DISTRIBUTED FORCING ON A 3D BLUFF BODY WITH A BLUNT BASE
AN EXPERIMENTAL ACTIVE DRAG CONTROL APPROACH

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ABSTRACT OF THESIS

Distributed Forcing on a 3D Bluff Body with a Blunt Base
An Experimental Active Drag Control Approach

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This paper seeks to explore the effects of an active drag control method known as distributed forcing on a 3D bluff body with a blunt base. The 9.5 x 15.25 x 3 inch aluminum model constructed for this experiment has an elliptically shaped nose and rectangular aft section. The model is fitted with four, 12 Volt fans, forcing the freestream air into and out of 1 mm thick slots on the upper and lower trailing edges. The forcing is steady in time, held at a constant forcing velocity though all Reynolds numbers, but varies roughly sinusoidally in the spanwise direction across the model. Testing was conducted at Reynolds numbers of 50,000, 100,000 and 150,000 at California Polytechnic State University, San Luis Obispo in the Aerospace Engineering Department's subsonic 3' by 4' wind tunnel.

Effectiveness of the distributed forcing method was evaluated by measuring the base pressure on the model using a Scanivalve system. By measuring multiple static pressure ports, it was found that base pressure increased by 15.3% and 4.2% at Reynolds numbers of 50,000 and 100,000 respectively, and showed a decrease of 2.7% at a Reynolds number of 150,000.

Total drag on the model was also measured using a sting balance mount fitted with strain gauges. This test showed a drag reduction of 15.8% and 5.5% for Reynolds numbers of 50,000 and 100,000 respectively, and an increase in drag of 2.0% at Reynolds number of 150,000, when omitting external power required to run the forcing assembly. The forcing assembly was shown to require nearly 12 times the power to operate than it saves in drag reduction at Reynolds number of 50,000. In addition, a thermal anemometry measurement of streamwise velocity of the near wake behind the bluff body was conducted to qualitatively assess the attenuation of the vortex street behind the model. Distributed forcing shows that as the freestream velocity is increased as compared to the forcing velocity, the change in energy spectral density is lessened, and as such, the largest attenuation in vortex shedding is at Reynolds number of 50,000 while nearly no change is seen at the Reynolds number of 150,000.
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**NOMENCLATURE**

| 2D, 3D | Two-dimensional, Three-dimensional |
| A | Frontal planform area |
| B, S, Ø | Blowing, Suction, Off, (of slot fan direction) |
| BSNTS | Blowing Slot Next to Strut |
| b | Forcing slot width |
| \(C_D\) | Coefficient of Drag |
| \(C_P\) | Coefficient of Pressure |
| \(C_\mu\) | Momentum Coefficient |
| CAD | Computer Aided Design |
| CFD | Computational Fluids Dynamics |
| CNC | Computer Numerically Controlled |
| CTA | Constant-temperature anemometer |
| D | Drag Force |
| \(F_{\text{Drag}}\) | Drag force |
| FFT | Fast Fourier Transform |
| f | Frequency |
| GUI | Graphical User Interface |
| h | Height (thickness) of model |
| ID, OD | Inner diameter, Outer diameter |
| J | Steady jet momentum |
| Kpt | Kilapoints, \(10^3\) points |
| LES | large eddy simulation |
| \(P_T\) | Total Pressure |
| \(P_\infty\) | Static Pressure |
| PVC | Polyvinyl chloride |
| psid | Pounds per square inch, differential |
| \(q_\infty\) | Freestream Dynamic Pressure |
| \(\text{Re}_h\) | Reynolds number based on model height |
| \(\text{Re}_L\) | Reynolds number based on model length |
| SSNTS | Suction Slot Next to Strut |
| St | Strouhal number |
| t | Slot thickness |
$U_j$  jet velocity
$u_\infty$  Freestream velocity
$V, V_{DC}$  Volts, Volts Direct Current
VFD  Variable Frequency Drive
$X, Y, Z$  Coordinates in streamwise, normal, spanwise directions, respectively

**Greek**

$\beta$  Forcing angle
$\lambda_z$  Forcing wavelength
$\rho$  Density
$\mu_\infty$  Freestream dynamic viscosity
$\Phi$  Forcing velocity
1. INTRODUCTION

This document describes the experimental wind tunnel tests exploring the effects of an active drag control method. The tests will be performed in the subsonic, incompressible flow regime on a three-dimensional bluff body with a blunt base. These parameters follow many previous studies in drag reduction and vortex attenuation, which are to be described later in this introduction.

The bluff body, dominated by pressure drag, presents itself in many real world applications. These include, but are not limited to, tractor-trailers, helicopter fuselages, large transport aircraft, and many high-rise buildings. These bluff bodies have a large base region, perpendicular to the freestream flow which creates a large area of separation and low base pressure. This leads to a substantial increase in the drag associated with the body. In addition, above a Reynolds number of approximately 100, bluff bodies will generate a von Kármán vortex street, named after the engineer and fluid dynamicist, Theodore von Kármán for his work in studying the phenomenon. This repeating, unsteady oscillation of eddies is caused by the slow moving boundary shear layer rolling up on the trailing edge of the blunt base. This adds complexity to the flow and increases the drag on the body as well. Therefore, it is advantageous to attenuate or delay the vortical structure as much as possible in reducing drag.

The blunt base describes the aft end of the object where the flat-plate, boundary-layer flow suddenly becomes a wake flow. This occurs when the separation point is at the trailing edge of the body. An example of a bluff body with a blunt base is shown in Fig. 1-1 [6]. This figure is of a tractor-trailer aft section fitted with passive drag control fins, and also shows the CFD (Fluent) generated velocity vectors. One can notice the separation point at the trailing edge, and the bluff body design indicates a low pressure area near the base of the trailer. In this particular study, twelve configurations of different rear-end tapering panels and side fairings facilitated the reduction of semitrailer-under-body drag. The configurations were studied experimentally at a
1:15 scale and one rear-end tapering panel was emphasized with a Fluent simulation. In the best case, tangent drag coefficient savings maxed at 12%.

Figure. 1-1. CFD generated velocity vectors of a tractor-trailer bluff body with a blunt base fitted with aft control fins; Shown looking down from top of trailer at a Reynolds number of 1.1e6.

The need for drag reduction is an ever-present requirement in the aerospace industry as well as other transportation fields. A seemingly small improvement in drag could have a dramatic cost savings and a positive environmental impact in the fuel saved. For this reason, many studies have been performed analyzing different techniques to control the flow around bluff bodies in the attempt to increase base pressure behind the body. These endeavors to decrease drag can be separated into two main categories, passive and active drag control. Passive drag control is adding or removing material to or from the body to change the geometry, and thus the flow around it. There are many passive flow control devices which, for instance, are intended to either delay or advance transition, suppress or enhance turbulence, or prevent or promote
separation depending on the application. In the references that follow, the passive flow control devices are intended to reduce drag and to eliminate or attenuate the vortex shedding which occurs. One example includes splitter plates, which are a continuous, stiff plate attached to the model in the spanwise direction. In the experiment performed by P.W. Bearman in 1964 [2], a two-dimensional model was fitted with various lengths of splitter plates. The splitter plates were shown to be effective in delaying the formation location of the vortex street occurring at Reynolds numbers of 1.4e5 and 2.45e5 and also shown to increase the base pressure for a specific range of splitter plate length to base height ratios.

Another example of a passive flow control device intended to reduce drag on a bluff body is the Master’s thesis work which preceded this paper’s experiments. In the wind tunnel tests performed by J. Pinn in 2011 [5], a three-dimensional bluff body was fitted with end-plate tabs protruding off the model at the separation point, normal to the freestream. A total of six tabs were placed on the top and bottom edges, spaced approximately 1.7 base heights apart along the span. The tabs were shown to eliminate all vortex shedding off the body, in both the normal and spanwise directions, and to increase the base pressure. Pinn’s experiments were based heavily on the work done by Park et al. [14] which study the effect of end-plate tabs in both a two-dimensional large eddy simulation (LES) and a two-dimensional wind tunnel experiment. Both of these passive drag control studies were referenced in developing this experiment and will be talked about later in this paper.

Active flow control has much the same application-specific objectives as passive flow control; however there is an external expenditure of energy into the flow around the body in the attempt to influence it favorably. A handful of active flow control focuses may include mechanical or acoustical vibrations [16], thermal heating or cooling [18], or any type of air forcing, through positive or negative pressures [17]. All these topics try to manipulate the pressure differentials around the body to influence the freestream flow and boundary layer in a beneficial way. The main advantage for active flow control is the ability to manipulate the
amount of energy being put into the system, for whatever method is chosen. For example, an active flow control technique can be turned on or off, affecting the flow instantaneously. A feedback loop can even be placed in the system in order to try and achieve a more optimal performance for changing flow conditions. The main disadvantage in using active flow control is the energy being spent manipulating the flow around the body. In the case of drag reduction, the energy savings must exceed the energy put into the active flow control, otherwise a net loss of energy will be realized in overcoming the freestream fluid.

As stated previously, there are many different methods to actively control flow just as there are for passive flow control. One example of an active flow control is base bleed. This term was coined to describe the addition of a secondary flow in the freestream direction on the base of the model. An experiment by P.W. Bearman in 1967 [3] details a two-dimensional cut-off ellipse for the bluff body rendering the back section rectangular. The “bleed”, which is steady in time, is powered by a small centrifugal fan which pushed air though a slot travelling along the entire span of the model. Bearman was able to show that as the bleed quantity is increased, the formation of the vortex street occurs further downstream. As a result, a greater proportion of vortex energy is diffused before the street is formed, which decreases the drag on the model.

Another example of active flow control is the work done in 2005 by Choi et al. [4]. A two-dimensional circular cylinder was analyzed in a LES with a flow control technique called distributed forcing. The forcing, which is steady in time, varies sinusoidally in the spanwise direction and applied on the upper and lower surfaces of the cylinder. The authors attempted to optimize parameters associated with the forcing such as the forcing angle, forcing velocity, slot width, and forcing wavelength, by sweeping through ranges of values for each of the parameters. Drag savings were reported in this study although the energy required to actively control the flow in this manner was not. In addition, it was also concluded that the drag savings for laminar and turbulent flow over the cylinder was not due to the delay of separation, but to the modified
vortical structure in the wake trailing the body. This finding suggested the distributed forcing technique may provide drag reduction over a body having a fixed separation point.

Therefore, another study was conducted by J. Kim et. al in 2004 [1] where an elliptically-nosed bluff body with a rectangular trailing edge was tested in a two-dimensional large eddy simulation (LES) and a two-dimensional wind tunnel experiment. The forcing is applied on the upper and lower trailing edges where the boundary layer flow suddenly turns into a wake flow. Fig. 1-2 shows a schematic from this particular paper illustrating the distributed forcing technique on the bluff body with a blunt base. In-phase forcing is defined as the same direction of forcing, either suction or blowing, at the same spanwise location on the body. Out-of-phase forcing is one wavelength out of phase in the spanwise location, so a suction slot is at the same spanwise location as a blowing slot.

