AN EXPERIMENTAL STUDY OF BLADE TIP CLEARANCE SUCTION APPLIED TO A HIGH SPEED COMPRESSOR

by

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ABSTRACT

A computational and experimental study was carried out to determine whether blade tip suction can increase compressor efficiency. Compressor flow fields contain concentrated regions of high entropy fluid near the blade surfaces. Blade suction can remove this fluid, lowering the entropy of the compressor throughflow, thereby avoiding a thermodynamic inefficiency, ultimately raising the compressor efficiency 0.25 percentage points for 1% flow suction.

Blade suction also can reduce unwanted aerodynamic flow structures such as secondary flows and the tip clearance vortex. Low speed fan blade tip suction analysis using a three-dimensional viscous solver calculates a lower tip entropy and higher tip efficiency with fluid suction of only 0.2%.

Implementation of blade tip suction on a high speed transonic compressor proved the possibility of blade tip suction. Results show strong effects on the upstream shock structure, including the disappearance of one shock wave entirely. Measurements downstream of the rotor indicate a change in the flow field exiting the suction blade rows, possibly due to a weaker tip vortex.
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Nomenclature

A  Area
C_p  Constant pressure specific heat
D  Hydraulic Diameter
f  Friction factor
k  constant
M  Mach number
m  Compressor mass flow rate
\delta m  Suction mass flow rate
P  Pressure
R  Gas Constant
Re  Reynold's Number
r  radius
T  Temperature
s  Specific entropy
\Delta s  Specific entropy change
V  Cross Gap Flow Velocity
V_1  Supply Tank Volume
V_5  Suction Dump Tank Volume
W  Work
\pi  Compressor Pressure Ratio
\delta  Bypass bleed percentage
\tau  Compressor Temperature Ratio
\gamma  Specific heat ratio
\lambda  Tip gap/Maximum blade thickness ratio
\rho  Density
\rho_1  Gap entrance (pressure side) density
\rho_2  Gap exit (suction side) density
\eta  Efficiency
subscripts

A Initial point
B Bleed point
C Core flow after bleed
C' Suction flow after bleed
D Final point, no bleed
D' Core flow final point, after bleed
E Suction flow final point, after bleed
g Tip clearance gap
H Hub
P Suction Passage
PS Pressure side of blade
T,t Blade tip
1 initial point
2 final point
ref Reference point
b Compressor with bleed
nb Compressor without bleed
v viscous stream
nv non-viscous stream
s stage
\( \theta_s \) Swirl or circumferential flow after the compressor
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Chapter I
Introduction

Reduction of compressor losses and improvement in the overall Brayton cycle efficiency are always goals in turbomachinery research. A simple cycle analysis shows that the ideal Brayton cycle efficiency is just a function of the compressor pressure ratio. As the compressor pressure ratio increases, so does the compressor exit temperature. Today's gas generators push the allowable temperature limit. Increases in compressor efficiency then also increase the cycle efficiency. Gas generator performance is so closely linked to the compressor efficiency that even a half percent gain in polytropic efficiency is significant. A 'new' idea, blade surface suction, might offer the chance to improve the compressor efficiency by much more than just a half percent.

Losses in modern gas turbines are fairly well correlated and tabulated, and yet very poorly understood. Bladerow losses are currently generalized into four main categories as follows:

- friction losses; Friction retards the motion of fluid flow, increasing the entropy of the fluid in the process. This higher entropy fluid then forms boundary layers on compressor surfaces.
The work deficit associated with the higher entropy fluid appears as blades wakes, centers of low total pressure trailing in roughly spanwise columns behind the blades. Friction only accounts for three to four percentage points of the ten percent inefficiency found on modern compressors.

- Secondary flow losses; Secondary flow losses are also associated with the viscous boundary layers. Instead of generating the loss directly, the secondary flows turn the boundary layers away from the mean flow direction. A shearing layer between the two flows then generates a loss, commonly carried downstream in the core of a vortex.

- Tip clearance losses; Similar to the secondary flow losses, tip clearance flow introduces a mixing loss. A fluid jet emerging from the clearance gap at an angle mixes with the core flow in the neighboring blade passage. The mixing losses increase with cross gap mass flow, where the cross gap mass flow is directly proportional to the tip clearance.

- Shock losses; In transonic machines, shock waves increase entropy, thus generating loss in the flow. Shock losses differ from the other losses mentioned in that they occur in the mainstream, or core, flow, while the other losses originate in the boundary layers.

Of these losses, only frictional losses are theoretically unavoidable. Indeed the improving ability of computational fluid dynamics has already reduced the losses from levels present just a few years ago. Still, other methods should exist, but remain to be found to eliminate these other losses.

Kerrebrock suggests a way to minimize the impact of these inefficiencies on the thermodynamic cycle (ref. 10). In the Brayton cycle, compression of high entropy fluid requires more work than the same compression of low entropy fluid. This occurs because the
Brayton cycle has constant pressure heat addition while the isobars diverge with increasing entropy. The easiest way to see this is to consider the work of compression as shown below.

\[ W = m \ CpT_1 \left( \frac{T_2}{T_1} - 1 \right) \]  

Equation 1.1 relates the compressor work input to the change in enthalpy of the fluid. At higher entropies the inlet temperature \( T_1 \), increases due to the diverging isobar. Consequently, for a given pressure ratio, represented by \( T_2/T_1 \), the work of compression increases as the fluid entropy increases. Hence, compression of higher entropy fluid is less efficient than the compression of low entropy fluid.

Fluid suction from compressor blade surfaces has the potential to reduce many of the inefficiencies of the modern compressor. Fluid suction is a perennial idea, always seeming to receive lots of attention but producing very little. Studies of casing suction concluded that better results were obtained by adding an additional blade row (ref. 10). The current study of suction differs in that the suction slots will be located on the blade surfaces themselves. This is particularly important in that most of the high entropy fluid is generated on these blade surfaces. Careful placement of the suction slots then permits targeting of particularly inefficient areas in the compressor. Blade surface suction will remove the high entropy fluid, raising the thermodynamic efficiency. Also, through proper placement of the
slots, suction may reduce aerodynamic losses. For these reasons, blade surface suction provides an inspirational topic of research.

Other benefits of blade surface suction may exist that are beyond the scope of the current research, such as a) an increase in stage pressure ratio and b) an increase in stall margin. Both ideas are closely coupled to the current idea of blade surface suction. Indeed, the stage pressure ratio will increase with increasing efficiency due to lower blade losses. Here, however, suction might be able increase stage pressure ratio by a larger amount than the increase due to efficiency. These ideas are presented briefly below.

Current stage pressure ratio is limited by an abrupt increase in loss. This jump in loss correlates to a D-factor around 0.55. The flow an adverse pressure gradient in diffusion on the blade suction surface. Eventually the pressure gradient increases enough that it reverses the direction of the flow in the boundary layer near the blade surface. The resulting boundary layer separation produces a large inefficiency. It follows then that the stage pressure ratio is limited by the separation of the boundary layer. Flow removal on the suction surface can suppress this boundary layer growth, potentially allowing for a larger stage pressure rise.

Studies of airfoil stall show that suction surface boundary layer control does permit a larger pressure rise. With wings, the adverse pressure gradient, or pressure rise, increases with angle of attack. The wing stalls, reducing lift, when the boundary layer
separates. Wings with suction achieve a much larger angle of attack than those without, all other conditions equal. Even so, airfoil designers choose passive measures such as turbulence generators over suction to control the boundary layers.

Compressors may gain more benefit from stall suction control than do wings. Compressor viscous layer suction reduces the thermodynamic inefficiency which is not a concern for wings, since there is no further compression of the fluid. Compressor airfoils are consistently highly loaded, while wings only occasionally operate near stall. Additionally, a compressor suction system should be lighter than that of an airfoil. Compressor blades have much smaller aspect ratios, reducing both the amount of suctioned fluid and the weight and complexity of the suction system. Finally, compressors operate above ambient pressure, suggesting the use of a passive suction system.

More controversy surrounds the potential gain in stall margin from blade surface suction. Currently there is no definitive explanation for compressor stall. Experiments show that compressors stall for differing reasons. Methods that increase stall margin in some compressors are completely ineffective in other cases. Recent and ongoing studies at MIT suggest that flow 'blockage' leads to stall. In this case, stall originates as boundary layers expand, crowding out the core flow. Measures that reduce stall on one machine are geared toward a particular stalling boundary layer. Thus the measure does not work if a different
boundary layer causes stall. Given that blockage causes stall, blade surface suction should increase stall margin through boundary layer control. This gain can only be realized if suction controls the boundary layer inciting stall on the particular compressor.

In light of the simple explanation presented above, a more detailed study is required. The removal of high entropy fluid will lower the average entropy remaining in the flow path, thus reducing the thermodynamic inefficiency. At the same time, however, extra fluid must be compressed to the bleed point, requiring extra work. Considering the goal, evaluating the effect of blade suction on compressor efficiency, several steps summarizing the work described within are necessary. The first step is to determine the potential thermodynamic gain from suction. This calculation does not account for gains in aerodynamic efficiency as a result of the suction. The second step is an attempt to calculate the effects of tip suction with a three-dimensional computational code. The last step is an experimental high speed rotor test of the effects of tip suction. Evaluation of a complete fluid suction system is beyond the scope of the current work, which only establishes the preliminaries for an exciting new approach in compressor design.
Thermodynamic analysis shows that the entropy compression loss exists and results in up to several percentage points in efficiency for a compressor. The derivation commences with the standard Brayton cycle plotted on a T-s diagram. Using the diagram as a guide, the work for each cycle is derived. The ratio of bleed work to the baseline compressor work provides an accurate comparison by which to evaluate bleed. This cycle analysis shows that the removal of high entropy fluid from modern compressors can indeed reduce the work required by the compressor. The work reduction results in an efficiency gain of a few percent.

The baseline compression follows a standard Brayton cycle representation with inefficiencies. The compression progresses smoothly from point A, the initial pressure $P_i$, to point D, the delivery pressure $P_f$, as plotted in Figure 2.1. The polytropic efficiency, $\eta_D$, provides a measure of the thermodynamic efficiency.

The bleed cycle follows the same path as the baseline case up until the bleed pressure $P_B$, at point B. Bleed of the high entropy fluid occurs at constant pressure, as shown in Figure 2.2. As high entropy fluid leaves the flow, the entropy of the remaining fluid falls from point B to point C before resuming the compression to point $D'$. The high entropy fluid, now at point $C'$, expands
isentropically back to the initial pressure at point E. While the baseline compressor follows one curve, three separate processes describe the bleed compression with work recovery.

