Aerodynamic Investigation of Upstream Misalignment over the Nozzle Guide Vane in a Transonic Cascade

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ABSTRACT

The possibility of misalignments at interfaces would be increased due to individual parts' assembly and external factors during its operation. In actual engine representative conditions, the upstream misalignments have effects on turbine’s performance through the nozzle guide vane passages. The current experimental aerodynamic investigation over the nozzle guide vane passage was concentrated on the backward-facing step of upstream misalignments. The tests were performed using two types of vane endwall platforms in a 2D linear cascade: flat endwall and axisymmetric converging endwall. The test conditions were a Mach number of 0.85, $Re_{ex} 1.5 \times 10^6$ based on exit condition and axial chord, and a high freestream turbulence intensity (16%), at the Virginia tech transonic cascade wind tunnel. The experimental results from the surface flow visualization and the five-hole probe measurements at the vane-passage exit were compared with the two cases with and without the backward-facing step for both types of endwall platforms.

As a main source of secondary flow, a horseshoe vortex at stagnation region of the leading edge of the vane directly influences other secondary flows. The intensity of the vortex is associated with boundary layer thickness of inlet flow. In this regard, the upstream backward-facing step as a misalignment induces the separation and attachment of the inlet flow sequentially, and these cause the boundary layer of the inlet flow to reform and become thinner locally. The upstream-step positively affects loss reduction in aerodynamics due to the thinner inlet boundary layer, which attenuates a horseshoe vortex ahead of the vane cascade despite the development of the additional vortices. And converging endwall results in an increase of the effect of the upstream misalignment in aerodynamics, since the inlet boundary layer becomes thinner near the vane's leading edge due to local flow acceleration caused by steep contraction of the converging endwall. These results show good correlation with many previous studies presented herein.
Aerodynamic Investigation of Upstream Misalignment over the Nozzle Guide Vane in a Transonic Cascade

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GENERAL AUDIENCE ABSTRACT

In response to climate change and limited resources, fossil fuel prices are expected to rise and energy policies are expected to change. Under these circumstances, there is a growing demand in the industry to provide an affordable option for improving the efficiency of technology. Energy efficiency is one of most cost effective ways to improve the competitiveness of all businesses and reduce energy costs for consumers.

Regarding the current study topic in particular, the gas turbine is an internal combustion engine that extracts energy, which is resultant from the liquid fuel flow, and is then converted into mechanical energy to drive a compressor or other devices. Gas turbines are used in many applications such as, to power aircraft, electrical generators, pumps, and gas compressors in industrial fields.

Because the gas turbine has a probability of unaligned connections of components due to assembly characteristics of its huge size, performance is affected. To consider issue, an experimental study was conducted related to the energy efficiency for an actual engine’s representative conditions; the current study focuses on the upstream backward facing step of the unaligned connections and highlights the practical effects of the unaligned connection and converging geometry.

These backward facing unaligned connections are shown to have positive effects for reducing aerodynamic losses by weakening a main source of the loss, even despite the development of the additional losses. And, the application of converging geometry to the gas turbine also results in loss reduction due to local flow acceleration. These results show good correlation with the many previous studies presented herein.
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Furthermore, my gratitude especially goes for the support of my company, Korea Gas Corporation. It has given me a great opportunity to research my interests at Virginia Tech. During my years at Virginia Tech, I have experienced, and received, a great deal of various values through my interactions with fellow students.

In conducting my research, I owe a great debt to others as below for their support:

I would like to express my gratitude to my academic advisor, Dr. Wing Ng for encouraging me to think more deeply about my research and offering constant assistance during this research endeavor. My research would never have been completed without his help and encouragement. I have experienced and learned ways to better perform as a manager in charge of projects.

Additionally, I would like to thank the other members of my committee, Dr. Srinath Ekkad and Dr. Alfred Wicks, for their guidance on the research presented herein.

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Finally, I express my more-than-grateful recognition to the sponsor, Solar Turbines Inc. for allowing me to participate in this research project. I hope to collaborate with related businesses after performing the research presented here.
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<td>$C_p$</td>
<td>Loss coefficient</td>
</tr>
<tr>
<td>$LE$</td>
<td>Leading edge</td>
</tr>
<tr>
<td>$Ma$</td>
<td>Mach number</td>
</tr>
<tr>
<td>$NGV$</td>
<td>Nozzle Guide Vane</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure</td>
</tr>
<tr>
<td>$Ps$</td>
<td>Static Pressure</td>
</tr>
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<tr>
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<td>Density</td>
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<td>exit</td>
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<td>$s$</td>
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1. Introduction

It is becoming increasingly obvious that energy efficiency is of growing importance in all industries around the world. With transition to other types of energy, energy efficiency can make conversion cheaper and more beneficial across all sectors of the energy industry. Increasing efficiency is desired throughout the industry in this regard. To add to this operational burden, there have been forecasts of rising oil prices in the long term due to the limitation of existing oil reserves and the expansion of oil production within the limitation. As well, with increasing efficiency, the rise in oil prices will have a significant impact on gas turbine operations.

Within the context of the global energy trend, specifically the need for increasing efficiency in the turbine industry, and the necessity for survival of high-efficiency machinery, consequently leads to a variety of associated future research activities. The turbine industry continues to research and develop their fuel efficiency with regard to energy losses and recovery, since these types of efforts for reducing costs are also contributing to productivity and profitability.

In general, one of these research methods is the modification of engine geometry for the improvement of the gas turbine's efficiency, since it is relatively difficult to improve materials designed to withstand the extreme high pressures and temperatures of turbine operating conditions. From this point of view, converging endwall is representative of such methods. These improvements have been proceeding by minimizing both thermal heat loads and the effects of secondary flow on the endwall and the passage.

In addition, because the components of the combustor and nozzle guide vanes are manufactured and assembled separately and the large size of gas turbines, there would be misalignments at the interface between the combustor exit and the nozzle guide vane entry. The possibility of misalignments would additionally be increased during operation, because of external factors, which include engine transient, manufacturing tolerance, vibration, and maintenance. In real-engine representative conditions, the misalignments have effects on aerodynamic performance through the nozzle guide vane passages in transonic conditions. Practical studies under these realistic conditions have been carried out recently to compare previous experiments with low Mach numbers.

This study intends to examine how upstream misalignments affect the aerodynamic performance across the nozzle guide vane. The following section briefly presents a literature
review on secondary flow, regarding upstream misalignment and converging endwall as a means to determine the best efficiency for turbine engines.

2. Relevant literature review

2.1. Secondary flow physics

Much research has been ongoing for finding more details of secondary flow under a variety of conditions in order to improve turbines' efficiency. Most secondary flow models have been obtained from cascade experiments, and related inlet flow conditions. The various models have been proposed to evaluate secondary flows to aid in the understanding of this complicated flow.

The vortex-flow models of the cascade passage were detailed by many researchers, whose findings were finally summarized by Sieverding [1]. These were further extended by Langston et al. [2], Langston [3], Gregory-Smith [4], Sharma and Butler [5], Goldstein and Spores [6], and Takeishi [7].