It is worth noting that in the LES, both in-phase and out-of-phase forcing were simulated, and in-phase forcing was shown to have the greatest drag-reducing potential. Consequently, for the experimental portion, only one in-phase forcing model was built and tested. The wind tunnel experiment was shown to substantially increase the base pressure by 36 and 18% at Reynolds numbers of 20,000 and 40,000 respectively. The experiment also used a smoke wire to make a visual determination of the significantly modified and reduced strength of the vortex street. It should be noted that the values for the variables in the LES differed from those reported in the experimental portion of the paper. The authors explicitly state that they “did not make a significant effort in matching conditions ... with those in LES, because the purpose of the present experiment is to show that the distributed forcing can be realized in a practical situation” [1]. They however, do not report the power required to run the distributing forcing assembly, or if there is enough of a drag savings to overcome the energy being put into manipulating the flow.
The following portion of this paper explores the distributed forcing flow control technique as it pertains to a three-dimensional bluff body with a blunt base. Following the work done by J. Kim et al [1], and Choi [4], this document describes the experimental wind tunnel tests performed on a model designed, manufactured, and tested to show how in-phase distributed forcing affects the flow around a three-dimensional bluff body. This paper will also compare how much power is being put into the flow through the distributed forcing assembly, to how much power is saved in the reduction in overcoming drag, if there is a drag savings.
2. EXPERIMENTAL APPARATUS

2.1 UNIVERSITY WIND TUNNEL

The wind tunnel in the Aerospace Engineering Department at California Polytechnic State University, San Luis Obispo was used for the testing of this project. The open-circuit tunnel, housed in a dedicated laboratory building on the campus has a rectangular test section that measures nominally 3’ by 4’. A photograph of the wind tunnel and laboratory building is shown in Fig. 2-1. It is a draw-down tunnel with a 10:1 contraction ratio which draws air in from vents through the ceiling and roof of the building and exhausting air out through a large rollup door in the far wall of the room. It is powered with a single 440 Volt, 3-phase, 150 horsepower electric motor that drives a fixed-pitch 9-blade fan. The electric motor’s speed is controlled by a SquareD Altivar66 variable frequency drive (VFD). A schematic diagram of the wind tunnel laboratory is shown in Fig. 2-2.

Figure 2-1. Photograph of the Cal Poly Aerospace Engineering wind tunnel as shown from inlet.
This wind tunnel was constructed starting in 1974 by Professor Jon Hoffman, other faculty, and students. It is manufactured almost entirely out of wood, meant to be reconfigurable with additional test sections that can be interchanged to allow a wide range of experiments. It also has a permanent test section that houses a sting balance by Aerolab. The wind tunnel was designed to have a peak velocity of 100 mph (44.7 m/s), but is currently only capable of achieving 78 mph (34.9 m/s) with the current fan unit. The wind tunnel was housed inside the hangar, building 4, until about 2005 when it was moved to its current location, building 41 on the Cal Poly campus.

Figure 2-2. Schematic layout of the Cal Poly Aerospace Engineering 3’x4’ draw-down wind tunnel. Airflow is right to left.

Within the past two years, a new flow straightening inlet of the wind tunnel was installed. The new flow straightener consists of polycarbonate honeycomb and four stainless steel screens. The first two screens are 18-mesh with 1” thick by 1/8” diameter honeycomb in between. A gap of 1.5” follows the second screen where a 20-mesh screen is located. Another gap of 1.5” is then followed by a 22-mesh screen. For a complete description of the inlet remanufacture, see Altmann’s work [8]. The flow quality was then measured to quantify the new inlet flow straightener in Thomas et al. [7]. The reported freestream turbulence intensity was measured to be below 0.5% for most of the test section area. One area in the top center of the tunnel had a
peak turbulence intensity of 2.7% due to undetermined causes, while the freestream velocity variation was found to vary considerably, with several areas over 2% and a peak velocity variation of 3.1%. This did not negatively affect the testing of distributed forcing on the model, as the boundary layer was tripped in order to induce turbulence.

Incline manometers were used to measure the total and static pressures in the freestream flow using a 1/8 inch pitot-static probe mounted at the testing location. The incline manometers are model 40HE35 from Meriam Instrument Co and have a range up to 8 in*H2O, differential. A thermometer and barometer in the wind tunnel lab facility were used to estimate ambient conditions. These two instruments are model number 453, manufactured by Princo Instruments, Inc.

2.2 TEST MODEL

2.2.1 DESIGN

The geometry of the model designed and manufactured for this study was based on the model described in Kim et al [1]. This geometry also follows the studies performed by Tombazis [10]. The main difference is the three-dimensionality of the model tested in this study. The models cited previously span the entire length of the wind tunnel, imitating infinite span by eliminating the vortices generated on the sides of the model. These are commonly referred to as two-dimensional models; only the vortex street shedding off the top and bottom edges is studied. Three-dimensionality is yet another step closer to realizing experimental aerodynamics in a practical situation outside of a laboratory building. Because of this, the aspect ratio becomes of importance and several revisions of the design for this experiment were produced.

Before design work started, a coordinate and direction convention was established which followed many of the same papers previously published. The streamwise direction in the freestream flow is X, the vertical normal direction to the freestream is Y, and horizontal normal freestream direction is labeled Z, with (0, 0, 0) always centered on the middle of the back plate of
the model. The span of the model corresponds to the width of the model, the length of the model is the chord, and the thickness of the model is height, h. The height will be used throughout the paper as the length to nondimensionalize about. The coordinate system and model directions are shown in Fig. 2-3.

All of the design work for this model was accomplished using ProEngineer, a three-dimensional CAD program. This provided a platform which was very easy to manipulate and adjust as more information about the study was obtained. The final outer dimensions of the model depended on many factors including fan size, wind tunnel dimensions and pressure tubing size. The as-tested model has a 15.25 inch length, 9.5 inch span, and a height of 3 inch. The nose is a half ellipse with its ratio of the major to minor axis of 4. The overall dimensions yield a very low blockage ratio of under 2% to avoid wall effects and corrections for buoyancy, velocity, and the boundary layer growth in the wind tunnel.

![Model Diagram](image)

Figure 2-3. Model dimensions and defined coordinate system.

Because this model was based on previous studies in distributed forcing a table is shown below with defining model parameters of similar models created. These parameters will be referenced later as the basis of how the model for this experiment was designed. See figure 1-2 for a pictorial representation of these parameters.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Kim et al. [1] LES model</th>
<th>Kim et al. [1] experimental model</th>
<th>3D model in this experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td>h</td>
<td>h</td>
<td>2.36”</td>
<td>3”</td>
</tr>
<tr>
<td>Length</td>
<td>Not defined in computational domain</td>
<td>15.0”</td>
<td>15.25”</td>
</tr>
<tr>
<td>Width</td>
<td>Not defined in computational domain</td>
<td>12.3”</td>
<td>9.5”</td>
</tr>
<tr>
<td>Nose</td>
<td>Not defined in computational domain</td>
<td>Half-ellipse</td>
<td>Half-ellipse</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Major to minor axis = 8</td>
<td>Major to minor axis = 4</td>
</tr>
<tr>
<td>$\lambda_z$, forcing wavelength</td>
<td>4h</td>
<td>2.5h</td>
<td>1.6h</td>
</tr>
<tr>
<td>b, slot width</td>
<td>0.1h</td>
<td>0.039”</td>
<td>0.04”</td>
</tr>
<tr>
<td>$\beta$, forcing angle</td>
<td>45°</td>
<td>45°</td>
<td>45°</td>
</tr>
<tr>
<td>$\Phi$, forcing velocity</td>
<td>0.1$u_\infty$</td>
<td>4.1 ms$^{-1}$</td>
<td>1.8 ms$^{-1}$</td>
</tr>
<tr>
<td>Reynolds number, $Re_h$</td>
<td>4200</td>
<td>20,000</td>
<td>50,000</td>
</tr>
<tr>
<td></td>
<td></td>
<td>40,000</td>
<td>100,000</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>150,000</td>
</tr>
</tbody>
</table>

Table 2-1. Comparison of parameters from previous distributing forcing literature on bluff bodies with blunt bases.

2.2.2 INTERNAL FORCING ASSEMBLY AND BASE PLATE

Following Fig. 1-2(b) in the Introduction, only in-phase forcing was tested in this study as this was the method which showed the greatest potential for drag reduction in the two-dimensional studies by Choi et al. [4] and Kim et al. [1]. In addition, using these previous works as a guideline, the decision was made to allow for two sets of suction-blowing pairs. In using suction-blowing pairs, the air from the suction slots could be routed to a blowing slot, thus allowing for momentum conservation between pairs. The forcing mechanism was based on small electronic cooling fans because they are inexpensive and readily available. A total of four fans
were used in the model and two different types of fans were tested. Kim et al. [1] also used four small fans in its experimental studies to provide forcing.

One of the key differences between the three-dimensional model designed for this study and the model designed by Kim et al. [1] is the forcing wavelength, $\lambda_z$. Kim et al. [1] states that the maximum drag reduction from the circular cylinder LES [4] was achieved with $\lambda_z = 3 \sim 4d$ where $d =$ cylinder diameter. Therefore, $\lambda_z$ was chosen to be $4h$ for the model vehicle in the LES [1]. Because the authors did not try to precisely match parameters between the CFD simulation and the wind tunnel tests, $\lambda_z$ ended up being $2.5h$ [1] for the experimental model in the same paper. In this three-dimensional model, the aspect ratio is designed to be much smaller than the two-dimensional models, and thus $\lambda_z$ cannot be realized as high as $4h$ without sacrificing aspect ratio. Therefore the parameter $\lambda_z$ was not chosen, but made to be as high as possible in the given model without increasing aspect ratio. This is also shown in Table 2-1 comparing model parameters.

The base plate on the model contains many static pressure ports to allow for the base pressures to be measured. A ProEngineer-produced drawing of the base plate is shown in Fig. 2-4 with the locations of the pressure ports drilled in the model and the locations of the forcing slots. An engineering drawing complete with all dimensions is shown in Appendix E. There are a total of 45 pressure ports on the base with 19 in the spanwise direction spaced roughly 0.5 inch apart and 15 in the normal direction centered on the model spaced roughly 0.2 inch apart. There is also a set of 13 ports centered on a middle slot in the normal direction. This allows for pressures to be recorded directly over a suction or blowing slot depending on the fan orientation. The model was designed so the fan direction can be reversed or the base plate rotated, to allow for any configuration of forcing to be tested. To have only one set of off-centered ports saves space by eliminating potential tubing within the model as well as simplifying the design. However, this is at the expense of ease of configuration changeability.
The design of the interior of the model was mainly determined by fitting both the forcing fans and the pressure tubing within the shell. The experiment called for measuring the base pressure using the static pressure ports and creating the forcing using the fans simultaneously. This created a finite volume issue and leak issue to allow for the tubing and the fans to exist in the same space, while still sealing the cavities between the fans and between the fans and the tubing. Using ProEngineer as the main workhorse in the design process, the best solution found to this problem was installing a triangular covering which routes the pressure port tubing through the dividing blocks which were machined to have an inner cavity. The flexible tubing would then travel through the holes placed in the fan mounting brackets and out through the strut. This design separated each fan to prevent flow interference, and accommodated the flexible tubing routed from the base plate through the strut. Fig. 2-5 illustrates this with screenshots from ProEngineer, and Fig. 2-6 shows a cross-section of one fan chamber and the air flow path through it. Drawings complete with model dimensions for the major components of the model are included in Appendix E for reference.
Figure 2-5. (a) ProE screenshot of model with top-plate coverings removed to allow for inside access. (b) ProE screen shot of cross-sectional view of internal forcing assembly, and coordinate reference.
Figure 2-6. Spanwise cross-section of airflow(red) through fan assembly in model. Covering for static pressure port inserts and forcing slot geometry is also shown.