The major assumption so far is that the fluid can be divided into two streams. One stream, the core flow, notices hardly any entropy rise while the second stream, the viscous surface stream, contains most of the entropy rise. The entropy increase from the inlet conditions is divided into the constituent entropy rises of the non-viscous ($\Delta s_{nv}$) and the viscous layers ($\Delta s_v$)

$$\frac{\Delta s}{C_p} = \frac{m_{nv} \Delta s_{nv}}{m C_p} + \frac{m_v \Delta s_v}{m C_p}$$ (2.1)

The second assumption is that it is possible to remove only the high entropy fluid, leaving the low entropy fluid behind. The mass-averaged entropy in the remaining flow then decreases with fluid suction, while the suctioned fluid has a higher entropy as shown in Figure 2.3.

This assumption appears to be valid in compressors. A major source of entropy rise in the compressor is the frictional drag on the blade surfaces. Along the stage, the entropy remains fairly close to the blade surface, only mixing into the core flow after the end of the blade row. As this is the case, only the entropy rise of the suctioned blade row can be removed. This model then does not capture large accumulations of high entropy fluid as might be found on, for example, stator pressure sides.
The average compressor efficiency is a measure of the mass-averaged entropy rise in the compressor. Starting with a standard definition for entropy,

\[ \Delta s = C_p \ln \frac{T}{T_{ref}} R \ln \frac{P}{P_{ref}} \]

substitution leads to Equation 2.3 which relates the average entropy rise to the polytropic efficiency of the compressor for the given pressure ratio.

\[ \frac{\Delta s}{C_p} = \frac{\gamma - 1}{\gamma} \left[ \frac{1}{\eta_d} - 1 \right] \ln \pi_B \]  

(2.3)

Setting the initial entropy to zero ( \( s_A = 0 \) ), then Equation 2.3 establishes the entropy of the fluid at point B before any fluid is removed.

The disposition of the suctioned fluid influences the effectiveness of blade suction. During compression to the suction pressure, the withdrawn fluid absorbs work. In removal some work might be recoverable, further increasing the effectiveness of fluid suction. Two cases limit the potential gains from blade surface suction. First, the suctioned fluid is dumped without recovering any work (essentially a recovery efficiency of zero). Second, the suctioned fluid expands isentropically back to the initial conditions at \( P_i \), but at the higher temperature \( T_E \). In the following analysis, the expansion term will be kept separately to
the right side of the equations to facilitate easy analysis of the two
limits.

The first step is to establish a basis of comparison between
the two compression cycles. Figure 2.1 shows the baseline
compression cycle for mass \( m_d \) compressed from \( P_i \) to the final
pressure, \( P_f \). Equation 1.1 shows the work for the baseline
compressor, rewritten here in terms of the compressor pressure
ratio.

\[
W_{nb} = m_D C_p T_A \left( \frac{\gamma - 1}{\gamma m_D} - 1 \right)
\]  

(2.4)

With bleed, the initial mass flow compressed from \( P_i \) to \( P_b \) is
\( m_D + \delta m \). At point B, suction removes the excess mass \( \delta m \) from the
flow, moving the remaining fluid along the isobar to point C and
temperature \( T_c \). The mass \( m_D \) is then compressed the remainder
of the way to the final pressure \( P_f \) (point \( D' \)) while the suction
mass flow expands to point E. Equation 2.5 shows the work
required by the compressor with bleed.

\[
W_b = (m_D + \delta m) C_p T_A \left( \frac{T_B}{T_A} - 1 \right) + m_D C_p T_A \left( \frac{T_D'}{T_A} - \frac{T_C}{T_A} \right) + \delta m C_p T_A \left( \frac{T_E}{T_A} - \frac{T_C'}{T_A} \right)
\]  

(2.5)

Then the basis for comparison between the two compressors is the
work ratio of the bleed compressor referenced to the baseline case.
The compressors then are equal if the work ratio is 1.0, with
smaller work ratios indicating better performance with the bleed cycle.

This last equation is unwieldy in its current form. Carrying the derivation further, the temperature ratios become

$$\frac{T_B}{T_A} = \left(\frac{\pi_B}{\gamma \eta_D}\right)^{\frac{1}{\gamma - 1}}$$  \hspace{1cm} (2.6)

$$\frac{T_D}{T_A} = \frac{T_D T_C}{T_B T_A} = \frac{T_D T_B T_C}{T_C T_A T_B}$$  \hspace{1cm} (2.7)

$$\frac{T_D}{T_B} = \frac{T_C}{T_A} = \frac{T_B}{T_A} = \frac{T_B}{T_A}$$  \hspace{1cm} (2.8)

Three temperature ratios remain. These temperature ratios follow isobars. The standard definition of entropy given in Equation 2.2 then relates the missing temperature ratios to the entropy change along the isobars, for any two points, 1 and 2, as shown below:

$$\frac{T_2}{T_1} = e^{\left(\frac{S_2 - S_1}{C_p}\right)}$$  \hspace{1cm} (2.10)

For the bleed cycle depicted in Figure 2.3, the temperature ratios are as follows:

$$\frac{T_C}{T_B} = e^{\left(\frac{S_C - S_B}{C_p}\right)}$$  \hspace{1cm} (2.11)
\[
\frac{T_C}{T_B} = e^{\left(\frac{S_C - S_B}{C_p}\right)}
\] (2.12)

\[
\frac{T_F}{T_A} = e^{\left(\frac{S_B - S_A}{C_p}\right)} = e^{\left(\frac{S_C - S_B}{C_p} + \frac{S_B - S_A}{C_p}\right)} = e^{\left(\frac{S_C - S_B}{C_p}\right)}e^{\left(\frac{S_B - S_A}{C_p}\right)}
\] (2.13)

Substitution of all the temperature ratios results in the blade work equation for one suction pressure,

\[
W_b = m_D C_p T_A \left[ \frac{\gamma - 1}{\eta_D} e^{\left(\frac{S_C - S_B}{C_p}\right)} + \left(\frac{\gamma - 1}{\eta_D}\right) \frac{S_C - S_B}{C_p} - \left(\frac{\gamma - 1}{\eta_D}\right) e^{\left(\frac{S_B - S_A}{C_p}\right)} \right]
\]

\[
+ \delta m C_p T_A \left[ e^{\left(\frac{S_C - S_B}{C_p}\right)} e^{\left(\frac{S_B - S_A}{C_p}\right)} - e^{\left(\frac{S_C - S_B}{C_p}\right)} \left(\frac{\gamma - 1}{\eta_D}\right) \right]
\]

where common terms have been collected. For comparison, divide the bleed work by the baseline compression work, giving:

\[
\frac{W_b}{W_{nb}} = \left[ \left(\frac{\gamma - 1}{\eta_D}\right) \frac{S_C - S_B}{C_p} + \left(\frac{\gamma - 1}{\eta_D}\right) \left(1 - e^{\left(\frac{S_B - S_A}{C_p}\right)}\right) \right]
\]

\[
+ \delta m \left[ e^{\left(\frac{S_C - S_B}{C_p}\right)} e^{\left(\frac{S_B - S_A}{C_p}\right)} - e^{\left(\frac{S_C - S_B}{C_p}\right)} \left(\frac{\gamma - 1}{\eta_D}\right) \right]
\]

(2.15)
The term on the left is the compression of the main flow to the final pressure. The last term is the work required to compress the bleed fluid up to the compression point and the middle term represents work gained by an isentropic expansion of the high entropy fluid.

The equation contains two entropy differences which are related via the mass average entropy in the compressor. By definition, the flow at point $C'$ must have the same entropy as the viscous stream represented by $s_v$. Therefore, the entropy difference from point B to point $C'$ is simply the viscous entropy minus the entropy at point B, as shown in Equation 2.16.

$$\frac{(s_{C'}) - (s_B)}{C_p} = \frac{[s_v - s_B]}{C_p}$$

(2.16)

The entropy difference from B to C must be related to the entropy withdrawn as given in Equation 2.16. In fact, the entropy difference is inversely proportional to the withdrawn entropy as given by Equation 2.17.

$$\frac{s_{C'}}{C_p} - \frac{s_B}{C_p} = \frac{-\delta m}{m} \frac{(s_{C'} - s_B)}{C_p}$$

(2.17)

In order to evaluate the effects of suction on actual compressor performance, two additional assumptions are required. The difference between the average compressor entropy and the viscous entropy $(s_B - s_v)$ and the percentage of high entropy fluid
$\frac{m_v}{m}$ remain unknown. An upper bound can be placed on the difference of high entropy fluid by assuming that all of the entropy rise is contained in the viscous layer. The viscous entropy then becomes

$$s_v = \frac{m}{m_v} \left( \frac{\gamma - 1}{\gamma} \right) \left[ \frac{1}{\eta_D} - 1 \right] \ln \pi_s$$

(2.18)

where $\pi_s$ represents the pressure rise over only one blade row. A moderate guess for the viscous flow percentage now allows a rough calculation of the effects of fluid suction on the Brayton cycle compressor performance.

Making several assumptions about compressor performance, the effects of suction are plotted against the bleed pressure for a pressure ratio 30 compressor. For comparison, Figure 2.4a shows the effects of 1% fluid suction assuming that the entire flow has the same entropy $s_B$. In this case, the work ratio starts above one and only rises. Bleed without recovery is especially bad, as shown by the square points. The bled compressor has a stage pressure ratio of 1.6 with a viscous mass ratio of 5%. Figure 2.5a shows the work ratio for a one percent bleed. Like the control case, the work ratio always rises. In contrast, however, the work ratio remains below one for the case of ideal recovery, represented by the circle points. The trend continues as the suction mass flow increases until all of the viscous flow is removed. Efficiency, as plotted, is
determined by the polytropic efficiency divided by the work ratio. The trend in efficiency follows that of the work ratio. Efficiency always declines with increasing bleed pressure. In the case of five percent suction, the efficiency shows a gain of 1 to 2 per cent at low bleed pressures.

Suction during the compression process can remove some of the high entropy fluid, thereby increasing compressor efficiency. However there is a tradeoff to determine whether or not suction is viable. A comparison of two compressors operating under the same conditions shows that gains in efficiency occur at low bleed pressures. As the bleed pressure increases, the efficiency gain depends more and more upon the recovery of work from the suctioned fluid. The ability to recover work from the suctioned fluid will determine the bleed pressure above which fluid suction becomes inefficient. At low bleed pressures, the compressor efficiency increases with bleed as Kerrebrock suggested (ref. 10).
Chapter III
Tip Clearance Flows

3.1 Introduction

In nearly every experiment, the compressor tip region exhibits low efficiency. The existence of this high loss region draws considerable research interest. Cascade studies first identified several vortical flow patterns near the end walls. Work by Herzig et al (ref. 7) clearly photographed three different vortex structures in a blade passage. Following the cascade studies, Rains noticed that the driving pressure gradients in the tip region cut across the blade. He further observed the formation of a vortex in the tip region. Rains' research produced the fluid jet model of the tip region\(^1\). Storer and Cumpsty refine the model even more to develop an expression for the loss in the tip region based on several design parameters. These investigations, while hardly encompassing the wealth of the tip clearance literature, do paint a picture of the flow complexity in the tip region.

