Design, Analysis, and Initial Testing of a Fiber-Optic Shear Gage for 3D, High-Temperature Flows

by

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Abstract

This investigation concerns the design, analysis, and initial testing of a new, two-component wall shear gage for 3D, high-temperature flows. This gage is a direct-measuring, non-nulling design with a round head surrounded by a small gap. Two flexure wheels are used to allow small motions of the floating head. Fiber-optic displacement sensors measure how far the polished faces of counterweights on the wheels move in relation to a fixed housing as the primary measurement system. No viscous damping was required. The gage has both fiber-optic instrumentation and strain gages mounted on the flexures for validation of the newer fiber optics. The sensor is constructed of Haynes® 230®, a high-temperature nickel alloy. The gage housing is made of 316 stainless steel. All components of the gage in pure fiber-optic form can survive to a temperature of 1073 K. The bonding methods of the backup strain gages limit their maximum temperature to 473 K. The dynamic range of the gage is from 0-500 Pa (0-10g) and higher shears can be measured by changing the floating head size.

Extensive use of finite element modeling was critical to the design and analysis of the gage. Static structural, modal, and thermal analyses were performed on the flexures using the ANSYS finite element package. Static finite element analysis predicted the response of the flexures to a given load, and static calibrations using a direct force method confirmed these results. Finite element modal analysis results were within 16.4% for the first mode and within 30% for the second mode when compared with the experimentally determined modes. Vibration characteristics of the gage were determined from experimental free vibration data after the gage was subjected to an impulse. Uncertainties in the finished geometry make this level of error acceptable. A transient thermal analysis examined the effects of a very high heat flux on the exposed head of the gage. The 100,000 W/m² heat flux used in this analysis is typical of a value in a scramjet.
engine. The gage can survive for 10 minutes and operate for 3 minutes before a 10% loss in flexure stiffness occurs under these conditions.

Repeated cold-flow wind tunnel tests at Mach 2.4 with a stagnation pressure from 3.7-8.2 atm (55-120 psia) and ambient stagnation temperature \((Re=6.6\times10^7/m)\) and Mach 4.0 with a stagnation pressure from 10.2-12.2 atm (150-180 psia) and ambient stagnation temperature \((Re=7.4\times10^7/m)\) were performed in the Virginia Tech Supersonic Wind Tunnel. Some of these tests had the gage intentionally misaligned by 25° to create a virtual 3D flow in this nominally 2D facility. Experimental results gave excellent agreement with semi-empirical prediction methods for both the aligned and 25° experiments. This fiber-optic skin friction gage operated successfully without viscous damping. These tests in the supersonic wind tunnel validated this wall shear gage design concept.
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<tr>
<td>$b$</td>
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<td>Skin Friction Coefficient</td>
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<td>Specific Heat at Constant Pressure</td>
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<td>Boundary Layer Thickness</td>
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<tr>
<td>$\mu$</td>
<td>Laminar Viscosity</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Standard Deviation</td>
</tr>
</tbody>
</table>
\( \tau_w \) Shear Force
\( \nu \) Laminar Kinematic Viscosity
\( \omega \) Frequency
\( \zeta \) Damping Ratio

Subscripts

Compr. Compressible
d Damped
e Boundary Layer Edge
FO Fiber-Optic
i Initial
Incompr. Incompressible
head Head
n Undamped, Normal
o Stagnation Condition
SG Strain Gage
w Wall
X In X Direction
Y In Y Direction
Z In Z Direction
1 Introduction

1.1 Rationale for Measuring Wall Shear

The drag of any object moving through a fluid can be broken up into two primary components: the pressure drag and the viscous drag. The pressure drag is the streamwise component of pressure force integrated over the entire body. The wall shear drag is the streamwise component of shear force integrated over the whole vehicle. Measuring both quantities is important for vehicle as well as engine performance prediction and analysis. This study focuses on the measurement of the wall shear drag in high-temperature flows.

The need for hot flow measurements is illustrated in Figure 1. Figure 1 shows the range of very high-temperature applications and the need to create wall shear gages that can operate in these environments. For example, the efficiency of a turbine engine is highly dependent on the internal loses. To improve the efficiency of the engines the wall shear must be known so that the desired characteristics can be designed into the engine components. Aircraft, spacecraft, and munitions and anything that travels through a fluid or gas have some portion of its drag due to viscosity. All could benefit from wind tunnel or flight measurements of the local viscosity to refine or change the designs. Anything that moves through a fluid or that has fluid flowing through it has the potential to benefit from accurate measurements of the wall shear. Efficiencies could be improved and costs reduced, allowing reduced operating costs.

Figure 1: High-Temperature Shear Applications and Measurement Limitations
Frictional loses are important for all flows. At high-temperatures fiber optics are one of the only measurement techniques which can survive. Conventional strain gages will fail at about ~620 K and piezoelectric techniques will fail at only ~820 K. Figure 1 shows the need for measurements at higher temperatures as all of the listed devices operate at temperatures above 1573 K. These high-temperatures test the limits of modern materials. Creating a sensor to survive in these environments, much less make accurate measurements, is an extremely challenging proposition.

1.2 Wall Shear Background

The wall shear is the actual quantity to be measured by a skin friction gage and is shown in Equation 1 below. The relation is for a Newtonian fluid where the shear stress varies linearly with the rate of strain. A non-Newtonian fluid will have a nonlinear variation of this profile. The skin friction coefficient \( C_f \) is shown in Equation 2. This is simply a convenient dimensionless form of the wall shear and is the wall shear divided by the dynamic pressure at the edge of the boundary layer of the flow in question. This may not be possible to state for all flows as measuring the local dynamic pressure may be extremely difficult or impossible. If a simple Pitot probe is directly exposed to a 1500 K flow it will likely melt within a very short period of time.

\[
\tau_w = \mu \frac{\partial V}{\partial n}
\]

**Equation 1**

\[
C_f = \frac{\tau_w}{\frac{1}{2} \rho_e U_e^2}
\]

**Equation 2**

Another important quantity is the friction velocity and the relation for this is displayed in Equation 3. The friction velocity is a way of normalizing parameters to show the influence of the wall shear. For turbulent flows, the friction velocity leads to results such as the law of the wall which is shown in Equation 4. The friction velocity is
also used in the development of models. When examining turbulent boundary layers, a 
Reynolds number of the form shown in Equation 5 can be used. Equation 6 shows the 
flow velocity scaled by the friction velocity. Figure 2 shows the defect law plot for 
turbulent velocity profiles over flat plates. This plot also demonstrates the importance of 
$u_*$. Correlations such as the law of the wall and the defect law are central to the 
development of turbulence models for Computational Fluid Dynamics (CFD).

\[ u_* = \sqrt{\frac{\tau_w}{\rho_c}} = U_e \sqrt{\frac{C_f}{2}} \]

**Equation 3**

\[ \frac{U}{u_*} = g\left(\frac{y u_*}{V}\right) \]

**Equation 4**

\[ y^+ \equiv \frac{y u_*}{V} \]

**Equation 5**

\[ u^+ = \frac{U}{u_*} \]

**Equation 6**
1.3 Wall Shear Measurement Methods

There are two broad categories of measurement types that are used to measure the wall shear. The first type directly senses the wall shear using some sort of sensor. The second broad category measures another flow quantity and uses assumed relationships to infer the value of the shear at the wall. These methods are known as direct and indirect measurement methods respectively. Each has its own advantages and disadvantages over the other depending on the situation.2,3

1.3.1 Direct Wall Shear Measurement

The way a direct wall shear gage generally operates is that some portion of the gage is exposed to the flow. The portion exposed to the flow is called the floating head and it is allowed to move slightly in reaction to the shear force. A schematic of this gage type is shown in Figure 3 below. The floating head is affixed to a flexure which deforms under the load of the shear force. A small gap surrounding the floating head allows movement of the floating head. The housing holds all of the gage components together and keeps the head of the gage flush with the wall. Some type of measurement system

Figure 2: Defect Law Plot of Turbulent Velocity Profiles 1
then records a strain or displacement and the wall shear is known from a calibration. One significant advantage of direct wall shear measurement is the ability to calibrate a gage using a direct force method. Historically a number of different systems have used this general method with varying degrees of success. 

**Figure 3: Direct Wall Shear Gage General Features**

Within the broad category of direct wall shear measurement there are two primary sub-categories. The first is the nulling skin friction gage. The nulling gage uses some type of mechanical device to move the head of the gage back to a known center position. The restoring force, which is required to return the head to the neutral position, will be equal and opposite to the wall shear force. The advantage to this gage type is that the flow-field is largely undisturbed. Thus, misalignment errors between the sensing head and the housing are eliminated or reduced. In practice, this type of gage is very complex and slow to respond. An example of a nulling skin friction gage is shown in Figure 4 below. The large number of internal components is evident when compared with the non-nulling gage schematic displayed by Figure 5. A non-nulling gage simply moves to a displaced position when exposed to a shear force and returns to the original position when the force is removed. The small displacements required by displacement measurement systems or sensitive strain gages keep the misalignment between the floating head and the housing to a minimum.
1.3.2 Indirect Wall Shear Measurement

Indirect wall shear measurements are categorized in Figure 6 below. All of these methods either rely on a correlation or another underlying principle to deduce the wall shear from another measured quantity. Several of the techniques rely on hot-wires or small tubes to measure the flow velocity or dynamic pressure. These techniques change the local flow field and can only be used in relatively benign environments. In a high-speed flow, the high dynamic pressure will challenge the strength of materials. In a very high-speed or combusting flow, obtaining a material to survive those hot conditions directly can be impossible or at best, extremely difficult. Many of the assumptions about the fluids begin to break down in these extreme conditions. For 3D, transitional, unsteady flows or flows with mass transfer, combustion, shocks or separated regions, the underlying correlations required for indirect methods are not available. This makes indirect methods impractical in many flows and impossible, with current materials, in some flows.
One of the more robust physical arrangements is the Stanton tube technique shown in Figure 7. Stanton tube measurement involves using a razor blade and a static pressure port. The idea of this technique is to measure the total pressure at a known location in the laminar sub-layer of the flow. However, the leading edge of the razor in a very hot and high-speed flow would be a very vulnerable location. In a scramjet combustor the temperatures can reach 3273 K. Creating materials that can survive at those temperatures is a daunting challenge. The best high-temperature metals may survive at temperatures close to 2273 K but they would fall far short of the needed temperature range.
Figure 7: Stanton tube technique\textsuperscript{53}
1.4 History of Direct Wall Shear Gage Concepts and Designs

Wall shear measurement has a long history. In 1872, Froude developed one of the first wall shear measurement techniques which involved pulling planks of various dimensions with the apparatus shown in Figure 8. He used the device to tow the planks and then ship models down a towing tank he had constructed for the purpose. This is one of the first recorded investigations to systematically measure wall shear.