2.2.3 MODEL MANUFACTURING

Most of the model is built from 6061 series aluminum including the elliptic nose and body. Because aluminum is relatively lightweight, strong, and can be easily tapped to provide threads for screws, it was a natural choice in this application. Manufacturing was fairly straightforward with the exception of the slots in which the forcing occurs and the elliptical shaped nose. The nose was machined by converting the ProEngineer file into G-code through which a milling machine could read and thus produce with minimal human interaction. The forcing slots are 0.04” wide and are at 45° to freestream. The width and angle of the slots was taken from [1], shown to have the highest potential for drag reduction. Due to the depth of the slots and the very high rotational speed recommended for using small diameter milling bits, a 3/4” diameter, 1/32” thick keyway cutter was used instead of a flat end bit. This prevented breakage of the bit and significantly decreased machining time. A close-up view of the forcing
slots and other internal components of the model is shown in Fig. 2-7. The rest of the model pieces were machined on a CNC mill by hand with some minor G-code programming to manufacture some specific cuts.

![Figure 2-7. Close-up view of inside of manufactured model.](image)

As explained previously, the base plate on the model houses the forcing slots as well as the static pressure ports. These ports have stiff metal tubing inserted in order to act as a connection point for the flexible tubing routed through the model. The inserts are stainless steel with dimensions, 1/16 inch OD, 0.022 inch ID. They were cut using a jeweler’s saw to approximately 5/8 inch in length, sanded orthogonally, and bent if needed to avoid interference with the other internal components. The inserts were mounted flush to the outside of the model and held in place with a drop of cyanoacrylate glue. The tubing which connects to the stainless steel inserts is a flexible PVC tube with 1/32 inch ID and 3/32 inch OD, manufactured by Tygon. This provided an interference fit to minimize leaks to the Scanivalve pressure transducers which are talked about later in this section. Fig. 2-8 shows the model without side plates, giving a view of the internal structure.
Figure 2-8. Internal structure of vertically mounted model with side plates removed, 28 mm fans, PVC tubing, and fan power wires connected.

The 180° flow turners are simply hard PVC elbow joints used in plumbing applications. Two elbows are used on each pair of fans, glued together with PVC specific glue and then epoxied onto the aluminum fan mounting bracket. There is also a boundary layer tripping strip of 1/2 inch 3M anti-slip tape model 7551NA located at approximately 50% of the model length.
2.2.4 MOUNTING STRUT

The mounting strut, which elevates the model in the wind tunnel, is a product of Jarred Pinn’s previous work [5]. This 4130 steel strut attaches the sting balance to the model to provide a secure stand for testing. Because the strut was previously designed for Pinn’s studies, the model in this experiment was designed to accommodate the dimensions of the already existing strut. This particular strut is elliptically shaped to decrease the drag associated with it. However, after some preliminary testing, which will be discussed later in the paper, the portion closest to the model was made more streamlined by adding aluminum to taper the aft section of the strut. This helps allow for a smoother transition to freestream, with a fixed stagnation point on the aft section of the strut, decreasing strut interference and providing data from the static pressure ports on the base plate more solely based on the model performance. The strut is also hollow to provide a housed area for tubing, preventing interference to the freestream flow. An additional set of tubes connects the flexible PVC tubing inside the model to the flexible PVC tubing connected to the Scanivalve pressure transducers. This tubing, which is housed inside the strut, is 3003 aluminum tubing with 1/16 inch OD, and 0.0345 inch ID. This tubing was chosen because it is stiff enough to be pushed through the small interior of the strut, yet flexible enough to follow the bend in the strut. Fig. 2-9 shows the strut modification as well as the aluminum tubing inside the strut.
2.2.5 FORCING FANS

The distributed forcing technique requires air to be moved from the boundary layer and freestream flows to another location on the model’s aft section and forced back into the flow. Following the previous studies done with two-dimensional models, a series of fans was used to achieve this effect. Using the methods described by Choi et al. [4] in which many variations were tested in LES modeling, a forcing wavelength of 3h-4h provided the best results when looking at vortex attenuation. However, in a three-dimensional model, the forcing wavelength is minimized considerably because of the finite span that can be used in the model. In addition, the thickness of the model has to allow for the fans to be mounted inside. Because of this, the model used in the experiment only has a forcing wavelength of 1.6h. This is far from ideal when compared to
previous literature, but as the aspect ratio decreases, this trade-off must be realized. Again, the
dimension of the fans was a large limitation in model size but also in the velocity of the fan
forcing through the slots. From several sources, [1], [4], the forcing velocity which showed the
greatest potential is $0.1u_{\infty}$. This is also difficult to achieve, especially at higher freestream
velocities, because the smaller the fans, the less mass flow rate produced and the less static
pressure differential across the fan. For the model manufactured in this study, two sets of fans
were chosen to be used for testing. The first is 40x40x10mm Sunon 12VDC electronic cooling
fan. The model KDE1204PFVX was chosen because it could be powered easily by a lab bench-
top power supply, and produced the highest volumetric flow rate at 9.5 CFM and static pressure
of 0.25 inch*H2O when compared to similar sized models. It also used low power rated at 1.8
Watt, important in active drag control. The second fan was chosen due to its higher static
pressure. Because the system in which the fan is located is a high obstruction flow, a higher static
pressure differential would be more beneficial than a higher volumetric flow rating. The second
fan is 40x40x28mm 12 VDC electronic cooling fan. The model number PMD1204PQB2 has a
rating of 15.3 CFM and static pressure of 0.51 inch*H2O. This also comes at a higher power
rating of 2.6 Watt. These two fans will be referred to as the 10 mm fans and 28 mm fans from
this point on in the report. Photographs of the two types of fan and their respective performance
curves are shown in Fig. 2-10. Additional performance and specification information can be
found in [11] or in Appendix D.
Figure 2-10. Photographs of the forcing fans used in the experiment. Left: 28 mm fan with respective performance curve below. Right: 10 mm fan with respective performance curve below.
2.2.6 LABORATORY BENCH-TOP POWER SUPPLY

A laboratory bench-top power supply was chosen over an internal battery pack to power the fans within the model. The laboratory power supply, being external to the model provided voltage adjustment in real time, without having to turn off the wind tunnel and open the model. The digital readout display also provided a power output value essential for use in the drag power savings calculations. The disadvantage was the wiring to the fans had to travel through the already crowded strut. This however was a minor inconvenience compared to an internal battery pack. The power supply used is a GPS 2303 two-channel from Goodwill Instrument Co., LTD. A photograph of this power supply is shown in Fig. 2-11.

Figure 2-11: Photograph of laboratory bench-top power supply
2.3 SCANIVALVE

The Scanivalve system samples and records pressures from the static pressure ports on the model to see how distributed forcing affects base pressure. The system is a Scanivalve ZOC33/64Px-X1 Ethernet-based, pressure-scanning unit consisting of 8 individual modules with 8 piezoresistive pressure sensors channels per module. This yields a total of 64 separate pressure sensors available. The first four modules, or the first 32 ports, have input ranges of ±10 inches of water (0.3613 psid) with stated accuracy of ±0.15% of full scale (±0.00054 psid). The last four modules have a range of ±1 psid with a stated accuracy of ±0.12% of full scale (±0.0012 psid).

One attribute of this system is it does not read all 64 ports simultaneously. Instead, one port is read at a time, reading all 64 ports every scan, while the user has the option of which ports to write to file. Additionally, the ZOC33 operates with a regulated 65psi feed to operate a pneumatic solenoid that operates a logic gate internally. This pressure was connected directly to a designated port on the unit and fed through a laboratory air supply in the Cal Poly wind tunnel facility. Another part of the Scanivalve system is the RAD3200 analog-to-digital amplifier. This piece of equipment samples the ±2.5 VDC output, amplifies the signal, and then sends it through a USB Extender 3200 to the computer. The RAD software V2.10 runs on the computer connecting the ZOC33 system to a GUI. RadLink controls the system entirely by selecting which ports are written to file and at what rate. A complete description of how to operate the Scanivalve and RadLink software is detailed by Roepke [12] in his master’s thesis work. Fig. 2-12 shows the main hardware components of the Scanivalve system. The adapter plate is described in the Scanivalve Setup section.
2.3.1 SCANIVALVE SETUP

Leaks between the static pressure taps on the model and the Scanivalve ports can cause wildly inaccurate data. Therefore, a thorough leak check was conducted before any testing commenced. This was accomplished by placing a piece of malleable rubber over the model’s pressure tap and the tubing disconnected at the tubing adapter plate. The tubing adapter plate was manufactured to ease installation of flexible tubing into the ZOC33 Pressure Scanner by allowing a larger diameter tubing to be connected. After disconnecting the tubing from the adapter plate, a vacuum pump applies at least 20 inch-Hg to the open end of tubing and the dial observed for 10 seconds. If the vacuum pressure lost more than 1 inch-Hg in that time, the entire line was investigated to fix the leak.

To verify that all ports on the Scanivalve were reading precisely, a pressure manifold with 26 connections was used, and is shown in Fig. 2-13. An accuracy test to the tolerance magnitudes stated by Scanivalve could not be performed due to lack of necessary equipment, and thus only a precision test conducted. The pressure manifold was connected to 25 of the ports on
the Scanivalve tubing adapter plate, while the single lead input into the pressure manifold was attached to a Cal Poly-manufactured, U-shaped manometer. On the opposite side of the manometer, a vacuum pump or a positive pressure pump was connected to apply approximately ±5.5 in-H₂O. Once the constant pressure was applied, the readings were recorded in a new data file on the computer and verified the results were within tolerance. The vacuum pressure data from this test can be seen in Fig. 2-14. All of the port pressures are within 0.0005 psi of each other, indicating the pressure transducers are reading the same value respective to one another. This is beneficial when calculating pressure coefficient, which is measured from, and nondimensionalized by the values recorded from the Scanivalve.

Figure 2-13: Upper Left: 64 port tubing adapter plate connected to ZOC33 Pressure Scanner. Lower Left: 26 connection pressure manifold. Right: U-manometer with vacuum pump.
2.4 STING BALANCE

The Aerolab sting balance is used to directly measure the forces on a mounted body within the wind tunnel. The sting balance houses six strain gauges, one for each degree of freedom: normal force, axial force, side force, pitching moment, rolling moment, and yawing moment. The sting balance also has an angle of attack indicator and motor to adjust the angle of attack by increments of 0.1°. The yawing can be adjusted manually with a hand wheel to within 0.1° as well. All of the voltages from the angle of attack indicator, motor, and the strain gauges are controlled and displayed using LabView. The LabView GUI created for the sting balance is shown in Fig. 2-15.
For this experiment, the drag force is of most importance. Since, by definition, the drag force is parallel to the freestream flow, it is a combination of trigonometric functions of the normal force and the axial force. Ideally, the sting balance would be set at 0° angle of attack, and the axial force would be the only drag force component. However, the normal force did have a component in the drag calculation because there was a slight angle of attack applied to level the model within the tunnel due to machining tolerance accumulations. This angle is verified using a digital level placed on top of the model, calibrated to the wind tunnel floor and accurate to 0.1°.

Figure 2-15. Screenshot of the LabView GUI which controls the sting balance angle of attack and strain gauge data collection.

