A Global Approach to Turbomachinery Flow Control: Loss Reduction using Endwall Suction and Midspan Vortex Generator Jet Blowing

Dissertation

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Abstract

A flow control scheme using endwall suction and vortex generator jet (VGJ) blowing was employed in an effort to reduce the turbine passage losses associated with the endwall flow field and midspan separation. Unsteady midspan control at low $Re$ had a significant impact on the wake area-average total pressure losses, decreasing the losses by 54%. Initially, the focus of the endwall control was the horseshoe vortex system. The addition of leading edge endwall suction resulted in an area-average total pressure loss reduction of 57%. The minimal additional gains achieved with leading edge endwall suction showed that the horseshoe vortex was a secondary contributor to endwall loss production (primary contributor - passage vortex).

A similar flow control strategy was then employed with an emphasis on passage vortex ($PV$) control. During the design, a theoretical model was used that effectively predicted the trajectory of the passage vortex. The model required inviscid results obtained from two-dimensional CFD. It was used in the design of two flow control approaches, the removal and redirection approaches. The emphasis of the removal approach was the direct application of flow control on the endwall below the passage vortex trajectory. The redirection approach attempted to alter the trajectory of the $PV$ by removing boundary layer fluid through judiciously placed suction holes. Suction hole positions were chosen using a potential flow model that emphasized the alignment of the endwall flow field with inviscid streamlines. Model results were validated using flow visualization and particle image velocimetry (PIV) in a linear turbine cascade comprised of the highly-loaded $L1A$ blade profile.
Detailed wake total pressure losses were measured while matching the suction and VGJ massflow rates, for the removal and redirection approaches at $Re_{Cx}=25000$ and blowing ratio, $B$, of 2. When compared with the no control results, the addition of steady VGJs and endwall suction reduced the wake losses by 69% (removal approach) and 68% (redirection approach). The majority of the total pressure loss reduction resulted from the steady spanwise VGJs, while the suction schemes provided modest additional reductions (<2%). At $Re_{Cx}=50000$, the endwall control effectiveness was assessed for a range of suction rates without midspan VGJs. Area-average total pressure loss reductions of up to 28% were measured in the wake at $Re_{Cx}=50000$, $B=0$, with applied endwall suction employed using the removal scheme (compared to no suction at $Re_{Cx}=50000$). At which point, the total pressure loss core was almost completely eliminated. Two-dimensional PIV showed that the endwall suction changed the location of the PV eliminating its influence on the suction surface of the turbine blade. More significantly, suction with the removal approach removed the corner vortex (CV) increasing the available span by more than 10%. The redirection approach was less effective at higher suction rates due to the continual presence of the CV.

A system analysis was also performed that compared the power needed to operate the flow control system (combined suction and VGJs) to the power gained by the system. The power gains were assessed by comparing the change in lift and wake total pressure losses (available work of the fluid) with and without flow control. The resultant power ratio showed that only 23% of the total power gained was needed to operate the flow control system for an L1A rotor at $Re_{Cx}=50000$, $B=2$. 

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Dedication

I dedicate this dissertation to my family, Mary Catherine, Claire, and Luke. The frustrations and stress of work quickly vanish in their presence. I am also indebted to my parents, Mark and Kerry Bloxham, for their encouragement and support.
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Nomenclature

\begin{align*}
A & \quad \text{area} \\
B & \quad \text{blowing ratio } \left( \frac{U_{\text{jet}}}{U_{\text{local}}} \right) \\
Cx & \quad \text{axial chord } (0.1435 \text{m}) \\
Cp & \quad \text{coefficient of pressure} \\
D & \quad \text{characteristic diameter} \\
F_{\text{Lift}} & \quad \text{lift force} \\
H & \quad \text{shape factor} \\
K_{li} & \quad \text{inlet loss coefficient} \\
K_{le} & \quad \text{exit loss coefficient} \\
L & \quad \text{characteristic length} \\
P & \quad \text{pressure} \\
Q & \quad \text{volumetric flow rate } [\text{m}^3/\text{s}] \\
Re & \quad \text{Reynolds number} \\
Re_{d} & \quad \text{Reynolds number based on diameter } \left( \frac{U_{\text{in}}}{d} \right) \left( v \right) \\
Re_{Cx} & \quad \text{Reynolds number } \left( \frac{U_{\text{in}}}{Cx} \right) \left( v \right) \\
S & \quad \text{span} \\
SR & \quad \text{suction rate } [\text{kg/s}] \\
T & \quad \text{temperature or blade thickness} \\
U & \quad \text{velocity } [\text{m/s}] \\
P_{\text{Pump}} & \quad \text{pump power requirement} \\
P_{T,\text{Gain}} & \quad \text{power gain to reduced } P_T \text{ loss} \\
P_{\text{Gain}} & \quad \text{total power gain from control} \\
W & \quad \text{rate of available work} \\
cp & \quad \text{specific heat} \\
d & \quad \text{diameter } (\text{VGJ or cylinder}) \text{ or differential change} \\
e & \quad \text{energy per unit mass} \\
f & \quad \text{Darcy friction factor} \\
g & \quad \text{gravity} \\
hL & \quad \text{head loss} \\
h_s & \quad \text{pump head} \\
\dot{m}_p & \quad \text{passage mass flowrate} \\
\dot{m}_s & \quad \text{suction mass flowrate} \\
\hat{n} & \quad \text{wall normal direction} \\
\hat{p} & \quad \text{pitch direction} \\
p & \quad \text{pitch} \\
q_{\text{rev}} & \quad \text{reversible heat addition per unit mass} \\
r & \quad \text{radius or radial direction} \\
s & \quad \text{entropy per unit mass} \\
t & \quad \text{time} \\
u & \quad \text{x velocity component} \\
v & \quad \text{y velocity component} \\
w & \quad \text{z velocity component} \\
w_{\text{rev}} & \quad \text{reversible work per unit mass} \\
x & \quad \text{Cartesian coordinate} \\
y & \quad \text{Cartesian coordinate} \\
y^* & \quad \text{wall normal direction in wall units} \\
z & \quad \text{Cartesian coordinate} \\
\Gamma & \quad \text{circulation or the potential flow suction rate} \\
\Delta & \quad \text{change} \\
\Omega & \quad \text{stage rotational rate} \\
\beta & \quad \text{increase in available span} \\
\gamma & \quad \text{wake loss parameter} \\
\delta_{\text{gg}} & \quad \text{boundary layer height} \\
\delta & \quad \text{displacement thickness} \\
\eta & \quad \text{pump efficiency} \\
\theta & \quad \text{radial coordinate} \\
\mu & \quad \text{dynamic viscosity} \\
v & \quad \text{kinematic viscosity} \\
\rho & \quad \text{fluid density} \\
\sigma & \quad \text{solidity } (p/Cx) \\
\phi & \quad \text{flow coefficient} \\
\omega & \quad \text{vorticity} \\
\text{subscripts} \\
A & \quad \text{suction location} \\
B & \quad \text{VGJ location} \\
S & \quad \text{static} \\
T & \quad \text{total} \\
ax & \quad \text{axial} \\
ex & \quad \text{exit}
\end{align*}
Abbreviations

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<td>CFD</td>
<td>computational fluid dynamics</td>
</tr>
<tr>
<td>CV</td>
<td>corner vortex</td>
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<tr>
<td>HV</td>
<td>horseshoe vortex</td>
</tr>
<tr>
<td>LIA</td>
<td>high performance blade profile</td>
</tr>
<tr>
<td>LIM</td>
<td>high performance blade profile</td>
</tr>
<tr>
<td>Pack B</td>
<td>high performance blade profile</td>
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<td>PIV</td>
<td>particle image velocimetry</td>
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<td>PSHV</td>
<td>pressure side horseshoe vortex</td>
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<tr>
<td>PV</td>
<td>passage vortex</td>
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<tr>
<td>SSHV</td>
<td>suction side horseshoe vortex</td>
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<tr>
<td>SV</td>
<td>secondary vortex</td>
</tr>
<tr>
<td>TSV</td>
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<tr>
<td>TV</td>
<td>tertiary vortex</td>
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Chapter 1: Introduction

Aerodynamic loss in the turbomachinery environment can be divided into three main categories, profile, endwall (secondary), and tip losses (will not be addressed). The distinction between profile and endwall loss is largely based on the location of loss generation rather than the mechanisms that actually generate the loss. In fact, in both cases the primary mechanism for loss generation is the influence of pressure gradients on the maturation of impinging or newly developing boundary layers. Profile (midspan) loss refers exclusively to aerodynamic losses that are generated along the middle span of the turbine or compressor blade. Endwall loss occurs in the tip and hub region of turbine and compressor blades due to the presence of vortical structures. Many attempts have been made to mitigate both profile and endwall losses with varied success. A more detailed discussion of each of these loss generators and some of the loss-mitigation attempts is provided below.

A. Endwall Losses

Horseshoe Vortex

According to Sharma and Butler [1], 30-50% of the aerodynamic losses in an axial turbine blade or stator row are attributed to secondary flows in the endwall region. Secondary flows are dominated by a system of interacting vortical structures; the most significant of which are the horseshoe ($HV$) and passage ($PV$) vortices. The $HV$ (also referred to as the wing-body junction vortex or necklace vortex) occurs in the endwall region as the boundary layer impinges on a bluff body in the flow. The boundary layer is characterized by low-momentum fluid which is retarded by the adverse pressure gradient due to the downstream presence of the turbine blade leading
edge. The adverse pressure gradient causes the boundary layer to separate and roll up into the HV system. A useful schematic of the HV system was provided by Sabatino and Smith [2] (Fig. 1.1).

![Fig. 1.1 Representation of the horseshoe vortex system provided by Sabatino and Smith [2].](image)

The HV system is comprised of at least four vortical structures; the most prominent member of this vortex system is the HV. After the HV forms near the stagnation point, it wraps around the leading edge of the turbine blade. The portions of the vortex that wrap around the suction and pressure surfaces are referred to as the suction (SSHV) and pressure (PSHV) side legs respectively. The pressure side of the HV migrates into the turbine passage and merges with the PV. Figure 1.2 contains a smoke wire visualization image of the horseshoe vortex system upstream of a cylinder in cross flow. The cylinder leading edge is visible on the right side of the image. The flow direction is from left to right. Both the HV and TV are labeled on the image. The image is included because it provides a physical manifestation of the HV, TV, and the horseshoe vortex legs wrapping around the cylinder leading edge.

Devenport and Simpson [3] showed that the HV system is subject to large-scale, low-frequency, bistable unsteadiness upstream of the obstruction. They attributed the bimodal distribution to the reversal of turbulent flow as it impinged on the downstream obstruction. The recirculation of the turbulent flow depended heavily on its origin (boundary layer or freestream fluid). Sabatino and Smith [2] generated two-dimensional probability density functions of the position of the HV (Figs. 1.3a and 1.3b) on the symmetry and 10° planes. Three hundred digital
images of the \( HV \) system were obtained using a particle image velocimetry system on each plane. The \( HV \) position was assumed to be the region of peak negative vorticity. Based on an integration of the bimodal regions in Fig. 1.3a, Sabatino and Smith determined that the \( HV \) resided in the upstream location 70-80\% of the time. They also noted that the bimodal distribution was not evident beyond the symmetry plane. They attributed this to vortex stretching and the influence of the impinging turbulent boundary layer on the structure.

Fig. 1.2 Smoke wire visualization of the horseshoe vortex system.

Fig. 1.3 Probability density function from Sabatino and Smith [2].
There are many detrimental effects of the HV. These effects depend largely on the HV location and the application of the obstruction that causes its formation. In external aerodynamics, a HV is found at the wing/body juncture of an airplane. The vortex reduces the wing’s ability to create lift in its immediate vicinity. The legs of the HV can remain coherent far downstream of the wing trailing edge. This can be problematic depending on the structures/machinery that reside in the downstream wake of the vortex cores. This is especially pertinent when one considers the unsteady nature of the HV. In water applications, HV systems develop on the ocean floor at the base of pier pilings. The HV scours the sand and sediment from the piling. If not properly set, the scouring motion can have a catastrophic impact. Submarines develop HV systems at the hull/sail juncture. The unsteadiness of the vortex can result in unwanted noise (loss of concealment). In turbomachinery applications, the HV system develops at the endwall leading edge of compressor and turbine blades. The structure scours low-momentum boundary layer fluid from the near wall leading edge region. This low-energy fluid congregates in the vortex which is enveloped by the PV. The HV also increases local heat transfer, pulling high temperature freestream fluid into the wall. This can be especially damaging to high-pressure turbine blades that already function at freestream temperatures beyond the melting point of the base metal.

Passage Vortex

There are two approaches to describing the formation of the passage vortex system, a vorticity approach and a pressure/boundary layer approach. Although both descriptions embody the same dynamics, understanding both provides a greater appreciation of the physical mechanisms involved. The vorticity description attributes the formation of the passage vortex to the intrinsic vortical motion of the incoming boundary layer and the velocity gradients in the passage. As the boundary layer enters the turbine passage the velocity gradients in the passage accelerate the
suction side of the boundary layer vortex tube. This causes an inclination in the vortex tube which results in streamwise vorticity near the suction surface.

The pressure/boundary layer approach describes the motion as pressure gradient induced. The pressure gradients of a turbine passage are primarily defined by the inviscid midspan flow field. These pressure gradients are imposed through the endwall boundary layer. The balance of the passage pressure gradients and fluid element momentum dictate the element path. Boundary layer fluid elements have less momentum than their freestream counterparts. Consequently, as the particles travel through the passage, the inviscid pressure gradients overcome their momentum and drive the particles toward the suction surface.

The presence of the PV results in a three-dimensional boundary layer separation near the endwall/suction surface junction. The separation reduces the turbine blade’s ability to produce lift in that region. Due to its sense of rotation, the tendency of the PV is to climb the suction surface wall. Consequently, the trailing edge endwall separation can cover a significant amount of the total blade span for short aspect ratio blades, as shown by Sharma and Butler [1], Langston et al. [4], and more recently by Palafox et al. [5].

Gostelow [6] provided an informative smoke visualization image of the PV in a linear turbine cascade as seen in Fig. 1.4. In this case, the flow moves from bottom to top. The smoke follows the fluid streamlines toward the leading edges. Once the smoke traces enter the turbine passages, they are swept up by the PV. Their motion in the passage clearly outlines the PV. The structures remain coherent far downstream of the exit plane. This can be problematic for subsequent blade rows. The PV can also alter the fluid exit angle at the trailing edge. The downstream blade row sees that deviation as an inlet incidence angle. Depending on the severity of the deviation, the losses can be significant. The downstream coherence of the PV can also lead to unsteady loading on the subsequent blade row.
The PV also has an impact on the heat transfer in the turbine passage. The rotation of the vortex pulls high temperature freestream fluid into the near-wall region. At the same time, the vortex collects the near-wall fluid and pushes it out into the freestream. This is very disruptive to film coolant strategies which rely on coolant in the near wall region.

The interaction of the horseshoe and passage vortices has been a topic of debate for the last 60 years. Although the debate has not been entirely resolved, the general consensus is that, to some degree, the flow field is represented by the secondary flow model provided by Langston [7] and shown in Fig. 1.5a. Sharma and Butler [1], Goldstein and Spores [8], and Wang et al. [9] have also contributed similar models to the scientific community (Fig. 1.5b-d). The main difference between these models is the treatment of the interaction of the counter-rotating suction side leg of the HV and the PV. Langston’s model suggests that the suction side leg of the HV remains in the corner region below the PV. Sharma and Butler’s model suggests that the suction leg moves under the PV initially but then wraps multiple times around the PV by the exit of the passage. The model provided by Goldstein and Spores suggests that the suction leg migrates up the span of the suction surface and remains above the PV. A flow visualization study performed by Wang et al. suggests that the suction leg migrates up the span above the PV but then wraps once around the
PV by the exit of the passage. The existence of so many models attests to the complexity of the flow field and the difficulty of measuring the flow structures.

*Endwall Flow Control Techniques*

Many researchers have attempted to mitigate the total losses due to these structures by altering the development of the HV or PV with varying success. A majority of these studies have utilized passive control devices including leading edge modifications (fillets, bulbs, etc) [10-15], mid-pitch endwall fences [16-18], and endwall profiling [19-24]. Studies have also been performed to assess the impact of forward and backward facing steps [25-27] and leakage flows [27, 28] between adjacent blade rows. These phenomena are characteristic of any axial turbomachinery environment. Examples of the control techniques mentioned above can be found in Fig. 1.6.

![Secondary flow field models](image-url)
Becz et al. [10] compared the loss characteristics of two different turbine blades with leading edge modifications. Their comparison included blades modified with a leading edge fillet, a small bulb, and an unmodified baseline blade. Their work furthered the efforts of Sauer et al. [11] (leading edge bulb modifications) and Zess and Thole [12] (leading edge fillet modifications).

Leading edge bulb modifications are implemented to impact the development of the horseshoe vortex structure. Sauer et al. measured a reduction in secondary losses of approximately 50%. These results were obtained in a turbine cascade consisting of twelve T106 blades with small leading edge bulbs. They attributed this reduction to the strengthening of the suction side.
horseshoe vortex. As was mentioned previously, this side of the vortex has a counter rotation to the passage vortex. Sauer et al. postulated that the net losses were reduced due to the interaction of these counter rotating structures. These results were later validated by the same group using bulb modifications in a compressor cascade [13]. Becz et al. found that implementing the bulb modification did not reduce the overall loss but did enable slightly higher blade loading. They suggested further work into the design and characteristics of the leading edge bulb.

Becz et al. also showed that a leading edge fillet modification reduced the mass-averaged total pressure loss coefficient by ~7% and reduced the strength of the passage vortex. Like bulb modifications, leading edge fillets are implemented to accelerate the impinging boundary layer fluid. They typically extend far upstream of the leading edge. The acceleration reduces the flow’s susceptibility to separate and form the horseshoe vortex system. Zess and Thole used computational methods to study the impact of various leading edge fillet geometries on the development of the horseshoe vortex. They chose the most effective fillet design and then validated their result experimentally. The experimental data did not show the presence of a leading edge horseshoe vortex. They also noted an order of magnitude reduction in the secondary kinetic energy associated with the vortex and a delay in the formation of the passage vortex.

Devenport et al. [15] obtained similar results at zero angle of attack but noted that the benefits of the leading edge fillet diminished as the angle of attack increased. Devenport et al. [14] also studied the impact of a simple fillet at the wing/body junction. Unlike a leading edge fillet that extends upstream of the turbine leading edge, the simple fillet used was a quarter circle centered equidistant from the wing surface and endwall. The simple fillet encompassed the entire junction. Devenport et al. showed that the simple fillet did not prevent leading edge separation or inhibit the development of the HV.

Another secondary flow technique was employed by Chung et al. [16, 17] and more recently by Govardhan [18]. Their technique sought to block the cross-passage migration of the PV with a
small endwall fence or obstruction. Chung et al. used a triangular shaped streamwise fence with the objective of lifting the vortex up into the freestream fluid. The freestream fluid would then weaken the structure and change its trajectory. Instead of impinging on the suction surface, the structure would follow the fence and be led out of the passage. Chung et al. showed experimentally that the structure did not reach its full strength with the endwall fence, nor did it have as dramatic an impact on the suction surface flow field. Consequently, the aerodynamic losses were reduced. One of the main obstacles to this technique in the turbine environment is that the fence would require extensive cooling.