Figure 8: Apparatus used by Froude to measure wall shear on planks

Table 1 shows a summary of direct skin friction measurement devices and a summary of their important characteristics. The size of the floating head, test conditions, style of measurement, year, author, nulling or non-nulling type, and if the gap between the floating head was filled with any type of media isolation compound or damping fluid are recorded below. An interesting design was made by Bruno et. al. in 1969 using a single wheel-flexure in a nulling gage design. It should be noted that gages for 3D flow do not show up until 1990.
<table>
<thead>
<tr>
<th>YEAR</th>
<th>AUTHOR</th>
<th>FLOW</th>
<th>2D/3D</th>
<th>NULLING</th>
<th>HEAD (MM)</th>
<th>MEASUREMENT METHOD</th>
<th>FLUID</th>
<th>NOTES</th>
</tr>
</thead>
<tbody>
<tr>
<td>1929</td>
<td>Kempf11</td>
<td>Bottom of 77m pontoon</td>
<td>2D</td>
<td>Yes</td>
<td>309 x 1010</td>
<td>Pulleys and Springs</td>
<td>None</td>
<td>Measurements made in water</td>
</tr>
<tr>
<td>1932</td>
<td>Schoenherr12</td>
<td>“Catamaran Friction Plates” 0.3-3.0 ft/s Re&lt;2x10^6</td>
<td>2D</td>
<td>Yes</td>
<td>914 x 457 to 1829 x 914</td>
<td>Hanging Weights</td>
<td>None</td>
<td>Towing tank with water or glycerin</td>
</tr>
<tr>
<td>1940</td>
<td>Schultz-Grunow13</td>
<td>U=20 m/s 1.6 x 10^6 Re&lt; 16 x 10^6</td>
<td>2D</td>
<td>Yes</td>
<td>300 x 500 (Estimated)</td>
<td>Manual Torsion Bar</td>
<td>None</td>
<td>Very Large</td>
</tr>
<tr>
<td>1952</td>
<td>Weiler and Hertwig14</td>
<td>Supersonic Wind Tunnel</td>
<td>2D</td>
<td>Yes</td>
<td>25 Diameter (D)</td>
<td>Flexure with Linear Variable Differential Transformer (LVDT)</td>
<td>None</td>
<td>DRI University of Texas</td>
</tr>
<tr>
<td>1953</td>
<td>Chapman and Kester15</td>
<td>Subsonic and supersonic axial flow over cylinders</td>
<td>2D</td>
<td>Yes</td>
<td></td>
<td>Cylinder Length to Diameters of 8,13, and 23</td>
<td>Transducer</td>
<td>None</td>
</tr>
<tr>
<td>1953</td>
<td>Coles16</td>
<td>M=1.97 4x10^5 &lt; Re &lt; 1x10^7 M=2.57 4x10^5 &lt; Re &lt; 9x10^6</td>
<td>2D</td>
<td>Yes</td>
<td>6 x 38</td>
<td>Parallel linkage flexure with Schaevitz LVDT</td>
<td>Oil</td>
<td>Figures 15-17 of (Coles 1952)</td>
</tr>
<tr>
<td>1953</td>
<td>Dhawan17</td>
<td>Low speed 6 x 10^6 &lt; Re &lt; 6 x 10^7 Subsonic 0.2 &lt; M &lt; 0.8 3 x 10^5 Re &lt; 1.2 x 10^6</td>
<td>2D</td>
<td>Yes</td>
<td>12 x 63 2 x 20</td>
<td>Parallel LVDT</td>
<td>None</td>
<td>GALCIT “Correlation” Tunnel</td>
</tr>
<tr>
<td>1953</td>
<td>Eimer18</td>
<td>Hypersonic M=5.8 laminar flat plate with condensation</td>
<td>2D</td>
<td>Yes</td>
<td>51 x 25</td>
<td>Schaevitz LVDT Chemical balance and flexures</td>
<td>None</td>
<td>GALCIT-Hypersonic 5”x 5” Tunnel</td>
</tr>
<tr>
<td>1953</td>
<td>Hakkinen19</td>
<td>Turbulent boundary layer of flat plate at high subsonic to Mach 1.75 3.3 x 10^5 &lt; Re &lt; 1.2 x 10^6</td>
<td>2D</td>
<td>Yes</td>
<td>2 x 20</td>
<td>Schaevitz LVDT</td>
<td>None</td>
<td>GALCIT-Similar to Dhawan’s Gage</td>
</tr>
<tr>
<td>1954</td>
<td>Blumer20</td>
<td>Axially symmetric conical turbulent boundary layer M=2.6 1 x 10^6 &lt; Re &lt; 8 x 10^6</td>
<td>2D</td>
<td>Yes</td>
<td>254 Long 15° Cone</td>
<td>Schaevitz LVDT</td>
<td>None</td>
<td>University of Minnesota, Rosemount Aeronautical Laboratory</td>
</tr>
<tr>
<td>1956</td>
<td>Wolfe21</td>
<td>64 to 286 ft/s</td>
<td>2D</td>
<td>Yes</td>
<td>1006 x 749</td>
<td>Baldwin cantilever spring strain gages</td>
<td>None</td>
<td>University of Minnesota</td>
</tr>
<tr>
<td>1957</td>
<td>Lyons and Fenter22</td>
<td>Supersonic Flight</td>
<td>2D</td>
<td>Yes</td>
<td>50 (D)</td>
<td>Double Parallel inter-connected linkage to eliminate sensitivity to linear and rotational accelerations</td>
<td>None</td>
<td>Tested on Rockets</td>
</tr>
<tr>
<td>Year</td>
<td>Author</td>
<td>Flow</td>
<td>2D/3D</td>
<td>Nulling</td>
<td>Head (mm)</td>
<td>Measurement Method</td>
<td>Fluid</td>
<td>Notes</td>
</tr>
<tr>
<td>-------</td>
<td>-------------------------------</td>
<td>----------------------------------------------------------------------</td>
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<td>---------</td>
<td>-----------</td>
<td>-------------------------------------------</td>
<td>-------------</td>
<td>----------------------------------------------------------------------</td>
</tr>
<tr>
<td>1958</td>
<td>Everett²¹</td>
<td>Incompressible liquid flow with variable channel height</td>
<td>2D</td>
<td>Yes</td>
<td>25.4 and 25.15 (D)</td>
<td>Beryllium-copper flexure Schaevitz LVDT</td>
<td>Silicon Fluid</td>
<td>Variable gap size and thickness</td>
</tr>
</tbody>
</table>
| 1958  | Smith and Walker²⁴            | 0.11 < M < 0.32  
10⁴ < Re < 40 x 10³                      | 2D    | Yes     | 50 (D)   | Parallel linkage LVDT  
Kelvin current balance                     | None       | Incompressible flow                                                  |
| 1963  | MacArthur²⁵                    | Shock Tunnel/Impulse Facilities                                   | 2D    | Yes     | 6.4 (D)   | Parallel linkage lead zirconium titanate piezoelectric ceramic | None       | Overcame difficulties in measuring small forces. It was complex with limited success and also know as Calspan gage |
| 1963  | Moulic²⁶                       | M=6 Low Density                                                    | 2D    | Yes     | 0.25 x 25 | Side flexure pivot LVDT                    | None       | Studied strong interaction region near leading edge of flat plate    |
| 1964  | O’Donnell²⁷                    | M=2.67  
Re=100070                                              | 2D    | Yes     | 25 (D)   | Parallel linkage                           | None       | Studied effects of misalignment of floating head                        |
| 1965  | Young and Westkaemper²⁸        | Supersonic flow with heat transfer and surface roughness            | 2D    | Yes     | 25 (D)   | Parallel linkage                           | None       |                                                                      |
| 1966  | Dershin et al.²⁹               | Supersonic flow with mass transfer                                 | 2D    | Yes     | “Pointed ellipse” | Parallel linkage LVDT                     | None       |                                                                      |
| 1966  | Moore and McVey²⁰             | High-Temperature hypersonic flows                                   | 2D    | Yes     | N/A      | Flexure pivot Pneumatic position sensor    | None       |                                                                      |
| 1967  | Brown and Joubert²¹           | Low-speed adverse pressure gradients                                | 2D    | Yes     | 19 (D)   | Parallel Linkage LVDT                      | None       | Studied gradient forces in gap                                         |
| 1969  | Fowke³²                        | Supersonic                                                          | 2D    | Yes     | 127 (D)  | Flexure pivot LVDT                         | None       |                                                                      |
| 1969  | Bruno, Yanta and Risher³³      | Supersonic flows with heat transfer                                 | 2D    | Yes     | 20.3 (D) | Flexure pivot LVDT                         | None       | Force variable by changing loading spring                             |
| 1970  | Winter and Gaudet²⁴           | 0.2 < M < 2.8  
1.6 x 10⁴ < Re < 2 x 10⁵                     | 2D    | Yes     | 368 (D)  | Parallel linkage Resistance strain gages    | None       | Surface roughness tests                                               |
| 1970  | Hastings and Sawyer³⁵         | M=4  
1 x 10⁴ < Re < 3 x 10⁴                         | 2D    | Yes     | 7.9 (D)  | Parallel Linkage LVDT                      | None       |                                                                      |
<p>| 1970  | Paros (Kistler)³⁶             | Used in a wide range of conditions including flight. Cooling system available | 2D    | Yes     | 9 (D)    | Pivoted about crossed-spring flexure       | None       | Schematic Figure 4 of Winter (1977)                                    |</p>
<table>
<thead>
<tr>
<th>Year</th>
<th>Author</th>
<th>Flow</th>
<th>2D/3D</th>
<th>Nulling</th>
<th>Head (mm)</th>
<th>Measurement Method</th>
<th>Fluid</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1971</td>
<td>Miller</td>
<td>Low speed flow with a favorable pressure gradient</td>
<td>2D</td>
<td>Yes</td>
<td>25 (D)</td>
<td>Parallel Linkage LVDT</td>
<td>None</td>
<td>Extension of Brown and Joubert (1961)</td>
</tr>
<tr>
<td>1973</td>
<td>Franklin</td>
<td>Subsonic wind tunnel and water</td>
<td>2D</td>
<td>Yes</td>
<td>16 (D)</td>
<td>Pivoted variable geometry electric valve</td>
<td>None</td>
<td></td>
</tr>
<tr>
<td>1973</td>
<td>Waltrup and Schetz</td>
<td>Complex supersonic flow with a strong adverse pressure gradient and with injection through porous and tangential slots and M=2.4</td>
<td>2D</td>
<td>Yes</td>
<td>25.4 (D)</td>
<td>LVDT</td>
<td>Silicon Oil</td>
<td>VT Supersonic Wind Tunnel (VT-SST)</td>
</tr>
<tr>
<td>1974</td>
<td>Morsy</td>
<td>Low speed flow past circular cylinder</td>
<td>2D</td>
<td>Yes</td>
<td>50.1 x 3.2</td>
<td>Jewelied pivots clock springs</td>
<td>None</td>
<td></td>
</tr>
<tr>
<td>1975</td>
<td>Schetz and Van Overeem</td>
<td>Complex supersonic flow with a strong adverse pressure gradient and with injection through porous and tangential slots and M=2.4</td>
<td>2D</td>
<td>Yes</td>
<td>25.4 (D)</td>
<td>LVDT</td>
<td>None</td>
<td>VT-SST</td>
</tr>
<tr>
<td>1977</td>
<td>Schetz and Nerney</td>
<td>Low speed turbulent flow on smooth, rough, and porous surfaces with and without injection</td>
<td>2D</td>
<td>No</td>
<td>25.4 (D)</td>
<td>Semi-conductor strain gages</td>
<td>None</td>
<td>VT-SST</td>
</tr>
<tr>
<td>1981-</td>
<td>Kong and Schetz</td>
<td>Hot high-speed flow</td>
<td>3D</td>
<td>Yes</td>
<td>12.7 (D)</td>
<td>Piezoelectric transducers</td>
<td>Silicon Oil</td>
<td>Water Cooling</td>
</tr>
<tr>
<td>1984</td>
<td>Kelly, Simmons, and Paull</td>
<td>Impulse test in hypersonic flow</td>
<td>2D</td>
<td>No</td>
<td>10 (D)</td>
<td>Piezoelectric transducers</td>
<td>Thermal cover</td>
<td>University of Queensland Australia</td>
</tr>
<tr>
<td>1992</td>
<td>Chadwick, DeTurris, and Schetz</td>
<td>Very High Heat Flux</td>
<td>3D</td>
<td>No</td>
<td>6.35 (D)</td>
<td>Piezoresistive Strain Gages/Transducer</td>
<td>Silicon Oil</td>
<td>Quartz Beam, Water Cooling</td>
</tr>
<tr>
<td>1994</td>
<td>Bowersox, Chadwick, Diewert, and Schetz</td>
<td>Impulse tests M=14 NASA Ames 16&quot; Shock Tunnel T=2.0 milliseconds Hypulse Facility T=0.3 milliseconds</td>
<td>2D</td>
<td>No</td>
<td>8.13 (D)</td>
<td>Semi-conductor strain gages</td>
<td>Silicon Oil</td>
<td>Low thermal conductivity plastic</td>
</tr>
<tr>
<td>Year</td>
<td>Author</td>
<td>Flow</td>
<td>2D/3D</td>
<td>Nulling</td>
<td>Head (mm)</td>
<td>Measurement Method</td>
<td>Fluid</td>
<td>Notes</td>
</tr>
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<tr>
<td>1995</td>
<td>Paik and Schetz[^49]</td>
<td>Supersonic flow with heat flux M=2.4 (T_e=300k) (Re_e=4.9 \times 10^7)</td>
<td>2D</td>
<td>No</td>
<td>12.7 (D)</td>
<td>Foil Strain Gages</td>
<td>None</td>
<td>Water cooled with heat flux micro-sensor</td>
</tr>
<tr>
<td>1996</td>
<td>Novean, Schetz, Hazelton, and Bowersox[^50]</td>
<td>Shock Tunnel M=12-14 (r=200-3000) Pascals Hypulse M=14 (r=800-3000) Pascals Supersonic flow M=2.4 (T_e=300k)</td>
<td>2D</td>
<td>No</td>
<td>4.55, 4.6, and 5.4 (D)</td>
<td>Semi-conductor strain gages</td>
<td>Silicon oil and rubber RTV615</td>
<td>Successful except when shocks impinge on floating head</td>
</tr>
<tr>
<td>1999</td>
<td>Remington and Schetz[^60]</td>
<td>Transonic M=0.7-0.99 (r=0.3-1.45) Pascals</td>
<td>2D</td>
<td>No</td>
<td>4.2, 6.35, and 9.53 (D)</td>
<td>Kistler-Morse Strain gages</td>
<td>None</td>
<td></td>
</tr>
<tr>
<td>1999</td>
<td>Pulliam and Schetz[^53],[^54]</td>
<td>Supersonic Flow</td>
<td>3D</td>
<td>No</td>
<td>1.63 (D)</td>
<td>Fiber-Optic Displacement Sensors</td>
<td>Oil</td>
<td>Sensitive to Vibration</td>
</tr>
<tr>
<td>2001</td>
<td>Smith and Schetz[^5],[^31]</td>
<td>Supersonic flow M=2.4 (T_e=300k) Scramjet Engine Model M=6.4 (T_e=1555 K)</td>
<td>2D</td>
<td>No</td>
<td>7.6 (D)</td>
<td>Flexure Ring with Metal Foil Strain Gages</td>
<td>None</td>
<td>No cooling and could operate for limited duration</td>
</tr>
<tr>
<td>2002</td>
<td>Sang and Schetz[^5,[^69]</td>
<td>Supersonic flow M=2.4 (T_e=300k) Transonic Flight Test (F-15)</td>
<td>3D</td>
<td>No</td>
<td>19 (D)</td>
<td>Aluminum Flexure Rod with Semiconductor Strain Gages</td>
<td>Thin Silicon RTV Sheets And/Or Glycerin Fill</td>
<td>Flight Test Gage Used Only RTV Sheet for Damping</td>
</tr>
<tr>
<td>2002</td>
<td>Goldfeld, Nestoulia, and Falempin[^52]</td>
<td>Supersonic Flow Mach 2, 4, 6 (Re_e=2.5-10x10^6)</td>
<td>3D</td>
<td>No</td>
<td>6-10 (D)</td>
<td>Semiconductor resistive-strain Sensors</td>
<td>Rubber Damper to Isolate the Gage from the Tunnel and Fill Oil</td>
<td>Temperature Sensitivity Issues No Return to Zero</td>
</tr>
</tbody>
</table>