A serial cable connects the sting balance strain gauges to an amplifier that increases the voltage signal produced by the sting balance. The signal is then transmitted to the computer through a data acquisition card and recorded through a LabView program. The angle of attack components are controlled through separate cabling, but use an integrated portion of the same LabView program.
In order to ensure repeatability in the sting balance portion of the experiment, a calibration was performed on the strain gauges. The normal and axial forces were both calibrated in order to find the linear relationship between the voltage output of the sting balance and the force applied. At zero angle of attack, standard weights from 0 to 7 pounds, in 1 pound increments, were placed on the sting balance in the manner dictated by the force direction being calibrated. A schematic diagram in Fig. 2-16 shows how the calibration is performed to obtain relationships for both the axial and normal forces, and a photograph of the axial force calibration is shown in Fig. 2-17. For the axial calibration, a window was opened in the floor of the tunnel to allow the top of the pulley to be parallel to the streamwise direction. Successive calibration runs were conducted while increasing and decreasing the weight amounts to verify that hysteresis has no effect on the sting balance strain gauges. The calibration curves and their respective linear regression curves can be found in Appendix B.

Figure 2-16. A schematic diagram showing calibration of the Aerolab sting balance strain gauges in the normal and axial directions.
2.5 HOT WIRE ANEMOMETER

A hot wire anemometry system was used in this experiment to measure the velocity fluctuations off of the bluff body. These velocity fluctuations correspond to the vortex street shedding at a particular frequency and the velocity of the air within with the vortices formed. It is of interest to plot the Strouhal number against the energy of the vortices to determine if distributed forcing has any effect in attenuating or intensifying the vortex street.

Hot wire anemometers use a very fine wire, on the order of several micrometers, electrically heated up to a temperature above the ambient. Air flowing past the wire lowers the temperature and thus changes the electrical resistance. A relationship between the velocity of the air and the resistance of the wire can be obtained and the equipment used in velocity determination. Because the wire is very thin, it allows for very high frequency response and fine spatial resolution. In addition, dual sensor wires or three channel wires can be manufactured in order to obtain 2D or 3D velocity directions.

This lab uses a constant-temperature anemometer (CTA) in which the wire maintains a set temperature of 250 °C. The primary control unit is the IFA-300 by TSI, which currently
supports a single channel. The hot wire probe used in testing was a TSI model 1241-20 X-wire probe with serial number 70611547. Because this particular sensor is a dual channel sensor and was the only readily available probe for this experiment, a single channel is activated on the IFA-300 to observe the change in velocity. Fig. 2-18 shows the TSI hot wire used in this experiment.

Figure 2-18. Photograph of TSI X-wire probe used in testing.

To mount the hot wire probe in the wind tunnel, a window was modified to allow for a transverse beam to be installed. This allows the probe to be placed in a specific location behind the model with three degrees of freedom: spanwise, streamwise, and normal. This modification is a very simple design to achieve three spatial directions at the expense of positional accuracy, which is not as important for this portion of the experiment.

The window is held in place by four manual toggle clamps and sealed with aluminum tape on the inside of the tunnel and window seam, preventing air leaks. The transverse beam which travels through a milled slot in the window is held in place by four wing nut screw clamps. These are tapped into the acrylic window to prevent slippage. To adjust the vertical or normal direction, these screw clamps can be loosened and the beam assembly slid up or down on the window. To locate the hotwire in the spanwise direction, the red C-clamps outside of the wind tunnel can be loosened, and the transverse beam slid in or out of the window. Finally, to adjust the streamwise location of the hot wire probe, the four screws which clamp the probe support rod
at the end of the transverse beam can be loosened and the probe support rod can be located
towards or further away from the model. Once the hot wire probe was placed in the tunnel,
aluminum tape was placed over the milled slot in the window to seal leaks in that location as
well. Fig. 2-19 shows the modified window used in mounting the hot wire probe and the hot wire
mounting setup.

Figure 2-19. Photographs of modified window and mounting setup of hotwire testing equipment.

2.5.1 HOT WIRE CALIBRATION

Calibrating the hot wire anemometry system is essential is obtaining a correlation
between output voltage and flow velocity. The hot wire probe was calibrated using a TSI model
1128A, with a 0 to 10 mm*Hg pressure transducer. The calibration equipment integrates a secure
mount for the probe support with adjustable angles to allow for multi-channel probes, and a
pressure transducer to measure differential pressures. A compressed air supply is fed though a
filter and pressure regulator, and then controlled with two valves on the 1128A. The coarse and
fine adjustment valves allow for changing the flow velocity output through the nozzle, and the output pressures are read through the pressure transducer module. The calibrator also contains a thermocouple to measure the offset needed at atmospheric conditions. Fig. 2-20 shows the hardware used in calibrating the hot wire anemometry system.

Figure 2-20: Photograph of hot wire anemometry calibration equipment.

Thermalpro software was installed to calibrate, take data, and compile data from the thermal anemometry system. In calibrating the probe, 17 points were generated by the software ranging from 0 to 40m/s, with clustering at slower flow velocities. The flow velocity though the
nozzle, controlled by the adjustment valves on the 1128A, was changed until the differential pressure matched the pressure displayed by the software. This change in resistance of the hot wire probe was measured and recorded, until all 17 points were completed. A curve was fitted to this data using King’s law, which accurately describes the trend between velocity and probe voltage. For a more in depth routine on how to use the thermal anemometry system, see the system instructional manual [9]. Following the calibration routine, the probe and probe support were removed from the calibration equipment and placed in the mounting system in the tunnel for testing.
3. ANALYSIS

This section is meant as a brief reference describing the major coefficients and nondimensional numbers in this paper and how they are calculated.

3.1 DRAG COEFFICIENT

Drag coefficient is a dimensionless quantity that is used to quantify the drag or resistance of an object in a fluid environment. In this experiment, it is calculated after utilizing the established linear relationship between voltage and force on the strain gauges. The equation for drag coefficient is as follows:

\[ C_D = \frac{F_{Drag}}{q_\infty A} \]

Where \( F_{Drag} \) is the drag force, \( q_\infty \) is the freestream dynamic pressure, and \( A \) is the frontal planform area. For the model in this experiment, \( A \) is calculated as the span multiplied by height:

\[ A = h \times \text{span} = 3 \times 9.5 = 28.5 \text{ in}^2 \]

An example of the propagation of uncertainties for drag coefficient is shown in Appendix A.

3.2 PRESSURE COEFFICIENT

The pressure coefficient is a dimensionless number which describes the relative pressures throughout a flow field. The pressure coefficient is calculated for all the pressure ports on the model and plotted to show the pressure distribution on the base plate. The pressure coefficient is given as:

\[ C_p = \frac{P - P_\infty}{q_\infty} = \frac{P - P_\infty}{P_T - P_\infty} \]

Where \( P \) is the relative pressure as measured by the static pressure ports on the model, \( P_\infty \) is the freestream static pressure, and \( P_T \) is the total or stagnation pressure. In the experiment, \( q_\infty \) is calculated as the total pressure less the freestream static pressure by using a pitot-static probe at
the test section connected to an incline manometer. An example of the propagation of uncertainties for pressure coefficient is shown in Appendix A.

3.3 STROUHAL NUMBER

In dimensional analysis, the Strouhal number is a dimensionless number describing oscillating flow mechanisms. It is useful in the hot wire portion of this experiment to analyze the vortex shedding frequency as a function of body height and freestream flow velocity. The equation for Strouhal number is as follows:

$$St = \frac{f \cdot h}{u_{\infty}}$$

Where $f$ is the frequency of the vortex shedding, $h$ is the model height and $u_{\infty}$ is the freestream velocity.

3.4 MOMENTUM COEFFICIENT

One can represent the ratio of forcing velocity to freestream velocity by using a momentum coefficient. The equation for steady momentum coefficient is given by,

$$C_u = \frac{J}{h \cdot q}$$

Where $J$ is the steady jet momentum, $h$ is the base height and $q$ is the dynamic pressure. $J$ is given by the equation,

$$J = \rho \cdot U_j^2 \cdot t$$

Where $\rho$ is density, $U_j$ is jet velocity as found using the hot wire equipment, and $t$ is thickness of jet slot. This equation, normally used for forcing constant along the span of the model, is assumed quasi-2D in this instance, and is the measure of the local forcing strength. When reducing this equation for the model in this experiment, and using the values, $t = 0.04”$ and $h = 3”$, the following is obtained,
Using this equation for the hot wire-measured forcing velocity of 1.846 m/s and freestream velocities of 10, 20 and 30 m/s, the resulting momentum coefficients are 8.94e-4, 2.24e-4, and 9.94e-5, respectively. This shows that the forcing velocity has less and less effect on the freestream flow and therefore less effect on the base pressure.

3.5 REYNOLDS NUMBER

The Reynolds number is a dimensionless number that gives a measure of the ratio of inertial forces to viscous forces and consequently quantifies the relative importance of these two types of forces for given flow conditions. In this experiment, the Reynolds number is calculated based on model height, instead of model length. Because of this, the equation for Reynolds number is as follows:

\[ Re_h = \frac{u_\infty \cdot h}{v} \]

This equation yields Reynolds numbers of 5.0e4, 1.0e5, and 1.5e5 at freestream velocities of 10, 20, and 30 m/s. This assumes a kinematic viscosity of 1.52e-5 m²/s, which is approximately the viscosity of air at 70 °F, close to the usual operating conditions of the wind tunnel in San Luis Obispo during the summer season.
4. RESULTS AND DISCUSSION

4.1 WIND TUNNEL CALIBRATION

It is necessary to provide consistent and repeatable results during testing, and therefore a relationship between velocity of the wind tunnel and tunnel control panel needed to be established. Jarred Pinn, in his thesis work [5], recently performed an experiment to determine the linear correlation of the input frequency of the SquareD Altivar66 wind tunnel control panel to the dynamic pressure of the freestream flow at the testing location. This test was verified using a sweep of frequency inputs from 10 Hz to 45 Hz in 5 Hz increments. A pitot-static probe at the model location measured the total and static pressures which were recorded using the Scanivalve pressure transducers, and the final plot of frequency input to velocity is shown in Fig. 4-1. The frequency inputs which corresponded to 10, 20, and 30 m/s were the same as those found in [5], thus verifying the work previously completed. This corresponds to Reynolds numbers based of model height, h, of approximately 5.0e4, 1.0e5, and 1.5e5 respectively. Once this relationship had been determined, experimental testing on distributed forcing could commence.

![Figure 4-1. Experimentally-determined correlation graph of input frequency to test-section velocity.](image)
4.2 TESTING OVERVIEW

This section is meant as a guide to document the testing schedule, as the mounting orientation and model were changed throughout the process. Two main testing phases can be separated into the first round of testing and the second round of testing. Table 4-1 organizes this information and the main differences in the testing parameters.

The mounting orientation refers to the orientation of the spanwise pressure ports on the model. ‘Vertically’ denotes the span of the model in the direction from the top of the wind tunnel to the bottom, while ‘horizontally’ denotes the span of the model in the direction from the left to the right walls in the wind tunnel. This difference is important because of the strut mounting location on the model. It will be found in the following section, the strut has a significant effect on the base pressure distribution.

