rotates counterclockwise while the vortex leg on the pressure-side rotates clockwise, when
viewed from downstream. The two vortex legs interact with the blade boundary layer, entraining viscous boundary layer fluid into the vortex. As the pressure-side leg enters the main passage, it is convected toward the suction-side of the adjacent blade because of the pitchwise pressure gradient. As the pressure-side leg moves across the passage it merges with the passage vortex. The passage vortex meets the suction surface near the minimum static pressure point, then moving away from the endwall and growing rapidly as it travels along the suction surface. The suction-side leg of the horseshoe vortex follows the contour of the blade suction surface until it meets the passage vortex. The suction-side leg of the horseshoe vortex then leaves the surface and moves around the outer edge of the passage vortex, as the passage vortex moves downstream.

In an effort to understand how these effects are produced, it is important to see how losses are related to flow properties through a blade row.

1.4 Losses and Entropy Generation

The formation of losses are attributable to three factors, as cited by Denton [9]:

1. Viscous friction that occurs in boundary layers or free shear layers,
2. Heat transfer across finite temperature differences, and
3. Nonequilibrium processes such as rapid expansions or shock waves.

Each of these mechanisms will have a different influence on the overall loss, but for this research only the first two mechanisms are present. To quantify the loss through a stage, many authors, including Denton, list the total pressure loss coefficient. This is defined in a turbine as

$$c_p = \frac{P_{ti} - P_{te}}{P_{te} - P_{se}}. \quad (1.2)$$

This coefficient is used primarily because of the ease of calculating pressure loss from cascade testing. A better metric of loss is the energy, or enthalpy, loss coefficient
defined as

\[ \zeta = \frac{h_{se} - h_{se,}^{\text{isen}}}{h_{te} - h_{se}} \] (1.3)

where \( h_{se,}^{\text{isen}} \) is the final enthalpy of an isentropic expansion or compression to the same final static pressure as the real process. The enthalpy-entropy diagram, as given by Denton, is shown in Figure 1.3. Fluid at inlet state 1 with \( P_{01} \) and \( P_1 \), if brought to rest isentropically would arrive at the exit state 2s, or the isentropic final static pressure with \( P_{02} \) and \( P_{2s} \). However, the fluid actually goes to the exit state 2 with \( P_2 \) and \( P_{02} \), introducing entropy with the change in static pressure. This analysis could also be considered with an irreversible change in total pressure, which would introduce entropy similarly. In either case, the definition given in Equation 1.3 is not sufficient in a rotating machine, where the relative stagnation pressure and relative stagnation enthalpy may vary due to changes in radius, and not as a result of losses.

Figure 1.3. Enthalpy-Entropy Diagram, taken from Denton, 1993

A better definition of loss can be written in terms of the isentropic efficiency. Because the isentropic efficiency is defined as the ratio of actual work to the isentropic work, only deviations from isentropic flow produce loss. These are a result of heat
transfer or thermodynamic irreversibilities. For this research, the tip clearance flow was investigated in a stationary, non-rotating cascade facility. In this environment the flow is adiabatic, so the only loss generation mechanism is via entropy creation.

It is helpful to be able to give a quantitative measure of the change in entropy based on measurable flow parameters. Considering the flow as a calorically perfect gas, the change in entropy through a turbine is defined by Gibb’s Equation from Oates [13] as

\[ s - s_{ref} = C_p \ln\left(\frac{T}{T_{ref}}\right) - R \ln\left(\frac{P}{P_{ref}}\right), \]  

(1.4)

where \( C_p \) is the specific heat at constant pressure and \( R \) is the gas constant. In an adiabatic flow through the stationary cascade facility, the stagnation temperature is constant, and so entropy change is only a function of the change in total pressure. Therefore Equation 1.4 becomes

\[ \Delta s = -R \ln\left(\frac{P_{te}}{P_{ti}}\right). \]  

(1.5)

From this definition of entropy and from Equation 1.3, the turbine entropy loss coefficient can be defined as

\[ \zeta_s = \frac{T_{se}\Delta s}{h_{te} - h_{se}}. \]  

(1.6)

For low Mach numbers, below 0.3, the above definitions for Equations 1.2, 1.3, and 1.6 approach the same value, even where changes in stagnation temperature are known to occur.

1.5 Turbine Efficiency

As an extension of the definitions of loss above, it is also important to be able to quantify the turbine efficiency in order to understand the overall impact on engine performance. The following derivation was made by Stephens [14].

The definition of efficiency for aeronautical applications is the total-to-total
efficiency through a turbine stage (rotor-stator combination) given by Lakshminarayana [3] as

\[ \eta_t = \frac{\text{Actual Work Output}}{\text{Ideal Work Output}} = \frac{T_{t1} - T_{t3}}{T_{t1} - T_{t3ss}} \]  

(1.7)

Here state 1 refers to the inlet flow to the nozzle, state 2 to the nozzle exit flow, which is also the rotor inlet flow, and state 3 is the rotor exit flow. State 3ss refers to the condition when the flow is isentropic through the entire rotor-stator turbine stage. For this research in the stationary cascade facility, the test blade simulates a rotor of a turbine stage in a blade relative frame of reference. For this, there is no stator section, so for the calculations of the turbine stage efficiency given below the stator efficiency will be fixed at 100% and the state variables will be redefined. State 1 will be redefined as state ‘i’, for the cascade inlet, and state 2 will be redefined as state ‘e’, for the cascade exit. In a turbine, this efficiency equation takes the form

\[ \eta = \frac{1 - \tau}{1 - \pi_i \left(\frac{\gamma - 1}{\gamma}\right)} \]  

(1.8)

where \( \pi \) is the ideal pressure ratio through a full rotor-stator stage, \( \tau \) is the actual temperature ratio through the stage, and \( \gamma \) is the ratio of specific heats. For a typical low pressure turbine, the pressure ratio through the entire stage is approximately \( \pi \approx 0.4 \), which will be assumed for this calculation. The temperature ratio is defined here as

\[ \tau = \frac{T_{te}}{T_{ti}} \]  

(1.9)

Incorporating Equation 1.5, the total temperature ratio is then

\[ \tau = \frac{T_{te}}{T_{ti}} = \left(\frac{P_{te}}{P_{ti}} \exp \left(\frac{\Delta s}{R}\right)\right)^{\frac{1}{\gamma - 1}} \]  

(1.10)

The parameter that is measured downstream of the cascade is the total pressure loss coefficient at state 2, given previously in Equation 1.2. This change in total pressure may be used to determine the corresponding efficiency change. To find the ratio of
total pressures, the total pressure loss coefficient (Equation 1.2) is rearranged to give
\[ \frac{P_{te}}{P_{ti}} = \frac{1}{1 + c_p \left( 1 - \frac{P_{se}}{P_{te}} \right)}, \]  
\[ (1.11) \]
Now the downstream pressure ratio, \( P_{te}/P_{se} \), is given by the isentropic relation
\[ \frac{P_{se}}{P_{te}} = \left( 1 + \frac{\gamma - 1}{2} M_e^2 \right)^{\frac{\gamma - 1}{\gamma}}, \]  
\[ (1.12) \]
where the Mach number is defined as
\[ M_e = \frac{V_e}{\sqrt{\gamma R T_e}}. \]  
\[ (1.13) \]
The change in entropy may be calculated from the mass-averaged quantities over the flow measurement area. The efficiency is then calculated via Equations 1.2 - 1.13 as
\[ \eta = \frac{1 - \left( \pi \exp \left( \frac{\Delta S}{R} \right) \right)^{\frac{\gamma - 1}{\gamma}}}{1 - \pi \left( \frac{\Delta S}{R} \right)}, \]  
\[ (1.14) \]
or equivalently
\[ \eta = \left[ 1 - \pi \left( \frac{\Delta S}{R} \right) \right]^{-1} \left[ 1 - \pi \left( 1 + c_p \left( 1 - \left[ 1 + \left( \frac{\gamma - 1}{2} M_e^2 \left( \frac{\gamma - 1}{\gamma} \right) \right) \right]^{-1} \right) \right]. \]  
\[ (1.15) \]
For this research, the improvement of the flow was measured in terms of the pressure loss coefficient, \( c_p \), as defined in Equation 1.2. The \( c_p \) results can be converted to an equivalent change in overall efficiency based on Equation 1.15 if desired. However, for this calculation the values computed will be biased toward an efficiency of 100% because the remainder of the turbine stage was idealized in the calculation.

Starting from these general results, it is helpful now to examine specific flow studies to aid in understanding the blade tip region flow development in more detail. This is presented in the following sections.
1.6 Literature Review of Tip Leakage Flow in a Cascade

Over the last half-century much attention has been given to the tip leakage and other secondary flows within turbines. The majority of research has been performed in cascades, because of the ease of experimental setup and measurements. Only within the last few decades has the research focus shifted to studies within axial flow rotating machines. Because of the continued use of cascades to study this flow problem, a review of the turbine rotor tip flow research within a cascade will be given. A summary of the pertinent cascade results along with a comparison to axial rotating machines will be given after.

Moore and Tilton [15] studied the tip clearance flow on the blade tip and endwall surfaces in a linear turbine rotor cascade. The tip clearance tested was $2.1\% \, c_x$ and the Reynolds number, based on axial blade chord ($c_x$) and exit velocity, was $4.5 \times 10^5$. Static pressure measurements were taken on the blade and endwall surfaces at one chordwise plane across the clearance, which represented one flow streamline of leakage flow.

The flow entered the tip clearance at $60\% \, c_x$ on the pressure surface and exited the clearance at $72\% \, c_x$ on the suction surface. The static pressure decreased as the flow approached the clearance entrance. Fluid accelerated into the clearance and continued to accelerate along the endwall until a distance of $18\%$ from the pressure side corner where the static pressure at that location reached a minimum value. For a distance of approximately $20\%$ through the clearance gap, both the endwall and blade tip static pressures began to rise rapidly. The pressures continued to rise, becoming approximately equal to the clearance gap exit pressure. The reduction and subsequent increase of pressure through the clearance conveyed the existence of a contraction in the flow streamlines and a vena contracta (minimum streamtube area) at the location of minimum pressure in the tip clearance. In the final $80\%$ of
the tip clearance, a region of rising pressure existed that represented mixing of the clearance leakage flow with the passage flow. This produced a total pressure loss in the fluid that passed through the clearance.

Oil dot flow visualization was also performed on the endwall surface in this study that indicated a complicated endwall flow pattern. Figure 1.4 shows an example of the flow physics. In the upstream portion of the passage, the flow separated at point L, along L1 and L2. The flow subsequently moved toward the adjacent blade or proceeded into the tip clearance. An endwall separation bubble was noticeable as a laminar separation of the endwall boundary layer at S3 and a turbulent reattachment at R1. The flow remained attached along S1 until it met the passage flow on the suction side of the blade. Here, this flow became part of the tip leakage flow which moved downstream and dissipated as S1 and R1 merged together.

![Figure 1.4](image)

Figure 1.4. Oil Dot Flow Visualization of the Tip Leakage Flow on the Endwall, taken from Moore & Tilton, 1988

Rains [16] originally speculated that a vena contracta existed, and this phenomenon was first partially visualized by Graham [17], and then by Sjolander and Cao [6] for an idealized circular arc blade. The endwall separation bubble is distinct
from the one on the blade tip. The endwall separation bubble is located further in the clearance gap away from the pressure-side corner than the tip separation bubble. Heyes, Hodson, and Dailey [18] also observed an endwall separation bubble form due to the flow acceleration and eventual flow expansion in the clearance.

Heyes and Hodson [19] conducted an experimental study in a linear turbine cascade using tip static pressure distributions and five-hole pressure probe measurements downstream of the cascade blade. They presented contours of total pressure within and downstream of the clearance. In the clearance, Heyes and Hodson saw that the lowest pressure on the blade tip coincided with the vena contracta under the blade. The low pressure region extended into the clearance gap for a distance from the pressure side corner of approximately 1.5 times the tip clearance height. As the flow approached the clearance exit for a small clearance of 2.18%, the pressure rose until it reached the value at the blade suction-side. For a larger clearance of 6.16%, the pressure did not increase to that at the blade suction-side indicating that the mixing process was not complete. Thus the loss generated within the clearance is a function of the clearance height, and is likely a function of the mass flow through the clearance gap.

The authors believed that the flow mixed at the exit of the vena contracta at a distance of 1.5\(\tau\) from the pressure side corner until a distance of 6\(\tau\), where \(\tau\) is the tip clearance height. If the blade thickness was shorter than 6\(\tau\), the flow would not be completely mixed at the clearance exit. A blade thickness of 6\(\tau\) corresponded to a tip clearance ratio of \(\tau/c_x = 5.1\%\). For shorter widths, corresponding to tip clearance ratios greater than 5.1\%, it was thought that an isentropic jet with a loss-free core was present and mixing was incomplete.

Sjolander and Cao [6] conducted measurements inside of an idealized, large scale tip clearance gap in a turbine cascade to better understand the clearance flow physics. The blade had a flat, plain tip, a constant thickness, and formed a
circular arc of 90° from leading edge to trailing edge. The blade chord Reynolds number was on the order of $8 \times 10^5$. The tip clearance was tested for heights from 0.292 to 0.667 of the maximum blade thickness. The pressure differential through the clearance gap was varied using screens that were placed at the pressure-side exit.

The authors found a separation bubble that occurred near the pressure side edge of the blade tip. The flow reattached about three-quarters of the way through the clearance. This separation bubble was unique in that it showed two counter-rotating regions of flow within the bubble, as shown in Figure 1.5, with a separation line (S) and a reattachment line (R). The mass flow through the clearance caused the static pressure difference across the tip to vary along the axial blade chord. This meant that there was a variation in the amount of leakage flow along the chord as well, which would be expected for this complex three-dimensional flow. Although the separation and reattachment line locations varied for the different clearance heights and pressure differentials tested, for all test conditions examined the flow structures were qualitatively similar.

Based on clearance total pressure surveys, Sjolander and Cao [6] found that the vena contracta plane experienced almost no pressure loss. High losses were however found in the shear layer at the edge of the separation bubble where it reattached on the blade tip. This was similar to the findings of Yaras et al. and Yaras and Sjolander [7, 20].

Sjolander [1, 21] completed a review of the tip leakage flow literature in a two part series as part of the Von Kármán Institute lectures. These two lectures were entitled “Physics of Tip Clearance Flows I and II.” In Part I, Sjolander discussed the flow within the clearance region. Fluid near the blade tip pressure surface was strongly accelerated into the tip clearance, as illustrated by Figure 1.6. Surface oil flow visualization by Sjolander and Amrud [22] illustrated this result. The flow exiting the clearance was essentially normal to the main passage flow direction.
The endwall boundary layer around the leading edge was modified by the tip clearance flow. For large tip clearances, the stagnation point disappeared, and the adverse pressure gradient was weakened. This reduced the horseshoe vortex, and its formation was not discernable above a specified clearance. Sjolander and Amrud [22] found that horseshoe vortex formation occurred at 1% clearance, but not at a 2.9% clearance. Similarly, Govardhan et al. [23] saw the horseshoe vortex disappear at tip clearances above 1.5%. It was thought that endwall boundary layer fluid that would become part of the horseshoe vortex evolved into the tip leakage flow. The horseshoe vortex strength was a function of the approaching boundary layer thickness and the clearance height. This showed that the tip leakage flow was dependent on the approaching boundary layer thickness as well.

Graham [17] noticed a strong suction peak just past midchord near the blade tip for a clearance of 3.5% that was not noticeable for midspan locations. This was a characteristic of the tip clearance leakage flow on the blade surface measurements.
There was also unloading of the pressure side within about 0.3 clearance heights of the tip. Graham found that the blade loading was a function of tip clearance height. As the clearance increased, the location of the suction peak moved reward from the leading edge and the blade loading generally decreased.

Based on clearance total pressure surveys, Yaras et al. [7] found that most of the clearance fluid was seen to pass in the half of the clearance height closest to the endwall. Additionally, most the total pressure loss was seen in the half of the clearance height closest to the blade tip. They also saw high losses in the shear layer at the edge of the separation bubble where the flow reattached on the blade tip. This was similar to the findings of Yaras and Sjolander [20].

Yamamoto [24] performed measurements within the tip clearance and near the tip endwall region of a linear turbine cascade using a miniature five-hole probe for a Reynolds number of $1.7 \times 10^5$ at tip clearances of 1.3%, 2.1%, and 2.7%. Yamamoto classified the different major regions of loss in the blade-tip-endwall region and reported the five highest loss sources. The highest loss occurred at the suction surface where the leakage flow interacts with the passage vortex, resulting in flow separation from the endwall. The second highest loss occurs where the flow enters the pressure side corner, in the region from 50% axial blade chord to the trailing edge. Yamamoto went on to say that the majority of the leakage flow that forms

Figure 1.6. Separation Bubble Formation and Tip Leakage Flow Path through the clearance, taken from Sjolander & Amrud, 1987
the tip leakage vortex occurred at the rear part of the tip where the passage and leakage flows interact, but this interaction region depended on the clearance height.

Yamamoto [24] found that as the vortices moved downstream the passage vortex was pushed away from the suction surface by the tip leakage vortex, and there was a strong shear between them because the two vortices had opposite senses of rotation. This interaction led to considerable mixing as well as the production of turbulence and losses. Depending on the downstream location, the degree to which the two structures interacted and mixed varied greatly.

Sjolander [21], in the second part of the Von Kármán Institute series, discussed loss formation in the tip clearance region. He attributed losses to two sources: those within the clearance region and those downstream of the blade tip clearance.

An important parameter that governed the tip clearance flow was the corner radius of the pressure side of the blade. Bindon and Morphis [25, 26, 27, 28] documented the formation and disappearance of the separation bubble for blades with varying corner radii, including a flat, unradiused tip, at various tip clearance heights. They observed that the bubble disappeared only for a radius at or greater than $2.5\tau$. Heyes et al. [18] and Sjolander and Amrud [22] saw similar behavior.

For an unradiused tip, Bindon [27] observed a separation bubble on the pressure side corner over which the tip leakage flow passes, reattaching to the blade tip surface and traveling virtually normal to the blade chord line. There was a large low pressure region at around 50% $c_x$ that was roughly 3 times lower than the cascade outlet pressure and approximately 1.7 times lower than the lowest suction surface value.

Bindon [27] was able to measure the various components of the loss both inside and around the tip clearance region of a linear turbine cascade with unradiused flat tip blades for a tip clearance of 2.5% $c_x$. Bindon found that the internal clearance loss was approximately 39% of the total tip-endwall flow loss, showing that entropy
generation within the clearance played a role in the overall losses incurred for a blade. Within the clearance, the loss at 40% $c_x$ was ten times less than at 70% $c_x$, where the boundary layer had grown to fill the entire clearance region. At the 40% $c_x$ region, a loss free core existed within the clearance. At this chord location, the fluid in the separation bubble mixed with the loss-free core fluid. Bindon credited this mixing for much of the internal clearance loss.

In the rear part of the blade clearance, where the clearance height narrowed, the separation bubble may not have reattached. Bindon [27] believed that the pressure gradient within the bubble forced the flow toward the leading edge, and the loss-free core and separation bubble interaction were more severe, causing higher losses in that region. It was also found that the loss due to the suction corner mixing of the leakage vortex was 48% of the total loss. Endwall losses due to shear and other secondary losses made up the remaining 13% of tip clearance flow loss. Mixing losses were attributed to separation bubble fluid being unable to diffuse around the suction-side corner. This loss was only truly present in the last 20% of blade chord. The loss is shown in Figure 1.7, which details the accumulated loss development through the tip clearance. Bindon [27] recommended investigating the reduction of losses by minimizing entropy generation and maximizing the flow deflection in the tip region, through the use of improved blade tip geometries.

1.7 Summary of Tip Leakage Flow in a Cascade

In conclusion, the studies given above present a broad picture of the tip leakage flow. A few general statements about the leakage flow within a cascade can be made from the above results.

- The leakage vortex introduces a characteristic overturning and underturning of the near-endwall flow with respect to the main passage flow.
Figure 1.7. Accumulated Loss Contributions to Tip-Endwall Clearance Loss for a Clearance of 2.5%, taken from Bindon, 1989

- The leakage flow is a function of the mass flow through the clearance, which is in turn dependent on the tip clearance gap height.

- The fluid near the blade tip is strongly accelerated around the pressure side corner into the clearance.

- Separation of the fluid off of the blade pressure-surface corner can occur depending on the blade pressure-surface corner radius.

- For blades with sharp tip edges, a virtually loss-less vena contracta and characteristic laminar separation bubble exist from the pressure side corner to approximately 20% of the blade thickness into the clearance. Reattachment is turbulent, and a shear layer is generated near the reattachment point.

- An endwall separation bubble is also present, which contributes to the clear-
ance gap loss.

- Away from the blade surface, a loss-free, inviscid region of flow exists within the clearance.

- Most of the clearance mass flow occurs in the half of the clearance height nearest to the endwall, while most of the tip clearance total pressure loss takes place near the tip surface.

- Losses are present within the clearance endwall and tip separation bubbles, tip leakage vortex, passage vortex, and horseshoe vortex, however the dominant loss is contained within the leakage vortex.

- Downstream mixing of the leakage vortex and interaction with the horseshoe and passage vortices add to the overall loss.

From these results, the tip clearance flow should be sensitive to reduction by flow control via separation bubble reattachment, weakening of the clearance shear layer, reduction of the clearance mass flow, or increased flow turning in the clearance.

1.8 Limitations of a Linear Turbine Cascade

Although the bulk of flow studies have focused on research within turbine cascades, there are a number of features that are found in rotating machinery that are not present with a cascade. By observing the differences between the flow through a linear cascade and an axial flow machine, the limitations of a cascade environment are briefly noted.

Generally, the tip clearance flow and leakage vortex formation are similar in an axial machine as in a stationary cascade. For both an axial turbine and a cascade, a separation bubble forms on the blade tip surface due to pressure-corner separation.
Also, the loss in the blade row rises substantially in the last 20% of axial blade chord for both [27, 29, 30].

However, in an axial flow rotating machine, the presence of blade rotation causes the leakage vortex to be pushed nearer to the endwall and blade suction surfaces than would be seen in a stationary cascade [31]. The decay of the leakage vortex as it convects downstream is faster in an axial machine. Also, the boundary layer was highly skewed and usually covered the entire tip leakage clearance, while in cascade experiments, the boundary layer height varied between experiments and may fill the entire clearance height or only a small fraction of it [29, 30, 31]. The interaction between the secondary flows and the blade and endwall boundary layers was significant in a rotating turbine, because the passage and leakage vortices entrained a large amount of boundary layer fluid. Additionally, compressibility is a factor that is not included in the cascade tests, however research on this effect is very limited [21].

Despite the simplification of the leakage flow within a cascade, the general similarities allow the use of a planar stationary cascade in place of a rotating axial stage. Therefore, the effects of rotation will not be discussed further.

1.9 Flow Control Methods

Flow control is traditionally divided into two categories: passive and active control. Passive flow control describes a steady state actuation device that alters the air flow in a fixed, permanent manner and offers no control authority input by the user. Active flow control describes an actuation device with variable forcing authority that can be tailored to the flow when required. Active flow control devices have the benefit of being able to be turned off when unneeded, unlike passive control methods. Additionally, active control can force the flow using either temporally steady or unsteady modes of operation. The flexibility of variable forcing would be beneficial during in-service operation, where an active flow control device could be
incorporated into a feedback loop that senses the state of the tip clearance flow and responds accordingly with the proper forcing amplitude and frequency.

1.9.1 Passive Flow Control

The initial treatment in the literature to mitigate the tip leakage vortex utilized passive flow control methods. The most common forms of passive flow control devices include squealer tips, winglets, steady blowing or suction, and labyrinth seals. Much research has been focused on the squealer tip geometry, particularly the suction-side squealer tip, because of its positive benefit [32, 33]. This research will investigate the effect of a partial suction-side squealer on the tip clearance flow, as a benchmark in assessing the merits of active flow control.

Squealers are effective because they increase the flow turning in the clearance region, reduce the mass flow through the clearance, causing a reduction in the strength of the tip leakage vortex and help reduce direct blade damage during a blade rub with the endwall. These effects have a direct impact in reducing the tip leakage flow loss. A squealer tip is a spanwise extension of a small portion of the blade tip outward from the blade tip pressure- or suction-surfaces toward the endwall. A suction-side squealer is an extension of the suction-surface profile outward, while a pressure-side squealer is the equivalent pressure-surface extension. A full squealer extends from blade leading edge to trailing edge, while a partial squealer extends over only some fraction of the full blade chord.

In the thesis by Dey [2], the effect of various types of flow control were investigated. Winglets, or partial shrouds, as well as squealer tips were found to be viable in altering the tip leakage flow. Pressure-side winglets reduced the leakage flow through the clearance by lowering the pressure differential across the blade surfaces. From an understanding of the leakage flow physics, this most likely lowered the amount of accelerated fluid that entered the clearance, and perhaps the size of
the tip separation bubble. Suction-side winglets did not show any increased benefit. Suction-side squealer tips produced lower momentum through the tip clearance than an untreated plain tip. This reduced the leakage vortex without affecting the passage vortex or the main passage flow away from the tip region. In testing different geometries, Dey noticed that the optimized squealer tip case consisted of a partial suction-side squealer tip extension that extended from 4% to 80% axial blade chord. In a separate study, Key and Arts [34] examined a double squealer tip and a suction-side squealer, finding that the velocities over the clearance as well as the aerodynamic losses attributed to the clearance flow were reduced.

It is also important to consider the effect that placing a squealer tip on the blade has on the thermal environment around and in the clearance. Aft of the combustor, the operating gas temperature in a turbine stage can reach upwards of 1600 K with blade temperatures as high as 1200 K [35]. Turbine blades undergo high thermal stresses and fatigue, and are difficult to cool. In this harsh, high temperature environment, hot gases may recirculate at the corner of the squealer and blade tip surface. The recirculation may lead to hot spots that could cause oxidation and burnout of the squealer tip and blade surface. Any device that improves the aerodynamics of the tip flow through a flow blockage will likely have an adverse effect on the heat transfer properties.

Nasir et al. [36] examined the effect a squealer tip on the heat transfer to a turbine rotor at various clearance heights. The authors found that in comparison to a flat tip blade, a pressure-side squealer had increased the heat transfer coefficient, whereas a suction-side squealer tip showed a decrease in the heat transfer coefficient. Increased coefficients would lead to increased heating transferred to the blade surface and greater cooling difficulties. For small clearances, the occurrence of a squealer rub with the endwall may partially remove or completely destroy the squealer tip. This would reduce the aerodynamic effectiveness of the squealer tip.
While squealer tips do have some unwanted effects, the aerodynamic gain of these devices is notable. With an understanding of the tip clearance flow, squealer tips can be improved upon by incorporating only their positive effects into an active flow control solution, which can be activated only when aerodynamically necessary. The utilization of the benefits of the squealer tips in active flow control devices is the objective of this research. Active flow control will be discussed below.

1.9.2 Active Flow Control

Active flow control methods have been applied to the tip clearance, including steady and unsteady blowing and suction, endwall clearance control through heating or cooling of the endwall, and synthetic jets [37, 38, 39, 40]. However, these methods are undesirable for a variety of reasons. For blowing, suction, and use of thermal expansion or contraction of the endwall, there is an increase in the overall weight of the engine from the addition of the necessary piping and valve systems for these methods. In addition, the use of blowing requires air to be bled from the engine compressor, which reduces the overall engine efficiency because the air does not contribute to rotation of the turbine rotor blades.

Another type of actuation that was favorably applied in many other flow studies, including trailing edge separation control in low pressure turbines, leading edge separation control of helicopter retreating blade stall, and boundary layer instabilities [41, 42, 43, 44, 45, 46] is the weakly-ionized single dielectric barrier discharge (SDBD). The SDBD plasma actuator works by inducing a body force on the fluid and the creation of near-wall flow velocity on a surface.

The plasma actuator appeared to be a feasible method of reducing the tip leakage clearance flow loss, and so was benchmarked against the passive squealer flow control. Plasma flow control is advantageous because the actuators are solid-state, compact, and may be activated only when needed. The specific passive and active
flow control designs used in this research are described in Chapter 3.

1.10 Objectives of Current Research

Based on the previous research in the study of the tip clearance flow and associated near-wall flow, the current research consisted of a feasibility study of the use of plasma actuator and squealer tip flow control on the tip clearance flow.

The overall goals of this research are fourfold in nature:

1. Understand the state of the baseline flow near the tip clearance within a linear turbine cascade over a range of inflow parameters and tip clearance heights,

2. Determine the physical mechanisms by which the turbine tip clearance flow is receptive to flow control,

3. Increase the cascade efficiency by decreasing losses within the tip gap flow through the use of plasma flow control, and

4. Determine the range of applicability of flow control with regard to inflow parameters and tip clearance height.

The experimental facility setup and the details of the flow measurement techniques used for this research are described in Chapter 2. The flow control designs are discussed in Chapter 3. These objectives are addressed using endwall boundary layer surveys (Chapter 4), endwall and blade tip surface flow visualization (Chapter 5), blade surface and blade tip pressure measurements (Chapter 6), and downstream blade wake surveys (Chapter 7). The final conclusions are given in Chapter 8.
CHAPTER 2

EXPERIMENTAL FACILITY AND TEST CONDITIONS

2.1 Introduction

The tip clearance leakage flow was investigated in a large-scale, linear turbine cascade at the Hessert Laboratory of the University of Notre Dame. This chapter explains the experimental methods and test conditions used in this research.

2.2 Experimental Facility

The tip clearance flow was experimentally investigated in a wind tunnel that incorporated a stationary, rectilinear cascade of nine Pratt & Whitney Pack-B blades, with nonrotating, fixed endwalls. A diagram of the tunnel is shown in Figure 2.1. The cascade geometry is given in Figure 2.2.

This facility is an open-return tunnel that was originally designed and built for research involving low pressure turbine (LPT) blade suction surface separation control by Huang, Corke, and Thomas [41, 47]. The tunnel draws in air and discharges air to atmospheric pressure. The inlet flow conditioning consists of a 15.54 cm thick honeycomb with 6.35 mm diameter cells followed by five low-solidity screens. A 6:1 area ratio contraction with a 5th-order polynomial profile follows the inlet screens. After the contraction is the test section with the cascade array, which has a 0.927 m square cross-sectional dimension. The distance between the end of the contraction section to the center blade of the cascade array is 970 mm, or approximately 6.08
Figure 2.1. Low Pressure Turbine Cascade Tunnel

Figure 2.2. Schematic of the Blade Cascade Arrangement
times the axial blade chord ($c_x$). The cascade blades have inlet and exit flow angles of 55 and 30 degrees, producing a blade turning angle of 95 degrees. The stagger angle, $\theta$, is 26.16 degrees. Table 2.1 lists the relevant cascade properties.

Table 2.1

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. blades</td>
<td>9</td>
<td>$\alpha_i$</td>
<td>55°</td>
</tr>
<tr>
<td>$c$</td>
<td>0.1778 m</td>
<td>$\alpha_e$</td>
<td>30°</td>
</tr>
<tr>
<td>$c_x$</td>
<td>0.1596 m</td>
<td>$\theta$</td>
<td>26.16°</td>
</tr>
<tr>
<td>$s$</td>
<td>0.924 m</td>
<td>$t_{max}$</td>
<td>22.3 mm</td>
</tr>
<tr>
<td>$p$</td>
<td>0.141 m</td>
<td>$Zw$</td>
<td>1.08</td>
</tr>
</tbody>
</table>

The flow is drawn downstream past the cascade section through a diffuser by a 1.52 m diameter, 100 HP fan, that features manual variable pitch blades.

The Pack-B blade, seen in Figure 2.3, was designed to yield a blade pressure distribution that is representative of distributions typically found in a LPT stage. The blades have flat tips with unradiused, 90 degree edges.

The center blade of the array was used as the test blade in all measurements and was cantilevered to allow for a clearance at one end. The clearance between the end of the blade and wall was adjusted using shims to between one to four percent of the axial blade chord. The two blades adjacent to the center blade were cantilevered to a fixed clearance of 1.25% $c_x$ to provide reasonable boundary conditions for the center blade tip clearance. Four part-span spacers located at approximately 45% of span were used to hold these three blades in place. The spacers were made of 2.4 mm phenolic material that was machined to the pressure- and suction-surface contours of the blades. These can be seen in Figure 2.4.
Figure 2.3. Pack-B Airfoil Shape

(a) Wide View  (b) Planform View

Figure 2.4. Part-Span Spacers
Two geometrically similar blades were created for use as test blades in the center blade location.

The first blade incorporated static pressure tubulations cast into the blade to provide static taps on the suction and pressure surfaces at midspan and near the tip. The near-tip taps are shown in Figure 2.5. The midspan taps were located at the same axial chord positions as the near-tip taps. The locations will be given further on in this chapter.

The second blade was shorter in span to allow a tip endpiece, as shown in Figure 2.6(a) to be mounted to the blade. The endpiece formed the outermost part of the blade, over which the tip clearance flow leaked. Several tip pieces were created, each 38.1 mm in height, or 4.1% of the blade span, and had the same Pack-B airfoil shape as the rest of the cascade blades. Each endpiece incorporated a unique flow

![Figure 2.5. Images of the Pack-B Near-Tip Pressure Taps](image)

(a) Pressure-Side Taps

(b) Suction-Side Taps

Figure 2.5. Images of the Pack-B Near-Tip Pressure Taps
control device that was tested in the cascade facility.