**Table 1: History of Skin Friction Gages**

Previous skin friction gage designs have been largely designed to measure a single component of the wall shear. Only two of the gages shown in Table 1 measure
two-components of shear and also operate in hot, high-speed flows. The designs of Chadwick\textsuperscript{8} and DeTurris\textsuperscript{46} are the only other two gages capable of measuring two-components of wall shear in supersonic and hypersonic flows with a high heat flux. These designs used a Kistler-Morse piezoresistive strain sensor in a Deflection Sensor Cartridge (DSC) in the case of Deturris\textsuperscript{46} and piezoresistive semiconductor strain gages in the case of Chadwick\textsuperscript{8} who also used oil fill. Both designs used water to cool the gages in order to mitigate the effects of a high heat flux. They both also used silicon fill oil to provide additional thermal stability for the sensitive strain gages and DSC unit and viscous damping to minimize the effects of vibration. Using fill oil is particularly undesirable as it requires constant maintenance to keep the gage in operating condition. Refilling can be impossible when the gage is installed in a flight test vehicle. The water cooling can be used but adds to the complexity of the gage and is not available for all applications.

A two-component fiber-optic gage was developed by Pulliam\textsuperscript{54} but this design was for low-temperature tests. No high-temperature materials were used in this design. His gage used a cantilever flexure and measured the tilt of the floating head. Satisfactory results were only obtained with the use of oil fill. The gage design also showed sensitivity to total pressure of the flow and was rendered useless under certain conditions. This gage type only allows the use of a fiber-optic measurement system.

A high-temperature single-component gage was developed by Smith\textsuperscript{69} for use in the X-43A hypersonic flight test vehicle. The single-wheel-flexure concept used a single wheel flexure with the head balanced by a counterweight. The wheel was constructed of aluminum and the head and counterweights were made from copper. The housing was made of brass. All of these are comparatively low temperature materials which was acceptable as the test duration of the vehicle is only 5-7 seconds. Beyond this duration the thermal errors become large. The lack of cooling or viscous damping made this gage design unique for use in hot flows but it is limited to very short duration tests and can only measure a single component of wall shear. The proceeding discussion establishes the need for a two-component wall shear gage capable of operating in high-temperature flows without water cooling or viscous damping and measuring two-components of wall shear.
1.5 Design Goals for a High-Temperature 3D Gage

Wall shear is a small quantity. 250 Pascals of shear at Mach 2.4 with a total pressure of 7.2 atm is only 0.01257 Newton (0.00128 kg or 0.00282 lb) distributed over a 16 mm diameter floating head. Wall shear is an extremely small quantity to measure when you compare it with other forces in the flow. When a supersonic wind tunnel starts, the static pressure in the test section may go from atmospheric (101 kPa) to 25.9 kPa in a fraction of a second. A gage must be able to survive such a violent change in conditions and still be sensitive enough to measure the wall shear with reasonable accuracy. Once this formidable challenge has been overcome, there are still more design challenges. The extremely sensitive shear force measurement device must be insensitive to large changes in temperature, pressure gradients, or any other possible external effects.

The goal of this research is to develop an un-cooled and undamped fiber-optic wall shear gage capable of operating at temperatures up to 1073 K (1472° F) and measuring two components of the wall shear in 3D flows. The ambitious temperature requirement is for the entire gage, not just the surface exposed to the flow. The thermal goal represents a substantial advance over previous gage designs. Much higher flow temperatures can be measured for a limited duration. Fiber-optic displacement sensors were chosen as they can survive in the harsh 1073 K design conditions. In addition, fiber-optic sensors are much less sensitive to temperature changes than more conventional strain gages.

The basic concept behind the fiber-optic inferometry system is first that laser light propagates down an optical-fiber. The light then passes through the end of the optical fiber and across a gap. A portion of the light is returned to the core of the fiber and propagates back to a spectrometer. Extrinsic Fabry-Perot interferometry (EFPI), as invented by Murphy, relies on the formation of a low-finesse Fabry-Perot cavity in the gap between the end of the fiber and the reflective surface as shown in Figure 9.57. Some of the light is reflected at the end of the fiber and propagates back down to a collector. The remainder of the light hits an optically reflective surface and reflects back into the fiber. The resulting fringe pattern can be processed to determine the distance between the end of the fiber and the reflector.
Figure 9: Fiber and Reflector Geometry

Figure 10 illustrates the fiber-optic system used to measure the gap between the fiber and reflector. The outgoing signal is shown in the first portion of the figure. The light passes through the sensor, which consists of a fiber with a gap between it and a reflector. The light, which returns to the fiber core, is then measured by a spectrometer and the resulting fringe pattern is shown in the second portion of Figure 10 as the returned signal. The detailed definition of a fringe pattern can be found in Reference 53. This signal is then processed with a proprietary Luna Innovations Algorithm and the user receives a display of the gap between the fiber and reflector.

Figure 10: Fiber-Optic Gap Measurement System Diagram
One of the major disadvantages of fiber-optic interferometry systems is their sensitivity to vibration. The spectrometer used to record the fringe pattern returned from the sensor has a finite sample rate. If the amplitude and frequency of the gap change are at a certain value then smearing can occur. This results in the loss of the gap measurement for the instant that the signal is smeared. Figure 11 shows a normal fringe pattern in a quiescent environment. Figure 12 shows the degradation of the fiber-optic signal with severe vibration. The fringes appear smaller and the gap can no longer be resolved by the system. When the fringe pattern is lost a bad data point is created that is away from the rest of the recorded samples.

![Figure 11: Good Fringe Contrast (No Vibration)](image1)

![Figure 12: Poor Fringe Contrast (Severe Vibration Present)](image2)

The design goal of this gage is to be able to operate at 1073 K indefinitely. The design shear level is 250 Pa. This was selected to be representative of a hot, high-speed flow in a SCRAMjet engine and this level can be achieved in the Virginia Tech supersonic wind tunnel at Mach 2.4. A wide range of shear values can be measured as the sensor design allows the sensing area to be changed, i.e. a larger head for smaller shear values and a smaller head for flows for higher shear values.

Past studies have used conventional metal foil strain gages to measure the wall shear. Other studies have used fiber-optic interferometry to measure the displacement of a floating head but only for low temperature applications. No prior shear gage design has used both strain gages and fiber-optic measurements in a single package or successfully used fiber optics without damping. In addition no fiber-optic wall shear gage has ever been made for high-temperature high-speed flows.
Testing of this sensor will consist of rigorous a multi-step process. The sensor will first receive a static calibration of the fiber-optic and strain gage measurement systems. This will consist of hanging known masses from the head of the sensor and then measuring the resulting displacements and voltages. This calibration will be performed for both axes of the gage. The results of the modal and spectral analysis will be compared with experiment. The gage will then be thermally cycled in an environmental test chamber to quantify its sensitivity to temperature changes. Cold flow tests will be performed at Mach 2.4 and Mach 4.0 in the Virginia Tech Supersonic Wind Tunnel. Results from cold flow tests will be compared with semi-empirical prediction methods which calculate the $C_f$ values from the boundary layer thickness measured from experiment. The last step in the gage validation testing will be hot flow testing.

### 1.6 Outline

The following chapters detail the work performed in this investigation. Chapter 2 describes the instrumentation and equipment used during the experiments. Chapter 3 describes the first skin friction gage design process and testing. This design uses a parallel-beam flexure and attempted to avoid the need for viscous damping by raising the first vibration mode as high as possible. Chapter 4 details the design and analysis of the second gage design which uses a balanced wheel-flexure. Chapter 5 describes the construction, assembly, and static calibration of gage #2. Chapter 6 shows the results of thermal testing and Chapter 7 details testing in the Virginia Tech Supersonic Wind Tunnel for the second design. Conclusions and suggestions for future work are shown in Chapter 8.
2 General Facilities and Instrumentation

2.1 Electronics

National Instruments data acquisition cards were used to acquire data for supersonic wind tunnel tests and accelerometer testing. Detailed technical information is available in Reference 59. A summary of uncertainty for the two card types is shown in Table 2. The uncertainty is based on the full scale signal with the input limits listed as 0.50, 5.0 and 10.0 Volts which are typical for the measurement devices used during testing.

<table>
<thead>
<tr>
<th>Use</th>
<th>Card Type</th>
<th>Uncertainty ± 0.50 V</th>
<th>Uncertainty ± 5.0 V</th>
<th>Uncertainty ± 10.0 V</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wind Tunnel</td>
<td>NI6031E</td>
<td>± 0.043%</td>
<td>± 0.0415%</td>
<td>± 0.0115%</td>
</tr>
<tr>
<td>Accelerometers</td>
<td>NI6023E</td>
<td>± 0.169%</td>
<td>± 0.105%</td>
<td>± 0.165%</td>
</tr>
</tbody>
</table>

Table 2: Data Acquisition Card Uncertainty

The excitation and signal conditioning requirements for all electrical transducers and strain gages in this investigation were provided by the Micro Measurements Model 2310 signal conditioning amplifiers. This amplifier can provide a 0.7-14 Volt DC excitation to any device and can amplify a signal from its original value to 11,000 times its original value. The error information and all technical specifications can be found in Reference 55.