The fan type refers to either of the two different fans used in this experiment, 10 mm or 28 mm thick fans. Different voltages are applied to the fans to get an idea of how the forcing velocity influences base pressure. The sweep of voltages was only recorded in the very preliminary stages of testing. The direction of the fans is the direction of the forcing through each of the four pairs of slots on the trailing edges. Since only in-phase forcing was considered, the direction of the forcing is the same at the same spanwise location on both the upper and lower trailing edges. ‘B’ denotes a blowing slot, ‘S’, a suction slot, and ‘Ø’ refers to the fans turned off, but not removed from the model.

The model length was initially much shorter than the final configuration. A center section of 4 inches was added to the model in the second round of testing in order to achieve the same length-to-height ratio as Jarred Pinn’s model [5]. This was in part due to some concern about not fully-developed flow. Lengthening the model increases the Reynolds number based on length to allow for turbulent flow to develop by the time the boundary layer reaches the forcing slots at the separation point on the edge of the model. It is expected that turbulent flow is present.
at all testing velocities with the addition of a tripping strip of anti-slip tape, and Reynolds number based on length of 250,000 at 10 m/s, the slowest testing velocity.

<table>
<thead>
<tr>
<th></th>
<th>FIRST ROUND</th>
<th>SECOND ROUND</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mounting orientation</td>
<td>Span Vertical</td>
<td>Span Horizontal</td>
</tr>
<tr>
<td>Fan type</td>
<td>10 mm</td>
<td>28 mm</td>
</tr>
<tr>
<td>Fan Directions</td>
<td>BSBS, SBSB, BSSB, BØØB</td>
<td>BSSB, BØØB</td>
</tr>
<tr>
<td>Fan Voltages, Volts</td>
<td>5, 8, 10, 12, 13.5</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>Only 12 Volt tested in BSSB, BØØB cases</td>
<td></td>
</tr>
<tr>
<td>Model Length, inch</td>
<td>11.25</td>
<td>15.25</td>
</tr>
</tbody>
</table>

Table 4-1. Parameter differences between the first and second rounds of testing.

4.3 FIRST ROUND OF TESTING

The first series of tests are a preliminary approach to see how the distributed forcing technique influenced the base pressure. These results are not documented in the abstract or conclusion. For the results of the testing of the model as described in the experimental apparatus section, see Second Round of Testing.

To begin this series of tests, the model was first mounted with its spanwise direction vertically; the mount secured to the thinner side of the model. A schematic and photo of this mounting orientation is shown in Fig 4-2, which followed the mounting orientation of [5]. Additionally, the smaller 10 mm thick fans were installed on the model. The Scanivalve was used to measure the base pressure on the static pressure ports to acquire an initial evaluation of the distributed forcing technique on a three-dimensional model. The sting balance and hotwire were only used in the Second Round of Testing.
Following the distributed forcing definition, there are two main circumstances with the model mounted in this orientation: a suction slot next to the strut (SSNTS), or a blowing slot next to the strut (BSNTS). The pressure results of these two circumstances are documented in Fig. 4-3 through 4-8 below. BSNTS is presented first in Fig. 4-3, Fig. 4-4, and Fig. 4-5 at 10, 20 and 30 m/s, or Re_h of approximately 5.0e4, 1.0e5, and 1.5e5 respectively. The results of SSNTS are shown in Fig. 4-6, Fig. 4-7, and Fig. 4-8. Both the spanwise pressure distribution and the normal distribution at Z/h = 0 are shown in each figure. The normal distribution at Z/h = 0.3 is not presented in this report and will be discussed later. A sweep of voltages was performed for each test from 4.5 Volts to 13.5 Volts, which is just after the rated start-up voltage of the fan to just under the rated maximum voltage of the fan respectively. It should also be noted that the strut is located at approximately Z/h = 1.6 for each orientation.

Fig. 4-3 shows the centered normal and spanwise pressure distributions on the base plate of the model for the orientation BSNTS. The spanwise vortex shedding presents itself in the data as pressure dips at approximately Z/h = ±1. The normal vortices are not as conspicuous, but there
are traces from the “W” shape of the data. However, in both sets of data, the benefits of increasing the voltage to the fans can be seen. As the voltage of the fans increases, base pressure increases. It is interesting to note that in the normal direction, the highest base pressure is recorded at 8 Volts, indicating that exceeding a maximum forcing velocity for the given freestream speed may have adverse effects on the base pressure. That is, there may be a specific forcing velocity as a function of Reynolds number to have the greatest effect on base pressure.

Fig. 4-4 and Fig. 4-5 show the centered normal and spanwise pressure results at 20 and 30 m/s respectively. The same trends occur as found in the lower Reynolds number case; however they are much less dramatic. There is still a larger increase in base pressure when a higher voltage is supplied to the fan assembly in both the normal and spanwise directions on the base plate; however the same 13.5 Volts applied in a freestream of 30 m/s doesn’t nearly have the same effect on the model at 10 m/s. This shows the forcing velocity is important in reducing the strength of the vortical structure behind the model, and thus increasing base pressure.

The next three figures, (Fig. 4-6, Fig. 4-7, Fig. 4-8), show the pressure results of the configuration SSNTS. This configuration was achieved by switching the direction of the fans within the model. Fig. 4-6 shows the same pressure results as Fig. 4-3 but in the configuration SSNTS. This figure shows the same trend of increased voltage yielding a higher increase in base pressure. However, instead of the location of the largest increase in base pressure being next to the strut, as seen in BSNTS configuration, the largest increase in base pressure is away from the strut in SSNTS configuration. This seems to indicate the largest base pressure increase can be achieved when a blowing slot is at the end of the span in a three-dimensional model. This will be addressed in the next fan configuration.

It is also of interest to note the pressure distribution in the normal direction along Z/h = 0. In the BSNTS case, there was an overall increase in base pressure when distributed forcing was applied to the model. In the SSNTS orientation, there is an overall decrease in base pressure along Z/h = 0. This contradictory information can be explained by looking at the spanwise
pressure distribution at $Z/h = 0$. Looking at this location on the spanwise plots and noting the relative order of applied voltages, one can see the same order of data on the normal distribution graphs. This is due to the discrepancy in symmetry between the model in the wind tunnel and the forcing which is applied. Since the model is symmetric geometrically, and mounted orthogonally in the wind tunnel, the forcing applied asymmetrically will skew the pressure on the model depending on the direction of the forcing. The strut has little primary influence in the normal direction with the model mounted vertically, however if one of the side vortices is influenced more than the other, an asymmetry can be shown in the spanwise data. This influences the location of the center of symmetry of the pressure data in the spanwise direction and therefore the relative order of applied voltages in the normal direction. This strut interference is also shown in the case where no forcing is applied. Looking at the spanwise data at 0 Volts for all speeds, in both BSNTS and SSNTS orientations (as they should be the same without forcing), there is still a larger pressure drop at $Z/h = 1$ when compared to $Z/h = -1$. $Z/h = 1.6$ is approximately where the strut is located in all configurations. For this reason, the normal direction data does not provide a global interpretation of the effect the fans have on base pressure. Because the normal direction is centered on the model, while the vortex shedding is not, the trend of base pressure increase to fan speed is shown best in the spanwise direction. Therefore, only the spanwise data is presented following this section. This is also the reason why the normal pressure data at $Z/h = 0.3$ is not shown at all. The overall increase or decrease in base pressure was not a good indication in the global effects of distributed forcing and in addition, did not show any vortex shedding. This was most likely due to the position of the ports and the location of the first port from the edge. The first port at $Z/h = 0.3$ is located nearly $0.12h$ from the edge of the model, unable to capture effects from distributed forcing. This also unfortunately was the closest a static pressure port could be manufactured without interfering with the forcing slots.
Figure 4-3. Graph of back-plate normal and spanwise $C_p$ results for BSNTS, 10 m/s. Strut located at $Z/h = 1.6$. 
Figure 4-4. Graph of back-plate normal and spanwise $C_p$ results for BSNTS, 20 m/s. Strut located at $Z/h = 1.6$. 
Figure 4-5. Graph of back-plate normal and spanwise $C_p$ results for BSNTS, 30 m/s. Strut located at $Z/h = 1.6$. 
Figure 4-6. Graph of back-plate normal and spanwise $C_p$ results for SSNTS, 10 m/s. Strut located at $Z/h = 1.6$.  

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Figure 4-7. Graph of back-plate normal and spanwise $C_p$ results for SSNTS, 20 m/s. Strut located at $Z/h = 1.6$. 
Figure 4-8. Graph of back-plate normal and spanwise $C_p$ results for SSNTS, 30 m/s. Strut located at $Z/h = 1.6$. 
The flatlining of the spanwise plots at a blowing slot near the end of the model shows an attenuation of the vortex street. This attenuation has been linked to drag reduction and of interest in this study. Because the fan direction can be changed easily and blowing slots created at the span ends of the model, the orientations BSSB and BØØB were explored. Fig. 4-9, Fig. 4-10, and Fig. 4-11 show the pressure results as recorded in the spanwise direction for BSSB and BØØB at 10, 20, and 30 m/s.

Fig. 4-9 shows the pressure results at Reₐ = 5e4 with good pressure recovery away from the strut and at the suction location Z/h = 0.3. However, the blowing location closest to the strut does not show the same pressure recover as was found in BSNTS, Fig. 4-3. The BØØB case in Fig. 4-9 shows the same trend as the BSSB at the same freestream velocity, but not as high of pressure recovery. This could be beneficial when calculating the drag power savings as two of the fans are disconnected, halving the amount of power used while still maintaining a relatively high pressure recovery as compared to all four fans connected.

Fig. 4-10 and Fig. 4-11 show the spanwise pressure distributions for the 20 and 30 m/s cases. The differences between BSSB and BØØB are minimal with an overall pressure decrease with the forcing fans powered. There is still asymmetry found in the data, which is most likely due to the strut interfering with the vortex shedding off that edge of the model, and will be addressed in the second part of testing. At this point in the experiment, with the knowledge gained in the first round of testing, some changes were made to the model to attempt to increase the benefits of distributed forcing as much as possible.
Figure 4-9. Graph of spanwise back-plate $C_p$ results at 10 m/s. Strut located at $Z/h = 1.6$. Top: BSSB configuration. Bottom: BØØB configuration.
Figure 4-10. Graph of spanwise back-plate $C_p$ results at 20 m/s. Strut located at $Z/h = 1.6$. Top: BSSB configuration. Bottom: BØØB configuration.
Figure 4-11. Graph of spanwise back-plate $C_p$ results at 30 m/s. Strut located at $Z/h = 1.6$. Top: BSSB configuration. Bottom: BØØB configuration.
4.3.1 MODEL CHANGES

For all previous cases where the fans are unpowered, the spanwise data is asymmetrical about $Z/h = 0$, even with care to align the model with the wind tunnel walls to 0.1° tolerances. This asymmetry is not evident in the normal direction which seems to indicate interference of the strut with the vortices shedding off the strut side of the model. In order to lessen this interference, an aluminum insert was machined to better satisfy the Kutta Condition, keeping the separation point off of the strut fixed to one position. The insert also helped to decrease the interference drag with the addition of a wax fillet to the interior corners between the model and strut. The strut was also moved relatively closer towards the nose, away from the base of the model by adding a section to elongate the length. This was achieved by adding a 4 inch rectangular section between the nose and aft fan assembly. Strut mounting holes were machined in this section to allow the model to be placed in the wind tunnel as before. The longer model was tested vertically in the same orientation as previously shown, Fig. 4-2, however the data from that experiment suggested there was still an unacceptable level of strut interference. Therefore, the model was mounted horizontally.