Endwall profiling has also shown promise in the realm of passive secondary flow control. The aim of endwall profiling is to alter the pressure gradients in the turbine passage with the intent of reducing the strength of the passage vortex. Harvey et al. [19] attempted to reduce the passage vortex by increasing the static pressure near the suction surface (concave geometry) and decreasing the static pressure near the pressure surface (convex geometry). Ingram, Gregory-Smith, Rose, Harvey, and Brennan [20] compared the impact of two modified endwall designs and a planar endwall in their high-pressure turbine cascade. The first modified design had profiling that began far upstream of the turbine leading edge and ended downstream of the trailing edge. The second modified design focused the profiling in the region of the passage vortex. The experimental pitch averaged losses suggested that the localized profiling was more effective in reducing the loss (up to 24%).

Compressor and turbine sections of modern axial gas turbine engines consist of numerous parts. The machining tolerances allow for small gaps and steps between parts to allow for assembly and expansion/compression during thermal cycling. The impact of steps and gaps on the aerodynamic flow field has been investigated [25-27, 29]. de la Rosa et al. [25] studied the impact of a pitchwise step located upstream of the inlet plane of a turbine cascade. They showed that the direction and size of the step were important considerations. Backward facing steps reduced,
while forward facing steps increased, the losses compared with a flat endwall. Thus, careful
design and assembly could ensure that beneficial steps are included to reduce secondary losses.

The impact of leakage flows through tolerance gaps has also been studied by various groups
[26-28]. Rehder et al. [28] studied the impact of leakage flow near the leading edge of a three
blade low pressure turbine cascade. In their study, Rehder et al. measured the impact of various
leakage rates and leakage gap orientations, tangential or perpendicular leakage. The blade gap
was simulated with a backward facing step. Leakage flow rates were studied from 0-2% of the
freestream flow. It was found that leakage rates above 1% were necessary to impact the
secondary flow structures. Consequently, results were only presented for the no leakage (0%) and
maximum leakage (2%) cases. Using oil flow visualization, static pressure taps, PIV, and total
pressure measurements, Rehder et al. were able to show that tangential leakage upstream of the
leading edge of the turbine cascade greatly reduced the horseshoe vortex and consequently the
secondary losses. The reduction of the horseshoe vortex was attributed to an increase in the near
wall momentum due to the leakage flow. The passage vortex appeared to remain unaffected by
the tangential blowing. Perpendicular injection increased the size of the horseshoe vortex and the
secondary losses. The results of this study suggest that mass injection can reduce the impact of
the HV but careful consideration needs to be taken in the design of the injection sites.

The flow control techniques that have been mentioned to this point are considered passive.
Passive techniques refer primarily to geometric modifications (fillets, endwall contouring, steps,
etc) that impact the flow over the full regime of operation. At design conditions, passive
techniques can be very effective. Unfortunately, as the operating conditions change, passive flow
control can add to the system losses. Conversely, active flow control techniques adapt to the
operating conditions, maximizing their effectiveness at any condition. These systems are more
complex requiring a feedback loop. Often these types of flow control schemes are studied without
the additional complexity of the feedback loop. Instead, the flow control concept becomes the
primary focus accompanied by the concession that the system is not truly active but lends itself to a feedback loop. This manuscript employs a similar practice. Examples of active flow control systems are flow injection [30, 31] and removal [32-37] schemes.

Aunapu et al. [30] and Doerffer et al. [31] used endwall blowing in an attempt to mitigate the \( PV \) and \( HV \) respectively. Aunapu attempted two different jet hole configurations. The first configuration employed six wall jets near the saddle point of the \( HV \). This approach had no discernible impact on the location or size of the \( PV \). They also employed a midpitch row of jets in an attempt to mimic the presence of an endwall fence. This was done in an attempt to lift the \( PV \) up into the freestream fluid for trajectory redirection. Surface dot visualization suggested that for certain blowing ratios this technique was effective. Doerffer attempted to induce streamwise vorticity using blowing from an endwall jet. The jet was placed upstream of the leading edge of an airfoil. Depending on the location of the jet, control of the \( HV \) could be achieved.

Although the primary focus of this study is secondary flow mitigation in turbomachinery, attempts have also been made to reduce the impact of the \( HV \) system in external flow applications. For example, attempts have been made to minimize the \( HV \) system formed at wing-body junctions. These efforts have included the use of fillets and leading edge fairings [14, 15], techniques similar to those discussed earlier for turbomachinery flows. Of particular interest to the present study are the flow removal schemes [32-38].

Philips et al. [32] used boundary layer removal at a wing-body junction to reduce the impact of the horseshoe vortex system. They used a rectangular hole (150 mm by 190 mm) located on the tunnel wall upstream of the leading edge of a wing model (max airfoil thickness, \( T \), of 140 mm). Five different non-dimensional volumetric suction rates were applied. Suction rate was defined as the volume of the boundary layer that was removed relative to the volume flow rate of an undisturbed boundary layer (without the airfoil). A five-hole total pressure probe was used to map the total pressure losses 215 mm downstream of the leading edge adjacent to the airfoil. The
results were used to calculate the strength of the horseshoe vortex. The effectiveness of boundary layer suction was then quantified by calculating net circulation in the region of the vortex. The study showed that net circulation decreased with increasing suction rates and that a non-dimensional volumetric flow rate of ~190% was required to effectively eliminate the HV. Unfortunately, the edges of their finite width suction slot acted as streamwise vorticity generators. Consequently, they were unable to fully eliminate the net circulation in the data plane.

Johnson et al. [33] compared the impact of perpendicular blowing and suction through a round hole (20 mm diameter) that was positioned in the endwall upstream of the leading edge of an airfoil (T = 15 mm). They studied the flow field with three different Reynolds numbers, four angles of incidence, and four non-dimensional blowing and suction rates. They noted that greater HV control was achieved with mass flow removal compared with injection. They also observed that the HV strength increased (but size decreased) with increasing Re, and that the effectiveness of both blowing and suction decreased with increasing incidence.

Barberis et al. [34] compared the impact of a leading edge fillet and leading edge boundary layer removal (through a 100 mm x 82 mm rectangular suction slot) on the formation of the HV system for a symmetric airfoil (T = 180 mm). They found that boundary layer removal was more effective than a fillet and that the effectiveness of the boundary layer removal increased as the suction slot was moved closer to the endwall-body junction. Seal et al. [35] used a much narrower slot in their study (64 mm x 2 mm). Similar to the previous studies, their suction slot was offset (90 mm) from the leading edge. They found that surface suction was a viable option for HV control. They noted that surface suction weakened the instantaneous HV structure, effectively eliminated the time-average symmetry plane vortex structure, and weakened the time-average downstream legs of the HV.

Each of these boundary layer removal studies is mentioned by Simpson [36] in his annual review of junction flows. Simpson concludes that leading edge fillets are a more viable option for
HV control due to the decreasing effectiveness of surface suction at increasing incidence as mentioned by Johnson et al. [33]. Simpson’s conclusion was based on the results for suction slots that were offset from the leading edge of the wing/body junction. The boundary layer removal scheme’s dependence on incidence angle could be reduced by placing the suction slot nearer the endwall-body junction.

Flow removal schemes have also been developed in an attempt to eliminate secondary flows in turbomachinery. Gummer et al. [37] performed a computational study that looked at the impact of suction in the endwall of a compressor stator. They used a variety of different bleed geometries to compare the impact on the three-dimensional endwall flow field. The result of two bleed geometries (circular and tailored) was presented and compared with the no bleed stator results. The circular bleed was located in the middle of the endwall blade passage near the aft portion of the blades. This bleed configuration increased the size of the three-dimensional boundary layer and consequently aggravated the pitchwise-averaged deviation, total pressure losses, and loss coefficient. This suggests that geometry and position are important factors when choosing an effective bleed. The tailored bleed that resulted from their study looked very different than the original circular design. The tailored bleed had the appearance of a rectangular slot and resided in the endwall region near the suction surface. This geometry reduced the size of the three-dimensional endwall flow field and had a positive impact on the loss coefficient, total pressure losses, and pitchwise deviation.

Gbadebo et al. [38] performed a similar computational study to identify the impact of suction location on the hub corner stall of a compressor blade. They chose five different slot locations, two on the suction surface and three in the endwall region. The suction rate for the slots was obtained by gradually increasing the suction until the three-dimensional separation in the hub corner region was minimized. This was done at two different incidence angles, 0.0° (0.7% of bulk inlet mass flow) and -7.0° (0.4%). The most effective slot was located about 2% chord from the
suction surface and was 2% chord wide. The slot extended from 16% to 90% axial chord. With the slot at this location, the three-dimensional separation in the corner was completely eliminated from the suction surface of the compressor. The computational results were then validated using tuft flow visualization. Gbadebo et al. attributed the effectiveness of the slot to its position relative to the intersection of the limiting streamline from the leading edge saddle point and the suction surface. Removing the three-dimensional separation reduced the total pressure losses downstream of the exit plane. It also decreased the deviation angle and the flow blockage. Although not as effective, the other slot locations also decreased the three-dimensional separation region.

B. Profile Losses

Profile losses in the turbine environment also contribute significantly to the cumulative passage total pressure losses. As the working fluid moves through the turbine passage, momentum is transferred to the turbine walls. As $Re$ decreases, laminar boundary layers form on the blade surface even in the presence of a highly turbulent freestream. Laminar boundary layers are characterized by low momentum fluid near the wall. Under certain conditions, this fluid is unable to overcome the adverse pressure gradient typical of an aggressive turbine blade. Ultimately, the flow separates from the surface resulting in a significant decrease in lift from the turbine.

Passive control techniques are very effective at controlling boundary layer separation. One of the objectives of passive techniques is to transition the boundary layer. Turbulent boundary layers are characterized by elevated near wall momentum (compared to laminar). The momentum allows the boundary layer to remain attached in spite of an adverse pressure gradient. Passive control techniques include dimples, protrusions, vortex generators, boundary layer trips, and gurney flaps.
Volino [39] employed two-dimensional rectangular bars to control low-pressure separation. He used three different bars with heights up to 0.7% of the suction surface length. The bars were effective at transitioning the boundary layer and reattaching the separation bubble. As the bars increased in height the losses also increased. Volino showed that smaller bars did not immediately transition the boundary layer but were still effective at minimizing the separation region. The bars created small disturbances that propagated downstream causing the downstream boundary layer to transition. Volino also showed that a small reattaching separation region does not generate significant loss.

As was previously mentioned, a major defect of passive control devices is that they are present regardless of $Re$ or need. At design conditions, turbines are designed to be separation free. The addition of boundary layer trips or other passive devices to any attached flow field unnecessarily increases the loss without benefit. This defect gave rise to an emphasis in adaptive or active flow control techniques. Active flow control techniques for separation control include synthetic jets [40], slot blowing [41], suction [42], vortex generator jets [43-56], and more recently, plasma actuators [57]. Coupled with a feedback loop, active flow control can be adapted to any flight condition allowing for a controlled flow field regardless of the situation. These systems have the added benefit that they can be shut off when flow control becomes unnecessary. Of these separation control techniques, vortex generator jets (VGJs) have shown significant promise. Lin et al. [43] compared various passive and active flow control techniques to control a turbulent boundary layer on a diffusing ramp. Of the active flow control techniques, VGJs were most effective in controlling the separation.

One arrangement consists of a spanwise row of VGJs near the peak $C_p$ of the turbine airfoil. VGJs are typically designed to have an aggressive skew angle and a low pitch angle where pitch angle is the angle between the jet and its spanwise projection on the turbine and skew angle is the angle between the jet and the freestream fluid direction. Steady VGJs in this configuration result
in the creation of streamwise vorticity. A pair of vortex structures is created with each vortex having opposite sense and differing vorticity magnitude. The dominant leg remains coherent far downstream of the injection site. According to an experimental study performed by Compton and Johnston [44] and numerical results by Henry and Pearcy [45] the dominant vorticity leg’s rotation pulls high momentum fluid into the near wall region reenergizing even a highly separated boundary layer. This phenomenon has been demonstrated for a range of VGJ blowing ratios.

Experiments have also shown that pulsed vortex generator jets are effective at controlling boundary layer separation for a wide range of operating parameters. The mechanisms of control for pulsed VGJs are currently not completely understood. Computational studies performed by Postl et al. [47] suggested that the primary mechanism of control for unsteady VGJs was boundary layer transition rather than streamwise vortical structures. These results were obtained at VGJ blowing ratios, \( B \), below unity. Blowing ratio was defined as the ratio of the jet exit velocity and the local freestream velocity at the VGJ location. Postl et al. did note that vortical structures began to play a more important role as the blowing ratios were increased. They also noted the formation of a two-dimensional (spanwise) disturbance in the separation bubble. This disturbance formed after VGJ actuation and helped to accelerate reattachment.

The results of Postl et al. were later validated by Bloxham et al. [54] and Reimann et al. [55, 56]. Bloxham et al. was able to show the formation and propagation of streamwise vorticity with unsteady VGJs at a \( B=2 \). Using stereoscopic particle image velocimetry, they were able to show the formation of streamwise vorticity for a range of VGJ frequencies and duty cycles. Duty cycle was defined as the ratio of the jet on time to the total jet period. The streamwise vorticity caused localized boundary layer attachment on the downwash side of the vortex. The reattached region then spread laterally resulting in a fully attached boundary layer. Once attached the boundary layer did not immediately separate in the absence of flow control. After a brief delay, the
boundary layer was shown to recover its separated state prior to the subsequent VGJ-induced disturbance.

In a companion study, Reimann et al. used hot-film anemometer excursions to show that boundary layer transition also had a significant impact on the separation bubbles of both the Pack B and L1M blade profiles. The Pratt and Whitney Pack B, which has a non-reattaching separation bubble, was most affected by the control. After reattachment, there was a delay before separation re-growth of 35% of the pulsing period. The delayed re-growth was attributed to an increase in the flow inertia of the large amplitude oscillations inherent with a non-reattaching boundary layer. In contrast to the Pack B, the L1M has a smaller separation bubble that reattaches prior to the turbine trailing edge. Unsteady VGJ control was less effective due to the reduced separation. After reattachment, the boundary layer of the L1M immediately began to re-separate.

Bons et al. [50] studied the impact of VGJs on a separation bubble using the Pack B blade profile. They used boundary layer traverses and static pressure taps to monitor the changes in the separation zone with both steady and unsteady VGJ control. They reported reductions in the wake loss profile of over 50% with unsteady control, which was later substantiated by the results of Volino [40] obtained using synthetic jets. The unsteady result obtained by Bons et al. compared favorably to the control achieved with steady VGJs but at a fraction of the mass flow requirements. These results were obtained over a range of forcing frequencies and duty cycles with the conclusion that both variables had little impact on the time-averaged wake losses. Bons et al. further showed that the extent of the control was more profoundly impacted by the starting and ending of the jet pulse rather than the amount of time the jet remained active.

**C. Combined Endwall and Midspan Flow Control**

One of the major concerns with VGJ control schemes is the source of the jet air. High energy fluid from the compressor could supply the VGJs but at a cost to the work output of the cycle. Ideally, it would be more beneficial to allow this high energy fluid to move through the
combustor and turbine. The previous discussion on flow control of endwall flows suggested that endwall fluid removal could be used to mitigate the secondary losses. The removed fluid becomes a prospective source for the VGJs. A combined (global) flow control scheme could potentially mitigate the losses of both the endwall and midspan without adversely affecting the engine work output. In order for a combined system to be useful, it must provide substantial benefit compared to its operational costs.

D. Research Objectives

The objectives of this study are outlined below.

1. Design an effective flow control scheme that simultaneously mitigates endwall and profile total pressure losses.
2. Assess the impact of leading edge suction on the formation of the horseshoe vortex system.
3. Assess the impact of passage suction on the formation and migration of the passage vortex system.
4. Identify the primary loss mechanism in the endwall of a turbine passage.
5. Identify the physics responsible for the reduction in total pressure losses.
6. Use another tool (theory or computational fluid dynamics) to aid in the design of the passage vortex endwall flow control scheme.
7. Perform a system analysis to assess the benefits of a combined flow control scheme.
Chapter 2: Experimental Facility

A. Wind Tunnel

The open-loop wind tunnel is powered by a centrifugal blower. After the flow passes through a heater and cooling section (not used in this study), the tunnel branches, providing two distinct flow paths as depicted in the schematic found in Fig. 2.1. A series of gates are provided at the branch juncture to control the flow path. Downstream of the juncture, the wind tunnel branches incorporate flow straighteners and converging nozzles to condition the flow prior to entry into the respective test sections. After the converging nozzles, the wind tunnel branches transition into 0.15 m² clear acrylic ducts. The upper branch leads to a straight acrylic duct that is primarily used for flat plate studies. This test section was used for the proof-of-concept cylinder study. The lower acrylic duct leads to a linear turbine cascade test section. The remainder of this chapter will be used to discuss the data acquisition techniques employed during the study. Specifics of each test section will be discussed in detail in the subsequent chapters.

Fig. 2.1 Schematic of Ohio State University’s low speed wind tunnel

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B. Linear Cascade

The three passage linear cascade is comprised of four L1A turbine blades (Fig. 2.2). The L1A has an axial chord of 143 mm, a span of 380 mm, and a solidity of 0.99. Designed by Clark for use in flow control studies [52], the aft loaded L1A exhibits massive separation just downstream of the minimum pressure location near 57% $C_x$ for $Re_{C_x}$ below ~50000. It should be noted that the $L1A$ blade shape is proprietary. Consequently, any blade shape found in this document is either another blade shape or a manipulated $L1A$ blade profile.

The two fully-immersed blades (labeled inner and outer) house spanwise rows of VGJs near 59% $C_x$. The VGJs have a diameter, $d$, of 2.6 mm and are spaced $10d$ apart. They are injected with a pitch angle of 30° and skew angle of 90° (see Fig. 2.2). Pressurized air is fed to the VGJs from a compressor. In a steady blowing configuration, a pressure regulator is used to regulate the VGJ blowing. For unsteady actuation, a solenoid control valve is also incorporated to control the timing characteristics of the VGJ (duty cycle and frequency). Blowing ratio is defined as the ratio of the jet exit velocity to the local freestream velocity and duty cycle is the ratio of the jet on-time to the total period. The wind tunnel was configured with an upstream turbulence generator that

![Fig. 2.2 Schematic of the L1A cascade and VGJ angles.](image-url)
provided 3% freestream turbulence at the turbine passage inlet.

**C. Total Pressure Measurements**

Total pressure data were used to assess the impact of the endwall flow control schemes in the cylinder and cascade test sections. The data were collected with a United Sensor Corporation Kiel probe similar to the schematic found in Fig. 2.3. The probe had a shroud and shaft diameter of 3.2 mm and a shaft length of approximately 0.3 m. The total pressure port is positioned inside the shroud. Unlike a conventional pitot total pressure probe, Kiel probes are insensitive to flow direction in the pitch and yaw directions. This characteristic affords accurate total pressure measurement in flow fields with an unknown flow direction (like the turbine cascade endwall). According to United Sensor, the model used in this study was insensitive to direction up to ±45° in pitch and yaw. The yaw specification was validated experimentally as shown in Fig. 2.4 below. The validation was achieved by placing the Kiel probe alongside a conventional pitot probe in the wind tunnel. The total pressure difference (y axis of Fig. 2.4) was compared as the Kiel probe was rotated in the yaw direction ±50°. Measurements outside of the specified range were shown to increase the measurement error dramatically.

