Characterization and verification of a closed loop wind tunnel with a linear cascade and upstream wake generator

Christopher Foreman
Louisiana State University and Agricultural and Mechanical College

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CHARACTERIZATION AND VERIFICATION OF A CLOSED LOOP WIND TUNNEL WITH A LINEAR CASCADE AND UPSTREAM WAKE GENERATOR

A Thesis

Submitted to the Graduate Faculty of the Louisiana State University and Agricultural and Mechanical College in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering

in

The Department of Mechanical and Industrial Engineering

by

Christopher Michael Foreman
BSME, Louisiana State University
May 2013
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Abstract

A closed loop wind tunnel designed to study film cooling was completed in May 2011 along with a removable wake generating device. The test section featured a three blade, four passage linear cascade utilizing the Air Force Office of Scientific Research L1A low pressure turbine blade. The wake generator is unique because its blades are flat plates with round leading and trailing edges instead of circular rods. In this report, the test section of the wind tunnel is characterized and validated through velocity and pressure measurements in the test section. Hot-wire surveys were used to characterize the velocity and turbulence intensity. An in-house designed pressure blade along with Scanivalve pressure sensing equipment was used to acquire the static pressure on the surface of the center blade in the cascade. Incoming velocity results showed the profiles were uniform in both the horizontal and vertical directions. The turbulence intensity in the incoming freestream air was 0.20%. Installing the wake generator had no effect on the incoming velocity profiles. Experimental coefficient of pressure results with the wake generator installed were very close to the AFOSR results and fluctuated slightly when the wake generator was running. However, the overall shift in the coefficient of pressure magnitude as the wakes passed was small. Downstream velocity measurements in the wake of the cascade confirmed periodicity in the cascade. When the wake generator was running, the passing wakes effectively decreased the velocity and increased the turbulence intensity in the cascade wake, and caused the cascade wake to shift towards the suction side. It also increased the width of the cascade wake. Based on these results, the wind tunnel appears to be operating as designed and should produce adequate results in future film cooling measurements.
1 Introduction

1.1 Motivation

A gas turbine is an engine designed to either primarily produce thrust or shaft power. An example of a thrust producing gas turbine or turbojet is shown below in Figure 1.1.

![Figure 1.1 Example of a Turbojet (Source: history.nasa.gov)](image)

The three main components every gas turbine has are the compressor, the combustion chamber, and the turbine. Thrust producing gas turbines usually incorporate a diffuser upstream of the compressor and a nozzle downstream of the turbine. The nozzle is used to convert energy in the flow exiting the turbine into momentum; thus, producing a net increase in momentum in the flow direction. The goal of this research facility is to improve gas turbine efficiency. One way to do this is to increase the turbine inlet temperature (TIT), $T_3$ shown below in Figure 1.2, because it is proportional to the thermal efficiency of the gas turbine.

![Figure 1.2 Diagram of the Brayton Cycle (Source: learnthermo.com)](image)

To increase the TIT, turbine and combustor components need to be protected with special thermal barrier coatings and/or cooled. The primary focus of this research facility is to increase thermal efficiency of gas turbines through better film cooling. Film cooling is a cooling technique used on gas turbine blades. It uses air derived from a compressor stage and ejects it through small holes on the blade surface. The ejected air forms a thin layer of relatively cooler air at and downstream of the film cooling hole. The film
temperature is locally cooler than the combustion gases and acts as a protective layer between the blade surface and the environment. It is this film temperature that researchers are trying to predict and lower. When the film temperature is lower than the hot gases in the turbine section, gas turbines are capable of operating near the melting point of blade materials. However, approximately 20 – 25% of air passing through the compressor is used for turbine component cooling\(^1\). This reduction in airflow out of the compressor affects the amount of power/thrust the turbine can produce. For this reason, it is important to use as little air as possible to cool turbine components. If the amount of coolant used for film cooling is reduced, a further increase in gas turbine output will result. To help designers, this facility is being used to provide a better understanding of the complex fluid dynamics and heat transfer associated with film cooling.

1.2 Background

A closed loop wind tunnel was constructed in May 2011 in the Engineering Lab Annex Building on Louisiana State University’s Baton Rouge, LA campus.

![2D Cross-Section View of the Closed Loop Wind Tunnel](image)

Figure 1.3 2D Cross-Section View of the Closed Loop Wind Tunnel

The wind tunnel will eventually be used to study film cooling on gas turbine rotors and is currently in the validation stage. It is capable of reaching wind speeds up to 55 m/s in the test section; however, it is designed to operate at 50 m/s. The test section consists of a three blade, four passage linear cascade and an upstream wake generating device. The cascade blades are constructed using the 2D airfoil geometry of the Air Force Office of Scientific Research (AFOSR) L1A low pressure turbine blade. The L1A blade is a rotor but in the cascade, it is stationary. The coordinate system, in this case, is attached to the rotor; therefore, the upstream wake generator blades are the ones moving. It is set up like this because stationary blades are a lot easier to study than rotating ones; however, periodicity must exist between the cascade blades for the cascade to be valid. To provide adjustability and the ability to “tune” the flow, tail boards and inlet bleeds were included in the test section design.
The L1A blade is an aft loaded rotor blade designed to operate at low Reynolds numbers around 50,000. For the purposes of this project, the blade operating Reynolds number is 500,000 based on the axial chord of the cascade blades. The blades were designed with an axial chord of 152.4 mm and a span of 304.8 mm. The cascade solidity, the ratio of the axial chord to the blade spacing, is 1 and the flow turning angle is 95°.

![Figure 1.4 2D Schematic of Cascade Test Section with the Wake Generator Installed](image1)

The wake generator is designed to simulate upstream nozzle guide vanes (NGV) in a gas turbine. It uses a conveyor belt built using Intralox series 800 polypropylene slats. The housing for the conveyor seals it from the ambient environment and was constructed using sheet metal. The case is sealed with weather stripping and foam. The foam was installed on the interior of the case to enclose the blades and prevent a path of least resistance between the test section air and the case from forming. The belt consists of forty-nine, 50.4 mm x 152.4 mm slats with seventeen rectangular blades attached to it. A 261.62 mm diameter acetal sprocket connected to a Sew-Eurodrive gear motor drives the belt at speeds up to 2.72 m/s. Figure 1.5 shows a CAD rendering of the wake generator.

![Figure 1.5 CAD Rendering of Wake Generator](image2)

The rectangular blades are constructed out of Nylon and have removable acrylic leading and trailing edges. The blades are 304.8 mm (H) x 136.525 mm (W) and are spaced 152.4 mm apart. The gear motor drives the blades in and out of the test section.
This motion creates periodic wakes that impinge on the downstream rotor blades. Vortical structures are shed from the trailing edge of the blades and are of particular interest to the research in the facility. These vortical structures have been shown to have an adverse effect on film cooling because they increase the perturbations in the shear layer between the film and freestream air flow. These perturbations lead to more mixing between the hot freestream and relatively cooler film causing the blade surface temperature to increase. For more information on the wind tunnel, see Reference 2.

<table>
<thead>
<tr>
<th>Table 1.1 Characteristics of Sew-Eurodrive Gear Motor^2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max RPM of motor</td>
</tr>
<tr>
<td>Power Requirements</td>
</tr>
<tr>
<td>Horsepower</td>
</tr>
<tr>
<td>Gear Reduction Ratio</td>
</tr>
<tr>
<td>Maximum Velocity</td>
</tr>
</tbody>
</table>

1.3 Literature Survey

Researchers are trying to mimic the conditions that exist in a gas turbine with the intention of better understanding how certain ones such as turbulence intensity and passing wakes effect film cooling performance. Those two conditions can have an adverse effect on film cooling due to a passive pulsing effect^3^-^5^. The research facility at LSU is particularly interested in vortical structures shed by the trailing edge of upstream NGV and their effects on downstream film cooled rotors. Upstream NGV are used to direct the incoming flow onto the leading edge of the rotor blades to produce optimum rotor lift. However, this also causes the vortical structures to impinge on the leading edge of the rotor. A rotors leading edge is typically the most heavily film cooling portion because it is the location of the local stagnation temperature. These vortical structures can have detrimental effects on the effectiveness of film cooling jets because they increase turbulent mixing in the shear layer between the film and the freestream^6^.

In order to better understand the effects of an unsteady environment on film cooling jets, references 3-5 created freestream static pressure fluctuations using a system of shutters upstream of the jets. The jets were arranged in a spanwise row of five and angled at 35° to a flat surface on the floor of their wind tunnel. The static pressure fluctuations caused the exit velocity of the jets to fluctuate. Visualizations of the jet exit profile showed a wavy shape hinting at the development of an unstable shear layer between the freestream and jet flows. An increase in mixing between the jet and freestream flows occurred there which is undesirable in a gas turbine because it increases the film temperature.

A few researchers have explored the effects of film cooling in an unsteady environment on turbine blade profiles^7^,^6^. Reference 6 used stationary cylindrical rods upstream of a linear cascade to simulate wakes shedding from upstream NGV and hitting the leading edge of downstream rotors.
In this study, the coolant was injected through a labyrinth seal upstream and along the wall of the cascade. A decrease in adiabatic film cooling effectiveness was observed due to an increase in turbulent mixing caused by the rods. They also used delta wings in place of the cylindrical rods to produce vortices. The delta wings were rotated to angles of 30° and 45° relative to the incoming flow and produced vortices of different sizes. The vortical structures had a detrimental effect on film cooling effectiveness.

Recently, there has been an interest in a new film cooling concept called pulsed film cooling. Pulsed film cooling has the potential to decrease coolant usage and still adequately cool a turbine blade. The idea is to force the film cooling jet in either an on/off or high/low blowing configuration instead of allowing them to steadily blow. In the on/off configuration, the jet would alternate from a zero flow rate to a predetermined high flow rate. The jet alternates from a low flow rate to a high flow rate and, thus, has an average value in the high/low blowing configuration. Because the jet operates in a cyclic fashion, other control parameters such as the duty cycle and pulse rate or pulsing frequency are needed. The addition of these extra control parameters is both good and bad. The increased number of parameters increases the optimization capability of the jets compared to steady film cooling, but it also increases the complexity of the system. However, if it can successfully decrease coolant usage and maintain or surpass the performance of film cooling, it would contribute to increased turbine performance.

Reference 8 was the first to examine active control of pulsed film cooling jets. They forced a single, compound angle jet located at the leading edge of a half cylinder model into a steady (no wakes) environment. The jet was angled 20° to the surface in the spanwise direction, 90° in the streamwise direction, and it was forced in an on/off configuration. All the control parameters previously mentioned were varied in the experiment. The forced jet performed about the same as the steady jet at lower blowing ratios. In some cases, mostly at higher blowing ratios, the forced jet performed better than the steady jets. The increased performance at higher blowing ratios was caused by
mitigating jet liftoff through high frequency pulsing. The high frequency pulsing prevented the forced jet from shutting off producing a low momentum flow during the “off” period of the cycle very similar to the high/low configuration mentioned above.

Reference 9 also investigated pulsed film cooling in a steady environment. They used a single row of cylindrical film cooling jets inclined 35° relative to a flat surface on the floor of their wind tunnel. Again, all the control parameters were varied, and the jets were forced in an on/off configuration. The pulsed jets, however, were less effective compared to the steady jets at the same blowing ratios. The only exceptions were at high blowing ratios in conjunction with high forcing frequencies were jet liftoff was mitigated by the forcing of the jets. They also showed a longer duty cycle increased the effectiveness in most cases.

Reference 10 was the first to investigate the effects of a periodic wake on forced film cooling jets. The addition of the periodic wake is important because it exists outside of the laboratory setting in gas turbines. They used the same geometry as reference 9 with the addition a wake generator upstream of the film cooling jets. The wake generator created wakes using a spoked wheel with detachable cylindrical rods threaded into it.