The final change to the model dealt with the fan static pressure. The data in the first round of testing suggests that a more powerful fan, creating a faster forcing velocity to freestream velocity ratio will increase base pressure further. Therefore the more powerful 28 mm thick fans were installed in the model. In addition, the forcing velocity was measured using the hotwire and compared with the work previously completed by Kim et al. [1].
4.4 SECOND ROUND OF TESTING

The second round of testing follows with an elongated model and 28 mm fans installed. The horizontal mounting orientation was achieved by rotating the model 90° to align the strut with the normal pressure ports, rather than the spanwise pressure ports. A schematic diagram of the mounting orientation is shown in Fig. 4-12. A new section was manufactured adding 4 inches to the length between the elliptical nose and the aft fan assembly. An additional strut mounting hole was machined into the new section, and the other strut mounting holes were filled with aluminum plugs and sealed with aluminum tape. The second round of testing began with recording data from the static pressure ports in the spanwise direction.

![Diagram of horizontal mounting orientation and fan directions.](image)

Figure 4-12. Schematic diagram of horizontal mounting orientation and fan directions.

4.4.1 SCANIVALVE PRESSURE RESULTS

As stated previously, the model was fitted with the more powerful fans and only 12 Volts was applied instead of sweeping through a range of voltages. BSSB and BØØB were only tested as well because of the advantages shown in the pressure data with a blowing slot next to the end of the span.
Fig. 4-13, Fig. 4-14, and Fig. 4-15 show the spanwise pressure results for 10, 20, and 30 m/s for BSSB and BØØB. The normal direction was recorded but not presented in this paper, as it was very similar to the first round of testing cases.

Fig. 4-13 at 10 m/s showed the greatest attenuation of vortices and highest increase in base pressure as compared to the higher Reynolds number cases. In addition, BSSB showed a much higher pressure recovery than compared to BØØB. These are the same trends as established previously; however the graphs are much more symmetrical with the strut mounted on the larger-faced side of the model.

As the freestream velocity increased, the ratio of forcing velocity to freestream velocity decreased, and the effect of distributed forcing lessened, actually having a slightly negative effect on base pressure. These differences between distributed forcing in the BSSB orientation and the no forcing control orientation were a 15.3% and 4.2% increase in mean base pressure at Reynolds numbers of 50,000 and 100,000 respectively. At the Reynolds number of 150,000 a 2.7% decrease in mean base pressure was recorded.

In Fig. 4-14 and Fig. 4-15, BØØB showed the same trends however, for both 20 and 30 m/s, a slight decrease in base pressure was recorded. BØØB doesn’t have the effect that all four fans have, even though the forcing velocities measured are similar on the blowing locations. This shows that suction slots, in addition to the blowing slots, in the BSSB orientation have an effect the base pressure, and should be powered for the greatest increase in base pressure.

The small discrepancies of symmetry about Z/h = 0 are due to having to plug and tape the old strut mounting locations. Care was taken to make the plugs flush but the tape edges and dips from gaps affect the boundary layer and flow slightly. However, having showed a promising case in drag reduction in the lower Reynolds numbers in the BSSB orientation, the sting balance was then used to directly measure the forces on the model.
Figure 4-13. Graph of spanwise back-plate $C_p$ results at 10 m/s. Strut located at $Z/h = 0$. Top: BSSB configuration. Bottom: BØØB configuration.
Figure 4-14. Graph of spanwise back-plate $C_p$ results at 20 m/s. Strut located at $Z/h = 0$. Top: BSSB configuration. Bottom: BØØB configuration.
Figure 4-15. Graph of spanwise back-plate $C_p$ results at 30 m/s. Strut located at $Z/h = 0$. Top: BSSB configuration. Bottom: BØØB configuration.
4.4.2 STING BALANCE TESTING

The sting balance was used to directly measure the forces on the model with and without forcing at 10, 20 and 30 m/s. The voltage signal from the strain gauges was converted to a force using an experimentally determined linear relationship. This relationship can be found in Appendix B. The drag coefficient was found by nondimensionalizing force by dynamic pressure and frontal area. In all previous studies researched, the power required to run the distributed forcing technique has been omitted from the drag savings calculation. This is mostly due to the difficulty in estimating the power for a fan, as most of the studies are in two-dimensional CFD simulations.

In this particular experiment, the power required to run the fans for the distributed forcing method is presented and compared to any drag savings encountered in Table 4-2. This provides an insight to see if the active drag control method was installed on a vehicle, enough drag savings would be sufficient to overcome the power required to run the fans.

Briefly omitting the power used to drive the fan assembly, the total power needed to overcome drag is estimated using the force of drag at each Reynolds number and the velocity of the flow at that Reynolds number. The drag power savings is simply the power needed to overcome drag without the fans less the power needed to overcome drag with the fans. This can be written as,

\[
Total \ Drag \ Power \ Saved
= \ Power \ needed \ to \ overcome \ drag \ without \ fans
- \ Power \ needed \ to \ overcome \ drag \ with \ fans
\]

Where,

\[
Power \ needed \ to \ overcome \ drag = F_{Drag} \cdot u_\infty
\]

\[
F_{Drag} = C_D \cdot q_\infty \cdot A
\]
Table 4-2 shows the drag power calculations for each of the cases, the drag power savings and the power required to run the fans at 12 Volts.

<table>
<thead>
<tr>
<th>Drag power without forcing, Watts</th>
<th>Drag power with forcing, Watts</th>
</tr>
</thead>
<tbody>
<tr>
<td>10 m/s</td>
<td>20 m/s</td>
</tr>
<tr>
<td>3.99</td>
<td>24.6</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Drag Power Difference, Watts</th>
<th>10 m/s</th>
<th>20 m/s</th>
<th>30 m/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Percent Difference, %</td>
<td>15.8</td>
<td>5.4</td>
<td>-2.1</td>
</tr>
</tbody>
</table>

Table 4-2: Power required to overcome drag with and without forcing at different Reynolds numbers for BSSB orientation.

Table 4-2 shows that the distributed forcing technique is effective at the lower Reynolds number flows, given a constant fan forcing power. This is consistent with the previous findings of Kim et al. [1], and other literature published in this field of study. Kim et al. [1] reports a drag savings of approximately 36% and 18% at Reynolds numbers of 20,000 and 40,000 for two dimensional models. This experiment showed a drag reduction of 15%, 5% and -2% for Reynolds numbers of 50,000, 100,000 and 150,000, without taking into account the power required to run the forcing assembly. When taking into account this power expenditure, a large loss in overall power is realized. Nearly 7.5 Watts is required to power the fan which is more than 11 times the drag power savings at \( \text{Re}_b = 50,000 \), and more than 5 times the savings at \( \text{Re}_b = 100,000 \). This shows distributed forcing is highly impractical, unless a more efficient forcing technique can be developed. It is interesting to note that a higher power savings is shown at 20
m/s than at 10 m/s, however the percent difference of drag power savings at that particular Reynolds number is lower since the power required to overcome drag at 20 m/s is approximately 6 times that at 10 m/s.

Because BSSB used four fans for the forcing technique, the power required to run them is naturally twice that as compared to BØØB. Performing a sting balance force calculation for drag power savings at the freestream velocity of 10 m/s yields a drag power difference of 0.078 Watts. This still yields a drag savings of 1.6%, without taking into account the power required to run the fans. Because of this, the BØØB orientation is further from achieving an overall drag reduction when compared with BSSB.

The data from the sting balance shows that although the drag reduction is present at the lower Reynolds number flows, the power needed to run the fans requires spending much more energy for drag reduction than is saved with the distributed forcing technique.
4.4.3 HOT WIRE TESTING

The hot wire was meant as a demonstration to verify the attenuation of the vortex structure and an exercise in using the hotwire equipment. ThermoPro software was used in order to calibrate, record, and process the voltage signal from the probe. See TSI’s manual [9] for more information on how this is achieved. The hotwire collects data at 1 kHz with a total data size of 16 Kpts and sampling period of 100 µs. Again, this data is recorded at freestream velocities of 10, 20, and 30 m/s, approximate Reynolds numbers of 50,000, 100,000 and 150,000 respectively.

Thermalpro is able to analyze the data collected and conduct a Fast Fourier Transform (FFT) to produce energy spectral density. This value was plotted against Strouhal Number using the model height of 3 inches for h. 1,024 points per FFT were used resulting in a 0.977 Hz resolution. The dominant peak in the graphs is the result of the vortex shedding at the particular Strouhal number.

Exercising the coordinate system established in Fig. 2-3, with (X, Y, Z) directions and origin centered on the middle of the base, the hotwire was placed at specific locations to show the vortices in both the spanwise and normal coordinate directions. A sweep of locations was performed to attempt to isolate the two vortical structures. The final location for the normal vortex street is (h, h/2, 0). The spanwise data collection point is located at (h, 0, h) consistent with the largest spanwise pressure dip location on the Scanivalve pressure results at Z/h = 1.

Fig. 4-16, Fig. 4-17, and Fig. 4-18 show the results of plotting the energy spectrum against the Strouhal number at 10, 20, and 30 m/s respectively. Each figure shows the spanwise and normal energy spectrum plotted against Strouhal number. The dominant peak in all of the graphs is the nondimensionalized shedding frequency, and occurs at Strouhal numbers of just under 0.3. The secondary peaks are the harmonic frequencies at twice and thrice the shedding frequency. Comparing the three freestream velocities, the largest difference between forcing and no forcing is at the slowest freestream velocity in Fig. 4-16. This is expected from the previous tests of the Scanivalve and sting balance. The greatest attenuation of the vortex shedding is in the
spanwise location at 10 m/s in Fig. 4-16 which shows the dominant peak much smaller in magnitude and less defined than the results without forcing. As the peak is still present, distributed forcing does not eliminated the vortex structure entirely. This is also consistent with the spanwise vortex pressure results of the Scanivalve. As the freestream velocity is increased, the result of vortex attenuation is lessened, to the point where distributed forcing shows nearly no difference between forcing and no forcing. Fig. 4-18 at Reynolds number of 150,000 shows how similar the data between forcing and no forcing is, indicating distributed forcing has nearly no effect on vortex attenuation. A table showing the Strouhal numbers for the run cases presented in BSSB orientation is shown below as well.

<table>
<thead>
<tr>
<th>Position</th>
<th>Freestream Velocity, m/s</th>
<th>Forcing</th>
<th>Dominant Peak, St</th>
<th>Secondary Peaks, St</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(h, h/2, 0)</td>
<td>10</td>
<td>No Forcing</td>
<td>0.261</td>
<td>1.042</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Forcing</td>
<td>0.290</td>
<td>0.573</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>No Forcing</td>
<td>0.264</td>
<td>0.525, 0.785</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Forcing</td>
<td>0.275</td>
<td>0.547, 0.789</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>No Forcing</td>
<td>0.265</td>
<td>0.528</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Forcing</td>
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<td>0.538</td>
</tr>
<tr>
<td>Spanwise</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>(h, 0, h)</td>
<td>10</td>
<td>No Forcing</td>
<td>0.260</td>
<td>0.528</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Forcing</td>
<td>0.290</td>
<td>None</td>
</tr>
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<td>No Forcing</td>
<td>0.264</td>
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</tr>
<tr>
<td></td>
<td>30</td>
<td>No Forcing</td>
<td>0.265</td>
<td>0.533</td>
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<td></td>
<td></td>
<td>Forcing</td>
<td>0.270</td>
<td>None</td>
</tr>
</tbody>
</table>

Table 4-3. Strouhal numbers from spectral density plots for normal and spanwise locations at all Reynolds numbers tested for BSSB orientation.
Figure 4-16. Energy Spectrum and Strouhal Number at 10 m/s for BSSB orientation. Top: Normal location (h, h/2, 0) Bottom: Spanwise location (h, 0, h).
Figure 4-17. Energy Spectrum and Strouhal Number at 20 m/s for BSSB orientation. Top: Normal location (h, h/2, 0) Bottom: Spanwise location (h, 0, h).
Figure 4-18. Energy Spectrum and Strouhal Number at 30 m/s for BSSB orientation. Top: Normal location (h, h/2, 0) Bottom: Spanwise location (h, 0, h).
The three-dimensional bluff body with a blunt base found in Pinn’s studies [5], and the model studied in this paper are compared to one another using Strouhal number and Reynolds number. The control data was compared, i.e. the model in Pinn [5] without end-plate tabulations and the model in this study without forcing. Over this small range of Reynolds numbers, it is expected that Strouhal number remain fairly constant. It is unknown why the Strouhal number in Pinn [5] differs by nearly 14% around Reynolds number of 80,000 – 85,000.