Due to the geometry of the Kiel probe, the nearest wall measurement was 1.8 mm. Total pressure loss was measured using Druck brand differential pressure transducers with one port connected to an upstream total pressure reference pitot and the other port connected to the Kiel probe.
probe. The difference in total pressure was recorded for each location in a position array and normalized by the dynamic inlet pressure. An assessment of the total loss was performed using an area-average approach. Although a mass-average assessment would have been more appropriate, the necessary hardware was not available. The error in the total pressure measurements was calculated to be ±4%.

United Sensor provides a specified time constant for the Kiel probe of 15 seconds. This time constant was shown to be extremely conservative for the pressure excursions in this study. In the end, the probe was allowed to settle for three to five seconds (depending on the study) after traverse-controlled motion. Data were then collected for six seconds at a sample rate of 1000. The average value was then written to a text file.

The Kiel probe was fastened to a traverse located on top of the test section. The traverse consists of three stepper motors which control the probe motion in all three Cartesian directions. The traverse allows controlled motions of as small as 0.05 mm. The position is measured using Sony precision Magnescales® which are accurate to 0.0001 mm.
D. Particle Image Velocimetry

Particle Image Velocimetry Concept

Detailed velocity measurements were made with a two-dimensional LaVision particle image velocimetry (PIV) system mounted to a three-axis traverse below the test section. LaVision [58] provides a useful schematic that illustrates the principle of particle image velocimetry. It is provided in Fig. 2.5 below.

![Particle image velocimetry schematic from LaVision [58].](image)

The flow field is seeded with particulate that travels with the fluid. A double pulsed Nd:YAG laser is used to produce consecutive laser pulses at a known time delay. Laser light is passed through spherical and cylindrical optics to produce a sheet of laser light. The sheet is very thin (on the order of 1 mm) so as to isolate a plane of particulate. A high resolution camera is used to capture images of the particulate laden fluid for each of the laser pulses. The particulate images are then broken up into interrogation windows as depicted on the left side of Fig. 2.6 (also from LaVision [58]). The windows are typically from 64x64 to 8x8 pixels and are overlapped by up to 50%. Cross-correlation functions are applied to consecutive interrogation windows to indentify
the particle displacement. These correlations depend on fast Fourier transforms and provide a correlation map similar to the one found in Fig. 2.6. If the velocity is well resolved, a dramatic peak is obtained and the velocity is calculated from the displacement and time delay. Stereoscopic data is obtained following the same pattern with the addition of an extra camera (image pair).

![Fig. 2.6 Interrogation windows and correlation map from LaVision [58].](image)

*Particle Image Velocimetry System*

The PIV system used in this study has a CCD camera with a resolution of 1376 x 1040 pixels. The camera was outfitted with a Scheimpflug lens which allows the camera to be misaligned with the image while still maintaining focus. A single camera is only capable of measuring velocity in the plane of the laser sheet. The camera is capable of onboard storage and rapid transport of data. This is necessary in order to capture images for each laser pulse since the time delay between laser pulses can be very brief (as low as 1 µs).

The PIV system has a dual head Solo 120mJ Nd:YAG laser. The class IV laser emits a laser beam with a wavelength of 532 nm (green laser) and a diameter of 4.5 mm. The laser can operate at frequencies up to 15 Hz. According to the manufacturer, the laser is stable to ±4% (pulse-to-pulse for 98% of shots after 30 min) and produces pulse widths of 3-5 ns. The laser is mounted to
a three-axis traverse as illustrated by Fig. 2.7. The beam is altered using optics until it passes through a cylindrical lens. The cylindrical lens converts the concentrated beam into a laser sheet. This sheet is then introduced into the region of interest for particle illumination. The time delay between laser pulses is selected to ensure adequate particle motion between the image pairs (LaVision suggests ~8 pixels). This can be difficult with a single camera if the dominant velocity component is normal to the laser plane. A balance is required that accounts for in-plane particle motion while simultaneously retaining the particulate.

![PIV system prepared to take single camera data. The green caricature represents the laser path. View of the camera is obstructed by the Unistrut® system.](image)

Fig. 2.7 The *PIV* system prepared to take single camera data. The green caricature represents the laser path. View of the camera is obstructed by the Unistrut® system.
The particulate (olive oil) is added to the flow field using a seeder purchased from LaVision. The seeder is attached to a regulator-controlled high-pressure air line which introduces a high-velocity jet into the seeder’s oil reservoir. The jet impinges on the oil causing some of the oil to atomize. The particulate then leaves the reservoir and is introduced into the wind tunnel inlet. According to the manufacturer, the particulate diameter is 1-2 µm.

The PIV uncertainty was estimated using the calibration results and a t-statistic. The LaVision software outputs a pixel error during calibration that can be used to assess the bias error of the calibration. The pixel error can be directly converted to a velocity error once the user identifies the particle displacement between consecutive laser sheets. The precision error of the PIV system was assessed using a t-statistic in the steady freestream fluid. The velocity magnitude at a pixel location was tracked for all of the images in the data set. Since there were hundreds of images the t-value was approximated to be two. The combination of bias and precision error assessments provided an uncertainty in the PIV of 4%.

E. Flow Visualization

Both smoke wire and oil flow visualization techniques were employed during this study. These techniques were used to provide insight into the flow structures of interest. A brief overview of the techniques is provided here. The details of the setups and results will be provided in subsequent chapters.

Smoke Wire Visualization

Smoke wire visualization is used to seed the fluid. The smoke (seed) allows visualization of fluid streamlines and structures. The technique requires a wire with a small diameter, oil, and a Variac. The wire is placed upstream of the region of interest. It is brushed with oil. Typically, the oil beads up along the wire resulting in localized smoke production. The Variac provides a voltage potential to the wire which results in Ohmic heating. As the wire temperature rises, the oil
burns. This results in smoke production. The smoke is then illuminated with the PIV laser sheet and a video recording is obtained.

As part of this study, an automated oiling system was designed and manufactured. The system employs a DC motor and slider assembly that continuously dispenses oil onto the wire. An electrical switch was included in the design to ensure the oil assembly did not obstruct the fluid path during oil burn. The system is capable of continuous operation for hours.

*Oil Flow Visualization*

Oil flow visualization is used to assess shear stresses on a surface. The surface is coated with a mixture of oil and color pigments. As the fluid moves over the mixture, the shear stresses remove the oil from high shear locations and deposit it in regions of low shear. As the oil mixture moves, the pigment leaves streak-like patterns on the endwall indicating flow direction. In the current study, oil flow visualization was used to chart the location and trajectories of the endwall flow field in the LIA turbine passage. The regime of $Re$ used in this study required a very low viscosity fluid (brake fluid). The brake fluid was mixed with laser jet ink toner and applied to the endwall using an atomizer. Still images of the endwall were captured once the ink toner had adequately outlined the endwall flow field.
Chapter 3: Proof-of-Concept Cylinder Study

This chapter presents a proof-of-concept cylinder study performed to aid in the design of a $HV$ reduction/removal scheme in the turbine environment. The reduction scheme employed leading edge boundary layer removal at the base of a faired cylinder in crossflow. The mass flow was removed to reduce the adverse pressure gradient responsible for boundary layer separation and the $HV$ formation. The objective of the preliminary study was to validate the effectiveness of leading edge suction, assess its impact on the $HV$ system, and identify the necessary suction rates to reduce or remove the $HV$. The results of the proof-of-concept faired cylinder study were ultimately used to implement a similar $HV$ removal scheme in a linear turbine cascade.

A. Cylinder Test Section

The cylinder study $Re_d$ was $7.6\times10^3$ based on the cylinder diameter ($d=76.2$ mm) and inlet velocity, $U_{in}$. The inlet flow was shown to have a velocity uniformity of $\pm2\%$, and a freestream turbulence level of $<1\%$. An acrylic duct extension containing the test section (Fig. 3.1) is fastened to the end of the wind tunnel duct. The test section height is smaller than the acrylic duct leaving a 20 mm gap at the base of the interface. This gap was added to bleed off the boundary layer that develops in the acrylic duct. The base plate of the test section has a sharp leading edge and is outfitted with a boundary layer trip. The undisturbed (without the cylinder) boundary layer parameters at the cylinder leading edge were measured using a pitot tube ($\delta_{pp}=41$ mm and $H=1.54$). These parameters were shown to be uniform across a majority of the duct width.
A cylinder is mounted to the base plate of the test section duct. The cylinder is hollow with an outer diameter of 76.2 mm. Its leading edge is located approximately 500 mm from the leading edge of the base plate. The cylinder has a 1 mm wide slot near the base of the leading edge (base plate/cylinder interface) that extends to approximately $\theta=\pm 30^\circ$ as shown in Fig. 3.2. The slot is recessed in a trough that was removed from the base plate (inset of Fig. 3.2). Suction is applied through the slot with a vacuum pump connected to the top of the hollow cylinder. The suction rate is controlled with a rotameter and a series of valves. Fairings with a 10:1 taper are attached to the sides of the cylinder to eliminate vortex shedding.

Fig. 3.1 Schematic of the duct and cylinder test section. Flow moves from left to right.

Fig. 3.2 Top view schematic of the faired cylinder and coordinate system ($+z$ out of page). The inset depicts the suction slot position relative to the base plate trough.
B. Data Acquisition

Total pressure losses were measured on the 90° plane of the cylinder with and without boundary layer removal using a Kiel probe. The Kiel probe was mounted to a two-axis traverse above the cylinder test section. The traverse was used to collect a grid (2mm increments in z and r) of total pressure measurements in the r-z plane. The grid extended 60mm in r and 40mm in z. The total pressure data were used to gauge the effectiveness of boundary layer removal and to identify an effective suction rate.

Data were also collected using the LaVision particle image velocimetry (PIV) system described in chapter 2. The laser was positioned below the test section and emitted two consecutive laser sheets (1 mm thick) into the test region with a time separation of 200 µs. The camera captured a total viewing window of approximately 70 mm x 50 mm. Data were collected on the symmetry plane and the 90° plane with and without leading edge boundary layer removal. At each location 300 images were captured and averaged together using the LaVision software [58]. Vector processing was initially performed with 64 pixel x 64 pixel interrogation windows. The interrogation windows were then refined to 32 pixels x 32 pixels and finally 16 pixels x 16 pixels. A 50% overlap was used during the vector processing. The velocity vectors obtained from the LaVision software were then used to calculate the vorticity with a central difference approximation. A top-hat filter was applied to the resultant vorticity field to eliminate erroneous large-scale fluctuations of vorticity. A band-pass filter was also applied to the vorticity field to eliminate small fluctuations about zero. Net circulation was then calculated with the vorticity data using the net circulation equation provided by Philips et al. [32] (Eq. 3-1) where Δ^2 is the smallest differential area of the PIV grid and i and j are position indices.

\[ \Gamma = \Delta^2 \sum \omega_{ij} \]  
(Eq. 3-1)
C. Smoke Wire Visualization Results

Smoke wire visualization was used to obtain a qualitative assessment of the effectiveness of boundary layer removal on the HV formation for the cylinder test section. The wire (~0.25 mm) was placed 50 mm upstream of the cylinder and oriented perpendicular to the direction of the flow. The laser sheet was introduced in the \(\theta-r\) plane 10 mm above the base plate. A standard video camera was used to take footage of the HV with and without suction. Figure 3.3 contains two still images from the video. The left and right images were taken while the suction was turned off and on respectively. It should be noted that the boundary layer flow in these images is laminar and the \(\text{Re}_d\) is 3750 (50% lower than the nominal value). The \(\text{Re}_d\) was decreased and the boundary layer trip was removed to facilitate the visualization of the HV system. The remainder of the data presented in this work was obtained with the boundary layer trip.

A comparison of the images in Fig. 3.3 shows that removing a portion of the impinging boundary layer at the cylinder base significantly alters the near wall flow field. The image without suction clearly depicts a horseshoe vortex that wraps completely around the bluff body. As the boundary layer and smoke propagate toward the cylinder the adverse pressure gradient causes boundary layer separation. Once the near wall fluid separates the higher momentum fluid rolls over the stagnant air. The tertiary vortex is also visible upstream of the HV. When suction is
applied the smoke traces do not suggest the formation of a HV. Instead, the smoke traces move directly to the suction slot or around the cylinder.

D. Total Pressure Loss Results

Total pressure loss data were collected for five different suction rates (0, 6, 11, 15, and 23%). The suction rate was varied using a rotameter and a vacuum pump. Effectively suction rate (Eq. 3-2) was the percentage of the undisturbed (without the cylinder) boundary layer fluid removed through the suction slot.

\[ SR = \frac{\dot{m}_s}{\rho U_{in}d(\delta_{99} - \delta^*)} \times 100\% \]  

(Eq. 3-2)

The area-average total pressure losses on the cylinder were calculated for each suction rate and then normalized by the area-average total pressure losses without suction. Figure 3.4 contains the normalized integrated total pressure losses obtained for each of the suction rates measured at 0.5\(d\) (90° plane) and 2.5\(d\) downstream of the leading edge of the cylinder. A value of 1 on the ordinate axis indicates no effect of suction on the integrated total pressure losses. The trend of both data sets appears to be an asymptotic decrease in the total pressure losses with increasing suction rates. The uncertainty bars (±4%) are included to illustrate that the losses at a suction rate of 11% may not represent an optimum. The trend of both data sets (0.5\(d\) and 2.5\(d\)) suggests that there may be an inflection in the loss curves between suction rates of 11 and 20%.

The total pressure surveys on the 90° plane were also used to create contour plots. Figure 3.5 contains plots of the total pressure losses for the 0% and 11% suction rates normalized by the domain inlet total pressure. The cylinder wall and base plate border the data at \(r/d=0.5\) and \(z/d=0\) respectively. The approximate position of the vortex and its rotation is denoted by the arrow in the no suction plot. The impact of the HV on the total pressure losses is most evident between \(r/d=0.7\) and 0.9 (rounded contours). In this region the downwash of the vortex pulls high
momentum (low loss) fluid into the near wall region. The upwash of the HV scrapes low momentum (high loss) fluid away from the base plate and pushes it upward into the boundary layer. The low momentum fluid accumulates in the boundary layer resulting in a net increase of the total pressure losses. There is no evidence of the HV in the total pressure loss data with suction (11%). Consequently, the boundary layer is thinner and does not exhibit the same rounded contours observed when the vortex is present.

Fig. 3.4 The impact of suction rate on the area-average total pressure losses on the cylinder, \((P_{\text{in}}-P_{T})/(P_{\text{in}}-P_{T})_{SR=0}\).

![Graph showing normalized P_T losses for suction rates 0.5d and 2.5d.](image)

\begin{align*}
\text{Suction Rate} & \quad 0 & \quad 5 & \quad 10 & \quad 15 & \quad 20 & \quad 25 \\
\text{Normalized } P_T \text{ Losses} & \quad 1.05 & \quad 1.00 & \quad 0.95 & \quad 0.90 & \quad 0.85 & \quad 0.80
\end{align*}

Fig. 3.5 Total pressure losses at the 90° plane with and without boundary layer removal, \((P_{\text{in}}-P_{T})/(0.5\rho_{\text{in}}U_{\text{in}}^2)\)

![Graph showing total pressure losses with and without suction at 90° plane.](image)
Although it is difficult to see from these contour plots, applying suction at the endwall of the cylinder also reduced the size and magnitude of the corner vortex. The corner vortex is the small vortex very near the cylinder-wall junction as seen in Fig. 1.1. In Fig. 3.5, a slight bulge is evident in the plots near $r/d=0.55$ and $z/d=0.05$ for the no suction case. This structure is reduced with 11% suction. According to Goldstein and Spores [8] the highest mass transfer (and heat transfer due to the Heat-Mass transfer analogy) occurs in the leading edge corner vortex region. Similar to the $HV$, the corner vortex remains coherent downstream of the leading edge. As it propagates downstream, it increases the near wall heat transfer. Goldstein and Spore’s results were recently validated by Sabatino and Smith [2] and Praisner and Smith [59, 60] using liquid crystal thermography. Praisner and Smith observed a 350% increase in the Stanton number in the corner vortex region relative to the impinging boundary layer. Elevated regions of heat transfer caused by the corner vortex occur because of the strength of the vortex and the high temperature freestream fluid that forms it.

E. Particle Image Velocimetry Results

Figure 3.6 contains contour plots of the time-mean symmetry plane vorticity taken from the PIV data with and without leading edge boundary layer removal ($SR=11\%$). The vorticity is normalized by the ratio of the freestream velocity and the diameter of the cylinder. The black region near $r/d=0.5$ is a shadow caused by the obstruction of the base plate suction slot on the laser sheet. The leading edge of the cylinder and the base plate are located at $r/d=0.5$ and $z/d=0$ respectively. Streamline topologies were added to further define the flow structures.

The $HV$ is visible in the no suction plot between $r/d=0.6$ and 0.8 as a region of elevated negative vorticity. The vortex resides in two different regions as illustrated by the elevated vorticity near $r/d=0.75$ (primary position) and $r/d=0.63$ (secondary position). This bimodal behavior has been observed in the symmetry plane by various groups [2, 3, 35, 59, and 61]. The position of the primary time-mean $HV$ is about 0.24 $r/d$ from the cylinder wall and 0.06 $z/d$ from
The vertical position is similar to that observed by Praisner and Smith\cite{59}, but the streamwise position is closer to the cylinder in their data (0.18 \( r/d \) from the cylinder).

The streamline topologies clearly capture the primary position of the HV. The location of the TV is also visible in the streamlines (though not as clear). Its farthest downstream extent is near \( r/d = 0.85 \). At this point the streamlines change direction delineating the vortex. The SV is evident between the HV and TV as a region of positive vorticity in the near wall region near \( r/d = 0.77 \). Its location is further evidenced by the bending of the streamlines away from the HV near \( r/d = 0.80 \). The band of negative vorticity that extends from \( r/d = 0.9 \) to \( r/d = 1.3 \) is the bound vorticity in the boundary layer.

Applying boundary layer suction (\( SR = 11\% \)) significantly alters the time-mean symmetry plane vorticity field. The primary difference is the negative vorticity associated with the HV is no
longer present in the flow field. Consequently, the SV and TV are also no longer present. The streamline topologies demonstrate flow migration down to the suction slot. This flow migration suggests that the adverse pressure gradient has been removed. The band of negative vorticity embedded in the boundary layer is still present and extends all the way to r/d=0.7.

As further evidence of the impact of suction on the symmetry plane flow field, normalized boundary layer profiles were extracted from both sets of PIV data at select locations (r/d=0.59, 0.75, 0.91, 1.06, and 1.22). The profiles (Fig. 3.7) were normalized with the inlet velocity. Each plot is labeled at the top with its corresponding streamwise location. The black squares and circles correspond to the velocity profiles with and without suction respectively. The dashed line on each of the plots indicates the zero velocity position.

The shape of the no suction velocity profile at r/d=1.22 suggests that the boundary layer is on the verge of separation. At this location separation has not yet occurred since the velocity vectors remain positive. By r/d=1.06, there is a small region of reverse flow which increases in size as the position of the HV is approached (r/d=0.75). Praisner and Smith [59] found that boundary layer

![Fig. 3.7 Boundary layer profiles, u/U_in, taken from the PIV data with (squares) and without suction (circles). The profiles are normalized with the inlet velocity.](image)

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separation occurred between 1.05<$r/d<$0.95. Their results correspond well with the separation location of $r/d=1.0$ measured by Devenport and Simpson [61]. According to the PIV results of the current study, separation occurs between 1.22<$r/d<$1.06. This is not unexpected given that the HV in the present study is also positioned farther upstream.