Each tip piece had a cavity milled on the lower side of the part, so that it could receive and be adhered to a plug piece, as seen in Figure 2.6(b). The cavity sidewalls were 3.175 mm in thickness. The tip piece and plug combination was attached to the blade using two set screws fastened through the plug into the blade. This allowed the tip piece to be repeatedly affixed to the blade tip with a consistent tip clearance height. For this blade, the tip clearance was adjusted by placing airfoil-shaped plastic shims between the plug and the rest of the blade. The tip assembly, including the tip piece, plug, shims, and test blade can be seen in Figure 2.7.

![Tip Piece](image1.png)  
(a) Tip Piece

![Plug](image2.png)  
(b) Plug

Figure 2.6. Pack-B Tip Piece Drawings

The flow control methods used on the second blade will be addressed in Chapter 3. Because of the location of the shims and flow control devices at the tip, pressure taps were not incorporated into the second blade.
Figure 2.7. Image of the Pack-B Tip Piece Assembly
The cascade blades were created from a urethane mixture of equal parts polyol resin and isocyanate resin. The resins were thickened with molecular sieves and glass microspheres, poured into a mold, and allowed to cure. The endwall of the test section was made of Lexan plastic, while the sides of the test section were machined from Plexiglass. A removable endwall section as shown in Figure 2.8 was also made of Plexiglass to allow quick access to the tip clearance region of the center test blade. Static pressure taps were incorporated into this endwall section. The purpose and location of the pressure taps will be addressed later on in this chapter.

![Figure 2.8. Removable Endwall Section, with Static Pressure Taps](image)

2.3 Experimental Flow Conditions

Two trailing edge “tail boards” were located behind the innermost and outermost blades to allow for fine adjustment of the pressure distribution around the blades. The tail boards were set so the experimental pressure distribution matched that computed from a high Reynolds number, inviscid Euler numerical calculation by Susan-Resiga and Frunza [48, 49] for a two-dimensional infinite aspect ratio blade.
The tip flow was experimentally documented at four tip clearances of approximately 1, 2, 3, and 4% cₓ for four inflow velocity conditions given in Table 2.2. The blade Reynolds number was varied by changing the inflow velocity to the tunnel. The allowable operating range of the tunnel was confined between Re = 5.3 × 10⁴ and Re = 1.04 × 10⁵. This range was based on limitations of low flow velocity at the lowest Reynolds number and excess tunnel vibration at the highest Reynolds number. The tunnel vibration was due to the position of the “tail boards” that were used to adjust the pressure distribution over the Pack-B blades. The placement of the “tail boards” restricted some of the flow to the fan which caused unloading over part of the fan blades. The loading-unloading cycle created on the fan blades as they rotated caused the tunnel to vibrate. With these limitations on Reynolds number, the Reynolds number operating range was chosen to uniformly cover the full velocity range of the facility.

Table 2.2

<table>
<thead>
<tr>
<th>Test Condition</th>
<th>U∞ (m/s)</th>
<th>Re</th>
<th>M</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5.26</td>
<td>5.3 × 10⁴</td>
<td>0.015</td>
</tr>
<tr>
<td>2</td>
<td>6.84</td>
<td>6.9 × 10⁴</td>
<td>0.020</td>
</tr>
<tr>
<td>3</td>
<td>8.61</td>
<td>8.7 × 10⁴</td>
<td>0.025</td>
</tr>
<tr>
<td>4</td>
<td>10.27</td>
<td>1.04 × 10⁵</td>
<td>0.030</td>
</tr>
</tbody>
</table>

This experimental setup was beneficial for this research because of the large blade size involved and the large aspect ratio of the blades. The large blade size permitted detailed measurements of the tip clearance flow and wake structures of the vortices. The large aspect ratio of the blades allowed the secondary flow effects near the tip to be completely separated from the secondary flows at the blade fixed
2.4 Experimental Measurements

Various measurements were conducted in the LPT tunnel to understand the secondary flow through the turbine cascade. The important parameters of interest for the tip clearance flow include:

1. the Inflow Boundary Layer to the Clearance,
2. the Blade Pressure Distribution near the Tip Secondary Flows and at Midspan for a 2-D Flow,
3. the Leakage Flow Velocity through the Clearance,
4. the Downstream Total Pressure Loss due to the Secondary Losses, and
5. the Location of Separated Flow Regions on the Blade Tip and Endwall surfaces.

In order to characterize the relative effect of these parameters, the following experiments were conducted:

1. Hot-wire Anemometry Surveys of the Clearance Inflow Boundary Layer,
2. Ensemble-averaged Static Pressure measurements on the Blade Suction and Pressure Surfaces near the Tip and at Midspan,
3. Static Pressure Measurements on the Endwall Surface within the Clearance,
4. Oil Flow Visualization on the Endwall and Blade Tip Surfaces, and
5. Total and Static Pressure Surveys of the Tip Clearance Flow downstream of the Test Blade.

Each of these experimental investigations will be described in the following sections.
2.5 Inflow Endwall Boundary Layer Surveys

Hot-wire anemometry was used to measure the upstream boundary layer that enters the cascade in order to determine the characteristics of the flow that is drawn into the tip clearance. The effect of the inflow is important, as the clearance velocity is strongly affected by the state of the boundary layer.

Measurements were conducted at distances of 1.0, 1.5, and 2.0 \( c_x \) upstream from the leading edge of the center test blade of the cascade in the boundary layer of the endwall surface at the four test conditions listed in Table 2.2. Velocity measurements were recorded using a single constant-temperature hot-wire that was welded to a Dantec Dynamics boundary layer probe, model 55P15, as shown in Figure 2.9. The Tungsten wire had a length of 1.25 mm and a diameter of 5 \( \mu \text{m} \). It was oriented perpendicular to the mean inflow velocity and parallel to the endwall, as shown in Figure 2.10(a). An AA Lab Systems hot-wire anemometer system, model AN-1003, was used to operate the hot-wire at an overheat ratio of 1.65.

For each boundary layer profile the probe was traversed in the spanwise direction normal to the endwall surface using a computer-controlled linear traverse, shown in Figure 2.10(b). The survey covered a distance of 2.54 cm adjacent to the endwall surface. The spacing between survey points was 0.3175 mm, giving 80 measurement locations per survey. The hot-wire signal was acquired at 20kHz for 200K samples per survey point.

The probe was moved using a MicroMo Electronics two-phase miniature stepper
Figure 2.10. Boundary Layer Measurement Experimental Setup
motor, model AM1524, geared to a Dantec guide tube, part 55H136. This yielded a traverse step distance of 1.016 µm per step. The guide tube held a 4mm outer-diameter L-shaped probe support, Dantec part 55H22, that was inserted into the tunnel. The motor was operated using an in-house built motor controller. The traverse control and data acquisition of the hot-wire signal was performed by a 16-bit National Instruments analog-to-digital converter board, series model 6034E, using LabView.

The hot-wire was calibrated using an adjacent Pitot-static tube offset laterally 60 mm from the hot-wire, from a zero velocity condition up to the full Reynolds number range of the wind tunnel. Based on pre- and post-calibrations the measured variation in the freestream velocity was found to be only 1.22% \( U_\infty \).

2.6 Flow Visualization Techniques

The flowfield in the tip clearance of the center blade was studied by observing patterns of surface flow visualization mixtures that were applied to the endwall and blade tip surfaces within the clearance. The removable endwall in Figure 2.8 allowed easy access to the tip clearance to apply the mixtures.

2.6.1 Flow Visualization: Endwall Surface

For the endwall visualization, the flow visualization mixture consisted of 15 parts kerosene, 1 part TiO₂ powder, and 5 parts oleic acid. The three-part mixture was applied to the endwall using a foam brush, and when exposed to the oncoming flow, the TiO₂ was transported with the kerosene-oleic acid liquid in proportion to the local shear stress. Eventually the liquid evaporated, freezing the titanium dioxide pattern in place on the endwall. The endwall was then removed and the pattern photographed. Each test case was performed three times, and the three photographs were averaged by digitally overlaying the images. The averaging was done to remove
any small variations in the individual flow visualizations.

2.6.2 Flow Visualization: Blade Tip Surface

The surface flow visualization technique used on the blade tip involved applying a thin coating of methyl salicylate, also called “oil of wintergreen,” onto a pattern of ink dots, a technique credited to Langston and Boyle [51]. In this method, ink dots were applied in a pattern onto white contact paper using a permanent pen marker. The contact paper was then affixed to the flat blade tip and a thin coating of methyl salicylate was applied using an air brush, covering the dots. The ink is soluble in the methyl salicylate, so that when the flow was turned on, the ink dots moved in proportion to the local surface shear stress. The oil eventually evaporated leaving permanent ink tracks on the paper. The contact paper with the ink traces was then removed and digitally recorded using a computer scanner. Three runs were performed for each test case and the pattern of ink tracks for each of these were digitally averaged together.

2.7 Pressure Measurements

This section describes the method used to acquire static pressures on the blade and endwall surfaces as well as total and static pressures in the downstream blade wake. Pressures were measured using Validyne variable reluctance differential pressure transducers, all of type DP103, which featured interchangeable transducer diaphragms to accommodate various pressure levels. A companion carrier demodulator, model CD23, converted each transducer output to a measurable DC voltage. The pressure signals were digitized and stored to a computer using a United Electronics Incorporated analog-to-digital converter, model PD2-MFS-8-500/14DG with a 32kS on-board FIFO.

Tygon microbore tubing of 1.27 mm (0.05 in.) inner diameter and 2.29 mm
(0.09 in.) outer diameter was used to form the blade and endwall static pressure taps. This tubing was also used to connect the blade and endwall taps as well as the ports from the downstream probe to the pressure transducer. Because of the large number of surface pressure taps, a Scanivalve port multiplexer, model JS4-48, connected each port to the pressure transducer and the ports were read sequentially in order. The Scanivalve was computer controlled using a solenoid stepper driver, model CTRL2/S2-S6. The Scanivalve was housed in a homebuilt Faraday cage to shield the unit from external electromagnetic interference.

For the pressure acquisition, the following flow diagram in Figure 2.11 shows how the measurements were performed. The variables $k_1$ and $G_1$ denote the calibration constant (in. H$_2$O/Volt) and gain setting for the transducer during calibration. For each pressure port, the acquisition code recorded ensemble averages of 100K samples measured at 20kHz to obtain an ensemble average. Subsequent ensembles were recorded until the pressure converged to within 1%.

The following sections explain the type of pressure measurements that were performed which all use the described underlying pressure acquisition method. For the endwall and blade pressure measurements the Validyne transducer diaphragm # 14 was chosen, which has a dynamic range up to 0.89 in. H$_2$O.
Specify Transducer Gain ($G_2$)

Set Raw voltage = 0,
Set scan rate (SR),
Set number of scan samples (NSPC),
Set convergence value ($\epsilon = 0.01$).

Allocate computer memory in buffer for data storage,
Set Channel Average = 0.

Step Scanivalve to first Port (index j).

Set Total Average1 = 0,
Total Average2 = 0,
Channel Loop = 0.

Acquire NSPC at SR, compute mean and store to Channel Average.

Set Temporary Average = Total Average1,
Compute Total Average1 = (Total Average1 *(Channel Loop - 1) + Channel Average)/Channel Loop,
Set Total Average2 = Temporary Average, and compare: $Z = \frac{\text{Total Average}_1 - \text{Total Average}_2}{\text{Total Average}_1}$.

Is $Z < \epsilon$?

Yes

Record Raw Voltage[j] from converged Total Average1,
Record Channel Loop[j] from Channel Loop.

No

Find Pressure (in. H$_2$O): $P = F \times \text{Raw Voltage}[j]$, where $F = k_1 G_1 G_2$.

Store Pressure and number of iterations for each port.

Figure 2.11. Pressure Acquisition Flowchart Schematic
2.8 Endwall Surface Pressure Measurements

The endwall was instrumented with thirty-six static taps in order to survey the flow through the tip clearance. The location of the taps and their position in relation to the blade can be seen in Figure 2.12. A picture of the endwall with the taps was shown above in Figure 2.8.

![Figure 2.12. Endwall Surface Pressure Measurement Locations](image)

The static pressure was sampled using the pressure acquisition code defined in Section 2.7, and the results are given in terms of an endwall static pressure coefficient, defined in Equation 2.1 as

\[
c_p^{\text{wall}} = \frac{P_{\text{wall,s}} - P_i}{P_{\text{t}} - P_i}.
\]  

(2.1)

Each port pressure was sampled using the Scanivalve multiplexer and pressure transducer, and the static pressure coefficients were recorded to a file. The uncertainty in the pressure measurements is detailed in Appendix A, with a range of 1 to 5% depending on Reynolds number.
The pressure coefficient values were area-averaged to produce an average wall static pressure coefficient, denoted as $\overline{C_{pt}}$. To see the improvement in the area-averaged value with actuation, a percent difference equation was defined as

$$\Delta \overline{C_{pt}} = \frac{(\overline{C_{pt}}_b - \overline{C_{pt}}_a)}{\overline{C_{pt}}_b} \times 100 \text{ (%) (2.2)}$$

where subscript ‘a’ refers to the actuated flow control case and subscript ‘b’ denotes the baseline case. An increase in the area-averaged value of the wall static pressure coefficient with flow control will cause a resulting decrease in the percent difference $\Delta \overline{C_{pt}}$.

The endwall static pressure coefficient values were also area-averaged for only the clearance region directly under the blade tip, as it was thought that most of the pressure change occurred in this region. The blade area-averaged coefficients were given the symbol $\overline{C_{pt,bl}}$, and the percent change in the blade-area-averaged values is given by a similar equation as the one presented just above, defined as

$$\Delta \overline{C_{pt,bl}} = \frac{(\overline{C_{pt,bl}}_b - \overline{C_{pt,bl}}_a)}{\overline{C_{pt,bl}}_b} \times 100 \text{ (%) (2.3)}$$

where the subscript ‘bl’ refers to the blade-area and all the other terms are defined as before.

2.9 Blade Surface Pressure Measurements

The pressure on the blade pressure and suction surfaces was also important to record, in order to understand how the clearance affects the near-tip blade loading. The blade static pressure was acquired first with the urethane test blade, mentioned briefly above. This blade was cast with forty static taps on the pressure and suction surfaces near the blade tip, at a spanwise location of $H = 0.998$, and forty additional taps at midspan, $H = 0.5$. Here, $H = 1$ corresponds to the blade tip and $H = 0$ corresponds to the blade fixed end. Images of the near-tip taps were shown in
Figure 2.5. For each set of forty taps, thirty taps were located on the suction surface and ten taps were located on the pressure surface. The chordwise tap locations over the blade suction and pressure surfaces are listed in Table 2.3.

Table 2.3

PACK-B BLADE SURFACE PRESSURE PORT LOCATIONS

<table>
<thead>
<tr>
<th>Pressure Surface</th>
<th>Port</th>
<th>x/c (%)</th>
<th>Suction Surface</th>
<th>Port</th>
<th>x/c (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Surface</td>
<td>Port</td>
<td>x/c (%)</td>
<td>Suction Surface</td>
<td>Port</td>
<td>x/c (%)</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>1.0</td>
<td>11</td>
<td>50.0</td>
<td>21</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>11.0</td>
<td>21</td>
<td>75.0</td>
<td>22</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>21.0</td>
<td>12</td>
<td>52.5</td>
<td>23</td>
</tr>
<tr>
<td>4</td>
<td>4</td>
<td>31.0</td>
<td>13</td>
<td>55.0</td>
<td>24</td>
</tr>
<tr>
<td>5</td>
<td>5</td>
<td>41.0</td>
<td>14</td>
<td>57.5</td>
<td>25</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>51.0</td>
<td>15</td>
<td>60.0</td>
<td>26</td>
</tr>
<tr>
<td>7</td>
<td>7</td>
<td>61.0</td>
<td>16</td>
<td>62.5</td>
<td>27</td>
</tr>
<tr>
<td>8</td>
<td>8</td>
<td>71.0</td>
<td>17</td>
<td>65.0</td>
<td>28</td>
</tr>
<tr>
<td>9</td>
<td>9</td>
<td>81.0</td>
<td>18</td>
<td>67.5</td>
<td>29</td>
</tr>
<tr>
<td>10</td>
<td>10</td>
<td>91.0</td>
<td>19</td>
<td>70.0</td>
<td>30</td>
</tr>
</tbody>
</table>

Each pressure port was sampled in sequential order around the blade using the multiplexer and pressure transducer. For each measured static pressure, a blade static pressure coefficient was calculated as defined in Equation 2.4 by

\[ c_{ps} = \frac{P_{\text{blade},s} - P_i}{P_{ti} - P_i}, \] (2.4)

and the data was recorded to a file. The uncertainty in the blade pressure measurements varied between 1.4% and 5.7%. The details are given in Appendix A.
2.10 Wake Profiles: Multi-hole Probe Measurements

The ability to measure the downstream secondary flow affected by the presence of the tip clearance was very important, as this data would give the resultant effect of flow control on the tip clearance leakage flow. To measure the tip clearance flow, a five-hole pressure probe was installed downstream of the test blade tip clearance. The probe had a 3.175 mm in diameter and featured a cobra type conical shaped head, which allowed the probe to be traversed close to the endwall surface where the secondary flow exists. This probe was manufactured by United Sensor Corporation [52], model DC-125-12-A, and is shown in Figure 2.13.

The probe was placed downstream of the center test blade and oriented in the primary flow direction of the oncoming flow, in a non-nulling mode. The probe was traversed in the spanwise and pitchwise directions over a uniformly-spaced grid of points in a two dimensional plane oriented parallel to the blade array trailing edge plane.

A representation of the traverse measurement plane in relation to the cascade array is illustrated in Figure 2.14. The orientation and location of the probe with
respect to the test blade is shown in Figure 2.15.

![Figure 2.14. Pack-B Cascade Array Drawing](image)

The distance of the measurement plane downstream from the test blade in the streamwise direction was one axial chord \( (1 \, c_x) \). This value was chosen after surveying the flow at varying distances between \( 0.25 \, c_x \) to \( 1.2 \, c_x \) downstream. Due to the high gradients in the flowfield very close to the tip, measurements could not be recorded within the calibrated angle range of the probe for survey locations at close streamwise distances downstream of the blade. At \( 1 \, c_x \), where the measurements were recorded, the flow angles were not as high and the measurements were within the calibration range of the probe.

Movement of the probe was performed using a computer-controlled two-axis linear traverse comprised of two linear slide mechanisms from Velmex Inc. and two
Figure 2.15. Cascade Coordinate System Orientations
stepper motors with stepper motor drives from Applied Motion Products Inc. The pitchwise, horizontal slide was Velmex Unislide model number MB4030CJ-S4 and the spanwise, vertical slide was Velmex Unislide model number MB2515CJ-S2.5. The stepper motors were model number 4023-833D and the stepper motor drive model was PDO 3540. In addition, two linear variable differential transducers, purchased from Schaevitz Sensors, were incorporated as position feedback to the traverse, in order to verify the precise location of the probe, to within ± 0.254 mm. The horizontal transducer was model number DC-EC 5000 and the vertical transducer was model DC-EC 2000. These were powered by power supply model number PSD 4-15. The output from the transducers were passed through two voltage dividers to reduce the voltage signals from ± 10 V to ± 5 V for digitization by the data acquisition board.

The probe measured the local values of the total and static pressure. From the difference in the five port pressures the local velocity vector was computed. The upstream total pressure was also measured from a wall-mounted pitot-static tube. From the measured pressures, the mass averaged total pressure loss was calculated according to a method adapted from Yamamoto [53]. First, the pressure loss coefficient was calculated at each grid point, similar to that defined in Equation 1.2 from Chapter 1, but based on the downstream mass-averaged velocity, $v_{m2}$, as:

$$c_{pt} = \frac{(P_{ti} - P_{te})}{(\frac{1}{2}\rho v_{m2}^2)} \quad (2.5)$$

where $P_{ti}$ is the inlet total pressure, $P_{te}$ is the probe total pressure, $\rho$ is the density, and $v_{m2}$ is defined by

$$v_{m2} = \frac{\sum_i \sum_j (Vv_{ax}A)_{i,j}}{\sum_i \sum_j (Av_{ax})_{i,j}} \quad (2.6)$$

Here V is the local resultant velocity vector, and $v_{ax - ij}$ is the local streamwise velocity. The quantity $A_{ij}$ is the diamond-shaped local flow area around each grid
point as shown by the dashed lines in Figure 2.16. In this figure, for a given grid point, here shown as a square, this area extends out to the adjacent grid points surrounding the point under consideration. From these quantities, the overall total pressure loss coefficient is calculated as

\[ c_p \equiv \frac{\sum_i \sum_j (v_{ax} c_{pt} A)_{i,j}}{\sum_i \sum_j (A v_{ax})_{i,j}}. \] (2.7)

To see the change in the total pressure loss coefficient with applied flow control, the percent change is defined as

\[ \Delta c_p(\%) = \left( \frac{c_{pbaseline} - c_{pactuated}}{c_{pbaseline}} \right) \times 100, \] (2.8)

where a reduction in the total pressure loss coefficient with actuation compared to the baseline yields a positive value of \( \Delta c_p \).

With the acquisition of pressure and velocity at a number of equally spaced points in the flowfield the streamwise vorticity could also be computed. Streamwise vorticity was determined from the velocity gradients as

\[ \Omega_x = \frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}. \] (2.9)
Here, \( w \) and \( z \) are the Spanwise velocity and direction, while \( v \) and \( y \) are the Pitchwise velocity and direction. In this coordinate system, a positive streamwise vorticity occurs where the secondary flow had a counterclockwise rotation. A nondimensional form of Equation 2.9 was defined using the freestream inlet velocity and the kinematic viscosity:

\[
\hat{\Omega}_x = \frac{\Omega_x \nu}{U_\infty^2} = \frac{\nu}{U_\infty^2} \left( \frac{\partial w}{\partial y} - \frac{\partial v}{\partial z} \right). \tag{2.10}
\]

In addition to the calculation of vorticity, the identification of secondary vortices was also performed through the use of the \(-\lambda_2\) criterion, as given in the paper by Jeong and Hussain [54]. This method consistently identifies a vortex region as the zero contours of the second largest eigenvalue in the velocity gradient tensor that defines the two-dimensional flow measurement plane. With the velocity information, contours of the mass flow rate were also computed as

\[
\dot{m}_{vax} = \rho A_{ij} v_{ax}, \tag{2.11}
\]

where \( \rho \) is the fluid density, \( A_{ij} \) is the area around each grid point, and \( v_{ax} \) is the streamwise velocity at each point.

For the pressure measurements, ensemble averaged mean pressure readings were taken for the five probe ports and the upstream total pressure from the wall-mounted pitot-static tube. The pressures recorded were differential pressures that were referenced to the static pressure from the upstream pitot-static tube. For the five-hole probe readings, each ensemble consisted of 10K samples recorded at 5kHz, with a convergence in pressure of 1%. For the upstream dynamic pressure, each ensemble contained 50K samples recorded at 10kHz, with the same 1% convergence criteria.

There was some delay in the equalization in the pressure at the pressure transducer to the pressure measured by the probe ports. This was due to both the variation in pressure between adjacent grid points in the wake and the considerable
length of pressure tubing between the probe and the pressure transducer. Therefore, a settling time was added to allow the pressure within the transducer measurement volume to equalize to that measured by the probe. The value chosen was 23 seconds, as this was found to be suitable for the measured pressure signal to asymptotically approach the true measured pressure. The uncertainty in wake pressure measurements is given in Appendix A. The uncertainty in the total pressure loss coefficient ranged between 0.14% and 2.8%, depending on Reynolds number.

2.10.1 Five-Hole Probe Calibration Technique

The five-hole probe was calibrated at a Reynolds number of \(5 \times 10^4\) using a calibration method developed by Bryer and Pankhurst [55]. A schematic of the five-hole probe port configuration is shown in Figure 2.17, which shows the port numberings as viewed when looking at the front of the probe face. The port numbering described here follows that of Bryer and Pankhurst, which differs from the port numbering given by United Senor in Figure 2.13. Subscripts ‘1’ and ‘3’ represent the lower and upper static ports on the probe, while ‘2’ and ‘4’ represent the two side static ports on the probe. Subscript ‘5’ denotes the total pressure port on the probe.

Calibration consisted of rotating the probe over a large range of pitch (\(\alpha\)) and yaw (\(\beta\)) angles with respect to a known one-dimensional flowfield and recording all five port pressures for each combination of pitch and yaw angle. Yawing of the probe caused changes in the side port pressures ‘2’ and ‘4’, while Pitching of the probe caused changes in the upper and lower port pressures ‘1’ and ‘3’. A calibration map which correlated the pitch and yaw angles to the change in port pressures was then created, and is depicted in Figure 2.18. Each dot on this map corresponded to one calibration point, and the lines connecting each dot are lines of constant \(\alpha\) or constant \(\beta\). Pitch and yaw angles were interpolated using functions that relate the
Figure 2.17. Five-Hole Probe Port Location Schematic, from Bryer & Pankhurst [55]

\[ f(a) = \frac{(P_3 - P_1)}{(P_5 - P_m)} \]

\[ f(b) = \frac{(P_2 - P_4)}{(P_5 - P_m)} \]

Figure 2.18. Five-Hole Probe Calibration map for \( f(\alpha) \) vs. \( f(\beta) \)
angles to the port pressures, given by

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

(2.12)

and

\[ f(\beta) = \frac{P_2 - P_4}{P_5 - P_m}. \]  

(2.13)

Here, \( P_m \) is defined by

\[ P_m = \frac{P_1 + P_2 + P_3 + P_4}{4}. \]  

(2.14)

Static and dynamic flow coefficients over the range of interpolated flow angles and pressures from Equations 2.12 and 2.13 were defined by

\[ S_p = \frac{H_t - P_5}{P_5 - P_m}, \]  

(2.15)

\[ Q_p = \frac{P_5 - P_m}{\frac{1}{2} \rho V^2}, \]  

(2.16)

where \( H_t \) was the total pressure measured during the calibration from neighboring pitot tube. Contours of \( S_p \) and \( Q_p \) are shown in Figures 2.19 and 2.20.

When the probe was subjected to an unknown flow velocity at a point in the flow, the five-hole probe port pressures were sequentially recorded using the Scanivalve and pressure transducer, as described earlier. From these five pressure values, the flow angles were interpolated from Figure 2.18. Knowing the flow angles, the static and dynamic pressure coefficients were determined using Figures 2.19 and 2.20. From these two coefficients, Equations 2.15 and 2.16 were inverted to find the calibration-corrected dynamic pressure and static pressure at the point of interest. The velocity was computed from these two pressures using Bernoulli’s equation. Using this method, Bryer and Pankhurst state that the accuracy of the probe was within \( \pm 3\% \) of the measured velocity, provided that the range of \( \alpha \) and \( \beta \) were limited to \( \pm 25^\circ \). The majority of the flow vectors from this experiment remained within these limits.
Figure 2.19. Contour of the Static Pressure Coefficient over the Range of Yaw and Pitch Angles, for Re = 5×10^4

Figure 2.20. Contour of the Dynamic Pressure Coefficient over the Range of Yaw and Pitch Angles, for Re = 5×10^4
2.10.2 Five-Hole Probe Traverse Code

Shown in Figure 2.21 is a flow chart which depicts the traverse code used in the five-hole probe measurements. This code uses the pressure acquisition method described above in Figure 2.11.

![Flow Chart](image)

Figure 2.21. Five-Hole Probe Traverse Code Schematic
CHAPTER 3

FLOW CONTROL METHODS

This chapter describes the aerodynamic flow control devices that were implemented to reduce the losses related to the tip clearance. Because of the advantages of active flow control over passive control methods it was of interest to investigate active control in comparison with passive control to understand their relative impact on the tip clearance flow and associated losses. In this research both active and passive flow control were applied to the center blade of the cascade in order to affect the tip clearance flow, tip leakage vortex, and regions of separated flow within the tip clearance, which are the main loss production mechanisms. The aim of using flow control was only to affect these high-loss regions, while not influencing the horseshoe or passage vortices.

In the following sections, the types of passive and active flow control that were used during this research will be described, along with an explanation of the different test configurations that were investigated. In addition, the active flow control performance is documented, based on velocity measurements in the vicinity of the device.

3.1 Plain Tip

In both the active and passive flow control studies, the flow control devices were tested against a plain, flat tip blade to document the change in flow properties with
applied control. The plain tip attached to the free end of the second test blade installed as the center blade in the cascade, as described in Section 2.2 of Chapter 2 and shown in Figure 2.6(a). Several geometrically identical tip pieces were machined from Teflon Polytetrafluoroethylene (PTFE) or Macor machineable glass ceramic, both excellent dielectric materials. Dielectrics were chosen as a material because the plain tip was incorporated as part of the active flow control device described below.

3.2 Passive Flow Control: Suction-Side Squealer Tip

The passive flow control device used in this experiment was a partial suction-side squealer tip. The benefits of this flow control device were described in Chapter 1.

In this research, the squealer was investigated solely as a benchmark used to interpret the effectiveness of the active flow control. The partial suction-side squealer, seen in Figure 3.1, was made of a thin aluminum strip 2.0 mm (1.25% cₚ) in span-wise height and 0.64 mm in width, with an arc length of 173 mm. The strip was bent into the shape of the suction-side surface and bonded to the blade tip surface using cyanoacrylate glue. The squealer extended from approximately 4% to 83% axial chord and was offset from the suction-side surface approximately 1.5 mm.

![Figure 3.1. Suction Side Squealer Placement](image)
### 3.3 Active Flow Control: Plasma Actuation

The active flow control investigated in this research was the single dielectric barrier discharge (SDBD) plasma actuator. Prior research studies that incorporated SDBD plasma actuators for flow control were described in Chapter 1.

A SDBD plasma actuator is an electrohydrodynamic (EHD) device that injects momentum into an airflow by creating a localized wall jet on a surface, through an applied electric potential. The actuator consists of two metal electrodes arranged in an asymmetric fashion, separated by a dielectric layer. A cross-sectional schematic of the actuator is given in Figure 3.2. The lower or covered electrode is encapsulated underneath the dielectric surface while the upper or exposed electrode is open to the airflow where flow control is needed. When a high amplitude, high frequency a.c. voltage is applied across the electrodes, the electric potential generates an electric field gradient between the electrodes. Typical voltage values are on the order of a few tens of kilovolts and a frequency between one and ten kilohertz. At sufficiently high voltages the air between the upper electrode and dielectric surface breaks down, undergoing weak ionization (on the order of $10^{-5}$ [56]) into charged particles. The ionized air appears as a diffuse, uniformly distributed blue plasma, but occurs as a

![Figure 3.2. Schematic of the SDBD Plasma Actuator](image)

underneath the dielectric surface while the upper or exposed electrode is open to the airflow where flow control is needed. When a high amplitude, high frequency a.c. voltage is applied across the electrodes, the electric potential generates an electric field gradient between the electrodes. Typical voltage values are on the order of a few tens of kilovolts and a frequency between one and ten kilohertz. At sufficiently high voltages the air between the upper electrode and dielectric surface breaks down, undergoing weak ionization (on the order of $10^{-5}$ [56]) into charged particles. The ionized air appears as a diffuse, uniformly distributed blue plasma, but occurs as a
series of many filamentary microdischarges in which the charged particles move in order to try to neutralize the electric potential. A typical SDBD plasma discharge can be seen in Figure 3.3. During this reorganization of charges, the charged particles collide with the neutral gas. Baughn [57] states that the mean free path between ions is very small, on the order of 60 nm, and the collision rate of these ions with neutral particles occurs at $10^{10}$ collisions per second, so that the plasma acts on the entirety of the air and not only the ions.

The movement of plasma creates a localized wall jet, in which air is entrained along the dielectric surface as indicated by the arrow in Figure 3.2. This forcing is induced by the interaction of the generated electric field and the charged particle distribution. The direction of the induced flow is dictated by the asymmetric arrangement of the electrodes, and is always oriented from the upper electrode out across the dielectric surface toward the far side of the lower electrode.

3.3.1 Dielectric Barrier Discharge Physics

The previous section described the general characteristics of the dielectric barrier discharge. However, for use as a flow control device, it is important to understand the physical mechanism behind the SDBD plasma actuator. This section will give

Figure 3.3. Image of the SDBD Plasma
a more detailed description of the SDBD and develop the equations for the induced body force by the plasma on the air above the actuator. The following derivation is taken primarily from Enloe et al. [46] and Orlov [58], with additional information from Enloe et al. and Kogelschatz [45, 59].

The SDBD discharge is a quasi-neutral gas, and so is governed by Maxwell’s Equations:

\[
\nabla \times \vec{H} = \vec{j} + \frac{\partial \vec{D}}{\partial t},
\]

\[
\nabla \times \vec{E} = -\frac{\partial \vec{B}}{\partial t},
\]

\[
\nabla \cdot \vec{D} = \rho_c,
\]

\[
\nabla \cdot \vec{B} = 0,
\]

where \( \vec{H} \) is the magnetic field strength, \( \vec{E} \) is the electric field, \( \vec{j} \) is the electric current, \( \vec{D} \) is the electric induction, \( \vec{B} \) is the magnetic induction, and \( \rho_c \) is the charge density.