![Figure 1.7 Schematic of Test Section from Reference 10](image)

The wheel sliced into the wind tunnel at a constant rate. Rods were added or removed to vary the wake passing frequency. Typically, film cooling holes on rotors are of the order as the trailing edge of the NGV; therefore, the cylindrical rods had the same diameter as the film cooling holes. Film cooling effectiveness results were gathered when the wakes impacted as the jets were forced, and when they impacted between jet pulsations. Steady cases and cases without wakes were also presented for comparison. The results were acquired using different control parameter combinations, and the jets were operated in an on/off configuration. In general, pulsed film cooling in tandem with periodic wakes showed decreased performance when compared to a similar film cooling blowing ratio.

1.4 Objectives

The research group at Louisiana State University is interested in further studying pulsed film cooling applications on gas turbine rotors. There are multiple studies underway within the group including a computational study on the experimental facility presented in this report and an experimental jet in crossflow (JICF) facility. There is also
a computational study of the (JICF) facility attempting to produce a reduced order model of a pulsed jet in crossflow.

In this report, the existing cascade facility will be characterized and validated though measurements of the incoming air flow, the pressure distribution along the surface of the blades and the wake of the cascade. The energy spectrum in the frequency domain at the test section walls, test section center, and in the wake of the blades will also be presented. The incoming velocity profiles will be measured to confirm turbulence intensity levels and air flow distributions across the test section. Pressure profiles will be compared with those received from the AFOSR, and the above mentioned CFD study, to validate the construction of the cascade blades. Finally, wake measurements will be used to confirm the existence of periodicity in the cascade.

After the cascade is validated, the wake generator will be installed and the above measurements repeated. The energy spectrum in the frequency domain will also be measured in the wake of the wake generator blades. Phase averaged velocity and pressure profiles will be acquired with the wake generator running to characterize the effects the wake generator has on them.
2 Test Section Designs

2.1 Three Axis Linear Traverse

A three axis linear traverse was designed and built to accommodate constant temperature anemometry (CTA) measurements. The possibility of other uses such as particle image velocimetry (PIV) and infrared thermography (IRT) measurement techniques was taken into account in its design as well. Figure 2.1 demonstrates the traverse, its supporting structure, and other components attached to the supporting structure. The main components of the traverse are shown in Appendix I CAD Drawings.

![2D Schematic of Traverse and Supporting Structure](image)

The supporting structure was made using Futura extruded aluminum framing parts also called TSLOTs. The frame was constructed out of TSLOTS because of the ease of assembly and versatility it offered. Any new part can be face mounted on any TSLOT using a fastener and T-nut. The TS20-40 member and hidden anchor fasteners were used to make the supporting structure frame. Hidden fasteners were chosen over brackets so the least amount of face mounting space was occupied. This left plenty of room for additions to the frame and/or future expansions. The hidden anchor fasteners are also considered Futura’s strongest. The dimensions of the structure are 100 inches (H) x 72 inches (W) x 38 inches (D).

The traverse has two vertical axes coupled with a timing belt. The timing belt is connected to each axis via a sprocket. The sprocket on the right axis is fastened directly to the drive screw and a stepper motor is connected to the sprocket on the left axis. This configuration allows the motor to drive both axes instead of just one resulting in a smooth, lag and jam free motion.

A counter-weight system (not shown) was also implemented to increase the payload of the traverse by supporting the 60 lb. weight of the X and Y axes. Two 15 lb. steel cylinders are suspended on the far side of the traverse using a pulley. The pulley system was designed to double the counter weight so the size of the cylinders was reduced.

The linear stages that make up the individual axes are Velmex Bislides. They are a lead screw and ball design with 2 mm/rev leads. The X, Y and Z axes have travel lengths
of 40, 20, and 50 inches respectively. Each axis supports a 500 in-lb. cantilever load, a 300 lb. normal loading and a 100 lb. thrust load (40 lb. to motor).

The cantilever loading on the X axis was an area of concern because the Y axis extends so far off its centerline as shown in Figure 2.1. To limit this effect, the Y axis was offset by \( x_0 = 3 \) inches. With the Y axis in this position, a moment with a sense out of the page is generated by the weight of the Y axis. Of course, the location the center of mass of the Y axis will shift, but this configuration will still help mitigate the moment induced by the load. The heaviest expected loads are the particle image velocimetry (PIV) cameras which only weigh 10 lbs. max.

![Figure 2.2 Current Setup of X and Y Axes to Reduce Cantilever Loading Near Wind Tunnel Test Section](image)

\[
M = mgx_0 - Fd
\]

**Figure 2.2** Current Setup of X and Y Axes to Reduce Cantilever Loading Near Wind Tunnel Test Section

<table>
<thead>
<tr>
<th>Axis</th>
<th>Axis Weight, lbs.</th>
<th>Motor Weight, lbs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>X</td>
<td>20.4</td>
<td>2.3</td>
</tr>
<tr>
<td>Y</td>
<td>11.5</td>
<td>2.3</td>
</tr>
<tr>
<td>Z</td>
<td>23.9</td>
<td>3.7</td>
</tr>
</tbody>
</table>

Table 2.1 Weight Data for Traverse Slides and Motors

Stepper motors equipped with encoders (no encoder on Z axis) are used to power the axes of the traverse. NEMA 23 Danaher Motion stepper motors (Model # T22NRLG-LDN-NS-00) are used on the X and Y axes. A NEMA 34 Oriental Motors stepper motor (Model # PK296-03BA) is used on the Z axis. National Instruments quadrature encoders (Model # 15T-01SA-1000-N5RHV-F00-CE) are used on the X and Y axis, only. A BEI encoder shipped with the parallel coupled Z axes but was not compatible with the National Instruments hardware. The data sheets for the motors and encoders are available in Appendix II Data Sheets. The National Instruments (Danaher Motion) 3rd party stepper motors were selected because of their robustness and relatively low cost when compared to servo motors. They are also relatively easy to install and set up compared to servo motors. Lastly, they met the sizing requirements of the system.

The sizing calculations were carried out in the following manner. For example, if the Y axis is loaded with an 80 lb. payload, the X axis will support the weight of the Y axis and the 80 lb. payload. From Table 2.1 this adds up to 93.8 lbs. Looking at Figure 2.3a, a sliding force of 100 lbs. requires about a 30 oz-in torque from the motor to move it.
Multiplying the torque by a safety factor of 1.5 gives a new torque of 45 oz-in. Assuming a traverse speed of 10 mm/s (pretty fast), the RPM required from the motor would be about 300 (remember: 2 mm/rev lead screws). From Figure 2.3b, this results in a torque output of about 0.75 N-m or 106 oz-in which is greater than the required 45 oz-in.

National Instruments (NI) hardware and software is used to control the positioning of the traverse. It was chosen because of LabVIEW, which provides the ability to program a stepper axis and a data acquisition system simultaneously. This allowed full integration of the traverse and data acquisition systems. A NI compact RIO 9178 chassis is used as a remote chassis for the traverse. Virtual instruments are created on the host PC.
and deployed on the RIO chassis via an ethernet connection. This takes the computational load off the host PC so it can be used to process data. NI 9512 stepper drive C series modules mounted in the RIO chassis are used to issue move commands to the stepper power drives, monitor limit switches and receive encoder pulses. Danaher Motion P70530 power drives are used to power and relay move commands to the stepper motors. More information about the above equipment can be found on the National Instruments website (NI.com) or the Oriental Motor website (orientalmotor.com).

2.2 Pressure Measurement Blade

A pressure tap blade was constructed to measure the pressure profile on the central test blade and compare it to documented results. The blade was designed in several components to make assembly of the pressure taps and pressure tap tubing easier. Pressure taps were to be located at three spatial locations: 0.25H, 0.50H and 0.75H where H is the height of the blade (from left to right or vice-versa in Figure 2.5). Forty-one taps should be placed at each spanwise location totaling 123 taps.

Several iterations were attempted before the final design was realized. The next few paragraphs of this section will cover these attempts to provide the reader with a basis for the rational of the final design. It is also hoped this section will provide the reader with helpful insights to further his/her own design(s).

The original design for the blade was very similar to the final design displayed in Figure 2.5. The blade design itself changed very little throughout the design process. The main difference between the first iteration and final design is twofold. The first iteration consisted of 7 components: the 0.167H cap spacer; three, 0.167H pressure tap sections; two, 0.083H spacers; and the 0.167H base spacer. The final design had 9 components: the .083H cap spacer; three, 0.167H pressure tap sections; four, 0.083H spacers, and the 0.083H base spacer. Lastly, the first iteration had thinner walls, in general, then the final design. The final design wall thickness was double that of the first design for one reason: manufacturing difficulties (get to that in a second).

The blade was first manufactured using an in house 3D printer. The printer was a Dimension Elite Series printer which used a stereolithography technique. The parts were built from an ABS plastic like material that was fed into the machine and melted. The melted plastic was ejected out of a small nozzle attached to a 3 axis traverse mechanism. The mechanism traversed back and forth constructing the part layer by layer. The layer resolution on the machine could be set to either 0.01” or 0.007”. The layer resolution represents the thickness of each layer the printer puts down. The printer also had a full or sparse density build capability. The capabilities of the printer were not up to the standards the writer was looking for. The parts tended to be brittle and the blade parts were typically warped due to the thermal expansion/contraction the plastic layers experience during build. However, the rational for its use was the relatively low cost of each part.

The second iteration involved using the same design and manufacturing technique with one difference. To correct the warping of the blade profile, the parts were printed oversize on the Dimension printer and machined to size on a 3 axis CNC mill. A few things happened when this was attempted. The first problem was the parts that were printed were built with the sparse density setting (hence the discovery of the sparse density setting). The second problem was supporting the parts on the mill table. There
was no easy way to support them and because the printer accuracy was low, it was hard for the machinist to get a good starting point for the milling process. So, a few zero points were incorporated in a later design but the low accuracy of the printer produced results like the ones shown in Figure 2.4 below.

![This is the desired part](image1)
![This is the actual part](image2)

An attempt at machining using the center as a zero point would produce this

Figure 2.4 Example of Manufacturing Woes Encountered on Second Iteration

The parts that resulted from machining typically had varying wall thicknesses themselves and were no better than the printed parts.

For the third attempt, the spacer parts were cut with a water jet from a solid block of aluminum then machined to size using a 3 axis CNC mill. The thickness of the spacer walls was oversized by 1/8”. The internal surfaces were left “rough” but the external surfaces of the spacer parts were milled for a smooth surface. However, the same problem occurred; the walls of the resulting blade parts had varying thicknesses. Because the blades were machined out of rolled aluminum plate, which has a lot of internal stresses, the parts warped when cut due to stress relief of the material. Also, the blade walls were very thin (~0.030 in.) which makes them prone to warping during the machining as well so even if stress relieved plate was used, the same problem was likely to occur.

Finally, the entire blade was redesigned. The pressure tap blade was modified to allow for thicker spacer walls. Internal ribs were incorporated on the spacer parts to increase the rigidity of the walls so they didn’t warp during machining. More holes were added to promote a more stable part so it didn’t move when it was machined. Also, the spacer heights were limited to 1 inch. This was so the mill could cut the external surface in one pass with minimal deflection of the milling bit. The next few paragraphs talk more about the final design and the manufacturing process used to make and assemble it.