Contrary to the no suction boundary layer profiles, when 11% suction is applied the boundary layer does not separate. In fact, the velocity profile remains relatively unchanged in the near wall region until $r/d=0.75$. At this point in the flow the near wall fluid begins to accelerate toward the suction slot. By $r/d=0.59$ the maximum velocity in the near wall region is almost equal to the freestream velocity. Endwall suction at this suction rate appears not only to have eliminated the adverse pressure gradient but has created a favorable pressure gradient that accelerates the flow near the cylinder.

Localized acceleration in the boundary layer creates a new vortex pair that is visible near the suction slot in the PIV image with suction (Fig. 3.6). As the suction rate is further increased this vortex pair increases in size and magnitude (see Fig. 3.8). It also migrates away from the suction slot. As noted earlier, Philips et al. [32] concluded that the edges of their cross-stream suction slot acted as sources of streamwise vorticity. The present study suggests that the act of removing fluid can also be a source of cross-stream vorticity at the stagnation point. This is particularly pertinent as the suction rate increases.

Although downstream PIV data did not show the presence of this new vortex pair, it is possible that it is responsible for the asymptotic trend seen in the total pressure losses of Fig. 3.4. This suggests that an improper suction rate could augment the total pressure losses, which could be challenging for flow control applications which operate over a wide range of flow conditions.

In a turbine passage, the pressure side leg of the HV migrates toward the suction surface of the adjacent turbine blade. As it crosses the passage it interacts with the PV resulting in the complex flow field described earlier (Fig. 1.5). In the faired cylinder configuration, there are no adjacent
blades or curved passages to strengthen the vortex. Regardless, the downstream impact of boundary layer removal is still of interest to validate the effectiveness of leading edge suction. Figure 3.9 contains time-mean normalized vorticity plots at the 90° plane with and without leading edge suction. The no suction plot still clearly shows the presence of the HV. The strength of the vortex has decreased (by ~50%) but the bimodal shape is still evident. The position of the vortex in the 90° plane is similar to the results of Praisner and Smith [60]. Also of interest in the no suction vorticity contour of Fig. 3.9 is the positive corner vortex seen near \( r/d = 0.55 \). This region of positive vorticity, spanning the right side of the plot from \( z/d = 0 \) to \( z/d > 0.25 \), is also consistent with the results of Praisner and Smith. Its maximum vorticity value of 15 is near \( z/d = 0.1 \). The band of negative vorticity that extends from \( r/d = 0.9 \) to 1.3 is attributed to the reorientation of the negative vorticity found in the symmetry plane boundary layer (Fig. 3.6).

The corresponding normalized vorticity plot with suction (11%) does not exhibit a concentrated negative vortex structure, although the negative vorticity band is still present. It
extends from near the cylinder wall to $r/d=1.3$. By this point the vorticity has almost completely dissipated. Vorticity in the corner vortex region has decreased by ~33%. The maximum vorticity in the corner vortex with suction is found at the same spanwise location as the no suction data.

PIV data were also taken 1.5$d$ downstream of the leading edge of the faired cylinder. These data were used to calculate the net non-dimensional circulation ($\Gamma/d\cdot U_{in}$) for each of the suction rates used in this study. The results are compared with the results of Philips et al. [32] in Fig. 3.10. The present study fills a void in the data of Philips et al. The data depict a more rapid decrease in the net non-dimensional circulation for lower suction rates. Unlike the total pressure losses presented in Fig. 3.4, the circulation decreases monotonically with increasing suction up to suction rates of 200%. This result bodes well for applications of secondary flow control using suction.
Seal et al. [35] similarly calculated circulation in their cylinder study with and without surface suction. Unlike Philips et al., they calculated circulation for non-dimensional streamwise vorticity values above a threshold (2.84 for the 90° plane (0.5d) and 1.42 for the 1.5d plane). Their results are presented in Fig. 3.11 along with the circulation results of the present study (employing the same minimum thresholds). Without applying suction, Seal et al. measured a maximum non-dimensional circulation on the 90° plane of 0.055. Applying suction at 17% reduced the circulation on the 90° plane by ~9%. Seal noted the presence of two vortical structures in the 90° plane. One of the structures was attributed to the horseshoe vortex (circulation of 0.023); the other structure was attributed to vorticity generation at the extremities of the suction slot (circulation of 0.027). The additional circulation from the suction slot vortex reduced the impact of suction on the overall circulation results. By neglecting the circulation of the suction slot vortex, the impact of suction solely on the horseshoe vortex can be isolated. This results in a circulation reduction of ~58%.

The present study measured a higher circulation (0.089) on the 90° plane without suction. Applying a 15% suction rate resulted in an 87% reduction in circulation on the 90° plane. In contrast to Seal et al., there was no indication of suction slot generated vorticity in the 90° plane.

![Net non-dimensional circulation](image)
Although there is a discrepancy in the magnitudes and effectiveness of the two suction schemes on the horseshoe vortex circulation, the trends of both studies are consistent.

The cylinder study provided insight into the design of a similar control scheme for the turbine environment. The main conclusion learned from the cylinder study is that applying leading edge suction can effectively eliminate the formation of the HV system. Suction alters the adverse pressure gradient that causes the vortex to form. Instead, a favorable pressure gradient develops which accelerates the near wall fluid. An inflection in the total pressure loss data shows a minimal increase in pressure loss reduction for suction rates greater than 11%. Thus, suction rate can be optimized to have maximum benefit while minimizing fluid extraction. Sabatino and Smith [2] suggested that Re scaling of horseshoe vortex systems on cylinders and turbine blades should be based on the cylinder diameter and the maximum thickness of the turbine blade as the characteristic lengths respectively. Accordingly, a similar scaling arrangement was used in the present study to set both Reynolds numbers near 10000.
Chapter 4: Horseshoe Vortex Removal Scheme

A. Turbine Test Section

This portion of the study involved use of the turbine cascade mentioned in chapter 2 and shown in Fig. 4. A splitter plate was also added to the experimental turbine cascade to incorporate the endwall suction scheme.

![Diagram of the LIA cascade](image)

**Fig. 4.1 Schematic of the LIA cascade outfitted for the horseshoe vortex control scheme.**

The splitter plate is placed 20 mm above the base plate of the test section. It is used to condition the boundary layer that impinges on the turbine endwall and house the suction slots. The splitter plate has an elliptical leading edge to minimize flow separation and, additionally, fluid is removed below the plate to ensure uniform inlet flow. The plate extends 290 mm upstream of the leading edge of the turbine blades, where a 3.2 mm trip is placed to ensure the development of a turbulent boundary layer at the turbine inlet. Mid-pitch boundary layer parameters were measured in the leading edge plane with a boundary layer pitot tube ($\delta_{99}=20$ mm, ...
\( \delta^* = 3.4 \text{ mm}, \) and \( H = 1.44 \) for \( Re_{Cx} = 25000 \) and \( \delta_{99} = 17 \text{ mm}, \delta^* = 2.7 \text{ mm}, \) and \( H = 1.45 \) for \( Re_{Cx} = 50000 \).

A 10 mm radius fillet was created at the splitter plate/turbine junction to make the passage geometry more representative of an actual engine \((r/Cx \sim 1/14 \text{ is typical})\). Leading edge suction slots were cut into the splitter plate 12 mm (to slot center) upstream of the turbine leading edge. The slots were connected to a vacuum system with variable suction rates. The suction slot geometry was selected based on the results of the cylinder study and oil flow visualization studies discussed later. The slots have a nominal width of \( \sim 4 \text{ mm} \) and length of \( \sim 35 \text{ mm} \). They also contain screens to ensure uniform suction across the slots.

This study employed unsteady VGJ actuation with a jet frequency of 5Hz, a 25% duty cycle, and blowing ratios of \( B_{\text{max}} = 2 \) at \( Re_{Cx} = 25000 \) and \( B_{\text{max}} = 1 \) at \( Re_{Cx} = 50000 \). In the case of unsteady VGJs, blowing ratio is defined as the ratio of the maximum jet exit velocity to the local freestream velocity.

Since the turbine blades could not be removed for an undisturbed boundary layer measurement, the boundary layer was measured mid-pitch in the inlet plane. The characteristic length for the suction rate equation is taken to be the max thickness of the turbine \((T = 65 \text{ mm})\). Similar to Sabatino and Smith [2], max thickness (see Fig. 4.1) was defined as the maximum distance between a normal to the true chord and the suction surface. Maximum airfoil thickness was chosen because it was deemed an appropriate representation of the width of the fluid area that interacts with the turbine. The suction rate equation is provided in Eq. 4-1 below. The uncertainty in suction rate was calculated to be \( \pm 1\% \).

\[
SR = \frac{\dot{m}_s}{\rho U_{in} T (\delta_{99} - \delta^*)} \times 100\% \tag{Eq. 4-1}
\]

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Oil flow visualization was performed on the endwall of the turbine passage following the process described in chapter 2. In order to keep the turbine passage clean, white contact paper was placed on the turbine splitter plate. After application of the oil/pigment solution, the plate was placed in the test section and the wind tunnel was turned on. The resultant flow field was captured with a standard digital camera.

The oil flow visualization results were used to obtain a qualitative assessment of the turbine endwall flow before the suction slots were added. Figure 4.2 contains an image of the center passage of the turbine endwall at \( Re_{C_1} = 50000 \). The black turbine blades protrude from the white contact paper-covered endwall. The leading edge and suction surface of the inner turbine blade are visible on the left side of the image. The outer turbine blade leading edge and pressure surface are clearly visible on the right. The dashed lines represent critical lines in the flow field. They were positioned with the aid of the streaks left behind by the pigment.

The streaks show the trajectory of the main flow features in the endwall including the PV, the three-dimensional separation region, and the suction and pressure side legs of the horseshoe vortex. The saddle point of the outer turbine blade is located upstream and slightly to the left of the turbine leading edge. This topological feature is the genesis of the PV and HV and a juncture of flow separation. The flow on the right side of the dividing streamline rolls into the suction side of the outer blade horseshoe vortex. The flow on the left side of the dividing streamline forms the PSHV, which is subsequently engulfed by the PV.

The cross passage migration of the PV and the suction surface three-dimensional separation are also visible. It is interesting to note that the upstream inception of the three-dimensional separation occurs where the suction side HV separation line meets the endwall/suction surface juncture instead of where the PV meets the suction surface. This was seen previously by Gbadebo.
et al. in their study on compressor endwall flow topology [62]. The three-dimensional separation is characterized by lower shear stresses. Consequently, the ink toner accumulates in this region.

![Diagram showing oil flow visualization of the inlet endwall of the turbine passage.](image)

**Fig. 4.2** Oil flow visualization of the inlet endwall of the turbine passage. The dashed lines represent critical lines. \( \text{Re}_{c_2} = 50000 \).

These topological features and the results of the cylinder study were used to position the suction slots (the black arcs directly in front of the turbine leading edges in Fig. 4.2). The slots were positioned midway between the saddle point and the turbine blade leading edges. Ideally, the suction slots would have been placed directly upstream of the leading edges to minimize the impact of incidence angle on the flow control scheme. Unfortunately, this was not possible due to the endwall/turbine fillets. The fillets were not included in the oil flow visualization study, but in the final configuration the 10 mm radius fillets extend to the suction slots. The slots also extend into the turbine passages on both sides of the turbine blades. The slots were constructed to alleviate the pressure gradients on the characteristic length (maximum blade thickness, \( T \)) of the flow field. Similar to the cylinder study, the nominal length of the suction slot was half of the characteristic length (\( d \) or \( T \)).
C. Inlet Total Pressure Surveys

Total pressure surveys in the inlet plane endwall region of the turbine passage illustrate the impact of suction on the horseshoe vortex. The inlet surveys were collected with a grid spacing of 5 mm increments in the pitch and span directions. Pressure data were collected across the entire pitch (145 mm) and about 6% of the total span. Figure 4.3 contains normalized total pressure data without leading edge suction (left) and with 11% leading edge suction (right) at $Re_{C*}=25000$. The contour plots are presented from a downstream (looking upstream) frame of reference. The leading edges of the inner and outer turbine blades are 0% and 100% pitch respectively. Due to geometry limitations the nearest wall proximity to the inner turbine blade leading edge was ~6% pitch. Data are not presented across the full extent of the inlet plane given that the HV only resides in small localized areas. The focus of the data window presented in Fig. 4.3 is the SSHV. The impact of the SSHV is clearly evident in the no suction total pressure data near 10% pitch. The SSHV collects high loss fluid from the boundary layer which results in elevated total pressure loss. When suction is applied at 11% the total pressure loss core is eliminated from the inlet plane. It can also be inferred that the SSHV is likewise eliminated. Similar results were also obtained at $Re_{C*}=50000$ and $SR=11%$.

![Fig. 4.3 Inlet plane total pressure loss surveys with (right) and without (left) leading edge suction at $Re_{C*}=25000$. The inner and outer blade leading edges represent 0% and 100% Pitch respectively. $(P_{Tin}-P_r)/(P_{Tin}-P_r)_{SR=0%}$](image-url)
D. Wake Total Pressure Surveys

Wake total pressure losses were also collected 30% $C_x$ downstream of the turbine blades at $Re_{C_x}=25000$ and 50000. Wake surveys were collected with a grid spacing of 5 mm increments in the pitch and span directions. Pressure data were collected across the entire pitch (145 mm) and almost half the total span (~150 mm). Figure 4.4 contains a schematic that depicts the location and frame of reference for the wake total pressure data relative to the turbine cascade. The contour plots are presented from a downstream (looking upstream) frame of reference.

![Schematic depicting the frame of reference of the wake total pressure loss contour plots. The contour plot contains total pressure loss results at $Re_{C_x}=50000$, $B=0$, $SR=0$.](image)

At $Re_{C_x}=25000$, the suction surface of the turbine exhibits a massive separation. Figure 4.5 contains contour plots of the normalized total pressure losses with (right) and without (left) unsteady VGJ actuation at $B_{max}=2$ and no endwall suction ($SR=0$). The no VGJ plot shows significant total pressure loss particularly in the spanwise separation region. The total pressure
losses are highest near midspan (loss near 4) and decrease in the near wall region where the PV resides. The presence of the PV reduces the spanwise extent of the two-dimensional blade separation as a result of momentum transfer due to the rotation of the vortex. When the unsteady VGJs are activated ($B_{\text{max}}=2$), there is a significant change in the downstream total pressure losses (54% decrease) and the wake trajectory. With unsteady VGJs, the two-dimensional blade separation is eliminated or greatly reduced. The wake trajectory shifts toward the right side of the data window. The loss core of the PV also changes magnitude and position. The loss core shifts to the right ~10% pitch and away from the endwall ~5% span. One might suppose that the addition of midspan unsteady VGJ actuation would negatively impact the endwall flow field. The present results show that, in fact, this particular VGJ configuration positively impacts the endwall flow field as seen by a 25% reduction in the magnitude of the loss core (from ~2 to ~1.5).

![Contour plots of the wake total pressure losses with unsteady VGJs and with and without leading](image)

**Fig. 4.5** Total pressure loss wake surveys with (right) and without (left) unsteady VGJ actuation. $Re_{C}=25000$, $B_{\text{max}}=2$, $SR=0$. $\frac{(P_{\text{Tin}}-P_{\text{T}})}{(P_{\text{Tin}}-P_{T})_{SR=0}}$.

When leading edge endwall suction at $SR=11\%$ was included with VGJs, there was an additional reduction of 3% to the total pressure losses (total decrease of 57%). Figure 4.6 contains contour plots of the wake total pressure losses with unsteady VGJs and with and without leading
edge endwall suction. The left image is the wake profile without endwall suction. It is the same as the right image in Fig. 4.5 except that the color scale has been changed from 0-4 to 0-1.5.

![Fig. 4.6 Total pressure loss wake surveys with (right) and without (left) endwall suction. \( \text{Re}_{\text{Cl}}=25000, B_{\text{max}}=2, (P_{\text{Tin}}-P_{T})/ (P_{\text{Tin}}-P_{T})_{SR=0} \).](image)

The application of leading edge endwall suction reduces the area-average total pressure losses by 7% compared with the no suction case \( (B_{\text{max}}=2) \). The loss core becomes more concentrated as suction is applied. At first, a more concentrated core would seem detrimental, but inspection of the spanwise losses in Fig. 4.6 suggests that the PV spanwise influence decreases (as noted by the decrease in \( P_{T} \) losses near 30% span). The migration of the loss core toward the endwall also results in a thinner wake. At these settings \( (B_{\text{max}}=2 \text{ and } SR=11\%) \) the massflow ratio between the VGJs and the suction slots was 1.8:1. Although not truly a zero-net mass flux system, this setup can be adjusted (without a loss of effectiveness) to ensure the mass flow removed through suction is equivalent to the mass flow supplied to the VGJs.

The impact of endwall suction without VGJs was also compared at \( \text{Re}_{\text{Cl}}=25000 \). Figure 4.7 contains contour plots of the total pressure losses with and without endwall suction. The addition of leading edge endwall suction again decreases the area-average total pressure losses by 4% when compared to the no control case. Unlike the previous case, endwall suction appears to
diffuse the $PV$ loss core (weakened the structure). Due to the weakened structure, the midspan losses cover more of the span. The position of the $PV$ loss core does not change.

![Contour plots of normalized total pressure loss in the turbine wake with (right) and without (left) leading edge suction at $SR=11\%$.](image)

Given the small impact of suction at $11\%$, data were also collected at suction rates approaching $20\%$. Similar to the results of the cylinder study, suction rates above $11\%$ had minimal additional impact on total pressure loss reduction. Given the significance of the two-dimensional profile losses compared with the endwall losses, the secondary loss reduction due to leading edge suction was difficult to gauge. An attempt was made to isolate the impact of suction on the total pressure losses by increasing the $Re_{C_s}$ of the flow to near $50000$. At this $Re_{C_s}$, the spanwise flow field does not exhibit boundary layer separation. Consequently, any change in the wake total pressure losses can be directly attributed to leading edge flow control.

Figure 4.8 contains contour plots of normalized total pressure loss in the turbine wake with (right) and without (left) leading edge suction at $SR=11\%$. Suction resulted in a $7\%$ decrease in the area-average total pressure losses. Subtle changes in the shape, magnitude, and position of the total pressure loss core are also noted. Unlike the previous data sets, the loss structures in the
wake at this $Re_{Cs}$ were not as well defined. In this case, the $Re_{Cs}$ did not result in a consistently attached flow field on the turbine blade.

![Figure 4.8 Total pressure loss wake surveys with (right) and without (left) leading edge suction. $Re_{Cs}$=50000, $B_{max}$=0. $(P_{Tin}-P_T)/(P_{Tin}-P_T)_{SR=0}$.](image)

Figure 4.9 shows a plot of the integrated wake loss parameter 24% $C_x$ downstream of the $L1A$ trailing edge plane for a range of $Re_{Cs}$. The definition of the integrated wake loss parameter, $\gamma$, is found in Eq. 4-2 below.

$$\gamma = \int_0^P \frac{P_{Tin}-P_T}{0.5 \rho u_{in}} dy$$  \hspace{1cm} (Eq. 4-2)

Figure 4.9 demonstrates that between $Re_{Cs}$=20000 and $Re_{Cs}$=50000 the $L1A$ exhibits varying degrees of separation. The figure illustrates the proximity of the collected data to a partially separated flow field. The proximity suggests that even subtle variations in the wind tunnel or ambient conditions could cause the flow to separate. This discussion is included because, unlike Fig. 4.8, the subsequent data sets collected at $Re_{Cs}$=50000 do not suggest separation. In fact, a later data set at the same $Re_{Cs}$ and conditions ($SR=0\%, B_{max}=0$) showed a completely attached flow field. The fully attached data set was not used in Fig. 4.8 because the data used in the right
image \((SR=11\%, B_{max}=0)\) was only measured once. The subsequent figures contain comparisons of data sets with the fully attached \((Re_{Cv}=50000, SR=0\%, B_{max}=0)\) total pressure results.