For a better understanding of the secondary flow behavior and features, Wang et al. [8], described the detailed vortex flow patterns indicating the formation and evolution of secondary flows at the cascade passage, as shown in Figure 1.

Figure 1: Secondary flow model [8]
As approaching inlet flow in the boundary layer encounters a leading edge of an airfoil, a horseshoe vortex is formed due to the adverse pressure gradient, and splits into two legs: a pressure-side leg, and a suction-side leg. At that juncture, a saddle point and a separation line are observed as evidence of development of the horseshoe vortex. The stronger pressure-side leg of the vortex near the leading edge propagates toward the suction-side of the adjacent airfoil. A corner vortex is induced under the influence of a passage vortex at the meeting point of the pressure-side leg vortex and suction-side leg vortex. The weaker suction-side leg of the horseshoe vortex meets the passage vortex near the suction-side surface and lifts off the airfoil. This is a small, but intense counter vortex (called a wall vortex by Wang et al. [8]). The horseshoe vortex pressure-side leg then becomes a passage vortex. It lifts off the endwall and grows in size as it moves downstream. This passage vortex has its boundary layer in the passage, and accompanies the mainstream flow. It can be shown that the horseshoe vortex is a primary source of the aerodynamic losses in the passages, and significantly affects other secondary flows.

Near the pressure side of the airfoil, the endwall-crossflow moves pitch-wise across the passage from the pressure side toward the suction side of the adjacent airfoil at a lower speed. The flow field near the endwall region of the passage is dominated by a boundary layer, and strong pressure gradients.

With regard to the secondary flow, Kang et al. [9] evaluated heat transfer and flow field characteristics with two exit Reynolds numbers near the leading edge of the vane endwall, where horseshoe vortices form. Their finding was that the horseshoe vortex forms closer to the leading edge at lower Reynolds numbers. The lower Reynolds number case shows larger velocity gradients near the core of the vortex.

The effect of inlet boundary-layer thickness on endwall heat transfer was investigated with flow visualization results in a large-scale-blade cascade by Graziani et al. [10]. It was noted that for a thin, inlet-flow boundary layer, a saddle point was formed closer to the pressure side and there was a broader coverage of the endwall crossflow. This is associated with the fact that the size and the intensity of the horseshoe vortex are dependent on the thickness of the inlet boundary layer. As the flow accelerates, the boundary layer becomes thinner.

In addition, the development of the secondary flow depends on the airfoil design endwall geometry and the inlet flow conditions into the passage. The flow features on the converging endwall have been extensively analyzed and reviewed. As a practical factor, a number of detailed
investigations have been conducted over many decades in an attempt to weaken secondary flows and minimize aerodynamic losses.

With regard to the endwall geometry, the converging endwall can generally be categorized as axisymmetric or non-axisymmetric. Non-axisymmetric converging endwalls are used to reduce pressure gradients across the passage, which causes endwall crossflow. The benefits of non-axisymmetric converging endwalls were investigated by many researchers [11–15].

While many of the existing investigations are associated with rotor blades, this study concentrates on transonic stator vanes. Of the two endwall types, axisymmetric converging endwall is typically utilized for the first stage guide nozzle, where the flow increases its velocity as the pressure drop across these nozzles. This is the focus of this thesis.

Spencer et al. [16] performed endwall heat transfer and aerodynamic measurements in an annular-cascade of nozzle guide vanes. The authors noted that the separation line for the converging endwall was closer to the stagnation point because the inlet flow was accelerated due to the abrupt narrowing, and it had a radial velocity component and pressure gradient at the leading edge.

The effect of the converging endwall over the nozzle guide vane was reported by Thrift et al. [17]. It was found that the boundary layer became thinner in the passage due to flow acceleration caused by endwall convergence, reducing the size of the horseshoe vortex.

2.2. Backward facing step flow

When it comes to upstream misalignment, the flow behind the upstream step is complex, and composed of different flow regions despite its geometric simplicity. The backward facing step is mostly considered as one of separation flow geometry. The development of the horseshoe vortex can be influenced by flow separation and reattachment around the step. Upstream of the vane cascade, the current study also includes the backward-facing step with different endwall-geometry configurations.

The low-speed two-dimensional reattachment and its flow field were reported by Bradshaw and Wong [18]. Eaton and Johnston [19] showed that the backward facing step induces reattachment flow, with separation region and reverse flow in the subsonic turbulence, as seen in Figure 2.
An investigation regarding the effects of the forward- and backward-facing steps on the endwall flows of a turbine-blade cascade was conducted by de la Rosa Blanco et al. [20, 21]. They found that the backward facing step leads to lower losses in comparison with a flat endwall. However, this effect depends on the step height, and the incoming boundary-layer thickness.

The effect of the misalignment and film-cooling leakage in the nozzle guide vane endwall were studied by Piggush et al. [22]. It was noted that the backward facing step causes local deceleration of inlet flow behind the step, allowing a boundary layer to thicken more rapidly, whereas a forward-facing step has the opposite effect on the local flow acceleration, with the boundary layer becoming thinner.

Arisi et al. [23] and Mayo [24] investigated the heat transfer effect of an upstream backward-facing misalignment between the combustor exit and the inlet-nozzle guide vane on the endwall surface for both flat and converging endwall geometries. Their studies revealed that the strength of the endwall secondary flows was reduced by the step, but an additional unique auxiliary vortex system developed upon interaction with a separation vortex, and a lateral and axial pressure gradient in the inlet plane in front of the vane.

The main objective of this study is to determine detailed aerodynamic effects of interface misalignment over the nozzle guide vane. The findings are presented for both general and detailed analytical and aerodynamic reviews of the inlet nozzle guide vane endwall platform misalignment on the secondary flow.
3. Experimental setup

3.1. Wind tunnel facility

An aerodynamic investigation is still dependent on testing in a wind tunnel. The Virginia Tech transonic-linear-cascade wind tunnel was primarily utilized for this study. A schematic of the wind tunnel is presented in Figure 3. A visual overview of this facility and more detailed capabilities of this facility can be found in Nix [25], Holmberg et al. [26], Anto et al. [27], Nasir et al. [28], Carullo et al. [29], and Jain et al. [30], amongst others.

The wind tunnel is two-dimensional, and is a blow-down type providing a maximum mass flow rate of 4.5 kg/s for approximately 15 s. The tunnel is operated with a LabVIEW code, which manages the inlet pressures and data acquisition. The flow inside the wind tunnel is mainly controlled using butterfly valves, located in front of the heat exchanger. The flow conditions of the wind tunnel are in a quasi-steady state, while the temperatures of the freestream and vane-endwall surface at the test sections are transient.