\(^1\)Rains fluid jet is only one of several models explaining tip clearance flow. Lakshminarayana is a major investigator into vortex shedding analysis. Comparable to a wing with lift, vorticity must be shed by the compressor airfoil at the tip. The amount of shed vorticity decreases towards zero as the tip gap vanishes. It is the shed vorticity then that accounts for the vortex formation.
3.2 Vortical Flows

An early study of cascade flows (ref. 7) showed the formation of three different vortices. First, in a cascade with no tip clearance or relative wall motion, a passage vortex formed in the suction surface corner. Secondary flows in the passage transported the high entropy boundary layer fluid from both blade surfaces and the endwall to the suction surface corner. The low momentum fluid from the suction surface formed the core of the vortex, while the pressure side boundary layer fluid became entrained in the outer folds (Figure 3.1). Next, with the addition of a clearance gap, a tip clearance vortex joined the passage vortex (Figure 3.2). The fluid jet emerging from the gap (or perhaps blade end unloading) formed into a vortex near the suction surface tip. The new vortex merely pushed the passage vortex further out into the flow field. Finally, simulating a moving endwall with a belt, the scraping vortex formed. The relative motion of the wall pulled fluid down the suction surface counteracting the effect of the secondary flows generating the passage vortex. This motion counteracts the natural tendency of the passage vortex to pull fluid up the suction side. As a result, the scraping vortex caused by the relative wall motion replaced the passage vortex (Figure 3.3). The relative motion in turbines has the opposite effect. Instead of canceling the passage vortex, the scraping vortex complements the passage vortex.
3.3 Fluid Jet Model

Rains, conducting experiments on a water rig, observed several flow phenomena relating to tip clearance in both non-rotating and rotating systems. The first question, and indeed the first assumption in any fluids problem is, is the flow inviscid? The ratio of tip gap to maximum blade thickness ($\lambda$) provides a distinction of the different flow regimes. A large ratio ($\lambda > 0.167$) indicates that the tip height is large compared to the blade thickness. Viscous forces do not have much time or area in which to act, and the flow is essentially a potential flow. On the other hand, if the ratio is small ($\lambda < 0.05$) the tip gap looks more like a channel flow and the flow is dominated by viscous effects.

Rains made several observations on the potential flow through the tip gap. First, the pressure difference is much greater normal to the chord than along the chord. The flow then travels across the blade tip in proportion to the driving pressure difference across the blade. It follows that as the blade loading increases, the velocity across the gap increases as well. Bernoulli's equation essentially determines the normal gap exit velocity (Equation 3.1).

$$V = \sqrt{\frac{2(P_0 - P)}{\rho}} \quad (3.1)$$

Second, the cross flow velocity is of the same order as the main flow velocity, a fact readily identifiable in Equation 3.2.
\[ \frac{V}{U} = \sqrt{\frac{2(P_0 - P)}{\rho U^2}} \]  

(3.2)

The potential flow field then is like the flow around a corner. Separation occurs after the corner and the stream tubes collapse to a fraction of the initial flow area. For ideal potential flow the exit area is 60% of the gap area. In reality, the exit streamtube is larger than the potential solution would predict. Bernoulli's equation only predicts the inviscid velocity profile.

Compressibility affects and the relative motion of the bladerow with respect to the wall also affect the velocity. For compressors, the relative motion of the wall complements the gap flow, while in turbines it retards the gap flow.\(^2\)

Additionally, Rains noticed the formation of a vortex issuing forth from the clearance region. The concentrated vortex formed from a vortex sheet shedding off the blade end, much as a wake trails from the wingspan of an airplane. The vortex is a localized flow structure and only impacts the local flow. The vortex is virtually hugging the case, so its image vortex is very near. The net effect cancels in the far field, in this case only a few tip clearances away. Interestingly, the vortex formed behind the quarter chord with the stationary wall, but moved forward to near the leading edge in the moving wall cases. The vortex core even changed sides of the blade when the flow angle of attack passed

\(^2\)Cascade experiments by Graham show that it is indeed possible to reverse the gap flow in turbines by increasing the rotational speed of the turbine, effectively increasing the relative motion.
through neutral. So Rains concluded that relative motion and viscous effects play a major role in vortex formation. Even though the vortex formation cannot be generalized, the method of formation did not alter the entropy increase associated with the vortical tip flow.

Storer (1991) investigates the tip clearance flow region, focusing his attention on the loss mechanisms and possible means to avoid the loss. As a starting point, his discussion cites a myriad of sources examining many aspects of tip clearance flow not even considered here. Storer continues with three-dimensional viscous calculations for a low speed compressor. Finally he compares the computational results with cascade data, showing that the computation captures the trends of the tip clearance flow. Two important results evolve from this work. The first is the detailed measurement of the flow in the tip region and the second is a model of the loss due to the tip clearance flow.

Experimental studies using a low speed cascade at several different tip clearances and blade loadings provided a plethora of flow measurements in the tip gap itself. In particular, the experiment measured the flow angle and loss factor as a function of distance in the tip gap at several axial locations. Measurements showed two separate flow regions. A strong cross gap flow associated with Rains' fluid jet model exists near the casing. At the minimum pressure point the cross flow reaches its maximum angle, greater than 60° with respect to the blade suction surface.
Near the blade itself, the flow follows the core flow in the streamwise direction. A shear layer separating the two flows generates virtually all of the loss in the tip gap.

In light of these results, Storer and Cumpsty (ref. 20) develop a mixing jet model to predict the tip clearance flow losses. The model assumes that the core flow and the cross gap flow fully mix. The core flow turns the cross gap flow towards the streamwise direction when the two flows mix. A simple momentum balance predicts a large total pressure deficit as a result of the mixing. The momentum balance shows that the loss between the two flows increases as the incidence angle varies from $0^\circ$ to $180^\circ$, the maximum loss angle. Rains' simple model is used to calculate the relative angle of incidence. The gap flow emerges with a velocity described by the vector sum of the streamwise velocity and the cross gap velocity. The model permits the calculation of the loss associated with the tip clearance flow.

The loss calculated by the model compares favorably with experimental results. The model overpredicts the loss associated with the tip clearance up until the minimum pressure point, at which point the model gains considerable accuracy. Storer and Cumpsty posit that the over prediction along the first portion of the blade stems from the fact that the streamwise pressure gradient is large. At the leading edge, the two pressure gradients have nearly equal values while Rains' model assumes that the cross gap pressure gradient is always much larger than the
streamwise pressure gradient. Even with the slight disagreement near the leading edge, the model produces several important pieces of information.

1) The tip clearance loss is proportional to the percentage of axial flow passing over the tip gap.
2) The loss is virtually fixed for a given loading and clearance gap.
3) Viscous modeling and mesh spacing play only minor roles in the loss determination.
4) The loss depends on the angle of the cross flow to the main flow, which can vary up to 60 degrees.
5) The loss is smaller than the loss associated with blade surface boundary layer growth as the loading increases.

The results of the model suggest that the loss due to the tip clearance flow might be smaller than the measured loss in the tip region. Also, the loss is virtually unavoidable, and alternate aerodynamic designs are not likely to increase the overall compressor efficiency.
3.4 Tip Design

Bindon and Morphis studied the effects of different tip shapes on the blade tip losses. In particular, attention focused on the shear layer that contains the majority of the lossy fluid. Using a smoothly contoured tip shape, Bindon and Morphis hoped to reduce the tip loss by prevention of the pressure corner separation and elimination of the shear layer. In the attempt they conducted tests using several modified turbine cascade blades. One blade with a standard square edged contour provided the control. The second blade had a smooth finite radius contour leading into the tip gap. The contour prevented separation as the flow entered the clearance gap. This blade nevertheless performed worse than the unchanged configuration. The lack of a separated region significantly reduced the loss in the gap itself. However the total loss increased due to a larger mixing loss in the neighboring blade passage. The tip of the third blade looked more like a set of front porch steps. It had a smooth ninety degree turn into the blade, then another smooth ninety to straighten back out, at which point it ended in a sharp edged tip. This contour also reduced the loss in the gap in comparison with the regular blade shape. Unlike the rounded contour blade, the passage mixing loss for the step contoured blade did not offset the reduction of the gap loss. The step contoured blade performed the best while the smoothly contoured blade produced the worst results.
The results of Bindon and Morphis tend to confirm the work of Storer and Cumpsty. Changing the blade shape does little to change the tip mass flow. Consequently there should be little change with the new blades. The model even predicts that the smoothly contoured blade should exhibit more loss. The lack of a separated region allows for increased mass flow through the blade, hence increasing the loss. The experimental results appear to confirm this. As both theory and experiment show, the tip clearance loss strongly correlates with the cross gap mass flow.
3.5 Crook loss box

Using a steady three-dimensional viscous code, Crook studied the tip clearance flow for a stator to aid casing treatment research. Crook's analysis also complements the boundary layer research performed by Gertzig et al. Crook traced fluid particles released near the tip gap. Particles released just under the gap from the 10% chord up to the 40% chord location formed the core of the ensuing vortex. These particles traced their way to the low total pressure region, which Crook traced out as a box. Particles released between the 50% and 80% chord locations followed helical paths that encircled but remained outside of Crook's loss box. Particles emerging from the last 20% of the clearance gap scooted all the way across the blade passage, eventually being driven radially closer to the hub by the vortical flows.
3.6 Stalling Characteristics

Although stall is not the topic of this report, several studies conducted at MIT aimed at increasing stall margin contain relevant information for tip suction research. Lee and Greitzer study the effects of casing treatment on the endwall flow field. Casing suction and fluid injection model casing treatment grooves. Results from this experiment can help predict the effects of blade tip suction. In conjunction with NASA Lewis, Greitzer examines the growth of the clearance vortex as a rotor approaches stall. The tip clearance vortex increases, suggesting that the vortex might induce the stall through flow blockage. Taken together, these two experiments loosely tie blade tip suction with efforts to increase compressor stall margin.