Thermocouples were used to monitor the temperatures of the various test conditions. Most thermocouples used were the Type-K. Some low-noise Type-E thermocouples were used in supersonic wind tunnel tests as these are standard for this facility. A special wall mounted Medtherm Type-K thermocouple was used to measure the wall temperature of the wind tunnel. This thermocouple has a flat head and is closely aligned with the surface of the tunnel wall. All thermocouples were purchased from Omega and meet ANSI limits of error.56

Various strain gage pressure transducers were used in conjunction with the 2310 amplifiers. One was used to measure the total pressure in the wind tunnel. The static
pressure was also measured with a transducer. The pressure transducers used were all calibrated with a deadweight pressure calibration rig. A listing of the pressure transducers used and their properties is shown in Table 3. Some uncertainties are listed as a typical value for a diaphragm type of pressure transducer. Several of the transducers were purchased at surplus auctions and the manufacturers either do not exist or no longer make the product listed.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Manufacturer</th>
<th>Model/Type</th>
<th>Serial</th>
<th>Range (atm)</th>
<th>Excitation</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_o$</td>
<td>Viton</td>
<td>Diaphragm</td>
<td>317</td>
<td>0-15 (est.)</td>
<td>12 V</td>
<td>± 0.5%</td>
</tr>
<tr>
<td>$P_{static}$, $P_{cs}$</td>
<td>Statham</td>
<td>PA285TC-50-350</td>
<td>50036</td>
<td>0-3.4</td>
<td>7 V</td>
<td>± 0.5%</td>
</tr>
<tr>
<td>$P_{o2}$</td>
<td>Genisco</td>
<td>PB-923</td>
<td>81-1299</td>
<td>0-6.8</td>
<td>10 V</td>
<td>± 0.5%</td>
</tr>
<tr>
<td>$P_o$ (HST)</td>
<td>Mahotomi</td>
<td>IP65-2002 50007</td>
<td>398</td>
<td>0-250</td>
<td>28 V</td>
<td>± 0.5%</td>
</tr>
</tbody>
</table>

Table 3: Pressure Transducer Technical Information

2.2 Fiber-Optic Data Acquisition System

The Luna Innovations Fast Fiber Scan (FFS) fiber-optic interferometry system diagram is shown in Figure 13 below. The FFS system is capable of measuring and recording the gap data for 4 independent channels simultaneously. The Luna Innovations FiberPro USB was also used for some setup and single channel testing. Both systems measure the absolute gap between a fiber and a reflector and can save the gap data as well as the raw spectrometer data of the signal. The gap and the raw spectrometer data are then saved. Saving both the gap data and the spectrometer data is imperative. The data can only be re-reduced if the spectrometer logs are saved. The system interfaces with a Dell 8200 2.4 GHz computer and is independent from the tunnel control computer. Therefore it must be manually time synchronized with the strain gage data during post-processing.
2.3 Virginia Tech Supersonic Wind Tunnel

The flow facility used to test the shear gage was the Virginia Tech Supersonic Wind Tunnel. This is a cold-flow facility with a total temperature of about 300 K. The facility is a blow-down tunnel with nominal test duration of approximately 10 seconds. Normal test conditions are a total pressure of 3.4 atm (~50 psia) and a total temperature of 300 Kelvin at Mach 2.4. At Mach 4.0 the tunnel can operate at 10-13 atm (150-200 psia) with approximately the same total temperature.

Figure 14: Schematic of Supersonic Wind Tunnel
The airflow through the facility is regulated with a hydraulically actuated valve in series with a fast-acting butterfly valve. The stagnation pressure is variable from 3-20.5 atm and the Reynolds number per meter can vary from $2 \times 10^6$ to $5 \times 10^6$. The compressor is an Ingersoll-Rand Type 4-HHE-4 4-stage air compressor driven by a 500 hp Marathon Electric Company motor. Dried air is stored in tanks of 23 m$^3$ volume.

The test section is 23x23 cm (9x9 in.) in cross section, and the floor of the tunnel is removable to facilitate a wide variety of test apparatus. A number of shear sensors have used the supersonic tunnel at Mach 2.4 in the past so a large body of experimental evidence is available for comparison. In addition extensive boundary layer surveys have been performed and the flow is well quantified in the test section. The shear values on the floor of this tunnel have been documented in great detail using both with direct and indirect methods. It will provide validation for the gage and prove the basic design. A photograph of the tunnel test section and nozzle block is shown in Figure 15 below.

![Figure 15: Virginia Tech Supersonic Wind Tunnel](image)

A 450 MHz computer serves as the data acquisition system with a National Instruments 16 bit data acquisition card and multiplexer which allows up to 32 channels at a sample rate of up to 500 Hertz. The card used is the National Instruments NI 6031E and detailed technical information can be found in Reference 59. The computer also
controls the operation of the wind tunnel. It has provision to operate a traverse with probes for surveying the tunnel boundary layer. The total pressure in the plenum chamber and the total temperature are measured and recorded with each tunnel run.

Figure 16 to Figure 18 below shows the acceleration loads in the Virginia Tech supersonic wind tunnel at Mach 2.4 in Power Spectral Density (PSD) format. This data was taken by Remington\textsuperscript{60} in a previous study. These figures show where the vibration in the tunnel is at its worst and which frequencies should be avoided when designing the structure of the gage flexures. If the modes of the gage coincide with the peaks in vibration resonance could occur. If this happens at a high enough frequency the fiber-optic system will be rendered useless due to smearing of the returning signal as the change in the signal may be faster than the spectrometer can sample. Figure 16 shows the streamwise (X-axis) vibration of the tunnel. There is a peak of $-1 \times 10^{-6} \text{G}^2/\text{Hz}$ at 2500 Hertz in the X-axis. Figure 17 shows the transverse (X-axis vibration) and there is a peak in vibration $\sim 900 \text{ Hz}$ and another around $1700 \text{ Hz}$ which is an order of magnitude higher than typical values found in skin friction gages\textsuperscript{70,69}.

![Figure 16: Measured Streamwise Vibration at Mach 2.4\textsuperscript{60}](image1)

![Figure 17: Crosswise Vibration in at Mach 2.4\textsuperscript{60}](image2)
Figure 18 shows the out of plane (Z-axis) vibration. The out of plane vibration has four peaks, which should be avoided. They are at 800 Hz, 1650 Hz, 1950Hz, and ∼2500 Hz. All of the tunnel loads are loads at which the gage must be able to operate successfully.

Figure 18: Out of Plane Vibration in at Mach 2.4

Figure 19 shows a NASA flight test PSD curve for survivability. The B curve is of interest as it is the one for atmospheric flight test vehicles. Even the lowest magnitude of 0.04 G²/Hz is 40,000 times greater than the vibrations experienced in the supersonic wind tunnel. This curve represents what a gage must be able to survive in order to be considered airworthy by NASA. These loads do not necessarily reflect the conditions at which the gage must operate. The greatest loads may simply be experienced while the aircraft is taxiing out to the runway and passing over expansion joints in the pavement.

2.4 Vibration Test Apparatus

To conduct vibration tests, a Ling Dynamic System LTD Model V103 force transducer was used to shake the gage. Technical information is shown in Reference 61. The transducer creates one pound of force for every 100 millivolts applied. A Harmon Kardon HK770 Twin Torordal Ultra-wideband DC Amplifier was used to power the
transducer. Details can be found in Reference 62. The source for the signal was a HP3566A Dynamic Signal Analyzer. The input signal was measured with an accelerometer, which has an output of 10 millivolts per G. The displacement output was measured with the gage fiber-optic system. A diagram of the vibration test setup is shown in Figure 20.

![System Diagram of Vibration Test](image)

**Figure 20: System Diagram of Vibration Test**

### 2.5 Accelerometer Test Apparatus

Accelerometer measurements require that an additional data acquisition system be utilized. This system is shown in Figure 21 and consists of a Pentium II computer with a National Instruments 6023E 200 kHz 12 bit data acquisition card with detailed information shown in Reference 59. This card is plugged into a National Instruments BNC-2090 adaptor, which allows BNC cables to be attached to the data acquisition card. An overview of the system is shown in Figure 21 below. The next portion of this system is the PCB Model 482A16 4-channel accelerometer power unit and two PCB Model 484B I.C.P. power units. These were used to power the accelerometers themselves.
The accelerometer numbers, orientation, type, and calibration data are shown in Table 4. The calibration values, shown as millivolts per “G” were used in the data reduction program to calculate the number of G’s, of acceleration, measured on the various points on the gage and wind tunnel.

<table>
<thead>
<tr>
<th>#</th>
<th>Type</th>
<th>Serial #</th>
<th>Range G/s/Hz</th>
<th>mV/G</th>
<th>Calibration</th>
<th>Date</th>
<th>Axis</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>353B17</td>
<td>10335</td>
<td>500/1-10,000</td>
<td>10.15</td>
<td>0.0010350</td>
<td>03/18/94</td>
<td>Z-Tunnel</td>
</tr>
<tr>
<td>5</td>
<td>303A03</td>
<td>4758</td>
<td>500/1-10,000</td>
<td>9.94</td>
<td>0.0010136</td>
<td>12/20/82</td>
<td>X-Gage</td>
</tr>
<tr>
<td>6</td>
<td>303A03</td>
<td>5602</td>
<td>500/1-10,000</td>
<td>11.22</td>
<td>0.0011441</td>
<td>07/29/83</td>
<td>Y-Gage</td>
</tr>
<tr>
<td>7</td>
<td>A353B17</td>
<td>10187</td>
<td>500/1-10,000</td>
<td>9.93</td>
<td>0.0010125</td>
<td>03/11/94</td>
<td>Z-Gage</td>
</tr>
<tr>
<td>9</td>
<td>308B09</td>
<td>11428</td>
<td>50/.05-3,000</td>
<td>100.8</td>
<td>0.0102784</td>
<td>05/23/83</td>
<td>X-Tunnel</td>
</tr>
</tbody>
</table>

Table 4: Accelerometer Characteristic Data

2.6 Boundary Layer Survey Equipment

Boundary layer surveys were conducted using a traverse to move a probe from the wall to about 23 mm (0.9 in.) above the wall. This traverse consists of a stepper motor controlled by the tunnel computer and a shaft with notches for the driven gear to grip it. The head of the traverse is shown in Figure 22. The first probe, shown from left to right, is the cone-static probe. The second probe is a total temperature probe which is made from a hollow tube with a small bleed hole and a Type-E thermocouple inside. The last probe is a Pitot or total pressure probe which measures the total pressure of the flow behind the normal shock forming at the tip of the probe. Compressible flow relations can
be used to calculate $P_{o1}$ from the measured $P_{o2}$. The data from the Pitot and con-static probes can be processed to yield the local Mach number and all other flow quantities. The probes are separated by a distance of 7mm from centerline to centerline. Figure 23 shows the assembled probe and traverse in the test section of the wind tunnel.

Figure 22: Traverse Probe for Boundary Layer Survey

Figure 23: Probe mounted to Traverse in Supersonic Wind Tunnel
3 Parallel Beam Gage Design and Analysis

3.1 Design and Background

A parallel- flexure wall shear sensor design was selected initially as the most logical choice of a flexure for the first attempt at a high-temperature fiber-optic wall shear sensor. This choice was driven by several considerations. First and most important, this configuration involves a simple geometry so it might be relatively easily fabricated with high-temperature materials, such as ceramics, see Figure 24 (a). Second, this design was an attempt to raise the first mode of the sensor out of the vibration range typical of supersonic laboratory and flight tests (~100-2000 Hz) through the use of a light head and relatively stiff flexures. This design can only measure a single component of shear.

![Figure 24: Schematic Drawing of Skin Friction Flexure Types](image)

The general arrangement is shown in Figure 24 (a) compared to a cantilever beam design in Figure 24 (b). The parallel beam design uses a sensing head resting on two thin beams. These beams are compliant in the direction of desired motion. The head of the sensor essentially translates for small displacements instead of translating and rotating as would the head of a sensor with a cantilever beam flexure. This should minimize misalignment errors. An advantage of the parallel plate design is its resistance to a pressure gradient applied to the top of the sensing head and is displayed in Figure 24. Figure 24 (a) would be more resistant to a pressure gradient as it has a beam on each end
of the floating head, making it much more difficult to cause misalignment due to the
tilting of the head. Other workers have used this configuration with success. Figure 25
shows an example.