The wind tunnel comprised a compressor, two air storage tanks, control valves, a heater, and a test section. The experimental runs were conducted using a high-pressure compressor. First, a four-stage high-pressure compressor was used to raise pressure in an air storage tank to 1380 kPa (200 psig). The compressed air was then transferred to a second air tank, and was allowed to stabilize between 20.7 kPa (3 psig) and 69.0 kPa (10 psig), at which point the air moved to the test section through the pipeline. The isentropic exit Mach numbers in the wind tunnel can be varied from
approximately 0.7 to 1.2. The isentropic exit Mach number was specifically set at 0.85 for this study.

A passive heat exchanger and a heater are located between the inlet control valve and the test section. The heater can heat the cascade inlet flow up to 120 °C. This heating process was not required for this experiment; therefore, it was bypassed.

3.2. Turbulence grid

As shown in Figure 4, a passive square mesh turbulence grid was applied to generate approximately 16% freestream turbulence intensity at 0.48 $C_{ax}$ upstream of the cascade leading edge. The turbulence grid was located at 5.5 $C_{ax}$ from the leading edge of the vanes, between the plenum and test sections. This grid design was based on the turbulence-intensity research carried out by Baines et al. [31], and performed by Nix et al. [27] in the Virginia Tech wind tunnel.

![Square-mesh turbulence grid](image)

Figure 4: Square-mesh turbulence grid

3.3. Test section

Figure 5 presents a schematic and photograph of the cascade test section, showing four vanes and three passages. The passage of interest is the center passage between the third and fourth airfoil from the top. The pressure data were collected on the central passage to ensure the flow's periodicity around this central passage. The transition geometry was located at 0.48$C_{ax}$ upstream of the leading edge.
The main components consisted of three pieces of vane-endwall platforms. They were placed adjacent to each other at the test section. The vane-endwall platforms were interchangeable at the test section to allow varied experiments on turbine parts. The platforms were secured to the test section windows with screws on the backs. The gaps between the platforms were smoothed using epoxy glue in order to avoid the leakage of the air and reduce resistance. Twelve brass tubes of 1/16” size were inserted into the holes of the test section window and fixed in place with adhesive in order to measure inlet- and exit-static pressures in the cascade. Further details on the test section can be found in Appendix A. A tailing board was placed near the trailing edge exit of the first vanes to guide exit flow and keep flow periodicity for each passage. Upstream of the vane leading edge, an upper slot located at 1 C_{ax} was used to ascertain inlet total-pressure and static-pressure profiles, as well as the turbulence conditions.

![Figure 5: Vane cascade geometry and details](image)

### 3.4. Vane endwall design

The test-vane endwall platform has a geometry similar to a first-stage nozzle guide vane of a gas turbine. The vane cascade was scaled up by a factor of 1.5. The vane-cascade geometry specifications are summarized in Table 1. The set’s vane-endwall platforms, from the designs provided by Solar Turbines Inc., were manufactured by a Stratasys Fortus 250mc 3D-printing machine using ABS P-430.
Table 1: Vane geometry

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Chord</td>
<td>91.2 mm (3.59 in.)</td>
</tr>
<tr>
<td>Axial Chord</td>
<td>50.0 mm (1.97 in.)</td>
</tr>
<tr>
<td>Pitch</td>
<td>83.0 mm (3.27 in.)</td>
</tr>
<tr>
<td>Span</td>
<td>152.4 mm (6.00 in.)</td>
</tr>
<tr>
<td>Step Height</td>
<td>6.78 mm (0.68 in.)</td>
</tr>
<tr>
<td>Inlet and Exit Angle</td>
<td>0° and 73.5°</td>
</tr>
</tbody>
</table>

3.5. Measurements and instrumentation

The experimental technique of oil-surface-flow visualization and five-hole-probe measurements downstream were used in this work to analyze flow features.

3.5.1. Surface flow visualization

The surface flow visualization was conducted in order to qualitatively investigate the effects of the backward-facing step as seen in Figure 6. Yellow, pink, green, and blue fluorescent oils were used for this study. A line of each of the four oils was painted onto the endwall surfaces at each of the designated locations, as shown in Table 2, and the lines were tracked until the wind tunnel was stopped.

Figure 6: Before (Left), and after (Right) pictures for the flow visualization tests
Table 2: Oil application for flow visualization

<table>
<thead>
<tr>
<th>Location</th>
<th>Color</th>
</tr>
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<tbody>
<tr>
<td>Upstream ahead of the static-pressure ports</td>
<td>Yellow</td>
</tr>
<tr>
<td>Endwall entry behind backward-facing step</td>
<td>Red</td>
</tr>
<tr>
<td>Pressure-side curve on endwall</td>
<td>Green</td>
</tr>
<tr>
<td>Suction-side curve on endwall</td>
<td>Blue</td>
</tr>
</tbody>
</table>

The oils, after being released, were swept along by the airflow, which was set for an exit Mach number of 0.85 for all test points. The oil generated certain visible flow patterns on interaction with the airflow. The qualitative flow structures become visible from the observed patterns, which is particularly useful for detecting specific flows regarding secondary flows at specific positions.

The streaks of colored oil followed the vane exit angle smoothly in the wake region behind the trailing edge of the vanes. The oil traces showed the flow patterns flowing up the suction surface, and curving back toward the trailing edge of that surface. Near to the cascade exit, the oil streaks on the endwall moved directly across the passages in a direction tangential to the vanes. This could suggest periodicity for the cascade vane at a transonic exit-Mach number.

3.5.2. Upstream measurement

For temperature applications, a NI SCXI-1320 model was utilized for calculating the flow properties for the compressible flow, which included the speed of sound, velocity and density.

A NetScanner Model 98RK PSI system was used for sampling pressure measurements at 10 Hz during all of the experimental runs. Upstream, the total inlet pressure and static pressure were measured by a stationary pitot-static probe, which was located at 3 C_{ax} of the vane leading-edge, and mid-span. Static pressure data at six pressure ports on the surface of both the inlet and the exit were obtained from -0.91C_{ax}− +1.26 C_{ax} from the vane’s leading edge, as shown in Figure 5.

3.5.3. Downstream measurement (five-hole probe measurement)

At the exit of the nozzle guide vane, a five-hole probe measurement was employed to analyze aerodynamic performance based on the loss-coefficient value, since the five-hole probe method is already well proven.
The probe was positioned at $1.25 \text{ C}_{\alpha x}$ from the leading edge of the vane cascade through the slots of the test section window and a traverse box, which contained the five-hole probe and two traverses for the five-hole probe measurements. The probe's angle was set to the exit flow angle of the vane exit row. Total exit pressures and velocity components across the central passage were measured while the probe traversed approximately 15 cm in a pitch-wise direction along the measurement slot.

These measurements were carried out at 11 logarithmically spaced span-wise positions, from near the endwall to 75 mm above midspan, repeating the same method under identical conditions. The details of the five-hole probe are presented in Appendix B.