The charges within the plasma can be assumed to have sufficient time to redistribute themselves, so the problem is quasi-steady. This means that the set of four equations simplifies to an Electrostatic model

\[
\nabla \cdot \vec{D} = \rho_c
\]

because all other terms are zero. The electric induction is a function of the electric field by

\[
\vec{D} = \epsilon \vec{E},
\]

where \( \epsilon \) is the dielectric coefficient. The dielectric coefficient is defined as

\[
\epsilon = \kappa \epsilon_o,
\]

where \( \kappa \) is the relative static permittivity, also referred to as the dielectric constant,
and $\epsilon_o$ is the permittivity of free space ($\epsilon_o \approx 8.85 \times 10^{12}$ F/m). The electric field $\vec{E}$ is proportional to the gradient of the potential, $\phi$

$$\vec{E} = -\nabla \phi,$$

so the governing equation becomes

$$\nabla (\epsilon \nabla \phi) = -\frac{\rho_c}{\epsilon_o}$$

(3.10)

The net charge density at a particular point in the plasma is defined as the local charge imbalance created between positive ions and negative electrons

$$\rho_c = e(n_i - n_o)$$

(3.11)

where $n_e$ and $n_i$ are the electron and ion densities in the plasma. Using the quasi-steady assumption stated earlier, the charge densities can be related to the local electric potential through the Boltzmann relation

$$n_{i,e} = n_o \exp \left[ \mp \left( \frac{e\phi}{kT_{i,e}} \right) \right] \approx n_o \left[ 1 \mp \left( \frac{e\phi}{kT_{i,e}} \right) \right]$$

(3.12)

where $n_o$ is the background plasma density, k is Boltzmann’s constant, and T is the temperature of the particular charged particles. Here a minus sign applies to the ions while the plus sign applies to electrons. Then the net charge density is given by

$$\rho_c = e(n_i - n_o) \approx -en_o \left[ \frac{e\phi}{kT_i} + \frac{e\phi}{kT_e} \right].$$

(3.13)

Substituting Equation 3.13 into Equation 3.10 gives

$$\nabla (\epsilon \nabla \phi) = \frac{e^2n_o}{\epsilon_o} \left[ \frac{1}{kT_i} + \frac{1}{kT_e} \right] \phi = \frac{1}{\lambda_D^2} \phi,$$

(3.14)

with

$$\frac{1}{\lambda_D^2} = \frac{e^2n_o}{\epsilon_o} \left[ \frac{1}{kT_i} + \frac{1}{kT_e} \right],$$

(3.15)
where $\lambda_D$ is the Debye length, which is the characteristic length for electrostatic shielding in plasmas. At values greater than the Debye length, the plasma cancels the electric potential. In the immediate vicinity of the electrodes, for dimensions smaller than $\lambda_D$, the potential is not canceled and the charge density is nonzero. Then the governing equation becomes

$$\nabla(\nabla \phi) = -\frac{\rho_c}{\epsilon_o} = \frac{1}{\lambda_D^2} \phi. \quad (3.16)$$

Because there is a net charge density within the electric field, a Lorentz force is generated as

$$\vec{f}_b = \rho_c \vec{E} = -\frac{\epsilon_o}{\lambda_D^2} \phi \vec{E}. \quad (3.17)$$

where $\vec{f}_b$ is a volumetric quantity based on the volume of plasma generated between the two electrodes.

This body force will always be oriented in the direction from the exposed electrode toward the covered electrode. The body force is the product of the electric field and the electric potential. During each half cycle of the applied voltage waveform, both the electric potential and the electric field change sign. Thus the product of these two quantities is always of the same sign. Therefore the body force is then always directed in the same direction during both halves of the a.c. cycle.

### 3.3.2 Plasma Actuator Designs

With an understanding of the physical mechanism of the plasma actuator, the specific designs tested for this active flow control device will be given. As stated in Section 3.1, two materials were used to create the blade tip pieces: Teflon PTFE and Macor glass ceramic. These materials were chosen because of their high values of dielectric constant and dielectric strength.

From Equations 3.8 and 3.10, the net charge density is directly proportional to the material dielectric constant. So as the dielectric constant is increased, there is a
corresponding increase in the sustainable net charge density at a given electric potential, which results in a stronger body force and induced flow velocity. Therefore materials with a high dielectric constant are desired to maximize the momentum coupling. For Teflon and Macor the dielectric constants are 2.1 and 6.03. In comparison, air has a value of 1.0. Dielectric strength was also important, to ensure that the materials could withstand large peak potentials. Teflon PTFE has a dielectric strength of 500 V/mil (19.7 kV/mm) while Macor has a value of 785 V/mil (30.9 kV/mm) [60].

All of the actuator configurations used the plain tip listed above as the dielectric onto which the electrodes were applied. The electrodes were made of copper foil tape and were approximately 0.127 mm thick. The dielectric thickness between the electrodes was set to a fixed value of 3.175 mm by milling the tip endpiece cavity to the proper depth. The thickness of the dielectric resulted in dielectric breakdown voltages of 62.5 kV for Teflon and 98 kV for Macor.

The electrical input to the actuators was a 2.0 kHz a.c. positive-going ramp waveform, which is the upper signal in the voltage time traces shown in Figure 3.4. This waveform shape was chosen because it is very similar to a positive sawtooth wave, in that it exhibits a very high forward-going voltage slope. It has been shown by Enloe et al. [45] that a positive-going sawtooth input produces greater thrust at a given applied voltage than a similar but negative-going sawtooth waveform. The signal was set at an optimal value of 2.0kHz because this frequency maximized the body force achieved for the ramp waveform at a given peak voltage input while reducing the input power that was lost due to heating [61]. The power consumption for this type of actuator is relatively low, approximately 66 W/m, because the amperage draw is very low for the required high voltage. For these electrode configurations, this translates into a power consumption of roughly 10 W, which makes the actuator a low power option for flow control.
Six actuator designs were tested for plasma flow control. All plasma actuators were run in a steady mode of operation. The actuators utilized the same covered electrode geometry that was encapsulated within either a Macor or Teflon tip piece. The fixed location of the covered electrode is illustrated for one of the actuator configurations in Figure 3.5 by the dashed line. It was offset from the pressure and suction surfaces by 3.175 mm and extended from 4% to 79% axial chord, which fixed the axial extent of plasma formation to only over this region. The upper electrode geometries were changed for each actuator configuration to vary the direction and location of the induced actuator flow velocity.

The six designs are shown in Figures 3.6 and 3.7 and are described below in terms of the dielectric material, forcing direction, geometry of the exposed electrode(s), and the method of flow control. These will be referred to as Actuators 1 through 6 in the remainder of this document.

1. **Actuator 1: Pressure-Side Edge Tip Actuator**, Fig. 3.6(a)

   **Dielectric material:** Teflon PTFE.

   **Forcing direction:** Across the blade tip from pressure-side to suction-
Figure 3.5. Blade Tip Plasma Actuator Encapsulated Electrode Location, denoted by the dashed line

side, normal to the camber line.

**Electrode geometry:** Exposed electrode located on the tip surface at the pressure-side edge, which extended 3.6 mm in width toward the suction-side edge.

**Method of flow control:** Suppression of the tip surface separation bubble.

2. **Actuator 2: Pressure-Side Edge Tip Actuator,** Fig. 3.6(b)

**Dielectric material:** Macor.

**Forcing direction:** Identical to Actuator 1.

**Electrode geometry:** Identical to Actuator 1, except that electrode was cut back away from the pressure-side edge to reduce plasma formation on that corner of the electrode, where plasma was not desired. The cut back amount was approximately 1.8 mm.

**Method of flow control:** Identical to Actuator 1.

3. **Actuator 3: Partial Pressure-Side Edge Tip Actuator,** Fig. 3.6(c)
4. Actuator 4: Dual Pressure Surface/Pressure-Side Edge Tip Actuator, Fig. 3.7(a,b)

**Dielectric material:** Macor.

**Forcing direction:** Two-actuator design, incorporating Actuator 2 and an additional pressure-surface actuator facing toward the fixed end of the blade.

**Electrode geometry:** The pressure-side edge exposed electrode width was 2.8 mm. The pressure-surface actuator had an encapsulated electrode that covered the inner pressure-side surface of the cavity and was 19.1 mm in height. The exposed electrode for the pressure-surface actuator extended from the pressure-corner of the blade tip radially downward 4.8 mm. The two encapsulated electrodes as well as the two exposed electrodes were each electrically connected, so that both actuators were driven in phase with each other.

**Method of flow control:** Reduce the mass flow entering the clearance using the pressure-surface actuator and suppress the tip separation bubble using Actuator 2.

5. Actuator 5: Suction-Side Edge Tip Actuator, Fig. 3.7(c)

**Dielectric material:** Macor.

**Forcing direction:** Along the tip surface from suction-side to pressure-
side, in a direction normal to the camber line.

**ELECTRODE GEOMETRY:** Exposed electrode extended 2.8 mm away from suction-side edge and included a cutback amount of approximately 0.8 mm.

**METHOD OF FLOW CONTROL:** Mimicked the suction-side passive squealer by increasing the flow turning and opposing the leakage flow through the clearance.

6. **Actuator 6: Pressure-Side Edge/Suction-Side Edge Cavity Tip Actuator**, Fig. 3.7(d)

**DIELECTRIC MATERIAL:** Macor.

**FORCING DIRECTION:** From both the suction-side and pressure-side toward the blade camberline.

**ELECTRODE GEOMETRY:** Combination of Actuators 2 and 5.

**METHOD OF FLOW CONTROL:** Create an airflow from the tip toward the wall to obstruct the leakage flow through the tip. Opposing actuators should create a stagnation point at the camberline and a resulting upward motion of fluid as a jet of air toward the endwall.

In addition to the images above of the actuator configurations, the following images showing the regions of plasma formation for the four major actuator types (Actuator 1, 2, and 3: Pressure-Side Edge Tip, Actuator 4: Dual Pressure Surface/Pressure-Side Edge Tip, Actuator 5: Suction-Side Edge Tip, and Actuator 6: Pressure-Side Edge/Suction-Side Edge Cavity Tip) are pictured in Figure 3.8.
(a) Actuator 1: Pressure-Side Edge Tip Teflon Actuator

(b) Actuator 2: Pressure-Side Edge Tip Macor Actuator

(c) Actuator 3: Partial Pressure-Side Edge Tip Macor Actuator

Figure 3.6. Plasma Actuator Configurations
(a) Actuator 4: Dual Pressure Surface/Pressure-Side Edge Tip Macor Actuator, Top View

(b) Actuator 4: Dual Pressure Surface/Pressure-Side Edge Tip Macor Actuator, Side View

(c) Actuator 5: Suction-Side Edge Tip Macor Actuator

(d) Actuator 6: Pressure-Side Edge/Suction-Side Edge Cavity Tip Macor Actuator

Figure 3.7. Plasma Actuator Configurations
(a) Actuator 2: Pressure-Side Edge Tip Macor Actuator

(b) Actuator 4: Dual Pressure Surface/Pressure-Side Edge Tip Macor Actuator

(c) Actuator 5: Suction-Side Edge Tip Macor Actuator

(d) Actuator 6: Pressure-Side Edge/Suction-Side Edge Cavity Tip Macor Actuator

Figure 3.8. Images of Plasma Formation
3.3.3 Plasma Actuator Velocity Measurements

In order to understand the effect of the plasma actuator on the clearance flow, it was important to first document the performance of the blade tip actuator given the unusual three-dimensional blade-shaped actuator geometry. Also important was the need to understand how the induced velocity profile might be affected by the presence of a finite clearance over the flow control device. The actuator works by entraining air from above the dielectric surface, therefore operating the actuator with a finite clearance should have a strong effect on the induced velocity distribution.

Therefore, mean velocity surveys of the actuator-induced flow field in quiescent air were recorded for three different clearance values over the blade tip. Actuator 1 was the actuator configuration that was tested, as shown in Figure 3.6(a). Measuring the velocity for various tip clearance values allowed an understanding of the effect of clearance height on the induced flow. Velocity surveys were also completed at the same three clearances for a ‘flat plate’ actuator, identical to Actuator 1 except for the size of the dielectric. This second actuator was created with the same curved electrode geometry as Actuator 1, however the electrodes were placed on a large, flat 3.175 mm thick dielectric surface of 300 mm width by 300 mm length. This actuator differed from Actuator 1 in that the step change in height the fluid experienced as it exited the clearance at the edge of the blade tip was removed. These measurements were used to understand the unique effect that the blade tip/cascade endwall flow geometry has on the flow control device.

The flow velocity was measured using a miniature 0.85 mm outer diameter glass Pitot probe mounted to the two-axis computer-controlled traversing mechanism described in Chapter 2. The three clearances tested were 2% c_x, 4% c_x, and an infinite clearance (uncovered actuator). The location of the measurements was at the 35.7% axial chord location, with the probe placed at the suction surface edge of the blade,
and oriented perpendicular to the camber line. For the flat plate actuator, the probe location was kept in the same position as for the blade tip test at the location where the suction surface would have existed, which is approximately 22 mm away from the exposed electrode.

First the velocity surveys over the blade tip with Actuator 1 will be considered. All measurements were taken with an applied actuator voltage of 39.4 kV peak-to-peak (pk-pk). The results are shown in Figure 3.9. The uncovered case was documented twice for repeatability. In the figure, the horizontal dotted lines represent the two positions of the endwall that correspond to 2% $c_x$ (3.19 mm) and 4% $c_x$ (6.38 mm) clearances. The actuator was located at the 0 mm vertical position.

![Figure 3.9. Mean Velocity Profiles for the Plasma Actuator at 40kV pk-pk, Steady, Positive Ramp Wave Operating at 2.0kHz](image)

Without the endwall present, the velocity profile (Open Circle and Asterisk Symbols) took on a nearly symmetric shape. The maximum velocity was approximately
2.7 m/s, which occurred at a vertical position of about 2.6 mm. When the endwall was added to produce a 4% $c_x$ clearance (Cross Symbols), the maximum velocity increased slightly to approximately 3 m/s, and the location of the maximum velocity moved down a small distance to approximately 2.3 mm above the actuator surface. Finally, when the clearance was closed to 2% $c_x$ (Star Symbols), the maximum jet velocity increased significantly to a value of approximately 4.4 m/s. For this condition the location of the maximum velocity decreased to approximately 1.6 mm above the actuator surface, which was about the mid-height of the clearance. In the case of the 2% $c_x$ clearance, it should be noted that the velocity falls to only 2 m/s near the endwall. It is thought in this case that the probe does not capture the flow closest to the wall where the velocity goes to zero.

These results demonstrated that without an external flow both the blade tip actuator induced velocity maximum and velocity profile depended on the clearance height. For a 4% $c_x$ clearance, the effect of the endwall appears to be negligible. However for a 2% $c_x$ clearance, the reduced clearance has a dramatic effect of increasing the velocity profile. This shows that the clearance height is a significant parameter in analyzing the effect of the flow control.

Next the results of the flat plate case are investigated. In the uncovered case the flat plate actuator (Diamond Symbols) exhibited a maximum velocity of 3.16 m/s at approximately 1.6 mm above the surface. However the overall velocity profile was slightly reduced from the uncovered blade-tip actuator case. For the 4% $c_x$ clearance flat plate actuator (Plus Symbols), the maximum velocity increased to 4.4 m/s at a height of 1.5 mm above the actuator surface.

In comparison of both 4% $c_x$ cases, the flat plate actuator had better flow acceleration near the dielectric surface, with a small performance reduction farther away, around 4 to 5 mm outward from the dielectric. This meant that the step change reduced the effectiveness of the blade tip actuator to force the fluid, probably be-
cause of the merging of the induced flow with the quiescent air off of the suction-side surface. In considering the flat plate actuator (Left Triangle and Hexagon Symbols) at a clearance value of 2% $c_x$, the flow was greatly reduced over any other of the flow conditions tested. The velocity profiles also were fairly uniform across the height of the clearance, showing a maximum velocity of 1.4 m/s, which may mean the flow was inhibited by the close proximity of the endwall to the dielectric surface. This is in contrast to the results for the blade tip actuator at a 2% $c_x$, where the flow strongly increased. It is thought that at this smaller clearance, the negative effect of the step height was completely reversed and for the flat plate actuator the flow was restricted, while for the blade tip actuator the fluid was able to leave the clearance without this restriction.

In general, the flow increased for the blade tip actuator with a decrease in clearance, which caused the maximum velocity to move slightly toward the dielectric surface. For the flat plate actuator case, the flow velocity increased with a reduction in clearance height, until a critical value between 4% $c_x$ and 2% $c_x$. At the critical value, the flow was impeded by the small clearance and the actuator effectiveness to entrain flow along the surface was greatly reduced. Therefore the change in step height for the tip actuator actually caused a flow improvement in the induced flow velocity for small clearances, while for larger clearances the step height created a modest reduction in the induced velocity for the blade tip actuator.

Finally, it should be noted that the maximum actuator-induced velocities documented in Figure 3.9 are of the same order of magnitude as the 5 to 10 m/s inlet velocities into the cascade. It will be important to keep in mind the characteristics of the actuator effectiveness presented here during the interpretation of the flow control results in later Chapters, as the flow control in the tip clearance is dependent on the actuator forcing velocity, the state of the inflow boundary layer, and the velocity of the inflow into the cascade array.
CHAPTER 4

INFLOW BOUNDARY LAYER RESULTS

4.1 Boundary Layer Profiles

Hot-wire anemometry was used to measure the boundary layer upstream to the cascade in order to determine the characteristics of the flow that entered the tip clearance. The experimental setup of the hot-wire measurements was discussed in Section 2.5 of Chapter 2.

Figures 4.1 and 4.2 present the velocity and turbulence intensity profiles for the four Reynolds numbers tested at three upstream locations (2.0 \(c_x\), 1.5 \(c_x\), and 1.0 \(c_x\)). In both figures, the y-axis height is normalized by the boundary layer height, \(\delta\), and the x-axis velocity is normalized by the freestream velocity, \(U_\infty\), for each Reynolds number case. Figure 4.1 also displays the extrema of the clearance height to boundary layer thickness ratio, \(\tau/\delta\), that were tested based on the ranges of the inlet boundary layer thickness and the tip clearance height values. This indicates the extent to which the boundary layer covered the clearance height for the range of clearances tested.

Figure 4.1 of the boundary layer velocity profile, will be examined first. What is initially noticeable about the profiles is that the lowest three Reynolds numbers tested: \(Re = 5.3 \times 10^4\) (Blue Symbols), \(Re = 6.9 \times 10^4\) (Red Symbols), and \(Re = 8.7 \times 10^4\) (Green Symbols), have a different profile shape than at \(Re = 1.03 \times 10^5\). Also, for these lower three Re values the velocity profiles collapse to a relatively
Figure 4.1. Boundary Layer Velocity Profiles normalized by the Layer Height ($y/\delta$) and Freestream Velocity ($U/U_\infty$), at 2.0 $c_x$ (○), 1.5 $c_x$ (▲), and 1.0 $c_x$ (+) Upstream of the Test Blade at $Re = 5.3 \times 10^4$ (Blue), $Re = 6.9 \times 10^4$ (Red), $Re = 8.7 \times 10^4$ (Green), and $Re = 1.03 \times 10^5$ (Black)
Figure 4.2. Boundary Layer Turbulence Intensity ($\sqrt{\bar{u}^2}/U_\infty$) normalized by the Layer Height ($y/\delta$) and Freestream Velocity ($U/U_\infty$), at 2.0 $c_x$ (o), 1.5 $c_x$ (△), and 1.0 $c_x$ (+) Upstream of the Test Blade at Re = 5.3 ×10⁴ (Blue), Re = 6.9 ×10⁴ (Red), Re = 8.7 ×10⁴ (Green), and Re = 1.03 ×10⁵ (Black)
uniform profile shape at all three upstream locations: $2.0 \, c_x \, (\circ)$, $1.5 \, c_x \, (\triangle)$, $1.0 \, c_x \, (+)$. This profile shape is characteristic of laminar flow. It is also important for these lowest three Re cases to note that the profiles do not collapse perfectly over each other when nondimensionalized, but that there is a small increase in the profile velocity as the Reynolds number increases, which may indicate a transition toward a turbulent flow as the inlet velocity is increased. At the highest Reynolds number of $\text{Re} = 1.03 \times 10^5$ (Black Symbols), the profiles are clearly more full near the endwall surface. This signifies that the boundary layer is turbulent at this Reynolds number condition.

Next the turbulence intensity profile will be discussed in Figure 4.2. For the lowest two Reynolds numbers, the turbulence intensity remains relatively flat over the entire boundary layer. For $\text{Re} = 8.7 \times 10^4$ the turbulence intensity grows significantly as the wall is approached. This compliments the idea above that as the Reynolds number is increased, the flow begins to transition from an initially laminar state to a turbulent boundary layer. However, the velocity profile shows only a small increase in the boundary layer velocity, so the flow is not fully turbulent. At $\text{Re} = 1.03 \times 10^5$, the turbulence intensity recedes from the higher values at $\text{Re} = 8.7 \times 10^4$, where it is thought that at the higher Reynolds number the flow has past the transition phase and is now fully turbulent, as indicated by the velocity profiles.

The overall state of the boundary layer can therefore be summarized by stating that the flow is laminar at $\text{Re} = 5.3 \times 10^4$ and $\text{Re} = 6.9 \times 10^4$, is transitional from laminar to turbulent flow at $\text{Re} = 8.7 \times 10^4$, and turbulent at $\text{Re} = 1.03 \times 10^5$. Based on the velocity profiles at all three upstream locations surveyed the flow does not change significantly in the streamwise direction for the lowest three Reynolds numbers. This indicates that if the flow is laminar or transitional, it is well established as it enters the tip clearance region of the test blade.

For these measurements, the exact location of the probe height from the wall
was unknown. The results at the lowest three Reynolds number cases indicated that the velocity profile near the wall was linear. For these cases, the velocity profiles were linearly extrapolated down to the wall \((y/\delta = 0)\), and the profile heights were adjusted in the vertical direction to meet the no-slip condition at the wall. The extrapolation to the wall consisted of fitting a line to the five data points closest to the wall for each mean-velocity profile. The distance between the origin and the y-axis crossing of the extrapolated line was the distance the profile was adjusted vertically. For the highest Reynolds number case of \(Re = 1.03 \times 10^5\), the flow was turbulent and the velocity profile was not linear near the wall, so for this case the absolute height above the wall was not adjusted.

### 4.2 Boundary Layer Characteristics

The overall characteristics of the boundary layer are important in describing the state of the boundary layer. For the cases tested, the flow properties are given in Table 4.1. The important parameters are the boundary layer height \(\delta\), the boundary layer displacement thickness, \(\delta^*\), the momentum thickness, \(\theta\), and the shape factor, \(H\). These are defined as:

\[
\delta = y \Big|_{u=0.99U_\infty}
\]

\[
\delta^* = \int_{y=0}^{\infty} \left(1 - \frac{u(y)}{U_\infty}\right) dy
\]

\[
\theta = \int_{y=0}^{\infty} \left(\frac{u(y)}{U_\infty}\right) \left(1 - \frac{u(y)}{U_\infty}\right) dy
\]

\[
H = \frac{\delta^*}{\theta}
\]

For the lowest three Reynolds numbers, \(\delta\) and \(\delta^*\) do not vary substantially over the three streamwise locations surveyed. At the highest Reynolds number, \(\delta\) and \(\delta^*\) both increase as the boundary layer develops from location 2.0 \(c_x\) to 1.0 \(c_x\). The
Table 4.1

BOUNDARY LAYER PROPERTIES UPSTREAM OF THE TEST BLADE LEADING EDGE

<table>
<thead>
<tr>
<th>Location</th>
<th>$U_{\infty}$ [m/s]</th>
<th>Re $\times 10^{-4}$</th>
<th>$\delta$ [mm]</th>
<th>$\delta^*$ [mm]</th>
<th>$\theta$ [mm]</th>
<th>$H = \delta^*/\theta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.0 $c_x$</td>
<td>5.24</td>
<td>5.3</td>
<td>8.32</td>
<td>2.98</td>
<td>1.18</td>
<td>2.52</td>
</tr>
<tr>
<td>1.5 $c_x$</td>
<td>5.20</td>
<td>5.3</td>
<td>9.12</td>
<td>3.05</td>
<td>1.19</td>
<td>2.57</td>
</tr>
<tr>
<td>1.0 $c_x$</td>
<td>5.22</td>
<td>5.3</td>
<td>9.11</td>
<td>3.09</td>
<td>1.19</td>
<td>2.60</td>
</tr>
<tr>
<td>2.0 $c_x$</td>
<td>6.81</td>
<td>6.9</td>
<td>8.24</td>
<td>2.80</td>
<td>1.12</td>
<td>2.49</td>
</tr>
<tr>
<td>1.5 $c_x$</td>
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<td>6.9</td>
<td>8.13</td>
<td>2.83</td>
<td>1.09</td>
<td>2.59</td>
</tr>
<tr>
<td>1.0 $c_x$</td>
<td>6.80</td>
<td>6.9</td>
<td>8.21</td>
<td>2.82</td>
<td>1.11</td>
<td>2.55</td>
</tr>
<tr>
<td>2.0 $c_x$</td>
<td>8.57</td>
<td>8.7</td>
<td>8.17</td>
<td>2.69</td>
<td>1.06</td>
<td>2.53</td>
</tr>
<tr>
<td>1.5 $c_x$</td>
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<td>8.6</td>
<td>8.41</td>
<td>2.67</td>
<td>1.14</td>
<td>2.33</td>
</tr>
<tr>
<td>1.0 $c_x$</td>
<td>8.51</td>
<td>8.6</td>
<td>9.37</td>
<td>2.82</td>
<td>1.15</td>
<td>2.45</td>
</tr>
<tr>
<td>2.0 $c_x$</td>
<td>10.25</td>
<td>10.4</td>
<td>11.43</td>
<td>1.53</td>
<td>1.17</td>
<td>1.31</td>
</tr>
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<td>10.11</td>
<td>10.2</td>
<td>12.70</td>
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<td>1.43</td>
<td>1.53</td>
</tr>
<tr>
<td>1.0 $c_x$</td>
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<td>10.3</td>
<td>14.29</td>
<td>2.28</td>
<td>1.55</td>
<td>1.47</td>
</tr>
</tbody>
</table>

boundary layer thickness at $Re = 1.03 \times 10^5$ also increased over the typical values seen for the three lowest Reynolds numbers. The momentum thickness, $\theta$, was constant over $Re = 5.3 \times 10^4$ to $8.7 \times 10^4$ at a value of around 1.1 to 1.2. Once the flow becomes turbulent at $Re = 1.03 \times 10^5$, the momentum thickness increased substantially, illustrating the increased entrainment of higher momentum freestream fluid into the boundary layer.

In looking at the shape factor, $H$, for lowest two Reynolds numbers the value is approximately 2.5 to 2.6 for all of the streamwise locations. These values are comparable to the value of $H = 2.59$ for a Blasius boundary layer, and the flow is proven to be laminar. At $Re = 8.7 \times 10^4$, $H$ decreases to approximately 2.3 - 2.5 where the flow is transitioning to a turbulent profile, and the shape factor deviates slightly from the Blasius value. For the highest Reynolds number, the shape factor
value decreases to about 1.4, which indicates the boundary layer has become fully turbulent. Thus lower shape factor value coupled with the significant increases in $\delta$ and $\theta$ all confirm the turbulent nature of the flow at this Reynolds number.

4.3 Pressure Gradient

The streamwise pressure gradient is another parameter of importance in characterizing the boundary layer beyond the mean velocity profile discussed above. To determine the pressure gradient for the lowest three Reynolds number cases where the flow is laminar, a series of Falkner-Skan similarity solutions were computed for a range of pressure gradient values. The Falkner-Skan family of boundary layer profiles describes a laminar boundary layer in the presence of a pressure gradient in the streamwise direction. The equation for Falkner-Skan flow is

$$f''' + ff'' + \beta (1 - (f'^2)) = 0,$$

(4.5)

where $f'$ is the boundary layer velocity normalized by the inlet velocity and $\beta$ is a function of the streamwise pressure gradient:

$$\beta = \frac{d \delta^2}{dx \rho u'}.$$

To understand how the pressure gradient varies, the solutions were fit to the boundary layer profiles and the gradient that most closely matched the experimental data determined the pressure value.

The velocity profiles and Falkner-Skan solutions are plotted in Figure 4.3 for four values of $\beta$. For the lowest two Reynolds numbers, the best-fit solution was a value of $\beta = 0$, which defines a zero-pressure-gradient Blasius boundary layer profile. The Blasius profile is denoted by the solid black line. As the profile begins to transition to a turbulent flow at $Re = 8.7 \times 10^4$ and becomes turbulent at $Re = 1.04 \times 10^5$, the application of the Blasius solution does not apply. This can be
Figure 4.3. Boundary Layer Velocity Profiles normalized by the Layer Height ($U/\delta$), at 2.0 $c_x$ ($\circ$), 1.5 $c_x$ ($\triangleleft$), and 1.0 $c_x$ ($+$) Upstream of the Test Blade at Re = 5.3 \times 10^4 (Blue), Re = 6.9 \times 10^4 (Red), Re = 8.7 \times 10^4 (Green), and Re = 1.03 \times 10^5 (Black) from Figure 4.1 with Falkner-Skan similarity solutions for various values of $\beta$, as defined in Equation 4.6
seen in the velocity profiles that both fall below the Blasius profile: \( \text{Re} = 8.7 \times 10^4 \) (Green Symbols) and \( \text{Re} = 1.04 \times 10^5 \) (Black Symbols). At \( \text{Re} = 8.7 \times 10^4 \) the flow began to transition to a turbulent flow, but was nearly laminar in nature. At \( \text{Re} = 1.04 \times 10^5 \), the flow was turbulent, and the laminar flow Falkner-Skan solution is not applicable. Thus the flow has a zero-pressure gradient for all Reynolds numbers tested.

### 4.4 Scaling of Inflow Velocity to Applied Actuation Level

It is worth stating that the blade tip clearances tested (1% \( c_x \) (1.60 mm), 2% \( c_x \) (3.19 mm), 3% \( c_x \) (4.79 mm), and 4% \( c_x \) (6.38 mm)) were only a fraction of the typical boundary layer heights tested (8 to 14 mm). This means that the boundary layer significantly covered the tip clearance for all Reynolds numbers and tip clearance values tested. Therefore the flow velocity that passed through the tip clearance region was greatly reduced from the mean freestream velocity values measured outside of the boundary layer.

The clearance velocity could not be measured directly, so the mean clearance velocity was estimated by averaging the boundary layer velocity profile data from Figure 4.1 recorded at 1 \( c_x \) upstream of the test blade leading edge over the tip clearance height of interest. Table 4.2 shows the calculated mean clearance velocity at each of the tip clearance and Reynolds numbers combinations tested. The mean clearance velocity, denoted as \( U_{\text{gap}} \), is a function of both of these parameters. The mean plasma actuator velocity was taken as the mean of the velocity profile for the tip actuator at a 2% clearance (Star Symbols) from Figure 3.9 in Chapter 3. The inlet velocity \( U_\infty \) and boundary layer height \( \delta \) values were also taken from the boundary layer data recorded at the 1 \( c_x \) upstream location.

The ratio of the mean clearance velocity to the mean inflow velocity varied greatly over the range of test conditions, from 0.17 to 0.70. The resulting ratio of
the mean forcing velocity to the mean clearance velocity varied from 3.1 to 0.38. The lower Reynolds number cases illustrate the greater effect the actuator forcing had on the flow relative to the clearance velocity, when compared with the higher Reynolds number tests. As the clearance is increased, the relative forcing level of the actuator was also diminished, as the mean clearance velocity increased.

Table 4.2

SCALING OF THE PLASMA ACTUATOR MEAN FORCING VELOCITY WITH MEAN INFLOW AND MEAN CLEARANCE VELOCITIES

<table>
<thead>
<tr>
<th>( r ) [mm]</th>
<th>( U_\infty ) [m/s]</th>
<th>( U_{gap} ) [m/s]</th>
<th>( U_a ) [m/s]</th>
<th>( U_{gap}/U_\infty )</th>
<th>( U_a/U_\infty )</th>
<th>( U_a/U_{gap} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.60</td>
<td>5.22</td>
<td>0.87</td>
<td>2.70</td>
<td>0.17</td>
<td>0.52</td>
<td>3.1</td>
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<tr>
<td></td>
<td>6.80</td>
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<td></td>
<td>0.18</td>
<td>0.40</td>
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</tr>
<tr>
<td></td>
<td>8.51</td>
<td>1.63</td>
<td></td>
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<td></td>
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</tr>
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<td></td>
<td>8.51</td>
<td>5.04</td>
<td></td>
<td>0.59</td>
<td>0.32</td>
<td>0.54</td>
</tr>
<tr>
<td></td>
<td>10.21</td>
<td>7.17</td>
<td></td>
<td>0.70</td>
<td>0.26</td>
<td>0.38</td>
</tr>
</tbody>
</table>
4.5 Normalized Velocity Spectral Density

Another metric that was used to characterize the inflow boundary layer was the velocity spectra of the fluctuations in the boundary layer. Viewing the spectral content is a good measure of the relative energy content of velocity fluctuations at varying heights in the boundary layer. The velocity spectra were computed at five distinct wall-normal heights within the boundary layer, for the four Reynolds number cases at the 1.0 c_x location. The spectra was computed using Welch’s averaged modified periodogram method of spectral estimation, as defined in MATLAB. This method divided the fluctuating velocity vector data made up of 200K points into equal sized records of 8192 points, with no overlap. Each record was then windowed using a Hamming window of the same length as the record. The resultant rms velocity was normalized by the freestream velocity. The frequency was normalized by the inflow freestream velocity and the boundary layer displacement height as $f\delta^*/U_\infty$.

Figures 4.4 through 4.7 show the results of the spectral analysis at the four tested Reynolds number conditions. The subplot in the upper left of the figures illustrates the boundary layer positions that were surveyed. The subplot in the upper right of the figures is the time trace history of the fluctuating velocity over the ten second sample period. The bottom subplot in the figures shows the resulting velocity spectra. The line color in each of the subplots indicates the height in the boundary layer where the spectra were computed. The magenta line is the farthest out from the wall, and represents the freestream velocity spectra, while the red line denotes a location very close to the wall.