The final blade, depicted in Figure 2.5, is an assembly of nine parts spanning 12 inches. The pressure tap sections are located at 0.25H, 0.50H and 0.75H along the blade height. Each pressure tap section has forty-one pressure taps built into it totaling one hundred twenty three pressure taps. The large number of pressure taps enabled a high resolution capture of the pressure profile along the length of the blade. Having the taps
spaced every three inches allowed the straightness of the blade to be validated by comparing the pressure acquired by each section.

Figure 2.5 Labeled Parts of Assembled Pressure Tap Blade

Two Scanivalve DSA3217 differential pressure scanners were used to measure the pressure profiles. Each DSA3217 has sixteen pressure channels, a reference channel, and an onboard A/D converter to directly output the pressure in engineering units via an ethernet connection.

The blade was broken into nine parts for both manufacturing and assembly purposes. The base spacer, cap spacer, and spacer sections were manufactured at LSU in the Mechanical Engineering Machine Shop out of Al6061-T6 extruded bar. Because of its complexity, the pressure tap sections had to be manufactured using a rapid prototyping process called stereolithography (SLA). This process uses a vat of UV curable liquid to build the part layer by layer. Basically, a mold of the part sits on a vertically traversable table. The part is lowered into the vat until a layer of liquid of a specified thickness sits on the top surface of the mold. A UV light passes over the part and cures the layer of liquid on the surface of the part. The part traverses farther down and the process is repeated. For the pressure tap blade, an acrylonitrile butadiene styrene (ABS) like material called Accura 55 (details in Appendix II Data Sheets) was used. The layer resolution was either 0.004” or 0.003”. GPI Prototype and Manufacturing Services, Inc. located in Lake Bluff, IL made the pressure tap sections. After the parts were finished, GPI used their proprietary post processing routine to put a smooth finish on the blade. The process included bead blasting, sanding, and priming (optional). The finish was equivalent to the machined aluminum spacer parts.
The pressure tap section of the blade was especially difficult to design and manufacture. Looking at Figure 2.6, there are tunnels through the inside of the blade from the pressure taps on the surface to the holes labeled “surface tap output locations”. The SLA manufacturing method was chosen because of these tunnels and the complex manufacturing challenges they imposed. Stainless steel tubulations were inserted into the surface tap output locations and connected to vinyl tubing that went to the inputs of the DSAs. This section was designed to have a height of 2 inches so that any imperfections at the mating interface with the other blades would be far away from the pressure taps.

The other components of the pressure tap blade were easier to manufacture but also imposed their own unique complexities. For example, the spacer blade needed a thin outside wall to allow the tubulations to fit inside of it but strong enough to avoid warping or stress relief during the manufacturing process (see Figure 2.7).

Ribs were also put into the design of the spacer sections to prevent flexing of the walls during manufacture which was a problem with some of the old designs. Keep in mind that any increase in the wall thickness of the spacers automatically induced an increase in the wall thickness of the pressure tap section. This becomes problematic because there is not enough room to put the output holes linked to the taps near the leading and trailing sections. Instead, those outputs had to be located on the inside of the pressure tap section blade as shown in Figure 2.6.

To ensure proper mating and alignment of the blade, a square lip with a height and width of 0.0625 inches was designed. The lip wrapped around the sections starting at about 0.5 – 1.0 inches from the leading and trailing edge of each section. Looking at Figure 2.5, the cap spacer had a male lip only; the base spacer had a female lip only; the parts in between had a female lip on the left side and a male lip on the right side. The pressure tap blade had a threaded rod running through its center and was secured in the section on both sides using a nut and washer. This causes the parts to compress onto each other and closes any gaps between the blade sections.
Table 2.2 Location of Pressure Taps as Percent of Axial Chord (X/Cₜ)

<table>
<thead>
<tr>
<th>Tap #</th>
<th>% of Cₓ</th>
<th>Tap #</th>
<th>% of Cₓ</th>
<th>Tap #</th>
<th>% of Cₓ</th>
<th>Tap #</th>
<th>% of Cₓ</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>19</td>
<td>65</td>
<td>1</td>
<td>10</td>
<td>71</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>20</td>
<td>67.5</td>
<td>2</td>
<td>30</td>
<td>76</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>30</td>
<td>21</td>
<td>70</td>
<td>3</td>
<td>30</td>
<td>81</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>30</td>
<td>22</td>
<td>72.5</td>
<td>4</td>
<td>30</td>
<td>86</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>30</td>
<td>23</td>
<td>77.5</td>
<td>5</td>
<td>30</td>
<td>91</td>
<td></td>
</tr>
</tbody>
</table>

2.3 Film Cooling Blade

The film cooling blade is shown in Figure 2.8 below.

![Film Cooling Blade Diagram](image)

Figure 2.8 (Top) Suction/Pressure Side Views, (Middle) Left/Right Side Views, (Bottom) Left/Back View w/ Hidden Lines Shown of the Film Cooling Blade
The film cooling blade was designed to accommodate future measurements using PIV and IRT techniques. The blade is mounted inside the test section using a ¼"-20 threaded rod and two ¼"-20 screws. The rod is secured on both sides of the blade with nuts and washers. The screws connect with threaded inserts on the far side face of the blade as shown in Figure 2.8. The blade has thirty 1/8” diameter film cooling holes connected to an internal plenum. The holes are angled 35° to the local tangent of the surface and are spaced 3/8” laterally. The length to depth ratios of the holes are listed in Table 2.3 assuming a 1/8” diameter. The internal plenum is connected to an air supply via a ¼” NPT fitting and is completely isolated from the other components of the blade such as the thru-rod hole and the threaded insert holes as shown in Figure 2.8.

The blade was manufactured using the same process, material and company as the pressure tap sections. The manufacturing process produced undersized film cooling holes with an average diameter of 0.118 inches.

### Table 2.3 Locations and L/D Ratio of Film Cooling Holes

<table>
<thead>
<tr>
<th>Row/Side</th>
<th>% Chord Location (X/C&lt;sub&gt;x&lt;/sub&gt;)</th>
<th>Length to Diameter Ratio, L/D</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction</td>
<td>1</td>
<td>12.5</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>29.1667</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>50</td>
</tr>
<tr>
<td>Pressure</td>
<td>1</td>
<td>16.6667</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>33.3333</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>50</td>
</tr>
</tbody>
</table>

#### 2.4 PIV Side Blades

The PIV side blades were built to accommodate future PIV measurements in the test section using the film cooling blade. There are two of them, one on either side of the central test blade in the cascade, and they are manufactured in five separate parts as shown in Figure 2.9 below. The plastic base spacer and acrylic sections are optically transparent to allow a laser sheet to pass through them. The blade was designed to allow this laser sheet to enter from the left of the plastic base spacer, travel through the blade to about the mid-height of the blade and then be redirected 90° through the pressure side of the bottom blade (suction side of top blade) to illuminate the film cooling holes on the film cooling blade. Refer to Figure 2.10 for an illustration.

![Figure 2.9 Assembled PIV Side Blade](image)
Figure 2.10 Illustration of PIV Side Blade Operation w/ PIV Optics. The illustration on the right is a left side view of “A”

The laser sheet enters from the near wall of the test section, goes through the plastic base spacer, impact a PIV mirror located on the inside of the blade and is redirected 90° through the surface of the blade. This allows the laser sheet to illuminate a 2D slice of the film cooling blade. The total blade height is 12” but the individual part heights vary. Refer to Table 2.4 for the length of each part.

Table 2.4 PIV Side Blade Part Heights

<table>
<thead>
<tr>
<th>Part</th>
<th>Height (inches)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plastic Base Spacer</td>
<td>1</td>
</tr>
<tr>
<td>Plastic Spacer</td>
<td>3.25</td>
</tr>
<tr>
<td>Acrylic Section</td>
<td>3.5</td>
</tr>
<tr>
<td>Cap Spacer</td>
<td>1</td>
</tr>
</tbody>
</table>

There were several considerations that drove the design of the blade including clearance inside the blade, laser sheet width, and the area of interest on the film cooling blade. Because the mirror had to be a certain width to fit inside the blade, the laser sheet expansion angle was also limited. It was desirable to have the laser sheet stretch across the entire width of the mirror. If the mirror width is equal to the width of the laser sheet when it impacts the mirror, w, and the distance the laser sheet traveled before impacting the mirror, L, is known, the expansion angle, θ, could be determined for different values of w and L (see Figure 2.11). Due to the constraint on the width and height of the PIV mirror, a right angle mirror with a width of 5 mm and a leg size of 5 mm was chosen. However, the actual width of the mirror is 15 mm because the “mirror” inside the blade is actually an assembly of three mirrors. The mirror was sold by Thor Labs and the part number is MRA05-M01. Based on w = 15 mm, the expansion angle of the laser sheet is estimated to be θ = 2.4° assuming L = 7 inches.
The acrylic center section was manufactured with a height of 3.5 inches so that all five film cooling holes could be examined. The total width of the area of interest is 2.25 inches starting 3/8” outside the first hole and going to 3/8” outside the last hole. All parts were manufactured using the SLA process except the cap spacer which was made in house in the ME machine shop using Al6061-T6. The 3.25” plastic spacer was made using Accura 55. The plastic base spacer and acrylic sections were made using Accura 60. Accura 60 is a polycarbonate like material designed to be transparent. After production, the Accura 60 part is post processed with GPI’s proprietary method to make the part transparent. The Accura 60 data sheet is located in Appendix II Data Sheets.

2.5 PIV Mirror Traverse

To study different slices of the film cooling jets or different rows of film cooling jets, or to adjust the laser sheet, the previously mentioned PIV mirror will need to be moved inside the PIV side blades. To do this, a small traverse mounted outside of the blade and test section was designed to accommodate translation and rotation about one axis. The rest of the adjustments can be made using another traverse called the “Laser Sheet Generator Optics Traverse” discussed in the next section. There were size constraints and location constraints on the traverse because of the internal size of the PIV side blade. These restraints required the traverse controls to be operated outside of the wind tunnel and they also required those controls to be relatively compact.

To force the mirror to translate and rotate about one axis, the small slot shown in Figure 2.12 below was designed into the acrylic section of the PIV side blades.
This slot was designed in accordance with the mirror carriage shown below in Figure 2.13. The Thor Labs mirrors previously mentioned fasten to the top of the carriage using glue. The carriage is made out of Accura 55, the same material used to make the pressure tap section of the pressure tap blade, and the PIV side blades. It was also manufactured using the same company and stereolithography technique.

The mirror carriage has an internal 4-40 thread that allows it to attach to the end of the drive shaft. Figure 2.14 below shows a section of the traverse device with each component labeled.
The drive shaft is an 8-32 x 12 inches long threaded rod with a 4-40 x 0.75 inches thread at one end of it. It has a small keyway 0.065” (W) x 0.03125” (H) starting 1 inch from the left end (Figure 2.15) and extending 4.5 inches down the shaft.

![Figure 2.15 Side/Front View of the Drive Shaft used in the PIV Mirror Traverse](image)

A set screw with a 0.0625” diameter flat top Nylon piece attached to the end of it sits in this keyway and prevents the rod from rotating while the drive nut turns. The set screw is threaded through the inner housing part. There are two set screws set apart by 90° which thread through the outer housing and intersect with the inner housing as shown below in Figure 2.16. These two set screws stop the inner housing from rotating when engaged.