For this measurement, the traverse box, to which the five-hole probe was secured, was attached to the window and sealed with screws and gaskets to avoid leakage during experimental runs. The two traverses inside the traverse box were used to measure pressures at specific grids of exit flow for two axial movements, span-wise, and pitch-wise. A traverse from Velmex, Inc., with a stepper motor and a manual traverse, was used for the two directions. In the pitch-wise direction, a motor-driven traverse was utilized for keeping the probe at a constant velocity of 1 in/s. The probe was shifted to the adjacent span-wise positions by a manual traverse for a total of 11 locations. The exit total pressure of data obtained from the five-hole probe measurements for each run is presented in Appendix C.

![Figure 7: Five-hole probe setup details](image-url)
For this study, a standard five-hole probe data-reduction algorithm for the probe was applied in conjunction with the calibration database provided by the manufacturer, Aeroprobe. The provided calibration data are for a range of Mach numbers from 0.4–1.2.

The calibration involves the identification of three dimensionless pressure coefficients in order to interpolate between three coefficients obtained from the calibration database and the experimental data. The three-dimensional curve-fits from the calibration data were used to determine the velocity-magnitude and -direction from the five pressures measured. The result of the calibration procedure is a set of three calibration coefficients mapping the dimensionless pitch angle, yaw angle, and pressure, to pressure data obtained from the probes. The coefficients from the experimental data can be determined comparing the pressure differences between these tubes. The pressures are converted into dimensionless coefficients, which are defined as follows;

\[
C_{p_{\text{yaw}}} = \frac{P_2 - P_3}{P_1 - \bar{P}}, \quad C_{p_{\text{pitch}}} = \frac{P_4 - P_5}{P_1 - \bar{P}}, \quad C_{p_{\text{Ma}}} = \frac{P_1 - \bar{P}}{P_1}
\]

where, \(\bar{P} = \frac{P_2 + P_3 + P_4 + P_5}{4}\)

The subscripts indicate the five-hole indices. The stagnation pressure \(P_1\) was collected as the pressure at the central hole. From the other tubes, \(P_2\) and \(P_3\) were measured in the yaw plane, and \(P_4\) and \(P_5\) were measured in the pitch plane. The four averaged values provide static pressures, \(P_2, P_3, P_4, P_5\), as shown in Figure 21.

Based on pressure coefficients, the pitch- and yaw-plane, and Mach coefficients from the calibration database provided by Aeroprobe, the data between the span-wise locations of the experimental runs were interpolated against the calibration database to generate pictures of the exit flow conditions. From the calculated pressure data, a layer of the data grid with local pitch-wise and span-wise locations was created to generate interpolation plots of the exit flow conditions of the center passage.

The interpolation data were represented by three-dimensional spline curves passed through the individual data points to show the loss coefficients. The mass-averaged loss-coefficient values at each local point of the central passage grids were calculated according to equation (1).
The nominator of the equation is the difference between the total inlet and total exit pressures, and the denominator values gives the normalizing factor, which is the difference between exit total pressure weighed by mass flow, and the exit static pressure averaged by the area for the test grid in the exit plane.

The uncertainty analysis for the averaged loss coefficient was conducted by means of Kline and McClintock’s method [32], presented in Appendix D. The uncertainty of the averaged loss coefficient varies within the range of 3.8–6.1%.

3.6. Flow condition inside the wind tunnel

Through the wind tunnel, the flow condition during the experimental runs was assumed to be compressible and isentropic, which means that the total pressure remained constant through inviscid core of the nozzle guide vane passage.

The exit static pressure data from the six wall pressure ports were averaged at each interval step; then the exit Mach numbers were calculated at each interval, using Equation (2), in conjunction with the total inlet pressure data over the 6 s data-reduction window. This window was under quasi-steady-state flow conditions.

\[
M_{\text{isen}} = \sqrt{\frac{2}{\gamma-1} \left( \frac{P_t}{P_\infty} \right)^{\gamma-1} - 1} \tag{2}
\]

Similarly, equation (2) can be used to calculate the inlet Mach number using the inlet static pressure taken from a pitot-static tube. The reported exit Mach number is used for the results in this study. As shown in Table 3, these flow properties were computed using the total temperature measured the upstream of the cascade, static pressure from the pressure ports, and exit total pressure derived from the five-hole probe. All experiments were conducted with an isentropic exit Mach number of 0.85. The results from each test will be discussed in the following section.
Table 3: Flow properties

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Exit Reynolds number (Re)</td>
<td>$1.5 \times 10^6$</td>
</tr>
<tr>
<td>Exit Mach number (Ma)</td>
<td>0.85</td>
</tr>
<tr>
<td>Turbulence Intensity (Tu)</td>
<td>16%</td>
</tr>
</tbody>
</table>

3.7. Test configuration

For the present study, the test configurations were set up to conduct two investigations of aerodynamic performance: the effects of the upstream backward-facing step, and the converging endwall. The configuration lists, presented in Table 4, give the geometry condition for each case. Two sets of tests, for both the flat and the axisymmetric converging endwall representing the inner casing of the engine, were conducted to compare the two effects.

Table 4: Test configurations

<table>
<thead>
<tr>
<th>Cases</th>
<th>Note</th>
<th>Endwall</th>
<th>Entry Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case BF</td>
<td>Baseline</td>
<td>Flat</td>
<td>No Gap</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>No step</td>
</tr>
<tr>
<td>Case MF</td>
<td>Misaligned</td>
<td>Flat</td>
<td>No Gap</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Step</td>
</tr>
<tr>
<td>Case BC</td>
<td>Baseline</td>
<td>Converging</td>
<td>Gap</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>No step</td>
</tr>
<tr>
<td>Case MC</td>
<td>Misaligned</td>
<td>Converging</td>
<td>Gap</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Step</td>
</tr>
</tbody>
</table>

Figure 8: Geometry details for 4 cases
As shown in Figure 8, the height between the upstream surface and the vane endwall platform surface was the same for all of the misaligned cases. The endwall platform was recessed from the upstream surface, creating a misalignment. In terms of the inlet entry condition for the tested cases, the distance from the step to the vane's leading edge was 43.8 mm (a) and the misalignment step height was 6.78 mm (Δ).

4. Study results and discussion

4.1. Experimental results

The principal features of secondary flow were viewed with the surface-flow-visualization experiments and five-hole-probe measurements.

4.1.1. Case BF results: flat endwall (baseline)

The flow-visualization results for the baseline case BF show close aerodynamic agreement with the flow field features of the typical, flat endwall, as shown in Figure 9.

![Figure 9: Flow visualization results for case BF](image_url)

The inlet flow is stagnated in front of the leading edge. As the associated stagnation pressures are increased, the inlet flow is decreasingly decelerated and then decreases. The downward flow rolls up near the endwall surface and a horseshoe vortex is generated at this juncture. The saddle point (A) also formed where the reverse flow meets an incoming flow.

A separation line (Sp1) is seen from the saddle point to the adjacent suction side along the passage as indicated by the formation and evolution of a horseshoe vortex. Two different flows are
observed on either side of the separation line (Sp1). Similar to reported by Spencer et al. [16], the upstream side region of this separation line (Sp1), the pressure-side leg of the horseshoe vortex became the passage vortex, which traveled towards the adjacent-vane suction side. It lifted off to the suction-side surface at 0.35 $C_{ax}(B)$ from the leading edge. The endwall crossflow with a new boundary layer is shown on the endwall surface of the pressure-side, downstream-side region of the separation line (Sp1). This may suggest that the newly formed boundary layer gets influenced by the strong horseshoe vortex in the passage.