Figure 25: Parallel-linkage balance

Figure 24 (b) would tend to tilt about the center of the head when displaced or
exposed to a pressure gradient. Prior skin friction gage designs utilized a head mounted
on a cantilever beam. This cantilever beam flexure had to be quite long in order to
accommodate the needed strain values. Older gage designs used conventional strain
gages and their lower sensitivity dictated the needed values of strain and corresponding
deflections.

The flexible cantilever beam design, while successful in the past, can create a
number of difficulties. The first difficulty is the need for viscous damping. The beam
has fairly low natural frequencies and when these modes are excited there must be
something to damp them out. This need for damping greatly increases both the
complexity of the sensor and the maintenance required. One solution that is successful in
laboratory tests but not for flight tests, is to fill the inside of the gage housing with
viscous oil. The oil fill also aids in thermal protection and stabilization. But, the oil
levels have to be constantly monitored and the sensor refilled when they get low.

Alumina was the original material of choice for the construction of the sensor as it
has well-known capability for high-temperature and the ability to be diffusion bonded.
Alumina is a high-temperature ceramic which cannot be easily machined. This material
allows a much broader range of sensor concepts to be investigated. Cantilever beam
designs and several other concepts were considered before deciding on a parallel beam
design. The method of construction for an Alumina sensor involves diffusion bonding
various small parts together. This parallel plate design provides flat sections which have
ample surface area for the bonding process. The multiple step bonding procedures
proved to be problematic in practice and the use of this material was deferred in favor of
aluminum to first demonstrate the gage design concept.

Once the general concept for a parallel plate design was established as the most
logical choice, a decision was made to create a working model of the design out of
aluminum. This model would provide the opportunity for investigation into the sensor’s
response and the ability of the available design tools to accurately predict the response in
a locally available test facility. The aluminum gage design also had the advantage of
being able to use both a fiber-optic displacement sensor and semi-conductor strain gages.
By using the two, independent measurement systems, the chances of learning more about
the design are greatly increased. The experiments would not simply show failure or
success but would provide a wide range of information useful for future design. The
aluminum gage also had the advantage of being much cheaper and faster to construct than
a comparable Alumina Gage.

The operating conditions of the Virginia Tech Supersonic Wind Tunnel were the
design conditions for the aluminum version of this gage concept. These conditions are
Mach 2.4 flow with a wall shear of approximately 250 Pascals.
3.1.1 Analytic Design

The analytic solution to find the head displacement of the parallel beam sensor was developed. Small displacements relative to the beam size were assumed. The material is presumed to be linear-elastic and isotropic. Essentially one end of the beam is anchored to the sensor base and the other end is constrained against rotation as shown in Figure 26. It is free to move only in the Y axis direction.

![Figure 26: Schematic of Structural Problem](image)

The following equations can be inferred from Figure 26.

\[
\begin{align*}
v(0) &= 0 \\
v'(0) &= 0 \\
v'(L) &= 0 \\
-EHv''(L) &= F
\end{align*}
\]

\[
v(x) = \frac{A_3}{3}x^3 + \frac{A_2}{2}x^2 + Ax + A_0
\]

Equation 7

Solving this system of equations yields the following values for the four unknown coefficients.
\begin{align*}
A_0 &= 0 \\
A_1 &= 0 \\
A_2 &= \frac{FL}{2EI} \\
A_3 &= -\frac{FL}{2EI}
\end{align*}

Equation 8

This yields the following expression for the end displacement, where X=L, on the beam in question.

\[ v(L) = \frac{FL^3}{12EI} \]

Equation 9

The moment of inertia used for this computation is the following\textsuperscript{64}.

\[ I = \frac{1}{12}bh^3 \]

Equation 10

Figure 27 shows the change in beam end displacement with change in beam length. There are two parallel beams. The thickness was chosen as the lowest possible with a machining tolerance of \(\pm 1\mu m\) so the machining error would be less than 1%. The width was chosen to be 6 mm due to make it fit under the sensing head. Each one is 0.2 mm thick, 4.22 mm tall and 6 mm wide. The final length was chosen to provide a displacement of 430 nanometers. The head of the sensor, which is exposed to the flow, has a 14 mm diameter. The actual fiber cannot be at the exact end of the beam as it would interfere with the movement of the floating head. Therefore the measured displacement will be less than this predicted value. The addition of a reflector and associated adhesive to the side of the beam will also increase its stiffness and reduce its displacement.
Once the rough sizing of the sensor had been established using the analytical solution, a higher fidelity model was needed to verify the displacement results and predict the strains on the structure. This need was met through the use of ANSYS, a finite element analysis package.\textsuperscript{65} This package has the ability to perform static structural calculations, modal analysis, spectral analysis, thermal analysis, and computational fluid dynamics. A solid model of the sensor was created in ANSYS using its native drawing package. This solid model is shown below in Figure 28. The flexures are 125 \(\mu\)m thick, 6\,mm wide, and 4.22 \,mm tall. The floating head is 14 \,mm in diameter and there is a 100 \,\mu m gap between the head and housing. The gage fits into a hole in a wall and the floating head is flush with the wall’s surface.
The first concern with ANSYS after the program roughly agreed with the analytical solution, was the effect of the grid on the results. The lines which define the outline of the webs were divided up into segments. This defined the grid spacing at the edges of the webs. The element used for this model is the SOLID 92 element, which is a ten-node tetrahedron with quadratic displacement behavior. This element choice provided a balance between computer resources available, computational time, and accuracy.

The grid design in ANSYS with the SOLID 92 element is a partially mapped mesh. The program uses a sizing routine to keep the rest of the element shapes within the parameters required for a useful solution. In addition to changing the number of line segments dividing up the webs the effect of changing the smart mesh was also examined. The smart mesh parameter is in ANSYS and lets the user define the coarseness of the mesh. The value 1 is the finest automatic mesh and 10 is the coarsest. The displacement results and corresponding number of solid elements are shown in Figure 29. These values agree closely with the analytical solution which is plotted on the figure. The difference is within the margin of error of the fiber-optic sensor. The small changes in the results indicate that the chosen grids are sufficient for the design work performed here and should provide a good basis for comparison with experimental results.
A view of the grid used is shown in Figure 30. This grid uses 55,164 elements. The majority of the elements are concentrated on the parallel beams as most of the movement and stress in the sensor is concentrated there. The shear load was calculated using the Excel spread sheet and then applied to the nodes on the head of the finite element model. The shear load is distributed over 585 nodes on the head and should provide a good approximation of the actual distributed load. This grid choice provides an FEA model which can be solved on a modern personal computer in a matter of hours and still yield results very close to the analytic prediction.
3.1.2.1 Static Analysis

The displacement and strain results are shown in Figure 31 and Figure 32 respectively. The load is applied in the positive direction of motion for the sensing head. The calculated displacement of the sensor head where the reflector will be placed for the fiber-optic sensor, is 428 nanometers. This is only two nanometers less than the 430 nanometer value predicted by the analytical solution. This value of displacement was chosen so that the error in the fiber-optic system measurement would be small and sufficient strain would be produced to use the semiconductor strain gages that were later abandoned. The actual measured displacement will be 2mm from the sensing head and ANSYS predicted a value of 238 nanometers at this location. The uncertainty in the fiber-optic displacement measurement is on the order of a nanometer. The strain value to be measured by the strain gages will be approximately 16 microstrains on the thin beams. This is a reasonable value for semi-conductor strain gages but far below the 50-75 microstrains required for metal-foil strain gages.

![Figure 31: Displacement (meters) for a 250 Pa Wall Shear](image1)
![Figure 32: Strain for a 250 Pa Wall Shear](image2)

3.1.2.2 Modal Analysis

Once the static analysis and design of the skin friction gage was completed, a modal analysis of the skin friction sensor was performed in ANSYS. This analysis
predicted the natural frequencies within a specified frequency range. The input needed for the modal analysis consisted of the constraints on the boundary of the sensor. In this application, this consisted of constraining the sensor from translation or rotation at the base. The next specification is the number of modes to be extracted and the frequency range over which to search for those modes. For this portion of the project, the only real concern was that the first mode fall outside the frequency range specified in NASA flight test requirements in Figure 19. The first five modes were calculated in ANSYS and the frequency range of 0-50,000 Hertz was searched. The elemental results were calculated so that the maximum amount of information would be available and then a spectral analysis could be run using the results from the modal analysis.

The results of this modal analysis are shown in Table 5 below. Three cases with progressively fewer elements were run. There was a noticeable effect on the modal results. However, the effect is insignificant for the purpose at hand. The only goal is to show that the first mode will be out of the range needed for NASA flight tests.

The first mode is out of the 15-2000 Hertz flight test regime specified in Figure 19. This is desirable as if the natural frequency fell within this range damping would likely be required to prevent damage to the sensor. This was the result of careful structural design and involved several iterations of sensor head designs to make the first mode fall out of this regime.

<table>
<thead>
<tr>
<th>MODE</th>
<th>Frequency Hz (41,240 Elements)</th>
<th>23,275 Elements</th>
<th>14,373 Elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2122.8</td>
<td>2129.4</td>
<td>2147.1</td>
</tr>
<tr>
<td>2</td>
<td>14,135</td>
<td>14,152</td>
<td>14,176</td>
</tr>
<tr>
<td>3</td>
<td>19,477</td>
<td>19,497</td>
<td>19,529</td>
</tr>
<tr>
<td>4</td>
<td>22,855</td>
<td>22,867</td>
<td>22,903</td>
</tr>
<tr>
<td>5</td>
<td>26,681</td>
<td>29,707</td>
<td>26,781</td>
</tr>
</tbody>
</table>

Table 5: Results of Modal Analysis
3.1.2.3 Spectrum Analysis

Figure 19 provides the range to be analyzed for the spectrum computations in ANSYS. These calculations are used to provide a random vibration analysis of the sensor. This is a statistical analysis instead of the deterministic analyses performed previously so the results will show the probability of a stress or displacement in response to a given range of loading. This analysis uses power spectral density (PSD) to quantify the loading. The PSD is a “…statistical measure defined as the limiting mean square value of a random variable.”

The elemental results from the modal analysis are used in the spectrum analysis portion of ANSYS. The same element used in the modal analysis was used in the spectrum analysis. This element is the SOLID92, which is a 10-node tetrahedron which has nodes at the endpoints and midpoints of each line. The spectrum analysis lets one specify the PSD curve, the number of modes to expand upon, boundary constraints, and then solves for the participation factors. Reference 65 can be consulted if more detail is needed on the solution process. It details the entire process in a step by step guide with mathematical details and instructions for code usage. The participation factors are then used to solve the random vibration problem. After this is solved, the modes are combined in ANSYS as the final solution step for the spectrum analysis.

A summary of results from the spectrum analysis is Table 6. This shows the 1σ displacement of the sensor when it is subjected to the PSD curve shown in Figure 19. The 1σ results mean that the values will be at or below those shown in the table 68.2% of the time. The values will be between 1σ and 2σ 27.2% of the time. The stresses, throughout the flexure, are shown in Figure 33. These spectrum results were computed using the mesh with 41,240 elements in ANSYS. The main point taken from the spectrum analysis is that if the gage is subject to the NASA B curve it will be destroyed unless the motion is limited. The ultimate shear strength for 6061 T6 aluminum is 205 MPa and this will be exceeded. The housing around the floating head provides a mechanical stop and prevents the head from moving as far as shown in Table 6. A soft stop or bumper could be included in the design to prevent damage to the head. This was not felt necessary for a proof of concept gage.
<table>
<thead>
<tr>
<th>$\sigma$</th>
<th>Displacement (µm)</th>
<th>Max. Stress (Pa)</th>
<th>Max. Strain</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.39</td>
<td>$2.87 \times 10^9$</td>
<td>0.0413</td>
</tr>
<tr>
<td>2</td>
<td>2.77</td>
<td>$5.74 \times 10^9$</td>
<td>0.0826</td>
</tr>
</tbody>
</table>

Table 6: Results from Spectrum Analysis

![Figure 33: 1-$\sigma$ Stress Results](image)

### 3.1.3 Thermal Analysis of Aluminum Sensor

Thermal effects on both the sensor itself and the instrumentation affixed to the sensor dictate that an investigation of these effects be performed prior to testing. This will give a baseline prediction for the expected results. Both the semiconductor strain gages and the fiber-optic systems are sensitive to temperature changes. The temperature changes will also cause expansion or contraction of the sensor itself. The powerful ANSYS finite element package is again used to accomplish this task.