![Figure 2.16 Centerline Section of Traverse Depicting Set Screws Used to Control Rotation](image)

To break it down further, the “cap spacer mount” secures with the cap spacer utilizing a 1/16” NPT connection. The “outer housing” secures with the other side of the cap spacer mount through a 1/16” NPT connection on the opposite side. The “inner housing” is connected to two ball bearings mounted inside of the “outer housing”. It is free to rotate unless the set screws are engaged. The “drive shaft” slides inside the “inner housing” and is threaded through the “drive nut”. The small set screw threaded through the “inner housing” keeps the “drive shaft” from rotating. The “drive nut” is also connected to the “inner housing” by a bearing compression fitted inside the nut. So, when the “outer housing” set screws are engaged the only thing allowed to rotate is the “drive nut”. The threads of the “drive nut” force the “drive shaft” to translate left or right (in Figure 2.16). Now, as mentioned before the mirror needs to be able to rotate as well. Rotation is possible when the two “outer housing” set screws are disengaged. This allows the “inner housing” to rotate and since the “drive shaft” and “inner housing” are coupled
through the small set screw, the “drive shaft” also rotates. To keep the “drive shaft” from translating when it is rotated, the “drive nut” is secured to the “inner housing” via two set screws set apart by 90° and threaded through the nut (not shown in Figure 2.16).

![Figure 2.17 Illustration of Installed PIV Side Blade and PIV Mirror Traverse Mechanism](image)

### 2.6 Laser Sheet Generator Optics Traverse

In order to account for any misalignment of the PIV mirrors, another traverse was built to provide fine adjustment to the sheet generator optics. The sheet generator optics consist of a set of mirrors to transform an incoming laser beam into an expanding parabolic sheet similar to the one shown below in Figure 2.18.

![Figure 2.18 Simple Illustration of How the Laser Sheet Generator Optics Work](image)

To acquire 2D or stereoscopic PIV measurements, the laser sheet and camera have to be very close to perpendicular. How close depends upon the nature of the measurements and the facility. Because the sheet or camera or both will almost always need to have slight adjustments made, the traverse was specifically designed to be able to make small, precise adjustments. The traverse has the ability to move the optics up/down, left/right, and rotate about two axes as shown below in Figure 2.19. When mounting, it can be positioned in or out of the page. It is mounted on the traverse support frame as shown in Figure 2.1, and the entire assembly can be adjusted up or down depending on the location of the measurements.
The traverse basically accounts for any adjustments the PIV mirror traverse cannot make. The two translation stages are Thor Labs PT1 manual translation stages. They are driven with a micrometer and feature stainless steel ball bearings for long life and precise movement. The Z axis is mounted vertically and has a max load carrying capability of 20 lbs. Rotation about the X axis is generated using a Thor Labs PR01 precision rotary stage. The PR01 allows continuous 360° rotation while the set screw shown in Figure 2.19 is loose and fine adjustment using the micrometer drive, in increments of 5 arcminutes, when it is tightened. Finally, rotation about the Z axis is created using a Thor Labs MSRP01 mini rotation stage. It has continuous 360° rotation and a set screw to lock it in place. The laser sheet generator optics holder mounts to this stage using the “Base Plate Adaptor” shown in Appendix I CAD Drawings.

2.7 Wake Generator Blades

The last design presented are changes to the wake generator blades. The previous student and designer of the wind tunnel, Jean-Philippe Junca-Laplace, designed and built a set of wake generator blades with the body constructed out of MDS filled Nylon (see Figure 2.20 for a depiction of the blades). The blades warped because of stress reliefs when they were machined. To correct the warping problem, a new design was proposed that consisted of the same core concepts as JP’s design:

- a central blade piece
- removable leading and trailing edges
- an Aluminum intermediate plate between the central blade piece and the wake generator slats
- threaded inserts press fit into the slats to secure the intermediate plate
However, instead of using Nylon for the central blade part, it was machined out of stress relieved Aluminum.

Figure 2.20 Labeled Parts of the Old Design for Wake Generator Blades

A concern for the Aluminum body design was weight especially because of the high acceleration/deceleration the blade experiences when making a turn on the wake generator. Aluminum is about 1.5x denser than Nylon so the blades are roughly 1.5x times heavier than the original Nylon parts. Figure 2.21 below shows an illustration of the turn the blades have to make going around the wake generator drive sprocket.

Figure 2.21 Illustration of a Wake Generator Blade Making a Turn around a Sprocket Top View (left) and Front View (right)

The highest loading will be on the bottom turn because the normal acceleration and gravity are both in the downward direction. The normal acceleration was calculated to be $4.23 \text{ m/s}^2$ based on a belt velocity of 1 m/s and the center of mass of the blade located 9.3 inches from the sprocket center. This normal acceleration, combined with the component of gravity in the same direction, gives a force of 2.65 lbs. in the direction normal to motion. There are other load considerations to be concerned about especially vibrations. To combat vibrations and to help keep all fasteners tight, Loctite 243 blue was applied to all fasteners connecting the intermediate plate to the threaded inserts. This compound is not permanent and is removable.

As mentioned earlier, the wake generator blades are made in several parts. Figure 2.22 below is a blown up view of the entire assembly.
The new blade has essentially the same parts as the old one with a few exceptions. The main differences are the number of fasteners used and the material used for the central blade part. The old design also only had three internal pockets machined into it. These pockets extended into the central blade piece and left a 1/16” thick back wall. The new design has six pockets that extend all the way through the central blade part to prevent warping. Three, 10-32 fasteners and two, 3/16” dowel pins are still used to fasten the central blade part to the intermediate plate. The leading and trailing edges are fastened to the central blade part using five, 6-32 fasteners instead of the previous three. A 1/8” dowel pin was also added at the top and bottom of each edge to help with alignment. The 6-32 fasteners are made from Nylon to preserve the threads on the polycarbonate leading and trailing edges. Eleven, 4-40 fasteners instead of the previous six are used to keep the top and bottom plates firmly attached to the central blade part. The exact dimensions of all of these parts can be found in detailed drawings section located in Appendix I CAD Drawings.

These parts were manufactured in the Chemical Engineering Machine Shop located in the Chemical Engineering building near the co-generation power facility on the Louisiana State University Baton Rouge campus.
3 Characterization of the Linear Cascade without the Wake Generator

Pressure and velocity results are presented to characterize the test section inlet flow and the linear cascade. Velocity profiles are acquired at the test section inlet and downstream of the cascade to verify cascade periodicity. Pressure profiles are captured on the central test blade and compared to known results. The central test blade is designated as the second blade in the three blade cascade shown in Figure 3.1 below.

Velocity profiles were acquired at slot s0 in both the vertical (Z) and horizontal (Y) directions; s2 in the direction along the slot and in the horizontal direction; and at s3 in the direction along the slot only. A TSI IFA 300 constant temperature anemometer (CTA) and hot film/wire probe were used to acquire the velocity profiles with the 3 axis linear traverse.

Pressure profiles were recorded using two Scanivalve DSA 3217 Pressure Scanners and the “Pressure Measurement Blade”. A Pitot-static tube and thermocouple were used to monitor the velocity and temperature during the pressure acquisitions.

3.1 Fan Blade Pitch and Test Section Inlet Velocity Correlations

Before any tests were run, the inlet velocity was correlated to the control pressure used for the fan blade pitch control (see reference 2). The Pitot-static tube and thermocouple were also calibrated using the CTA system.
The incoming velocity versus pitch control pressure is plotted below in Figure 3.2. The Pitot-static tube and thermocouple were calibrated by adjusting the coefficient used to convert the measured voltage to pressure, and the thermocouple reading until it matched the CTA reading. Air density was calculated from the temperature and used to calculate the air velocity from the dynamic pressure. The pitch control pressure signal for both tests was varied from 7.0 psig to 10 psig and then from 10 psig to 7 psig. The CTA readings were acquired at Z/L = 0.50, and Y/H = 0.50 at slot s0 in Figure 3.1. The results from both acquisitions are plotted in Figure 3.2.

![Graphs showing velocity and temperature vs. pitch pressure](image)

**Figure 3.2 Test Section Inlet Velocity (a) and Temperature (b) vs. Fan Pitch Control Pressure**

At the operating velocity, 50 m/s, the pitch pressure was about 8.30 psig. Hysteresis occurred coming back down from 10 psi in both measurement techniques. The hysteresis is most likely occurring because of a response delay in the wind tunnel. The temperature in the wind tunnel reacts very slowly and takes a long time to equilibrate. Air density is affected by the temperature and will cause a change in velocity because of the closed-circuit nature of the wind tunnel (fixed mass flow rate). If the temperature were given more time to equilibrate, the hysteresis would mostly likely decrease.

### 3.2 Spectra and Velocity Profiles

The energy spectrum at all four walls and the center of the test section was measured in slot s0 and presented in Figure 3.3. Time records were captured using a 100 kHz sample rate and a 50 kHz low pass filter for 100 s. A discrete fast Fourier transform of the time record resulted in a frequency resolution of 0.02 Hz. The original and the smoothed spectrums at each location are presented first, followed by a comparison of all the locations.
In all cases, a peak occurred at ~15 Hz. At the center of the test section, a peak occurred at 470 Hz as well. A more distributed set of peaks also occurred from 200 – 300 Hz.
Hz in the center of the test section. In Figure 3.3f, the energy in the boundary layer near the walls of the test section was higher than the center of the test section. This is expected because the turbulent fluctuations, which are directly related to energy dissipation in the flow, are not as large in the center of the test section.

To further investigate the peaks in energy, the natural frequency of the wind tunnel ducts, and test section walls were computed using Equation 1 below. Table 3.1 shows the resulting frequencies.

\[
\omega_n = \frac{a_\infty}{2L_c}
\]

Equation 1

The sound of speed is \(a_\infty\) in Equation 1, and \(L_c\) is the characteristic length of the square duct. The speed of sound was calculated assuming an ideal gas and ignoring compressibility effects for a temperature of 25°C. A characteristic frequency based on the freestream velocity and the characteristic length of the ducts was also calculated using Equation 2 below.

\[
f = \frac{U_\infty}{L_c}
\]

Equation 2

In Equation 2, \(U_\infty\) is the freestream velocity. For the upstream wind tunnel ducting, the freestream velocity was calculated based on the mass flow rate through the test section at slot s0, and the cross sectional area of the ducts. Since the boundary layers at slot s0 make up only about 5% of the cross sectional area, they were ignored when calculating the mass flow rate. The freestream velocity through the ducts was estimated to be about 8 m/s.

Table 3.1 Results of Frequency Analysis on Wind Tunnel Ducting and Test Section Walls

<table>
<thead>
<tr>
<th>Location</th>
<th>Characteristic Length, (L_c) (m)</th>
<th>Natural Frequency, (\omega_n) (Hz)</th>
<th>Characteristic Frequency, f (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upstream duct</td>
<td>0.9652</td>
<td>180</td>
<td>10</td>
</tr>
<tr>
<td>Bottom &amp; Top Test Section Wall</td>
<td>0.3048</td>
<td>570</td>
<td>164</td>
</tr>
<tr>
<td>Left &amp; Right Test Section Wall</td>
<td>0.4953</td>
<td>350</td>
<td>100</td>
</tr>
</tbody>
</table>

Based on the results in Table 3.1, it appears the 15 Hz peak may be coming from the wind tunnel ducting and the 470 Hz peak may be coming from the test section walls. The “hump” from 200 – 300 Hz may also be coming from a combination of the natural frequency and the characteristic frequency of the test section walls.

The autocorrelation coefficient, defined as Equation 3 below\(^{11}\), was calculated before each measurement.

\[
r_{11} = \frac{u_1(x_1, t_1)u_1(x_2, t_2)}{\sqrt{u_1^2(x_1, t_1)}\sqrt{u_1^2(x_2, t_2)}}
\]

Equation 3
The integral time scale was approximated from the results of the autocorrelation coefficient, and the velocity profiles were acquired with the corresponding sampling frequency. To ensure proper sampling, all estimated integral time scales were multiplied by two.