At the suction side of the leading edge, as the horseshoe vortex suction-side leg enters the passage, the suction-side leg encounters the passage vortex where it reaches the opposite suction side. The suction-side leg then becomes the counter vortex. It sweeps up onto the suction-side surface and moves along the surface, indicated by the blue oil streaks in Figure 10.

![Figure 10: Flow visualization results on suction side surface for case BF](image1)

On the suction-side surface of the vane, three vortices are shown from the endwall in the order of corner vortex, a passage vortex, and a counter vortex. These flow visualization results can be confirmed as consistent results of the five-hole probe measurements as presented in Figure 11.

These three vortices can be observed at each position in the loss-coefficient distribution. The highest peak near the endwall is associated with the corner and passage vortices. The corner and passage vortices appear between 2–8% span. The region of the corner and passage vortices can be identified by a highly concentrated shape. These two vortices are located close to each other, and the corner vortex is seen to be lower on the closer-pressure side. The corner vortex and the passage
vortex are clearly visible near the endwall under the counter vortex. At around 8–15% span, loss core is seen due to the presence of the counter vortex. The counter vortex relatively shows lower intensity than other vortices.

![Figure 11: Loss-coefficient distribution for case BF](image)

### 4.1.2. Case MF results: flat endwall with step (misalignment)

Unlike the baseline case BF, the separation and reattachment of the inlet flow can be observed in the inlet plane ahead of the leading edge as a result of the presence of the backward facing step, as seen in Figure 12. The separation flow behind the upstream step is the typical flow, and leads to the reverse flow from the attachment region over the inlet plane. It can be confirmed by the reverse flow (a) in the flow visualization results. This result is consistent with the flow patterns investigated by Eaton and Johnston [20].

The inlet flow's attachment (a) after the separation flow appears in the vicinity of the vane's leading edge. The reattachment region shifts axially farther into the passage, and the reverse flow diverges to lateral flow when compared with the BF case. At the stagnation region in front of the vane, a horseshoe vortex originates near a saddle point (a) closer to the leading edge, and the vortex's pressure-side leg grows into another vortex; a passage vortex. Since an incoming boundary layer induced by the step is thinner, the strength of the horseshoe vortex is significantly weakened.
The moving passage vortex faces the suction-side leg of the vortex near the suction-side surface. At this point, the suction-side leg becomes a source of a counter-rotating-vortex pair. The counter vortex, which travels along the suction-side surface, is shown above the passage vortex in Figure 13. Downstream of the separation line, the broader endwall crossflow is seen near the pressure side. This may suggest that it is affected by the reduced strength of the horseshoe-vortex pressure-side leg.

In the mid-pitch region of the inlet plane, the auxiliary vortex system (c), which was found by Arisi et al. [23], is formed due to the interaction with the lateral- and axial-pressure gradients. The passage flow in Figure 12 indicates that the auxiliary-vortex system gets weakened as the flow
passes into the passage with the passage vortex. These results show good correlation with the findings of Arisi et al. [23].

As the pressure-side leg of the horseshoe vortex becomes the core of the passage vortex, the passage vortex encounters the auxiliary vortex, and the passage vortex reaches the adjacent suction-side surface at $0.68 \ C_{ax}(b)$. The passage vortex starts to lift off near the passage vortex and suction-side leg of the horseshoe vortex, and travels along the surface. These three vortices have the same order as those of the baseline case. It is also consistent with the results of the five-hole probe measurement as shown in Figure 14.

![Figure 14: Loss-coefficient distribution for case MF](image)

As observed between 2–5% span, this peak region, closer to the endwall, corresponds to interaction between a corner vortex and a passage vortex. Another loss core between 5–11% span, is found due to a counter vortex. The counter vortex has a relatively lower intensity than the corner vortex and passage vortex. These vortices spread out and are more broadly and lower distributed than case BF. The loss distribution above the counter vortex shows the tendency to increase gradually to the mid span.

4.1.3. Case BC results: converging endwall with gap (baseline)

The flow visualization results are presented in Figure 15. A saddle point (A), and two separation lines (Sp1, Sp2), which are associated with a horseshoe vortex, are clearly visible.
Figure 15: Flow visualization results for case BC

The separation flow caused by the interface gap attaches near the end of the interface gap, which represents the combustor-turbine transition geometry as an actual engine representative condition. As a consequence of the separation, a recirculation flow (C) is generated within the interface gap. The attachment of the mainstream flow causes two boundary layers to be formed in opposing directions. One boundary layer goes into the gap created by the recirculation flow, and the other boundary layer is formed in the passage.

The flow behind the attachment point was locally decelerated due to the interface gap. This indicates that the boundary layer becomes thicker before the flow after the attachment reaches the leading edge of the vane. The inlet flow with the thicker boundary layer causes the formation of a stronger horseshoe vortex near the leading edge. A saddle point (A) is presented as the evidence of the horseshoe vortex at around 0.14 \( C_{ax} \). A passage vortex, which developed from the pressure-side leg of the horseshoe vortex, travels to the passage along a separation line (Sp1). The main flow in the passage is accelerated from the beginning of the curvature of the converging endwall due to the nozzle and the steep narrowing of the converging endwall through the passage, as noted by Spencer et al. [16].

Upstream of the separation line (Sp1), the passage vortex and the inlet flow converge in the passage, and the passage vortex reached the suction-side surface of the vane at 0.43 \( C_{ax} \) (B). At this juncture, the passage vortex lifts off the suction-side surface and moves along the surface. The suction-side leg of the horseshoe vortex meets the passage vortex and becomes a counter vortex. From the endwall surface, three different oil streaks on the suction-side surface represent a corner vortex, a passage vortex, and a counter vortex, as shown in Figure 16.
As shown in the positions of these oil streaks in Figure 16, three vortices are shown near the endwall under 15% span in loss coefficient distribution, as presented in Figure 17. The highest loss core found near the endwall can be attributed to interaction with the corner vortex and passage vortex at the position between 2–8% span. The counter vortex is located from 8–15% span above the passage vortex.
4.1.4. Case MC results: converging endwall with gap and step (misalignment)

As opposed to the previous case, BC, the presence of the upstream backward-facing step causes separation flow and shows different flow patterns over the nozzle guide vane. The flow visualization results are presented in Figure 18, and are consistent with the studies by Eaton and Johnston [20].

![Figure 18: Flow visualization results for case MC](image)

With the introduction of the backward-facing step, the separation flow reattached closer to the leading edge of the vane. The reverse flow (b) can be observed, and the horseshoe vortex is formed near the stagnation region in the vicinity of the leading edge. As evidence for the horseshoe vortex, a saddle point (a) at around 0.1 $C_{ax}$, where the reverse flow encounters the incoming flow, is also shown closer to the leading edge. The horseshoe vortex is bifurcated to the pressure-and suction-side legs of the horseshoe vortex.