For Re = 5.3 \times 10^4 in Figure 4.4, at the lowest frequencies from $10^{-3}$ to $10^{-2}$ the characteristic roll off of energy in the flow can be noticed. There is a noticeable peak around a nondimensional frequency of $1.5 \times 10^{-2}$ and many peaks for frequencies
Figure 4.4. Normalized Velocity Spectra at varying heights within the Boundary Layer, at 1.0 $c_x$ Upstream Location at Re = 5.3 $\times 10^4$
Figure 4.5. Normalized Velocity Spectra at varying heights within the Boundary Layer, at 1.0 c_\infty Upstream Location at Re = 6.9 \times 10^4
Figure 4.6. Normalized Velocity Spectra at varying heights within the Boundary Layer, at 1.0 c_x Upstream Location at Re = 8.7 \times 10^4
Figure 4.7. Normalized Velocity Spectra at varying heights within the Boundary Layer, at 1.0 c_x Upstream Location at Re = 1.03 \times 10^5
at or above $2 \times 10^{-2}$. The lowest frequency peak at $1.5 \times 10^{-2}$ decreases as the boundary layer position is increased. The multiple remaining sharp peaks to the right of this main peak are electromagnetic noise that occurred at 60 Hz and higher multiples. It is also important to note that the second boundary layer position as denoted by the blue line shows increased spectral levels over much of the frequency range, meaning the boundary layer velocity fluctuations are larger at this location than the other locations. Overall for this profile, the flow contains low levels of velocity fluctuations except at the lowest frequency for a few isolated positions in the layer.

Next at $Re = 6.9 \times 10^4$ in Figure 4.5, the lowest frequency peaks observed in Figure 4.4 are again noticeable. The magnitude of the peak height increases closer to the wall in the boundary layer, as before. For this profile the energy level over the range of frequencies was generally low as in the previous figure.

At $Re = 8.7 \times 10^4$ in Figure 4.6, the velocity spectra exhibit a different shape as compared to the lower two Reynolds numbers. Here is observed an increase in the energy content over most of the frequency range. As the boundary layer survey position is moved closer to the wall there is a growth of energy in higher frequencies. Drawn for reference on the spectra is a line signifying a $-5/3$ power law slope. This $-5/3$ slope characterizes the inertial subrange of the turbulent fluctuations in a boundary layer. These spectra show a clear $-5/3$ slope region which signifies the energetic turbulent fluctuations in accordance with Kolmogorov theory. In contrast to the laminar flow at the lower two Reynolds numbers, this further supports the transitory turbulent nature of the boundary layer at higher Reynolds numbers in this experiment.

At the highest Reynolds number where the boundary layer flow was clearly shown to be turbulent by the mean velocity profile, the spectra in Figure 4.7 show slightly increased broadband energy levels compared to the previous lower Reynolds
number. The same characteristic Kolmogorov inertial subrange scaling is observed, again signifying the energetic turbulent fluctuations in this case.

4.6 Summary

In conclusion, the approaching boundary layer to the linear cascade was shown to be laminar at the two lower Reynolds numbers. At the next highest Reynolds number the flow began to transition to a turbulent flow, although was still highly laminar in nature. At the highest Reynolds number the flow was shown to be turbulent. The blade clearance heights ranged from 1.60 mm to 6.38 mm. The boundary layer thicknesses ranged from 8.3 mm to 14.3 mm. Therefore in all cases the blade clearance height was smaller than the approaching boundary layer. Thus the local velocity near the gap was a fraction of the freestream velocity. The exact amount varied with the clearance height and the inflow Reynolds number.
CHAPTER 5

FLOW VISUALIZATION RESULTS

5.1 Flow Visualization Description

This chapter presents the results obtained by flow visualization performed on the blade tip and endwall surfaces. It is used in conjunction with the surface pressure measurements presented in the next chapter to characterize the coherent structures of the flow in the tip clearance region of the blade. The flow visualization results on the tip surface using the methyl salicylate ink-dot tracer method are shown together with the corresponding endwall surface flow visualization involving the kerosene, oleic acid, and TiO$_2$ mixture. Figures 5.1 and 5.2 illustrate the type of surface visualization patterns that were obtained for the two techniques applied to the blade tip and endwall. These are for a 2% clearance at $Re = 5.3 \times 10^4$. These figures will be used to present the general features of the flow field and, more importantly, illustrate how to interpret the flow visualization images that follow.

First Figure 5.1 will be examined. This shows the tip surface flow visualization. The streaklines of the individual dots indicate the magnitude and direction of the velocity shear stress on the blade tip. From these, a number of coherent features can be deduced.

Near the leading edge of the blade at an axial chord of 15%, the ink lines are long and directed approximately tangent to the mean camber line of the blade. In this region there is no indication of a tip surface separation bubble. Since the lines
are long, they indicate that the flow passes through the clearance gap near the tip, most likely without separating from the pressure-side edge.

![Blade Tip Surface Flow Visualization Reference Diagram, 2% Clearance, Re = 5.3 × 10^4](image)

Figure 5.1. Blade Tip Surface Flow Visualization Reference Diagram, 2% Clearance, Re = 5.3 × 10^4

On the pressure side of the blade tip, the boundary between the dark smeared region and the coherent dot traces suggests that the flow separates off of the pressure-side corner of the blade tip and reattaches on the tip surface. This pressure-side tip separation bubble and reattachment line have been highlighted in Figure 5.1. The dark smeared region is indicative of a flow recirculation tube that wraps along the pressure-side edge of the blade tip. Eventually the fluid in the bubble exits through the clearance near the trailing edge of the blade, into the tip leakage vortex region, which is marked on the figure.

On the suction side of the blade, the flow that passes through the clearance gap near the leading edge of the blade meets the oncoming main passage flow and rolls up to form the tip leakage vortex. The tip leakage vortex clings to the suction-side edge of the blade tip and, depending on the Reynolds number, can influence the flow in the clearance near the suction-side edge. The dark pooling of the ink dots near the suction-surface edge suggests that the flow might be recirculating on the
tip surface in the vicinity of the leakage vortex.

The endwall surface flow visualization that corresponds to the same flow conditions in Figure 5.1 is presented in Figure 5.2. For all of the endwall surface flow visualization images the position of the blade has been superimposed in yellow, to provide a reference to the location of the blade tip in the flow visualization patterns.

![Endwall Flow Visualization Reference Diagram](image)

Figure 5.2. Endwall Flow Visualization Reference Diagram, 2% Clearance, $Re = 5.3 \times 10^4$

The surface visualization on the endwall shows many coherent features. The first noticeable characteristic is the many horseshoe vortex structures on the suction-side of the blade. These were termed “lift-off” lines by Harrison [62] and are the result of multiple horseshoe vortices wrapping around the blade leading edge, as labeled in the figure.

Also evident in the endwall surface visualization is the pattern of a separation bubble that exists under the blade location. This separated flow region was discussed in Chapter 1 and is denoted in the schematic in Figure 1.6. This bubble is different from the tip surface separation bubble.

Another important feature to note in Figure 5.2 is the demarcation line that is
offset from the suction-side edge of the blade. Because the tip leakage vortex and the passage vortex have opposite signs of circulation, and are located adjacent to each other, the movement of fluid by these vortical structures forms an observable demarcation, or interface, line on the endwall surface. The location of this line marks the trajectory of the tip leakage vortex. In addition it may indicate the size and strength of the tip leakage vortex that can be useful in evaluating flow control approaches.

5.2 Baseline Flow Cases

With this introduction on the interpretation of the surface flow visualization, the results from the tip and endwall visualization images will now be examined for the different tip clearance and Reynolds number conditions.

5.2.1 Baseline 1% Clearance

Figures 5.3 and 5.4 show the flow visualization patterns for a 1% clearance at four Reynolds numbers from $5.3 \times 10^4$ to $1.04 \times 10^5$. Comparing the tip surface flow visualization images at this small 1% clearance with the reference case of a 2% clearance in Figure 5.1, it is notable that there is no visible separation/reattachment line on the blade tip surface for any of the Reynolds numbers tested. Near the leading edge of the blade, the flow initially moves across the tip tangential to the camber line. The flow through the gap is driven by the blade-tip pressure gradient. This flow first exits through the gap beginning at approximately 20% $x/c_x$. Comparing Figures 5.3(a-d), the characteristics of the clearance flow appear to be similar for all of the Reynolds numbers. This indicates that the small 1% gap is the dominating factor.

The associated endwall surface visualization images are shown in Figure 5.4. For these, it is observed that an endwall separation line is present and uniformly offset
Figure 5.3. Tip Surface Visualization, 1% Clearance, Plain Tip

(a) Re = $5.3 \times 10^4$
(b) Re = $6.9 \times 10^4$
(c) Re = $8.7 \times 10^4$
(d) Re = $1.04 \times 10^5$

Figure 5.4. Endwall Surface Visualization, 1% Clearance, Plain Tip

(a) Re = $5.3 \times 10^4$
(b) Re = $6.9 \times 10^4$
(c) Re = $8.7 \times 10^4$
(d) Re = $1.04 \times 10^5$
from the blade pressure-side edge. Less noticeable, but still detectable with the small clearance, is an indication of the presence of the tip leakage vortex. This is evident from the darker region in the surface visualization on the suction side of the blade.

In Figures 5.4(a&b), which corresponds to the two lower Reynolds numbers, the multiple “lift-off” lines at the leading edge are very visible, indicating strong horseshoe vortices. The passage and tip leakage vortex locations also are more noticeable at these Reynolds numbers. The demarcation line between the vortices has been indicated by the dashed white line.

At the larger Reynolds numbers of $8.7 \times 10^4$ and $1.04 \times 10^5$ in Figure 5.4(c&d), the “lift-off” lines of the horseshoe vortices are less visible. This may be due to the transitional and turbulent boundary layers that were documented at these two Reynolds numbers in the previous chapter. The thicker boundary layer is expected to weaken the horseshoe vortex. Also evident is the downward movement of the demarcation line towards the suction side of the blade clearance gap. This shows that as the Reynolds number increased, the tip leakage vortex is able to cling to the suction-surface corner of the blade tip for a greater axial extent.

The surface flow visualization also reveals the separation/reattachment line that occurs under the blade on the endwall. Its location and extent seem to be independent of the Reynolds number. The separation/reattachment line appears to extend to approximately 95% $x/c_x$. However the fluid that is ejected at the downstream end of the blade looks to be a function of the Reynolds number. The point of ejection seems to affect the trajectory of the leakage vortex. As will be evident with the other cases to follow, the ejection point will move upstream on the blade as the clearance height increases.
5.2.2 Baseline 2% Clearance

Figures 5.5 and 5.6 document the flow visualization for the 2% $c_x$ tip clearance. The tip surface visualization images in Figure 5.5 indicate that the separation bubble near the pressure-side edge of the blade extends from approximately 10% $x/c_x$ to 85% $x/c_x$. Its location and extent appears to be independent of the Reynolds number as was observed with the smaller 1% clearance.

The corresponding endwall flow visualization shown in Figure 5.6 reveals a number of noticeable differences that result from increasing the clearance from 1% to 2%. One of these is a more visible indication of the passage and tip clearance vortices adjacent to the suction side of the blade. The demarcation line that occurs between these two vortices is more visible than with the 1% $c_x$ clearance. This implies that the strength of these vortices has increased with the 2% clearance.

The endwall surface visualization reveals a similar separation/reattachment line under the blade. Its location and extent are different from the smaller gap case. In particular the downstream end where the recirculating fluid exits has moved slightly upstream. This coincided with an upstream movement of the location where the fluid is ejected near the blade trailing edge. As pointed out, this affects the trajectory of the leakage vortex. The characteristics of the separation/reattachment line are again observed to be weakly dependent on the Reynolds number, also again indicting the dominance of the small clearance gap.

5.2.3 Baseline 3% Clearance

The surface flow visualization images for the 3% $c_x$ clearance gap are presented in Figures 5.7 and 5.8. In Figure 5.7, it is evident that the blade tip separation/reattachment line has moved further under the tip towards the suction side of the blade, compared to the previous cases. This is especially evident near the 60% $x/c_x$ location.
Figure 5.5. Tip Surface Visualization, 2% Clearance, Plain Tip

(a) Re = 5.3 \times 10^4

(b) Re = 6.9 \times 10^4

(c) Re = 8.7 \times 10^4

(d) Re = 1.04 \times 10^5

Figure 5.6. Endwall Surface Visualization, 2% Clearance, Plain Tip

(a) Re = 5.3 \times 10^4

(b) Re = 6.9 \times 10^4

(c) Re = 8.7 \times 10^4

(d) Re = 1.04 \times 10^5
Figure 5.7. Tip Surface Visualization, 3% Clearance, Plain Tip

(a) Re = 5.3 × 10^4
(b) Re = 6.9 × 10^4
(c) Re = 8.7 × 10^4
(d) Re = 1.04 × 10^5

Figure 5.8. Endwall Surface Visualization, 3% Clearance, Plain Tip

(a) Re = 5.3 × 10^4
(b) Re = 6.9 × 10^4
(c) Re = 8.7 × 10^4
(d) Re = 1.04 × 10^5
The endwall surface flow visualization in Figure 5.8 indicate that the flow has not turned as much as with the smaller gap clearances. This is discernable from the location of the demarcation line that is now located further from the suction-side edge of the blade compared to the previous cases.

The endwall separation/reattachment line is clearly visible. Its location and extent are consistent with the blade-tip flow visualization. This reveals a further upstream movement of its downstream extent where fluid is ejected to the suction-side trailing-edge of the blade. In this case, the exit location is at approximately 85% x/c.<ref>

5.2.4 Baseline 4% Clearance

The surface flow visualization results for the large clearance investigated (4% c_x) are presented in Figures 5.9 and 5.10. These reveal further movement of the endwall separation/reattachment line towards the suction side of the blade. As a result, the separation bubble exits from under the blade at the earliest chord location, 80% x/c_x, compared to the previous cases with smaller clearances. This again was found to be independent of Reynolds number.

The endwall flow visualization indicates that the blade with the 4% clearance gap was least capable of turning the flow. This is apparent from the location of the demarcation line which has moved the farthest distance from the blade suction side compared to the previous cases.

5.3 Passive Control Case: Suction-Side Squealer Tip

The following presents flow visualization for blade tips that include a suction-side squealer tip. The tip clearances are the same as with the baseline case, so that a direct comparison can be made. These are limited to the endwall flow visualization because the presence of the squealer tip prohibited flow visualization from being
Figure 5.9. Tip Surface Visualization, 4% Clearance, Plain Tip

(a) Re = 5.3 × 10^4

(b) Re = 6.9 × 10^4

(c) Re = 8.7 × 10^4

(d) Re = 1.04 × 10^5

Figure 5.10. Endwall Surface Visualization, 4% Clearance, Plain Tip

(a) Re = 5.3 × 10^4

(b) Re = 6.9 × 10^4

(c) Re = 8.7 × 10^4

(d) Re = 1.04 × 10^5
performed on the tip surface.

5.3.1 2% Clearance With Squealer Tip

The effect of a squealer tip on the clearance flow was revealed in the endwall flow visualization images shown in Figure 5.11. The squealer tip geometry was discussed in Chapter 3 and was depicted in Figure 3.1. The squealer tip in this case had a height that occupied 66.5% of the clearance gap.

The endwall flow visualization revealed a dramatic change in the local flowfield with the squealer tip. In particular, the location of the passage vortex (marks 1d, 2d, 3b, and 4b) is observed to be closer to the suction side of the blade. This suggested that the squealer tip inhibited fluid from passing through the clearance gap. The tip leakage vortex resides between the passage vortex and the blade suction surface. Therefore it is fair to say that the size of the tip leakage vortex was also reduced by the presence of the squealer tip.

The endwall separation/reattachment line under the blade (marks 1e-1f, 2e-2f, 3e-3f, and 4e-4f) was still evident with the squealer tip, but its extent was shortened considerably. Specifically, the separation/reattachment line extended to 60% axial chord as compared to 85% for the baseline case. In the baseline case the flow exited from under the blade as indicated by mark (4d) in Figure 5.6(d). With the squealer tip, the flow exited the clearance gap at an angle that is more parallel to the trailing edge of the blade. This is especially evident in the high Reynolds number cases in Figures 5.11(c&d). This suggests that the squealer tip aided in turning the flow, which further reduces losses.

5.4 Active Control Case: Actuator 1: Pressure-Side Tip Actuator

Flow control using the plasma actuator on the blade tip is next presented. The results with plasma actuator designs 1 and 5, as described in Section 3.3.2 of Chap-
Figure 5.11. Endwall Surface Visualization, 2% Clearance, Case With Squealer Tip Height of 66.5% of the Clearance Gap
ter 3 are presented here. Again, the flow features are illustrated by visualization on the endwall under the blade.

5.4.1 2% Clearance with Actuator 1 Off

The flow visualization results with the passive actuator (without an applied voltage) are shown in Figure 5.12. These were documented to verify that the baseline state of the flow was not modified by the addition of the exposed electrode onto the blade tip surface.

A comparison of the baseline flow for the passive plasma actuator shown in Figure 5.12 comes by viewing the original baseline flow visualization images in Figure 5.6. For these, the relative locations of the horseshoe, tip, and passage vortices did not change with the addition of the exposed electrode. The location and extent of the endwall separation/reattachment line also remained the same. Therefore it can be concluded that the addition of the passive plasma actuator tip did not change the baseline clearance gap flow. From this, the influence of the plasma flow control can be understood and interpreted correctly by comparing the actuated cases to the baseline flow.

5.4.2 2% Clearance with Actuator 1 On

The effect of the plasma Actuator 1 on the tip-clearance flow is documented in the flow visualization records shown in Figures 5.13 and Figure 5.14. The plasma actuator was operated at 40kV pk-pk, which produced a maximum induced velocity of 4.4 m/s. Additional details of the induced velocity profile were given in Section 3.3.3 in Chapter 3.

Actuator configuration 1 was designed to suppress the tip separation bubble by producing a tangential wall jet directed from the pressure-side to the suction-side of the blade. Normally the tip separation bubble was evident by the separa-
Figure 5.12. Endwall Surface Visualization, 2% Clearance, Actuators Off
Figure 5.13. Endwall Surface Visualization, 2% Clearance, Re = $5.3 \times 10^4$, $6.9 \times 10^4$, Pressure-Side Plasma Actuator On
Figure 5.14. Endwall Surface Visualization, 2% Clearance, Re = $8.7 \times 10^4$, $1.04 \times 10^5$, Pressure-Side Plasma Actuator On
tion/reattachment line under the blade. In the surface visualization, it is observed that on the endwall with the flow control there were not one as before but two separation/reattachment lines. This indicates that there were then two smaller laterally spaced separation bubbles that form as a result of the plasma actuator. The two separation/reattachment lines are indicated by labels R2 and R3 in the Figures.

To illustrate the difference in the surface flow structure, Figure 5.15 shows two schematics of the separation/reattachment lines. The top schematic (a) shows the baseline plain tip. For the baseline case, the tip bubble and vena contracta induce the endwall separation bubble. The flow separates close to the entrance of the gap (S1), and reattaches somewhere downstream (R1). The separation/reattachment line observed in the flow visualization corresponds to this region. The bounds of the separation region are denoted by the two vertical, dashed lines.

The schematic for the plasma actuator effect is shown in Figure 5.15(b). It is speculated that the tip separation bubble is suppressed by the plasma actuator, weakening the vena contracta, but not completely eliminating the contraction of fluid as it enters the clearance gap. In this condition the endwall separation point (S2) remains fixed, but the reattachment point (R2) is moved closer to the pressure-side edge of the clearance. The pressure gradient was apparently not sufficient to maintain attached flow on the endwall so that a second smaller separation bubble formed immediately after the first, with a second separation point (R2) and reattachment point (R3). The extent of these separation regions are also shown with vertical, dashed lines in the figure. The total extent of the pair of separation bubbles appears to occupy more of the region under the blade, extending further towards the suction side of the blade.

Given the changes that occur in the clearance flow, the effect of the plasma actuator on the coherent vortices around the blade can now be examined. The first observation concerns the regular pattern of horseshoe vortices, shown in Figure 5.13
Figure 5.15. Tip Clearance Flow Physics for the Baseline Case and Plasma Actuated Tip Case. PS: Blade Pressure Surface, SS: Blade Suction Surface
Based on these, the plasma actuator had a minimal effect on the upstream endwall flowfield, similar to the action of the passive squealer tip.

A more important observation taken from the flow visualization involves the region near the trailing edge of the blade, shown in Figure 5.13. Here is observed the absence of a strong fluid ejection due to the single endwall separation bubble that was present in the baseline flow. Based on this, it is inferred that the plasma actuator reduced or re-distributed the fluid that was captured in the separation bubble under the blade and ejected near the blade trailing edge. The amount and location of ejected fluid was observed to affect the trajectory of the tip leakage vortex. This ultimately impacts the losses. From these points it is surmised that the plasma actuator configuration 1 had positively improved the tip clearance flow loss. Wake data that is presented later in Chapter 7 will clarify these suppositions.

5.5 Active Control Case: Actuator 5: Suction-Side Tip Actuator

Next, the casing surface flow visualization of the suction-side forcing using the plasma actuator configuration 5 is presented. This actuator configuration was used with a slightly larger gap clearance of 2.181% rather than the 2% value of the previous tests. This was due to limitations on the ability to set the clearance height precisely.

5.5.1 2.2% Clearance with Actuator 5 Off

The baseline flow visualization with the passive actuator is shown in Figure 5.16. Here are observed the same features that occurred with the baseline case at a 2% clearance, that was shown in Figure 5.6. For example, the existence of the endwall separation/reattachment line, the pronounced fluid ejection near the trailing edge of
the blade, and the demarcation line between the passage and tip-clearance vortices are all seen. Overall, the flow physics for this baseline case are essentially the same as the plain tip cases with the 2% clearance.

A point of note is the circular fluid mark that appears in these images at approximately 60% of axial chord. This is an artifact of the flow visualization technique, where the flow visualization fluid reacted with the acetone solvent that was used to clean the surface between test cases, which inadvertently remained on the surface during these baseline cases.

5.5.2 2.2% Clearance with Actuator 5 On

The results for plasma actuator configuration 5 are presented in the flow visualization records shown in Figures 5.17 and 5.18. This actuator was designed to perform in a similar manner as the suction-side squealer tip, in order to reduce the mass flow within the clearance region and aid in turning the flow under the blade.

In the flow visualization presented here, a higher forcing level of 42.2kV pk-pk was investigated, which induced a 44% higher velocity than Actuator 1. However, because Actuator 5 was designed to affect the flow in a fundamentally different way, it was not critical to match the forcing level of the previous configurations. In first examining the flow visualization at the lowest Reynolds number in Figure 5.17(a), it is observed that the demarcation line has moved very close to the suction surface of the blade compared to the baseline case. This indicates that the mass flow through the clearance gap has been substantially reduced in a manner similar to the passive squealer tip. Although this effect is most dramatic at the lowest Reynolds number, it is also clearly evident at Re = 6.9 × 10^4 case in Figure 5.17(b). At the highest two Reynolds numbers (Figure 5.18(a&b)), the flow control does not appreciably move the demarcation line.

Also noticeable is the additional separation bubble that forms near the suction-
Figure 5.16. Endwall Surface Visualization, 2.181% Clearance, Baseline

(a) $\text{Re} = 5.3 \times 10^4$

(b) $\text{Re} = 6.9 \times 10^4$

(c) $\text{Re} = 8.7 \times 10^4$

(d) $\text{Re} = 1.04 \times 10^5$
Figure 5.17. Endwall Surface Visualization, 2.181% Clearance, Re = $5.3 \times 10^4$, $6.9 \times 10^4$, Macor Suction Side Actuator, 42.2kV
Figure 5.18. Endwall Surface Visualization, 2.181% Clearance, Re = 8.7 \times 10^4, 1.04 \times 10^5, Macor Suction Side Actuator, 42.2kV
side edge of the blade. This bubble is indicated in Figures 5.17 and 5.18 by the separation line R2 and reattachment line at R3. This bubble is thought to result from the induced flow produced by the actuator that is counter to the leakage flow. This results in a flow separation/reattachment observed in the flow visualization. A similar effect was observed with the passive squealer tip.

Observable at all four Reynolds numbers is the pronounced reduction in the fluid ejection near the trailing edge of the blade. This has a strong effect on the trajectory of the tip leakage vortex and the related losses. Again, a similar favorable effect was seen for the passive squealer tip.

From these findings Actuator 5 is believed to have a beneficial effect on the flowfield by acting against the clearance flow in a manner that is similar to a passive squealer tip. The quantitative effect it has on the losses measured downstream will be presented in the wake measurement results in Chapter 7.
This chapter is an examination of the pressure measurements that were made on the blade near the tip, and on the endwall surface under the blade. The details of the pressure measurement locations were presented in Section 2.9 of Chapter 2. These measurements are a good indication of the clearance flow physics, as the majority of the tip clearance leakage fluid passes in the half of the tip clearance height closest to the endwall surface.

6.1 Blade Surface Pressure Measurements

The blade surface pressures at midspan (H=0.5) and near the tip (H=0.998) were measured to understand how the blade tip loading differed from the two-dimensional midspan loading, along with the sensitivity of the tip loading to changes in the tip clearance height and Reynolds number. These measurements were also compared with the pressure distribution for an infinite aspect ratio blade that was based on an Euler simulation, as described in Chapter 2.

6.1.1 Midspan Blade Pressure Distribution

The midspan pressure distributions are presented in Figure 6.1. These correspond to pressure coefficients, $c_p$, at different chord locations around the blade. The pressure coefficients are shown for two Reynolds numbers. The solid curve corre-
sponds to the Euler simulation for an infinite aspect ratio blade. It is observed that the \( c_p \) distribution on the pressure side of the blade matches the inviscid distribution well. There is also good agreement between the measured \( c_p \) distribution over a large portion of the suction side of the blade. However, at approximately \( x/c_x = 0.6 \), the suction-surface distribution deviates from the Euler distribution. This is due to a suction-surface separation bubble that occurs at these low Reynolds numbers [47] where the flow is not able to remain attached due to the high blade turning angle and strong adverse pressure gradient. It is known that the separation point is not dependent on Reynolds number, but the reattachment point is [47]. The location of the flow separation corresponds to where the \( c_p \) distribution becomes flat. The location of the flow reattachment occurs where the blade pressure returns to the Euler distribution [47, 63]. For \( Re = 5.4 \times 10^4 \), the reattachment occurs at approximately \( x/c_x = 0.96 \), while at \( Re = 1.05 \times 10^5 \), at around \( x/c_x = 0.87 \). This agrees well with measurements by Huang [47].

It should be noted that the tip clearance had no effect on the midspan blade pressure distribution. This is due to the large aspect ratio of the blades in the cascade, which are effectively two-dimensional away from the blade ends.

6.1.2 Near-Tip Blade Pressure Distribution

The pressure measurements near the blade tip at \( H = 0.998 \) are shown in Figure 6.2. Included for reference is the Euler distribution for an infinite aspect ratio blade. The Euler distribution is useful for highlighting differences that are associated with tip clearance effects.

The effect of the tip clearance at \( Re = 5.3 \times 10^4 \) is shown in Figure 6.2(a). In contrast to the mid-span pressure distribution, the pressure distribution near the blade tip falls inside that of the Euler distribution. The degree to which this occurs depends on the tip clearance gap.
Overall, the presence of the tip clearance acts to reduce the tip blade loading. Focusing first on the pressure-side distribution, increasing the tip clearance decreases the blade loading in a fairly uniform manner. On the suction-side of the blade, up to $x/c_x = 0.5$, the blade loading decreases uniformly with increasing blade clearance. However for $x/c_x > 0.5$, there is a different trend. In this region, increased blade clearance causes an increase in the blade tip loading with more negative $c_p$ values. This increase in blade tip loading is due to the tip leakage vortex, which is close to the suction-surface corner. There is also an increase in the suction peak with an increase in the blade clearance height, which agrees with the results reported by Sjolander [17].

The same measurements of the blade loading as a function of tip clearance for $Re = 1.03 \times 10^5$ are shown in Figure 6.2(b). At this Reynolds number, the same characteristics were observed as for the lower Reynolds number case just presented.
Overall the blade tip loading is highly sensitive to the clearance size. Generally increasing the clearance gap lowers the tip loading except on the suction side of the blade at $x/c_x > 0.5$, where the tip leakage vortex forms. In the region of the tip leakage vortex, increasing the clearance gap produces larger suction pressure. Flow control approaches which reduce the strength of the tip leakage vortex will then generally reduce the blade loading in the aft suction-surface region.

Next Figures 6.3 and 6.4 examine more closely the Reynolds number effects on the blade loading for a fixed blade clearance. Figure 6.3 corresponds to a 1.2% blade tip clearance. In this figure, on the pressure side of the blade there appears to be no dependence on Reynolds number for the range examined. On the suction side of the blade, up to approximately $x/c_x = 0.3$ there also appears to be no dependence on Reynolds number. However between $0.3 \leq x/c_x \leq 0.5$ there is observed a systematic decrease in the tip loading with increased Reynolds number. Then for $x/c_x > 0.5$ there is an increase in tip loading with increased Reynolds number. The former is associated with the tip leakage flow while the latter is associated with the tip leakage vortex.

The characteristics of the other tip clearance cases in Figures 6.3(b) and 6.4(c&d) are similar to the smaller tip clearance case. In general the effect of the leakage flow observed on the suction side at $0.3 \leq x/c_x \leq 0.5$ reduces the tip loading as the clearance height increases. Similarly, the effect of the tip leakage vortex observed for $x/c_x > 0.5$ increases the blade loading further as the clearance height increases. These effects are magnified as the Reynolds number increases.

Finally, with an increase in the blade tip clearance the increased effect of the tip leakage vortex produces a rearward shift in the blade pressure distribution, progressively increasing the pressure load towards the rear portion of the blade. This could possibly have structural implications on the blade design.
Figure 6.2. Effect of Tip Clearance on Near-tip (H = 0.998) Loading of Pack-B Blade at Re = 53K, 103K
Figure 6.3. Pack-B Blade Static Pressure Coefficient ($c_{ps}$) Distribution, Near-tip measurements (H = 0.998) at $\tau/c_x = 1.204\%$ and $1.899\%$
Figure 6.4. Pack-B Blade Static Pressure Coefficient ($c_{ps}$) Distribution, Near-tip measurements ($H = 0.998$) at $\tau/c_x = 2.833\%$ and $3.953\%$. 
6.2  Endwall Surface Pressure Measurements: Passive Control Cases

This section presents the results from surface pressure measurements on the endwall across from the blade. This includes cases with a passive suction-side squealer tip as well as all six plasma actuator designs. Also shown for comparison are the baseline cases with a flat blade tip. All of these cases were investigated for a tip clearance of approximately 2% $c_x$. The ability to set the tip clearance accurately at 2% $c_x$ was limited by the experimental setup, however this did not prove to be a critical issue.

The complete effect of the actuators on the tip clearance flow cannot be determined solely from these results alone. The effects of the actuator designs will be further understood and quantified through the downstream wake surveys presented in Chapter 7.

6.2.1  Suction-Side Squealer Tip, $\tau/c_x = 2.19\%$

The results with the passive suction-side squealer tip are presented first in Figures 6.5 and 6.6. In these the baseline is shown on the left side of the figure (a,c) and the results with the squealer tip are shown on the right side (b,d). The two lowest Reynolds numbers are shown in Figure 6.5. The two highest Reynolds numbers are shown in Figure 6.6.

The overall characteristics of the pressure distributions will be discussed for Figure 6.5(a). The contours correspond to constant levels of static pressure coefficient, $c_{p\text{-wall}}$, that was defined in Equation 2.1 of Chapter 2. A negative static pressure coefficient indicates a pressure that is lower than the static pressure in the freestream at the entrance to the linear cascade.

Based on the contours the lowest static pressure is observed to be near the center of the axial blade chord under the blade. In this region the static pressure coefficient
Figure 6.5. Endwall Static Pressure Coefficient, $c_{p\text{-wall}}$, 2.19% $\tau/c_x$, Suction-Side Squealer Tip, Re = $5.3 \times 10^4$, $6.9 \times 10^4$
Figure 6.6. Endwall Static Pressure Coefficient, $c_{p\text{-wall}}$, 2.19% $\tau/c_x$, Suction-Side Squealer Tip, $Re = 8.6 \times 10^4$, $1.03 \times 10^5$
reaches a value of approximately -5.0. It is in this region under the blade where the leakage flow is the largest.

The large negative $c_{p\text{-wall}}$ values near the last 20% of the blade also indicate a substantially higher local tip leakage flow velocity. In addition the lower $c_{p\text{-wall}}$ values in the region on the suction side of the blade near the trailing edge reflect the presence of the tip leakage vortex, which was observed to increase the blade tip loading.

Comparing the $c_{p\text{-wall}}$ distributions for the four Reynolds numbers, the characteristics of the baseline flow are similar. Thus the Reynolds number effect on the wall pressure coefficients was minimal. The effect of the suction-side squealer tip on the wall pressure distributions is shown on the right part of Figures 6.5(b,d), 6.6(b,d). The squealer tip produces a dramatic change in the endwall $c_{p\text{-wall}}$ contours. The $c_{p\text{-wall}}$ level present at the blade tip leading edge now extends well under the tip. In addition, the highly negative $c_{p\text{-wall}}$ region observed in the baseline cases is completely removed by the squealer tip. The squealer tip acts to turn the flow under the blade and to prevent part of the flow from leaking through the clearance. These effects are evident in the $c_{p\text{-wall}}$ distribution.