As mentioned in the introduction to this work, the \(HV\) has both suction-side and pressure-side legs in the turbine environment. Each leg interacts with the \(PV\) differently and, consequently, could impact the total pressure losses differently. An attempt was made to gauge the relative impact of each leg on the exit plane total pressure losses by pulling suction from solely the inner or outer blade suction slot. The results are compared with the aforementioned no control data set in Fig. 4.10. The left contour plot is the \(P_T\) loss for the no control data set, the middle plot is the \(P_T\) loss for outer suction slot only, and the right plot is \(P_T\) loss for the inner suction slot only. Caricatures in the corner of the contour plots depict the associated plot’s suction scheme. The suction rates for each data set were matched.

The addition of suction solely to the outer slot results in an area-average total pressure loss reduction of 7\%. The structure maintains a similar shape but moves down toward the endwall. Removal of the pressure side \(HV\) eliminates the leading edge high-loss fluid that is fed into the \(PV\). Without this high-loss fluid addition, the \(PV\) doesn’t reach its uncontrolled size and position.
When suction is applied solely to the inner blade at the same $SR$, the losses decrease by 4%. Contrary to suction from the outer slot, the loss core appears to remain in the same location while still maintaining the same shape. Given the subtle changes between the contour plots of the no control loss core and the inner suction loss core, difference plots were created to show the location of the loss reduction. The difference plots were created by subtracting the no suction total pressure loss results from the outer and inner suction data sets. A positive difference in the plots suggests a decrease in the total pressure losses from the no control case. The results are presented in Fig. 4.11.

![Contour plots of the total pressure losses without endwall suction (left), with only outer blade suction (middle), and with only inner blade suction (right). $Re_{Cl}=50000$, $B=0$.](image)

The difference plot for inner suction (right) shows that the loss reduction does not occur in a concentrated location. Instead, there is a large region of subtle pressure loss reduction that extends from 40% to 60% pitch and from 10% to 30% span. This corroborates Sharma and Butler’s [1] secondary flow model presented in the introduction to this work (Fig. 1.5). In their model, they suggested that the $SSHV$ wrapped itself multiple times around the $PV$ structure by the exit of the turbine passage. This phenomenon would distribute the $HV$-contributed loss around the $PV$-induced loss core. If this is the case, elimination of the $SSHV$ would result in a broad difference plot decrease.
Contrarily, the removal of the pressure side HV results in a concentrated change in the difference plot. The majority of the loss reduction occurs between 40\% and 50\% pitch and 10\% to 20\% span. The loss reduction can be attributed to a combination of structure migration and total pressure loss reduction. All of the secondary flow models depicted in Fig. 1.5 suggested that the PSHV was enveloped by the PV as it entered the turbine passage. The concentrated reduction in the loss core described by the difference plot suggests that this may be the case.

Flow visualization was used to further support the influence of the pressure side HV fluid on the PV. In this case, concentrated olive oil particles were introduced into the wind tunnel directly upstream of the outer blade leading edge. The PIV laser was used to capture the location of the particulate as it exited the turbine passage. The laser sheet was introduced in a plane parallel to the trailing edge plane. Roughly two hundred images were captured and then averaged together to identify regions of particulate concentration. The result was superposed on the no control loss contour plot from Fig. 4.10 along with a caricature showing where the particulate was released (white arrow).
The vertical line in Fig. 4.12 near 35% pitch is the laser reflection on the trailing edge of the outer blade upstream of the laser sheet plane. The figure clearly shows the impact of the passage pressure gradients on the introduced particulate. The pressure gradient sweeps the particulate into the passage vortex which then migrates to the suction surface. The loss core (PV structure) is clearly outlined by the particulate. Unfortunately, when suction was applied to the outer suction slot, the flow visualization did not suggest any impact on the structure.

E. Cascade Horseshoe Vortex Conclusions

Unlike the cylinder study, leading edge suction in the turbine cascade did not eliminate the downstream loss core. It was shown in Fig. 4.3 that the HV was eliminated in the cascade inlet plane with a suction rate of 11%. The continued presence of the majority of the loss structure in the wake (with suction) suggests that the horseshoe vortex is a secondary contributor to endwall loss generation in the turbine environment. Consequently, an effective secondary loss reduction scheme should focus on altering the mechanisms that drive the formation of the PV system as well as the HV system.
Chapter 5: Theoretical Endwall Model

The complexity of the passage vortex system was mentioned previously. Given today’s computational capabilities, it is possible to model fully three-dimensional turbine passages with flow control. However, a project of this magnitude would require an experienced scientist with significant computational resources. Often, these resources are not available or affordable.

In an effort to facilitate the design of a passage vortex flow control scheme, a simple theoretical model was developed. The model requires the solution of the mid-span (two-dimensional) inviscid flow field in the turbine passage rather than the entire viscous three-dimensional flow field. The model tracks endwall boundary layer fluid elements as they enter the turbine passage and are influenced by the passage pressure gradients. The mechanisms that drive the trajectory of the passage vortex are largely inviscid. Thus, the fluid viscosity is neglected and the model assumes that the element’s path is solely dictated by its initial momentum and the forces that the pressure gradients exert on it. Endwall boundary layer fluid elements have a lower initial momentum compared with their freestream counterparts. Consequently, they lack the necessary momentum to follow the mid-span streamlines. Instead, the pressure gradient overcomes the element momentum and pulls the element toward the suction surface side of the turbine passage. The endwall element trajectories are tracked through the passage until they reach the suction surface.

The model assumes that the boundary layer elements remain in the spanwise plane where they originated. In reality, the elements are swept up into the passage vortex which carries them out of the original plane. Although the elements are carried out of their original plane, the model assumes their trajectory is still influenced primarily by the inviscid pressure gradients.
At this point, the model can be used for two different flow control approaches. The first approach ("removal approach") attempts to weaken the vortex after it has already formed. This is done by applying flow control on the endwall along the path of the passage vortex. The second approach ("redirection approach") attempts to alter the trajectory of the vortex. The redirection approach requires the modeling of the $PV$ and the flow control scheme. This is accomplished using potential flow theory. In the case of the turbine passage, the element trajectories are used as a baseline flow field and suction holes are treated as potential flow sinks. Thus, the impact of the suction holes is treated as a planar effect (i.e. no out of plane motion). The model selects suction holes from a user-defined array that “best” align the endwall element trajectory streamlines with the mid-span inviscid streamlines. The model result is then used to create an endwall suction pattern for the wind tunnel.

A. Inviscid CFD

The two-dimensional inviscid flow field through a turbine passage can be easily computed using any computational fluid dynamics (CFD) package. The CFD package used during this study was Fluent™. An unstructured, triangular cell mesh was created using Gambit™ which contained 37,000 cells and covered one turbine passage. The mesh extended upstream of the turbine leading edge ~55% $Cx$ and downstream of the trailing edge ~55% $Cx$. The unstructured mesh and the mesh boundary conditions are provided in Fig. 5.1.

The inlet boundary condition was defined as a velocity inlet with a specified inlet velocity magnitude (5 m/s). The exit boundary condition was treated as an outflow condition and the turbine blade suction and pressure surface boundaries were defined as walls. The remaining boundaries were defined as periodic surfaces. In order to illustrate the position of the CFD domain, the LIA turbine blade shapes are also included in the figure in black. An expanded view of the mesh directly above the suction surface is included at the bottom of the figure for
visualization of the mesh density. After the mesh was generated, it was then exported and loaded into Fluent™.

An inviscid solver was used in Fluent™ to calculate the midspan flow field. The residuals of continuity, x-velocity, and y-velocity are provided in Fig. 5.2 to demonstrate the ease of convergence. In this case, convergence was achieved after 600 iterations and just a few minutes on a single processor. For comparison, a fully three-dimensional viscous L1A calculation would take closer to 8000 iterations or roughly 12 hours on 16 processors [63].

After convergence was achieved, the computational $C_p$ distribution was compared with a high $Re_{cs}$ experimental $C_p$ distribution to benchmark the accuracy of the numerical solution. The experimental data were measured in the L1A linear cascade (described in chapter 2) at
$Re_{Cx}=58,000$. At this $Re_{Cx}$, the LIA is fully attached. The comparison (Fig. 5.3) suggests that the numerical simulation adequately resolves the flow field.

**Fig. 5.2** The residuals of continuity, $x$-velocity, and $y$-velocity for the inviscid Fluent™ calculation.

**Fig. 5.3** $C_p$ comparison between the inviscid Fluent™ calculation and the experimental data.
After the computational $C_p$ results were validated, the static pressure, $u$ and $v$ velocity components, and streamline data were extracted. Unfortunately, the exported data from an unstructured mesh can be very difficult to use. To assuage this problem, line rakes were created in Fluent™ every 5% $Cx$ throughout the turbine passage mesh. The pressure, velocity, and stream function data along the line rakes were exported as text files. Contour plots of the static pressure and velocity magnitude, along with a plot of the inviscid streamlines, are provided in Fig. 5.4. The turbine passage inlet is located on the left side of the images.

Fig. 5.4 Static pressure, velocity magnitude, and streamline results from Fluent™.
**B. Removal Approach**

The element trajectories were predicted using a time-marching Lagrangian approach. The elements were released ~45% $C_x$ upstream of the turbine leading edge plane with a user-defined percentage of the corresponding inviscid freestream velocity. This user-defined velocity percentage corresponds directly with the element’s placement in the boundary layer. A second-order finite difference approximation was made to Newton’s second law in both the pitch and axial directions. In the inviscid flow field, the pressure gradient becomes the sole external force acting on a fluid element. The resultant $u$ and $v$ velocity equations are shown in Eqns. 5-1 and 5-2 below.

$$u^{t+1} = u^t - \Delta t \frac{\rho}{\rho} \left( \frac{P_{x+dx} - P_{x-dx}}{2*dx} \right)$$ \hspace{1cm} \text{Eq. 5-1}

$$v^{t+1} = v^t - \Delta t \frac{\rho}{\rho} \left( \frac{P_{y+dy} - P_{y-dy}}{2*dy} \right)$$ \hspace{1cm} \text{Eq. 5-2}

Using a time step of $t=0.0001$s, the paths of 20 elements (at 0.5$U_{in}$-red and 0.75$U_{in}$-blue) were predicted and then superposed on the inviscid streamlines. The resultant vector fields are provided in Fig. 5.5 along with the inviscid streamlines. To aid in clarity, only every thirtieth vector is included.

A cross-passage migration is clearly evident once the elements enter the turbine passage. The lower initial velocity elements (red vectors) are more susceptible to the influence of the pressure gradients. Consequently, the elements embedded deeper in the boundary layer move more drastically. This is most evident in the leading edge region of the lower turbine blade. Here, the elements actually move away from the turbine blade due to the high stagnation pressure at the leading edge. The elements then gain momentum as the pressure gradient propels them into the
passage. This increase in momentum allows the suction side stagnation region elements to move farther into the passage than their mid-pitch counterparts.

The initial trajectory of some of the leading edge elements leads them directly toward the stagnation region of the outer turbine blade leading edge. The model results show that, near the leading edge of the upper turbine blade, a few element trajectories actually completely stagnate. Once the pressure gradient completely overcomes the element’s axial momentum, the model stops predicting that element’s trajectory.

Figure 5.5 further suggests that the impingement of lower momentum elements on the suction surface occurs over a broader area (25% $Cx$ to 80% $Cx$). As the initial velocity is increased to $0.75U_{in}$ the envelope shifts downstream but also narrows (45% $Cx$ to 90% $Cx$). The overlap of element paths with the same initial velocity is attributed to the two-dimensional modeling of a three-dimensional flow field. In reality, the structure of the passage vortex is highly three-dimensional. Fortunately, modeling the three-dimensional nature of the $PV$ does not appear to be
essential in predicting its general trajectory. This was shown by superposing the element trajectories with a flow visualization image captured in the turbine passage (Fig. 5.6). The similarities between the modeled fluid trajectories and the flow visualization image provided by Gostelow [6] (Fig. 1.4) are also noted.

Once the PV trajectory is known, suction holes in the region of the trajectory can be activated. These suction holes are used to remove low momentum fluid from the PV with the intent of weakening the structure. Application of the removal approach requires the designer to balance the number of suction holes with the total suction available. This study employed the hole configuration shown in Fig. 5.7b. The filled circles were the active suction holes. The remaining holes were covered during the study.
C. Redirection Approach

The focus of the redirection approach is to alter the cross-passage trajectory of the vortex. Ideally, if the PV trajectory were altered, the structure could pass out of the turbine passage at a mid-passage location. Thus, the detrimental impact of the PV’s presence along the suction surface could be eliminated. The redirection approach incorporates a potential flow analysis that models the impact of flow control on the element velocity field. The ultimate objective is to use the endwall flow control to align the boundary layer velocity field with the inviscid velocity field.

Potential Flow Theory

Potential flow theory requires that the flow field be both irrotational and inviscid. After reading the endwall flow field description in the introduction to this work, the reader is well aware that the turbine passage endwall region is highly rotational and dominated by viscosity. Consequently, any attempt at modeling in this region of the passage requires these limitations to be addressed.

This analysis does not attempt to model the highly three-dimensional structure of the PV. Instead, the model is used to track the cross-passage trajectories of the low momentum elements that comprise the vortex. The analysis assumes that the cross-passage migration is solely...
dependent on the passage pressure gradients and is not influenced by the rotation of the vortex. The relative importance of the pressure gradient versus the vortex rotation is evident in flow visualization images of the $PV$. The bulk direction of the fluid is two-dimensional across the passage (pressure induced) while the secondary three-dimensional motion merely causes rotation about the bulk fluid direction. This fluid description is analogous to the shape of a corkscrew. Although the corkscrew shape is highly three-dimensional, the general direction of the screw is longitudinal. Similarly, regardless of its rotation, the $PV$ fluid largely follows the same path. When the three-dimensional rotation of the $PV$ is neglected, the remaining motion can be treated as planar, irrotational motion.

Fluid flow over any surface results in the formation of shear stress-induced boundary layers. In a turbulent channel flow, the wall shear stress has two primary contributors, viscosity and Reynolds stresses. In the regions very near the wall ($y^+<50$), the wall shear stress is dominated by viscosity with minimal contributions from the Reynolds stresses (Fig. 5.8). Outside of this region, the Reynolds stresses become the primary contributor to the overall shear stress. Unfortunately, the model does not account for the contribution of Reynolds stresses. In this case, it is expected that the Reynolds stress contribution is minor given the relatively low $Re_{Cx}$ and freestream turbulence levels of the experimental portion of this study. Prior to direct application of the model

![Fig. 5.8 Profiles of the viscous and Reynolds shear stresses in a turbulent boundary. Image from Pope [64], data from Kim et al. [65].](image)
to a real engine, an assessment of Reynolds stress contribution to the total shear would be advisable.

The redirection approach methodology assumes that the viscous stress contribution is negligible. Figure 5.8 suggests that this assumption is acceptable for $y/\delta > 0.08$ (less than 10% of the total shear). Velocity fractions of $U/U_m=0.75$ and $U/U_m=0.50$ correspond with $y/\delta=0.2$ and $y/\delta=0.04$ respectively (for a turbulent boundary layer). At these velocity fractions, the viscous stress contribution to the total wall shear is 5% and 30% respectively. In order to avoid violation of the inviscid assumption, the remaining model results will use the velocity fraction 0.75.

When velocity is expressed as the gradient of the velocity potential, $\varphi$, the continuity equation takes the form of the Laplace equation, $\nabla^2 \varphi = 0$, for an incompressible, irrotational flow. Since the Laplace equation is linear, the superposition of two distinct solutions becomes a solution itself. Simple fluid flow features are easily modeled using potential functions. These potential functions include uniform freestream flows, line sinks and sources, and free (potential) vortices. These functions are developed so that they also satisfy the Laplace equation. When multiple potential function solutions are combined, it is possible to describe complex flow fields. In the case of the turbine passage, the vector field from Fig. 5.5 is considered to be a baseline potential flow field. Although the vector field, by itself, is not one of the basic building blocks of potential flow theory, it is not unreasonable to imagine the construction of a similar flow field using the potential functions. A small concession is taken with the region of the background flow field that overlaps. It is noted that such a flow feature is not achievable with potential flow functions.

The analysis was performed using the $0.75U_m$ element paths as the background flow field, and the suction holes were modeled as potential flow sinks. The individual impact of each sink on the trajectory of the elements was assessed by calculating the deviation of the angle of each velocity vector in the boundary layer field (with suction applied) from its corresponding velocity vector in the inviscid mid-passage field. The deviations at each location were squared. Then, the entire
deviation field was summed together. The result was compared with the sum of the deviations without flow control and expressed as a percent reduction.

Figure 5.9 (left) contains a plot of each individual hole’s impact on the percent reduction of angle deviation and a schematic that describes the numbering system used (right). It is readily apparent from the percent reduction plot, that some holes reduce the overall angle deviation (e.g. holes 1 and 2) while other holes aggravate it (e.g. holes 3-9). The red dashed line represents the cutoff if the most influential twenty two holes were selected. The model would then select the holes that provided a percent reduction above the red dashed line for the potential flow analysis. A major caveat of this method is that it neglects inter-hole interactions. Although not considered in this work, this issue could be addressed with an optimization scheme.

![Figure 5.9 Plot of each hole’s impact on the percent reduction of angle deviation (left) and the hole numbering sequence (right). The red dashed line represents the cutoff if the most influential twenty two holes are selected.](image)

After an assessment was made of the individual hole’s impact on the angle deviation, an assessment was made of the percent reduction of angle deviation as a function of the number of user-selected active holes and the total suction rate. As described previously, when twenty two holes are activated, the model selects the twenty two, independently, most influential holes for a
given suction rate. Then, the model calculates a percent reduction in angle deviation employing those holes collectively. The collective results are provided in Fig. 5.10 for a range of the number of active holes and suction rates. The potential flow suction rate, $\Gamma$, is defined as the ratio of the total volumetric suction flow rate and a characteristic length. In this case the characteristic length was chosen to be the height of the passage inlet boundary layer from the wall to 0.75$U_{in}$. This height was chosen because the sinks are employed in an effort to remove the low-momentum fluid below 0.75$U_{in}$. As a comparison between the two suction designations, $\Gamma=0.7$ is equivalent to $SR=29\%$.

According to the model results in Fig. 5.10, almost a 65\% realignment of the $PV$ vectors is possible for suction rates greater than 0.7 with at least 22 active holes. Figure 5.10 also shows that as the suction rate increases the trend of the number of active holes (to maximize the percent reduction) also increases. This result is not wholly unexpected since as suction rate increases so

![Fig. 5.10](image-url)
too does the extent of influence of each individual hole. The results for less than ten suction holes are not included to avoid violation of the assumption that the potential flow sinks only remove the lower-momentum boundary layer fluid. If a high suction rate with only a few holes were employed, it is more likely that fluid would also be removed outside of the boundary layer.