The energy spectrum of the central test blade wake was also investigated at different angles of angles of attack ranging from +2.5° to -2.5°. A velocity time record was acquired at the position of minimum velocity for all three cases using a 100 kHz sampling frequency with a 50 kHz low pass filter for 100 s. All cases were acquired twice; once at an earlier time with a 10 kHz sampling rate, and again at a later time with a 100 kHz sampling rate. However, only the +2.5° case is presented because the other two cases showed no change.

A small peak occurred at about 15 Hz in Figure 3.4 for the 0° and -2.5° angle of attack cases. This peak is the same peak shown in Figure 3.3a-e. For the +2.5° [2] case, a hump occurred from 200 Hz to 500 Hz. This hump or peak in the energy spectrum is possibly the wake shedding frequency of an unstable or separated boundary layer. The supposed reason for the +2.5° wake to produce two different energy spectrums is an error in the blade position. It is believe the case labeled “+2.5 AOA” was actually at an angle of attack ~ 2°. However, the fact that the hump disappeared when the angle of attack was slightly decreased shows the blade becomes very sensitive when the angle of attack is increased to ~ 2°.

The velocity and turbulence intensity are grouped based on slot position and are presented in the next few pages. The velocity and turbulence intensity for each slot were plotted on a single figure. Error bars were calculated from the statistics of the each velocity time record using Equation 4. In Equation 4, $U_{std}$ is the standard deviation of the population, $N$ is the number of samples acquired, and $U_{uns}$ is the resulting confidence interval.

$$U_{uns} = U_{std} \cdot \frac{1.96}{\sqrt{N}}$$  \hspace{1cm}  \text{Equation 4}
The velocity and turbulence intensity were plotted for slot s0, s2 and s3 in Figure 3.5 - Figure 3.7. The temperature profile versus the vertical direction is shown in Figure 3.9 for slot s0. The normalized position was defined as the x, y or z position divided by the total length of the section. For slot s0, the length was measured to be 494 mm and the width 306 mm. The length of slot s2 was 610 mm and the width was 306 mm. For slot s3, the length was 608 mm. In Figure 3.6 and Figure 3.7, the term “slot” corresponds to the direction along the slot and was calculated using the following equation.

\[ s = \sqrt{(X_{\text{max}} - X)^2 + (Z - Z_{\text{min}})^2} \]  

Equation 5

![Figure 3.5 Mean Velocity and Turbulence Intensity at Slot s0](image1)

![Figure 3.6 Mean Velocity and Turbulence Intensity at Slot s2](image2)

![Figure 3.7 Mean Velocity and Turbulence Intensity at Slot s3](image3)

![Figure 3.8 Comparison of Blade Wake Profiles at Different Angle of Attack](image4)
Figure 3.9 Mean Temperature vs. Vertical Direction at Slot s0

Figure 3.5 showed a uniform velocity of 50 m/s across about 95% of the cross-section in the vertical direction. The horizontal direction showed much of the same except with a slight drop of about 1 m/s in the horizontal direction from Y/H = 0.8 to 0.95. This phenomenon is believed to be the result of the mate between the removable near wall and the settling chamber exit in the test section. The turbulence intensity in the freestream was about 0.20% for both cases.

Figure 3.6 indicated the incoming air flow at slot s2 was beginning “feel” the effects of the leading edge of the cascade blades. The velocity varied from 50 m/s upstream of the leading edge to 53 m/s between the leading edges of the cascade blades. The variance between the velocities at the leading edge, s/S = 0.25, 0.50 & 0.75, of the blades was approximately 1% of the freestream velocity. In the horizontal direction, the velocity was uniform from Y/H = 0.05 to 0.65. Beginning at Y/H = 0.65, the velocity dropped to about 49 m/s and stayed there until Y/H = 0.95. The turbulence intensity was about 0.20% in the freestream again for both cases.

The wake profile in Figure 3.7 had a periodic shape as hoped. The minimum velocity in the wake of each blade varied from 62 m/s to 59 m/s, from s/S = 0.25, 0.50 & 0.75. In between the blades, the velocity started at about 73 m/s then decreased in a curved, concave fashion to about 70 m/s on the edge of the blades wake. The turbulence intensity downstream of the cascade was approximately 0.25% in between the blades and peaked at about 2% downstream of the central test blade. The turbulence intensity downstream of the two side blades was about 1.5% each. The maximum turbulence intensity at the pressure side and suction side walls was about 10% and 2%, respectively.

The effect of altering the angle of attack on the central test blade was also investigated. Figure 3.8 showed decreasing the angle of attack from 0° to -2.5° decreased the velocity downstream of the trailing edge by about 1 m/s and, in general, decreased the suction side wake velocity by about 2 m/s; the pressure side wake was reduced by ~1 m/s. It also shifted the position of minimum velocity slightly to the left. When the angle of attack was increased from 0° to +2.5°, the minimum velocity in the wake was decreased by approximately 25 m/s. This increase in angle of attack also created a wake about 350% larger than the 0° wake. A second measurement of the wake at a later time,
shown in purple, produced a different result. In this case, the wake minimum velocity was about 15 m/s lower than the 0° angle of attack case. This result most likely occurred because the blade was not at a +2.5° angle of attack when the measurement took place, but instead was at an angle of attack slightly lower (~2.0°). However, this is still useful because it shows the blade is very sensitive to an increasing angle of attack. (See above discussion about Figure 3.4)

In Figure 3.9, the temperature was shown to stay at about 24°C from Z/L = 0.20 to 0.95. Starting at Z/L = 0.05 and extending to Z/L = 0.20, the temperature rose from 23°C to 24°C. This temperature rise is most likely due to an increase in the overall, bulk temperature of the air flow in the wind tunnel.

3.3 Pressure Profiles:

Pressure profiles were acquired using the in-house designed pressure tap blade and two Scanivalve DSA 3217 Pressure Scanners. The pressure scanners had two different ranges: one from 0 – 10 inches of H2O and one from 0 – 1 psid. The accuracy of the scanners is 0.02 inches of H2O for the 0 – 10 inches H2O range and 0.0012 psi (0.0332 inches H2O) for the 0 – 1 psid range. Based on these values, the estimated uncertainty in C_p from the measured values was 2%. For each acquisition, the velocity and temperature of the incoming flow was acquired using the previously mentioned Pitot-static tube and a type K thermocouple. They provided a way of checking to insure a significant deviation in the velocity did not occur during a test. The total pressure and static pressure ports on the Pitot-static tube were simultaneously acquired using the DSA to provide a means to calculate the coefficient of pressure using Equation 6.

\[ C_p = \frac{(P - P_{ref}) - (P_\infty - P_{ref})}{(P_o - P_{ref}) - (P_\infty - P_{ref})} \]  \hspace{1cm} \text{Equation 6}

In Equation 6, P is the measured local static pressure on the blade surface; P_o and P_\infty are the incoming total and static pressures, respectively. The reference pressure, P_{ref} for all cases was the room barometric pressure. A static bottle constructed out of 3” PVC pipe was used to provide a stable reference pressure.

The AFOSR supplied pressure data in the form of a non-dimensional pressure, normalized by the incoming (entering the test section) total pressure. This normalized pressure was converted to a coefficient of pressure to compare with the experimental results. A k-\epsilon, computational fluid dynamics (CFD) simulation was also run using the cascade geometry with a blade axial chord of 6 inches (the AFOSR simulation used C_x = 8.5 inches). The supplied data for both simulations was converted to a coefficient of pressure (C_p) using the atmospheric conditions in the simulation. A plot comparing the three C_p’s is below in Figure 3.10.
Figure 3.10 $C_p$ with AOA = 0° and $U_\infty = 49.04 \pm 0.11$ m/s on the Cascade Blade Compared to an AFOSR Simulation at $Re_{Cx} = 500,000$ and a k-\(\varepsilon\) 3D Simulation at $Re_{Cx} = 500,000$

In Figure 3.10, AFOSR represents the pressure data supplied by the U.S. Air Force Office of Scientific Research and CFD denotes the 3D, k-\(\varepsilon\) simulation. The experimental coefficient of pressure ($C_p$) shown has decent agreement with the AFOSR results but better agreement with the CFD results. The largest deviations occur on the suction side of the blade where the blade has high curvature, and at the trailing edge of the blade. The most likely cause of the discrepancies is the test section pressure bleeds. The bottom bleed was opened to balance the flow at slot s2 and slot s3. The bleed is reducing the flow rate and total pressure through the cascade; hence, most likely changing the dynamic pressure. Since the dynamic pressure used to calculate $C_p$ is measured upstream of the cascade at slot s0, the coefficient of pressure calculated on the blade surface may be off slightly.

Figure 3.11 Comparison of Y/H = 0.50 $C_p$ at Different Incoming Velocities and 0° AOA

Figure 3.12 Comparison of Y/H = 0.50 $C_p$ at Different Angle of Attack and $U_\infty = 48.71 \pm 0.33$ m/s
Figure 3.11 depicted the coefficient of pressure at three different incoming velocities. These velocities were chosen because they were within 5% of the nominal incoming Reynolds number. The experimental results collapsed on top of each other as expected.

Moving to Figure 3.12, $C_p$ on the suction side decreased as the angle of attack was increased and $C_p$ on the pressure side essentially stayed constant except near the leading and trailing edges. At the leading edge, $C_p$ moved closer to 1 at positions downstream of the stagnation point. $C_p$ decreased slightly at the last two trailing edge positions. When the angle of attack was decreased to -2.5°, $C_p$ on the suction side increased and $C_p$ on the pressure side didn’t seem to be affected except at the leading edge. There, $C_p$ decreased near the stagnation point and the stagnation point appeared to shift closer to the leading edge.

An indication of boundary layer separation occurred in Figure 3.12. Here, the final two pressure taps on the suction side for the +2.5° case indicated a dramatically lower pressure than expected. Separation is entirely possible at this location because a positive pressure gradient exists there and was actually the highest one observed out of all three cases. To further this point, in Figure 3.4 the wake spectra for the +2.5° case had a bump in the energy spectrum from 200 Hz – 600 Hz. These frequencies may be the wake shedding frequencies of the unstable or separated boundary layer occurring in Figure 3.12 for the +2.5° case.
4 Velocity and Pressure Measurements Acquired in the Test Section with the Wake Generator Installed

The measurements and results presented in the ensuing chapter were acquired with the setup depicted in Figure 4.1 below. Incoming velocity profiles were acquired at slot s0, located 24 inches upstream of the central test blades leading edge. Cascade wake measurements were acquired at the downstream slot located 3 inches from the cascade trailing edge plane. The slot was located 3 inches downstream of the cascade blade leading edges in the Z direction not the direction depicted in Figure 4.1 below.

![Figure 4.1 Illustration of Test Section with Wake Generator Installed and CTA Slots Labeled](image)

All “running” results were acquired when the wake generator was running at a translation velocity of 1 m/s. This corresponds to a wake passing frequency of 6.5 Hz based on a blade spacing of 6 inches. Other results were acquired with the wake generator at a fixed position to compare with the running results. These results are labeled “aligned” and “misaligned” and the positions are depicted below in Figure 4.2.

![Figure 4.2 Illustration of “Aligned” (left) and “Misaligned” (right) Configurations](image)
The spacing between the leading edge of the cascade blades and the trailing edge of the wake generator blades was designed to be 40% of the axial chord or 2.4 inches. This distance was measured for all 17 wake generator blades and the results are shown below in Table 4.1. The standard deviation, denoted Std, about the mean was ~3% and the standard deviation about 2.4 inches was ~8%.