Also, the squealer tip appears to be more effective at higher Reynolds numbers. This is based on observing a larger region of higher $c_{p\text{-wall}}$ levels as the Reynolds number increased.
6.2.2 Suction-Side Squealer Tip, $\tau/c_x = 3.28\%$

It was important to investigate the effect of the clearance height on the squealer tip performance. Therefore a second height corresponding to 3.28% $c_x$ was examined. These results are presented in Figures 6.7 and 6.8.

For the baseline cases in subfigures (a,c) for both figures, what is most apparent is the reduction in the negative $c_{p-wall}$ values under the blade, compared with the previous cases for the smaller blade clearance. Otherwise the characteristics of the $c_{p-wall}$ distributions remained similar to the smaller clearance.

With the addition of the squealer tip the pressure coefficient change is less dramatic compared to the smaller blade clearance case. This illustrates that the squealer is less effective with the larger blade clearance. When interpreting this result it needs to be pointed out that the height of the squealer was fixed in both blade clearances cases. Therefore the effective clearance gap height left unobstructed by the squealer tip was larger for the larger blade clearance. This undoubtedly affected its performance. Perhaps an improved test design would have been to keep a fixed effective gap by increasing the height of the squealer. However in engine design this is not generally done.
Figure 6.7. Endwall Static Pressure Coefficient, $c_{p-wall}$, 3.28% $\tau/c_x$, Suction-Side Squealer Tip, $Re = 5.3\times10^4, 6.9\times10^4$
Figure 6.8. Endwall Static Pressure Coefficient, $c_{p-wall}$, 3.28% $\tau/c_x$, Suction-Side Squealer Tip, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
6.3 Endwall Surface Pressure Measurements: Active Control Cases

The results of the endwall surveys of the pressure coefficient for the plasma actuator cases are presented in this section. As with the previous results for the squealer tip, here the baseline distributions are presented on the left and the flow control cases are presented on the right. Results from all six of the plasma actuator configurations are presented. Descriptions of these are contained in Chapter 3, Section 3.3.2.

6.3.1 Actuator 1: Pressure-Side Edge, $\tau/c_x = 1.94\%$

The results for the Teflon Pressure-Side Edge Actuator are presented in Figures 6.9 and 6.10. The applied voltage level was set at 40kV pk-pk for this actuator, with the induced flow oriented from the pressure-side towards the suction-side of the blade. For the lowest Reynolds number of $\text{Re} = 5.3 \times 10^4$ (Figure 6.9(a,b)) with the actuator on, there is observed higher negative $c_{p-wall}$ levels in the centermost area under the blade. This extends over a greater area of the endwall surface compared to the baseline case. This indicates that the actuator augmented the clearance flow velocity.

As the Reynolds number was increased (Figures 6.9(c,d), ??(a-d)), the relative amplitude of the actuator with respect to the inflow velocity reduced. Therefore the effect of the actuator on the tip clearance flow was also reduced. This second point is evident from the $c_{p-wall}$ distributions that show that the peak negative $c_{p-wall}$ levels under the blade with the actuator on are not changed as significantly from the baseline case at the two higher Reynolds numbers in comparison to the lower Reynolds numbers.
Figure 6.9. Endwall Static Pressure Coefficient, $c_{p_{wall}}$, 1.94% $\tau/c_x$, Actuator 1: Teflon Pressure-Side Edge Actuator, $Re = 5.3 \times 10^4$, $6.9 \times 10^4$.
Figure 6.10. Endwall Static Pressure Coefficient, $c_{p\text{-wall}}$, 1.94% $\tau/c_x$, Actuator 1:
Teflon Pressure-Side Edge Actuator, Re $= 8.7 \times 10^4$, 1.04$ \times 10^5$
6.3.2 Actuator 2: Pressure-Side Edge, $\tau/c_x = 2.15\%$

The results for Pressure-Side Edge Actuator 2 with the Macor dielectric are shown in Figures 6.11 and 6.12. The applied voltage for this design was set at 37kV pk-pk, in order to match the inducted velocity generated by Actuator 1. As with Actuator 1, the results indicate a small increase in the peak negative $c_{p-wall}$ levels under the blade. As before, the effect is most prominent at the lower Reynolds numbers. At the higher Reynolds numbers, the effect of the actuator forcing appears to be minimal, according to the wall $c_{p-wall}$ levels.

Based on these findings, Actuators 1 and 2 may be useful at the low Reynolds numbers. Ultimately the effectiveness of these two actuator designs will be evaluated by the downstream wake measurements that are presented in Chapter 7.
Figure 6.11. Endwall Static Pressure Coefficient, $c_{p_{wall}}$, 2.15% $\tau/c_x$, Actuator 2: Macor Pressure-Side Edge Tip Actuator, $Re = 5.4 \times 10^4$, $7.0 \times 10^4$
Figure 6.12. Endwall Static Pressure Coefficient, $c_{p_{-wall}}$, 2.15% $\tau/c_x$, Actuator 2: Macor Pressure-Side Edge Tip Actuator, Re = $8.7 \times 10^4$, $1.05 \times 10^5$
6.3.3 Actuator 3: Partial Pressure-Side Edge, $\tau/c_x = 2.19\%$

The results of Actuator 3 on the $c_{p_{-\text{wall}}}$ distribution on the endwall under the blade are presented in Figures 6.13 and 6.14. The design of Actuator 3 was presented in Chapter 3, Section 3.3.2. This design was intended to investigate if the configuration used with Actuators 1 and 2 required the full axial extent of the blade in order to be effective. It was stated that the roll-up process of the tip leakage vortex was found to begin around 30% $c_x$ in a study by Tallman and Lakshminarayana [64]. Therefore, Actuator 3 was designed to extend from 32% to 79% $c_x$. The actuator voltage was the same (37kV pk-pk) as with Actuator 2.

Because Actuator 3 was designed to influence the development of the tip leakage vortex, attention should be focused on the $c_{p_{-\text{wall}}}$ distribution on the suction side of the blade near the trailing edge. At the two lower Reynolds numbers in Figure 6.13, there is a slight increase in the area of moderate negative $c_{p_{-\text{wall}}}$ levels adjacent to the blade suction surface at $x/c_x = 0.9$, but otherwise the wall pressure contours remain essentially unchanged with actuation. This trend is also evident at the higher two Reynolds numbers in Figure 6.14, although to a lesser extent. Thus this actuator illustrates that the flow is receptive to actuation only in the rear section of the blade, where the leakage vortex forms, however the impact of this actuator design is not significant.
Figure 6.13. Endwall Static Pressure Coefficient, $c_\text{p-wall}$, 2.19% $\tau/c_x$, Actuator 3: Partial-Pressure Side Tip Actuator, Re = $5.3\times10^4$, $6.9\times10^4$
Figure 6.14. Endwall Static Pressure Coefficient, $c_{p_{wall}}$, 2.19% $\tau/c_x$, Actuator 3: Partial-Pressure Side Tip Actuator, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
6.3.4 Actuator 4: Pressure Surface/Pressure-Side Edge, $\tau/c_z = 2.26\%$

The results of the endwall $c_{p\text{-}wall}$ measurements for Actuator 4 are shown in Figures 6.15 and 6.16. Actuator 4 was designed to suppress the tip-surface pressure-side separation bubble similar to Actuators 1 through 3 while at the same time reduce the mass flow entering the tip clearance region under the blade. To accomplish this the actuator incorporated the plasma actuator geometry of Actuator 2 and a radially-directed actuator located on the pressure-side surface of the blade, oriented in the spanwise direction away from the tip clearance. The intent for the radially-directed actuator was to divert the flow away from the clearance. The actuator applied voltage was 36kV pk-pk, similar to the values for the previous actuator configurations.

Figure 6.15(a,b) shows the $c_{p\text{-}wall}$ distributions for the lowest two Reynolds numbers. For these it is observed that the peak negative $c_{p\text{-}wall}$ levels under the blade were larger than those with Actuator 2. This change in the $c_{p\text{-}wall}$ levels between Actuators 2 and 4 can be inferred to be the result of the additional pressure-side actuator, as this was the only difference between the two actuation designs. It was thought that the radially-directed actuation would reduce the mass flow through the clearance gap. As a result, it was believed that the natural acceleration of the fluid into the tip clearance gap would be reduced with actuation. The increase in the peak values of the negative $c_{p\text{-}wall}$ however seems to indicate that the velocity through the gap is greater than in the baseline.

At the two higher Reynolds numbers (Figures 6.15(c,d) and 6.16(a-d)), an increase in the peak negative wall pressure static coefficient is also observed, although the effect is minimal. Of the four actuator designs discussed to this point, Actuator 4 generated the most effect on the flow through the clearance gap at the lowest Reynolds number. The overall effect of this actuation design on the wake
loss coefficient will be discussed in Chapter 7.

Figure 6.15. Endwall Static Pressure Coefficient, $c_{p_{\text{wall}}}$, 2.26% $\tau/c_x$, Actuator 4: Pressure Surface/Pressure-Side Edge Actuator, $Re = 5.3 \times 10^4, 6.9 \times 10^4$
Figure 6.16. Endwall Static Pressure Coefficient, $c_{p_{\text{wall}}}$, 2.26% $\tau/c_x$, Actuator 4: Pressure Surface/Pressure-Side Edge Actuator, Re = $8.6 \times 10^4$, $1.04 \times 10^5$
6.3.5 Actuator 5: Suction-Side Edge, $\tau/c_x = 2.18\%$

This section gives the results for the Suction-Side Edge Actuator 5, which was designed to emulate a passive partial suction-side squealer tip. The intent of this actuator was to create a velocity that opposed the leakage flow, in the direction perpendicular to the blade camber line. It was thought that this induced velocity would reduce the leakage mass flow through the clearance and improve the flow turning in the clearance gap region. For Actuator 5, two voltage levels were used: 36.4kV pk-pk and 42.2kV pk-pk. Evaluating the actuator at two voltages allowed the effect of forcing amplitude to be understood.

For the lower voltage of 36.4kV, the results are presented in Figures 6.17 and 6.18. At the lowest $Re = 5.3 \times 10^4$, Actuator 5 created a strong reduction in the peak negative $c_{p-wall}$ level under the blade that was similar to the effect of the partial squealer. This change was evident of all four of the Reynolds numbers. In addition to the change in the $c_{p-wall}$ distribution under the blade, a change in the $c_{p-wall}$ distribution on the suction side near the trailing edge was also observed. This suggests that the tip leakage vortex was also affected.

Figures 6.19 and 6.20 display the results of Actuator 5 with the higher applied voltage of 42.2kV. Here, a similar reduction in the negative $c_{p-wall}$ levels are observed for all four cases. At the lower two Reynolds numbers the higher applied voltage produces a greater reduction in the negative $c_{p-wall}$ levels than the lower applied voltage case. This result shows a clear dependence of the pressure coefficients under the blade on the actuator forcing level. This is undoubtedly due to the ratio of the actuator induced velocity to the velocity entering the clearance gap.

Overall, Actuator 5 appears to be the most promising design of a blade-mounted flow control device that has the potential for reducing the wake losses. Since it was designed to emulate a suction-side squealer tip, it may be referred to as a plasma
Figure 6.17. Endwall Static Pressure Coefficient, \( c_{p \text{-wall}} \), 2.18\% \( \tau / c_x \), Actuator 5: Suction-Side Edge Tip Actuator, \( \text{Re} = 5.3 \times 10^4, 6.9 \times 10^4 \)
Figure 6.18. Endwall Static Pressure Coefficient, $c_{p_{\text{wall}}}$, 2.18% $\tau/c_x$, Actuator 5: Suction-Side Edge Tip Actuator, $Re = 8.7 \times 10^4$, $1.04 \times 10^5$
Figure 6.19. Endwall Static Pressure Coefficient, $c_{p\text{-wall}}$, 2.18% $\tau/c_x$, Actuator 5: Suction-Side Edge Tip Actuator, $Re = 5.3 \times 10^4$, $6.9 \times 10^4$
Figure 6.20. Endwall Static Pressure Coefficient, $c_{p_{wall}}$, 2.18% $\tau/c_x$, Actuator 5: Suction-Side Edge Tip Actuator, Re = $8.6 \times 10^4$, $1.04 \times 10^5$
6.3.6 Actuator 6: Pressure-Side/Suction-Side Cavity, $\tau/c_x = 2.25\%$

The effect of Actuator 6 on the pressure coefficient distribution under the blade is presented in Figures 6.21 and 6.22. Actuator 6 combines the effects of Actuators 2 and 5, and was operated at 36.4kV pk-pk. The objective of this actuator design was to create a jet of air from the blade-tip surface towards the endwall that would block the flow trying to pass through the clearance gap. The design of the actuator was described in Section 3.3.2, Chapter 3.

![Figure 6.21](image)

Figure 6.21. Endwall Static Pressure Coefficient, $c_{p-wall}$, 2.25% $\tau/c_x$, Actuator 6: Cavity Tip Actuator, Re = $5.3 \times 10^4$, $6.9 \times 10^4$

The effect of Actuator 6 on the $c_{p-wall}$ distribution under the blade are presented
Figure 6.22. Endwall Static Pressure Coefficient, $c_{p-wall}$, 2.25% $\tau/c_x$, Actuator 6: Cavity Tip Actuator, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
in Figures 6.21 and 6.22 for the full range of Reynolds numbers. Overall the peak $-c_{p\text{-wall}}$ level is reduced under the center region of the blade indicating that the flow through the clearance gap was reduced. The effect is evident even at the largest of the Reynolds numbers.

The actuator however appears to have only a minimal effect on the $c_{p\text{-wall}}$ distribution on the suction side of the blade near the trailing edge. This suggests that the tip clearance vortex is not affected by Actuator 6. The effect of this actuator on the wake loss coefficients will be examined in Chapter 7.

6.4 Endwall Area-Averaged Results

This section provides further analysis of the preceding results based on the $c_{p\text{-wall}}$ distributions on the endwall for the various actuator designs. Figure 6.23 shows $c_{p\text{-wall}}$ values that were averaged over the total endwall area region for each of the actuator cases as a function of Reynolds number.

In examining these, the majority of the baseline cases are clustered around a $\overline{Cpt}$ value of -1.2. Actuators 1 - 4 (Diamond, Downward-pointing Triangle, Hexagram, Cross, and Left-pointing Triangle Symbols) all showed a larger negative average pressure coefficient than their respective baseline cases. This result indicates that the velocity through the clearance gap in these four cases was larger than the baseline.

Actuators 5 and 6 (Circle, Upward-pointing Triangle, and Star Symbols) and both passive suction-side squealer tips (Square and Plus Symbols) all had smaller negative average pressure coefficients than the baseline. This shows that the velocity through the clearance gap in these cases was smaller than the baseline.

Figure 6.24 gives the percent change with respect to the baseline for the actuator cases investigated. Here a positive change in $\overline{Cpt}$ with respect to the baseline is interpreted to mean that the average $\overline{Cpt}$ values were less negative than those of the baseline. A negative change indicates that the average $\overline{Cpt}$ values were more
negative than the baseline. Therefore, a positive change in $C_{pt}$ would be beneficial because it signifies a lower velocity through the clearance gap, which is considered to be better from the point of view of lowering the loss coefficient.

In Figure 6.24 it is observed that the passive squealer tip for a $2.19\% c_x$ (Square Symbols) produces the best result, with an average $C_{pt}$ improvement of approximately 50%. The next best result was for the passive squealer tip at $3.28\% c_x$ (Plus Symbols) with an average $C_{pt}$ improvement of approximately 33%. Following these passive devices, both Actuator 5 cases, the suction-side edge actuators (Upward-pointing Triangle and Circle Symbols), and Actuator 6, the pressure-side/suction-side cavity actuator (Star Symbols) were most effective with average $C_{pt}$ improvements of approximately 20%. For the Actuator 5 configuration, there was a clear dependence on the applied voltage level, with the higher voltage case producing a few percentage point greater improvement than the lower voltage case. There was also a clear Reynolds number dependence for Actuator 5, with the improvement in the flow decreasing as the Reynolds number increased. Actuator 6 also exhibited a Reynolds number dependence, although this actuator was not as effective as Actuator 5 at the lower Reynolds number.

Actuators 1 - 4 resulted in negative changes in the average $C_{pt}$ level compared to the baseline. Actuator 4, the pressure surface/pressure-side edge design, had the greatest negative change. Actuator 3, the partial pressure-side edge actuator, and Actuators 1 and 2, the longer pressure-side edge actuators, produced less negative changes with respect to the baseline. All of these actuators (1 - 4) are expected to perform poorly toward reducing the wake loss coefficient.
(a) Endwall Area-Averaged Static Pressure Coefficient

(b) Legend of Symbols

Figure 6.23. Endwall Area-Averaged Static Pressure Coefficient Results for Baseline and Actuated Flow Control Cases
(a) Endwall Area-Averaged Static Pressure Coefficient Change with Actuation

(b) Legend of Symbols

Figure 6.24. Percent Change in Endwall Area-Averaged Static Pressure Coefficient Results for Actuated Flow Control Cases over the Baseline Flow
6.5 Interpolated Blade Area-Averaged Results

The majority of the flow control only affected the wall pressure within the region under the blade clearance. This is evident from the flow control results previously discussed in this chapter, as the static pressure on the endwall outside of the region under the blade tip did not vary significantly. In order to focus on the change in the endwall pressure that was exclusive to the region covered by the blade tip, the endwall static pressure was interpolated within the blade profile only. An example of a full endwall pressure coefficient contour and an equivalent interpolated pressure coefficient contour in the region under the blade are shown in Figure 6.25.

![Contour plots](attachment:contour_plots.png)

(a) Endwall Region

(b) Blade Interpolated Region

Figure 6.25. Static Pressure Coefficient Contours over the Endwall Region and Interpolated over the Blade Region, 1.94% \( \tau/c_x \), Teflon Pressure-Side Edge Actuator 1, \( \text{Re} = 5.3 \times 10^4 \)

Figures 6.26 and 6.27 present the new blade-interpolated area-averaged pressure coefficient results. These results for the blade region will be denoted by the variable \( \overline{C_{pt_{bl}}} \). Figure 6.26 gives the average \( \overline{C_{pt_{bl}}} \) values. Figure 6.27 gives the percent change in the average \( \overline{C_{pt_{bl}}} \) values with respect to the baseline.

In Figure 6.26 the data exhibit the same trends as previously seen in Figure 6.23. The baseline cases all have values from \( \overline{C_{pt_{bl}}} = -2.4 \) to \(-3\). These values are
lower than those in the full endwall averages given above, which indicates that the pressure coefficient values under the blade were biased high by the pressure coefficients on the wall outside the blade region. For the blade-interpolated results, the two passive squealer tip cases (Square and Plus Symbols) and Actuators 5 and 6 (Circle, Upward-pointing Triangle, and Star Symbols) all show lower negative $\overline{C_{pt_{bl}}}$ values, while Actuators 1 - 4 (Diamond, Downward-pointing Triangle, Hexagram, Cross, and Left-pointing Triangle Symbols) all show larger negative $\overline{C_{pt_{bl}}}$ values.

More importantly, Figure 6.27 presents the percent change in the average $\overline{C_{pt_{bl}}}$ with respect to the baseline with flow control. Focusing only on the region under the blade, the values of $\Delta \overline{C_{pt_{bl}}}$ are greater than those in Figure 6.24. The passive squealer tip produces a 50% to 80% improvement in the endwall pressure coefficient depending on the clearance height. The suction-side edge plasma actuator (Actuator 5) creates from 16 to 40 percent improvement, depending on the Reynolds number and forcing level. Actuator 6 resulted in as much as an 18% improvement in the wall pressure coefficient. These results confirm that the majority of the flow control effect was localized under the blade tip.
Figure 6.26. Blade Area-Averaged Static Pressure Coefficient Results for Baseline and Actuated Flow Control Cases
(a) Blade Area-Averaged Static Pressure Coefficient Change with Actuation

1.94%, Teflon Actuator 1, 40kV, Blade
2.154%, Macor Actuator 2, 37kV, Blade
2.186%, Macor Actuator 3, 36.8kV, Blade
3.11%, Macor Actuator 4, 36kV, Blade
2.26%, Macor Actuator 4, 36.4kV, Blade
2.181%, Macor Actuator 5, 36.4kV, Blade
2.181%, Macor Actuator 5, 42.2kV, Blade
2.250%, Macor Actuator 6, 36.4kV, Blade
2.192%, Macor SS Squealer Tip, Blade
3.284%, Macor SS Squealer Tip, Blade

(b) Legend of Symbols

Figure 6.27. Percent Change in Blade Area-Averaged Static Pressure Coefficient Results for Actuated Flow Control Cases over the Baseline Flow
This chapter discusses the wake survey measurements taken using the five-hole Pitot probe. These pressure surveys were used to obtain the velocity field, from which the vorticity field was computed. In addition, the Pitot probe surveys were used to calculate the total pressure loss coefficient. This loss coefficient is the metric of merit for evaluating the different flow control approaches investigated.

7.1 Wake Profile Description

Three types of plots are presented to document the flow in the wake of the blade. The first plot type contains contours of the normalized mass flow rate, \( \dot{m}_{vax} \), that was defined in Chapter 2, Equation 2.11. The next plot is contours of the streamwise component of the normalized vorticity, \( \hat{\Omega}_x \), that was defined in Equation 2.10. Finally, the last contour plot is of the total pressure loss, \( c_{pt} \), that was defined in Equation 2.5, is presented.

Figure 7.1 shows a representative plot of contours of the total pressure loss at the downstream location of the measurements. This depicts the viewpoint and features that were generally observed in the measurements. The surveys were taken at a streamwise distance of 1 \( c_x \) downstream of the center blade, as depicted in Figure 2.15. The survey region extends in the horizontal direction from the blade trailing edge over approximately 0.8% of the passage on the suction side of
(a) Location of the Spatial Survey Region

(b) Wake Survey Figure Description, for Total Pressure Loss Coefficient, 2.18% $\tau/c_x$, $Re = 1.04 \times 10^5$, Baseline

Figure 7.1. Schematics of the Wake Measurement Plane
the center blade, and in the vertical direction from the endwall over approximately 0.1% of the blade span, an equivalent distance of 0.6 \( c_x \). The blade trailing edge is located at \( y/p = 0 \), which corresponds to the left edge of the plot. Here ‘\( p \)’ denotes the blade pitch spacing. The endwall is located at \( z/c_x = 0 \), which corresponds to the top edge of the plot. These locations are both indicated by thin lines in the image.

In addition to the solid-filled contours in the plot, velocity vectors are overlayed and are shown as white arrows. These vectors give an indication of the local time-averaged velocity field. Finally, contours of zero levels of the \(-\lambda_2\) are shown as solid white lines. The definition of the \(-\lambda_2\) criterion was discussed in Chapter 2. Its purpose is to highlight regions that contain a coherent vortical structure.

Next, three distinct vortical structures can be identified: the tip leakage vortex, passage vortex, and horseshoe vortex. The tip leakage vortex is centered at \( y/p = 0.5 \) and \( z/c_x = 0.075 \). The passage vortex is located at \( y/p = 0.375 \) and \( z/c_x = 0.32 \). The horseshoe vortex is located at \( y/p = 0.12, z/c_x = 0.51 \).

The tip leakage vortex and horseshoe vortex have clockwise circulation (negative vorticity) while the passage vortex has a counterclockwise circulation (positive vorticity). It is noted that the \(-\lambda_2\) contours locate the passage and horseshoe vortices well. It does not locate the tip leakage vortex as well. This is likely due to the lack of measurement points near the endwall, which makes the \(-\lambda_2\) calculation less accurate.

The plots in Figure 7.1 form the basis of analysis of the wake region of the blade. The primary quantity presented in the contours will change, but the overall plot presentation will be the same.
7.2 Passive Suction-Side Squealer

The effect of the passive squealer on the wake flow is documented in this section for two tip clearances of 2.19\% and 3.28\% $c_x$. This will include baseline cases that are used to illustrate the changes produced by the squealer tips.

The results for the smaller clearance are presented first in Figures 7.2 through 7.7. Figures 7.2 and 7.3 show constant level contours of the mass flow rate ($\dot{m}_{va_x}$) in the wake of the blade. The baseline cases are shown at the left in the figure. For these, there is an observable ‘S’-shaped profile. This is particularly evident at $Re = 5.3 \times 10^4$. With the addition of the squealer, shown on the right in the figure, the overall size and strength of the mass flow deficit associated with the tip leakage vortex was substantially reduced. The mass flow associated with the passage vortex remained approximately the same.

Figures 7.4 and 7.5 show contours of normalized streamwise vorticity. In the baseline cases shown on the left as before, the tip leakage vortex and passage vortex contained the highest levels of vorticity. The tip leakage vortex was also visible in the vorticity distribution. As the Reynolds number increased, the level of vorticity in these three vortical structures decreased. A reason for this was the change in the boundary layer approaching the cascade, which changed from laminar to turbulent between the lowest and highest Reynolds number. This affected the flow through the clearance gap as well as the mixing of these three vortical structures.

The effect of the squealer tip is shown by the plots on the right. When the squealer tip was applied, the level of vorticity in the tip leakage vortex was reduced considerably. This is consistent with the reduction in the velocity through the clearance gap that was observed with the endwall pressure surveys in the previous chapter. This was also observed in the flow visualization with the squealer tips.

Interestingly, with the application of the squealer tip, the resulting reduction
in the vorticity level in the tip leakage vortex led to an increase in the levels in the passage vortex and horseshoe vortex. This was especially apparent at Re = $5.3 \times 10^4$. At the higher Reynolds numbers, the passage vortex also moved further into the center of the passage channel, away from the blade surface.

Contours of the total pressure loss coefficient are shown in Figures 7.6 and 7.7. Again, the baseline cases are shown on the left. For these the tip leakage vortex is observed to contain the largest pressure loss coefficient compared to the other vortical structures. As Reynolds number was increased, the peak $c_{pt}$ level in the tip leakage vortex decreased by 35%.

With the addition of the squealer tip, the $c_{pt}$ level of the tip leakage vortex was significantly reduced at all of the Reynolds numbers. Therefore the squealer tip was highly effective at minimizing the pressure loss in the wake of the blade associated with the tip clearance vortex.
Figure 7.2. Suction-Side Squealer: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{\text{max}}$), 2.19% $\tau/c_x$, Re = 5.3$x10^4$, 6.8$x10^4$
Figure 7.3. Suction-Side Squealer: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{v_{ax}}$),
2.19$\%$ $\tau/c_x$, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
Figure 7.4. Suction-Side Squealer: Wake Survey Contours of Normalized Streamwise Vorticity ($\tilde{\Omega}_x$), 2.19% $\tau/c_x$, Re = $5.3 \times 10^4$, $6.8 \times 10^4$
Figure 7.5. Suction-Side Squealer: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.19% $\tau/c_x$, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
Figure 7.6. Suction-Side Squealer: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pt}$), 2.19% $\tau/c_x$, $Re = 5.3 \times 10^4$, $6.8 \times 10^4$
Figure 7.7. Suction-Side Squealer: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pt}$), 2.19% $\tau/c_x$, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
The results with the larger clearance gap of 3.28% $c_x$ are presented in Figures 7.8 through 7.10. It must be noted that at the highest two Reynolds numbers, because of the increased size of the clearance, the velocities in the wake were too large, causing the exceedingly high pressure transducer voltages to be clipped by the computer analog-to-digital converter. Therefore the results for $Re = 8.6 \times 10^4$ and $Re = 1.03 \times 10^5$ were not included in this section. Also, note that the scale of the contour plots for this larger clearance is different from the previous case. The scale was changed for this case in order to best present the contours over the full range of the values.

The contours of the mass flow rate for the larger clearance gap are shown in Figure 7.8. For the baseline cases, the larger clearance gap allowed the tip leakage vortex mass flow deficit to increase considerably compared to the smaller clearance case. With the addition of the squealer tip, there was a significant decrease in the mass flow deficit associated with the tip leakage vortex.

Contours of the normalized vorticity are shown in Figure 7.9. For this larger gap, it is observed in the baseline cases that the level of vorticity in the tip leakage vortex has increased. In addition, near the endwall a small region of positive-signed vorticity can be seen. This is thought to be indicative of a secondary vortex that was induced by the stronger tip leakage vortex near the wall. This effect was not observed with the smaller clearance. As the Reynolds number was increased, the size of this secondary vortex decreased.

With the addition of the squealer tip, the location of the tip leakage vortex was observed to move closer to the blade. The vorticity contours indicate that the horseshoe vortex and passage vortex locations were not affected by the squealer tip at this larger clearance gap.

Contours of the total pressure loss coefficient are shown in Figure 7.10. For the baseline cases, a considerable increase in the $c_{pt}$ levels associated with the tip leakage
vortex with the larger clearance gap was observed. In contrast to the loss in the tip leakage vortex, the losses associated with the passage vortex and horseshoe vortex were relatively minor. As with the smaller clearance gap case, increasing Reynolds number produced a decrease in the $c_{pt}$ loss levels of the tip leakage vortex.

With the addition of the squealer tip, the levels of the $c_{pt}$ associated with the tip leakage vortex were significantly reduced. This was observed over both of the Reynolds numbers. This is consistent with the vorticity and mass flow rate contours.

![Contour plots](attachment:contours.png)

(a) $Re = 5.3 \times 10^4$, Baseline
(b) $Re = 5.3 \times 10^4$, Squealer
(c) $Re = 6.8 \times 10^4$, Baseline
(d) $Re = 6.8 \times 10^4$, Squealer

Figure 7.8. Suction-Side Squealer: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{v_{ax}}$), $3.28\% \tau/c_x$, $Re = 5.3 \times 10^4, 6.8 \times 10^4$
Figure 7.9. Suction-Side Squealer: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 3.28% $\tau/c_x$, Re = $5.3\times10^4$, $6.8\times10^4$
Figure 7.10. Suction-Side Squealer: Wake Survey Contours of Total Pressure Loss Coefficient \(c_{pt}\), 3.28\% \(\tau/c_x\), \(Re = 5.3 \times 10^4, 6.8 \times 10^4\)
7.3 Actuator 1: Pressure-Side Edge Tip Actuator

This section presents wake measurements for the Teflon Pressure-Side Edge Tip Actuator 1. This actuator was operated at 40kV pk-pk, which limited its lifetime to only a few hours. Because of this, full spatial wake surveys could not be achieved. As a result only the tip leakage vortex region was measured for this actuator configuration. Since the primary vortical structure of interest was the tip leakage vortex, measuring only this smaller window area was thought to be a reasonable compromise of the experimental procedure.

Contours of the mass flow rate are shown in Figure 7.11. For these, only a change at the highest Reynolds number was seen, where there was a small increase in the mass flow associated with the tip leakage vortex. Contours of vorticity are presented in Figure 7.12. Comparing the results with Actuator 1 against the baseline plots on the left of the figure, there appears to be negligible change. Contours of the $c_{pt}$ levels are shown in Figure 7.13. The actuator in this case seemed to increase the $c_{pt}$ levels and therefore the loss. This was most apparent at the lower Reynolds numbers. At the highest Reynolds number there appeared to be a small decrease in the $c_{pt}$ values. However the full effect of this actuator could not be determined because the full spatial survey could not be acquired.
Figure 7.11. Actuator 1: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{nur}$), 1.94% $\tau/c_x, 40kV, Re = 5.3\times10^4, 6.9\times10^4, 8.7\times10^4, 1.04\times10^5$
Figure 7.12. Actuator 1: Wake Survey Contours of Normalized Streamwise Vorticity \( \hat{\Omega}_x \), 1.94% \( \tau/c_x \), 40kV, \( \text{Re} = 5.3 \times 10^4, 6.9 \times 10^4, 8.7 \times 10^4, 1.04 \times 10^5 \)
Figure 7.13. Actuator 1: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pl}$), 1.94% $\tau/c_x$, 40kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$, $8.7 \times 10^4$, $1.04 \times 10^5$
7.4 Actuator 2: Pressure-Side Edge Tip Actuator

This section presents the wake measurements for the Macor Pressure-Side Edge Actuator 2. This actuator was only tested at the highest Reynolds number of 1.05×10^5 in order to measure the full spatial survey region, as Actuator 1 had the most beneficial effect at this Reynolds number condition.

The contours of the mass flow rate are shown in Figure 7.14. These show a negligible effect of the actuator on the mass flow associated with the tip leakage vortex. The contours of vorticity for Actuator 2 are presented in Figure 7.15. Based on these, there appeared to be no significant difference compared to the baseline. Finally, the contours of $c_{pt}$ are presented in Figure 7.16. These indicated a small increase in the $c_{pt}$ loss with the actuator on.

These results are similar to those of Actuator 1, which had the same design. The choice to use Macor as the dielectric was simply to eliminate the durability problems with the Teflon. Since the electrode geometry was the same, no differences in the flow with this actuator would be expected. While Actuator 2 produced unfavorable results, the ability of this actuator to affect the tip clearance flow was important.

![Figure 7.14](image.png)

(a) Re = 1.05×10^5, Baseline

(b) Re = 1.05×10^5, 37kV

Figure 7.14. Actuator 2: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{vx}$), 2.15% $\tau/c_x$, 37kV, Re = 1.05×10^5
Figure 7.15. Actuator 2: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.15% $\tau/c_x$, 37kV, Re = $1.05 \times 10^5$

Figure 7.16. Actuator 2: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pt}$), 2.15% $\tau/c_x$, 37kV, Re = $1.05 \times 10^5$
7.5 Actuator 3: Partial Pressure-Side Edge Tip Actuator

This section presents results for Actuator 3, which featured a partial-pressure side edge tip electrode. The contours of mass flow for Actuator 3 are displayed in Figures 7.17 and 7.18. At the lowest two Reynolds numbers there was a decrease in the mass flow rate within the tip leakage vortex. At the higher two Reynolds numbers, Actuator 3 only had a minimal effect on the wake mass flow affiliated with the tip leakage vortex.