Figure 5.7c contains a schematic of the hole pattern suggested by the theoretical model for a user input of twenty two holes. This value (22) was selected in an attempt to cover the optimum reduction for the lower range of suction rates ($\Gamma=0.1$ to $\Gamma=1.0$) found in Fig. 5.10. Figure 5.11 shows the velocity vector field results for the redirection approach with twenty two active suction holes and $\Gamma=0.7$. The figure includes the baseline velocity field (top) and a controlled velocity

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**Fig. 5.11** Theoretical model velocity vector fields with and without endwall control (redirection approach).
field (bottom). Each of the images includes an expanded view to aid in visualization of the vector field. As expected, the influence of the flow sinks realigns the boundary layer velocity field with the inviscid streamlines.

The results of the redirection approach were validated using two-dimensional particle image velocimetry at $Re_C_x=25000$. A laser sheet was introduced into the wind tunnel from an upstream location. The laser sheet was positioned in the boundary layer about 1% of the total span from the endwall. The camera was positioned above the test section. It captured two hundred images of the endwall flow field with and without endwall suction. These images were then used to calculate the average vector field using 16x16 interrogation windows with 50% overlap. As a qualitative gauge of the flow control effectiveness and the model prediction accuracy, the trajectories of the average $PIV$ velocity vectors were superposed on the prediction. The comparison is found in Fig. 5.12. The blue and red vectors are the model and experimental data velocity vectors respectively. Again, the open and filled circles represent unused and used suction holes respectively.

The $PIV$ data were measured at approximately $0.80U_{in}$ instead of $0.75U_{in}$ due to the presence of near wall glare in the lower regions of the boundary layer. Consequently, the cross passage migration occurs slightly farther downstream than the model prediction when $\Gamma=0$. Regardless,
the *PIV* data show a significant cross-passage migration with an uncontrolled endwall. When suction is applied the *PIV* velocity vectors align themselves with the model prediction and the inviscid streamlines demonstrating that locally the *PV* trajectory has been altered.

**D. Leading Edge Suction Modeling**

The model’s stagnation region depicted in Fig. 5.5 corresponds with the stagnation region of the *HV* system seen in the oil flow visualization image provided in Fig. 4.2. It was shown experimentally in chapter 4 that suction in this region did not significantly alter the exit plane total pressure losses. The redirection approach was also used to model the impact of leading edge suction on the trajectory of the *PV* system. Outer blade leading edge suction was modeled using a cluster of twenty flow sinks. The suction mass flow rate of the model was matched with the flow rate required to provide air to the VGJs at $B_{max}=2$. The inner slot was not included in the study since the *PT* loss data in chapter 4 showed that the *PV* was unresponsive to it.

The results of this modeling attempt are provided in Fig. 5.13. The left image contains the baseline velocity vector field without leading edge suction. An expanded view of the mid-passage *PV* velocity vectors is included to aid in visualization. The right image contains the velocity vectors that were modified by the suction slot potential flow sinks. A comparison of the expanded

![Fig. 5.13 The *PV* velocity vectors with (right) and without (left) leading edge suction.](image)
views shows that leading edge suction significantly alters the velocity vectors in close proximity to the suction slot, while the vectors far from the suction slot remain relatively unchanged. In the end, the ineffectiveness of leading edge suction is attributed primarily to the relatively localized impact of the suction slot. Unlike the passage suction scheme, leading edge suction only significantly altered 2 or 3 trajectories (aside from the two trajectories that completely stagnate). This corresponds with only 11% to 17% of the trajectories that form the PV. In contrast, the redirection passage suction scheme (Fig. 5.12) significantly impacts all of the velocity vectors that form the PV.
Chapter 6: Passage Vortex Flow Control Scheme

A. Experimental Facility

This portion of the study used the turbine cascade mentioned in chapter 2 and shown in Fig. 6.1. Similar to the HV study, a splitter plate was added to the test section to house the suction holes. The endwall suction holes are not included in the figure to minimize clutter.

![Schematic of the LIA cascade.](image)

Fig. 6.1 Schematic of the LIA cascade.

Unlike the HV study, steady VGJs were employed. In reality, a steady configuration is more conducive (and realistic) to a global scheme. Given that suction occurs continuously, unsteady VGJs would require a storage plenum to house the removed mass flow until VGJ actuation. If unsteady jets were employed, the complete system would be more complicated because of the storage needs. The steady VGJs had a $B=2$ and were injected in the positive $z$ direction (inclined toward the endwall, see Fig. 6.1).
Similar to the horseshoe vortex study, a splitter plate was placed 20 mm above the base plate of the test section to house the suction scheme. An array of 3.2 mm suction holes was drilled into the splitter plate in the configuration found in Fig. 5.7a and repeated below in Fig. 6.2a. The hole inlets were rounded to reduce the minor losses associated with an entrance region. The suction holes lead to a chamber connected to a vacuum system with variable suction rates. During testing, the unused suction holes were covered with a continuous sheet of contact paper. The endwall/blade juncture fillet prevented the placement of holes closer to the turbine blades.

![Diagram of hole patterns](image)

**Fig. 6.2 Schematics of splitter plate hole pattern (a), and the removal (b) and redirection (c) approach hole patterns.**

A variation of the previous suction rate definition was used. In this case, suction rate ($SR$) is defined as the percentage of the mass flow rate of the impinging boundary layer fluid that is removed by suction. Equation 6-1 is the mathematical definition. Unlike the $HV$ study, the characteristic length is taken to be the pitch ($p$) of the $LIA$ (145 mm). This was chosen because it represents the width of the fluid area that contributes to the $PV$ formation. The uncertainty in suction rate was calculated to be ±1%. The boundary layer parameters of Eq. 6-1 were measured mid-pitch in the turbine inlet plane.

\[
SR = \frac{\dot{m}_2}{\rho U_{in} p(\delta_{99} - \delta^*)} * 100\%
\]  

(Eq. 6-1)
B. Total Pressure and Two-Dimensional PIV Results

Total pressure loss wake surveys were collected 35% Cx downstream of the turbine blades at $Re_{C_x}=25000$ and 50000. Similar to the horseshoe vortex wake surveys, the total pressure losses were collected with a grid spacing of 5 mm increments across the entire pitch and about half the total span.

At $Re_{C_x}=25000$, the suction surface of the turbine exhibits a massive separation. Figure 6.3 contains contour plots of the normalized total pressure losses with (right) and without (left) steady $VGJ$ actuation and no endwall suction ($SR=0$). The contour plots are presented from a downstream (looking upstream) frame of reference similar to the schematic in Fig. 4.4. The no $VGJ$ plot shows significant $P_T$ loss particularly in the spanwise separation region. The $P_T$ losses are highest near midspan (loss near 4) and decrease in the endwall region where the $PV$ resides.

When the steady $VGJ$s are activated ($B=2$), there is a significant change in the downstream $P_T$ losses (68% decrease) and the wake trajectory. With steady $VGJ$s, the two-dimensional blade separation is eliminated or greatly reduced. The wake trajectory shifts toward the right side of the data window. The loss core of the $PV$ also changes magnitude and position. The loss core shifts to

![Fig. 6.3 Total pressure loss wake surveys with (right) and without (left) steady $VGJ$ actuation. $Re_{C_x}=25000$, $SR=0$. $(P_{T_{in}}-P_T)/(P_{T_{in}}-P_{Sim})$.](image-url)
the left ~10% pitch and toward the endwall ~5% span.

Total pressure measurements were obtained for the redirection and removal approaches at a suction rate sufficient to supply the VGJs at $B=2$ ($Re_{C_x}=25000$). The VGJ mass flow requirements were estimated for a turbine aspect ratio of four with a VGJ diameter of 2.6 mm and spacing of 10$d$. Since a turbine blade has two endwalls (casing and hub), each endwall must provide the necessary VGJ massflow for half the total span. This amounts to the removal of approximately 9.7% of the approaching endwall boundary layer fluid for $B=2$ at $Re_{C_x}=25000$. Contour plots of the wake surveys are provided in Fig. 6.4. The figure also includes a contour of steady VGJ actuation at $B=2$ (same as Fig. 6.1-right) for comparison. To aid in visualization, the color range was reduced from 0-4 (Fig. 6.3) to 0-1.5.

The bottom left contour plot in Fig. 6.4 contains the removal approach results at $SR=9.7\%$ and $B=2$. With these conditions the $P_T$ losses decrease by approximately 5% when compared to the $SR=0$, $B=2$ case and 69% when compared to the uncontrolled flow field ($SR=0$, $B=0$). The magnitude of the loss core decreases as well as the size of the structure. The loss core remains near 8% span but moves to the right ~5% pitch. The redirection approach (bottom right) had a similar outcome, resulting in a $P_T$ decrease of only 1% when compared to the $B=2$, $SR=0$ case and 68% when compared to the $B=0$, $SR=0$ case. The magnitude of the loss core is similar to the baseline case without suction. The loss core with the redirection approach resides closer to the endwall. When compared with the impact of the VGJs on the total pressure losses, the effect of endwall suction with either suction approach is minimal.

The vertical position of the vortex structure is an important consideration when gauging the effectiveness of a flow control approach. It was mentioned previously that the presence of the PV along the suction surface results in a three-dimensional separation, impeding lift production. Although the redirection approach at $SR=9.7\%$ proved to have more $P_T$ losses than the removal approach, the position of the structure nearer the endwall increases the available span.
The undulations evident in the mid-span wakes of Fig. 6.4 are caused by the steady VGJs. As mentioned earlier, steady VGJs promote mixing by the creation of streamwise vorticity and boundary layer transition. The vortical structures pull high-momentum fluid into the near-wall region while dragging low-momentum fluid into the freestream. The result is a broad wake with pockets of higher loss fluid downstream of the VGJ injection sites. These structures contribute to the overall loss measurement. As could be expected, the losses associated with streamwise vorticity increase as the blowing ratio increases for an attached flow. Thus, the designer could minimize the profile losses by employing lower blowing ratios while still maintaining a separation free flow field. At higher \( Re_{C_x} \), the VGJs are not needed for boundary layer
reattachment on the $L_{IA}$. Consequently, they have a detrimental impact on the total pressure losses.

In light of this, an assessment was made of the impact of endwall suction at an attached flow $Re_{C_x}$ without VGJs. In order to aid in the discussion, two-dimensional $PIV$ was obtained at six quasi wall-normal locations. The locations are identified in the schematic provided in Fig. 6.5 (labeled positions 1 through 6). The axial chord locations of the first four positions are also labeled in the schematic. Each $PIV$ plane contains five black dots representing the $y/C_x$ position labels that will be used in the subsequent $PIV$ plots. Each of these positions is labeled adjacent to the line representing $PIV$ position 2. These same numeric values correspond to the dots of the

![Fig. 6.5 Schematic of the relative positions of the wall-normal $PIV$ data (black dashed lines labeled Pos.1 through Pos. 6) and the total pressure surveys (vertical red line).]
other PIV planes in the schematic.

The PIV camera was positioned downstream of the turbine passage. The laser projected a sheet normal to the aft portion of the blade surface in the y-z plane (see schematic for coordinate system). It projected a 2 mm laser sheet with a time delay between laser pulses of 100 µs. At position 1, the camera was not able to capture the near wall region due to the blockage of the turbine blade. Consequently, the “zero” location for position 1 is actually 5 mm off the wall. PIV planes 5 and 6 are located 20 and 50 mm downstream of the trailing edge and are parallel to plane 4. The dashed line that extends downstream from the trailing edge of the turbine blade is the line of sight of the camera. It defines the zero location for PIV positions 5 and 6.

The red vertical line is the location (35% Cx downstream of trailing edge plane) of the total pressure surveys. The labels 0% through 100%, adjacent to the red line, correspond with the percent pitch labels on the abscissa of the total pressure contour plots. Although the PIV and PT data planes were not directly aligned, comparing the position of the loss cores and the vortex structures that form them is insightful.

Figure 6.6 contains normalized PIV vorticity (ωx · Cx/Ut) contour plots at ReCx=50000, SR=0%, and B=0 for all six wall-normal positions. Black velocity vectors are added to further describe the secondary flow field. The ordinate axis (expressed as %Span) is the spanwise direction where 0% Span is the endwall. The abscissa is normalized by the axial chord (y/Cx) and, for positions 2 through 4, y/Cx=0 represents the turbine blade wall.

At position 1, two distinct vortex structures are clearly visible. Both structures share the same rotational sense and, consequently, must be the PV (left) and CV (right). The PV resides near y/Cx=0.2 and 2% span, while the CV resides adjacent to the turbine blade surface. At position 1 there is indication that the CV has climbed up the suction surface. By position 1, the CV has traveled about ~50% of the total suction surface length. As the PV approaches, the CV begins to
rapidly climb the blade surface. The rotational sense of the $PV$ imparts momentum to the $CV$ forcing it to move more quickly up the span. This interaction continues through position 6.

By position 4, the passage vortex has pushed the $CV$ up the span. The $PV$ now resides below the $CV$ adjacent to the base of the turbine blade. It resides in the endwall from 0 to 5% span. The $CV$ extends from 5 to 15% span showing that, in this case, the $CV$ presence is responsible for a majority of the three-dimensional separation. Farther downstream (beyond the trailing edge), the

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**Fig. 6.6** Normalized vorticity ($\omega_x \cdot Cx/U_m$) contours of the wall-normal PIV planes. The secondary velocity field is represented with the black vectors. $Re_{C_x}$=50000, $SR$=0%, $B$=0.
PV diffuses increasing its spanwise presence. Simultaneously, the CV’s spanwise presence decreases.

Beginning near position 3, a negative vortex structure appears above the CV. The rotational sense of this vortex suggests that it could be the SSHV. However, if it were the SSHV, it would have been present in the earlier data planes. Instead, the negative vorticity is attributed to trailing shed vorticity [26, 66]. Trailing shed vorticity occurs due to the influence of the CV/PV on the bound vorticity of the suction surface boundary layers. The rotation of the CV/PV lifts the bound vortex off the turbine blade. As the vortex is lifted into the freestream, it is turned into the streamwise direction.

The initial trailing shed vortex (pos. 3) diffuses between positions 4 and 5. Once the CV/PV move beyond the trailing edge (between positions 4 and 5), a new negative vortex structure appears on the right side of the CV/PV. This structure has a similar vorticity magnitude to the CV and PV. At position 5, the magnitude is exaggerated due to the presence of glare in the seed images, but the glare was minimized by position 6 and had no impact on the results. This vortex is attributed to the reorientation of vortex rollers (TSV) shed from the trailing edge of the L1A.

Regardless of the origin of the TSV, its presence plays a significant role in the relative movement of the three vortices. This is evident in the change of the velocity vectors between PIV positions 4 through 6. At position 4, the velocity vectors on the right side of the PV move up the span of the turbine blade. At position 5, the TSV appears and the direction of the vectors changes drastically. Although the velocity vector magnitudes may be exaggerated, the general trend is consistent with the evolution of the vectors from position 4 to 6. By position 6, the mutual induction of the PV and the TSV push the vortices toward the upper left side of the image. The PV is more concentrated and has a higher vorticity magnitude. The result is a more rapid lateral movement of the CV and TSV and only a subtle movement of the PV.
The relative movement of the vortices results in the shape of the total pressure loss data. Figure 6.7 contains contour plots of the $P_T$ loss in the wake at $Re_C=50000$. The top plot in the figure is the $P_T$ wake losses without suction or blowing. The total pressure loss core is positioned near 55% pitch and 12% span. A comparison of this spanwise position with the position of the vortices in Fig. 6.6 (pos. 6) shows that the $PV$ resides below the loss core. Its rotations accounts for the endwall $P_T$ loss deposit near 80% pitch and the deformation of the loss structure. As it rotates it also pushes the loss core toward the left of the $P_T$ plot. The position of the $PV$ below the loss core results in the slight increase in the $P_T$ loss between 40 and 60% pitch and 0 and 5% span. Although this region of loss is relatively insignificant, its presence identifies the $PV$’s

![Fig. 6.7 Total pressure loss wake surveys. Baseline (top), removal approach (bottom left), and redirection approach (bottom right). $Re_C=50000$, $SR=11.3\%$, $B=0$. ($P_{Tail}-P_T$)/($P_{Tail}-P_{Sin}$).](image)
Referencing Figs. 6.5-6.7, the loss core of the $P_T$ data resides near the intersection of the $P_T$ data plane and the PIV data at pos. 6. Since the intersection is near $y/C_x=0.2$ in the vorticity plots and 55% in the $P_T$ contour plot, the position of the loss core is in the region of the intersection of the PV, CV, and TSV. This implies that either the vortices are depositing the high loss fluid in this region or the vortex interaction is a significant contributor to the total pressure losses.

When suction is applied using the removal approach at $SR=11.3\%$ (equivalent to the massflow needed for $B=2$ at $Re_{C_x}=50000$) the $P_T$ loss core moves to the right ~10\% pitch and up ~5\% span. The loss structure also changes shape when suction is applied. The structure is elongated and aligns itself with the profile loss structure. In the end, the area-average $P_T$ loss reduction using the removal approach is minimal (4\%) at $SR=11.3\%$. This may be attributed to the increase in near-wall losses below and to the left of the loss core.

The increased near wall losses suggest that the loss associated with the PV has increased due to the suction. Figure 6.8 contains contour plots of the normalized vorticity at all six PIV locations with the removal approach at $SR=11.3\%$. The increase in the near wall $P_T$ losses in Fig. 6.7 (60-80\% pitch and <5\% span) can be explained by the more coherent and concentrated PV structure in the data at position 6. A comparison of the PV structures without control (Fig. 6.6) shows that the removal approach actually increases the size and magnitude of the PV. At the same time, the new PV structure resides farther from the suction surface. By position 4 (near the trailing edge), the structure resides about 0.15 $y/C_x$ farther away from the suction surface than the no control case (Fig. 6.6). As a consequence, the PV is not as effective at moving the loss core laterally (see Fig. 6.7), leaving the loss core closer to the pitch location of the profile losses.
When suction is applied with the removal approach, the CV is completely eliminated. Consequently, there is not a vortex climbing the suction surface wall reducing the available span of the turbine. A comparison of the vorticity contours at position 4 with and without endwall suction suggests that the available span at the trailing edge has been increased at least 10%.

Although the removal approach did not eliminate the vortex at this suction rate, there are

Fig. 6.8 Normalized vorticity ($\omega_x \cdot C_x/U_m$) contours of the wall-normal PIV planes. The secondary velocity field is represented with the black vectors. Removal approach, $Re_{C_x}=50000$, $SR=11.3\%$, $B=0$.  

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significant benefits to the newly formed structure. Aside from its increased size and magnitude, its mid passage position is much more desirable. The PV position suggests that the vortex forms deeper into the passage. The schematics in Fig. 6.9 offer a potential explanation. The uncontrolled PV is represented in the left image by the red arrow. Without endwall suction, the PV quickly moves across the passage and impinges on the suction surface near the peak $C_p$. When suction is applied with the removal approach, the suction holes remove the low momentum fluid of the inlet boundary layer. At the same time, the suction holes reduce the momentum of the remaining boundary layer fluid. This fluid still has enough momentum to pass over top of the suction holes (middle schematic), but beyond the suction holes, the higher momentum fluid rolls over the lower momentum fluid creating a delayed PV. This PV forms beyond the influence of the pressure gradient so it exits mid passage.