Lee and Greitzer study casing treatment grooves as an effective means to improve stall margin. On some compressors, casing treatment grooves increase stall margin. Research determined that the casing treatment grooves suction fluid along the latter half of the blade chord, and then eject the fluid back into the flow path along the front portion of the blade. To separately determine the effects of both the suction and the injection, Lee and Greitzer tested a low speed compressor with variable fluid suction and injection from the casing. The experiment determined that both the suction and the injection increase the stalling pressure rise. Injection in the front middle of the blade provided the best improvement in stall margin.
Crook, as mentioned above, performed a calculation of the tip region in support of this casing treatment investigation. The computational results showed that the fluid injection altered the flow field in the tip region. The extra momentum pushed the low pressure, high entropy fluid away from the wall, such that less of the fluid entered the loss box. The increase in stalling pressure rise then correlated with a redirection of the high loss fluid from the vortex core.

Adamczyk et al (1992) examine the compressor flowfield as the rotor approaches stall. The study viewed several cases from no clearance to a large clearance. The larger clearance led as expected to a larger vortex and a lower stall margin. Another case proved interesting. The new blade has no clearance gap along the front half of the blade with a nominal clearance along the aft half. This case attained significantly better results, suggesting that the fluid passing over the front of the blade has the largest affect on stall margin. Indeed it is the fluid from this region that Crook traced into the center of the loss box.
3.7 Result Summary

Past research into the tip clearance flow yields several worthy items. First, the tip clearance flow can be modeled as a fluid jet. This model predicts the mass flow crossing the tip gap and the angle at which the flow exists the clearance gap. Second, the loss associated with the flow is essentially inviscid, so coarse, poor modeling of the tip region should produce reasonable results. Third, the loss depends strongly on the cross gap mass flow, and most attempts to reduce the loss produce ambiguous results at best. Fourth, the mass flow passing over the front half of the blade introduces the largest flow disturbance. This flow then is the most important to attenuate. Keeping the lessons in mind, it remains to assess the effectiveness of tip suction at reducing the loss in the tip region.
Chapter IV
Tip Suction Theory

4.1 Introduction

Limiting the loss in the compressor tip region is a concern for all design engineers. As shown above, the compressor tip loss is mostly a function of the tip clearance mass flow. The traditional means of reducing tip loss then has been to decrease the tip mass flow by a reduction of the gap height. However the threat of tip rubs limits the effectiveness of this option. Other attempts to reduce the tip loss include casing suction and tip slots. Neither method proved particularly successful. Tip suction is proposed as a new approach to control the tip clearance flow.

Blade tip suction may influence the tip clearance flow in several ways. The objective is to remove the high entropy fluid from the tip region. This would raise efficiency immediately as the worst fluid is no longer present. Removal of the high entropy fluid also increases the thermodynamic efficiency for later compressor stages. Second, tip suction could reduce the mass flow exiting the tip region. In this fashion then, tip suction acts as an extension of the blade tip, effectively creating a smaller tip gap than actually exists. Additionally, tip suction ought to influence the angle of the cross flow, thus reducing the mixing loss determined by Storer and Cumpsty. This last thought is a possibility not investigated here. Through the mechanisms discussed above, blade tip suction has the potential to reduce the tip loss.
4.2 Important Measurements

Both computational and experimental work has been done to assess the effect of tip suction on compressor performance. Computations using a three-dimensional steady viscous solver offer the best place to start since the computer modeling permits the study of several different suction configurations. The large array of data for each case enables detailed resolution and analysis of the compressor flow field. The disadvantage with the computations is the necessity to make several assumptions. Measurements on a compressor provide data with which to compare the computational results, but experiments are expensive and do not allow the detailed investigation of the flow field that the computations produce. The combined sources of data paint a better picture and establish more certainty in the final results.

Several fluid quantities significantly aid the analysis of blade tip suction. The primary goal is an increase of efficiency, which is measured directly. The entropy distribution is another important parameter, though it is virtually the same as the efficiency. As previously mentioned, efficiency scales with the tip mass flow, the third measure of interest. Finally, velocity vectors indicate the effects of suction. In particular, velocity vectors show the relative flow angles so important in the mixing calculations. The effectiveness of the blade tip suction is captured in these four parameters, considered now in more detail.

Only the throughflow efficiency measures the relative effects on the flow field cause by the blade tip suction. In the
thermodynamics section, efficiency is calculated by means of a rotor work balance. This 'cycle' efficiency is the ratio of useful work measured by delivery pressure, to the shaft work input. However the cycle efficiency depends strongly on the disposition of the withdrawn fluid. It follows then that the cycle efficiency, while being the ultimate measure of success or failure of blade tip suction, does not reflect the efficiency of the main or through flow. Fortunately the 'compressor' efficiency calculated based on the mass averaged entropy difference and total temperature ratio as shown below

\[
\eta = \frac{\Delta_s}{T_D E_{C_r} - 1} \quad (4.1)
\]

measures the efficiency in the flow exiting the blade row. The compressor efficiency is a scalar measuring the change from the inlet to the exit. Efficiency is also a function of the radial distance. The compressor efficiency then is a useful yardstick to compare the effects of tip suction at this stage of the research.

Entropy possesses two important properties that differentiate it from the efficiency. One, entropy is a fluid property, taking on the same value regardless of the co-ordinate system. This is true because entropy is a property of state, dependent only on the local temperature and pressure. Two, entropy is a mass intensive property. The specific entropy remains unaffected by the removal of the suction fluid. Entropy contours graphically show the loss in axial rotor planes. Furthermore, the center of the clearance vortex is a high entropy region, easily seen on the contour plots.
Velocity vectors catch important features of the tip suction flow field. Velocity vectors in the tip gap serve to indicate the direction of the mass flow passing through the tip. A closer examination of the tip also shows the fluid jet's interactions with the immediate flow near the blade surface. Radial planes also show the interaction of the fluid jet and core flow. Passage view planes, axial planes viewed from the flow stagger angle, show the relative swirling of the vortical flow. The velocity vectors help depict what actually happens in a compressor with tip suction.

The effects of tip suction on the clearance mass flow are hard to deduce due to the complex nature of the tip flow field. Fluid jet models usually characterize the tip flow as two-dimensional potential flow. Theory then predicts a gap discharge coefficient of 0.6. Experiments on the other hand, report gap discharge coefficients up to 0.93. Indeed for a choked gaseous flow, discharge coefficients can approach 0.99. To further complicate matters, transonic compressors pull the casing boundary layer past the tip at sonic speeds, while the same no slip condition predicts no velocity directly on the blade tip. Thus a sonic line must exist somewhere in the clearance gap.

In spite of the uncertainty involved, several observations can at least suggest the effect of tip suction on the flow exiting the tip region. Without suction, flow entering the gap separates from the pressure corner with the main core flow hugging the casing wall. As the flow accelerates towards the suction side, the streamtube area contracts. The flow immediately above the blade does not follow the same path. Instead it follows the blade more closely. As suction is applied, first the undetermined flow near the blade tip is removed.
Now more flow area is open for the fluid jet. The core flow cannot just expand to fill the void, as it would never be able to match pressure at the exit plane. Consequently, more fluid is drawn into the gap from the pressure side to fill the void. As this occurs, the gap inlet velocity increases to maintain continuity. Up to this point, none of the core flow is being withdrawn, but more enters the gap. It follows then that with small amounts of suction, the gap exit flow increases in place of the desired reduction. From this point, more suction will decrease the flow exiting the gap, assuming that the suction port is downstream of the gap minimum area. The minimum area chokes, maintaining a constant entering mass flow. Now the increased suction decreases the flow exiting the gap. Taken to an extreme, large suction will remove flow from both sides of the blade.

Realizing that tip mass flow is significantly effected by flow area, fluid injection from the blade tip just might reduce the tip flow as well. Fluid injection spanwise from the blade tips creates a blockage in the clearance gap. This blockage reduces the available area for the core gap flow. Granted that the injected fluid leaves via the tip gap, the injected fluid still crowds out more core flow than it replaces. Tip injection reduces the clearance mass flow, but ignores several other factors.

Fluid injection, while decreasing the tip mass flow, is unlikely to help the overall compressor efficiency. Whereas blade suction removes the inefficient high entropy fluid from the flow path, thereby avoiding the thermodynamic inefficiency, fluid injection does not. While tip suction might eliminate the mixing loss, fluid
injection provides another surface shear layer at the boundary of the cross flow jet and the injection jet.

The discussion highlights several important measures of mass flow. First, the goal is to increase efficiency through mass suction. Obviously then, the suction mass flow is necessary to evaluate the effectiveness of fluid suction. Since non-dimensional numbers carry more meaning, the suction mass flow is measured as the percentage of the inlet flow. The other important mass measure is the mass flow exiting the tip gap, while the amount of flow passing over the camber line adds insight into the flow variations in the tip gap.
4.3 Computational Analysis

The recent advances in computational fluid dynamics (CFD) have produced several reliable solvers. The choice of code then depended on other factors. Many projects at MIT use the multi-stage turbomachinery solver developed at NASA by Adamczyk et al (1990), while Mulac explains the accompanying mesh generator program. At MIT, Crook also provides a good description of the NASA code as part of his work. Given the level of familiarity with the NASA code, it was chosen for the tip suction investigation.

Boundary conditions can severely affect the accuracy of the final results. The NASA code follows a center difference scheme, requiring a virtual or dummy cell at each edge. The flow variables must be specified in the dummy cells in order for the calculation to proceed. At the inlet, the total temperature, pressure and inlet flow angle must be specified. At the exit plane, radial equilibrium determines the flow values of the dummy cells, given the hub pressure. On the blade surfaces, the no slip, no flux conditions are specified. For blade tip suction, the no flux condition is modified, allowing for the specification of the suction mass flux.

Creating a useful computational mesh also induced several deviations from the standard NASA code. The customary mesh routine reported in Mulac provided the first grid iteration, but does not detail the tip region. For the blade tip suction calculations, the base grid was 61 axial gridpoints by 31 gridpoints in both the radial and circumferential directions. Figure 4.1 and Figure 4.2 show two blade passages plotted side by side. Of the 61 axial grid points, 31
points span the blade. A mesh post-processing program written by Crook and modified by Khalid provided greater detail in the tip region. Crook developed a method to bend the existing grid lines, forming a blade tip. The gridlines running up the side of the blade are pinched together to form a peaked tip at the blade camberline. From the peaked tip, these bounding gridlines continue to the casing. Several gridpoints can be placed in the gap and along the skewed tip. In the modification of the blade suction grid, three flow cells spanned the entire tip gap, while four more covered each side of the slanted blade tip as shown in Figure 4.3. The resulting tip clearance was 1.5% of the span, a nominal clearance for modern fans.

The large skewing of the grid in the tip region does not appear to invalidate the computational results. Crook explored several possibilities to determine the effects of non-orthogonality and grid skew in the tip region. He compared the computational results from several different grid modifications. He expected that large numerical errors due to grid skew and non-orthogonal grid lines would appear as obvious differences between the different flow fields. As the results appear essentially the same, the grid skewing is apparently a non-factor.