The vorticity contour plots for Actuator 3 are presented in Figures 7.19 and 7.20. Comparing the vorticity contours with the baseline distribution, it was observed that the size of the negative vorticity region of the tip leakage vortex increased with actuation at \( \text{Re} = 5.3 \times 10^4 \). At the higher Reynolds number cases, there appeared to be no appreciable change in the vorticity magnitude with actuation.

The plots of the \( c_{pt} \) contours are displayed in Figures 7.21 and 7.22. Here an increase in the loss coefficient was seen for the tip leakage vortex at the two lowest Reynolds numbers of \( 5.3 \times 10^4 \) and \( 6.9 \times 10^4 \). This result agrees with those of Actuator 2 and was expected since Actuator 3 was designed to utilize the same flow control method: effecting the tip surface separation bubble. There was no appreciable change in the total pressure coefficient contours at the higher two Reynolds numbers for Actuator 3.
(a) Re = $5.3 \times 10^4$, Baseline

(b) Re = $5.3 \times 10^4$, 36.8kV

(c) Re = $6.9 \times 10^4$, Baseline

(d) Re = $6.9 \times 10^4$, 36.8kV

Figure 7.17. Actuator 3: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{\nu x}$), 2.19% $\tau/c_x$, 36.8kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$
Figure 7.18. Actuator 3: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{\text{aux}}$), 2.19% $\tau/c_x$, 36.8kV, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
Figure 7.19. Actuator 3: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.19% $\tau/c_x$, 36.8kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$
Figure 7.20. Actuator 3: Wake Survey Contours of Normalized Streamwise Vorticity $(\hat{\Omega}_x)$, 2.19% $\tau/c_x$, 36.8kV, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
Figure 7.21. Actuator 3: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pl}$), $2.19\% \tau/c_x$, 36.8kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$
Figure 7.22. Actuator 3: Wake Survey Contours of Total Pressure Loss Coefficient $(c_{pl})$, 2.19% $\tau/c_x$, 36.8kV, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
7.6 Actuator 4: Dual Pressure Surface/Pressure-Side Edge Tip Actuator

The wake results for Actuator 4 are presented in this section. The design of this actuator included a dual, two-actuator design as shown in Figure 3.7(a,b). These results are presented for two clearance heights of 2.26% and 3.11% $c_x$.

First the smaller clearance results will be examined. The contours of the mass flow rate in the wake for a 2.26% clearance for Actuator 4 are presented in Figures 7.23 and 7.24. Comparing these to the baseline, it was observed that the mass flow rate decreased in the tip leakage vortex with actuation for all of the Reynolds numbers.

Contours of vorticity magnitude are presented in Figures 7.25 and 7.26. Comparing the results to the baseline it was seen that the magnitude of vorticity in the tip leakage vortex was increased by the actuator at the lowest two Reynolds number. The higher vorticity in the leakage vortex also induced the secondary vortex on the endwall which was observed before with the passive squealer tip at the larger 3.28% clearance gap. The changes with Actuator 4 were all observed at the lowest two Reynolds numbers. At the higher Reynolds numbers, the effect of the actuator on the vorticity was seen to be small.

Contours of the total pressure loss coefficient in the wake are presented in Figures 7.27 and 7.28. With actuation the total pressure loss coefficient associated with the tip leakage vortex increased considerably for all of the Reynolds number cases. These results from the wake survey are consistent with the endwall pressure measurements which indicated that the velocity through the clearance gap was higher with Actuator 4. The higher velocity through the gap would be expected to increase the loss associated with the tip leakage vortex, which was precisely what was observed in the wake measurements.

The results for Actuator 4 at the larger clearance of 3.11% are presented in
Figure 7.23. Actuator 4: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{\nu_x}$), 2.26% $\tau/c_x$, 36kV, Re = 5.3×10^4, 6.9×10^4
Figure 7.24. Actuator 4: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{\nu x}$), 2.26% $\tau/c_x$, 36kV, Re = $8.7 \times 10^4$, 1.04$ \times 10^5$
Figure 7.25. Actuator 4: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.26% $\tau/c_x$, 36kV, Re = $5.3\times10^4$, $6.9\times10^4$
Figure 7.26. Actuator 4: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.26% $\tau/c_x$, 36kV, Re = $8.7 \times 10^4$, $1.04 \times 10^5$
Figure 7.27. Actuator 4: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pl}$), 2.26% $\tau/c_x$, 36kV, $Re = 5.3 \times 10^4$, $6.9 \times 10^4$
Figure 7.28. Actuator 4: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pl}$), 2.26% $\tau/c_x$, 36kV, Re = $8.7 \times 10^4$, $1.04 \times 10^5$
Figures 7.29 through 7.31. Here, as with the passive squealer tip results at the larger clearance, the velocities in the wake were too large at Re $= 1.04 \times 10^5$, which caused the pressure transducer voltages to be clipped by the analog-to-digital converter. Because of this the Re $= 1.04 \times 10^5$ results are not included in this section.

The contours of the mass flow rate are shown in Figure 7.29. With the larger clearance gap there again was a decrease in the tip leakage vortex mass flow rate with Actuator 4, especially at low Reynolds numbers.

The contours of the normalized vorticity are shown in Figure 7.30. Comparing the vorticity to the baseline, it was observed that the actuator increased the magnitude of the vorticity associated with the tip leakage vortex again. The secondary vortex close to the endwall was also observed.

Finally, the $c_{pt}$ contours in the wake are presented in Figure 7.31. Here again was seen an increase in $c_{pt}$ associated with the tip leakage vortex with actuation for all Reynolds numbers.
Figure 7.29. Actuator 4: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{v,u}$), 3.11% $\tau/c_x$, 36kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$, $8.6 \times 10^4$
Figure 7.30. Actuator 4: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 3.11% $\tau/c_x$, 36kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$, $8.6 \times 10^4$
Figure 7.31. Actuator 4: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pl}$), 3.11% $\tau/c_x$, 36kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$, $8.6 \times 10^4$
7.7 Actuator 5: Suction-Side Edge Tip Actuator

This section documents the wake results with Actuator 5, which was designed to mimic a suction-side passive squealer tip. For this design the electrodes were configured to induce a velocity that opposed the leakage flow through the clearance gap. For this actuator, two applied voltages of 36.4kV pk-pk and 42.2kV pk-pk were examined. The effect of Actuator 5 at the lower voltage is presented in Figures 7.32 through 7.37. The results at the higher voltage are displayed in Figures 7.38 through 7.43.

For the lower voltage, contours of the mass flow rate for Actuator 5 are presented in Figures 7.32 and 7.33. Here there was a visible increase in the mass flux associated with the tip leakage vortex at all four Reynolds numbers.

Contours of vorticity are presented in Figures 7.34 and 7.35. At the lowest Reynolds number of $5.3 \times 10^4$, the actuator clearly reduced the vorticity magnitude of the tip leakage vortex. Here, changes in the horseshoe and passage vortices with actuation were also observed. At $Re = 6.9 \times 10^4$, the actuator continued to reduce the vorticity level of the tip leakage vortex. There appeared to be only a minimal change in the passage vortex and horseshoe vortex. At the higher two Reynolds numbers, the changes in the tip leakage vortex due to the actuator were small.

The contours of $c_{pt}$ for Actuator 5 are shown in Figures 7.36 and 7.37. At the lowest Reynolds number, a nearly complete suppression of the total pressure loss coefficient associated with the tip leakage vortex was noticeable. Actuator 5 continued to reduce the loss in the tip leakage vortex as the Reynolds number was increased, although the improvement diminished.

Overall, Actuator 5 performed similar to a passive squealer tip in reducing the effect of the tip leakage vortex. The effect of increasing the applied voltage to the actuator is presented in the next figures.
Figure 7.32. Actuator 5: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{v_{ax}}$), 2.18% $\tau/c_x$, 36.4kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$
Figure 7.33. Actuator 5: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{v_{ax}}$), 2.18% $\tau/c_x$, 36.4kV, Re = $8.7 \times 10^4$, 1.04$ \times 10^5$
Figure 7.34. Actuator 5: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.18% $\tau/c_x$, 36.4kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$
Figure 7.35. Actuator 5: Wake Survey Contours of Normalized Streamwise Vorticity $(\hat{\Omega}_x)$, $2.18\% \tau/c_x$, $36.4$kV, $Re = 8.7 \times 10^4$, $1.04 \times 10^5$
Figure 7.36. Actuator 5: Wake Survey Contours of Total Pressure Loss Coefficient $(c_p)$, $2.18\% \tau/c_x$, 36.4kV; Re = $5.3 \times 10^4$, $6.9 \times 10^4$
Figure 7.37. Actuator 5: Wake Survey Contours of Total Pressure Loss Coefficient \( (c_{pl}) \), 2.18\% \( \tau/c_x \), 36.4kV, Re = 8.7\( \times10^4 \), 1.04\( \times10^5 \)
Contours of the mass flow rate for Actuator 5 at the higher voltage level of 42.2kV are displayed in Figures 7.38 and 7.39. As with the lower voltage, there was an reduction in the mass flow deficit associated with the tip leakage vortex, however here the effect was more pronounced.

The vorticity contours in the wake of the blade with Actuator 5 at the higher voltage level are presented in Figures 7.40 and 7.41. For this the same trend of a reduced vorticity magnitude in the tip leakage vortex was apparent, although the effect was more significant at the higher forcing level.

Finally, contours of the $c_{pt}$ for Actuator 5 at the higher voltage are presented in Figures 7.42 and 7.43. This again showed a strong reduction in the loss associated with the tip leakage vortex.

In summary, Actuator 5 appeared to reduce the mass flow through the clearance and reduce the losses associated with the tip leakage vortex. It was also demonstrated that the effect of the actuator was sensitive to the level of the actuator voltage. This result is consistent with the endwall pressure measurements presented in the previous chapter.
Figure 7.38. Actuator 5: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{v_{ax}}$), 2.18% $\tau/c_x$, 42.2kV, Re = $5.3 \times 10^4$, $6.8 \times 10^4$.
(a) $\text{Re} = 8.6 \times 10^4$, Baseline

(b) $\text{Re} = 8.6 \times 10^4$, 42.2kV

(c) $\text{Re} = 1.03 \times 10^5$, Baseline

(d) $\text{Re} = 1.03 \times 10^5$, 42.2kV

Figure 7.39. Actuator 5: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{vz}$), 2.18% $\tau/c_x$, 42.2kV, Re = $8.6 \times 10^4$, 1.03$\times 10^5$
Figure 7.40. Actuator 5: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.18% $\tau/c_x$, 42.2kV, $Re = 5.3 \times 10^4, 6.8 \times 10^4$
Figure 7.41. Actuator 5: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.18% $\tau/c_x$, 42.2kV, $Re = 8.6 \times 10^4$, $1.03 \times 10^5$
Figure 7.42. Actuator 5: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pL}$), 2.18% $\tau/c_x$, 42.2kV, Re = $5.3 \times 10^4$, $6.8 \times 10^4$
Figure 7.43. Actuator 5: Wake Survey Contours of Total Pressure Loss Coefficient $(c_{pl})$, 2.18% $\tau/c_x$, 42.2kV, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
7.8 Actuator 6: Pressure-Side Edge/Suction-Side Edge Tip Actuator

This section presents wake measurements for Actuator 6. This design combined Actuators 2 and 5. It was comprised of electrodes that were located at both the pressure- and suction-side edges of the blade tip surface. The actuators each induced a velocity towards the camberline, causing the flow to stagnate near the center of the blade thickness and result in a flow directed away from the blade surface toward the endwall. This was intended to block the leakage flow through the clearance.

The contours of the mass flow rate for Actuator 6 are presented in Figures 7.44 and 7.45. In this case a decrease in the tip leakage vortex mass flow rate deficit was apparent, similar to the result with Actuator 5. The effect decreased with increasing Reynolds number.

The contours of vorticity in the wake for Actuator 6 are shown in Figures 7.46 and 7.47. With the actuation at \( \text{Re} = 5.3 \times 10^4 \), the peak negative vorticity associated with the tip leakage vortex was reduced. At the three higher Reynolds numbers, the effect of the actuator on the vorticity field was negligible.

Finally, the contours of the total pressure loss coefficient in the wake for Actuator 6 are shown in Figures 7.48 and 7.49. Here there was a strong reduction in the loss in the tip leakage vortex at the lowest Reynolds number. This improvement was visible at all Reynolds numbers, but again decreased with increasing Reynolds number.
Figure 7.44. Actuator 6: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{v_{ax}}$), 2.25% $\tau/c_x$, 36.4kV, Re = $5.3 \times 10^4$, $6.9 \times 10^4$
Figure 7.45. Actuator 6: Wake Survey Contours of Mass Flow Rate ($\dot{m}_{\nu x}$), 2.25% $\tau/c_x$, 36.4kV, $Re = 8.6 \times 10^4$, $1.03 \times 10^5$
Figure 7.46. Actuator 6: Wake Survey Contours of Normalized Streamwise Vorticity ($\hat{\Omega}_x$), 2.25% $\tau/c_x$, 36.4kV, Re = 5.3×10^4, 6.9×10^4
Figure 7.47. Actuator 6: Wake Survey Contours of Normalized Streamwise Vorticity $(\hat{\Omega}_x)$, 2.25% $\tau/c_x$, 36.4kV, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
Figure 7.48. Actuator 6: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pl}$), 2.25% $\tau/c_x$, 36.4kV, $Re = 5.3 \times 10^4, 6.9 \times 10^4$
Figure 7.49. Actuator 6: Wake Survey Contours of Total Pressure Loss Coefficient ($c_{pl}$), 2.25% $\tau/c_x$, 36.4kV, Re = $8.6 \times 10^4$, $1.03 \times 10^5$
7.9 Mass Averaged Total Pressure Loss Coefficient Results

This section provides the results of calculations of the mass-averaged total pressure loss coefficient values, $\overline{c_p}$, that were obtained by integrating the total pressure coefficients over the full spatial survey region. The mass-averaged parameter will be used to quantify the differences between the flow control approaches. These results will be presented first as absolute mass-averaged total pressure loss coefficients and then as percent differences in $\overline{c_p}$ with flow control from the respective baseline cases, based on Equations 2.7 and 2.8. As stated in Chapter 2, a reduction in $\overline{c_p}$ with applied flow control yields a positive change in $\Delta \overline{c_p}$.

The mass-averaged loss coefficient integrated over the entire spatial survey area ($0 < y/p < 0.8$, $0 < z/c_x < 0.6$) is presented in Figures 7.50 through 7.53. This includes all of the actuator configurations except Actuator 1. Surveys for Actuator 1 were limited to the smaller spatial survey region that encompassed the tip leakage vortex because the durability of the Teflon dielectric limited the wake survey times. The mass-averaged total pressure loss coefficient integrated over the smaller region encompassing the tip leakage vortex ($0.29 < y/p < 0.71$, $0 < z/c_x < 0.15$) are presented in Figures 7.54 through 7.57. This result includes all of the flow control designs.

Figure 7.50(a) presents the results for Actuators 2 through 4. Figure 7.50(b) lists the legend of symbols for this figure and Figure 7.51. Actuator 2 (Left-pointing Triangle Symbols) was observed to cause a small increase in the overall loss. For Actuator 3 (Hexagram Symbols), there was a substantial increase in the loss coefficient above the baseline for the lowest Reynolds numbers. For Actuator 4 at both the smaller 2.26% gap (Diamond Symbols) and the larger 3.11% gap (Star Symbols), there was a consistent increase in the loss for all four Reynolds numbers. Therefore all of these actuators increased the overall loss coefficient when operated.
The overall loss coefficient for Actuators 5 and 6 are presented in Figure 7.51. The results for Actuator 5 (Downward-pointing and Upward-pointing Triangle Symbols) showed a significant reduction of \( \bar{c}_p \) compared to the baseline. Also noticeable is the further reduction of \( \bar{c}_p \) as the applied voltage level to the actuator was increased. This was especially evident at the lower Reynolds numbers. The results for Actuator 6 (Right-pointing Triangle Symbols) indicate an increase in the loss coefficient compared to the baseline, particularly at the lower Reynolds numbers.

The total pressure loss coefficient results for the passive suction-side squealer tip are also presented in Figure 7.51. This includes the smaller 2.19% gap (Square Symbols) and 3.28% gap (Circle Symbols). Overall, the squealer tip significantly reduced the loss coefficient over the full range of Reynolds numbers. As the clearance was increased, the baseline loss increased almost uniformly (Square Symbols to Circle Symbols). The reduction in the loss coefficient was largest for the squealer tip applied to the smaller clearance. Also, when the squealer tip was added to the 3.28% clearance, the loss coefficient was reduced below the baseline for the smaller clearance. This illustrates that the squealer tips were beneficial in all instances.

The percent change in the mass-averaged total pressure loss coefficient for all of the flow control cases are presented in Figures 7.52 and 7.53. Figure 7.52 shows the results for Actuators 2 through 4. At the lowest Reynolds number, Actuator 2 (Left-pointing Triangle Symbols) produced an increase in the loss of approximately 3.5%. Actuator 3 (Hexagram Symbols) also produced an increase in the loss coefficient of a few percent at all Reynolds numbers. Actuator 4 at a 2.26% clearance gap (Diamond Symbols) created a clear increase in the loss coefficient ranging between 1% and 9% depending on Reynolds number. At a clearance gap of 3.11%, Actuator 4 (Star Symbols) increased the loss from between 1% to 4.2%.

The percentage change in the loss coefficients for Actuators 5 and 6 and the squealer tips are presented in Figure 7.53. For Actuator 5 (Upward-pointing and
Downward-pointing Triangle Symbols), there was a reduction in the loss coefficient of about 2% for the highest two Reynolds numbers, irrespective of the forcing level. For the lower applied voltage of 36.4kV, the loss coefficient decreased by 3% at \( \text{Re} = 6.9 \times 10^4 \) and by 9% at the lowest Reynolds number. When the forcing level was increased, the loss coefficient decreased by 12% over the baseline. Actuator 6 (Right-pointing Triangle Symbols) produced higher loss coefficients for all of the Reynolds numbers. Here the percent increase ranged from 0.05% to 3.5%.

The results for the squealer tips are also shown in Figure 7.53. For the 2.19% clearance gap (Square Symbols), the reduction in loss coefficient was fairly constant over the Reynolds numbers. The percent reduction in the loss coefficient was approximately 40% in this case. For the 3.28% clearance gap (Circle Symbols) the squealer tip reduced the loss coefficient by 16% to 19% over both Reynolds numbers tested.
(a) Active Results for Actuators 2 through 4

<table>
<thead>
<tr>
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</thead>
<tbody>
<tr>
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<td>★★</td>
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</tr>
<tr>
<td>□</td>
<td>2.19%, Plain Tip</td>
</tr>
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<td>3.28%, Plain Tip</td>
</tr>
<tr>
<td>○</td>
<td>3.28%, Suction–Side Squealer Tip</td>
</tr>
</tbody>
</table>

(b) Legend of Symbols

Figure 7.50. Overall Mass Averaged Total Pressure Loss Values for Baseline and Actuated Flow Control Cases, Part 1
Figure 7.51. Overall Mass Averaged Total Pressure Loss Values for Baseline and Actuated Flow Control Cases, Part 2. Legend of Symbols shown in Figure 7.50.
(a) Active Results for Actuators 2 through 4

2.15%, Macor Actuator 2, 37kV
2.19%, Macor Actuator 3, 36.8kV
2.26%, Macor Actuator 4, 36kV
3.11%, Macor Actuator 4, 36.4kV
2.18%, Macor Actuator 5, 36.4kV
2.18%, Macor Actuator 5, 42.2kV
2.25%, Macor Actuator 6, 36.4kV
2.19%, Suction–Side Squealer Tip
3.28%, Suction–Side Squealer Tip

(b) Legend of Symbols

Figure 7.52. Overall Percent Change in Mass Averaged Total Pressure Loss Values with Actuation over the Baseline Cases, Part 1
(a) Active Results for Actuators 5 through 6 and Passive Results for the Squealer Tip

(b) Legend of Symbols

Figure 7.53. Overall Percent Change in Mass Averaged Total Pressure Loss Values with Actuation over the Baseline Cases, Part 2
The flow control results of $c_p$ values integrated over the smaller spatial survey region encompassing the tip leakage vortex are presented in Figures 7.54 through 7.57. The absolute values of $c_p$ are presented in Figures 7.54 and 7.55. The percentage change in the loss coefficient with actuation is shown in Figures 7.56 and 7.57.

Figure 7.54 corresponds to Actuators 1 through 4. For Actuator 1 (Right-pointing Triangle Symbols), an increase in $c_p$ at low Reynolds numbers was seen, but a reduction in the loss coefficient compared to the baseline at the higher Reynolds was observed. Actuator 2 (Left-pointing Triangle Symbols) exhibited a small increase in the loss coefficient compared to the baseline. Actuator 3 (Hexagram Symbols) also showed an increase in the loss coefficient compared to the baseline over all of the Reynolds numbers. Actuator 4 (Diamond Symbols) for the smaller 2.26% clearance produced a modest increase in the loss coefficient. At the larger 3.11% clearance (Star Symbols) there was a significant change in the loss coefficient compared to the baseline.

The results for Actuators 5 and 6 are shown in Figure 7.55. With Actuator 5 operating at 36.4kV (Upward-pointing Triangle Symbols) a decrease in the loss coefficient was observed that further decreased as the Reynolds number was reduced. Actuator 5 at 42.2kV (Downward-pointing Triangle Symbols) yielded a similar trend, with an even further decrease in the loss coefficient at this higher voltage. Actuator 6 (Right-pointing Triangle Symbols) followed a similar trend as Actuator 5, but was not as effective as Actuator 5 in reducing the loss coefficient. Note however that Actuator 6 did not show a lowering of the loss coefficient for the larger survey area. These results illustrate that it was effective in reducing the tip leakage vortex loss only.

The effect of the squealer tip on the loss coefficient associated with the tip leakage vortex is presented in Figure 7.55. The addition of the squealer tip with the smaller 2.19% clearance gap (Square Symbols) again produced a strong reduction in the loss
coefficients over the full range of Reynolds numbers. This was similarly true for the squealer tip at the larger 3.28% clearance gap (Circle Symbols).

The percentage changes in the tip leakage vortex loss coefficient with flow control are presented in Figures 7.56 and 7.57. Figure 7.56 corresponds to Actuators 1 through 4. For Actuator 1 (Dot Symbols) a reduction in the loss coefficient at the higher Reynolds numbers was observed. At the highest Reynolds number the loss coefficient decreased by approximately 9.1%. This beneficial effect is contradicted by Actuator 2 (Left-pointing Triangle Symbols), which showed an increase in the loss at this Reynolds number. Actuator 3 (Hexagram Symbols) and Actuator 4 at the 2.26% clearance gap (Diamond Symbols) and 3.11% clearance gap (Star Symbols) also showed increases in the loss coefficient. A similar result was found with the larger spatial survey. Thus these actuator designs were not effective for reducing the losses due to the tip clearance flow but instead exacerbated the losses.

The results for Actuators 5 and 6, and the passage squealer tip are presented in Figure 7.57. Actuator 5 at 36.4kV (Upward-pointing Triangle Symbols) and at 42.2kV (Downward-pointing Triangle Symbols) both gave the same general trend as with the larger survey region, but with much greater effect ranging between 5% and 37% at 36.4kV and 5% to 51% at 42.2kV. Actuator 5 at 42.2kV produced approximately the same decrease in the loss coefficient as the passive squealer tip (Square Symbols). Actuator 6 (Rightward-pointing Triangle Symbols) also showed reduction in the loss coefficient between 3% and 26%, although this benefit did not extend to the larger spatial survey.

For the squealer tip, the results were similar to those at the larger survey area. With the smaller 2.19% clearance gap (Square Symbols) the squealer tip gave a maximum 57% reduction in the loss associated with the tip leakage vortex. The squealer tip with the larger 3.28% clearance gap (Circle Symbols) similarly produced a 54% reduction in the tip leakage vortex loss coefficient.
(a) Active Results for Actuators 1 through 4

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<th>Actuated Voltage</th>
<th>TLV</th>
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<td>TLV</td>
</tr>
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<td>TLV</td>
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<td>Macor Actuator 3</td>
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<td>36.8kV</td>
<td>TLV</td>
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<tr>
<td>Macor Actuator 4</td>
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<td>36kV</td>
<td>TLV</td>
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<tr>
<td>Macor Actuator 4</td>
<td>2.26%</td>
<td>36.4kV</td>
<td>TLV</td>
</tr>
<tr>
<td>Macor Actuator 5</td>
<td>2.18%</td>
<td>Baseline</td>
<td>TLV</td>
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<td>Macor Actuator 5</td>
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<td>42.2kV</td>
<td>TLV</td>
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<td>Macor Actuator 6</td>
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<td>Baseline</td>
<td>TLV</td>
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<td>Macor Actuator 6</td>
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<td>36.4kV</td>
<td>TLV</td>
</tr>
<tr>
<td>Plain Tip</td>
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<td></td>
<td>TLV</td>
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<td>Suction–Side Squealer Tip</td>
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<tr>
<td>Plain Tip</td>
<td>3.28%</td>
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<td>Suction–Side Squealer Tip</td>
<td>3.28%</td>
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</tbody>
</table>

(b) Legend of Symbols

Figure 7.54. Tip Leakage Vortex Region Mass Averaged Total Pressure Loss Values for Baseline and Actuated Flow Control Cases, Part 1
Figure 7.55. Tip Leakage Vortex Region Mass Averaged Total Pressure Loss Values for Baseline and Actuated Flow Control Cases, Part 2. Legend of Symbols shown in Figure 7.54
Figure 7.56. Tip Leakage Vortex Region Percent Change in Mass Averaged Total Pressure Loss Values for Baseline and Actuated Flow Control Cases, Part 1
(a) Active Results for Actuators 5 through 6 and Passive Results for the Squealer Tip

- 2.0%, Teflon Actuator 1, 40kV, TLV
- 2.15%, Macor Actuator 2, 37kV, TLV
- 2.19%, Macor Actuator 3, 36.8kV, TLV
- 2.26%, Macor Actuator 4, 36kV, TLV
- 3.11%, Macor Actuator 4, 36.4kV, TLV
- 2.18%, Macor Actuator 5, 36.4kV, TLV
- 2.18%, Macor Actuator 5, 42.2kV, TLV
- 2.25%, Macor Actuator 6, 36.4kV, TLV
- 2.19%, Suction–Side Squealer Tip, TLV
- 3.28%, Suction–Side Squealer Tip, TLV

(b) Legend of Symbols

Figure 7.57. Tip Leakage Vortex Region Percent Change in Mass Averaged Total Pressure Loss Values for Baseline and Actuated Flow Control Cases, Part 2
CHAPTER 8

CONCLUSIONS AND FUTURE RECOMMENDATIONS

8.1 Research Objectives and Experimental Investigations

This chapter presents a summary of the tip clearance flow control study using blade-mounted plasma actuation as well as recommendations for future work in this area. The objectives of this work were to:

1. Understand the state of the baseline flow near the tip clearance within a linear turbine cascade over a range of inflow parameters and tip clearance heights,
2. Determine the physical mechanisms by which the turbine tip clearance flow is receptive to flow control,
3. Increase the cascade efficiency by decreasing losses within the tip gap flow through the use of plasma flow control, and
4. Determine the range of applicability of flow control with regard to inflow parameters and tip clearance height.

In order to meet these objectives, the tip clearance flow was experimentally investigated using multiple experiments. These included: surveys of the endwall inflow boundary layer to the tip clearance, endwall and blade tip surface flow visualization, endwall and blade surface pressure measurements, and blade wake measurements.
downstream of the blade with a tip clearance. Two types of flow control were applied to the tip clearance flow in order to reduce the tip clearance losses. These were a passive partial suction-side squealer tip and a single dielectric barrier discharge plasma actuator. For the plasma actuation, six actuator configurations were tested in order to determine the receptivity of the flow to different actuation mechanisms. The outcomes from these tests will be explained in the following sections and future recommendations will be made.

8.2 Boundary Layer Results

The endwall boundary layer of the inlet flow to the turbine cascade was measured using a CTA hot-wire probe oriented parallel to the endwall. The flow was measured at three upstream distances from the test blade leading edge: 1 c, 1.5 c, and 2 c.

The flow did not vary considerably between the three test locations, indicating that the flow was reasonably well developed in the streamwise direction. The boundary layer thickness varied between 8 mm and 14 mm, depending on the inlet Reynolds number. The inlet boundary layer to the linear cascade was found to be laminar at the two lower Reynolds numbers. At the next highest Reynolds number, the flow was transitional in nature. At the highest Reynolds number the flow was turbulent.

The velocity profiles were compared with a Falkner-Skan laminar similarity boundary layer profile to determine the streamwise pressure gradient within the flow. For Re = 5.3 \times 10^4 and 6.9 \times 10^4, the boundary layer exhibited no pressure gradient, and the flow took on a Blasius boundary layer velocity profile shape. This result was confirmed by the measured shape factor for the flow, which was approximately 2.58 for the two laminar Reynolds numbers at the 1 c location. This shape factor agrees well with the shape factor for a laminar Blasius boundary layer velocity profile of 2.59. At Re = 8.7 \times 10^4, the flow was transitioning to a turbulent flow
and did not match the Blasius boundary layer profile seen at the lower Reynolds numbers, but was almost fully laminar. For \( \text{Re} = 1.04 \times 10^5 \) the flow was turbulent, so the Falkner-Skan similarity solution was not applicable in this case. These results illustrate that a zero-pressure gradient boundary layer existed for all Reynolds numbers.

The blade tip clearance was varied between 1.60 mm and 6.38 mm. For these values, the clearance was always completely submersed inside the boundary layer. The ratio of the mean clearance velocity to the mean inflow velocity varied significantly over the range of test conditions, from 0.17 to 0.70. This has implications on the effectiveness of the flow control in the clearance gap. The resulting ratio of the mean forcing velocity to the mean clearance velocity ranged between 3.1 to 0.38. At the lowest Reynolds number the plasma actuator flow control was able to exert a greater effect on the clearance flow because of the laminar nature of the boundary layer and because of the higher actuator forcing velocity with respect to the clearance gap flow velocity. As the tip clearance height or inlet velocity to the clearance were increased, the effectiveness of the actuator was diminished because of an increased mean clearance leakage velocity.

8.3 Flow Visualization Results

Surface flow visualization on the endwall under the blade and on the blade tip surface were performed to understand the effect of blade clearance height and Reynolds number on the tip clearance flow physics. For the endwall measurements, a mixture of kerosene, oleic acid, and TiO\(_2\) was used, while methyl salicylate was used for the tip surface visualization.

On the tip surface, the flowfield characteristics were weakly dependent with changes in Reynolds number. However, there were changes in the flow physics with variation in the tip clearance height. The experimental results revealed a persistent
near-pressure-side tip surface separation bubble for clearances greater than 1%. As the clearance increased, the separation bubble location extended further into the clearance toward the suction-side edge, covering a larger percentage of the blade tip surface. On the endwall surface, there were changes in the flow patterns outside of the clearance with an increase in Reynolds number. Multiple horseshoe vortices were found at lower Reynolds numbers, while these vortices collapsed to one vortex closer to the leading edge of the blade at higher Reynolds numbers.

On the endwall surface within the clearance, as the Reynolds number was increased the endwall separation bubble moved further into the clearance. This was due to the stronger flow acceleration and subsequently strengthened vena contracta formation. Additionally, the location of the endwall bubble fluid exit point, where the fluid migrated out of the clearance, moved forward axially as the tip clearance was increased. This illustrated that as the clearance gap was increased, the ability of the blade to turn the flow in the clearance was significantly reduced.

Beyond the baseline flow investigations, flow visualization was used to investigate some of the flow control designs on the blade tip: a passive suction-side squealer tip, a plasma actuator forcing with the leakage flow, and a plasma squealer forcing in opposition to the leakage flow. All three forcing methods qualitatively appeared to be successful at reducing the unwanted tip clearance flow physics and improving the secondary losses. The results implied that both the passive suction-side squealer and plasma squealer were able to reduce the flow leakage and improve turning of the flow within the clearance. The plasma actuator forcing with the leakage appeared to reduce the blade separation bubble strength and discharge of the high-loss bubble fluid into the downstream wake at \( \text{Re} = 1.04 \times 10^5 \). These results were confirmed by the downstream wake total pressure loss coefficient measurements presented in Chapter 7.
8.4 Surface Pressure Measurement Results

The static pressure on the blade pressure- and suction-surfaces was measured at midspan and at the tip. Away from the blade ends, a characteristic two-dimensional blade suction-surface separation bubble existed in the aft region of the axial blade chord, where the flow was not able to remain attached to the blade surface due to the adverse pressure gradient and high blade turning.

At the blade tip, the presence of the clearance caused blade unloading. Severe unloading occurred on the pressure surface and in the forward part of the suction surface. In the aft region of the suction-surface the blade tip experienced strong loading, which masked the suction-surface separation bubble. Higher loading of the blade in this region was caused by the tip leakage vortex that remained attached to the blade suction-surface corner. Overall, the unloading over the majority of the blade dominated the higher loading in the rear portion of the suction-surface. As the tip clearance height was increased, the unloading of the blade tip increased considerably. However, the blade tip unloading was only weakly dependent on Reynolds number.

The static pressure on the endwall under the blade was also measured to understand the clearance flow both with and without flow control. For the baseline flow, the endwall static pressure illustrated a high negative region of wall pressure coefficient at around $x/c_x = 0.6$, where the flow leakage through the clearance was the strongest. This region was likely the place where the separation bubble losses within the clearance and the tip leakage flow were the strongest, and also where the leakage vortex began to create considerable loss as it merged with the passage flow.