When suction is applied with the redirection approach at $SR=11.3\%$, the total pressure losses are reduced by 13%. In contrast to the removal approach, the loss core retains its shape while decreasing in size. The loss core migrates to the right $\sim5\%$ pitch and down slightly. Unlike the removal approach at $SR=11.3\%$, the redirection approach moves the loss core down toward the boundary layer.

![Fig. 6.9 Schematic describing the altered position of the PV when suction is applied with the removal approach.](image)

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endwall suggesting either a weaker structure or a decrease in the residence time of the vortex along the suction surface.

Figure 6.10 contains the normalized vorticity contours obtained while applying suction with the redirection approach. In this case, the PV does not have a concentrated core like the no control and removal flow fields. Instead there is a vorticity band that extends along the endwall the length of the data window. The CV is similar in size and magnitude to that of the no control case. This is
not wholly unexpected since the flow control is applied on the other side of the passage. Similar to the no control flow field, the CV climbs the suction surface wall with the aid of the PV. Since the PV is weaker, the CV is not able to climb as rapidly up the blade span. In the no control case

![Diagram of flow control](image)

**Fig. 6.11** Schematic describing the modified PV when suction is applied with the redirection approach.

the CV resided from 5 to 15% span (Fig. 6.7, pos. 4). In this case, the CV structure is closer to the endwall, residing from 3 to 12% span. Although not as effective as the removal approach, the redirection approach also increases the available span. Similar to the no control case, the CV/PV presence on the suction surface redirects the bound vorticity on the turbine blade resulting in the TSV. The shape of the total pressure losses reflects the presence of all three vortices. Again, the loss core resides above the PV in the interaction area between the three vortices.

The similarity between the location of the redirection PV structures and the no control structures suggests that the redirection was ineffective in its objective. The results of the potential flow model and accompanying PIV validation showed that in a localized area the PV was redirected, but similar to the removal scheme, unexpected consequences of the endwall suction arose. Figure 6.11 contains a schematic of a potential explanation for the altered vortex. Again, the left image is the no control PV position. When suction is applied with the redirection approach, the boundary layer in proximity to the suction surface is unaffected by the suction holes
resulting in the creation of streamwise vorticity. Similar to the removal approach discussion, in the suction vicinity low momentum fluid from the boundary layer passing near the holes is removed. The remaining boundary layer fluid passes over top of the control region losing some of its momentum to the suction holes. After the fluid passes over the suction holes, the higher momentum fluid rolls over the remaining lower momentum fluid resulting in the creation of vorticity (see middle schematic). The orientation of the suction holes does not encourage the vorticity to coalesce into one large, concentrated vortex. Instead, the broad band of vorticity seen in the PIV data results.

C. Increased Suction Rate Results

The losses at $Re_{c_x}=25000$ and 50000 were also measured with higher suction rates. Although these suction rates would result in an excess of fluid for the VGJs at $B=2$, if significant gains can be achieved, the fluid could be used elsewhere. For example, the designer could produce more aggressive blades that require multiple VGJ rows for attached flow. These designs have the potential for significant work output per stage.

Figure 6.12 contains the total pressure loss results for both suction schemes over the full range of suction rates collected. The upper series of plots correspond to the removal approach for suction rates 0, 11.3, 24.4, 32.5, 40.6, and 48.7% (moving left to right). The data extends from 40% to 90% pitch and includes 0% to 40% span. Only half of the total pitch is included in the figure so that the structures are more clearly visible. The removal approach effectively eliminates the loss core as suction is increased. Each successive suction rate results in a decrease in the size of the structure. The lateral movement of the structures away from the pitch position of the profile loss also decreases. By $SR=48.7\%$, there is little left of the loss core. In contrast, the redirection approach reduces the magnitude of the losses between $SR=0\%$ and $SR=24.4\%$, while greater suction rates provide little additional benefit.
Figure 6.13 quantifies the changes to the loss structures described in Fig. 6.12. The plot contains normalized $P_T$ loss data at $Re_{C_f}=25000$ and 50000 for the full range of suction rates and both flow control approaches. At $Re_{C_f}=25000$ the VGJs were employed at $B=2$ while at $Re_{C_f}=50000$ the VGJs remain inactive ($B=0$). Application of the removal approach at $Re_{C_f}=25000$ with a range of suction rates from ~9 to ~84% results in a maximum decrease in the normalized area-average total pressure losses of about 7%. In fact, the loss reduction asymptotes to 7% by $SR=32\%$. As suction is further increased, the increases in the normalized total pressure loss reductions are minimal. A similar trend is seen with the redirection approach where the losses asymptote near 8%.
In contrast to the removal approach at \( Re_{C_x}=25000 \), when endwall suction is applied using the removal approach at \( Re_{C_x}=50000 \), the \( P_T \) reductions decrease monotonically over the entire range of suction rates. The maximum reduction of \( \sim28\% \) is achieved with a \( SR \) near 50%. At this point, the total pressure loss wake surveys are dominated by the midspan profile losses. Although \( P_T \) losses with suction rates greater than 50% were not measured, it is anticipated that further reductions would be minimal. This suggests that the maximum achievable reduction in the total pressure losses due to secondary flow mitigation in the endwall is near 30%. In contrast, when the redirection approach is applied at \( Re_{C_x}=50000 \), the loss reduction asymptotes near 15%.

Particle image velocimetry data were also collected at higher suction rates (\( SR=24.4\% \) and 32.5%). The data for both the removal and redirection approaches at \( SR=24.4\% \) are provided in Figs. 6.14 and 6.15 respectively. The data at \( SR=32.5\% \) reflect the trends seen in the \( SR=24.4\% \) data and, consequently, the plots are relegated to the Appendix (Figs. A.1 and A.2).

A comparison of the removal approach at \( SR=11.3\% \) and 24.4% shows that as the suction rate is increased the magnitude and size of the \( PV \) structure decreases. The lateral position of the structure also changes. As suction is increased the \( PV \) resides farther from the suction surface. The structure’s position results in a decrease in the \( TSV \). The two structures still interact at this
suction rate resulting in a loss core, but as the distance between them grows, their ability to interact diminishes (as well as the losses).

The diminishing ability of the redirection approach is attributed to the continual presence of the CV. Regardless of suction rate, the CV continues to reside near the suction surface. It is able to climb the blade surface of its own accord, so the loss of the PV does little more than slow its
ascent. At the same time, the CV also encourages the bound vorticity to form the TSV. Consequently, the two structures interact feeding the loss core.

In summary, both endwall flow control schemes were effective at reducing the total pressure losses associated with the endwall flow field. Both scheme’s effectiveness were attributed to the influence of suction on the structure of the endwall vortex structures. The CV was shown to play a primary role in the three-dimensional separation. When the PV was also present, the two
structures worked together to increase the spanwise coverage of the separation region. The removal approach was more effective at higher $Re_{C_\lambda}$ due to the complete removal of the $CV$ structure and a delayed formation of the $PV$. In contrast, the redirection approach resulted in a diffused $PV$ but had little impact on the size/coherence of the $CV$ structure. This resulted in a diminishing impact at elevated suction rates for the redirection approach. The position of the loss core relative to the endwall vortex structures suggests that the interaction of multiple vortices is either a major source of loss or the interaction region is an effective accumulator of high loss fluid.
Chapter 7: System Analysis

In order for a combined VGJ/suction scheme to even be considered, the benefits of the scheme must outweigh the costs. A system analysis was performed to assess the overall impact of the control scheme at both $Re_{C_x}=25000$ and 50000. The cost was associated with the power required to remove the mass flow and provide the necessary pressure increase to obtain the desired blowing ratio for reinjection. At $Re_{C_x}=25000$, the power gains were assessed by estimating the increase in shaft power with the addition of the VGJs. The power increase is attributed to an increase in the $C_p$ distribution. The impact of suction (at $Re_{C_x}=25000$) on the available span was considered to be secondary and, consequently, was neglected. However, suction did provide a decrease in the wake total pressure losses adding available work to the system. This was quantified and included in the total power gain assessment. In contrast, at $Re_{C_x}=50000$, an increase in usable span (for lift) was considered the primary contributor to the power gain. The increase in usable span results from the removal of the separation region caused by the endwall flow field. A detailed system analysis at each $Re_{C_x}$ follows.

Fig. 7.1 Schematic of suction/VGJ scheme (left) and the coordinate system (right).
**Power ratio estimation for $Re_{C_x}=25000$**

The power required for suction/VGJ flow control is estimated by performing an energy balance on the suction and ejection path of one turbine passage (Fig. 7.1). The energy balance for an incompressible, steady flow field yields Eq. 7-1, where $h_L$ is the head loss, and $h_s$ is the head rise from the pump delivered to the fluid respectively.

$$P_A + \frac{1}{2}\rho U_A^2 + \rho g h_s - \rho g h_L + \rho g z_A = P_B + \frac{1}{2}\rho U_B^2 + \rho g z_B$$  \hspace{1cm} \text{Eq. 7-1}

The impact of elevation is neglected and the density and area of the tubing from position A to B is assumed constant. The result is that the kinetic and potential energy terms are eliminated from Eq. 7-1. The resultant equation, after rearrangement, is found in Eq. 7-2.

$$\rho g h_s = P_B - P_A + \rho g h_L$$  \hspace{1cm} \text{Eq. 7-2}

$P_A$ and $P_B$ can be expressed in terms of $C_p$ and the dynamic inlet pressure using the definition of $C_p$ found in Eq. 7-3 where $P_{S,local}$ is the static pressure at the suction location ($P_A$) or the injection location ($P_B$).

$$C_p = \frac{P_{T, in} - P_{S, local}}{\frac{1}{2}\rho U_{in}^2}$$  \hspace{1cm} \text{Eq. 7-3}

If $C_p$ expressions for $P_A$ and $P_B$ are substituted into Eq. 7-2, Eq. 7-4 results.

$$\rho g h_s = \frac{1}{2}\rho U_{in}^2(Cp_A - Cp_B) + \rho g h_L$$  \hspace{1cm} \text{Eq. 7-4}
The head rise is provided with a pump efficiency, $\eta$. It is expressed in terms of power, $P_{\text{pump}}$, using Eq. 7-5.

$$h_s = \frac{\eta P_{\text{pump}}}{\rho g Q} \quad \text{Eq. 7-5}$$

Head loss, $h_L$, is divided into major and minor losses. Minor loss contributions are assumed at the suction inlet and injection exit with loss coefficients of $K_{Li}$ and $K_{Le}$ respectively. Friction losses also play a significant role where $f$ is the Darcy friction factor, and $L$ and $D$ are the length and diameter of the tubing from position A to B respectively. Combining these loss terms with Eqns. 7-4 and 7-5 yields Eq. 7-6.

$$P_{\text{pump}} = \frac{\rho Q U_{\text{in}}^2}{2 \eta} (Cp_A - Cp_B) + \frac{\rho g Q}{\eta} \left( f \frac{L}{D} \frac{U^2}{2g} + (K_{Li} + K_{Le}) \frac{U_b^2}{2g} \right) \quad \text{Eq. 7-6}$$

Since a constant area duct was assumed, $U$ is equal to the jet exit velocity $U_B$. If $U_{in}$ is factored out of the loss terms and the suction flow rate, $\dot{m}_s$, is substituted in for $\rho Q$, Eq. 7-7 results.

$$P_{\text{pump}} = \frac{\dot{m}_s U_{\text{in}}^2}{2 \eta} \left[ Cp_A - Cp_B + f \frac{L}{D} \frac{U_b^2}{U_{\text{in}}^2} + (K_{Li} + K_{Le}) \frac{U_b^2}{U_{\text{in}}^2} \right] \quad \text{Eq. 7-7}$$

As previously discussed, blowing ratio, $B$, is defined as the ratio of the jet exit velocity to the local freestream velocity near the jet exit. The freestream velocity at the jet location of the L1A is approximately twice the inlet velocity. $U_{\text{in}}$ is replaced by the local freestream velocity, $U_{\text{local}}$. Blowing ratio is then obtained as shown in Eq. 7-8.
The final expression for the required pump power is obtained by combining Eqns 7-7 and 7-8. It is found in Eq. 7-9.

\[ B = \frac{U_B}{U_{local}} = \frac{U_B}{2 \cdot U_{in}} \]  Eq. 7-8

\[ P_{pump} = \frac{\dot{m}_s U_{in}^2}{2 \eta} \left[ Cp_A - Cp_B + 4B^2 \left( \frac{f}{D} + K_{Li} + K_{Le} \right) \right] \]  Eq. 7-9

The benefits of the combined suction/VGJ scheme are realized primarily by an increase in VGJ-induced lift reflected in the Cp plot found in Fig. 7.2. Figure 7.2 provides experimental data for the suction surface of the L1A blade profile with and without VGJ actuation at \( Re_{Cl} = 25000 \). The \( Cp \) of the pressure surface is approximately the same with or without the flow control. Consequently, it does not contribute to the change in lift and is not included in the analysis. A high \( Re_{Cp} \) CFD solution is also provided to show that, even at \( Re_{Cp} = 25000 \), the VGJs at \( b = 2 \) provide sufficient energy to the flow to regain the fully attached distribution. The difference between the attached and separated \( Cp \) profiles is the additional lift gained with the flow control scheme. This lift is used to assess the total power gained per turbine rotor passage.

Historically, any discussion of the benefits of VGJ actuation has been met with skepticism because of the belief that the required mass flow would come from the compressor (reducing the net work output of the cycle). This is not so for the present design since the massflow is obtained locally through endwall suction. Regardless of the power benefits of suction itself, it is important to note that the suction fluid is removed from a high-loss region instead of a high-energy region like the compressor. This allows the high-energy compressor fluid to pass through the combustor and turbine where its energy potential can be more fully utilized. Also, piping the fluid from a
local source could potentially reduce the total pressure loss of the fluid en route to the injection site.

The power gained, $P_{Gain}$, is assessed using Eq. 7-10, where $\Delta F_{Lift}$ is the additional lift gained with the VGJs, $r$ is the mean blade radius, $\Omega$ is the rotation rate of the rotor, and $P_{T,Gain}$ is an estimation of the usable energy that is gained from a reduction in the total pressure losses.

$$P_{Gain} = r\Omega \Delta F_{Lift} + P_{T,Gain}$$  \hspace{1cm} \text{Eq. 7-10}

The gain in lift ($\Delta F_{Lift}$) is obtained by integrating the $C_p$ distributions of Fig. 7.2 using Eq. 7-11, where $S$ is the blade span, $\hat{n}$ is the normal to the turbine blade surface, and $\hat{\phi}$ is the blade pitch direction (see Fig. 7.1).
\[ \Delta F_{lift} = \frac{1}{2} \rho S U_{in}^2 \left( \int C_{PV} \hat{n} \cdot \hat{p} \, dx - \int C_{P_{N_{O_{V_{G}}}}} \hat{n} \cdot \hat{p} \, dx \right) \]  
Eq. 7-11

The mass flow through the passage, \( \dot{m}_p = \rho U_{in,ax} Sp \), is obtained by substituting the axial velocity for \( U_{in} \) (\( U_{in,ax} = U_{in} \times \sin(55^\circ) \)) and multiplying the numerator and denominator by the pitch, \( p \). The first term of Eq. 7-10 becomes Eq. 7-12.

\[ r\Omega \Delta F_{lift} = \frac{\dot{m}_p r U_{in}}{2 p \sin(55^\circ)} \left( \int C_{PV} \hat{n} \cdot \hat{p} \, dx - \int C_{P_{N_{O_{V_{G}}}}} \hat{n} \cdot \hat{p} \, dx \right) \]  
Eq. 7-12

Following the methodology of Greitzer et al. [67], the formulation of \( P_{T,Gain} \) begins by assuming that, through an ideal reversible process, the fluid loss can be restored to its original (low loss) state. Equation 7-13 is for a reversible process at constant total temperature.

\[ q_{rev} = T_T \cdot \Delta s \]  
Eq. 7-13

The change in entropy for an ideal gas with constant specific heat, \( c_p \), can be evaluated with Eq. 7-14.

\[ ds = \frac{c_p \cdot dT_T}{T_T} - \frac{dP_T}{\rho_T T_T} \]  
Eq. 7-14

If the change in internal energy per unit mass \( (de=0) \) for the process is zero, then \( q_{rev} = w_{rev} \), where \( q_{rev} \) and \( w_{rev} \) are the reversible heat and work per unit mass respectively. Combining Eqns. 7-13 and 7-14 yields Eq. 7-15.
If the process has a constant total temperature from state 1 to state 2, Eq. 7-15 reduces to Eq. 7-16. This expression shows that the required reversible work to restore the initial state is entirely dependent on the change in total pressure through the process and the density.

\[
W_{rev} = T_T \left( \frac{cp \cdot dT_T}{T_T} - \frac{dP_T}{\rho_T T_T} \right) 
\]  

Eq. 7-15

For small changes in total pressure, Eq. 7-16 can be approximated as \( W_{rev} = -\Delta P_T / \rho_T \). Multiplying the relation by the mass flow rate of the passage yields the rate of work done (Eq. 7-17) during the process.

\[
W_{rev} = -\frac{\Delta P_T}{\rho_T} \dot{m}_p 
\]  

Eq. 7-17

Since the VGJs decrease the change in total pressure loss from state 1 to state 2, they also reduce the reversible work required to restore the fluid to its original state. If the reversible work is reduced, the system maintains that available work. Essentially, the VGJs (along with increasing the lift) create a higher level of available working fluid that can then provide additional work. The difference in the required reversible rate of work with and without suction/VGJs is the net power gained due to \( P_T \) loss reduction. This change in the rate of work done with and without suction/VGJs can be expressed using mass-average total pressure at the exit of the blade row (Eq. 7-18).
\[ \Delta W = \frac{\Delta P_{TB=2}^{m} - \Delta P_{TB=0}^{m}}{\rho} \dot{m}_p \]  

Eq. 7-18

Since the change in mass-average total pressure loss is a negative value and the absolute value of the change is greater without control, Eq. 7-18 yields the additional power benefit obtained with control. This additional power is available downstream of the controlled stage, and is available to the subsequent stages. Expanding the expression, rearranging, and adding the inlet dynamic pressure to the numerator and denominator yields Eq. 7-19.

\[ P_{T,\text{Gain}} = \Delta W = \frac{(P_{T_{\text{in}}}^{m} - P_{T_{\text{ex}}}^{m})_{B=0} - (P_{T_{\text{in}}}^{m} - P_{T_{\text{ex}}}^{m})_{B=2}}{\frac{1}{2} \rho u_{\text{in}}^2} \cdot \frac{1}{2} \rho u_{\text{in}}^2 \cdot U_{\text{in},a} \cdot S \cdot p \]  

Eq. 7-19

The wake loss parameter that was mentioned in chapter 4 (Eq. 4-2) is modified slightly to reflect the area-average of the total pressure at the exit data plane. The new expression is found in Eq. 7-20 below.

\[ \gamma = \frac{P_{T\text{in}}^{m} - P_{T\text{ex}}^{A}}{\frac{1}{2} \rho u_{\text{in}}^2} \]  

Eq. 7-20

Ideally, the wake loss parameter would utilize the mass-average total pressure data rather than the area-average. Unfortunately, the mass-average total pressure data were not obtained experimentally so the area-average total pressure will instead be used. In order to account for the exaggerated total pressure loss obtained with the area-average, it is assumed that sixty percent of the area-average wake loss parameter is equal to the actual mass-average wake loss parameter.
Combining this assumption and the wake loss parameter with Eq. 7-19 yields Eq. 7-21, which is the final expression for the additional power gained due to the change in \( P_T \) loss.

\[
P_{T,Gain} \equiv 0.6(\gamma_{B=0} - \gamma_{B=2}) \cdot \frac{1}{2} \cdot \rho \frac{U_{in}^2}{S \cdot \rho} \cdot \frac{U_{in,ax}}{m_p}
\]

Eq. 7-21

\( P_{Gain} \) is obtained by combining Eqns. 7-12 and 7-21. It is found in Eq. 7-22.

\[
P_{Gain} = \frac{\dot{m}_r \rho \omega U_{in}}{2 \rho \sin(55^\circ)} \left[ \int \frac{c_{p_B=2} \pi \rho dCx - \int c_{p_B=0} \pi \rho dCx}{2} \right] + 0.6(\gamma_{B=0} - \gamma_{B=2}) \frac{1}{2} U_{in}^2 \dot{m}_p
\]

Eq. 7-22

The ratio of Eqns. 7-9 and 7-22 provide an equation that assesses the amount of the gained power that is used to operate the flow control system. The ratio is provided in Eq. 7-23.

\[
\frac{P_{Pump}}{P_{Gain}} = \frac{\frac{\dot{m}_s U_{in}^2}{2 \eta} \left[ c_{p_A} - c_{p_B} + 4B^2 (f_{L} L_{1} + K_{L1} + K_{L2}) \right]}{\frac{\dot{m}_r \rho \omega U_{in}}{2 \rho \sin(55^\circ)} \left[ \int c_{p_B=2} \pi \rho dCx - \int c_{p_B=0} \pi \rho dCx \right] + 0.6(\gamma_{B=0} - \gamma_{B=2}) \frac{1}{2} U_{in}^2 \dot{m}_p}
\]

Eq. 7-23

Equations 7-24 through 7-26 depict the steps used to simplify to the final power ratio equation (Eq. 7-26).

\[
\frac{P_{Pump}}{P_{Gain}} = \left( \frac{\dot{m}_s}{\dot{m}_p} \right) \frac{U_{in}^2}{\eta} \left[ c_{p_A} - c_{p_B} + 4B^2 (f_{L} L_{1} + K_{L1} + K_{L2}) \right] \left( \frac{\pi \rho dCx - \int c_{p_B=0} \pi \rho dCx}{\rho \sin(55^\circ)} \right) + 0.6(\gamma_{B=0} - \gamma_{B=2}) U_{in}^2
\]

Eq. 7-24
The mass flow ratio, \( \dot{m}_S / \dot{m}_p \), is obtained by considering the mass flow rate of a single VGJ.