A coarse grid seemed suitable to capture the large flow changes due to tip suction. Coarse grids are very capable at capturing large potential effects. Tip flow is essentially potential flow governed by pressure gradients. Several reports also cite the ability of the coarse grid to properly capture tip flow features, including Adamczyk et al (1992), Crook and Storer. However, Khalid did noticed a failure of the code to accurately portray the flow in the tip gap itself, even
using a much finer mesh. The coarse mesh used in the calculations should reveal the important trends of tip suction, since large scale trends do not depend on the fine details overlooked by the coarse mesh.

Having created the grid, it remained to establish the initial conditions and the blade suction profile. Several parameters can be altered with the code. In the interest of evaluating tip suction, the flow parameter of greatest interest was the magnitude of the suction flow. Other input parameters were not changed throughout the experiment.

In spite of the consistent inputs, each solution had a different inlet mass flow while the exit mass flow hardly changed. The exit plane dummy cell values are calculated assuming radial equilibrium based on the specified hub pressure. As this value did not change, the rotor faced the same exit conditions for each solution. Changes due the removal or introduction of mass at the blade tip appeared almost entirely as changes in the inlet mass flow. For a sufficiently large fluid injection, the inlet mass flow decreased to the point that the rotor apparently stalled. Two important observations stem from this result. One, the blade suction introduces confusion into the definition of rotor operating point. Two, fluid injection apparently incited stall.

Unlike the focus of the experiment on high speed compressors, the calculations use a low speed fan. The fan has 54 blades with a hub/tip ratio of 0.84. The nominal fan speed was 3,445 rpm, with a base mass flow of 38.35 kg/s. The nominal tip clearance flow was 1.8% of the axial flow. The hub exit pressure was set to 101,426 Pa.
The computed fan efficiency was 94.25% with these operating conditions.

The chordwise suction profile resembles that of the experimental blades in most aspects. The experimental blades, described in section 5.5, have suction slots downstream of the gap minimum area, so suction slots were placed on the suction side of the camber line. Flow cells along half the chord were marked as suction cells, again mirroring the experimental blades. The experimental configuration used two-thirds of the blade thickness while only two of eight thickness cells were used for the computations, with one exception. To achieve the large suction of the last data point (0.90%), it was necessary to increase the number of thickness suction cells to three. In a departure from the experimental blades, the magnitude of the suction increased closer to mid-chord as shown in Figure 4.4. The experimental blades have a nearly constant suction profile design due to the nearly constant pressure gradient up until the shock impingement about the 2/3 chord location. In contrast, the low speed fan blades exhibit the traditional subsonic profile of a larger cross gap pressure gradient near mid-chord. The suction profile for the low speed fan followed that of the experimental blade designs as much as possible.

The computational results show several trends and a few surprises. Fluid suction at first increases the compressor efficiency, as shown in Figure 4.5. Unfortunately, the efficiency reaches a maximum quickly and begins to drop off. Apparently, the initial fluid removed comes from the high entropy low total pressure fluid hugging the blade tip. The removal of the high entropy fluid lowers
the mass-average entropy of the remaining flow as predicted. As the suction grows above a certain percentage, the suctioned fluid becomes that of the lower entropy fluid jet. At this point, the mass-average entropy increases, as shown in Figure 4.6. Thus the efficiency and entropy curves are much as expected.

Suction increases the mass flow entering the tip region, but only moderately changes the mass flow exiting into the flow field. The mass flux entering the suction side control volume shown in Figure 4.7, increases in direct proportion to the suction ratio. The mass flow exiting the tip gap increases at first, but then steadies (Figure 4.8). As suction increases, the mass flow exiting the gap region even decreases as predicted. At a suction ratio of 0.5%, the gap exit mass flow returns to its initial value. The suction mass flow offers few surprises.

The 0.90% suction data point raises a few questions. It appears to follows the mass flow trend quite well (Figure 4.11), but has a much larger efficiency than trend would suggest (Figure 4.9). This trend is potentially due to the extra suction ports. Several random solutions not presented captured the same trends as shown here, though efficiency values varied from case to case. On the other hand, the data point could be the harbinger of an upward turn in the efficiency curve. Higher suction levels could increased efficiency due to a lower mixing loss, as less mass flow actually passes across the blade tip at this point. The 0.90% suction removed only half of the initial gap flow, so there exists much room for improvement.

Another small series of cases studied pressure side suction using the same profile. Only two cases were run to evaluate the
difference between suction on different sides of the blade camber line. Pressure side suction followed the same trends, but inexplicably performed better than suction side suction over the evaluated range. The question remains whether or not this trend continues with higher suction levels.

The efficiency of the tip region shows the same trends as does the overall efficiency. The radial efficiency changes significantly in the tip region from case to case, while hardly any change is noticed over 80% of the blade span. This indicates that the trends noticed in the overall efficiency do occur in the tip region as expected for tip suction.

A series of radial efficiency plots for the top 20% span show the same trends noticed earlier. Figure 4.12 starts the series for the case of 0.20% fluid injection. The efficiency increases from this point. Figure 4.13 shows the nominal no-suction case while Figure 4.14 shows the 0.20% fluid suction case for comparison. Figure 4.15 depicts the lower efficiency of the 0.90% suction case. Radial efficiency exhibits the same trends as the overall efficiency, indicating that the efficiency change is associated with tip suction.

Mass averaged entropy contours show complementary results. Figures 4.16 to 4.19 show the entropy caught up in the vortex near the top of the blade passage at mid-chord. The same entropy contours are used for all graphs. The number of contours associated with the tip clearance vortex, and hence the entropy, decreases as the suction mass flow increases. In fact with 0.90% suction, the center of low entropy almost vanished. Figures 4.20 to 4.23 show the same trends just after the trailing edge. Unfortunately, the
vortex can be clearly distinguished in all cases. Contours show a
decrease in entropy due to blade tip suction, but suggest that the
vortex exists downstream of the blade row in all cases.

Viewing velocity vectors in axial cuts through the blade show
the flow direction through the tip gap. With fluid injection, the cross
gap flow is pushed away from the blade tip. Figure 4.24 shows that
the flow exits with an outward radial component. As suction
increases (Figures 4.25 to 4.27), the emerging fluid jet is pulled
closer to the blade metal. For the case of 0.90% suction, the flow
shows a large radial inflow that alters the local flow pattern in the
blade passage itself. Blade suction does change the local tip flow
field.

Thus the computations predict that blade tip suction provides
some beneficial results. First, small amounts of suction do increase
the efficiency. Second, larger amounts of suction are capable of
decreasing the cross gap mass flow. Third, as suction increases, the
magnitude of the entropy contained by the tip clearance vortex
decreases. Software written by Khalid to analyze the flow field even
shows that the blade tip vorticity downstream of the rotor blade row
decreases with suction. Since the software program is still in the
design phase, this data is not presented here. Computational results
imply that blade tip suction does work as Kerrebrock suggests.
Chapter V  
Experimental Facilities

5.1 Blowdown Compressor Theory

The blade suction tests utilize the Blowdown Compressor (BDC) Facility. The BDC is a transient transonic compressor test bed. Recognizing that the time scale for the development of compressor flow structures is smaller than the rotational period of the rotor, it is possible to measure the variations of compressor flow on a passage by passage basis. These flow structures emerge so quickly that a transient flow of fluid across a spinning rotor provides very reliable compressor characteristics, requiring few full revolutions for accurate resolution. With proper adjustment of the rotor's inertia, it is also possible to maintain a constant test section Mach number by choking the flow. A throttle plate downstream of the rotor serves this purpose.

Blowdown facilities emerged as an inexpensive alternative for testing rotating machinery at realistic conditions. Due to the transient test nature, the total air mass per run is small. The BDC, for example, uses a 100 ft³ gas supply at 500 mm Hg pressure and room temperature for the MIT rotor. This low mass requirement enables the use of pricier, more exotic gases. A Freon/argon
mixture, the working fluid of the BDC, has the same specific heat ratio as air but at a lower speed of sound. The reduced speed of sound permits transonic testing at lower rotational speeds, simplifying the rotor drive system. Additionally, because the rotor accelerates in vacuum, a small, low power electric motor performs quite well, avoiding any complexities associated with steam turbine drives. The Freon/argon also provides realistic Reynolds numbers.

The BDC is uniquely suited for the blade suction experiments for several reasons. Blade stresses are relatively low for a transonic compressor. The lower blade stresses in turn permit the testing of such novel concepts as blade suction slots. The facility is fairly simple. Installation of a fluid suction system proved to be easy, while removal of the system only takes a few minutes. The detailed resolution of each blade passage permits modification of only a few blades. For a more detailed description of the blowdown compressor see Kerrebrock, 1974. In order to examine the effects of blade tip suction on the flow in a transonic compressor, the BDC is a good facility.
5.2 Blowdown Compressor Facility

A general description of the BDC helps illuminate the use of the various components. For reference, Figure 5.1 is a schematic of the facility. At the beginning of the test, the supply tank contains the working fluid. The supply tank is a large tank that stores the working fluid and acts as a pressure plenum. A dead soft aluminum diaphragm separates the supply tank from the rest of the system until test time. To begin the test, strips of plastic explosive carefully laid on the diaphragm in an asterisk shape are remotely detonated. The pressure quickly pushes the resulting aluminum petals flush against the passage wall as the flow floods into the test section. The test section houses the rotor and instrumentation. After passing the throttle plate, the fluid enters the dump tank. The dump tank is just a large reservoir to hold the fluid. Since the test ends when the dump tank pressure equals roughly half the rotor outlet pressure, a larger dump tank increases the available test time for a given supply tank pressure and operating point. The BDC dump tank is 400 ft³, unchoking after 130 milliseconds when operating with the MIT rotor. For the blade suction experiments, a new isolated dump tank is added. The suction dump tank connects to the facility via three copper pipes. The suction dump tank’s sole purpose is to provide a reservoir for the suctioned fluid. It’s sizeable 11.4 ft³ volume
shows almost no pressure rise with the suction from just four blades.