The active flow control had a modest effect on the tip clearance flow. All of the plasma actuator designs exhibited an inverse relationship between Reynolds number and control authority, while the passive control showed only a relatively weak
sensitivity to Reynolds number effects. Actuator 1, the pressure-side edge tip actuator, showed the greatest negative change in wall static pressure, where the actuator increased the clearance flow velocity in an effort to reduce the separation bubble losses within the clearance. Actuator 5, the suction-side edge tip actuator, and Actuator 6, the pressure-side edge/suction-side edge cavity tip actuator, both produced the strongest increases in wall static pressure coefficient. These illustrate the effectiveness of the suction-side plasma squealer to turn the flow and the potential for the pressure-side actuator to reduce the losses in the separation bubble. The application of two forcing levels using Actuator 5 illustrated the dependence of the wall pressure signature on the induced actuator velocity. The results from Actuator 5 forcing were of a similar magnitude to the findings from the passive flow control for the lowest Reynolds number.

It was also found that the change in the endwall pressure due to flow control occurred mostly within the interior of the clearance region, as evidenced by the much higher change in the blade area-averaged wall pressure than for the full endwall area-averaged values. This confirmed the local effect of the flow control on the tip clearance flow. These favorable results illustrate the potential of using blade-mounted flow control techniques to mitigate the tip leakage flow problem.

8.5 Downstream Wake Survey Results

The flow downstream of the test blade was measured using a five-hole-Pitot probe to understand the effect of plasma flow control on the overall total pressure loss coefficient. The total pressure loss coefficient varies linearly with the efficiency loss within the turbine cascade, so these results are able to give the effect of the flow control on the overall loss.

The passive squealer showed a positive reduction in the loss, which was fairly constant over all Reynolds numbers. For the plasma actuator designs, Actuator
Actuators 3 and 4 negatively increased the tip leakage flow loss over all Reynolds numbers by exacerbating the tip surface separation bubble loss. Actuator 5, the plasma squealer design, delivered a favorable reduction in the total pressure loss coefficient, which was comparable to the passive squealer tip at the lowest Reynolds number. Actuator 5 also produced a beneficial reduction in the loss coefficient at the higher Reynolds numbers, although the effect was diminished. This illustrated that the tip leakage flow exhibited a clear Reynolds number dependence to plasma flow control.

Actuator 6 produced a beneficial effect on the loss within the tip leakage vortex as did Actuator 5. Yet Actuator 6 increased the overall loss when the passage and horseshoe vortices were included in the loss calculation. This shows that the tip leakage vortex and passage vortex were weakly coupled, due to the proximity of these vortices to each other. The reduction in the tip leakage vortex through flow control in the Actuator 6 case allowed the passage vortex to become stronger and
increase in loss, especially at low Reynolds numbers.

The data also show that the major loss producing mechanism is the tip leakage vortex. This vortex is also the location of the greatest loss improvement with applied flow control for the secondary flow region. From these wake results, it was found that the tip leakage vortex total pressure loss can be mitigated by: (1) improving flow turning in the clearance region, (2) reducing the clearance velocity and thus the strength of the tip leakage vortex, (3) decreasing the quantity of high-loss viscous fluid that is permitted to migrate into the tip leakage vortex, and by (4) increasing the mass flow rate within the tip leakage vortex. Overall, the plasma flow control was beneficial in reducing the tip clearance loss in a similar fashion as the passive squealer without the negative heat transfer penalty that passive squealer tips present.

8.6 Recommendations for Future Work

Based on these results the overall effect of the blade-mounted plasma flow control was determined. From this research there are a few recommendations that can be made to build on this work and within this research area.

Based on the beneficial results in the turbine cascade of some of these actuator designs, it would be advantageous to investigate the applicability of the plasma flow control on a rotating turbine. Particularly Actuator 1, which forced with the leakage flow, Actuator 5, the plasma squealer, and Actuator 6, which produced the cross-flow jet, showed positive improvements in the tip leakage flow, so it is thought that their usefulness may extend to the rotating environment. Although implementation of these actuators on the blade tip surface of a rotating machine would be impractical due to the small blade tip area, these designs could be integrated into the casing wall of the clearance gap opposite the turbine blade tip. These flow control actuators could be investigated alongside a favorable passive device, as was done in this research, to provide a useful reference point for the active flow control.
results. For additional information in the area of tip clearance flow control, a separate experimental study using plasma actuation on the casing wall opposite the blade tip will be published soon by Stephens [65].

For future work, in order to further understand the detailed physical mechanisms of the flow control that was used in this research, surveys of the velocity and pressure fields within the clearance gap with applied flow control would be necessary. Invasive studies of the baseline flow have been performed by other researchers [6, 27, 66], however, typically the results are highly dependent on blade geometry and inflow conditions. Also, often there is a lack of detail in these measurements of the relevant parameters in the tests, such as the inlet boundary layer thickness or inlet turbulence intensity, that make the applicability of these findings difficult to apply elsewhere.

To perform measurements such as this, it is suggested that a thorough, systematic parameter study of the flow within the clearance gap with and without flow control be conducted using a multi-hole pressure probe along with stereo particle image velocimetry (PIV). These would provide a comprehensive picture of the velocity and loss characteristics within the gap. Associated with this, the difficulty of such measurements is that a small physical dimension of the clearance gap is required to achieve a reasonable clearance height of a few percent of axial blade chord. A very large, oversized blade would eliminate this problem, and allow for a more invasive study of the tip clearance flow physics.

Additionally, it was shown in the time-averaged wake measurements that the passage and tip leakage vortices were weakly coupled. In order to understand the interaction of these vortical structures, time-resolved stereo PIV would elucidate the instantaneous mutual effect of these vortices. Three-component PIV would be necessary because of the need to capture both the radial and tangential velocity components that determine the vorticity contours, as well as the streamwise velocity component, which would allow mass flow rate contours to be measured. It would also
be necessary to take measurements using PIV at various downstream measurement planes from the test blade, in order to determine any change in the growth rate of the three structures due to interaction with each other, both with and without flow control. The drawback of PIV is the lack of pressure information, which would be vital to knowing the effect of flow control on the total pressure loss coefficient. Therefore it would again be necessary to couple the PIV measurements with pressure probe measurements.

The final recommendation is in the area of plasma actuator manufacturing. Past advances in the selection of suitable dielectric materials, actuator electronics, and plasma signal waveform have been made. However, the physical manufacture of a plasma actuator is time consuming and laborious. What is needed is a fast, accurate method for electrode placement onto or integration into dielectric surfaces. Work in the area of applicable surface coatings and deposition of material onto dielectric materials are being investigated, however, more research could be done to increase the utilization of plasma actuators.
APPENDIX A

UNCERTAINTY ANALYSIS

A.1 Introduction

The overall uncertainty in a measurement is a result of bias (systematic), $B_u$, and precision (random), $P_u$, errors. Bias error is defined as the average error in a series of repeated measurements [67]. Bias errors determine the accuracy of a measurement, and are minimized through calibration [68]. Precision error is defined as a measure of the random variation found during repeated measurements [67]. These errors are related to the scatter in the data. Precision errors characterize the measurement values with a confidence interval [68]. Bias and Precision errors are defined as

$$B_u = \sqrt{\sum_{i=1}^{N} (B_{ui})^2}, \quad P_u = t_{\nu, P} S_x$$  \hspace{1cm} (A.1)

respectively, where $B_{ui}$ is the $i^{th}$ elemental bias error, $P_{ui}$ is the $i^{th}$ elemental precision error, $t_{\nu, P}$ is the Student-t variable, and $S_x$ is the standard deviation of the means. The bias error is the sum of the $N$ elemental systematic errors inherent in the measurement system, and the precision error is defined by the confidence interval set by the normal distribution of samples over the finite data set. The total uncertainty in the measurement, $e_P$, is given using the Kline-McClintock root-sum-squares method as

$$e_P = \pm \sqrt{B_u^2 + (t_{\nu, P} S_x)^2}.$$  \hspace{1cm} (A.2)
A.2 Pressure Measurement Uncertainty

Pressure measurements consisted of sampling of a differential pressure over K number of samples to obtain a time-averaged mean. Uncertainty in these measurements is due to the errors in acquiring and digitizing the analog signal through the transducer and the analog-to-digital (A/D) converter.

The pressure was digitally sampled using a Validyne Differential Pressure Transducer, model DP103, with a companion carrier demodulator, model CD23. The digital voltage output from the transducer was digitized using a UEI analog-to-digital converter, model PD2-MFS-8-500/14DG, and stored to a computer. The relevant properties for the transducer and A/D converter are given in the following tables. Also, the typical properties that are found in this work are given in Table A.3.

Table A.1

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<tr>
<td>Full Scale Pressure, FS</td>
<td>0.89 in. $H_2O$</td>
</tr>
<tr>
<td>Accuracy, $u_A$</td>
<td>0.25% FS</td>
</tr>
<tr>
<td>Thermal Sensitivity Shift, $u_{TSS}$</td>
<td>5% $P_{typ}/100^oF$</td>
</tr>
<tr>
<td>Thermal Zero Shift, $u_{TZS}$</td>
<td>1% $FS/100^oF$</td>
</tr>
</tbody>
</table>

A.2.1 Bias Error, $B_2$

The relevant bias errors for the transducer are the uncertainties in accuracy, $u_A$, sensitivity, $u_S$, thermal zero shift, $u_{TZS}$, and thermal sensitivity shift, $u_{TSS}$.
Table A.2

ANALOG-TO-DIGITAL CONVERTER PROPERTIES

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_{FS}$</td>
<td>10 V</td>
</tr>
<tr>
<td>Number of bits, M</td>
<td>14</td>
</tr>
<tr>
<td>Allowed Input</td>
<td>$\pm 5V$</td>
</tr>
</tbody>
</table>

Table A.3

TYPICAL EXPERIMENTAL PROPERTIES

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Typical Operating Pressure, $P_{typ}$</td>
<td>0.059 in. H$_2$0</td>
</tr>
<tr>
<td>Typical Temperature Variation, $\Delta T$</td>
<td>2.2 K, 4°F</td>
</tr>
<tr>
<td>Typical Transducer Voltage, $E_{typ}$</td>
<td>2 V</td>
</tr>
</tbody>
</table>
**Accuracy Error**

The accuracy error is defined as

\[ B_{u_A} = u_A FS, \]  

(A.3)

where \( FS \) is the full scale pressure operating range of the transducer. It should be noted that the stated value for the accuracy takes into account the effects of hysteresis, linearity, and repeatability within the measurement. From Table A.1, this is

\[ B_{u_A} = 0.25 \times 0.89 \text{in.}H_20. \]

\[ B_{u_A} = 2.22E-3 \text{ in.}H_20. \]

**Sensitivity Error**

The sensitivity uncertainty is determined using a linear regression of the pressure versus voltage data recorded during the pressure transducer calibration, according to the method given in Dunn [68]. Based on the standard deviation of the linear curve fit, the error in the pressure reading, \( B_{us} \), is

\[ B_{us} = 2.24E-3 \text{ in.}H_20. \]

**Thermal Zero Shift Error**

The error of temperature effect on zero is defined as

\[ B_{uTZS} = u_{TZS} FS \frac{\Delta T}{100^\circ F}, \]

(A.4)

where \( \Delta T \) is the typical variation in operating temperature of the experiment. This becomes

\[ B_{uTZS} = 0.01 \times 0.89 \text{ in.}H_20 \frac{4^\circ F}{100^\circ F}. \]
\[ B_{uTSS} = 3.56 \times 10^{-4} \text{ in.}H_20. \]

**Thermal Sensitivity Shift Error**

The shift in span due to a temperature sensitivity is given by

\[ B_{uTSS} = u_{TSS}P_{typ} \frac{\Delta T}{100^\circ F}. \] (A.5)

Inserting values from the tables gives

\[ B_{uTSS} = 0.05 \times 0.059 \text{ in.}H_20 \times \frac{4^\circ F}{100^\circ F} \]

\[ B_{uTSS} = 1.18 \times 10^{-4} \text{ in.}H_20. \]

These errors can be combined to give the total bias error of the transducer,

\[ B_{21} = \sqrt{B_{uTSS}^2 + B_{uTZS}^2 + B_{uA}^2 + B_{uS}^2}. \] (A.6)

\[ B_{21} = \sqrt{1.18 \times 10^{-4}^2 + 3.56 \times 10^{-4}^2 + 2.22 \times 10^{-3}^2 + 2.24 \times 10^{-3}^2} \text{ in.}H_20. \]

\[ B_{21} = 1.31 \times 10^{-4} \text{ in.}H_20. \]

The only important bias error for the A/D converter is the quantization error, which is given as

\[ B_{uQ} = \frac{1}{2} \frac{E_{FS}}{2^M}. \] (A.7)

where \( E_{FS} \) is the full scale voltage range and M is the number of bits. The bias error in volts, when converted to a pressure value, yields a converter error of,

\[ B_{22} = B_{uQ}m. \]

\[ B_{22} = \frac{1}{2} \left( \frac{10V}{2^{14}} \right) 0.3768 \text{ in.}H_20/V. \]

\[ B_{22} = 1.15 \times 10^{-4} \text{ in.}H_20. \]
**Overall Bias Error**

The overall bias error is then

\[ B_2 = \sqrt{B_{21}^2 + B_{22}^2} \]  

(A.8)

\[ B_2 = \sqrt{3.18E-3^2 + 1.15E-4^2 \text{in.H}_2\text{O}} \]

\[ B_2 = 3.18 - 3 \text{ in.H}_2\text{O}. \]

**A.2.2 Precision Error, \( p_2 \)**

The precision error is a statistical approximation of the true error based on the error from the measurement of a finite sample size. Using the standard deviation of the mean and the sample size, the standard deviation from the true mean, or standard deviation of the means, can be calculated. The standard deviation is defined as

\[ S_x = \sqrt{\frac{\sum_{i=1}^{K} (P_i - P_m)^2}{K - 1}} \]  

(A.9)

where \( P_m \) is the mean pressure, \( P_i \) is the \( i^{th} \) pressure in a sample size of \( K \) pressures \((K = 10,000)\). From the transducer, the standard deviation of the mean is found to be \( S_x = 1.22E-3 \text{ V} \) or \( S_x = 4.60E-4 \text{ in. H}_2\text{O} \) when converted to pressure using calibration constant \( m \). The standard deviation of the means assumes a normal distribution of values within the sample size to determine the true error, which is given by

\[ S_{\bar{x}} = \frac{S_x}{\sqrt{K}} \]  

(A.10)

\[ S_{\bar{x}} = \frac{4.60E-4 \text{ in.H}_2\text{O}}{\sqrt{10000}} = 4.60E-6 \text{ in.H}_2\text{O}. \]

From a Student-t distribution for the large sample size being considered, the value of the Student t variable is \( t_{\nu,P} = 1.96 \) for a 95% confidence interval. The precision
error was defined in Equation A.1 as

\[ P_u = 1.96 \times 4.60E^{-6} \text{ in.} H_2O. \]

\[ P_u = 9.02E^{-6} \text{ in.} H_2O. \]

A.2.3 Overall Error

Combining the bias and precision errors from above yields

\[ e_P = \sqrt{\left(\%B_2\right)^2 + \left(\%P_2\right)^2} \]

\[ e_P = \sqrt{(3.18E-3)^2 + (9.02E^{-6})^2} \]

The overall error is nominally \( e_P = 3.18E-3 \text{ in.} H_2O \) for a typical pressure of 0.059 in. \( H_2O \), or \( \pm 5.24\% \) of the typical working pressure.

Completing a nearly identical analysis for \( Re = 1 \times 10^5 \), with a working pressure of 0.23 in. \( H_2O \), the error is nominally \( e_P = 3.21E-3 \text{ in.} H_2O \) or \( \pm 1.32\% \) of the typical working pressure.

A.3 Blade Surface Pressure Measurements

Measurements of the blade surface pressure were conducted using blade surface pressure taps of 0.050 in. ID/0.090 in. OD. The pressure coefficient is defined as

\[ c_{ps} = \frac{P_{blade,s} - P_i}{P_{ti} - P_i}. \]

where \( P_{blade,s} \) is the blade surface static pressure, \( P_i \) is the freestream inlet static pressure, and \( P_{ti} \) is the inlet total pressure. For each sample, the transducer measured the differential pressure between the blade static pressure and the inlet static pressure (numerator) as well as the differential pressure between the inlet total and
static pressures (denominator). Therefore the equation can be written as

\[ c_{ps} = \frac{\Delta P_{blade}}{\Delta P_i}. \]  

(A.13)

So the uncertainty is due to the propagation of errors from measuring the ratio of these two differential pressures. This error propagates as

\[ u_{c_{ps}} = \sqrt{\left( \frac{\partial c_{ps}}{\partial \Delta P_{blade}} u_{P_{blade,s}} \right)^2 + \left( \frac{\partial c_{ps}}{\partial \Delta P_i} u_{P_i} \right)^2}, \]

\[ \frac{\partial c_{ps}}{\partial \Delta P_{blade}} = \frac{1}{\Delta P_i}, \quad \frac{\partial c_{ps}}{\partial \Delta P_i} = -\frac{\Delta P_{blade}}{(\Delta P_i)^2} = -c_{ps}, \]

\[ u_{c_{ps}} = \sqrt{\left( \frac{1}{\Delta P_i} u_{P_{blade,s}} \right)^2 + \left( -\frac{\Delta P_{blade}}{(\Delta P_i)^2} u_{P_i} \right)^2}, \]  

(A.14)

Given that the uncertainties in each of these measured pressures are equal to \( e_P \) from above, the uncertainty in \( c_{ps} \) becomes

\[ u_{c_{ps}} = \sqrt{\left( \frac{1}{\Delta P_i} e_P \right)^2 + \left( -\frac{c_{ps}}{\Delta P_i} e_P \right)^2}. \]  

(A.15)

For a Reynolds number of \( 5 \times 10^4 \), The typical values for the blade pressure measurements are \( c_{ps} = -2.5 \), \( \Delta P_i = P_{typ} = 0.059 \text{in.} H_20 \), and \( e_P = 3.18E-3 \text{in.} H_20 \). This gives a nominal uncertainty in static pressure of

\[ u_{c_{ps}} = \sqrt{\left( \frac{1}{0.059 \text{ in.} H_20} 3.18E-3 \text{ in.} H_20 \right)^2 + \left( \frac{-2.5}{0.059 \text{ in.} H_20} 3.18E-3 \text{ in.} H_20 \right)^2}, \]

\[ u_{c_{ps}} = 1.42E-1 \text{ in.} H_20 = 5.70\% \ c_{ps}. \]

For a Reynolds number of \( 1 \times 10^5 \), The typical values for the blade pressure measurements are \( c_{ps} = -2.5 \), \( \Delta P_i = P_{typ} = 0.23 \text{in.} H_20 \), and \( e_P = 3.21E-3 \text{in.} H_20 \). This gives a nominal uncertainty in static pressure of

\[ u_{c_{ps}} = \sqrt{\left( \frac{1}{0.23 \text{ in.} H_20} 3.21E-3 \text{ in.} H_20 \right)^2 + \left( \frac{-2.5}{0.23 \text{ in.} H_20} 3.21E-3 \text{ in.} H_20 \right)^2}, \]
Therefore the blade static pressure measurements will give results within

\[ c_{ps} = 3.57E-2 \text{ in.}H_20 = 1.43\% \quad c_{ps}. \]

\[
c_{ps-true} = c_{ps} \pm 0.142 \text{ in.}H_20 \quad (Re = 5 \times 10^4) \tag{A.16}
\]

\[
c_{ps-true} = c_{ps} \pm 0.0357 \text{ in.}H_20 \quad (Re = 1 \times 10^5)
\]

A.4 Five-Hole-Probe Measurements
A.4.1 Mass-Averaged Pressure Loss Coefficient

Downstream pressure measurements were taken of the blade tip wake to determine the total pressure loss coefficient at each grid point, which is defined as

\[ c_p = \frac{P_{ti} - P_{te}}{P_{te} - P_{se}}. \tag{A.17} \]

The desired loss quantity is the mass-averaged total pressure loss coefficient, given by:

\[ \bar{c}_p = \frac{\sum c_p v_{ax} A_{ij}}{\sum v_{ax} A_{ij}}. \tag{A.18} \]

The uncertainty in the point-to-point loss coefficient is given as

\[
u_{c_p} = \sqrt{\left(\frac{\partial c_p}{\partial \Delta P_{ti}} u_P\right)^2 + \left(\frac{\partial c_p}{\partial \Delta P_{te}} u_P\right)^2 + \left(\frac{\partial c_p}{\partial \Delta P_{se}} u_P\right)^2}, \tag{A.19}\]

where the partial derivatives are

\[
\frac{\partial c_p}{\partial \Delta P_{ti}} = \frac{1}{P_{te} - P_{se}}, \quad \frac{\partial c_p}{\partial \Delta P_{te}} = \frac{-P_{ti} + P_{se}}{(P_{te} - P_{se})^2}, \quad \frac{\partial c_p}{\partial \Delta P_{se}} = \frac{(P_{ti} - P_{te})}{(P_{te} - P_{se})^2}.
\]

The uncertainty in the mass averaged loss coefficient is then

\[
u_{\bar{c}_p} = \sqrt{\left(\frac{\partial \bar{c}_p}{\partial v_{ax}} u_{v_{ax}}\right)^2 + \left(\frac{\partial \bar{c}_p}{\partial c_p} u_{c_p}\right)^2}, \tag{A.19}\]

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with derivatives

\[
\frac{\partial \bar{c}_p}{\partial c_p} = \sum v_{ax}A_{ij}u_{c_p}, \quad \frac{\partial \bar{c}_p}{\partial v_{ax}} = \sum \frac{u_{v_{ax}}A_{ij}c_p}{v_{ax}A_{ij}} - \bar{c}_p \sum \frac{u_{v_{ax}}A_{ij}}{v_{ax}A_{ij}}.
\]

But the streamwise velocity \( v_{ax} \) is computed from Bernoulli’s equation, as

\[
v_{ax} = \sqrt{\frac{2\rho}{P_{dyn,e}}} = \sqrt{\frac{2\rho}{P_{te} - P_{se}}},
\]

and is a function of the ambient pressure and temperature variation that propagates through as a variation in density from

\[
\rho = \frac{P_{Bar}}{RT} = \frac{P_{si}}{RT}.
\]

The uncertainties in these quantities are

\[
u_{\rho} = \sqrt{\left(\frac{u_{P_{Bar}}}{RT}\right)^2 + \left(\frac{-P_{Bar}u_T}{RT^2}\right)^2},
\]

and

\[
u_{v_{ax}} = \sqrt{\left(\frac{u_{P}}{\sqrt{2(P_{te} - P_{se})\rho}}\right)^2 + \left(\frac{u_{\rho}\sqrt{P_{te} - P_{se}}}{\sqrt{2\rho^{3/2}}}\right)^2}.
\]

For a Reynolds number of \( 5 \times 10^4 \), the typical values for the five-hole-probe pressure measurements are given in Table A.4. The uncertainties in streamwise velocity and density are then

\[
u_{\rho} = \sqrt{\left(\frac{16.93 Pa}{287.7 J/(kg \cdot K) \times 293.15 K}\right)^2 + \left(\frac{-2.2K \times 9.81E4 Pa}{287.7 J/(kg \cdot K) \times 293.15 K^2}\right)^2} = 8.55E-3 kg/m^3,
\]

and

\[
u_{v_{ax}} = \sqrt{\left(\frac{0.791 Pa}{\sqrt{2 \times 32.672 Pa \times 1.152 kg/m^3}}\right)^2 + \left(\frac{2.52E-4 kg/m^3 \sqrt{32.672 Pa}}{\sqrt{2(1.152)^{3/2}}}\right)^2} = 0.0967 m/s.
\]
Table A.4

TYPICAL MEASURED PROPERTIES

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{\text{dyn, e}} = P_{t_e} - P_{s_e}$</td>
<td>32.672 Pa</td>
</tr>
<tr>
<td>$P_{s_e} - P_{t_b}$</td>
<td>-38.524 Pa</td>
</tr>
<tr>
<td>$P_{t_i} - P_{t_e}$</td>
<td>5.845 Pa</td>
</tr>
<tr>
<td>$P_{s_i} = P_{\text{bar}}$</td>
<td>9.81E4 Pa</td>
</tr>
<tr>
<td>$u_p = \epsilon_p$</td>
<td>0.791 Pa</td>
</tr>
<tr>
<td>$u_{p_{\text{bar}}}$</td>
<td>16.93 Pa</td>
</tr>
<tr>
<td>$u_T$</td>
<td>2.2 K</td>
</tr>
<tr>
<td>$u_{U_{\text{vel}}}$</td>
<td>3% $U_{\infty}$</td>
</tr>
<tr>
<td>$R$</td>
<td>287.7 J/(kg·K)</td>
</tr>
<tr>
<td>$T$</td>
<td>296.65 K</td>
</tr>
<tr>
<td>$\rho$</td>
<td>1.152 kg/m³</td>
</tr>
<tr>
<td>$v_{ax}$</td>
<td>7.92 m/s</td>
</tr>
<tr>
<td>$c_p$</td>
<td>0.2284</td>
</tr>
<tr>
<td>$\bar{c}_p$</td>
<td>0.1593</td>
</tr>
</tbody>
</table>
Substituting in for the pressure uncertainties, the resulting uncertainties are:

\[ c_{p\text{-true}} = c_p \pm 1.08E-1 \quad (Re = 5 \times 10^4) \quad (A.20) \]
\[ c_{p\text{-true}} = c_p \pm 2.13E-2 \quad (Re = 1 \times 10^5) \]

\[ \bar{c}_{p\text{-true}} = \bar{c}_p \pm 4.42E-3 \quad (\pm 2.78\% \bar{c}_p) \quad (Re = 5 \times 10^4) \quad (A.21) \]
\[ \bar{c}_{p\text{-true}} = \bar{c}_p \pm 2.21E-4 \quad (\pm 0.134\% \bar{c}_p) \quad (Re = 1 \times 10^5) \]

### A.4.2 Normalized Streamwise Vorticity

The streamwise vorticity is computed from the point-to-point measurements using a second order central finite difference. The vorticity is given by

\[ \Omega_x = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \quad (A.22) \]

and is normalized as

\[ \hat{\Omega}_x = \frac{\nu}{U_\infty^2} \Omega_x = \frac{\nu}{U_\infty^2} \left( \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right) \]

The error in vorticity is a function of the linear spacing between the grid points, which propagates through the finite difference equations as a residual error that arises because of the approximation of a partial derivative using a Taylor expansion [68]. It is given by

\[ e_d = f'' dz^2 \quad (A.23) \]

where \( f'' \) is the value of the second derivative and \( dz \) is the distance between the grid points. Here, \( dz \) is a maximum of 0.4 inches. The method states that \( f'' \) is difficult to establish a value for, so a value of unity is taken as a first approximation. The maximum uncertainty in vorticity is therefore 1.03E-4 1/s. For normalize vorticity
uncertainty, the propagation of error gives:

$$u_{\hat{\Omega}} = \sqrt{\left(\frac{\mu\Omega}{\rho^2 U_\infty^2} u_\rho\right)^2 + \left(\frac{-2\mu\Omega}{\rho U_\infty^3} u_{U_\infty}\right)^2 + \left(\frac{\mu}{\rho U_\infty^2} u_{\Omega}\right)^2}$$

(A.24)

The uncertainties are therefore:

$$\hat{\Omega}_{x-true} = \hat{\Omega}_x \pm 1.44E^{-7} \quad (Re = 5 \times 10^4),$$

(A.25)

$$\hat{\Omega}_{x-true} = \hat{\Omega}_x \pm 9.11E^{-8} \quad (Re = 1 \times 10^5).$$
APPENDIX B

QUALITATIVE ANALYSIS OF TURBOMACHINERY EFFICIENCY

B.1 Introduction

This appendix is a qualitative analysis of the effect that component efficiency change has on the thrust specific fuel consumption and overall compressor and turbine efficiencies of a turbomachine.

In order to understand how the overall engine is affected by changes in component efficiencies, a study will be performed on both a typical turbojet engine and also an unmixed dual-spool turbofan. Both engines will incorporate losses incurred for each component as well as real gas effects, to simulate engines that are currently in service. Specific engine parameters will be given later on.

B.2 Component Analysis

The engine being considered will be tested at a single on-design operating point, as it is not the absolute efficiency of the engine that is important here, but the change in efficiency from a baseline or standard efficiency that is important. For this study, the design point will be a subsonic cruise Mach number that is typically seen in most flight envelopes of aircraft being used today. Real gas properties will also be used to more closely approximate in-flight conditions.

To understand how component efficiencies affect the overall engine, each component will be examined individually, and the engine will be stepped through from
inlet to exit nozzle. In Figure B.2.1, schematics of a turbojet engine and turbofan engine are given, showing the numbering sequence used. The following stations are defined for both engines. Station 0 denotes the ambient atmospheric freestream properties at altitude. Between station 1 to 2 is the Inlet, or Diffuser. Station 2 to 3 is the Compressor, 3 to 4 is the Burner, or Combustor, 4 to 5 is the Turbine, 6 to 7 is the Afterburner, if included, and 7 to 9 is the Exit Nozzle. For a turbofan, a few additional stations are needed to account for the bypass fan. These stations will be explained when the discussion of the turbofan equations is taken up. Subscripts used in the following sections are given in Table B.1.

First, the equations for a single spool turbojet with afterburner will be generally discussed. Then secondly, modifications of these equations will be performed for use in developing the equations for a dual-spool turbofan without an afterburner. A systematic variation in each component efficiency will be conducted and the change in overall efficiency documented.
Table B.1

SUBSCRIPT DEFINITIONS

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>o, R</td>
<td>Freestream</td>
</tr>
<tr>
<td>D</td>
<td>Diffuser</td>
</tr>
<tr>
<td>F</td>
<td>Bypass Fan</td>
</tr>
<tr>
<td>C</td>
<td>Compressor</td>
</tr>
<tr>
<td>B</td>
<td>Combustor</td>
</tr>
<tr>
<td>T</td>
<td>Turbine</td>
</tr>
<tr>
<td>AB</td>
<td>Afterburner</td>
</tr>
<tr>
<td>N</td>
<td>Nozzle</td>
</tr>
<tr>
<td>FN</td>
<td>Fan Nozzle</td>
</tr>
<tr>
<td>i</td>
<td>Isentropic Process</td>
</tr>
</tbody>
</table>

B.1 Turbojet with Afterburner and Losses

This section will discuss the general equations that govern the component efficiencies within a turbojet. This overview is meant to highlight important concepts that govern the component efficiencies and gas properties that occur within a turbomachine currently in service. A more thorough examination of these equations can be found in Oates [13], Jumper [69], and Mattingly [70].

Freestream Properties

In a turbomachine, changes in temperature and pressure within the engine are usually cast in terms of stagnation properties. To simplify equations, $\pi$ and $\tau$ are defined for each individual component as

$$\pi = \frac{P_{exit}}{P_{inlet}}, \quad \tau = \frac{T_{exit}}{T_{inlet}}.$$  \hspace{1cm} (B.1)  

which are the ratios of stagnation temperature leaving a component to the stagnation temperature entering a component, and of stagnation pressure leaving a component.
to the stagnation pressure entering a component, respectively. For the freestream, the stagnation temperature and pressure are defined in terms of freestream Mach number, $M_o$, as

$$\tau_R = 1 + \frac{\gamma - 1}{2} M_o^2, \quad \pi_R = \tau_R \left( \frac{\gamma}{\gamma - 1} \right),$$  \hspace{1cm} (B.2)

where subscript ‘r’ denotes recovery of these properties to their stagnation values and $\gamma$ is the ratio of specific heats. The Mach number, in turn, is a function of the speed of sound, $a_o$, and the Gas Constant, $R$, defined by

$$R = \frac{\gamma - 1}{\gamma} C_p, \quad a_o = \sqrt{\gamma RT_o},$$  \hspace{1cm} (B.3)

where $C_p$ is the freestream specific heat at constant pressure. Therefore, design variables for air moving into the inlet are the cruise Mach number and ambient conditions at altitude.

**Diffuser**

The purpose of the diffuser is to decelerate the airflow entering into the compressor while keeping the temperature and pressure as close to freestream values as possible. The pressure and temperature ratios are defined by

$$\tau_D = \frac{T_{i2}}{T_{i1}}, \quad \pi_D = \frac{P_{i2}}{P_{i1}}.$$  \hspace{1cm} (B.4)

For a subsonic inlet diffuser, the temperature ratio is usually chosen as one or near to unity, because it is closely adiabatic. Because of wall friction and separated flow, the flow is nonisentropic, and the stagnation pressure is reduced through the inlet. The pressure and temperature ratio values are determined from the geometry of the inlet and the Mach number of operation and will change with the operating point. Since only one operating point will be considered here, the two values can be chosen as design variables, as the geometry of the inlet can be tailored to meet the engine design specifications.
Moving from station 2 to 3, the task of the compressor is to pressurize the air as efficiently as possible to ready the gas for the burner. To determine the overall efficiency of the compressor, $\eta_C$, the temperature and pressure ratios, $\tau_C$ and $\pi_C$ are required. For the compressor, they are defined as

$$\tau_C = \frac{T_{t3}}{T_{t2}}, \quad \pi_C = \frac{P_{t3}}{P_{t2}}. \tag{B.5}$$

The compressor efficiency is written as

$$\eta_C = \frac{w_{Ci}}{w_C}, \tag{B.6}$$

which says that the efficiency is the work of the compressor done isentropically divided by the actual work the compressor performs. For a compressor made up of $N$ stages, applying the First Law with a summation of the efficiencies of each stage, the overall compressor efficiency is defined for $N$ stages as

$$\eta_C = \frac{\pi_C^{(\gamma_C-1)/\gamma_C} - 1}{\prod_{j=1}^{N} \left[ 1 + 1/\eta_j \left( \pi_j^{(\gamma_C-1)/\gamma_C} - 1 \right) \right] - 1}. \tag{B.7}$$

where the subscript ‘$j$’ represents properties of the $j^{th}$ stage and $\pi_C$ is the overall pressure ratio. However, assuming that the compressor has a multi-stage spool, the pressure ratio and efficiency across each stage will vary, and no assumptions about uniformity of these values through the spool can be validly made. Ultimately, it is not the specific row-by-row efficiency that is sought here, but the overall total compressor efficiency. So, to move past the restriction on how the ratios of pressure and temperature vary for each stage, the polytropic compressor efficiency is used. This quantity is defined as the ratio of ideal work of compression for a differential pressure change to that of the actual work of compression for a differential pressure change.