The conditions for this analysis are a nominal total temperature of 300 K, which is roughly what can be expected in the wind tunnel. A total pressure of 379 kPa, a wall temperature of 300 K, and a flow of Mach 2.4 are assumed. These approximate the conditions in the Virginia Tech Supersonic wind tunnel. The flow in the tunnel isn’t heated and this is known as a “cold flow.” This means that the wall of the tunnel will be cooled during the run, and so will the gage and associated instrumentation.
The skin friction coefficient can be calculated, using the Schultz-Grunow relation. This is shown in Equation 12 below. Equation 11 shows the expression for $Re_\delta$ which is based on the boundary thickness $\delta$. The boundary layer thickness is approximately 1 cm. It should be noticed that this predicts the skin friction coefficient for the incompressible case. A correction must be applied which is $C_{f,comp}/C_{f,incomp} = 0.7$ for $M=2.4$ from Figure 10-7 in Reference [1].

$$Re_\delta = \frac{\rho U e \delta}{\mu}$$

**Equation 11**

$$C_f = 0.0456(Re_\delta)^{-1/4}$$

**Equation 12**

These expressions yield a turbulent $C_f$ value of 0.0013 and combined with Equation 13 to Equation 15 can be used to estimate the turbulent film coefficient and heat flux values.

These values are very important for the thermal analysis.

$$St = \frac{C_f}{2 Pr^{0.4} (T_w / T_e)^{0.4}}$$

**Equation 13**

$$h = \rho U_e C_p St$$

**Equation 14**

$$q_w = h(T_w - T_e)$$

**Equation 15**

The values of $h$ and $q_w$ were calculated to be 201 J/(m$^2$sK) and 32,210 (W/m$^2$) respectively.

These values were input into ANSYS and the solution plotted below is the temperature of the sensor after ten seconds of exposure of the head to Mach 2.4 flow conditions in the wind tunnel. The sensor is assumed to start at a temperature of 300 K for the purpose of this analysis. This should be representative of the nominal conditions. Figure 34 below shows these results. The results shown are after twelve seconds of exposure to the flow conditions. This shows approximately a 5 K temperature change in
the webs. Additional temperature change would likely occur over repeated runs unless
the wall temperature was allowed to return to its equilibrium value. This indicates that
the change in temperature on the webs won’t cause a large error in the strain gages results
in the supersonic wind tunnel tests. It does show that this design changes temperature
rapidly as this case isn’t a severe condition.

![Figure 34: Contour Plot of Thermal Results (M=2.4, 12 Seconds)](image)

**3.1.4 Mechanical Gage Design**

Once the sensor analysis was performed, a housing and fiber-optic mounting
system were designed. The resulting final configuration is shown in Figure 35 and
Figure 36. Several design choices presented themselves. The first major choice was that
the housing would be a two-piece unit which bolts together and seals with a copper
gasket. This two-piece unit facilitates the installation of all the components in an open
environment. It also allows changes and maintenance to be performed. Previous fiber-
optic designs were rendered useless as the fiber could not be replaced if broken. This
design also uses connectors for all instrumentation. This keeps the installation simpler as
fragile fiber-optic lines are not left unprotected. Figure 35 shows the gage assembled and
in its housing. The floating head is labeled in the figure and it sits flush with the top of
the outer housing. The outer diameter of the housing is 50.8 mm (2 in.) and the outer
diameter of the housing, which surrounds the floating head, is 25.4 mm (1 in.).
The gage is shown again in Figure 36 with half of the housing removed for clarity. This view allows the red sensor portion to be seen next to the green fiber-optic ferrule. The pink objects are the connectors. The one on the left hand portion of the figure is for the optical fiber and the one of the right is for the strain gage wires. The white tubes protruding from this half of the housing are simply pins which hold the housing in alignment. The channel below the red sensor is for the routing of the strain gage wiring.

Figure 36: Half of Skin Friction Sensor Housing with all Internal Parts and Connectors

Figure 37 shows a partially dimensioned drawing of the left half of the skin friction gage housing. It is only about two inches in diameter and about two inches tall.
An effort was made to keep the gage compact while keeping within the abilities of the machines in the Aerospace Department’s machine shop.

![Figure 37: Drawing of Housing Half (all Dimensions in Inches)](image)

The drawing shown in Figure 38 below is the actual sensing portion of the skin friction gage. This part was built by the Damon Company located in Salem Virginia using electric discharge machining techniques. This allowed the 0.0049” thick webs to be constructed with a high degree of precision. The sensor is composed of 6061-T6 aluminum.

![Figure 38: Drawing of the Sensor for the Aluminum Gage (all Dimensions in Inches)](image)
3.1.5 Aluminum Cf Gage Fabrication and Testing

3.1.5.1 Fabrication and Assembly

The sensor portion of the aluminum skin friction gage was machined at Damon Company in Salem, Virginia. The first step in the process was to turn 6061-T6 aluminum on a lathe to create a blank for the gage. This defined the outer dimensions. The next step in the process was to use Wire Electron Discharge Machining (EDM) to cut out the webs and base of the sensor. The finished product can be seen in Figure 39. This figure shows the roughness of the portions of the sensor which underwent the EDM process. It also show the small size of the head and flexures. The EDM process was chosen as it was the only economical way to create such a small and delicate part out of aluminum.

![Figure 39: Side View of Sensor](image)

Figure 40 and Figure 41 show the completed housing for this skin friction gage. This was created in the Virginia Tech Aerospace and Ocean Engineering Machine Shop. The attempted dimensional tolerances were plus and minus 0.0005 in. This was quite a stretch with the available tools. The first figure shows half of the gage with the thin copper gasket. The second figure shows both halves of the housing. In Figure 41, the small overall size of the gage is demonstrated. Creating a gage of the absolute minimum external dimension was not an overriding goal in this portion of the project. The semiconductor strain gages necessitated a much larger housing. This is mainly due to the commercial 10-pin connector used. Without the strain gages, the housing could have been only slightly larger than the sensor itself.
Figure 40: Housing Half with Gasket

Figure 41: Housing Halves

Figure 42 and Figure 43 show the housing with the sensor in place. Figure 42 shows the overall picture of the completed gage. Figure 43 shows a similar view of the gage along with a scale so that the relatively small overall dimensions can be seen. Figure 42 shows a top view of the sensor and housing. This view of the gage assembly shows the mounting pattern of the six large bolts which will secure the gage to the wind tunnel floor plate in the Virginia Tech Supersonic Wind Tunnel. The sensor is held to the housing using eight, nylon-tipped 2-56 set screws. The nylon-tipped set screws were chosen as conventional steel set screws would dig into the comparably soft aluminum and make alignment difficult.

Figure 42: Top View of Housing and Sensor

Figure 43: Housing with Scale
3.1.5.2 Calibration

The sensor was instrumented with fiber optics alone and calibrated using a direct force method. The FiberPro USB system was used for this sensor. A flat silicon reflector, which is eight microns thick, was attached to each beam near the top with a one micron thick layer of adhesive.

The calibration setup is shown in Figure 44 below. The calibration consisted of hanging weights in a paper cup which was affixed to the head of the sensor by a string. The gage was slightly tilted to avoid the string rubbing on the outer housing of the gage. Small weights were then added to the bucket to simulate a wide range of shear values. The cup was shielded by a beaker in order to minimize the effects of any airflow on the bucket.

![Figure 44: Calibration of Gage with Fiber System](image)

The results of the calibration are shown in Figure 45. It can be noted that the displacement values at the predicted shear of 250 Pa are significantly less then those predicted by analysis and computation. This is due to two factors. When the reflectors were installed, the increase in beam stiffness due to the glue layer and the reflectors wasn’t accounted for in computations. The second factor is that the fiber wasn’t looking at the top of the reflector. It was positioned so that it would view the displacement approximately 2 millimeters from the top of the beam. The analytical model predicts a displacement of approximately 238 nanometers at that location which is in line with the calibration results.
3.1.5.3 Wind Tunnel and Vibration Results

The next step with the calibrated sensor was to test it in the Virginia Tech Supersonic Wind Tunnel at Mach 2.4. The total pressure for this test was 3.4 atm (50 psia) and the total temperature was 300 K. Several attempts were made to take measurements under these conditions. As soon as the tunnel was started, the fringe pattern disappeared and the gap could no longer be resolved. It was assumed that the vibration, transmitted through the wind tunnel walls, was the cause of the fringe pattern disappearing. Thus, vibrations can cause a “smearing” of the fringe pattern if the amplitude and frequency of the change in gap are inappropriate.

A vibration test was performed to quantify the conditions needed to disrupt the fringe pattern and make the gap measurement impossible. The test setup is shown in Figure 46 below showing the shaker and gage suspended from a tripod. A detailed schematic of this setup is given in Figure 20. The excitation was applied to give about 0.3 G’s of maximum acceleration and the excitation was applied on the sensitive axis of this single component gage. The load is a simple sinusoid of a single frequency.
The results of the vibration tests are shown in Table 7. These tests showed that the fringes disappear at about 2.6 kHz and are completely useless between 2.6 and 3.8 kHz. This is due to the design of the gage for the 15-2000 Hz flight test regime specified by NASA flight test requirements and mentioned earlier. It is hypothesized that the addition of damping such as with an oil fill would allow flow measurements with this current gage.

<table>
<thead>
<tr>
<th>Frequency (kHz)</th>
<th>Fringes</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-2.5</td>
<td>Good</td>
</tr>
<tr>
<td>2.524</td>
<td>Poor</td>
</tr>
<tr>
<td>2.592</td>
<td>Bad</td>
</tr>
<tr>
<td>2.632</td>
<td>Poor</td>
</tr>
<tr>
<td>2.688</td>
<td>Bad</td>
</tr>
<tr>
<td>2.944</td>
<td>Poor</td>
</tr>
<tr>
<td>2.984</td>
<td>Bad</td>
</tr>
<tr>
<td>3.086</td>
<td>Poor</td>
</tr>
<tr>
<td>3.8</td>
<td>Good</td>
</tr>
<tr>
<td>3.8-73.8</td>
<td>Good</td>
</tr>
</tbody>
</table>

Table 7: Vibration Test Results with Rigidly Mounted Gage
When rigidly mounted, the gage had been shown to be unacceptable for use in the Virginia Tech supersonic wind tunnel. The next step was to devise a method of isolating the gage from the tunnel vibration. It was hypothesized that a layer of silicon rubber, between the gage housing and the tunnel floor plate, might provide adequate damping. Figure 47 details this arrangement.

![Figure 47: Schematic of Gage Isolated with Silicon](image)

The gage was first mounted to a plate which had an oversize hole drilled in it for the sensor head. This allowed the sensor to move freely along any direction in-plane with the mounting plate. This first plate was small and light to allow the use of the previously described vibration test apparatus. A photograph of the gage and test plate in the vibration test rig is shown in Figure 48.

![Figure 48: Isolated Sensor in Vibration Test Rig](image)

It is worth noting that a connector system was devised for this gage. A previous fiber had broken at the junction of the cable and the housing. It was decided that a connector system should be devised which would allow the gage to take on a more robust
and user friendly form. Table 8 below shows the improvement that the addition of the silicon made to the response of the gage. The gage still has an area in which it isn’t desirable to operate, but this area is reduced and moved father away from the area of peak vibration in the wind tunnel. The load is the same 51.559 mV (.5138 lb) as used in the previous series of vibration tests. The previous 1.3 kHz dead-band has been reduced to 0.25 kHz with damping.