The test section has many noteworthy features. In front of the rotor there is a boundary layer bleed which removes the wall viscous layer. The bleed redirects about 10% of the supply tank flow into the dump tank through two bypass channels. Several access ports, or 'windows', on the sides of the test section provide instrumentation access to the flow field. The rotor mounts onto a rotating assembly housed in a centerbody cantilevered near the back of the test section. A small DC motor operating at room pressure accelerates the rotor in a vacuum. A shaft seal prohibits pressure leakage into the test section. Another cantilevered centerbody, the spider, sits ahead of the rotor. The spider interfaces between the suction dump tank and the suction passages on the rotor. The spherical front end joins with a long cylindrical body. At the end of the spider body rests a shroud that directs the flow from the spider diameter to the hub diameter. Near the front end, one inch diameter passages inside the three spider supports access the suction dump tank. Figure 5.2 displays the spider design.
5.3 Instrumentation

The primary instrument in use on the BDC is the pressure transducer. Different requirements on the transducers dictate the use of several differing models. First, measurements of the flow structures as they emerge behind the blade row require high frequency response transducers. Four extremely small Kulite transducers mounted in a four way probe described in the next paragraph meet the stringent response criteria but tend to have zero drifts time. In addition, three wall static probes also have high frequency response requirements. Again, small diameter Kulite pressure transducers are used. These probes can access any of the seven ports in the instrumentation window. The systems level guages sample the fluid state in the tanks and provide a steady reference. These probes are larger, steady guages as the frequency requirement is very low. Larger, more reliable Kulites are located in the supply tank, the suction dump tank, the dump tank, and just after the throttle plate to record the general system pressures.

A four way probe composed of four high response frequency Kulite pressure transducers is the backbone of the measurement system. The four pressure measurements, one total and three static, determine the flow angle and Mach number uniquely. This procedure is more fully reported in Thompkins (1976).
Additionally, the total temperature ratio is inferred from the data using Euler's equation as shown below;

$$
\tau_D = 1 + \frac{r}{r_t} \frac{(\gamma - 1) M_T M_{\theta_2} \sqrt{\tau_D}}{\sqrt{1 + \frac{(\gamma - 1)}{2} M_1^2}} \sqrt{1 + \frac{(\gamma - 1)}{2} M_2^2}
$$

Experiments using the aspirating probe, which measures both total pressure and total temperature at the same time, validate the Euler equation approach. Furthermore, a linear translator moves the probe radially. As the rotor spins, this one probe practically measures the entire flow field behind the rotor. A linear potentiometer attached to the probe outputs radial position.

Several other instruments complement the pressure transducers. Thermocouples in the supply tank and the suction dump tank read the initial temperatures. The supply tank temperature changes isentropically during the blowdown as the flow expands out of the supply tank. The supply tank temperature is used in conjunction with the supply tank pressure to calculate the test section mass flow as shown below

$$\dot{m} = \frac{V_1}{(1 + \delta) a_{i(0)}^2} \left( \frac{T_i(0)}{T_1} \right) \frac{dp_1}{dt}$$

(5.2)

where $\delta$ is the percentage of bleed flow, here taken to be 0.1. In a similar fashion, the suction mass flow is calculated as shown

$$\dot{m} = \frac{V_5}{a_{s(0)}^2} \frac{dp_5}{dt}$$

(5.3)
based on the suction dump tank pressure.

Finally a whole suite of instrumentation provides rotor location, speed and deceleration. Just as the linear position indicator sets the four way probe location, a modular quadrature shaft encoder (400 count) with index provides rotor angular position. Rate and acceleration can be determined through differentiation. The index aligns with the trailing edge of blade four, the third suction blade. Its trace bounds consecutive rotor periods. The index only corresponds to blade four for probes located at ports 4 and 5. At the other locations, the index leads blade four. Rotor deceleration measures the work of the rotor on the fluid, and provides one measure of rotor efficiency. Past attempts to measure efficiency this way proved inaccurate. A toothed wheel (115 teeth per revolution) and a magnetic pickup provide redundant shaft information.

Before being recorded, the pressure signals pass through tailored amplifiers. The systems level transducers use an Analog Devices operational amplifier. This amplifier has a gain of 50 up to 500 Hz, at which point the gains heads towards zero. At 10,000 Hz the gain is reduced to 15. The high frequency signals require a better response than this. They use Burr-Brown amplifiers. The gain on the Burr-Browns is adjustable. For this experiment a gain of 80 is used. The signals go directly from the amplifiers to the A/D system.
A lab PC equipped with high speed A/D cards records and processes the data. A Dell 486/50Mhz computer with 32 megabytes of extended memory provides the hardware. Two ADTEK 830 madcap A/D cards provide 8 data acquisition channels each. The cards can sample at a maximum frequency of 330 kilohertz. Unfortunately, all channels must sample at the same rate, so even the low frequency systems guages are sampled 330,000 times per second, the rate required to resolve the 400 count index. The cards read voltages from minus ten to plus ten, and can amplify the signal up to a gain of eight. The A/D is externally triggered by the starting pulse, but uses internal clocking. After sampling, the data is processed using Matlab.
5.4 MIT Rotor Description

The basic rotor is the MIT Blowdown Compressor rotor, with several modifications for the tip suction experiments. The rotor has 23 blades, of which five consecutive blades are modified for tip suction. The five blades now have seven flow passages running spanwise. A covering composed of two layers of fiberglass epoxy cloth is bonded over the grooves on the pressure surface. This fiberglass epoxy covers all blades in an effort to maintain uniformity. The covering adds 20 to 30 mils of thickness to the blade, except at the blade edges, where sanding removes the fiberglass. The covering extends beyond the blade tips, allowing for a static tip clearance of 45 mils at the tightest point. Red paint on the blade tips reveals that there is no tip rubbing during operation. Relevant data on blade shape can be found in Table 5.1. No additional changes are made to the blades, which are described in detail in Kerrebrock (1974).

Several pieces of the suction flow system replace the spinner on the rotor face. First, large 7/16 in diameter pipes depicted in Figure 5.3 collect the flow from the blade passages. Holes in the pipes, making them look like strange flutes, open into the blade passages. Pipes with different hole sizes can be inserted to change the suction profile, although this option has yet to be exercised. The pipes open into a cavity made by the addition of the rotor cover and rotor plate (see Figure 5.2). In turn, a cylindrical cup
resting on the shaft guides the flow into the spider. A labyrinth seal isolates the suction flow from the main flow at this boundary of the rotating system and the stationary spider.
5.5 Blade Modification

The blade suction passage design presented several unique problems from aerodynamics to structures and manufacturing. The primary goal of the suction passages is to remove some portion of the clearance mass flow. The pressure drop down the passage drives the flow. Size constraints imposed by blade thickness and structural integrity limit the allowable passage size. While mass flow increases and the pressure drop decreases with increasing passage area, the structural integrity of the blade decreases. Hence like many engineering problems, a trade-off existed between increasing the passage size for better flow conditions and retaining enough blade material for structural stability. One last constraint guiding the design was manufacturability.

An overview describes the general idea for seven or eight suction flow passages in the blades. As a first guess, approximately half of the tip area should be open to the suction passages. The flow passages should be long and slender for an even suction distribution along the tip. In theory, it would be nice to drill the suction slots from the blade ends, leaving the blade surfaces intact. Thin, twisting blades made this impossible. Instead, grooves were milled from the pressure side, with each groove of constant width for easier machining. A covering was then bonded over the grooves as a replacement pressure surface.
Calculating the desired mass flow provided the first challenge. The goal was to suction off an amount of fluid equivalent to the tip mass flow with the hope that most of this fluid comes from the cross flow. Rains’ fluid jet model described the blade crossing tip flow as a two dimensional potential flow. Bernoulli’s equation estimated the flow velocity exiting the gap, while continuity then determined the mass flow through the gap. An isentropic relationship related the pressure and density. Now three equations described eight variables, the tip inlet and exit streamtube area, velocity, pressure, and density. Measurements or assumptions provided the five missing variables. Epstein measured the pressure and suction surface densities for the MIT rotor. Haymann-Haber computationally determined the same density distribution, while Thompkins produced the blade pressure distribution. Measurements of the tip gap and chord provided enough information to calculate the inlet streamtube area. The inlet flow velocity is small and assumed to be zero. As the flow accelerates through the gap, the streamtube area contracts. Potential flow theory predicts an exit to inlet area ratio of 0.6, while measurements on low speed machines show higher values. A streamtube area ratio of 0.7 was used.

Instead of directly calculating the mass flow, the above procedure was incorporated into the pressure loss calculations. In the meantime, continuity provided a good approximation of the tip mass flow. First remember that the axial and tip velocities are of
the same order. Then the ratio of tip clearance flow to axial flow is proportional to the area ratio (equation 5.4).

$$\frac{m_g}{m_{\text{axial}}} \approx \frac{A_g}{A_{\text{axial}}} \quad (5.4)$$

For the MIT rotor, the tip radius is 12 inches, with a span of 6 inches. The chord is 3 inches while the new gap height is 45 mils. Approximately 0.039% of the compressor mass flow passes across each tip gap. The overall flow for 23 blades then is about 0.92% of the compressor mass flow, while the mass flow suction for only five blades is 0.16% of the compressor flow. As an initial estimate then, the desired suction flow was a small fraction of the entire compressor mass flow given by the ratio of gap area to axial flow area.

Knowing the mass flow, the passage dimensions that minimize the passage pressure loss were found. Two sources were responsible for the pressure loss: friction and the adverse pressure gradient due to rotation. Friction scales as the passage velocity squared, but also depends on the passage geometry. Miller presents the relation shown in equation 5.5.

$$\frac{dp}{dr} = \frac{fpU^2}{D} \quad (5.5)$$

Here D represents the hydraulic diameter, or the equivalent diameter a square or trapezoidal hole would have if it were round. The hydraulic diameter is defined as four times the passage area divided by the passage perimeter. The quantity f represents a
coefficient of friction as shown in equation 5.6, where \( k \) is an empirically determined value depending on passage smoothness and material. For the aluminum compressor blades, 0.025 mm appeared to be a reasonable value. A Reynolds number of 10,000 was used. The radial pressure gradient scales as the circumferential velocity squared. Since the fluid is constrained in the flow passages, the circumferential velocity is a measure of the tip speed and the radial location. Equation 5.7 shows the pressure gradient as a function of radius.

\[
f = \frac{0.25}{\left[ \log \left( \frac{k}{3.7 D} + \frac{5.74}{Re^{0.9}} \right) \right]^2}
\]

(5.6)