<table>
<thead>
<tr>
<th></th>
<th>Bottom</th>
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<td>0.02</td>
<td>0.03</td>
<td>0.03</td>
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<tr>
<td>$(\text{Std})_{2.4}/2.4$</td>
<td>0.07</td>
<td>0.09</td>
<td>0.08</td>
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### 4.1 Velocity Measurements

All results in this section were acquired using a TSI IFA 300 constant temperature anemometer (CTA) and a TSI 1211-20 hot film probe. The energy spectrums were acquired using a sampling rate of 100 kHz with a low pass filter of 50 kHz for 100 s. They were computed using a discrete fast Fourier transform (FFT) producing a frequency resolution of 0.02 Hz. Velocity profiles were acquired using a sampling rate based on the estimated local integral time scale. The integral time scales were computed from measurements acquired at several points along the vertical and horizontal profiles using a sampling rate of 10 kHz with a 5 kHz low pass filter for 100 s. The resulting integral time scales were multiplied by a factor of two, in some cases a factor of four was used, to ensure ample time between samples.

#### 4.1.1 Time Averaged Measurements

The energy spectrums computed at the walls and center of slot s0 are presented below when the wake generator was running and in the aligned position. Only the smoothed spectrums are presented to conserve space.

![Figure 4.3 Slot s0 Energy Spectrum at Different Positions](image_url)

a) Not Rotating (NR)  
b) Rotating (ROT)
In Figure 4.3a, the spectrums at the test section walls have the same very high frequency peaks observed in Figure 3.3 and a peak at 1 kHz. The center of the test section has those same peaks, although at lower amplitude, and peaks at 11 Hz, 30 Hz and 90 Hz. It also has lower overall energy in the center because the turbulence level is lower compared to the walls. The 11 Hz peak is mostly likely a shifted result of the 15 Hz peak in Figure 3.3, but the 90 and 1000 Hz peaks must be coming from the addition of the wake generator. If the same characteristic frequency used in Equation 2 is used to calculate a frequency based on the freestream velocity and the blade spacing, 2.4 inches, the resulting frequency is 833 Hz. Based on this result, it seems likely that the ~ 1000 Hz peak occurring in the spectra is coming from the blade spacing.

Looking at Figure 4.3b, the wall energy spectra are identical to the spectrums when the wake generator is not rotating except the peak around 1000 Hz is not visible in the boundary layer spectrums. However, the peak still exists in the center of the test section. The center spectrum also has a peak at 6.5 Hz which is the wake passing frequency. This shows that the wake generator is operating as predicted and producing a predictable wake passing frequency.

The energy spectrum in the wake of the central test blade and the wake generator blades is presented in Figure 4.4. The wake of the central test blade was acquired with the wake generator blades aligned, misaligned and running; the wake of the wake generator blade was acquired in the aligned and running positions. The central test blade measurement was acquired at the location of the minimum velocity in the wake. The wake generator measurement was at Y/H = 0.5 and about 1 inch upstream of the leading edge of the central test blade. The location is depicted as “A” in Figure 4.1.

The aligned case in Figure 4.4a has higher energy content compared to the misaligned case indicating a more turbulent wake. Peaks occur at about 30 Hz and 900 Hz in that case. Both of these peaks are most likely the same ones observed in Figure 4.3a for the center of the test section. The 900 Hz peak occurs in the misaligned case as well, although it appears to be slightly shifted to the right. The wake passing frequency was picked up along with the resulting harmonics in the running case. It is unclear
whether the 6.5 Hz peak shifts and amplifies the 30 Hz peak or if that peak is a harmonic of the 6.5 Hz peak.

In Figure 4.4b, the expected 6.5 Hz peak and its harmonics are present along with the same 900 Hz peak observed upstream of the wake generator for the running case. However, the energy content for that case is lower than in the aligned case. The peak at ~1000 Hz in the aligned case is also a full order of magnitude larger than in Figure 4.3 and Figure 4.4a. This makes sense because the blade spacing for that case is fixed at 2.4 inches. For the running case, the peak at about 900 Hz is actually a more distributed, wider peak and has lower peak energy because the blade spacing is fluctuating due to the motion of the wake generator blades.

The velocity profiles and turbulence intensities at slot s0 are presented in Figure 4.5 below. The velocity profile and turbulence intensity are presented with the wake generator blades in the aligned, misaligned, and running positions. The velocities were normalized with a computed weighted average velocity for each case.

![Figure 4.5 Velocity Profiles and Turbulence Intensities in Slot s0 for Different Cases](image)

Both velocity profiles in Figure 4.5 appear to be uniform across the test section. To check the fluctuation level in the test section, a single point measurement of the velocity was acquired at slot s0. The mean velocity when the wake generator was aligned was 50.029 ± 0.035 m/s and 49.972 ± 0.037 m/s when it was misaligned. When the wake generator was running, the velocity varied from 49.991 m/s to 49.840 m/s. However, phase averaged results acquired over 104 cycles at the same positions showed no cyclic pattern occurring with the wake generator running. The turbulence intensity in the freestream was about 0.20% with the wake generator running and about 0.15% with it off for the horizontal and vertical profiles. It increases, as expected, near the walls and in the boundary layer.

The wake of the cascade blades was acquired with the wake generator blades in the aligned and misaligned positions, and compared to the case without the wake generator installed. Again, the velocities were normalized using the computed weighted average of each velocity profile.
The velocity in the wake is a little bit higher with the wake generator installed than without it (not shown in Figure 4.6). The test section bleeds change the mass flow rate exiting the cascade and thus, seem to cause a decrease in the overall velocity downstream of the cascade. When the wake generator is installed, the bleeds are not used producing a higher mass flow rate; therefore, the velocity is also higher.

In Figure 4.6, the central test blade minimum velocity is about 5 m/s lower than the top and bottom test blades for the aligned case. In this same case, the velocity between the blades starts off at about 80 m/s on the pressure side and slowly decreases to about 78 m/s on the suction side before the wake of the blades begins. The aligned profile is also shifted to the left by $l/L = 0.02$ and has a lower minimum velocity for each blade than the misaligned profile. The expected shape between the cascade blades is present in the misaligned profile. It agrees very well with the “Velocity w/o Wake Generator” case, but does have higher turbulence intensity between the blades. The values there are from $\sim0.50\%$ - $\sim2\%$ compared to $0.25\%$ in the “Velocity w/o Wake Generator” case. The overall velocity profile, in the misaligned case, seems to decrease slightly with increasing position (from left to right) possibly due to flow restrictions caused by the wake generator blade locations. The aligned case turbulence intensity in Figure 4.6 is larger than the misaligned case with values of 5%, 9%, and 7% from bottom to top in the cascade.

4.1.2 Phase Averaged Measurements in the Wake(s)

Phase averaged central test blade wake profiles for a full cycle, are presented below. Phase locked results for the wake generator blade wake are also presented. The results were phase averaged over 208 cycles for the central test blade and 104 cycles for the wake generator blade. The wake generator blade wake velocity is plotted against the normalized time, $t/T$, with error bars, and the central test blade wake is plotted against the
normalized position. The period of one cycle was calculated to be $T = 0.15385$ s based on the 6.5 Hz wake passing frequency. The same TSI equipment was used to acquire the data. A sampling rate of 1300 Hz (200 times the wake passing frequency) was used to provide a high resolution capture of each cycle. Figure 4.7 below shows the phase averaged time record of the wake generator blades starting at an initial time $t_0$. The evolution of the central test blade wake through time is presented in Figure 4.8. The different time slices shown in Figure 4.8 are marked as vertical black lines in Figure 4.7. This helps provide a position reference of the wake generator blade wake when examining Figure 4.8.

![Figure 4.7 Phase Averaged Time Record of the Wake Generator Blade](image1)

![Figure 4.8 Phase Averaged Velocity Profile of the Central Test Blade Wake at Different Times](image2)

The wake profile in Figure 4.7 was measured at the leading edge of the central test blade, denoted position “A” in Figure 4.1, between the wake generator blade trailing edge and the central test blade leading edge. The passing wake is clearly captured in Figure 4.7 and shows an initial incoming velocity of 50 m/s at $t_0$ then a decrease starting at $t/T = 0.50$. The minimum velocity, $U = 38$ m/s, occurred at $t/T = 0.7175$. After this time, the velocity increases back to 50 m/s at $t/T = 1.0$. The effect of the passing wake on the central test blade wake is apparent in Figure 4.8. The wake profile initially looks very similar to the one shown in Figure 4.6 for the misaligned case. Then, starting at $t/T = 0.50$, the wake begins shifting to the left and growing wider, moving closer to the profile observed in Figure 4.6 for the aligned case. The wake was about $l/L = 0.25$ wide at $t/T = 0$, but grows to $l/L = 0.41$ at $t/T = 0.75$ where the wake looks very similar to the one shown in Figure 4.6 for the aligned case. In that figure, the minimum velocity is about 60 m/s and the wake is very wide. After time $t/T = 0.75$, the wake begins to move back to the right and eventually goes back to the same shape as at $t_0$.

### 4.2 Pressure Measurements

The pressure measurements in this section were acquired with two Scanivalve DSA3217s. Long samples with a sampling rate of 100 Hz were acquired for 100 s to
estimate the autocorrelation coefficient and the integral time scale. The results presented were acquired using a sampling frequency based off of an estimate of the largest integral time scale. Because the nature of the measurement system requires the acquisition of multiple points at once rather than one point at a time like the CTA system, the estimate was based off of the minimum sampling rate for the entire system which consisted of 123 pressure taps. The resulting sampling rate was 1.5 Hz for the misaligned case and 5 Hz for the aligned case.

4.2.1 Time Independent Measurements

Pressure measurements were acquired with the wake generator blades in the aligned and misaligned positions to compare with the phase averaged profiles acquired with the wake generator running.

![Coefficient of Pressure Distribution on the Pressure Tap Blade with the Wake Generator Blades in Different Positions](image)

Figure 4.9 Coefficient of Pressure Distribution on the Pressure Tap Blade with the Wake Generator Blades in Different Positions

![Comparison of the Coefficient of Pressure Distribution with the AFOSR and CFD Results](image)

Figure 4.10 Comparison of the Coefficient of Pressure Distribution with the AFOSR and CFD Results
In Figure 4.9, the coefficient of pressure distribution on the pressure tap blade is presented for three locations along the blade height: Y/H = 0.25, 0.50, and 0.75. The results are shown for the aligned and misaligned cases. In both cases, there is some slight disagreement between the spanwise sections on the pressure side and on the suction side near the trailing edge. However, the overall agreement is very good and the results are well within 10% of one another.

Figure 4.10 compares the experimental results with the previously presented computational and experimental results. Only the experimental results for the section at Y/H = 0.50 are presented to unclutter the graph. \( C_p \) on the pressure side appears to be the same for all cases, even at the leading edge. On the pressure side, the experimental results agree very well with the CFD results and slightly less with the AFOSR results. Again, the results at the trailing edge do not agree with the AFOSR data. \( C_p \) on the suction side when the wake generator is not installed is the lowest compared to the aligned and misaligned cases, most likely due to the test section bleeds. As expected, the aligned case resulted in the highest coefficient of pressure on the suction side compared to the other two cases; therefore, it the blade produced the least amount of lift in that case. The experimental results for the misaligned case follow the CFD curve almost exactly and are only slightly shifted compared to the AFOSR results.

4.2.2 Phase Averaged Results

The pressure distribution on the blade profile was captured when the wake generator was running with a wake passing frequency of 6.5 Hz. A sampling frequency of 65 Hz was used to capture the pressure distribution at 10 different times. A total of 104 cycles were captured and averaged for all of the pressure taps. The times shown below in Figure 4.11 were normalized with respect to one wake passing period, 0.1538 s. The black X’s on the figure represent the time average coefficient of pressure profile and are used for the purpose of comparison.