The pressure-side leg of the horseshoe vortex is weakened due to a thinner approaching boundary layer caused by the upstream step. This causes the intensity of the horseshoe vortex to be attenuated. At this juncture, as the weakened horseshoe vortex pressure-side leg encounters the more intensive inlet flow under the influence of the auxiliary vortex system, the flow path of the passage vortex is delayed and lifts off on the opposite suction side at 0.66 $C_{ax}$ (d) from the leading edge. The vortex moves to the passage exit along the adjacent suction-side surface. In addition, as another suction-side leg of the horseshoe vortex enters the passage, it lifts off from the point where the suction-side leg meets the passage vortex at the suction-side surface and becomes a counter
vortex. A corner vortex and a counter vortex were also observed, from oil traces, on the suction-side surface with the passage vortex, as seen in Figure 19.

![Figure 19: Flow visualization results on suction side surface for case MC](image)

The oil traces (c) in the interface gap may suggest that a new auxiliary vortex system, which was studied by Mayo et al. [24], was generated due to the interaction between the reverse flow and the lateral flow. The auxiliary vortices then enter the passage.

Near the pressure side of the vane, the endwall cross flow behind the separation line is broader than in case BC. It is broadly distributed near the leading edge due to the weaker horseshoe vortex, but narrows in the passage as a result of the deeper penetration into the passage of the flow with the auxiliary vortex.

The flow in the passage accelerates due to the nozzle and the converging endwall contraction, thereby thinning the boundary layer of the flow and increasing the strength of the secondary flows, which result in increases in aerodynamic losses.

Near the trailing edge of the vane, the three vortices can be observed from the visualization results. Each of the positions and the intensity of the vortices can be confirmed through the mass-weighted loss-coefficient distribution shown in Figure 20. The largest loss is observed as interaction with the corner vortex and the passage vortex on the endwall surface in the 2–5% span. Another loss core is observed at the 5–11% span. The higher loss above the largest peak represents counter vortex. These vortices have lower positions and strength than those of the baseline case.
4.2. Effect of backward facing step

4.2.1. Flat endwall: comparison between case BF and case MF

From comparing the visualization results for the two cases, the locations of the saddle points and separation lines is the major difference as shown Figures 9 and 12. The saddle point is located closer to the leading edge of the vane in the misaligned case than in the baseline case due to the attachment of mainstream flow caused by the upstream step. Because of a new boundary layer forming, and thinner-than-normal inlet flow of the baseline case after the attachment flow, the point moves farther towards the leading edge. The thinner boundary layer, due to local deceleration of the inlet flow, generates the weaker horseshoe vortex.

Another notable observation of the oil traces on the vane-endwall platform is the convergence length of the passage vortex on the suction side. This is associated with the auxiliary vortex system. In the misalignment case MF, the auxiliary vortex system, which was found by Arisi et al. [23], originates on interaction with the lateral- and reverse-flow induced by the backward facing step for case MF case at the inlet plane ahead of the leading edge. This affects the path of the passage vortex, and results in the delayed migration of the passage vortex to the adjacent suction-side surface. On the other hand, the stronger passage vortex from the stronger-momentum boundary layer reaches the suction side at a shorter distance from the leading edge as was seen with case BF.
Additionally, in the misaligned case MF, the endwall crossflow near the pressure side under the influence of weaker horseshoe vortex, has a broader coverage than that of the baseline case BF downstream of the separation line. These comparisons from the flow visualization results show the good correlation with the investigation results of Graziani et al. [10].

With regard to aerodynamic losses, both cases generated very similar losses, but show different positions and strengths of vortices when comparing the averaged loss-coefficient profiles seen in Figure 21.

![Figure 21: Averaged loss-coefficient comparison between baseline flat endwall (BF) and misaligned flat endwall (MF)](image)

The loss-coefficient profile shows lower vortex strength in the misaligned case, since the application of the upstream step greatly reduced the strengths of all the vortices due to a thinner inlet boundary layer caused by local flow deceleration. On the whole, the loss-coefficient profile of case MF shows a loss about 15% lower than that of case BF. This is particularly evident for the positive effect of the upstream, backward-facing step.

As seen in left-hand pictures in Figures 11 and 14, the vortices are positioned lower in the misaligned case MF than in the baseline case BF, and the vortices are less widely distributed. Likewise, for both cases, the peak in loss near the endwall can be attributed to the corner vortices. The passage vortex and counter vortex contributed two loss cores. The corner vortex and the
passage vortex are located adjacent to each other in both cases. The loss core region of both the passage- and counter-vortices is shifted closer to the suction side for the misaligned case. These loss-coefficient distribution results correlate closely with the findings of de la Rosa Blanco et al. [21].

These results can also be confirmed by the three oil traces on the suction-side surface, which show the position of the three vortices in the order of a corner vortex, a passage vortex, and a counter vortex. These are consistent positions of the vortices that are shown in the loss-coefficient distributions.

When compared with previous investigations [10, 21, 23], this study’s results show overall close correlation with their results in terms of the location of each flow pattern, and the presence of additional vortex system, despite different experimental conditions.

4.2.2. Converging endwall: comparison between case BC and case MC

The upstream, backward-facing step ahead of the vane cascade causes clear differences in the flow field over the nozzle guide vane passage when comparing the baseline case BC to the misaligned case MC.

Of particular importance is the difference in the formation position of the horseshoe vortex. Its position for the misaligned configuration is located closer to the leading edge of the vane than that of the baseline configuration due to the reattachment of the upstream flow after the separation flow, as seen in Figures 15 and 18. This is associated with the intensity of the horseshoe vortex, which affects the overall secondary flow over the vane passage. This indicates that the thinner boundary layer of inlet flow, which forms later, causes the horseshoe vortex to be attenuated. The intensity of the horseshoe vortex primarily affects other induced-secondary-flows' strengths and movement patterns. The total loss is eventually reduced by about 18% for the misaligned case, as shown in Figure 22.

Through a comparison with loss-coefficient distribution from five-hole probe measurements, two comparative studies have also confirmed the positions and strengths of the vortices, and the two experimental results are closely correlated to each other. Figure 22 illustrates the extent of the overall losses and their differences in position. The relatively highest peaks correspond to the corner vortex, and the passage vortex is positioned above the corner vortex. Another loss core represents the counter vortex.
A noteworthy difference is the delayed migration of the passage vortex in the passage. Since the separation line corresponds to the paths of the passage vortex, the flow pattern of the passage vortex can be confirmed through the line. For the misaligned case MC, the new auxiliary-vortex system is generated near the end of the interface gap due to reverse flow and lateral flow caused by the step. The incoming flow with the auxiliary vortex enters deeper into the passage, and this causes the passage vortex to reach the opposite suction-side surface of vane later, at $0.66 \, C_{ax}$. In the baseline case BC, the pressure-side leg of the horseshoe vortex meets the opposite-vane suction-side surface at a shorter distance, at $0.43 \, C_{ax}$.