Now that the equations are assembled, it remains to combine them in a useful format. Since blowdown data is very time dependant, only non-dimensional numbers hold any real physical meaning. Flow quantities were non-dimensionalized to the supply tank conditions, which also change with time. The resulting non-dimensional quantities do not vary during the actual test period. Non-dimenisonalizing the compressible Bernoulli's equation as found in Kuethe and Chow gave the first useful relation, Equation 5.8 which related the gap exit Mach number to the density ratios
measured by Epstein. The non-dimensional equation for the pressure loss is given in equation 5.9. Integration of equation 5.9

\[
\frac{dP_P}{dP_0} = \gamma \left( \frac{\rho_P}{\rho_0} \right) M_T^2 \frac{r dr}{r_T^2} + \frac{\gamma f}{2 D} M_p^2 \left( \frac{\rho_P}{\rho_0} \right) dr
\]

resulted in the pressure loss in the passage from the hub to the tip. Integrating the last term was not so simple, as f, D and Mp all depend on the local passage geometry as a function of the radius. Assuming that the geometry changes little implies that these values are nearly constant. The result with the friction loss independent of radius became

\[
\left( \frac{P_H}{P_0} \right)^{\gamma - 1} = \left( \frac{\rho_T}{\rho_0} \right)^{\gamma - 1} - \left( \frac{\gamma - 1}{2} \right) M_T^2 \left[ 1 - \frac{r_H}{r_T} \right]^2 - \frac{\gamma - 1}{2 D} M_p^2 (r_T - r_H)
\]

Continuity provided the last relationship between the suction mass flow and the passage pressure loss. The gap mass flow non-dimensionalized by the passage mass flow is just

\[
\frac{m_g}{m_p} = \left( \frac{\rho_2}{\rho_0} \right) \left( \frac{M_g}{M_p} \right) k A_{PS} \frac{k A_{PS}}{A_p}
\]

where k represents the streamtube contraction. Assuming that the desired suction mass flow is equivalent to the clearance gap mass flow, the equation can be rearranged to solve for the blade passage area as a function of the gap mass flow.
Equation 5.12 relates the passage area to the passage Mach number, which determines the friction loss, based on the suction of the entire clearance mass flow. Equation 5.13 relates the tip density at the suction point to the local Mach number and the upstream stagnation density. Substituting equation 5.13 into equation 5.12 and solving for the passage Mach number at the tip showed that the passage tip Mach number

\[ \frac{A_p}{A_{ps}} = k \left( \frac{P_2}{P_0} \right) \left( \frac{P_0}{P_T} \right) \left( 1 + \frac{\gamma - 1}{2} M_P^2 \right)^{-1} \frac{M_g}{M_P} \]  

(5.12)

Another design constraint was the structural integrity of the compressor blades. Blades are highly stressed components. Difficulty then arises when flow passages, which invariably increase the stresses, are machined into the blades. Most blade stress originates from the centripetal outward acceleration acting on the blade mass. This stress is mostly radial, so removing entire columns of blade material should only incrementally increase the actual stress, while skewed passages could greatly increase the
stress. The pressure gradient across the blade creates a plate bending situation which might prove more troublesome. Classical beam theory delineates that material away from the center line retards bending the most, while material on the center line helps the least. The blade passages removed the less useful material, leaving plenty of wall on the suction side to resist the bending motion. Finally the blade also needed to withstand torsion. Maintaining the outer skin helped, but the blades still showed some degradation.

Although the equations can be used on each blade passage independently, the design coupled the pressure loss for every passage. Since all the passages dump into a common reservoir, it seemed sensible to design the passages for the same hub pressure. Each passage then had a different pressure drop as the tip pressure changed. In order to match the hub pressures, only the friction loss mattered. The radial pressure loss was the same for all channels.

Desiring several independent flow channels, the blade chord was sectioned into eight regions. Each channel had the responsibility to remove the flow passing over its region. This was easily achieved in the mid-chord where the blade is thicker. The blades are so thin at the leading and trailing edges that the end flow passages could not capture the desired flow. This was of particular concern at the leading edge, where suction could offer a greater benefit. One hope was to reduce the amount of uncaptured
flow by extending the edge regions towards mid-chord where the blade is thicker. Placing the passages as close as the region allowed to the mid-chord enabled wider passages which removed a larger flow portion, but then the edges were left untouched. In an attempt to extend the suction to the very edge of the blades, small slanted channels teeing off from the main passage were considered. These slanted passages proved impossible to manufacture, and the idea was dropped. Manufacturing demands also caused the elimination of the eighth passage and reduced the first passage to half its design length.

In the spanwise direction, the blade was discretized into five sections for pressure loss calculations. At each level, the local blade thickness determined the passage area. Given the passage area, then, the Mach number and density were calculated. The pressure loss was then calculated at each section based on these flow values. This passage discretization should help reduce the pressure error incurred by assuming that the flow conditions do not change along the passage.

The friction pressure loss is a function of the blade passage area and depended on the chosen passage dimensions. Uncoupled, it is necessary to explicitly set each blade passage geometry before calculating the pressure loss. Once coupled, only the geometry of the first blade was specified given two other limiting assumptions. First, the blade passage depth was set to a maximum 75% of the blade thickness. Second, the entire blade passage was the same
width. Initially the width was set as 0.8 times the chord length for that region. This left material to bond with the cover.

At this point, iterations began in the coupled case. The hub pressure for the first passage is calculated. Then the next passage hub pressure is calculated and compared to that of the first. If it was larger, as it invariably was, then the passage width was halved. After the next calculation, if the hub pressure was still too small, then the passage width was increased until the hub pressure matched that of the first blade. On the other hand, if the pressure drop was still too small, then the width was reset to the original value and the depth was reduced. This ensured that the passage grooves were long and skinny. The initial design was complete.

After the initial design, several modifications proved beneficial. First, since the first passage had such a large pressure loss, hub pressures were matched to the second channel instead. Then the allowable depth for the two end passages, which have large pressure losses, was increased to 0.77 times the blade thickness. Unfortunately, these changes do not completely correct the problem. Next, changes in the passage depth near the hub produce almost no effect on the passage pressure loss. These portions of the passage were so large that the flow had almost no velocity. The depth of these blade passages then was reduced to half the blade root thickness. The extra blade material will better carry the high stresses near the blade root. The design also placed the fifth hole even with the predicted location for the shock
impingement. The suction should help stabilize the shock. The resulting blades are shown in Figure 5.4.

The blades were milled in the GTL machine shop. A standard Bridgeport milling machine provided the first three degrees of freedom for blade milling. This was not enough. A dividing head held the blade by the root, providing an additional two degrees of freedom. Passages were milled using a simple end mill. The two outer passages were machined first, and then each level of the remaining passages, starting at the tip. This maintained the maximum blade support while milling the most crucial passages first. At each level, the dividing head rotated the blade to permit straight cuts. The slight dislocation of the blade created small steps at the interfaces. A hand-held Dremel tool was used to smooth these changes. Finally sanding the passages created a smooth surface.

A fiberglass epoxy sheet was bonded to the pressure side surface after groove completion. Originally, the plan called for a thin aluminum sheet to cover the plate. The aluminum, pressed into the correct shape, would bond onto the blade. This procedure required forming the aluminum nearly perfectly and then establishing two good aluminum bonds. Aluminum bonds are difficult, and the procedure would be cumbersome. Instead a composite construction utilizes only one aluminum bond and is form-fitted during application. The composite needs a similar shear modulus to that of aluminum to reduce the possibility of
delamination. Fiberglass bonded with Epoxi-Patch has a similar shear modulus and high bonding strength.

Bonding to aluminum was tricky, and even the advice of several experts at MIT did not eliminate the difficulties. The most important aspect when working with aluminum is surface preparation. To begin, sanding with a coarse paper roughened and prepared the surface. Next, washing with methelethelkeytone degreased the blade. From this point, the blade was not handled by bare skin until after the epoxy cured. Epoxying followed this surface preparation immediately. A wooden stick proved sufficient to spread the epoxy on the blades. Many blades were coated while still mounted on the rotor, so vacuum bagging was impractical. The failure to vacuum bag left some air trapped in the composite, although bubbles do not cause any real problems. Two pieces of fiberglass cloth coat each blade. Surprisingly, the final thickness is around 25 mils, while each sheet of fiberglass cloth is only 5 mils. Vacuum bagging and external pressure could be used to reduce the composite thickness in future applications.

Preventing excess epoxy from blocking the flow passages provided the biggest challenge in the blade fabrication. Attempts to fill the grooves with both wax and rubber strips failed. The wax can be removed by heating the blade moderately, and the rubber strips contract when pulled, presumably then pulling away from the epoxy. As it turns out, the wax bonded better to the epoxy than the aluminum. The result, wax impregnated epoxy reduced
the bond strength between the aluminum and the epoxy. The
cured fiberglass debonded easily into perfectly formed sheets.
Rubber on the other hand did not weaken the bonds. Instead
several portions stuck to the fiberglass well inside the blade
passages. The resulting flow blockage was unacceptable, so
another method was developed.

Since these two methods do not work, a third alternative
emerged. The fiberglass epoxy placed over the wax filled passages
debonded cleanly. The resulting piece perfectly matched the blade
contour but had a waxy surface. Upon sanding the surface to
remove the wax, this piece was bonded onto the blade. Since the
piece was already cured, a thin layer of epoxy applied directly to
the bonding surface easily sufficed to cement the bond. Because
rebonding required so little epoxy, it is not possible for the epoxy
to block the blade passages. Finally the blades were manufactured
and ready for testing.
6.1 Introduction

In order to examine the effect of tip suction, four experimental runs were completed with the MIT rotor. The runs mirrored the initial runs reported by Kerrebrock, with the exception of the blade tip suction. The initial supply tank pressure was 500 mm Hg coupled with a rotor speed of 167 rps. The four runs fell within 2-3% of these initial conditions. As expected, the results of the test runs were nearly identical to the results reported by Kerrebrock (1974) and Thompkins (1976). Four initial test runs verified the successful implementation of blade tip suction on an experimental transonic rotor.

Each of the four runs utilized a slightly different probe setup as the initial probe configuration provided little useful data. For the first run, the wall static probes were located in ports 2, 5, and 7. No four way probe data was taken. The second run was identical with the addition of the four way probe, stationed near the case \((r/r_t=.96)\) during the useful test time. The third run moved the wall static probe from port 5 to port 4 and traversed the four way probe from the wall to the mid-chord and back again. For the fourth run, the four way probe was relocated to port 5 from port 6. This move from one chord length downstream to just after the rotor trailing edge increased the individual blade resolution immensely. Data analysis after each run enabled the relocation of the sensors to more useful positions, improving the quality of the data.
6.2 Blade Suction

The experiment showed that blade tip suction is possible, though not yet fully understood. The blade suction determined by Equation 5.3 is nearly constant, varying about 10% around the mean value which is the dashed line in Figure 6.1. The magnitude of the suction mass flow was lower than expected, due mostly to the fact that the throttle tube at the base of the first suction blade (blade 1) remained closed throughout the tests. Thus only four blades actually provided tip suction. The axial flow past the rotor declined with time as expected (Figure 6.2). Taken together, the suction mass flow percentage per blade passage increased during the test time. The suction percentage varied from 0.3% to 1.2%, with a design value of 1.05%. Due to the length of the suction pipes, a pressure wave moving at the speed of sound takes just about 15 msec to travel from the suction passages to the suction dump tank. Thus the suction mass flow at a test time of 60 msec will be given by the calculated value at 75 msec. As Figure 6.3 shows, the mass flow percentage increased from 0.5% to 0.9% over the test time.
6.3 Blade Identification

Since the results show individual blade behavior and the experiment modified only five blades, it was necessary to identify each blade. For the purpose of discussion, the 23 rotor blades were numbered sequentially, with blade 1 being the first suction blade. As discussed in section 6.2, the throttle tube in blade one shut off the suction passages in that blade, so only blades 2 through 5 actually provided suction.