<table>
<thead>
<tr>
<th>Frequency (kHz)</th>
<th>Fringes</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-2.7</td>
<td>Good</td>
</tr>
<tr>
<td>2.768</td>
<td>Bad</td>
</tr>
<tr>
<td>2.912</td>
<td>Poor</td>
</tr>
<tr>
<td>2.93-10</td>
<td>Good</td>
</tr>
</tbody>
</table>

**Table 8: Response of Damped Housing to Vibration**

The next step after damping had been shown effective was to test the new gage in the wind tunnel at Mach 2.4 and under the same conditions as the previous tests. Figure 49 shows a representative result from this series of tests. The gage had a greater response to this flow then predicted in the modeling. There is no level portion evident in this figure where the total pressure and wall shear reach a steady value. The gage also took a very long time to return to the zero position after the wind tunnel had stopped running, suggesting that thermal effects might have washed out the shear measurement.
The poor response led to the assumption that the large numbers of different materials in the sensor were contributing to sensitivity in changes due to temperature. The wind tunnel floor plate experiences approximately a 15°C temperature change during each wind tunnel run. A check on this hypothesis was made by utilizing a Teaney Environmental (Tenn II) environmental test chamber. A Type-K Thermocouple was used to measure the surface temperature of the housing. A Dell 7500 lap top computer with a FiberPro USB was used to record the response of the sensor to the thermal effects.

The test results are shown in Table 9, and they clearly show that the sensor response is strongly influenced by a change in temperature. This rendered the sensor nearly useless for supersonic wind tunnel flows where the wall temperature changes by any significant amount. The thermal test results were so poor that the aluminum parallel beam sensor was abandoned in favor of the design concept described in the next chapter.
A computational investigation was undertaken to see if a parallel beam flexure gage with a first mode higher than likely significant vibration was possible. The result of the computation was that it was not possible to create a light enough head with reasonable materials. Without an extremely light head or very stiff beams it wasn’t possible for a sensor to be created within the design space of this investigation. These results yielded the conclusion that an alternative shear sensor configuration was needed in order to satisfy the requirement for a sensor which can operate in a hot flow.

<table>
<thead>
<tr>
<th>Gap (micrometers)</th>
<th>Thermocouple (°C)</th>
<th>Chamber (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>143.682</td>
<td>25.0</td>
<td>25.7</td>
</tr>
<tr>
<td>143.725</td>
<td>21.3</td>
<td>16.3</td>
</tr>
<tr>
<td>143.805</td>
<td>15.6</td>
<td>11.3</td>
</tr>
<tr>
<td>5143.924</td>
<td>10.9</td>
<td>6.7</td>
</tr>
<tr>
<td>144.093</td>
<td>7.0</td>
<td>2.3</td>
</tr>
<tr>
<td>144.305</td>
<td>3.0</td>
<td>-1.1</td>
</tr>
<tr>
<td>144.601</td>
<td>0.1</td>
<td>-4.6</td>
</tr>
<tr>
<td>145.004</td>
<td>-4.1</td>
<td>-8.8</td>
</tr>
</tbody>
</table>

Table 9: Environmental Chamber Test Results
4 Dual Wheel Gage Design

4.1 Design Background

Following the problems encountered with the parallel-beam flexure design, a new concept was needed. The first problem with the design was its sensitivity to vibration with a corresponding smearing of the fringe pattern and loss of the fiber-optic gap measurement when the supersonic tunnel was run. The second problem was that the gage was extremely sensitive to temperature, and that rendered it unusable. The sensitivity to temperature was so severe that even a ~15 K total temperature change during a supersonic tunnel run overwhelmed the wall shear measurement. A gage that is insensitive to temperature changes, resistant to linear accelerations, and measured 3D flow was desired for future work.

Prior wall shear gage flexure designs were examined to see which might be used or adapted to meet the current requirements. Early in the design process it was decided that relying only on the fiber-optic measurement system was unwise. The fiber-optic system is essentially a prototype in itself and some diagnostic tools seem to be needed. This calls out for a skin friction gage that can accommodate both fiber optics and strain gages.

Figure 50: Schematic of Two Prior Shear Gage Concepts
Several previous designs used a single cantilever-beam flexure with a floating head on the top.\textsuperscript{5, 53, 68, 60} This type of gage design is shown in Figure 50. Some designs have used semi-conductor strain gages. These strain gages are about 100 times more sensitive than metal foil strain gages, but they are also much more sensitive to changes in temperature. They are also fairly difficult to work with and are applied with special techniques and equipment. The comparatively large deflections required by metal-foil strain gages would induce a large misalignment error between the head of the sensor and the housing with a cantilever beam type of structure. The second method of measurement for this type of gage is to measure the tilt of the displaced head with fiber optics.\textsuperscript{53} This method depends on slight misalignment of the head. Success with this concept was mixed as the gage was sensitive to vibration at high frequencies, and this caused difficulty with the fiber-optic measurement systems. The difficulties were similar to those experienced by the first gage design in this investigation described in the previous chapter.

Previous investigations have demonstrated a need for some form of external damping.\textsuperscript{60} A novel method of using rubber sheets as a damping and isolation material has been investigated.\textsuperscript{68, 69} This method is impractical for very high-temperature applications as rubber sheets are incapable of the needed temperatures without significant changes in material properties. Viscous damping is less desirable for optical shear gages as it adds one more level of complexity to the problem. The gage is filled with a viscous fluid that has an index of refraction which changes with temperature. The search moved elsewhere on the path to a high-temperature shear sensor.

When examining past shear gages, one stood out from the rest as the best choice for a gage that would combine fiber optics and metal-foil strain gages into a single package. A wheel with very flexible spokes was chosen as the best candidate for the flexure of this new gage. This general type of skin friction sensor was recently employed for use in the X-43 flight test vehicle.\textsuperscript{5, 70} See Figure 51 for a drawing of this gage. This gage measures the shear in one direction only. When a force is applied to the head of the sensor the assembly rotates about the center point of the wheel, which is fixed to the housing. The “spokes” of the wheel are extremely thin beams which are about 100 times longer than they are thick. They are tall so that they are quite strong for out of the plane
force. This combination of strength in the directions where no motion is desired and weakness in the direction of motion is desirable. It used conventional metal-foil strain gages and an aluminum wheel flexure with strain gages mounted on the thin spokes. These gages required ~75 microstrains to work well. Conventional metal-foil strain gages and the associated high-temperature adhesives can withstand temperatures of only approximately 588 K (600°F). The actual sensing portion of the gage was up away from the flexure to provide a thermal barrier between the hot flow and the strain gages, as active cooling was not possible for this design. Thermal effects were, however, one of the prominent sources of error in the prior design as the flexures were constructed out of aluminum. It could only provide useful measurements for a few seconds in the NASA Langley Arc-Heated Scramjet Test Facility before temperature effects become evident. This design was for a very limited duration test and more exotic materials were not needed to meet the project requirements. This gage was also limited to measuring a single component of shear. The limitation of a single component of shear and maximum gage temperature of 588 K are two of the main limitations of this prior gage design.

A key feature of the wheel flexure design concept is that the flexure is mass balanced, and this makes the gage less susceptible to vibrations and linear accelerations. This mass was on the opposite side of the wheel as the sensing head. The fact that no viscous damping was required allowed this design to be installed in a flight vehicle and not be touched or disturbed for a long period of time.
Figure 52 shows the current two-component wheel flexure gage design. Instead of a single component of wall shear two components can now be simultaneously measured. The new design is a direct extension of the earlier, single-component gage design.

The thin spokes are the only parts of both gages that are designed to flex. The blue arrow on the sensing head in Figure 52 will cause motion in the wheel assembly which has blue arrows, and this is the first and primary component of the shear force measured. These arrows show the direction of motion of the flexure when a load is applied in this direction. The blue disk at the center of the wheel represents the center of rotation. This point is affixed to the housing of the gage and constrained against rotation or motion in any direction. The red arrow corresponds in the same manner, to the red arrows showing the motion of the upper wheel. The red disk on the top wheel is the center of motion for the top wheel. The red arrow corresponds to the second shear component. The upper wheel is affixed to the lower wheel by a square peg inserted in a square hole and then held together with a small bolt. This constrains the center pivot against rotation. Metal-foil strain gages are mounted on the thin spokes of both the upper and lower wheels. The fiber-optic distance measurement system views the sides of the counterweights. These sides are polished to an optical finish. The optical fibers are mounted to the housing of the gage and do not move. The counterweights move relative to the fibers.
4.2 Material Selection

The project goals require that the gage can survive up to 1073 K. The most critical element of design for this requirement is the selection of the material of the structure. There are several broad categories of possible materials: ceramics, iron-based super-alloys, stainless steels, and nickel based super-alloys. All of these can survive to the needed temperatures. One of the difficulties is choosing a material that can survive in harsh conditions and still be possible to work with on a reasonable budget. It is quite easy to select a material that will meet the temperature requirement but makes fabrication all but impossible. Resistance to oxidation is important as the counterweight sides must be polished to an optical finish. If the reflectors oxidize, the fiber-optic system is rendered useless. The coefficient of thermal expansion must also be minimized. If the chosen material is insensitive to temperature changes the final gage will have less error over a wider temperature range.

Ceramics were investigated as a possible choice of a construction material. They have the capability of surviving at very high-temperatures. The ability to survive isn’t the only requirement. It must also be possible to create the desired structure and have it survive handling and calibration. Ceramic materials tend to be difficult to work with and therefore expensive to machine. They are also quite brittle and cannot displace very much before failure occurs. One of the design goals for this gage was to use conventional metal-foil based strain gages. The strain required by these gages is quite large and would be difficult to generate with ceramics. Initial gage concepts focused around the diffusion bonding of alumina ceramic blocks together. Experiments with the diffusion bonding processes yielded results that were unacceptable for a high precision device such as a skin friction gage. This is not to say that ceramics should be dismissed in the future. They just didn’t appear to be a viable option for the current wall shear gage design.

The next set of materials examined was iron-based, super-alloys such as Invar. These alloys possess the desired temperature range and strength properties at lower temperatures. They do not resist oxidation well and also lose strength with increased temperature. All of the metals of this category examined required a protective coating in
order to resist oxidation at elevated temperatures. This coating would be difficult to apply to very thin structural members and could change their mechanical properties. The strength also drops off fairly rapidly with increasing temperature. The combination of poor oxidation resistance, low temperature strength, and difficulty of machining led to the dismissal of these materials for further consideration.

A number of common and exotic stainless steels were examined for the construction of the flexures of the gage. Many stainless steels have a high melting temperature, but lose considerable strength at higher temperatures. They also can change properties with thermal cycling. This is especially true in the 600-900 K target range for this gage. They do, however, have excellent resistance to oxidation. 316 Stainless was chosen for the housing of the gage and the optical ferrule holders. There is very little stress on the housing, when compared to the flexures, so the lower strength at high-temperature is less of an issue. There are also no polished surfaces as there are on the counterweights of the dual wheel flexure skin-friction gage, so oxidation resistance is less important. This makes any minor oxidation less of an issue. The savings in manufacturing time and cost were two of the primary drivers for this decision.

This left the main material selection question focused on the wheels of the gage. Finding reliable data on the various categories of materials presents its own challenge. Experimental data at elevated temperatures isn’t readily available. A summary of Young’s Modulus versus temperature is shown in Figure 53. This shows Haynes 230 and Hastaloy-X which represent nickel-based, super-alloys. The stainless steels are 310, 304, and HR-120. 316 Stainless behaves similarly to 304 but has better oxidation resistance. Iron based alloys are not shown as they were dismissed based on the requirement for oxidation resistance.
Figure 53: Young's Modulus of Various Materials vs. Temperature\textsuperscript{71,72}

Haynes\textsuperscript{®} 230\textsuperscript{®} is the material chosen for all of the internal components of the new gage. This is a high-temperature strength nickel-based super-alloy which contains 57% nickel, 22% chromium, 14% tungsten, 5% cobalt, 2% molybdenum, and 3% iron. It resists oxidation up to 1422 K and has a low coefficient of thermal expansion, ranging from 12.7 μm/m-K at 293 K to 16.1 μm/m-K at 1273 K. This material was chosen for its good thermal and mechanical properties as well as for its ability to be machined with conventional techniques. The X-43 flight test vehicle uses Haynes 230 as the primary material of the wing structure.\textsuperscript{73}

A test of the oxidation resistance was performed by Luna Innovations.\textsuperscript{74} A polished sample of Haynes 230 was heated to 1143K and the finish remained sufficiently oxidation free to provide a reflection for an optical fiber. Welding was also successfully accomplished in case an alternative to fastening method was needed in assembly.