Finally, the broader endwall crossflow near the pressure side for both cases is observed for the misaligned case than that of the baseline case, but gets narrower in the passage due to farther penetration caused by the auxiliary vortex. This may suggest that the weakened horseshoe vortex has an indirect effect on the pressure-side flow.

It can be seen that this study shows reasonable correlation with the findings of Graziani et al. [10], de la Rosa Blanco et al. [20], and Mayo [24].
4.3. Effect of the converging endwall

The converging endwall is essential for the gas turbine to increase the velocity of the flow in order to increase the efficiency, when considering the fundamental function of the rotor.

This comparison show noticeable consistency with the influence of the converging endwall on the passage flow over the nozzle guide vane previously studied by Thrift et al. [17], and Spencer et al. [16]. As evidence for the good correlation, the separation line in the converging endwall configuration is located closer to the stagnation point than that of the flat endwall, as shown in Figures 9 and 15. This is associated with the thinner boundary layer of the inlet flow due to the flow acceleration through a converging contraction of the endwall.

![Figure 23: Averaged loss-coefficient comparison for converging endwall](image)

As seen from the loss-coefficients comparisons in Figure 23, the experimental results for the effect of the converging endwall show better performance, except for proximity to the endwall, despite the existence of the upstream interface gap, which is perceived as the loss. Two respective comparisons, one between the two baseline cases BF and BC, and the other between the two misaligned cases MF and MC, show the reductions in losses, except for the range 2.3–4.2%. For the region near the endwall where losses increase, it may suggest that this is due to interaction with the corner vortex and flow acceleration in the passage. The most significant benefit of the converging endwall in the passage is the reduction in losses. An improvement of 12.2% is
observed in a comparison between the baseline cases. There is 14.6% loss reduction in a comparison of the misaligned cases.

The comparison shows that the flow in the passage is accelerated due to the convergent geometry of the converging endwall, which leads to a substantially lower level of loss, affecting secondary flows and overall loss. This may suggest that the relatively increased loss induced by converging is nominal, though it increases the loss due to streamwise flow acceleration in the passage.

More detailed comparisons with literature will be given in the next section.

4.4. Comparison with literature

Comparisons between current study results and literature were made to validate the current study.

First, this comparison is associated with the flow visualization results based on the effects of the boundary layer thickness on inlet flow. Graziani et al. [10] reported a saddle point closer to pressure side of the airfoil and a broader coverage of endwall crossflow near the pressure side, specifically for the thin boundary layer when compared with the thick boundary layer, as illustrated in Figure 24.

![Figure 24: The flow visualization results of surface ink trace streamlines [10]](image-url)
The misalignment cases, including case MF and case MC, are comparable to the thin boundary layer; and the baseline cases, including case BF and case BC, are comparable to the thick boundary layer. Near the near the leading edge of the vane, the attachment flow caused by the backward facing step induces inlet flow with a thin boundary layer. Similar to the findings of Graziani et al. [10], the misalignment cases show separation lines closer to the leading edge of the vane and broader endwall crossflows near the pressure side, as seen in Figures 12 and 18. These two studies show consistent results even though different flow speeds and airfoils were used.

A second comparison is for showing similarity related to flow patterns and positions caused by the backward facing step near the vane trailing edge. De la Rosa Blanco et al. [20] noted that the backward facing step maintains endwall flow closer to the endwall and reduces loss, as shown in Figure 25. Both studies were applied at different flow speeds, airfoils, and step heights.

![Figure 25: Loss contours measured at the trailing edge [20]](image)

The two respective comparisons from the loss-coefficient distributions between base line cases and misalignment cases show lower positions of the vortices in the misalignment cases, as presented in Figures 11 and 14 and Figures 17 and 19. The lower positioned flow is also observed on the suction side surface in the flow visualization of the misalignment cases.

The current study gives reasonable agreement with their findings. The reason for the similarity is that the backward facing step leads to lower positions of vortices in the vane passage.
A third comparison to Spencer et al. [16], regarding flow patterns caused by the effects of a converging endwall, is that the separation line is closer to the stagnation point in the hub endwall than the casing endwall. In this study, a separation line is also closer to the leading edge in the converging endwall case than the flat endwall. This is associated with flow acceleration which is induced by converging geometry and causes a thinning of the inlet boundary layer. The present studies show consistent results with their findings in terms of lower flow patterns in the presence of the backward facing step.

The last comparison is related to loss-coefficient distributions between the flat endwall results in the current study and the results of a flat endwall, which was conducted by Nguyen et al.[32]. Both investigations were tested with the vane endwall platforms. As observed in Figures 11 and 26, both cases of flat endwalls show similar loss distribution ranges and positions of loss cores, except for loss patterns in mid-span. Both studies show similar results for loss coefficient distribution.

![Figure 26: Loss-coefficient distribution results for the flat endwall [32]](image)

The following conclusions are drawn from the results of this investigation.

5. **Conclusion**

This study showed the detailed changes on the aerodynamics of secondary flow, which was induced by an upstream, backward-facing misalignment step over the nozzle guide vane.
The reductions in the aerodynamic loss were associated with the thickness of the inlet boundary layer ahead of vane cascade. It was significantly dominant in loss reduction by weakening the horseshoe vortex locally in the vicinity of the vane leading edge by controlling the inlet flow.

With regards to the attenuating of the horseshoe vortex, the flow separation and attachment caused by the backward-facing step induced the local deceleration of inlet flow. The decelerated flow caused a thinned inlet boundary layer, and this significantly affected the loss reduction despite the formation of the additional secondary flows, and the auxiliary vortex induced by the step.

Fundamentally, the geometry and configuration of the vane endwall platform in the cascade passage were considered as the most important factors in reducing aerodynamic losses. Converging endwall reduced losses and positively influenced the effect of misalignment in aerodynamics. The aerodynamic loss for the flow acceleration caused by a converging endwall was relatively marginal for overall loss in the passage.

6. **Recommendations for future work**

Recently, there has been an increasing interest in research for film cooling in gas turbines. As a practical application for actual turbine condition and geometry, it would provide clearer overviews for the effects on secondary flow over nozzle guide vanes due to design changes caused by film cooling.

As regards the backward-facing step, there is a possibility that another type of recessed step may be more representative of real turbine conditions. The addition of the recessed step also affects the secondary flow with the development of the vortices. It would be beneficial to evaluate the influence on the flow field of the recessed step along the guide nozzle vane.
References


Polytechnic Institute and State University, VA.


Appendix A: Test section cascade setup

Test section

The test section consists of the body, two circular windows, and three pieces of the vane endwall platform. The test section body is fabricated out of 6061 aluminum as seen in Figure 27. One circular window is made of polycarbonate, and the other is made of 6061 aluminum. Their dimensions are given in Table 5.