![Figure 4.11 Phase Averaged Coefficient of Pressure Profile (Y/H = 0.5) at Different Times during a Cycle (Phase Averaged for 104 Cycles)](image)
In Figure 4.11, the stagnation point at the leading edge of the blade appeared to fluctuate as the wake passed. It varied from 0.9, initially, to 1.0 then down to 0.8 at t/T = 0.90. This indicates the freestream flow loses energy due to losses occurring in the wake of the wake generator blade. Most of the blade experiences the same initial increase in pressure until about t/T = 0.60. This increase in pressure is a result of the flow accelerating and recovering after a wake passes. After this time, the pressure decreases as the wake passes. The pressure decreases on the suction side first because the wake impacts it first. The suction side appears to be more sensitive to the pressure decrease because the propagation of the pressure change is captured along it. About t/T = 0.10 later, the wake impacts the pressure side and a decrease in pressure is observed there as well. Overall, the fluctuations appear to be very small and do not seem to affect the lift generation of the airfoil by very much. The effect on the wake profile is much more profound than the apparent effect on the pressure profile. However, this may be due to the sampling rate. The sampling rate was only 65 Hz and may not have been high enough to capture the higher order modes or higher order turbulent fluctuations that occur in the flow. It is apparent from Figure 4.3 that these higher frequency modes are important and
due contribute to the flow in the test section. The sampling rate used for the wake measurements was 1300 Hz which is higher than the majority of the peaks observed in Figure 4.3.
5 Conclusions

A closed loop wind tunnel with a four passage linear cascade and a removable wake generator was completed in May 2011. The wind tunnel was designed to study film cooling on low pressure turbine blades, and the interaction between these film cooled rotors and the upstream wake generator blades.

The purpose of this report was to characterize and verify the performance of the wind tunnel through velocity and pressure measurements in the test section. Measurements were made without the wake generator installed to characterize the test section by itself and to serve as a baseline for comparisons between results acquired with the wake generator installed. After the test section was characterized, the wake generator was installed to characterize it and quantify its effects on the cascade blades.

Several new designs were created to accommodate measurements in the test section including a three axis traverse. The traverse has a travel range of 40 inches (X) x 20 inches (Y) x 50 inches (Z), and it was specifically designed to make automated hot wire/film measurements in the test section. A pressure tap blade featuring 41 pressure taps located at three spatial locations along the blade height was also realized. The blade was used in accordance with Scanivalve software and hardware to make the above mentioned pressure measurements in the cascade.

Old designs, such as the wake generator blades, the test section near walls, and the acrylic test section inlet used in conjunction with the wake generator were updated to meet the requirements of the facility. The wake generator blades were slightly modified to include more fasteners and an all-Aluminum body. The Aluminum body helped solve a warping problem the old blades were having.

Spectral results revealed a 15 Hz peak occurring at all four walls and in the center of the test section at slot s0. The energy spectrum of the central test blade wake had the same 15 Hz peak, a distributed set of peaks from 200 – 300 Hz, and a peak at 470 Hz. It is believed these peaks are coming from the wind tunnel duct walls and the test section walls. Velocity profiles acquired upstream of the cascade without the wake generator installed indicated a uniform velocity distribution in the vertical and horizontal directions. Turbulence intensities outside of the boundary layers were around 0.20%. A velocity profile acquired downstream of the cascade verified periodicity in the wake of the cascade blades.

Pressure profiles were acquired at Y/H = 0.25, 0.50 and 0.75, and presented as a coefficient of pressure. They were compared with an AFOSR simulation and an in house CFD simulation. The AFOSR simulation was run with an axial chord of 8.5 inches, and the in house simulation with the same geometry used in the cascade (C_x = 6 inches). The experimental results had the same trend as the computational data, but the magnitudes were slightly off. This was attributed to the use of the test section bleeds to “balance” the flow into the cascade.

Spectral measurements at slot s0 with the wake generator installed in the test section showed a 900 Hz peak. The results were acquired at the walls and the center of the test section. A peak also showed up at about 11 Hz in the center of the test section. The 11 Hz peak is most likely coming from the wind tunnel duct walls much like the previously mentioned 15 Hz peak. The blade spacing between the leading edge of the cascade blades and the trailing edge of the wake generator blades is the cause of the 900 Hz peak.
Spectral measurements in the wake of the central test blade and the wake generator blade showed the same 11 Hz and ~900 Hz peaks. The wake passing frequency, calculated to be 6.5 Hz, also showed up when the wake generator was running. The wake generator blade wake energy spectrum confirmed the 900 Hz theory. The peak in energy that occurred at ~900 Hz was an order of magnitude larger than it was in any other spectrum. When the wake generator was running, the energy at the peak decreased and increased in the frequencies around it producing a more distributed set of peaks around 900 Hz.

With the wake generator installed, velocity profiles upstream of the wake generator were acquired. All cases showed a uniform velocity across the vertical and horizontal directions of the test section and turbulence intensity levels around 0.20%. The wake profile downstream of the cascade confirmed periodicity still existed with the wake generator installed. Wake results showed a lower minimum velocity and a thicker overall wake when the wake generator trailing edge was aligned with the leading edge of the cascade blades. The wake also shifted down (or towards the suction side) when the blades were in this position. When the wake generator blade trailing edges were in between the cascade blades, the wake profile was very similar to the wake profile without the wake generator installed, but the turbulence intensity increased compared to the case without the wake generator. Phase averaged results, taken in the wake of the central test blade, indicated a thicker wake occurred after the passing wake impacted the leading edge of the blade. Wake impact also caused a shift in position and a decrease of the central test blade wake minimum velocity.

Pressure measurements with the wake generator installed indicated lower overall lift production with the presence of the wake generator compared to the case without it. Phase averaged results showed a small fluctuation in the coefficient of pressure arose with the wake generator running. The overall trend was a decrease in pressure as the wake passed.

Pressure measurements with the wake generator installed indicated lower overall lift production with the presence of the wake generator compared to the case without it. Phase averaged results showed a small fluctuation in the coefficient of pressure arose with the wake generator running. The overall trend was a decrease in pressure as the wake passed.

The above results indicate the wind tunnel and its conditioning screens are performing as designed. The test section has a uniform incoming velocity with low turbulence intensities; the pressure profile matches to within 7% of the AFOSR profile and 1.5% to the CFD profile; and the cascade appears to have periodicity since the variance in the wake profile of each blade is only about ±3%. The temperature profile of the incoming flow is also uniform.

The wake generator is functioning as expected with a predictable wake passing frequency, and it does not cause significant fluctuations in the incoming flow. Cascade wake velocity profiles indicate periodic flow conditions still exist even with the wake generator installed. However, the wake generator does not seem to be creating any significant changes in the coefficient of pressure distribution along the blade.
References


2. LaPlace, Jean Phillipe Junca, *Design, Fabrication, and Characterization of a New Wind Tunnel Facility- Linear Cascade with a Wake Simulator*, in Mechanical Engineering Dept. 2007, Louisiana State University. p. 113.


Appendix I CAD Drawings
Wind Tunnel Test Section End Wall Parts
Film Cooling Blade Parts

For threaded insert, 1/4"-20.

\( \varnothing \) 0.3438 \( \times \) 5.625

1.75
6.00
12.00

\( \varnothing \) 0.475 \( \times \) 1.00

2X \( \varnothing \) 0.3438 \( \times \) 5.625

30X \( \varnothing \) 0.125

Pressure Side

For threaded inserts, 1/4"-20.
PIV Optical Mirror Traversing Mechanism Parts

SECTION A-A
SCALE 2:1

Drive Nut
Inner Housing
Outer Housing
Cap Spacer Mount
Drive Shaft
Mirror Carriage
Ball Bearings
PIV Side Blade Parts
Pressure Tap Blade Parts
Modified Test Section Parts (from Jean-Phillipe’s designs)
Traverse Parts
**For Hot-Wire Probe Holder**
Wake Generator Blade Parts
Appendix II Data Sheets

85 mm (3.35 in.)
Step Angle 1.8°
PK Series Standard Type

Specifications

<table>
<thead>
<tr>
<th>Model</th>
<th>Connection Type</th>
<th>Holding Torque Nm</th>
<th>Holding Current Phase A</th>
<th>Voltage VDC</th>
<th>Resistance Ωphase</th>
<th>Inductance mH/phase</th>
<th>Motor Inertia kgm²</th>
<th>Start Motor Load Nm</th>
<th>Start Motor Current A</th>
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<td>2.8</td>
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<td>1.04</td>
<td>36</td>
<td>1760 x 10⁻⁵</td>
<td>16.8</td>
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Dimensions

Unit = mm (in.)

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<thead>
<tr>
<th>Model</th>
<th>L1</th>
<th>L2</th>
<th>Mass (kg)</th>
<th>DIF</th>
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<tr>
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<td>1.7 (0.07)</td>
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</tr>
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<td>PK296-0-1BA</td>
<td>160 (6.3)</td>
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<td>PK296-0-3AA</td>
<td>94 (3.7)</td>
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<td>PK296-0-3BA</td>
<td>130 (5.1)</td>
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<tr>
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<td>PK296-0-4BA</td>
<td>160 (6.3)</td>
<td>81232</td>
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</tbody>
</table>

Wiring and Connections C-211 / Specifications, Characteristics, Dimensions C-218 / General Specifications C-268

C-253

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Note:
- Pay attention to heat dissipation from motor as there will be a considerable amount of heat under certain conditions.
- Be sure to keep the temperature of the motor case under 105°C (221°F).
Note: Motor leads are 12.6 in. (314.6 mm) minimum.

<table>
<thead>
<tr>
<th>H Part Number</th>
<th>Manufacturer Part Number</th>
<th>Dual Shaft</th>
<th>Max Length A in. (mm)</th>
<th>B max and min in. (mm)</th>
<th>C max and min in. (mm)</th>
<th>D max and min in. (mm)</th>
<th>Net Weight (lb)</th>
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<td>78008-01</td>
<td>N33HRP.L:KSN.M00-00</td>
<td>no</td>
<td>3.10</td>
<td>0.900 (7.50) to 1.600 (12.697)</td>
<td>0.120 (3.05) to 0.120 (3.05)</td>
<td>0.055 (14.96) to 0.055 (14.96)</td>
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<td>78008-01</td>
<td>N33HRP.L:KSN.M00-00</td>
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<td>3.10</td>
<td>0.620 (5.95) to 1.624 (12.849)</td>
<td>0.120 (3.05) to 0.120 (3.05)</td>
<td>0.055 (14.96) to 0.055 (14.96)</td>
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<td>0.620 (5.95) to 1.624 (12.849)</td>
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<td>11.9</td>
</tr>
</tbody>
</table>

Encoders for NEMA 23 and NEMA 34 Motors

**Electrical**

- Resolution: 1500 counts/revolution
- Input voltage: 5 V ±10%
- Input current: 150 mA max (50 mA typical) with no output load
- Channel configuration: Quadrature A, B, and Index
- Output type: Differential sine driver
- Noise immunity: Tested to BS EN60068-2-2, BS EN60068-2-3, BS EN60068-4-2, BS EN61000-4-3, BS EN61000-4-6, BS EN60068-11
- Symmetry: 180 deg (±10 deg) electrical
- Quadrature phase: 50 deg (±25 deg) electrical
- Minimum wire separation: 0.05 in. (0.127 mm)
- Accuracy: Within ±0.01 deg mechanical at 1 sec/minute from true position

**Industry Standards**

- Industrial standards: CE
- Sealing standards: IP40
- RoHS Compliance: Yes

**Physical**

- Operating temperature: -20 to 80°C
- Motor type: Thrust
- Bore size: 1.4 in. (52.2 mm) or 8 mm (7826-01)
- Mounting: 1.812 in. (140.6 mm) two-hole flange mount
- Maximum frequency: 200 kHz
- Operating temperature: -20 to 80°C
- Max shaft speed: 8000 rpm
Dimensions, Wiring and Timing Diagrams

<table>
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<tr>
<th>Pin #</th>
<th>Wire Color</th>
<th>Function</th>
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<tr>
<td>2</td>
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<td>WDC</td>
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<tr>
<td>3</td>
<td>Yellow</td>
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<td>5</td>
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<td>6</td>
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<tr>
<td>8</td>
<td>Blue</td>
<td>ZZ</td>
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Note: All dimensions have a tolerance of ±0.005 in. or ±0.1 mm, unless otherwise specified.
Modular Incremental Rotary Optical Encoder

MX21 Series

By the time you have read this first sentence, you could have installed BEI's model MX21 INSTA-MOUNT™ modular optical encoder. In addition to its quick and easy installation, the MX21 is designed to operate with jitter-free output signals without tight control on shaft endplay, runout, or perpendicularity. The new INSTA-MOUNT™ encoder is capable of operating within a temperature range of -10º to +40º C, requiring less than 30 millamps of L.E.D. current, without degradation of output signals and is short circuit protected. The MX21 is perfectly suited for motor manufacturers and other high volume OEMs.