In addition to a number, each blade had a unique angle associated with it. With 23 blades over the 360° arc, each blade passage spanned 15.6°. The once around index separated different revolutions, and was given the value of 360°. Physically, the once around pulse corresponded to blade 4. Unfortunately this relationship only applied just downstream (port 5) of the rotor. With the four way probe in port 5, the wake from blade 4 occurred near 365°. It then follows that the region enclosed by the suction blades starts at 334.6° (blade 2) and extends up until 381.2° (blade 5). This is the region of interest for the four way probe data.

Unfortunately the blade identification just mentioned does not hold at other measurement ports. All of the probes sample at the same time, while flow structures take a finite amount of time to propagate from generation at the rotor to the probe. Upstream of the rotor, with the absence of swirl, waves propagate axially. Thus port 2 will see the pressure signature from the blade 3 when the index pulses.
6.4 Wall Static Results

The upstream wall static probes showed the clearest effect of the blade tip suction. The wall static probes register a pressure spike associated with a shock wave emanating from each blade. Epstein first documented this effect. Ideally then, each upstream wall static probe should clearly register 23 pressure spikes per revolution, or one for each blade. This is no longer the case with blade tip suction. The cause remains unknown, but the suction reduced the magnitude of one of the pressure spikes well below that of the rest. The spike is so low, that it is missing at port 2. Several representative figures present this result found in every test.

Several figures show the effects of blade tip suction on the upstream wall static pressure. Figure 6.4 displays the static pressure trace at the blade leading edge for nine rotor revolutions. The vertical lines on the figure show the passage of the once per revolution index. As the figure shows, the mean static pressure rose around the four suction blades while the peak pressure dropped slightly. Figure 6.5 shows the same plot from 65 to 80 msec. Again, the base static pressure was higher for the suction blades, while the magnitude of the shock wave, measured by the peak pressure, was reduced. An ensemble average (Figure 6.6) shows the average static pressure in front of the rotor. Here the suction also appeared to increase the median wall static pressure where the suction blades are located, with the exception of the trough between blade 3 and blade 4, which instead registered a large decrease. The shock wave pressure was especially low at blade 4, slowly increasing back to the
mean value several blades away. Figure 6.7 shows that the wall static pressure measured at port 2 for the second run follows the same trends. The ensemble average for this run (Figure 6.8) clearly shows that one of the shock waves (blade 4) no longer exists at port 2. Figure 6.9 shows the same plot for run 4. All of the runs exhibited similar behavior.
6.5 Four Way Probe Results

The four way probe, the backbone of the blowdown compressor, provided only mixed results. On the first run, the probe never entered the flow, while on the second test the probe remained in the boundary layer the whole time. The third run showed without doubt that the resolution at port 6 was not good enough to illuminate the effects of suction. For the fourth run, then, the probe sat in port 5, at most a quarter inch behind the blades. In this location, the individual blade data becomes apparent. Figure 6.10 shows that the position of the four way probe for run four is $0.94 \ r/\ r_t$ during the useful test time.

The four way probe, as reported in Thompkins, measures several important quantities. A clear picture of the flow structure emerges only after consideration of the many different flow parameters. For the tip suction, the most apparent effects will be seen in the flow angle plots. Mach number plots should support the flow angle trends. Due to the many changing variables affecting efficiency and total pressure, these plots are less useful.

The circumferential flow angle shows the largest effect from suction. Figure 6.11 shows the circumferential, or swirl flow angle. The ensemble plot averages six full rotor passages. As the figure shows, the swirl angle exhibits large peaks for every blade except the four suction blades, which instead exhibit smaller peaks. In addition to the peaks associated with the blade wakes, the ensemble average indicates the existence of peaks in the blade passages. This
peculiarity merits closer examination using data from single rotor revolutions.

Four consecutive rotor revolutions from the test interval showed similar trends. Figure 6.12 displays the circumferential flow angle from the second revolution. From the figure it can be seen that the suction blade wakes exhibit the same peak flow angle as do the unsucked blade wakes. Enlarging the region around the suction blades shows the existence of relatively large peaks in flow angle in between the blades. Figure 6.13 shows the increase in flow angle in the blade passages for the suction blades during the first analyzed revolution. This figure also shows a moderate peak associated with blade 4 on the first revolution, though it has clearly established itself in the second revolution (Figure 6.12). Finally Figure 6.14 shows that the in passage flow angle does not exhibit the large increase in circumferential flow angle for blades 6 through 10. The increase in flow angle indicates larger turning of the main passage flow in the suction region.

The radial flow angle data exhibits some change over the range of the suction blades. Figure 6.15 shows the ensemble average of the radial flow data, where the radial or outward flow is a sign of vorticity in the flow field. As can be seen, the radial angle shows significantly lower maximum and minimum values with the suction blades. Large spikes exist at either edge of the suction blades. Confusing the issue, however, is the fact that several other blades exhibit the same behavior. These other blades with relatively low radial angle variations are all located on the same rotor half as the suction blades. Curiously, one blade on each side of the suction
blades exhibits the normal radial flow variations. The reduction in radial flow variation suggests that the vorticity in the tip region is moderated through blade suction, although this explanation does not explain the behavior evidenced by the neighboring blades.

In a related analysis, Smilg (ref. 17) calculated the relative magnitudes of the radial flow angle. He found, on average, that the suction blades exhibited more radial outflow than did the nonsuction blades.

Flow Mach numbers shed additional light upon the flow in the blade passages. Figures 6.16, 6.17 and 6.18 show respectively the axial, tangential and radial Mach numbers. The axial Mach number shows a large decrease with blade 2 and a large increase with blade 5. The large variations in axial Mach number indicate that there are two separate regions of flow with somewhat different velocities. Blades 2 and 5 represent the boundaries of these two regions. At the boundaries, the flow exiting from the suction passages must re-align with the unchanged flow creating large changes in axial Mach number as the results show.

The other two flow directions provide little data at this time. At best, the radial Mach number follows almost exactly the trends already mentioned for the radial flow angle.

The four way probe also gives the downstream total pressure which in turn leads to the total pressure ratio ($\pi_D$). The magnitude of the total pressure, 1.5, is reasonable and correlates well with previous data\(^1\). The blades with suction, and also several others,

\[^1\text{Blade 10, located at 460 degrees, consistently shows low total pressure and low efficiency. This is due to a bent blade tip at the leading edge.}\]
exhibit large regions of total pressure deficit in the blade wakes while the mean trend is consistent (Figure 6.19).

Knowing the total pressure ratio, the total temperature is calculated utilizing Euler's equation and several assumptions. The solution for the total temperature assumes that straight and uniform flow enters the rotor face. The calculated value of total temperature using Equation 5.1 was 1.12, with large increases measured in the blade wakes. Figures 6.20 and 6.21 show the total pressure and total temperature ratios for rotor revolutions one and three respectively. Both plots show larger total temperature and pressure ratios in the blade wakes. This trend in total pressure conflicts with that shown by the ensemble average.

The efficiency, as shown in Figures 6.22 and 6.23 for rotations one and two respectively, dances all across the graph. The efficiency is the standard compressor efficiency

$$\eta_D = \frac{\gamma - 1}{\pi_D^\gamma - 1}$$

as calculated with the total pressure and temperature ratios. The mean value is around 90%, consistent with expectations for the MIT rotor. Figure 6-24 shows an enlarged view of the efficiency for the suctioned blades. The mean value appears lower than that of the blades without suction, shown in Figure 6-25. The trend for lower efficiency is consistent with the computational results for the given suction ratio. Figure 6-24 also shows large downward efficiency spikes at 335° and 380° which correspond to the shear boundaries separating the suctioned region from the unaltered flow.
Although entropy and efficiency provide redundant information, the entropy plot has value in this case. Entropy, 

\[
\frac{\Delta s}{C_p} = \ln \left( \frac{\tau_D}{\pi_D^{\gamma - 1}} \right)
\]

(6.2)
calculated from the total pressure and temperature ratios captured the blade wakes (Figure 6.26), whereas the efficiency plot did not. For revolution 2, the suction blades exhibited larger entropy generation in the blade wakes, with exceptionally large increases in the wakes of blades 2 and 5.
6.6 Result Analysis

The measurements from the tip suction experiments show that the suction altered the flow field. While the results do leave room for interpretation, they consistently point in one direction, namely that the tip clearance was moderated as desired. First, there is a larger radial outflow, indicating more flow room near the casing. The loose interpretation is that the clearance vortex has been diminished, reducing the blockage and allowing more flow into the region. Sinically, the increase in outward flow must exist to offset the amount of suctioned fluid. However, the reduction in radial flow angle variation does seem to indicate that the vortex is definitely weaker.

Next, the flow exhibited more turning in the blade passages themselves. For the suction to induce a larger tangential variation in the center of the channel, the effect should be seen across the whole blade passage. Instead the increased turning is a localized event, indicating that the flow is no longer being pushed away from the suction surface by the tip clearance vortex. Again, the data supports the hypothesis that suction reduced the tip clearance vortex.
Blade tip suction holds promise to affect compressor performance. Initial calculations show that blade suction has the possibility to improve compressor performance.

1) Compression of higher entropy fluid is less efficient in the Brayton cycle, so that removal of the high entropy fluid can increase compressor efficiency.

2) Computations of the effect of blade tip suction on the flow field predict small increases in adiabatic efficiency, calculated from the computed downstream flow field.

3) Blade tip suction has been implemented on a transonic rotor in the Blowdown Compressor.

4) Blade suction does change the compressor flow field. Results conclusively show a change in the upstream shock structure. Downstream results measure a shift in the flow field for the suction blades. The variations suggest that the vortex strength waned as a result of the tip suction.
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Figure 2.1: A T-s diagram depicts the standard path of compression from \( P_i \) to \( P_f \) with a polytropic compressor efficiency.

Figure 2.2: With blade surface suction at pressure \( P_B \), the entropy of the compressor mass flow decreases (a) before continuing the compression to \( P_f \), while the high entropy suctioned fluid expands back to \( P_i \).

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Pressure Side Data
Mass Flow Through the Tip Region

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Suction Massflow Percentage

Time [msec]

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