The mass flow rate of the passage is obtained by taking a passage slice with a height of 10\( d \) (recall that this is the distance between VGJs). Equations 7-27 through 7-30 contain the steps used to obtain the final mass flow ratio (Eq. 7-30).

\[
\frac{\dot{m}_S}{\dot{m}_p} = \frac{\rho U_{jet} A_{jet}}{\rho U_{in,ax} A_{slice}} \quad \text{Eq. 7-27}
\]

\[
\frac{\dot{m}_S}{\dot{m}_p} = \frac{U_{jet} \pi d_{jet}^2}{U_{in} \sin(55^\circ)(10d_{jet}) p} \quad \text{Eq. 7-28}
\]

Recall from the \( P_{Pump} \) discussion that \( U_{jet} = 2U_{local} = 4U_{in} \).

\[
\frac{\dot{m}_S}{\dot{m}_p} = \frac{4U_{in} \pi d_{jet}}{U_{in} \sin(55^\circ) 10p} \quad \text{Eq. 7-29}
\]

\[
\frac{\dot{m}_S}{\dot{m}_p} = \frac{\pi d_{jet}}{10 \sin(55^\circ) p} \quad \text{Eq. 7-30}
\]
Two pairs of representative $L1A$ values ($Re_{C_x}=25000$) for the variables in Eq. 7-26 are provided in Table 7.1 along with the resultant power ratio.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\frac{\dot{m}_s}{\dot{m}_p}$</td>
<td>0.007</td>
<td>0.007</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>0.85</td>
<td>0.85</td>
</tr>
<tr>
<td>$\sigma = \frac{Cx}{p}$</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>$\eta$</td>
<td>0.85</td>
<td>0.85</td>
</tr>
<tr>
<td>$Cp_A$</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>$Cp_B$</td>
<td>5.5</td>
<td>5.5</td>
</tr>
<tr>
<td>$B$</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>$f$</td>
<td>0.02</td>
<td>0.04</td>
</tr>
<tr>
<td>$\frac{L}{D}$</td>
<td>20</td>
<td>40</td>
</tr>
<tr>
<td>$K_{L_l}$</td>
<td>0.2</td>
<td>0.4</td>
</tr>
<tr>
<td>$K_{L_e}$</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>$\frac{1}{p} \int C_{p_{VGJ}} \hat{n} \cdot \hat{p} dCx$</td>
<td>0.385</td>
<td>0.385</td>
</tr>
<tr>
<td>$- \int C_{p_{NoVGJ}} \hat{n} \cdot \hat{p} dCx$</td>
<td>0.3268</td>
<td>0.3268</td>
</tr>
<tr>
<td>$0.6\varphi (\gamma_{B=0} - \gamma_{B=2})$</td>
<td>0.3268</td>
<td>0.3268</td>
</tr>
<tr>
<td>Power Ratio, $\frac{P_{Pump}}{P_{Gain}}$</td>
<td>0.2272</td>
<td>0.4475</td>
</tr>
</tbody>
</table>

The power ratio results in Table 7.1 suggest that there is a net gain from a combined suction/VGJ flow control scheme. The power ratio obtained with the values in Case 1 was near 23% which means that about 23% of the power gain is used to actuate the flow control scheme.
The power gain formulation shows that removing fluid from the passage flow must be done judiciously. In Case 1 and 2 the removed fluid was 0.7% of the total passage flow rate (the estimated value to maintain $B=2$). If this value had been doubled (for $B=4$) to 1.4%, the power ratio results would also be doubled, significantly reducing the total power gain of the flow control scheme (new Case 1 power ratio = 0.46). It is interesting to note that the contribution of the change in the wake total pressure due to flow control is approximately 46% of the total power gain. This is an important factor that needs to be addressed when assessing power gain in the turbomachinery environment.

Although the analysis to this point has been formulated for the power gain of a rotor, the system analysis can also be applied to a stator. Equation 7-26 is modified to eliminate the influence of the change of lift in the power gain. The modification yields Eq. 7-31 where the power gain of the stator is primarily due to the $VGJ$-induced decrease in the wake total pressure losses (increase in the work potential of the fluid).

\[
\frac{P_{Pump}}{P_{Gain}} = \left( \frac{m_s}{m_p} \right) \frac{1}{\eta} \left[ C_{PA} - C_{PB} + 4B^2 (f_D + K_{Li} + K_{Le}) \right] \frac{0.6(\gamma_{B=0} - \gamma_{B=2})}{\eta} \]  

Eq. 7-31

The stator power ratio yields 0.49 for the conditions provided in Case 1 of Table 7.1. This result is significant given the relative ease of applying a suction/$VGJ$ flow control scheme to a stator compared to a rotor. Control of the exit angle is another additional benefit of the flow control scheme that is not reflected in the power ratio equation. This impacts both the stator and rotor and could allow the designer a tighter tolerance for the incidence angle (yielding more power gain).
**Power ratio estimation for Re_{Cx}=50000**

The equation for pump power is obtained using a similar methodology (to the \( Re_{Cx}=25000 \) analysis) except that at this \( Re_{Cx} \) it is not necessary to use the VGJs. Consequently, the designer has more control of the loss contributors. For example, instead of supplying the VGJs the excess air could be used for wake filling. It should be noted, however, that the original derivation of the loss equation assumed a constant area duct from position A to B. The derivation could easily be adapted to accommodate a different geometry at the exit (different exit velocity). This analysis maintained the assumption in an effort to gauge the relative benefits of an increase in span (compared to a change in \( Cp \)). Consequently, the pump power equation remains the same as Eq. 7-9.

Unlike the pump power equation, the power gain equation does change slightly. Instead of an increase in lift due to a VGJ-induced \( Cp \) change, the usable span increases with the endwall suction. This increase in usable span is represented in Eq. 7-32 as \( \beta \). As an example, a ten percent increase in available span would yield a \( \beta \) of 1.10.

\[
r\Omega \Delta F_{\text{Lift}} = \frac{\dot{m}_p \rho U_{in}}{2 \rho \sin(55^\circ)} \left[ \beta Cp\bar{\tilde{n}} \bar{\phi} dx - \int Cp\bar{\tilde{n}} \bar{\phi} dx \right]
\]

Eq. 7-32

The combination of Eqs. 7-9, 7-21, and 7-32 provide the power ratio at \( Re_{Cx}=50000 \). The ratio is provided in Eq. 7-33 below.

\[
\frac{P_{\text{Pump}}}{P_{\text{Gain}}} = \left( \frac{\dot{m}_s}{m_p} \right) \left( \frac{\phi}{\eta} \right) \left[ \frac{Cp_A - Cp_B + 4 B^2 (f \frac{L}{D} + K_L + K_e)}{(\frac{1}{\phi})(\beta - 1)} \right] \left( \frac{1}{\phi} \right) (\beta - 1) \int Cp\bar{\tilde{n}} \bar{\phi} dx + 0.6 \phi (\gamma_B = 0 - \gamma_B = 2)
\]

Eq. 7-33
Two pairs of representative LIA values ($Re_{C_x}=50000$, $SR=11.3\%$, $B=2$) for the variables in Eq. 7-12 are provided in Table 7.2 along with their resultant power ratios. In this analysis both suction schemes were attributed with a 10% increase in the usable turbine blade span.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Removal Approach</th>
<th>Redirection Approach</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\frac{\dot{m}_s}{\dot{m}_p}$</td>
<td>0.007</td>
<td>0.007</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>0.85</td>
<td>0.85</td>
</tr>
<tr>
<td>$\sigma = C_x/p$</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>$\eta$</td>
<td>0.85</td>
<td>0.85</td>
</tr>
<tr>
<td>$C_{p_A}$</td>
<td>3.6</td>
<td>3.6</td>
</tr>
<tr>
<td>$C_{p_B}$</td>
<td>2.25</td>
<td>2.25</td>
</tr>
<tr>
<td>$B$</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>$f$</td>
<td>0.02</td>
<td>0.02</td>
</tr>
<tr>
<td>$\beta$</td>
<td>1.10</td>
<td>1.10</td>
</tr>
<tr>
<td>$L/D$</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>$K_{Li}$</td>
<td>0.2</td>
<td>0.2</td>
</tr>
<tr>
<td>$K_{Le}$</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>$\frac{1}{p} \int C_{pV} \hat{n} \cdot \hat{p} dC_x$</td>
<td>0.2824</td>
<td>0.2824</td>
</tr>
<tr>
<td>$- \int C_{pNoV} \hat{n} \cdot \hat{p} dC_x$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$0.6\varphi (\gamma_{B=0} - \gamma_{B=2})$</td>
<td>0.0132</td>
<td>0.0036</td>
</tr>
<tr>
<td><strong>Power Ratio, $\frac{P_{Pump}}{P_{Gain}}$</strong></td>
<td><strong>0.5470</strong></td>
<td><strong>0.5655</strong></td>
</tr>
</tbody>
</table>

Similar to the low $Re_{C_x}$ case, the $Re_{C_x}=50000$ power ratio results are significant for both the removal and redirection approaches. The addition of usable span and the total pressure reductions
caused by endwall suction results in a net gain for the removal and redirection approaches, where 57% and 55% of the total power gain is used to actuate the flow control scheme respectively.

The area-average total pressure contribution to the total power gain accounts for only about 5% of the power gain with the redirection approach and about 1% with the removal approach ($Re_{Ct}=50000$, $SR=11.3\%$, $B=2$).

To this point, the author has assumed that a combined flow control system would require a pump to drive the suction and jet fluid. Another strategy could be to use the existing pressure differentials in the engine. The pressure gradients in a single stage are probably insufficient to drive the control fluid. However, from stage to stage the pressure differentials increase drastically. If the pressure differentials were large enough, the endwall suction array could be plumbed directly to a downstream VGJ row. This type of arrangement would result in a quasi zero-net mass flux system, but, more importantly, would reduce the need for an additional pump. The resultant system would be essentially operation free. The author leaves it to the reader to consider the benefits of such a system.
Chapter 8: Conclusions

The overall objectives of this study were outlined at the end of the introductory chapter of this manuscript. As a conclusion to this work, the results are discussed with respect to each of the original objectives.

**Objective 1: Design an effective flow control scheme that simultaneously mitigates endwall and profile total pressure losses.**

Throughout this research initiative, three main approaches were taken to designing a global flow control scheme. The first approach involved leading edge suction to eliminate the horseshoe vortex. Although the horseshoe vortex system was eliminated at the turbine passage inlet, the impact of endwall suction was minimal in the exit plane. This was observed even for elevated suction rates suggesting that the horseshoe vortex was a secondary contributor to the turbine passage loss. At this juncture, the focus of the research became endwall suction to control the passage vortex.

Two different approaches were attempted for PV control, the removal and redirection approaches. Both approaches had limited success at $Re_{c_1} = 25000$. It was hypothesized that their impact was masked by the VGJ-induced disturbance. This hypothesis was supported by the measured impact of suction at an attached $Re_{c_1}$ without VGJs. At $Re_{c_1} = 50000$, the removal approach altered the secondary flow field by delaying the impingement of the PV on the suction surface of the turbine and eliminating the CV. As a consequence, the area-average total pressure losses decreased monotonically with increasing suction rates. In contrast, the redirection approach
was very effective at diffusing the $PV$ but had little impact on the $CV$ or $TSV$. The result was an asymptotic reduction in the total pressure losses.

**Objective 2:** Assess the impact of leading edge suction on the formation of the horseshoe vortex system.

The application of leading edge suction altered the mechanisms (boundary layers and pressure gradient) that caused the $HV$ system to form. In the end, total pressure plots at the inlet of the cascade showed that the horseshoe vortex system (or the losses associated with it) was eliminated.

**Objective 3:** Assess the impact of passage suction on the formation and migration of the passage vortex system.

The removal approach delayed the formation of the $PV$ until deeper into the passage (beyond the suction holes). As a result, the $PV$ remained in the middle of the turbine passage until the exit plane. Suction with the removal approach also was responsible for an increase in the size and vorticity of the structure. In this respect, endwall suction actually encouraged vortex formation by increasing the velocity gradients downstream of the suction holes. The suction hole pattern also encouraged the entire passage to contribute to the same vortex. The redirection approach resulted in a diffused $PV$ structure that covered a majority of the endwall at the exit plane. The arrangement of holes impeded the formation of a concentrated structure but was unable to significantly alter the loss core. The author concedes that, in the end, the two approaches were mislabeled with the removal approach redirecting the structure and the redirection approach removing it.

**Objectives 4:** Identify the primary loss mechanism in the endwall of a turbine passage.
In the end, this objective is not completely resolved. The position of the endwall vortex structures \((PV, CV, \text{ and } TSV)\) with respect to the loss core suggests that the interaction of these three structures is a major contributor to loss production. Historically, the \(PV\) has been credited with collecting high loss boundary layer fluid as it moves through the passage. Although the total pressure losses do increase in the area of the \(PV\), the increase is slight compared to the loss core that is generated above the \(PV\). It seems possible that the \(PV\) is depositing the high loss fluid above it, but when flow control is applied and the \(PV\) moves away from the suction surface, there is little associated loss near its new position.

**Objective 5: Identify the physics responsible for the reduction in total pressure loss.**

The total pressure loss reduction occurs when the individual vortex structures are weakened or there is relative movement between the structures. For example, when the removal approach is applied at \(Re_{ci}=50000, SR=11.3\%\), a majority of the \(CV\) is removed. The result is a reduction in \(P_T\) loss and movement of the vortex structure. As suction continues to increase, the \(PV\) further weakens and moves away from the suction surface. The result is a significant reduction in the loss core. When the redirection approach is applied at low suction rates, the loss core retains its shape but changes magnitude. The magnitude reduction corresponds with a significant change in the \(PV\). At higher suction rates, the \(PV\) loses influence but the \(CV\) and \(TSV\) remain and continue to interact. Since these structures do not change with increased suction, the loss core remains the same regardless of the suction rate.

**Objective 6: Use another tool (theory or computational fluid dynamics) to aid in the design of the passage vortex endwall flow control scheme.**
This objective was met with the theoretical modeling approach. The approach employed both a computational fluid dynamics package and a theoretically based modeling approach. The results were validated with flow visualization and experimental data.

Objective 7: Perform a system analysis to assess the benefits of a global flow control.

An analysis of the power used to operate the suction/VGJ system was performed by applying an energy balance to the suction/VGJ flow control fluid path. The additional power achieved from the VGJs was assessed by calculating the change in lift of the $C_p$ distribution and the change in the total pressure losses in the wake of the turbine with and without suction and VGJs. The resultant power ratio equation provides an assessment of the operating cost of the flow control system relative to the power gained by employing the system. For a typical situation, 23% of the power gained by the flow control is used in its actuation.

The conclusion of this manuscript is not meant to signify the completion of this research subject. In fact, there are many other aspects of endwall flow control that have yet to be studied. The employment of a combined endwall suction and midspan VGJ system has the potential for significant total pressure loss reduction and increased blade lift. The power costs of actuation can be minimized so that the majority of the power gain goes directly to work extraction. Increased power extraction (stage loading) provides avenues for significant engine weight reduction either through the removal of stages or a reduction in blade count.

Although endwall suction provided minimal loss reductions compared to the VGJs, it does provide an alternative to compressor fed VGJs. In addition, future designs could be developed to account for the individual roles of both the PV and CV in total pressure lose production and decreased available blade span. These designs have the potential for an appreciative decrease in the total pressure losses while maintaining relatively low suction rates.
Although the combined flow control system was only necessary for low $Re$ flows with the $LIA$ blade profile, implementing a suction/VGJ flow control scheme makes it possible for more aggressive blade designs that require flow control to operate efficiently. Since both endwall and profile losses increase with increased blade loading, more aggressive blade designs will necessitate new and innovative approaches to minimizing performance loss. A combined suction/VGJ scheme could provide the necessary control. The added benefit of system adaptability is also noted.
References


24. knezevici, d.c., sjolander, s.a., praisner, t.j., allen-bradley, e., and grover, e.a., 2009, “measurements of secondary losses in a high-lift front-loaded turbine cascade with the implementation of non-axisymmetric endwall contouring,” asme turbo expo 2009: power for land, sea, and air, june 8-12, orlando, fl, gt2009-59677.


35. seal, c.v. and smith, c.r., 1999, “the control of turbulent end-wall boundary layers using surface suction,” experiments in fluids, vol. 27, pp. 484-496.


63 Memory, C., 2009, private communication.


Appendix A: Additional Data

Fig. A.1 Normalized vorticity ($\omega_x \cdot Cx/U_{in}$) contours of the wall-normal PIV planes. The secondary velocity field is represented with the black vectors. Removal approach, $Re_{C_x} = 50000$, $SR = 32.5\%$, $B = 0$. 
Redirection Approach, $Re_{Cy}=50000$, $SR=32.5\%$, $B=0$.

Fig. A.2 Normalized vorticity ($\omega_x \cdot Cy/U_{in}$) contours of the wall-normal PIV planes. The secondary velocity field is represented with the black vectors. Redirection approach, $Re_{Cy}=50000$, $SR=32.5\%$, $B=0$. 

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