![Figure 27: CAD model of test section](image)

<table>
<thead>
<tr>
<th>Item</th>
<th>Dimension</th>
<th>Magnitude (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Outer Size</strong></td>
<td>Total Length</td>
<td>0.737</td>
</tr>
<tr>
<td></td>
<td>Total Width</td>
<td>0.216</td>
</tr>
<tr>
<td></td>
<td>Total Height</td>
<td>1.22</td>
</tr>
<tr>
<td><strong>Flow Passage Size</strong></td>
<td>Inlet Width</td>
<td>0.152</td>
</tr>
<tr>
<td></td>
<td>Inlet Height</td>
<td>0.381</td>
</tr>
<tr>
<td></td>
<td>Exit Width</td>
<td>0.152</td>
</tr>
<tr>
<td></td>
<td>Exit Height</td>
<td>0.106</td>
</tr>
<tr>
<td><strong>Window Inserts</strong></td>
<td>Diameter</td>
<td>0.508</td>
</tr>
<tr>
<td></td>
<td>Thickness</td>
<td>0.0318</td>
</tr>
</tbody>
</table>
Tunnel design

The assembled test section of the cascade test section has five vanes and four passages as shown in Figure 5. The passage of interest is the center passage between the third and fourth airfoils from the top.

The tip of the pitot-static tube was placed one chord upstream of the leading edge, at the location of mid-span and mid-pitch (x/Cax = -1.0, y/P = -0.5). The inlet total and static pressures were measured by a pitot-static tube.

At the downstream exit, six static-pressure ports were positioned on the surface of the 3D-printed endwall of the center exit passage, as shown in Figure 5. Exit static pressures were taken at the location of these static pressure ports from the leading edge of the vane as given in Table 6.

<table>
<thead>
<tr>
<th>Ports</th>
<th>x/Cax</th>
<th>y/P</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.258704</td>
<td>-1.023421</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td>-1.223775</td>
</tr>
<tr>
<td>3</td>
<td></td>
<td>-1.424129</td>
</tr>
<tr>
<td>4</td>
<td>1.258704</td>
<td>-1.624483</td>
</tr>
<tr>
<td>5</td>
<td></td>
<td>-1.824837</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td>-2.025191</td>
</tr>
</tbody>
</table>
Appendix B: Five-hole probe details

The five-hole probe has a bundle of five tubes; a center tube surrounded by four tubes in the shape of a cross as shown in Figure 28. The pressures are sensed respectively at the five holes located on its tip. The specifications of this probe are presented in Table 7. Additionally, a large portion of the uncertainty came from the alignment of the five-hole probe. This uncertainty was estimated as ±0.25° in both the span-wise and pitch-wise directions.

Figure 28: View of five-hole probe and flow

<table>
<thead>
<tr>
<th>Table 7: Five-hole probe specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Items</td>
</tr>
<tr>
<td>Measurement variables and specified</td>
</tr>
<tr>
<td>- Velocity components ((u, v, w))</td>
</tr>
<tr>
<td>- Flow angles (pitch-wise &amp; span-wise)</td>
</tr>
<tr>
<td>Maximum flow angle deviation from tip</td>
</tr>
<tr>
<td>- Subsonic</td>
</tr>
<tr>
<td>- Supersonic</td>
</tr>
<tr>
<td>Bending Analysis (at Mach 1.2):</td>
</tr>
<tr>
<td>- Maximum deflection</td>
</tr>
<tr>
<td>- Max slope</td>
</tr>
</tbody>
</table>
Appendix C: Exit total pressure data

The case BF

The case MF

The case BC

The case MC
Appendix D: Uncertainty analysis in loss coefficient

The uncertainty analysis was performed by means of the Kline and McClintock’s method [33] for uncertainty of experimental measurement.

The uncertainty in the calculation of loss-coefficient values refers to the averaged experimental values. The level of the uncertainty follows a non-linear relation with the loss-coefficient value according to equation (1).

\[
\omega = \frac{P_{t, in} - P_{t, ex}}{P_{t, ex, M} - P_{s, ex, A}} \quad (1)
\]

When the loss coefficient was applied to the Kline and McClintock’s method, the uncertainty, \( W \) is shown in the following equation (2).

\[
W_\omega = \sqrt{\left(\frac{\partial \omega}{\partial P_{t, in}} W_{P_{t, in}}\right)^2 + \left(\frac{\partial \omega}{\partial P_{t, ex}} W_{P_{t, ex}}\right)^2 + \left(\frac{\partial \omega}{\partial P_{s, ex}} W_{P_{s, ex}}\right)^2}
\]

where \( \frac{\partial \omega}{\partial P_{t, in}} = \frac{1}{P_{t, exit, m} - P_{s, exit, A}}, \frac{\partial \omega}{\partial P_{t, ex}} = \frac{-(1+\omega)}{P_{t, exit, m} - P_{s, exit, A}}, \frac{\partial \omega}{\partial P_{s, ex}} = \frac{\omega}{P_{t, exit, m} - P_{s, exit, A}} \)

By reorganizing equation (2) with the three differential equations above, the equation becomes,

\[
\frac{W_\omega}{\omega} = \sqrt{\left(\frac{W_{P_{t, in}}}{\omega}\right)^2 + \left(\frac{-(1+\omega)W_{P_{t, ex}}}{\omega}\right)^2 + \left(W_{P_{s, ex}}\right)^2}
\]

\[
p_{s, ex}\left[1 + \frac{1}{2} \frac{M_{exit}^2}{1 - M_{exit}^2}\right]^{-1} - 1]
\]

The inlet total pressure was measured using a NetScanner model 98RK, which includes a 9816-2675 module with a 20 psi range, and a 9816-2686 module with a 10 psi range.

The accuracy of the module instrument was ±0.05% full scale, which was ±0.01 psi. The variations in the recorded pressure values are ±0.06 psi over the window of obtained data. The total uncertainty for inlet total pressure can be given as \( W_{P_{t, in}} = ±0.07 \) psi.

For the uncertainty of \( P_{t, ex} \), the same procedures as above were applied to calculate the uncertainty of them, since the same module was utilized for this experiment. The total uncertainty was \( W_{P_{t, ex}} = ±0.09 \) psi for ±0.01 psi of the module accuracy, and ±0.08 psi in the variation of exit total pressures.
Since the exit static pressure is under 2.5 psi, the instrument accuracy of the 9816-2686 module which is ±0.15% full scale, is 0.038 psi, according to the manufacturer's manual. The variation in static pressure between each run was 0.033 psi. Total uncertainty for $P_{s,ex}$ is ±0.071 psi.

Hence it can be seen that the uncertainty for the averaged-loss-coefficient varies from 3.8–6.1%. It can be observed that the uncertainty is inversely proportional to the loss coefficient value. Since an uncertainty of 6.1% $\omega = 0.064$ then $\Delta \omega = \pm 0.004$ and an uncertainty at 3.8% $\omega = 0.2128$ is $\Delta \omega = \pm 0.008$, Conservatively the uncertainty can be given as $\omega_{actual} = \omega_{measured} \pm 0.004$. 