Figure 4-19. Comparison of Pinn’s [5] results of Strouhal number against Reynolds number without flow control devices.
5. CONCLUSION

This paper explored the effects of an active drag control method known as distributed forcing on a 3D bluff body with a blunt base. A number of different forcing orientations were tested including BSBS and SBSB, BSSB, and BØØB, where “B” denotes a blowing location, “S” is a suction slot, and “Ø” denotes an unpowered slot. It was found early in the testing that the BSSB orientation showed the greatest drag reduction potential and was studied further.

The BSSB forcing orientation increased base pressure by 15.3% and 4.2% at Reynolds numbers of 50,000 and 100,000 respectively, and showed a decrease of 2.7% at a Reynolds number of 150,000. A drag reduction of 15.8% and 5.5% was realized for Reynolds numbers of 50,000 and 100,000 respectively, and an increase in drag of 2.0% at Reynolds number of 150,000 when not taking into account external forcing power in the BSSB orientation. The forcing assembly was shown to require nearly 12 times more power to operate than it saves in drag reduction at Reynolds number of 50,000.

A thermal anemometry measurement of streamwise velocity of the near wake behind the bluff body was conducted to qualitatively assess the attenuation of the vortex street behind the model. BSSB forcing shows that as the freestream velocity is increased as compared to the forcing velocity, the change in energy spectral density is lessened, and as such, the largest attenuation in vortex shedding is at Reynolds number of 50,000 while nearly no change is seen at the Reynolds number of 150,000. The ratio of forcing velocity to streamwise velocity was quantized using the forcing momentum coefficients, 8.94e-4, 2.24e-4, and 9.94e-5 for Reynolds numbers of 50,000, 100,000, and 150,000, respectively.

In summary, the distributed forcing mechanism was shown to effectively increase base pressure and attenuate the strength of the vortex immediately after the body. However, a much more efficient method to move air around the aft section of the body needs to be found in order for this to be realized in an increasingly practical setting.
6. RECOMMENDED FUTURE WORK

Even though much was learned from this experiment, much still needs to be explained on the topic of distributed forcing. This section intends to outline some aspects to continue the work started here, beginning with the references cited throughout this paper, in particular, the work completed by Kim et al [1] and Park et al [14]. The literature published by both parties outlines much of the same experiments in the same wind tunnel and using the same CFD modeling techniques. Kim et al [1] studies the effects of distributed forcing and Park et al [14] studies the effects of end-plate tabulations on two-dimensional bluff bodies with blunt bases. The study on passive drag control with the tabulations, seemingly performed after the active drag studies, even though it was published prior, stated the tabulations were in response to finding a better, easier method than active drag control with fans. This leads to the argument that the authors of both papers knew distributed forcing couldn’t be realized in a practical setting, even though it was not published in that way, and turned their attention to tabulations to effectively perform vortex attenuation and drag reduction. With the knowledge of the results of this study and the literature previously published, this author is in agreement that there should be more work done in passively controlling the flow with tabulations, as this seems to provide a more efficient method of drag reduction. Perhaps a model built with moveable tabulations would provide the best of both active and passive drag control. The tabulations could then be translated across the base plate of the model in response to the boundary layer thickness over the aft section.

In the study of active drag control, the main limitation was the forcing velocity. The forcing slot width has a large impact on this, and an effective way to control the slot width, and therefore slot velocity could be developed to study the effect the slot width has on drag reduction. In addition, as the aspect ratio decreases, the more realistic the model becomes to a practical application. Therefore lowering the aspect ratio and perhaps performing some ground effect testing could be done to evaluate distributed forcing or end-plate tabulations.
In this study and the studies performed previously, the vortex street is effectively delayed in some instances and strength of the street attenuated in others. It could be beneficial to future research to determine how the distributed forcing technique influences vortex shedding on a 3D model. There could also be work completed in flow visualization to determine this. Not mentioned in this paper is the quick study in flow separation using yarn tufts attached to the model. This was performed to make sure the flow remained attached to the body up to the trailing edge. This is important as the separation point on the body at the trailing edge is where the boundary layer rolls up on itself and the vortex is formed. In consequence, the yarn tufts were attached to make sure the flow over the body was acting as expected.

Another point to be made is the applicability of distributed forcing to higher Reynolds number cases. In North America, semi-trailer trucks are limited to a width of approximately 2.6 meters, and using this as the length to nondimensionalize around, Reynolds numbers of highway speeds approach 5e6. This is more than an order of magnitude higher than the Reynolds numbers tested in this particular study, and extrapolating the same trends as found would require a very strong forcing assembly. This is again another reason why the passive flow control device of end-plate tabulations should be studied further rather than the distributed forcing assembly. Perhaps future research will bring a currently unknown application of distributed forcing into reality.

These suggestions for future work on distributed forcing and end-plate tabulations are meant as a continuation for future research and student study in the increasing knowledge about flow control.
LIST OF REFERENCES


APPENDICES

A. ERROR ANALYSIS

The following calculations are derived from those presented in Taylor [15] as a propagation of uncertainties for independent, random errors.

\[
C_p = \frac{P - P_\infty}{q_\infty} = \frac{P - P_\infty}{P_T - P_\infty}
\]

\[
\frac{dC_p}{C_p} = \frac{d(P - P_\infty)}{P - P_\infty} - \frac{d(P_T - P_\infty)}{P_T - P_\infty} = \frac{dP}{P - P_\infty} - \frac{dP_\infty}{P - P_\infty} - \frac{dP_T}{P_T - P_\infty} + \frac{dP_\infty}{P_T - P_\infty}
\]

\[
\frac{dC_p}{C_p} \approx \frac{dP}{P - P_\infty} + \left[\left(\frac{-1}{P - P_\infty} + \frac{1}{P_T - P_\infty}\right)\sigma_{P_\infty}\right] + \frac{dP_T}{P_T - P_\infty}
\]

So,

\[
\sigma_{C_p} \approx |C_p| \sqrt{\left(\frac{\sigma_P}{P - P_\infty}\right)^2 + \left[\left(\frac{-1}{P - P_\infty} + \frac{1}{P_T - P_\infty}\right)\sigma_{P_\infty}\right]^2 + \left(\frac{\sigma_{P_T}}{P_T - P_\infty}\right)^2}
\]

For example,

\[
C_p = \frac{P - P_\infty}{P_T - P_\infty}
\]

\[
C_p = \frac{-0.1041923 - -0.08215374}{-0.00580908 - -0.08215374} = \frac{-0.0220386}{0.0763446} = -0.288672
\]

\[
\sigma_{C_p} \approx \left| -0.288672 \right| \sqrt{\left(\frac{6.965814 \times 10^{-4}}{-0.0220386}\right)^2 + \ldots}
\]

\[
\left[\left(\frac{-1}{-0.0220386} + \frac{1}{0.0763446}\right)2.342732 \times 10^{-4}\right]^2 + \ldots
\]

\[
\left(\frac{2.107345 \times 10^{-4}}{0.0763446}\right)^2
\]

\[
\sigma_{C_p} \approx 0.009976
\]
Similarly,

\[ C_D = \frac{D}{q_{\infty}A} = \frac{D}{(P_T - P_{\infty})A} \]

\[
\frac{dC_D}{C_D} = \frac{dD}{D} - \frac{d(P_T - P_{\infty})}{P_T - P_{\infty}} - \frac{dA}{A} = \frac{dD}{D} - \frac{dP_T}{P_T - P_{\infty}} + \frac{dP_{\infty}}{P_T - P_{\infty}} - \frac{dA}{A}
\]

Neglect A, so

\[
\frac{dC_D}{C_D} \approx \frac{dD}{D} - \frac{dP_T}{P_T - P_{\infty}} + \frac{dP_{\infty}}{P_T - P_{\infty}}
\]

So,

\[
\sigma_{C_D} \approx |C_D| \sqrt{\left(\frac{\sigma_D}{D}\right)^2 + \left(\frac{\sigma_{P_T}}{P_T - P_{\infty}}\right)^2 + \left(\frac{\sigma_{P_{\infty}}}{P_T - P_{\infty}}\right)^2}
\]

For example,

\[ C_D = \frac{D}{(P_T - P_{\infty})A} \]

\[ C_D = \frac{0.699715}{(-0.00580908 - -0.08215374) \times 28.5} = 0.307798 \]

\[
\sigma_{C_D} \approx |0.307798| \sqrt{\left(\frac{0.0168317}{0.307798}\right)^2 + \cdots + \left(\frac{2.107345 \times 10^{-4}}{-0.00580908 - -0.08215374}\right)^2 + \cdots + \left(\frac{2.342732 \times 10^{-4}}{-0.00580908 - -0.08215374}\right)^2}
\]

\[ \sigma_{C_D} \approx 0.016879 \]
B. STING BALANCE CALIBRATION RESULTS

Figure B-1. Sting balance calibration graph with linear regression for axial force.

<table>
<thead>
<tr>
<th>Axial Force, lbs</th>
<th>Voltage</th>
<th>Normalized Voltage</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>4.24196</td>
<td>0</td>
</tr>
<tr>
<td>1</td>
<td>4.19615</td>
<td>0.04581</td>
</tr>
<tr>
<td>2</td>
<td>4.1501</td>
<td>0.09186</td>
</tr>
<tr>
<td>3</td>
<td>4.10384</td>
<td>0.13812</td>
</tr>
<tr>
<td>4</td>
<td>4.05744</td>
<td>0.18452</td>
</tr>
<tr>
<td>5</td>
<td>4.01302</td>
<td>0.22894</td>
</tr>
<tr>
<td>6</td>
<td>3.96612</td>
<td>0.27584</td>
</tr>
<tr>
<td>7</td>
<td>3.92087</td>
<td>0.32109</td>
</tr>
</tbody>
</table>

Table B-1. Data for voltage-force calibration relationship for axial force
Figure B-2. Sting balance calibration graph with linear regression for normal force.