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change. The polytropic efficiency is written as

\[ \epsilon_C = \frac{dw_C}{dw_C} \text{.} \]  (B.8)

Assuming that \( \epsilon_C \) is constant through the spool, the overall efficiency of the compressor can be written as

\[ \eta_C = \frac{\pi_C \left( \frac{\gamma_C - 1}{\gamma_C} \right)}{\pi_C \left( \frac{\gamma_C - 1}{\gamma_C e_C} \right)} - 1 \text{,} \]  (B.9)

or as

\[ \eta_C = \frac{\pi_C \left( \frac{\gamma_C - 1}{\gamma_C} \right)}{\tau_C - 1} - 1 \text{,} \]  (B.10)

where

\[ \tau_C = \pi_C \left( \frac{\gamma_C - 1}{\gamma_C e_C} \right) \text{.} \]  (B.11)

Assuming a constant polytropic efficiency does not require that the pressure or temperature ratios for each stage be equal, but divides the work of the compressor up over the entire spool. It is valid over a single spool, or even a single stage, as it is primarily a function of technology. Therefore, the only design variables for each compressor spool are the pressure ratio and the polytropic efficiency. Since the technology level and pressure ratio are both set during the design process, \( \epsilon_C \) and \( \pi_C \) can be chosen as design variables in this analysis to yield the efficiency of the compressor.

**Burner**

Moving from station 3 to 4, the purpose of the combustor or burner section is to heat the air to higher temperature and pressure through the addition of fuel. Again using a First Law analysis, the burner efficiency is defined as

\[ \eta_B = \frac{1}{m_f h_v} [(\dot{m} + \dot{m}_f) C_{pr} T_{t_4} - \dot{m} C_{pc} T_{t_3}] \text{.} \]  (B.12)
where $\dot{m}_f$ is the fuel mass flow rate and $h_v$ is the heating value of the fuel. The efficiency of the burner is affected by pressure loss due to enthalpy addition at a finite Mach number as well as viscous losses. Also included are incomplete combustion losses that will also be present. Defining a fuel fraction
\[ f = \frac{\dot{m}_f}{\dot{m}}, \]  
and a new ratio of specific heats and temperatures
\[ \tau_\lambda = \frac{C_{pT} T_{t4}}{C_{pC} T_o}, \]  
Equation B.12 can be rewritten in terms of a fuel fraction as
\[ f = \frac{C_{pC} T_o}{h_v} (\tau_\lambda - \tau_R \tau_C \tau_D), \]  
or with simplification
\[ f = \frac{\tau_\lambda - \tau_R \tau_C \tau_D}{\left( \eta_B h_v \right) C_{pC} T_o} - \tau_\lambda. \]  
The fuel fraction will become important in determining the efficiency of the turbine. The efficiency of the burner is governed by the amount and type of fuel used as well as the mass flow and other properties of the incoming air. It is also affected by the total temperature specified at the inlet to the turbine section, which is governed by the technology level of the turbine.

Turbine

Between stations 4 and 5, the purpose of the turbine is to produce work through flow turning to drive the compressor while minimizing losses. For the turbine, the pressure and temperature ratios are defined as
\[ \pi_T = \frac{P_{t5}}{P_{t4}}, \quad \tau_T = \frac{T_{t5}}{T_{t4}}. \]
The efficiency of the turbine is given by

\[
\eta_T = \frac{w_t}{w_{ti}},
\]  
(B.18)

which is the actual work interaction of the air through the turbine divided by the ideal work interaction. Similar to the compressor, again using a the First Law, the efficiency of the turbine is defined by

\[
\eta_T = \frac{1 - \prod_{j=1}^{M} \left( 1 - \eta_{T,j} \left[ 1 - \pi_{T,j} \left( \frac{\gamma - 1}{\gamma T} \right) \right] \right)}{1 - \left( \prod_{j=1}^{M} \pi_{T,j} \right) \left( \frac{\gamma - 1}{\gamma T} \right)},
\]  
(B.19)

where \( \eta_{T,j} \) is the isentropic efficiency for each turbine stage. As with the compressor, the overall turbine efficiency is dependent on the efficiencies and pressure ratios across each stage. Therefore, the polytropic efficiency is again used. For the turbine, it is defined as

\[
e_T = \frac{dw_T}{dw_{Ti}},
\]  
(B.20)

since the actual differential work extracted will be less than the ideal differential amount. Assuming a constant polytropic turbine efficiency, Equation B.19 becomes

\[
\eta_T = \frac{1 - \pi_T \left( \frac{\gamma - 1}{\gamma T} \right)}{1 - \pi_T \left( \frac{\gamma - 1}{\gamma T} \right)},
\]  
(B.21)

with the temperature ratio given as

\[
\pi_T = \tau_T \left( \frac{\gamma T}{\gamma T - 1} \right).
\]  
(B.22)

Therefore the overall turbine efficiency is dependent on the temperature ratio and polytropic efficiency. The polytropic efficiency is again a design variable to be chosen based on technology. However, the temperature ratio is still undetermined. The governing relationship is the fact that the turbine and compressor are joined
mechanically. The work produced by the turbine directly feeds the compressor, minus some losses, which are combined into a mechanical efficiency, $\eta_M$ that is defined as the ratio of compressor work to turbine work, as

$$\eta_M = \frac{\dot{w}_C}{\dot{w}_T}. \quad (B.23)$$

Writing out both work rates and substituting into the above equation will allow the temperature ratio in the turbine to be solved for, giving

$$\tau_T = 1 - \frac{1}{\eta_M(1 + f)} \frac{\tau_D \tau_R}{\tau_\lambda} (\tau_C - 1). \quad (B.24)$$

The turbine efficiency is found from Equation B.21 and the pressure ratio from Equation B.22. Next, the equations through the exit will be considered, and afterward an afterburner will be added.

Nozzle

The purpose of the nozzle is to accelerate the flow for maximum thrust with minimum losses from the turbine to the exit of the engine. As with the inlet, the flow is subsonic, and the nozzle will be assumed to operate at the design point, meaning that the exit pressure of the nozzle is equal to the ambient atmospheric pressure. With no heat addition in the nozzle, the flow will be adiabatic. Using the First Law and Continuity equations, while assuming an isentropic expansion, a momentum equation gives an equation for the thrust, $F$, as

$$F = (\dot{m} + \dot{m}_f) V_9 - V_o + \frac{P_9 - P_o}{\dot{m}} A_9. \quad (B.25)$$

The pressure and temperature ratios are

$$\pi_N = \frac{P_9}{P_{t\tau}}, \quad \tau_N = \frac{T_9}{T_{t\tau}}. \quad (B.26)$$
With some algebraic manipulation, Equation B.25 can be cast in terms of Mach numbers as

\[
\frac{\dot{F}}{\dot{m}} = a_o M_o \left[ (1 + f) \sqrt{\frac{\gamma T R_T}{\gamma C R_C} - \frac{T_\theta}{T_o} \left( \frac{M_9}{M_o} \right) - 1} \right] \\
+ \left[ \frac{\sqrt{\gamma T R_T T_\theta}}{\gamma T M_9} (1 + f) \right] \left( 1 - \frac{P_o}{P_9} \right),
\]  

(B.27)

where the exit Mach number is given by

\[
M_9 = \sqrt{\frac{2}{\gamma T - 1} \left( \frac{P_{t_9}}{P_o} \left( \frac{\gamma T - 1}{\gamma T} \right) - 1 \right)}. 
\]  

(B.28)

with the temperature and pressure ratios being

\[
\frac{T_9}{T_o} = \frac{\tau_{\lambda T} C_{P_{T}}} {\left( \frac{P_{t_9}}{P_o} \right)^{\left( \frac{\gamma T - 1}{\gamma T} \right)}}, \quad \frac{P_{t_9}}{P_o} = \frac{P_o}{P_9} \pi_{R} \pi_{D} \pi_{C} \pi_{B} \pi_{T} \pi_{N}.
\]  

(B.29)

The design variables for the nozzle are the pressure ratio at the exit \( \frac{P_{t_9}}{P_o} \) and the nozzle pressure ratio, \( \pi_{N} \). Because this nozzle is subsonic, it will be assumed that it operates at the design point, that the exit pressure, \( P_9 \), is equal to the ambient pressure, \( P_o \).

Afterburner

To increase thrust, additional fuel is combusted prior to the nozzle between stations 6 and 7, which changes the temperature and pressure ratios here, defined as

\[
\pi_{AB} = \frac{P_{t_7}}{P_{t_6}}, \quad \tau_{AB} = \frac{T_{t_7}}{T_{t_6}},
\]  

(B.30)

as well as the Gas constant

\[
R_{AB} = \frac{\gamma_{AB} - 1}{\gamma_{AB}} C_{P_{AB}}.
\]  

(B.31)
Another ratio of specific heats and temperatures is also defined as

$$\tau_{\lambda AB} = \frac{C_{pAB} T_t}{C_{pC} T_o}. \quad (B.32)$$

The fuel fraction can be found by considering the First Law with the addition of fuel as a simple heat addition

$$f_{AB} = (1 + f) \frac{\tau_{\lambda AB} - \tau_{\lambda T}}{\frac{h_{\lambda AB}}{C_{pC} T_o}} - \tau_{\lambda AB}. \quad (B.33)$$

The thrust equation is modified since $T_t$ changes with the addition of fuel.

$$\frac{F}{m} = a_o M_o \left[ (1 + f + f_{AB}) \sqrt{\frac{\gamma_{AB} R_{AB} T_9}{\gamma_{C} R_{C} T_o}} \left( \frac{M_9}{M_o} \right) - 1 \right] + \left[ \frac{\sqrt{\gamma_{AB} R_{AB} T_9}}{\gamma_{AB} M_9} (1 + f + f_{AB}) \right] \left( 1 - \frac{P_o}{P_9} \right), \quad (B.34)$$

where the exit Mach number is given by

$$M_9 = \sqrt{\frac{2}{(\gamma_{AB} - 1)}} \left( \frac{P_9 \left( \frac{\gamma_{AB} - 1}{\gamma_{AB}} \right)}{P_o} \right)^{\frac{1}{(\gamma_{AB} - 1)}}, \quad (B.35)$$

and the pressure and temperature ratios are

$$\frac{T_9}{T_o} = \frac{\tau_{\lambda AB} \tau_{\lambda T} C_{pC}}{(P_9 \left( \frac{\gamma_{AB} - 1}{\gamma_{AB}} \right)) C_{pAB}}, \quad \frac{P_9}{P_o} = \frac{P_9}{P_9} \pi_R \pi_D \pi_C \pi_B \pi_T \pi_{AB} \pi_N. \quad (B.36)$$

The addition of afterburning will increase the temperature ratio and also decrease the pressure ratio, as $\pi_{AB} < 1$.

**B.2 Turbojet Design Variables**

Now that the component efficiencies have been discussed, all of the design variables for the turbojet will be listed here for convenience. They are listed in the following Table. This completes the analysis of the turbojet. Next, the unmixed Turbofan will be considered.
Table B.2

**TURBOJET INPUT VARIABLES**

<table>
<thead>
<tr>
<th>Variable</th>
<th>Definition</th>
<th>Variable</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
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<td>$P_o/P_9$</td>
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</tr>
</tbody>
</table>

B.3 Unmixed Dual-Spool Turbofan with Losses

The turbofan is primarily different from a turbojet because of the inclusion of a bypass stream created by the fan. This increases the overall efficiency by decreasing the core flow kinetic energy while increasing the mass flow through the engine.

Referring to Figure B.2.1 b, the additional turbofan stations will now be explained. Station 2 to 3′ refer to the Bypass Fan, and station 3′ to 9′ depict the bypass nozzle, because an afterburner is not included here. Note that the dual spool is not shown here, but is referenced by splitting stations up in half. So, station 3′ to 2.5 is the low pressure compressor spool, while 2.5 to 3 the high pressure compressor spool. Station 3 to 4 is the burner, as before, 4 to 4.5 is the high pressure turbine spool, 4.5 to 5 the low pressure spool, and 5 to 9 the exit nozzle, again without considering an afterburner.

In the analysis of the Turbofan, the equations for the inlet diffuser, burner, and nozzle remain the same, and because the equations have previously been stated above, will not be repeated here. However the values of these parameters will be altered by the addition of the fan and dual-spool sections. For the following discus-
tion, only the changes in the compressor and turbine equations will be addressed, in addition to the explanation of the fan and dual-spools sections. With regard to the spool sections, subscript ‘L’ will denote the low pressure spool variables, and subscript ‘H’ the high pressure spool variables.

Bypass Fan

The Bypass Fan takes air from the inlet and increases the mass flow of the overall engine while decreasing the core exit velocity. A new ratio must be defined to express the relation between the amount of mass flow through the core, $\dot{m}_C$, and the amount of mass flow through the bypass fan, $\dot{m}_F$, which is termed the Bypass Ratio. It is given by

$$\alpha = \frac{\dot{m}_F}{\dot{m}_C}. \quad (B.37)$$

The total amount of air entering the engine is then

$$\dot{m}_o = \dot{m}_F + \dot{m}_C = (1 + \alpha) \dot{m}_C, \quad (B.38)$$

and the air leaving the exit nozzle of the core becomes

$$\dot{m}_9 = (1 + f) \dot{m}_C. \quad (B.39)$$

The pressure and temperature ratios for the fan section are simply defined as

$$\pi_F = \frac{P_{t'}'}{P_{t2}}, \quad \tau_F = \frac{T_{t'}'}{T_{t2}}. \quad (B.40)$$

So, just as before, the equation for the temperature ratio becomes

$$\tau_F = \pi_F \left( \frac{(\gamma_C - 1)}{\gamma_C s_F} \right) \quad (B.41)$$

and the bypass ratio, pressure ratio, and polytropic efficiency of the fan become additional design variables.
Bypass Nozzle

The Bypass Nozzle accelerates the flow just as in the Turbojet. For this analysis it will be assumed that the exit pressure is the same as the inlet pressure $P_o$ both for the fan and for the core nozzle. Because flow aft of the nozzle remains unmixed with the core flow, the pressure and temperature ratios here can also be easily described by

$$\pi_{FN} = \frac{P_{t9'}}{P_{t9}}, \quad \tau_{FN} = \frac{T_{t9'}}{T_{t9}}.$$  \hfill (B.42)

These are a function of the fan geometry, and also become new design variables. The pressure ratio and temperature ratios from inlet to fan nozzle exit are

$$\frac{T_{t9'}}{T_o} = \tau_R \tau_D \tau_F \pi_{FN}, \quad \frac{P_{t9'}}{P_o} = \frac{P_o}{P_{t9'}} \pi_R \pi_D \pi_F \pi_{FN},$$  \hfill (B.43)

and the Mach number for the bypass flow is defined in the same manner as in Equation B.35

$$M_{9'} = \sqrt{\frac{2}{(\gamma_C - 1)} \left( \frac{P_{t9'}}{P_o} \left( \frac{\gamma_C - 1}{\gamma_C} \right) - 1 \right)},$$  \hfill (B.44)

with

$$\frac{T_9}{T_o} = \frac{T_{t9'}}{T_o} \left( \frac{P_{t9'}}{P_o} \right)^{-\frac{1}{\gamma_C}} C_{pc}, \quad a_{9'} = \sqrt{\frac{T_{9'}}{T_o}}.$$  \hfill (B.45)

Completing a momentum balance as with the Turbojet, the thrust equation for the fan is

$$\frac{F}{\dot{m}} = \left( \frac{a_o \alpha}{1 + \alpha} \right) \left( \frac{a_{9'} M_{9'} - M_o}{a_k} \right) + \sqrt{\frac{\gamma_C R_C T_{t9'}}{T_o} T_o} \left( \frac{1 - P_o}{P_{t9'}} \right).$$  \hfill (B.46)

This will be combined with the thrust equation of the core to determine the overall engine thrust produced.
Low Pressure Spool

The splitting of the compressor and turbine sections into two spools implies that two separate work balances govern the overall thermodynamic cycle of the engine, with one balance equation for each spool. As was mentioned in the discussion of the Turbojet equations, each spool has a unique polytropic efficiency and mechanical efficiency.

Considering first the front half of the low pressure spool, the pressure and temperature ratios are given by

\[ \pi_{C_L} = \frac{P_{t_{2.5}}}{P_{t_2}}, \quad \tau_{C_L} = \frac{T_{t_{2.5}}}{T_{t_2}}. \]  

(B.47)

The low pressure spool work balance is

\[ \dot{W}_F + \dot{W}_{C_L} = \eta_{M_L} \dot{W}_{T_L}. \]  

(B.48)

The fan is attached to the low pressure spool, and actually makes up the first few compression stages of the compressor. So the low pressure spool is made up the fan and low pressure compressor which are both driven by the low pressure turbine. The pressure ratio of the entire low pressure spool between stations 2 and 2.5 is dependent on the fan pressure ratio and the low pressure compressor pressure ratio, as given by

\[ \bar{\pi}_{C_L} = \pi_F \pi_{C_{LP}}, \]  

(B.49)

where ‘\(C_{LP}\)’ denotes the compressor section of the low pressure spool and ‘\(L\)’ the entire forward half of the low pressure spool. Both pressure ratios are design variables. The temperature ratios for this section are

\[ \tau_{C_L} = \tau_F \tau_{C_{LP}}, \quad \tau_{C_{LP}} = \pi_{C_{LP}}^{\left(\frac{\gamma - 1}{\gamma C_{C_{LP}}}\right)}. \]  

(B.50)
If the efficiency of the compressor is solved for, it is

\[ \eta_{CL} = \frac{\pi_{CL} \left( \frac{\gamma_{C} - 1}{\gamma_{C}} \right)}{\tau_{CL} - 1}. \]  

(B.51)

For the low pressure turbine, the temperature and pressure ratios are

\[ \pi_{TL} = \frac{P_{t_5}}{P_{t_{4.5}}}, \quad \tau_{TL} = \frac{T_{t_5}}{T_{t_{4.5}}}. \]  

(B.52)

Solving the work balance for this flow, gives

\[ \tau_{TL} = 1 - \frac{\tau_{R} \tau_{D} \left[ \alpha (\tau_{F} - 1) + (\tau_{CL} - 1) \right]}{\eta_{ML} (1 + f) \tau_{L} \tau_{TH}}, \]  

(B.53)

with

\[ \pi_{TL} = \tau_{TL} \left( \frac{\tau_{T} \left( \frac{\gamma_{T}}{(\gamma_{T} - 1)e_{TL}} \right)}{\gamma_{T}} \right). \]  

(B.54)

The efficiency of the low pressure turbine is

\[ \eta_{TL} = \frac{1 - \tau_{TL}}{1 - \pi_{TL}}. \]  

(B.55)

High Pressure Spool

For the high pressure compressor spool, the pressure and temperature relations are

\[ \pi_{CH} = \frac{P_{i_3}}{P_{i_{2.5}}}, \quad \tau_{CH} = \frac{T_{i_3}}{T_{i_{2.5}}}. \]  

(B.56)

or

\[ \tau_{CH} = \frac{\left( \frac{(\gamma_{T} - 1)e_{CH}}{\gamma_{T}} \right)}{\pi_{CH}}. \]  

(B.57)

while for the high pressure turbine, they are

\[ \pi_{TH} = \frac{P_{i_{4.5}}}{P_{i_4}}, \quad \tau_{TH} = \frac{T_{i_{4.5}}}{T_{i_4}}. \]  

(B.58)
The work balance for the high pressure spool is simpler than for the low pressure spool, and is similar to the Turbojet work balance. It is

\[ \dot{W}_{CH} = \eta_{M_H} \dot{W}_{TH}. \]  

(B.59)

Solving this work balance for the temperature ratio in the turbine yields

\[ \tau_{T_H} = 1 - \frac{\tau_R \tau_D \tau_{CL} (\tau_{CH} - 1)}{\eta_{M_H} (1 + f) \tau_{\lambda}}, \]  

(B.60)

where the pressure ratio is given by

\[ \pi_{T_H} = \tau_{T_H} \left( \frac{\tau}{\tau - 1} \right)^{\pi_{T_H}}. \]  

(B.61)

Now, the pressure ratio through the core from the inlet to station 9 is the combination of both spools

\[ \frac{P_{t9}}{P_o} = \pi_R \pi_D \pi_{CL} \pi_{BH} \pi_{TH} \pi_{TL} \pi_N. \]  

(B.62)

where \( \pi_C \) is the overall compressor pressure ratio comprised of the low and high pressure compressor spool ratios

\[ \pi_C = \pi_{CL} \pi_{CH}. \]  

(B.63)

Similarly, the temperature ratio is defined as

\[ \tau_C = \tau_{CL} \tau_{CH}. \]  

(B.64)
The combination of Thrust from the Core and the Fan leads to a slightly modified form of Equation B.34, which is

\[
\frac{F}{\dot{m}} = a_o M_o \left[ (1 + f) \sqrt{\frac{\gamma_T R_T T_9}{\gamma_C R_C T_o}} \left( \frac{M_9}{M_o} \right) - 1 \right] \\
+ \left[ \frac{\sqrt{\gamma_T R_T T_9}}{\gamma_T M_9} (1 + f) \right] \left( 1 - \frac{P_o}{P_9} \right) \\
+ \left( \frac{a_o \alpha}{1 + \alpha} \right) \left( \frac{a_o y'}{a_o} M_9' - M_o \right) + \frac{\sqrt{\gamma_C R_C T_o}}{\gamma_C M_9} (1 + \alpha) \left( 1 - \frac{P_o}{P_9} \right).
\] (B.65)

Also, the efficiencies for the high pressure turbine and high pressure compressor are given by

\[
\eta_{TH} = \frac{1 - \tau_{TH}}{1 - \left( \frac{(\gamma_C - 1)}{\gamma_T} \right) \tau_{TH}}, \quad \eta_{CH} = \frac{\pi_{CH} - 1}{\tau_{CH} - 1}.
\] (B.66)

B.4 Turbofan Design Variables

The list of design variables for the Turbofan are now listed in Table B.3 for convenience.

**Table B.3**

**TURBOFAN INPUT VARIABLES**

<table>
<thead>
<tr>
<th>Variable</th>
<th>Definition</th>
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<th>Definition</th>
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<tbody>
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<td>$M_o$</td>
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<td>$P_o/P_{9'}$</td>
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</table>
B.5 Metrics of Engine Improvement

From the above equations, the component efficiencies are defined by inflow properties and design choices. The metrics for overall engine improvement are the thrust specific fuel consumption, TSFC, which is the amount of fuel consumed per unit of thrust produced, the compressor efficiency, $\eta_C$, and the turbine efficiency, $\eta_T$. The lower the values of TSFC or the higher the values of $\eta_C$ and $\eta_T$, the more efficient the engine. TSFC is defined as

$$TSFC = \frac{\dot{m}_f}{T} = \frac{f_{total}}{F/m},$$

(B.67)

where $f_{total}$ is the sum total of all fuel added.

Because the improvement in these properties are important for this discussion, rather than the absolute value, the improvements in TSFC, $\eta_T$, and $\eta_C$ will be measured as the percent changes from an ‘original’ design to an ‘improved’ design, as defined by

$$\Delta TSFC = \frac{TSFC_{original} - TSFC_{improved}}{TSFC_{original}} \times 100,$$

(B.68)

$$\Delta \eta_C = \frac{\eta_C^{improved} - \eta_C^{original}}{\eta_C^{original}} \times 100,$$

(B.69)

$$\Delta \eta_T = \frac{\eta_T^{improved} - \eta_T^{original}}{\eta_T^{original}} \times 100.$$  

(B.70)

B.3 Engine Properties

A distinct baseline engine profile will be considered for the Turbojet and Turbopan in order to evaluate the above equations. The properties for the turbojet are given in Table B.4 and were taken from Jumper [69]. They apply to a subsonic cruise engine with optimal values for all parameters.

The unmixed dual spool turbofan that will be considered is a Pratt & Whitney
## Table B.4

**TURBOJET COMPONENT PROPERTIES**

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<td>J/(kgK)</td>
</tr>
<tr>
<td>$\pi_D$</td>
<td>0.9425</td>
<td>-</td>
</tr>
<tr>
<td>$\pi_B$</td>
<td>0.98</td>
<td>-</td>
</tr>
<tr>
<td>$\pi_{AB}$</td>
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</tr>
<tr>
<td>$\pi_N$</td>
<td>0.96</td>
<td>-</td>
</tr>
<tr>
<td>$\pi_C$</td>
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</tr>
<tr>
<td>$e_C$</td>
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<td>-</td>
</tr>
<tr>
<td>$e_T$</td>
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</tr>
<tr>
<td>$\eta_B$</td>
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<td>-</td>
</tr>
<tr>
<td>$\eta_{AB}$</td>
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<td>-</td>
</tr>
<tr>
<td>$\eta_M$</td>
<td>0.99</td>
<td>-</td>
</tr>
<tr>
<td>$\tau_D$</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>$\tau_N$</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>$T_{i4}$</td>
<td>1300</td>
<td>K</td>
</tr>
<tr>
<td>$P_o/P_9$</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>$\tau_{AA}$</td>
<td>8.8</td>
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</tr>
</tbody>
</table>

JT8D-219. Values for this engine are reported in Table B.5.

### B.4 Results and Analysis

In order to understand how the individual component variables affect the engine thrust specific fuel consumption, compressor efficiency, and turbine efficiency, all
Table B.5

TURBOFAN COMPONENT PROPERTIES

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M_o$</td>
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<td>$T_o$</td>
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<td>$P_o$</td>
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<td>Pa</td>
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<td>$\gamma_C$</td>
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<tr>
<td>$C_{pc}$</td>
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<td>$J/(kgK)$</td>
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<tr>
<td>$\gamma_T$</td>
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</tr>
<tr>
<td>$C_{pf}$</td>
<td>1105.3</td>
<td>$J/(kgK)$</td>
</tr>
<tr>
<td>$h_v$</td>
<td>45357000</td>
<td>$J/kg$</td>
</tr>
<tr>
<td>$\alpha$</td>
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<td>$\pi_D$</td>
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<tr>
<td>$\pi_{CLP}$</td>
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<tr>
<td>$\pi_F$</td>
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</tr>
<tr>
<td>$\pi_C$</td>
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<td>-</td>
</tr>
<tr>
<td>$\pi_{FN}$</td>
<td>0.98</td>
<td>-</td>
</tr>
<tr>
<td>$\pi_B$</td>
<td>0.91</td>
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</tr>
<tr>
<td>$\pi_N$</td>
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<td>-</td>
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<td>$e_F$</td>
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<td>$e_{CLP}$</td>
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<tr>
<td>$e_{CH}$</td>
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<tr>
<td>$e_{TH}$</td>
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<tr>
<td>$e_{TL}$</td>
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<tr>
<td>$\eta_B$</td>
<td>0.96</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{MB}$</td>
<td>0.96</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{ML}$</td>
<td>0.97</td>
<td>-</td>
</tr>
<tr>
<td>$P_o/P_0$</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>$T_{t_4}$</td>
<td>1300</td>
<td>K</td>
</tr>
<tr>
<td>$\tau_N$</td>
<td>0.99</td>
<td>-</td>
</tr>
<tr>
<td>$\tau_D$</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>$\tau_{FN}$</td>
<td>1</td>
<td>-</td>
</tr>
</tbody>
</table>

Independent component quantities were varied systematically between ±6% of their initial value for both the Turbojet and Turbofan. Each component value was varied, one at a time, while all others were kept constant, and the resulting dependence on
the compressor and turbine efficiencies as well as the TSFC were recorded. The results consist of a series of plots, showing the percent change of the independent component variable on the x-axis and the percent change of the measure of performance on the y-axis, with the original value of the component variable listed above the plot.

It is important to note that some of the original component parameters, for example, the component pressure ratios, have values near unity. Because of the way in which different variables are defined in the sections preceding, they are not theoretically able to go above a value of one. Therefore, small percentage changes in these values are restricted to keep the value at or below unity. This means that for some variables, the entire ±6% range is not plotted, because of the limitation of the value of that variable. Also, for plots that show no results and are blank, the component variables for these plots exhibited no dependence whatsoever on the independent variable, and so these results were not plotted.

B.1 Turbojet

Thrust Specific Fuel Consumption

Beginning with the Turbojet, the effect on Thrust Specific Fuel Consumption is examined. In Figure B.2.2, the dependence of engine parameters on TSFC, and therefore the inverse of the overall engine efficiency, are shown.

From a first inspection, the largest modifier of the TSFC is the polytropic compressor efficiency. For a 6% increase in $e_C$, the TSFC increases by almost 5%. The other highly dependent variables for TSFC are $\eta_{AB}$, giving a maximum increase of 2%, $\eta_B$ and $e_T$, giving a maximum increase of about 1.5%, and $\pi_D$, $\pi_B$, and $\pi_N$, all increasing TSFC by roughly 1%. However, with small increases, other parameters had an adverse effect on the fuel consumption.
Compressor and Turbine Efficiencies

For the compressor, $\eta_C$ is dependent only on $e_C$. A 6% increase in $e_C$ causes about a 9% increase in efficiency, which is very impressive. This can be understood by referring to Equation B.9. Similarly for the turbine, the most important parameter here is the turbine polytropic efficiency, $e_T$, which caused about a 5% improvement in efficiency of the turbine.

B.2 Dual-Spool Turbofan

Moving on to the Turbofan, the analysis here will look first at the improvement in TSFC, and then at the improvements in the individual high and low spool compressor and turbine efficiencies. For the Turbofan, the results for each performance parameter are split up into two figures because of the large number of independent
component variables.

Thrust Specific Fuel Consumption

The Turbofan results on TSFC are shown in Figures B.2.3 and B.2.4. From these plots, the maximum increase of TSFC is 4%, that is due to an increase in $\eta_B$ of 4%. Also important are $\pi_B$, $\pi_N$, $\eta_{MH}$, $e_F$, $e_{CH}$, $e_{TH}$, and $e_{TL}$, all giving about a 1.8–2.3% TSFC increase each. Thus, the significant parameters here include burner values, as well as the polytropic efficiencies, the spool efficiencies, and the majority of the pressure ratios.

![Figure B.2.3. Change in Turbofan TSFC with variation in Component Variables](image-url)

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Figure B.2.4. Change in Turbofan TSFC with variation in Component Variables
Compressor and Turbine Spool Efficiencies

Again for the Turbofan, the efficiencies of the compressor and turbine spools of both the high and low pressure sections are only dependent on the respective polytropic efficiencies because of the way in which the equations were defined. Additionally, the low pressure compressor is also dependent on the fan polytropic efficiency as well. The low pressure compressor efficiency was able to be increased by 4% with a 6% increase in $e_F$, while the effect of $e_{C_{LP}}$ was less, approximately 2%. For the high pressure compressor spool, a 6% increase in $e_{C_{H}}$ caused a 9.5% increase in the compressor efficiency here. For both the high pressure and low pressure turbine spools, the polytropic efficiencies each caused a 5.1% increase the spool efficiencies.

B.5 Conclusions

From the previous results, it is clear that immense gains in engine performance can be achieved if the proper variables are focused upon. For a Turbojet, the significant parameters of merit in order from most to least important are the polytropic efficiency of the compressor and to a lesser extent, of the turbine, and the pressure ratios. For a Turbofan, the important variables are the pressure ratios through the fan, burner, and low pressure compressor, the burner efficiency, the polytropic efficiencies, and the mechanical efficiency through the high pressure spool. Of special importance to both engines are the polytropic efficiencies for each component, which have an immense effect on all metrics of improvement. Tip clearance flow loss reduction would have a particular effect on the polytropic efficiencies in the compressor or turbine because of the benefit of loss reduction on improvement of the total pressure ratio through the stages.

To improve the overall Thrust Specific Fuel Consumption of an engine, the first place to try to affect the component efficiencies is within the compressor, where the
largest improvement was found. Next, the turbine should be concentrated on, as changes here were favorable as well. In addition, the burner efficiency and mechanical efficiency of any high pressure spools can effect positive changes in the engine efficiency, and should be addressed. If the turbine and compressor sections do not yield sufficient gains, looking to increase the fan efficiency or the burner efficiencies should benefit the engine designer.

What remains to be seen from this qualitative analysis is how feasible implementation of these efficiency gains are, as each section of the engine presents a unique set of issues to address. In a compressor, the adverse pressure gradient and probability of stall are complex problems, while in a turbine, the high temperature, high pressure environment causes issues with blade degradation and efficiency loss. However, it is promising to recognize that even a fraction of a 9% increase in TSFC that was found above would be extremely beneficial for engine improvement. Also, even just a 1% improvement in certain component parameters would cause essentially a proportional change in the overall engine efficiency.
REFERENCES


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