BEI's INSTA-MOUNT™ Series encoder offers 5V TTL compatible quadrature outputs with index and complements as options. Axial shaft movements during operation, of ±0.010", will not adversely affect the output signals. Shaft runouts of 0.002" TIR can also be absorbed by this device without affecting output signal performance.

Figure 1

Standard Features
- Resolution up to 1024 PPR
- Quick and easy installation
- Tolerant of axial and radial shaft movements often associated with less expensive motor designs
- Jitter-free outputs
- Increased MTBF
- Index and complementary output options
- 24LS31 line driver output from MXC16
- High frequency response
- 2-year warranty
Performance Specifications

Mechanical
- Dimensions: see Figure 1
- Weight: 2.7 oz. (75g max.)
- Moment of inertia: 2.6 x 10^-6 lb-in sec^2
- Box Size: see "Ordering Information"

Motor Interface
- Mount Holes: 3 & 4 [.40 or [.50 x 10^-6] on 1.012" dia B.C.
- Mount Hardware: 7 socket head cap screws
- Perpendicularity: 0.001" max. (with 0.0001” positive accuracy by 0.5 arc minutes)
- Shaft Length: 0.050" max (with 0.0001” minimum accuracy by 0.5 arc minutes)
- Shaft Enlargement: 0.001”
- Shaft Finish: 16 microinches or better
- End must be chamfered or rounded
- Shell Finish: minimum 0.0001”-0.0002”

Electrical
- Code: Incremental
- Pulse per Revolution: see "Ordering Information"
- Index Pulse Options: unselected index (7)
- Diameters: 5 volt vs 5K ohms max.
- Output Format: dual channel quadrature and index
  (M2013 & MC1233)
  (No Index as M2012)
  (MC1233)
- Output Format (M2013):
  dual channel quadrature and index
  with complements
  square wave (TTL, Open Collector)
  (M2013)
  (MC1233 & MC1253)
  (M2013)
- Output Type: TTL differential line driver (20LS3 or equiv.)
  (M2013)
  (MC1233 or equiv.)
  should be terminated into a line maker
  (20LS3 or equivalent circuit)
- Frequency Response: see graph Fig. 5
- Rise Time: 1.5 usec max.
- Environmental Temperature:
  operating: -40°C to +70°C
  storage: -40°C to +125°C

Termination
- Type: 28 AWG flat ribbon cable with 10-position
  connector Berg Pin 55557-165 or equiv.
  Wires with Berg Pin 55557-165 or equiv.
  (wiring connector not provided)

Ordering Information

Model: MX21 X - XX - XXXX - X

Basic Model No.
- Output Format:
  2 = Quadrature
  3 = Quadrature w/index
  6 = Quadrature w/index & complements

Base Rate:
- 25 = 25, 26 = 26, 30 = 30
- 60 = 60, 61 = 61, 108 = 108

Pulse Per Revolution (PPR):
- 200, 400, 500, 1000, 1024

Index Cycles:
- C = used as data A & B
- U = unused

EXAMPLE: M2013-36-1006-U

Specifications subject to change without notice. Printed in U.S.A. 11/03/M201/MX21/100/1

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Tooele, UT 84074-9795
(801) 287-8984 Fax: (801) 287-8984
email: sales@duncan.com
www.duncan.com
Accura® 55 Plastic
For use with solid-state stereolithography (SLA®) Systems

“Accura® 55 has proven to be an excellent resin for Harvest. Its low viscosity allows us to clean and finish parts more easily. Coupled with a high build success rate and the result is greater production efficiency and higher quality parts. Additionally, the mechanical properties allow the models to serve as functional prototypes that better meet the needs of our customers.”

Jason Morgan,
StereoLithography
Production Manager
Harvest Technologies

Look and feel of molded ABS.

Durable and rigid testing.

Technical Data

Liquid Material

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Condition</th>
<th>Value</th>
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<td>Appearance</td>
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<tr>
<td>Liquid Density</td>
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</tr>
<tr>
<td>Solid Density</td>
<td>@ 25 °C (77 °F)</td>
<td>1.20 g/cm³</td>
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<td>@ 30 °C (86 °F)</td>
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<tr>
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Post-Cured Material

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<td>9,200 - 9,800 PSI</td>
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<td>ASTM D638</td>
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<td>460 - 490 KSI</td>
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<td>Elongation at Break (%)</td>
<td>ASTM D638</td>
<td>5 - 8 %</td>
<td>5 - 8 %</td>
</tr>
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<td>Flexural Strength</td>
<td>ASTM D790</td>
<td>88 - 110 MPa</td>
<td>13,810 - 16,920 PSI</td>
</tr>
<tr>
<td>Flexural Modulus</td>
<td>ASTM D790</td>
<td>1,050 - 3,460 MPa</td>
<td>890 - 2,600 KSI</td>
</tr>
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<td>Impact Strength (Notched izod)</td>
<td>ASTM D256</td>
<td>12 - 12.5 ft-lb/in</td>
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<td>Impact Strength (Notched Izod)</td>
<td>ASTM D5420</td>
<td>1.1 J</td>
<td>0.81 ft-lb</td>
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<td>Heat Deflection Temperature</td>
<td>ASTM D648</td>
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<td>Hardness Shore D</td>
<td>ASTM D648</td>
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<td>85</td>
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<td>Coefficient of Thermal Expansion</td>
<td>TMA (100°C)</td>
<td>61 ± μm/°C</td>
<td>141 μm/in - °C</td>
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<tr>
<td></td>
<td>TMA (100°C)</td>
<td>163 μm/°C</td>
<td>3.3 μm/in - °C</td>
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<tr>
<td>Glass Transition (Tg)</td>
<td>DMA, E¹</td>
<td>54 °C</td>
<td>127°C</td>
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</tbody>
</table>

¹Dp/Ep values are the same on all solid-state laser SLA® Systems.

3D Systems Corporation
333 Three D Systems Circle
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www.3dsystems.com
Nasdaq: TRSC

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Accura® 60 plastic
For use with solid-state stereolithography (SLA®) systems

"From a production standpoint Accura 60 is very efficient. It has very low viscosity, which makes it easy to build up and easy to clean up, which ultimately results in good parts and rapid turnaround. The parts we have made so far have been impressive, with excellent accuracy, detail, rigidity, functionality and finish, and it has excellent optical clarity when clear coated."

-- Ron Clemens - Director of Business Development - Harvest Technologies

TECHNICAL DATA

Liquid Material

<table>
<thead>
<tr>
<th>MEASUREMENT</th>
<th>CONDITION</th>
<th>VALUE:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Appearance</td>
<td></td>
<td>Clear</td>
</tr>
<tr>
<td>Liquid Density</td>
<td>@ 25 °C (77 °F)</td>
<td>1.11 g/cm³</td>
</tr>
<tr>
<td>Solid Density</td>
<td>@ 25 °C (77 °F)</td>
<td>1.21 g/cm³</td>
</tr>
<tr>
<td>Viscosity</td>
<td>@ 50 °C (122 °F)</td>
<td>150 - 180 gpa</td>
</tr>
<tr>
<td>Permeation/depth (top)</td>
<td></td>
<td>0.4 mm</td>
</tr>
<tr>
<td>Critical Exposure (E*)</td>
<td></td>
<td>7.6 mJ/cm²</td>
</tr>
<tr>
<td>Tested Build Styles</td>
<td></td>
<td>EXACT®, FAST®, QuickCast®</td>
</tr>
</tbody>
</table>

Post-Cured Material

<table>
<thead>
<tr>
<th>MEASUREMENT</th>
<th>CONDITION</th>
<th>VALUE:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile Strength</td>
<td>ASTM D 638</td>
<td>58 - 68 MPA (840 - 9600 PSI)</td>
</tr>
<tr>
<td>Tensile Modulus</td>
<td>ASTM D 638</td>
<td>2000 - 3100 MPA (290 - 450 KSI)</td>
</tr>
<tr>
<td>Elongation at Break (%)</td>
<td>ASTM D 638</td>
<td>5 - 13 %</td>
</tr>
<tr>
<td>Flexural Strength</td>
<td>ASTM D 790</td>
<td>87 - 101 MPA (1260 - 1460 PSI)</td>
</tr>
<tr>
<td>Flexural Modulus</td>
<td>ASTM D 790</td>
<td>2700 - 3000 MPA (390 - 435 KSI)</td>
</tr>
<tr>
<td>Impact Strength (Notched izod)</td>
<td>ASTM D 256</td>
<td>15 - 25 J/m (0.3 - 3.5 ft-lb/ft)</td>
</tr>
<tr>
<td>Heat Deflection Temperature</td>
<td>ASTM D 648</td>
<td>53 - 55 °C (127 - 131 °F)</td>
</tr>
<tr>
<td>Hardness, Shore D</td>
<td></td>
<td>80</td>
</tr>
<tr>
<td>Co-efficient of Thermal Expansion</td>
<td>ASTM E 831-92</td>
<td>71 μm/m°C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>123 μm/m°C</td>
</tr>
<tr>
<td>Glass Transition (Tg)</td>
<td>DMA, E'</td>
<td>58 °C (136 °F)</td>
</tr>
</tbody>
</table>

* OpEic values are the same on all systems.
Vita

Christopher Michael Foreman was born in Fort Walton Beach, Florida to Michael and Christine Foreman. Shortly after his birth, his father retired from the United States Air Force and the family moved back to his father’s home town of Crowley, Louisiana. After a few more moves, Christopher and his family finally settled in the small town of Rayne, LA.

Christopher attended high school at Notre Dame High School of Acadia Parish and graduated in May 2007 with honors. In August 2007, he continued his academic career at Louisiana State University in Baton Rouge, Louisiana pursuing a Bachelor of Science degree in Mechanical Engineering. He decided he wanted to pursue a career in the Aerospace Industry because of his interest in jet fighters. His junior year he added a minor in Aerospace Engineering and joined the Accelerated Master’s Program. In June 2010, he began researching film cooling in gas turbines under the direction of Dr. Dimitris Nikitopoulos. Shortly after, he graduated Magna Cum Laude with his Bachelor’s degree in May 2011. He is presently pursuing his Master’s degree in Mechanical Engineering and plans to graduate in May 2013.

Christopher enjoys exercising, flying RC helicopters with his father, duck hunting, and wakeboarding every chance he gets. He wants to get his recreational pilot license and maybe even fly a jet someday. After graduation Christopher is starting employment with the Boeing Company in Houston, TX.