Table B-2. Data for voltage-force calibration relationship for normal force

<table>
<thead>
<tr>
<th>Normal Force, lbs</th>
<th>Voltage</th>
<th>Normalized Voltage</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-0.1218</td>
<td>0</td>
</tr>
<tr>
<td>1</td>
<td>-0.0688</td>
<td>0.053</td>
</tr>
<tr>
<td>2</td>
<td>-0.0156</td>
<td>0.1062</td>
</tr>
<tr>
<td>3</td>
<td>0.0374</td>
<td>0.1592</td>
</tr>
<tr>
<td>4</td>
<td>0.0898</td>
<td>0.2116</td>
</tr>
<tr>
<td>5</td>
<td>0.1427</td>
<td>0.2645</td>
</tr>
<tr>
<td>6</td>
<td>0.1961</td>
<td>0.3179</td>
</tr>
</tbody>
</table>

$y = 18.892x - 0.0022$

$R^2 = 1$
C. DISSOLVING A THREADING TAP IN NONFERROUS METAL

Unfortunately, hand tapping threads in metals frequently results in tap breakage, especially from smaller sized taps. When the tap cannot be welded onto a bolt, or unscrewed using pliers, the piece usually must be scraped and remanufactured. During the course of manufacturing the model used in this experiment, a tap was broken off in a piece that took approximately 6 hours of machining to make. Instead of starting over with a new piece of aluminum, some research was done on dissolving broken taps. The following method presented of dissolving a ferrous tap in a non-ferrous material is the compilation of research through online forums and through The Watchmaker’s Hand-book [13]. Published in London in 1881, this reference has a brief statement about using the chemical Alum in order to dissolve tiny, broken ferrous taps common in watchmaking applications. Fortunately, Alum can be purchased at most supermarket stores in 1.9 oz containers in the spice section. It is commonly used for pickling foods.

Alum is the shorthand notation for potassium aluminum sulfate whose chemical formula is KAl(SO4)2·12H2O. This forms a very weak sulfuric acid when dissolved in water (H2SO4) and is the basis of why this works to dissolve a tap. This acidic solution is highly reactive to iron as shown in the chemical equation: Fe (s) + H2SO4 (aq) → H2 (g) + FeSO4 (aq). Reference [13] states that in 100 parts boiling water, 75 parts of Alum will dissolve. In performing this experiment, it was learned the closer to this ratio, the faster the reaction took place and the quicker the tap dissolved. The acidic solution must be kept at a constant boil or just under boiling in order to prevent crystallization of the Alum, and to provide energy for the reaction. If the solution solidifies, raising the temperature will dissolve the Alum back into the water.

It is important that the sulfuric acid is in a container which is nonreactive. A simple borosilicate glass measuring cup was used for this and placed in a bath of boiling water. As the tap is placed in the sulfuric acid, it will start giving off hydrogen gas, indicating the reaction is occurring. After approximately 3 hours, the tap was dissolved enough to fall out of the hole and
tapping could continue where it was left before the tap broke. A photograph of the setup which was successful in dissolving about 3/4 inch of a 6-32 tap is shown below.

Figure C1. Photograph of tap dissolving setup with saturated water-alum solution.
D. FAN SPECIFICATION SHEETS

The two fans used in this experiment are manufactured by Sunon, and the characteristics, specifications, performance curves, material and dimensions of each of the fans are shown for reference. The 10 mm thick, model KDE fan is presented first, followed by the 28 mm thick, model PMD fan.
# DC BRUSHLESS FAN

**MODEL:** KDE1204PFVX  
**P/N:** 11.MS.A.GN  

## CHARACTERISTICS

<table>
<thead>
<tr>
<th>1. Motor Design</th>
<th>Patented single-coil DC brushless 4 pole motor design.</th>
</tr>
</thead>
<tbody>
<tr>
<td>2. Insulation Resistance</td>
<td>More than 500M ohm between internal stator and lead wire(+) measured at DC 500V.</td>
</tr>
<tr>
<td>3. Dielectric Strength</td>
<td>Applied AC 500V for one minute or AC 600V for 2 seconds between housing and lead wire(+)</td>
</tr>
<tr>
<td>4. Noise Level</td>
<td>Measured in a semi-anechoic chamber with background noise level below 15 dB(A). The fan is running in free air with the microphone at a distance of one meter from the fan intake.</td>
</tr>
<tr>
<td>5. Input Power, Current &amp; Speed</td>
<td>Measured after continuous 10 minute operation at rated voltage in clean air, and at ambient temperature of 25 degrees C.</td>
</tr>
<tr>
<td>6. Tolerance</td>
<td>±15% on rated power and current.</td>
</tr>
<tr>
<td>7. Air Performance</td>
<td>Measured by a double chamber. The values are recorded when the fan speed has stabilized at rated voltage.</td>
</tr>
</tbody>
</table>
### SPECIFICATIONS

**MODEL:** KDE1204PFVX  
**P/N:** 11.MS.A.GN  

<table>
<thead>
<tr>
<th>Item</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-1. Rated Voltage</td>
<td>12 VDC</td>
</tr>
<tr>
<td>1-2. Operating Voltage Range</td>
<td>4.5~13.8 VDC</td>
</tr>
<tr>
<td>1-3. Starting Voltage</td>
<td>4.5 VDC (25 deg. C POWER ON/OFF)</td>
</tr>
<tr>
<td>1-4. Rated Speed</td>
<td>8500 RPM ± 20%</td>
</tr>
<tr>
<td>1-5. Air Delivery</td>
<td>9.5 CFM / MAX 11.0 CFM</td>
</tr>
<tr>
<td>1-6. Static Pressure</td>
<td>0.25 Inch-H2O / MAX 0.29 Inch-H2O</td>
</tr>
<tr>
<td>1-7. Rated Current</td>
<td>0.15 AMP</td>
</tr>
<tr>
<td>1-8. Rated Power</td>
<td>1.8 WATTS</td>
</tr>
<tr>
<td>1-9. Noise Level</td>
<td>39 dB(A) / MAX 43 dB(A)</td>
</tr>
<tr>
<td>1-10. Direction of Rotation</td>
<td>Counter-clockwise viewed from front of fan blade</td>
</tr>
<tr>
<td>1-11. Operating Temperature</td>
<td>-10 to +70 deg. C</td>
</tr>
<tr>
<td>1-12. Storage Temperature</td>
<td>-40 to +70 deg. C</td>
</tr>
<tr>
<td>1-13. Bearing System</td>
<td>Vapo bearing system</td>
</tr>
<tr>
<td>1-14. Weight</td>
<td>17 g</td>
</tr>
<tr>
<td>1-15. Safety</td>
<td>UL/CUR/TUV/CE Approvals</td>
</tr>
<tr>
<td>1-16. Vibration</td>
<td>Vibration of acceleration 1.5G and frequency 5<del>50</del>5Hz is applied in all 3 directions(X,Y,Z), in cycles of 1 minute each, for a total vibration time of 30 minutes.</td>
</tr>
</tbody>
</table>
| 1-17. Locked Rotor Protection | Automatic Restart Capability  
Note: In a situation where the fan is locked by an external force while the electricity is on, an increase in coil temperature will be prevented by temporarily turning off the electrical power to the motor. The fan will automatically restart when the locked rotor condition is released.
MODEL: KDE1204PFVX
P/N: 11.MS.A.GN

PERFORMANCE CURVES

STATIC PRESSURE

<table>
<thead>
<tr>
<th>mm-H₂O</th>
<th>Inch-H₂O</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>4</td>
<td>0.141</td>
</tr>
<tr>
<td>8</td>
<td>0.282</td>
</tr>
<tr>
<td>12</td>
<td>0.40</td>
</tr>
</tbody>
</table>

CFM

m³/min

12 VDC

MAX.
MATERIAL

2-1. Frame : Thermoplastic PBT of UL 94V-0
2-2. Impeller : Thermoplastic PBT of UL 94V-0
2-3. Bobbin : Thermoplastic PBT of UL 94V-0
2-4. Lead Wire : UL2468,26 awg,+RED,-BLACK

DIMENSIONS


UNITS: mm
DC BRUSHLESS FAN
MODEL : PMD1204PQB2-A
P/N : (2).GN

CHARACTERISTICS

1. Motor Design : DC brushless 4 pole motor design.

2. Insulation Resistance : More than 10M ohm between internal stator and lead wire(+) measured at DC 500V.

3. Dielectric Strength : Applied AC 500V for one minute or AC 600V for 2 seconds between housing and lead wire(+)

4. Noise Level : Measured in a semi-anechoic chamber with background noise level below 15 dB(A). The fan is running in free air with the microphone at a distance of one meter from the fan intake.

5. Input Power, Current & Speed : Measured after continuous 10 minute operation at rated voltage in clean air, and at ambient temperature of 25 degrees C.

6. Tolerance : ±15% on rated power and current.

7. Air Performance : Measured by a double chamber. The values are recorded when the fan speed has stabilized at rated voltage.
## SPECIFICATIONS

**MODEL:** PMD1204PQB2-A  
**P/N:** (2).GN

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1-1.</td>
<td>Rated Voltage</td>
</tr>
<tr>
<td>1-2.</td>
<td>Operating Voltage Range</td>
</tr>
<tr>
<td>1-3.</td>
<td>Starting Voltage</td>
</tr>
<tr>
<td>1-4.</td>
<td>Rated Speed</td>
</tr>
<tr>
<td>1-5.</td>
<td>Air Delivery</td>
</tr>
<tr>
<td>1-6.</td>
<td>Static Pressure</td>
</tr>
<tr>
<td>1-7.</td>
<td>Rated Current</td>
</tr>
<tr>
<td>1-8.</td>
<td>Rated Power</td>
</tr>
<tr>
<td>1-9.</td>
<td>Noise Level</td>
</tr>
<tr>
<td>1-10.</td>
<td>Direction of Rotation</td>
</tr>
<tr>
<td>1-11.</td>
<td>Operating Temperature</td>
</tr>
<tr>
<td>1-12.</td>
<td>Storage Temperature</td>
</tr>
<tr>
<td>1-13.</td>
<td>Bearing System</td>
</tr>
<tr>
<td>1-14.</td>
<td>Weight</td>
</tr>
<tr>
<td>1-15.</td>
<td>Safety</td>
</tr>
<tr>
<td>1-16.</td>
<td>Vibration</td>
</tr>
</tbody>
</table>
| 1-17. | Locked Rotor Protection | Automatic Restart Capability  
Note: In a situation where the fan is locked by an external force while the electricity is on, an increase in coil temperature will be prevented by temporarily turning off the electrical power to the motor. The fan will automatically restart when the locked rotor condition is released. |
MODEL: PMD1204PQB2-A
P/N: (2).GN

PERFORMANCE CURVES

STATIC PRESSURE

mm-H₂O  Inch-H₂O

0  0.424  0.848
0.25  15  30
12 VDC

CFM

m³/min
MATERIAL

2-1. Frame : Thermoplastic PBT of UL 94V-0
2-2. Impeller : Thermoplastic PBT of UL 94V-0
2-3. Bobbin : Thermoplastic PBT of UL 94V-0
2-4. Lead Wire : UL2468.26 awg,+RED,-BLACK

DIMENSIONS


UNITs: mm

SPEC.NO: D04038480G-00
ISSUE DATE: 02.15.2005
REVISION DATE: 07.15.2005
E. ENGINEERING DRAWINGS OF EXPERIMENTAL MODEL

The drawings presented in this part of the appendix are meant as a guide to recreate the model built for this experiment. The author is not responsible for missing or implied dimensions, nor recommends sending these drawings as they are presented here, to be manufactured professionally. Tolerances and threaded holes are omitted. All material is 6061 aluminum and dimensions shown are in inches.