Every attempt has been made in this design to achieve the highest operating temperature possible for the entire gage. The design condition of 1073 K is for the entire
gage assembly. Grafoil graphite gaskets were chosen to seal the various parts of the housing. These gaskets are usable in conditions up to 3273 K in a neutral atmosphere and provide a good compromise among sealing ability, ease of use, and durability.

The physical limitation on temperature capability of this gage is the maximum temperature that the gold-coated optical fibers can withstand. Their maximum operating temperature is 1073 K. This is approximately double the maximum temperature of the gage created by Smith as he used only brass, aluminum, and copper in his single component gage design.⁵
4.3 Analytic Design

Once the basic configuration had been decided upon, the next step was to develop a simple model of the structure. This analytic model allowed the design space to be narrowed before the detailed analysis of the design with a finite element analysis (FEA) package began. The displacement of the head was the primary design variable for the preliminary design. This analytic solution was developed to perform the initial design of the gage flexures and to compare with the initial ANSYS results. This comparison provides a reference for of the finite element results and a check of the mesh. The gage must move enough so that the fiber-optic displacement sensors can sense the movement and secondly there must be enough strain in the webs for the strain gages to measure.

An analytic solution to find the head displacement of the dual wheel flexure sensor was developed. Small displacements relative to the beam size were assumed. The material is presumed to be linear-elastic and isotropic. Essentially, one end of the beam is anchored to the sensor base and the other end is free as shown in Figure 54.

![Figure 54: Schematic of Structural Problem](image)

The following equations can be inferred from Figure 54.
\begin{align*}
\nu(0) &= 0 \\
\nu'(0) &= 0 \\
\nu''(L) &= \frac{M}{-EI} \\
\nu'''(L) &= \frac{F}{-EI} \\
e^{-E}v''''(L) &= 0 \\
v(x) &= \frac{A_1}{3} x^3 + \frac{A_2}{2} x^2 + A_4 x + A_6
\end{align*}

**Equation 16**

Solving this system of equations yields the following values for the four unknown coefficients.

\begin{align*}
A_6 &= 0 \\
A_4 &= 0 \\
A_2 &= -\frac{FL + M}{EI} \\
A_3 &= -\frac{F}{2EI}
\end{align*}

**Equation 17**

This gives the following expression for the end displacement, where \(X=L\), on the beam in question.

\[
v(L) = -\frac{FL^3}{6EI} - \frac{L^2(FL + M)}{2EI}
\]

**Equation 18**

The moment of inertia used for this computation is.

\[
I = \frac{1}{12} bh^3
\]

**Equation 19**

Results of this analysis for a Haynes 230 wheel flexure gage are shown in Figure 55. A simple Excel spreadsheet was used to create this figure using the analytic solution.
The spreadsheet makes the assumption that the load is applied at a distance from the end of the beam and that the displacement is the end displacement of each individual beam or web. 12.3 microns of end displacement was chosen. This value is the maximum possible for the geometry of the gage in order to provide as much strain on the webs as possible for the strain gages. This corresponds to four beams each of which are 6 mm tall which is 10 mm long and 140 µm thick. The height and width were chosen based on machining tolerances. This will provide enough displacement for very high wall shear resolution even after the strain gages are affixed to the beams. The fiber-optic measurement system requires a minimum of 100 nanometers of displacement and has a maximum limit of 200 microns. The strain gages and associated wires will reduce the movement of the flexure and this is the reason a fairly high displacement value was chosen for the non-instrumented flexures.

Figure 55: Analytic Results for Dual-Wheel Gage
4.4 Components and Design Features

A drawing of the detailed gage design, showing the internal components, is given in Figure 56. The wheel flexure spokes are 10 millimeters long from the center post of the gage to the inside of the outer ring. They are 140 microns thick and 6 millimeters deep. The upper wheel is affixed to the lower wheel with a square peg and a small screw. The lower wheel is captured top and bottom, between the two halves of the housing, with a similar arrangement. In this figure, the red objects are the flexure wheels. The light blue object is the sensing head. The round disk on the sensing head is the moving portion of the gage exposed to the flow. The green rectangular parts are the counterweights. The top counterweight balances the sensing head mass. The bottom counterweight balances the mass of the top wheel, top counterweight, and the sensing head.

Figure 56: Open View of Wall Shear Gage
The sides of these counterweights have an optical finish and serve as the reflectors for the fiber-optic system. This optical finish was created after the machining process and was the result of a mechanical polishing process. The polish was found to hold up to temperatures greater than 1073 K without difficulty on Haynes 230. One fiber looks at each side of each counterweight. There are a total of four fibers with two for each component of shear measured. This will reduce the temperature sensitivity of the gage as any thermal expansion or contraction will be reduced with the redundant measurement of the displacements.

The small gray dowels shown in Figure 56 are simply alignment pins which aid matching up the two halves of the housing during assembly. The housing itself consists of three parts, a removable cylindrical bossing which facilitates changing the sensing head and then two halves of the main gage housing. The right (starboard) housing also has a bossing on the bottom. This is for a connector which is for strain gages and can also accommodate a thermocouple, if needed. The dark gray thin parts are gaskets which go between the various housing parts to seal them.

The tubes, which house the fiber-optic ferrules, are shown in Figure 57. This drawing shows a cross sectional view of the internal components of the fiber-optic assemblies. The outer housing is a 5mm (0.2 in.) diameter tube of 316 stainless steel, with a 1.8 mm (0.07 in.) inner diameter. The inner diameter was chosen as fused silica fiber-optic ferrules are readily available with a 1.8 mm outer diameter. These ferrules have a 128 micron inner diameter which corresponds to the optical fiber outer diameter. The metal ferrule holders were used to allow the optical fiber assembly to be glued or welded to the housing as a separate unit. This modular assembly facilitates assembly in multiple steps. The optical fiber is affixed to the fused-silica ferrule and the stainless ferrule holder with a silver paste. This high-temperature bonding agent has a maximum working temperature of 1143 K. The optical fibers are gold coated for high-temperatures, and are fragile when compared with conventional polymer coated fibers. They can withstand a maximum temperature of 1073 K and are the limiting factor in the temperature capability of the gage.
Figure 57: Fiber-Optic Assembly

Figure 58 shows a drawing of the entire gage assembled. The circular head of the gage is 1.00 in. in outer diameter (25.4 mm). The housing is tightly wrapped around the internal components of the gage while allowing a small amount of room for changes. Future gage designs of this type could use a smaller housing or different shapes if the installation requires it. The footprint of the gage inside of the test section or in the flow is a one inch diameter circle. This is adjustable as the head and top bossing are easily replaceable while the rest of the gage stays assembled.

Figure 58: Drawing of Assembled Gage (Dimensions in Inches)
The following figures show mechanical drawings of the gage flexures. Figure 59 shows details of the upper wheel. Figure 60 shows a drawing of the upper counterweight. Figure 61 is a machine drawing of the lower wheel. Figure 62 shows a drawing of the lower counterweight. Figure 63 shows the sensing head and Figure 64 shows the top bossing for the floating head. Figure 65 and Figure 66 show the right and left half of the housing respectively. These drawings are not completely dimensioned for production purposes but are shown to give the general size and scale of the gage parts.

Figure 59: Upper Wheel Machine Drawing (Dimensions in Inches)

Figure 60: Upper Counterweight Machine Drawing (Dimensions in Inches)
Figure 61: Lower Wheel Machine Drawing (Dimensions in Inches)

Figure 62: Lower Counterweight Machine Drawing (Dimensions in Inches)
Figure 63: Sensing Head Machine Drawing (Dimensions in Inches)

Figure 64: Top Bossing Machine Drawing (Dimensions in Inches)
Figure 65: Right Housing Machine Drawing (Dimensions in Inches)

Figure 66: Left Housing Machine Drawing (Dimensions in Inches)
4.5 Finite Element Analysis

The finite element method (FEM) software package ANSYS was used to study various facets of the gage. This proved essential to the design and verification of this new gage flexure concept.

4.5.1 Structural Analysis

A model internal structure was created to evaluate the displacement of the sensor to a given load. The model used for the wheels is shown in Figure 67. The counterweights are omitted in this figure for clarity. A sample load and the corresponding motion of the bottom wheel are shown. When the load is applied in this direction, only the bottom wheel turns and only the main component of shear will be measured by the gage. This sample load illustrates one of the conditions shown in the results.

![Figure 67: ANSYS Mesh of Internal Gage Structure](image)

The final mesh used in the ANSYS static analysis of the structure is shown in Figure 68. The actual counterweights aren’t modeled in detail, but they are included as simple, equivalent masses. The meshes used contain the maximum number of elements possible along the thin webs, which are the flexures and where almost all motion will
take place. The mesh is coarser in the more rigid sections. This was done to speed the computation as well as to stay within the maximum number of nodes allowed in our version of this software package. In this model 64,721 elements were used. This number of elements was the maximum possible with the student version of ANSYS available for use. Therefore a detailed convergence study could not be undertaken. Simple verification cases with simple analytic models were instead used. In ANSYS the element is a Solid 92. This is a ten-node tetrahedron with quadratic displacement behavior. Each of the ten nodes has three degrees of freedom.

Figure 68: Static Analysis Mesh (64,721 Elements)

A check of the mesh was run to validate the ANSYS model. In this case the top wheel was held fixed by the support connecting it to the bottom wheel. The result of this sample case was a 12.6 micron displacement at the end of each beam. This compares favorably to the results predicted by Equation 18 of 12.3 microns. The very fine finite element mesh is slightly more flexible than the rigid beam theory model; thus, the result compares satisfactorily with theory and the mesh is considered reasonable.

Some of the finite element analysis results are shown in Figure 69 and Figure 70. These figures show the effect of a 250 Pascal shear load, oriented so that only motion in the lower wheel is produced. This load was applied by selecting all 401 of the nodes on the sensing head, dividing the total load by the number of nodes and applying this nodal
load. This simulates a shear load distributed over the entire head. The center pivots of the bottom load are help rigid by restricting all degrees of motion. These parts are clamped in the actual gage and these boundary conditions are a good approximation. It should be noted that the displacements are not to scale. They are exaggerated for clarity in the figures. This is true for all of the figures generated with ANSYS.

The displacement results of the static analysis are shown in Figure 69. The maximum motion appears at the edge of the top wheel because it is at the far edge of the center of rotation about the bottom hub. Figure 70 shows the strain in each of the flexures. The peak strain in each of the bottom spokes is 55.3 microstrains, and this occurs near the fixed pivot of the assembly. This piece of information is important as the strain gages must be located as close to the end of the beams as practicable. The fiber-optic data is dependent on both axes as displacements on both wheels occur when you excite one or the other motions. The strain gages are less coupled. If a force is applied in the X-direction only there is no strain in the upper wheel. This means that whenever there is a change in strain of the upper wheel, it is caused by a force in the Y-direction. This is illustrated in Figure 71 and Figure 72.
Using a model which better represents the actual configuration allows the displacements seen by the optical fibers during the static calibration to be calculated. The actual shapes and sizes of the counterweights were modeled instead of an equivalent cube. The balancing becomes important when the model and spectral analysis is performed but is less critical for accurate static displacement results. Figure 73 shows the new mesh used for this analysis. It has fewer elements than the first mesh due to a change in the shape. There were fewer shared nodes, so the ANSYS limit of 128,000 nodes was reached with fewer elements.
Figure 73: Mesh with Actual Counterweights-55,204 Elements

Figure 74: Total Displacements (meters) (6g-X)

Figure 74 shows the total displacements, as a contour plot, when a 6 gram load is applied to the head of the gage as a point load in the X-direction. The coordinate system, with this ANSYS model, was created to match the coordinate system created for the gage. The Z-axis goes up from the X-Y plane and is parallel to the stem of the sensing head. Figure 75 shows a top view of the sensor. The mesh is shown in the original position and the colored portion of the figure is after the load has been applied. The approximate displacements that the fibers should measure are given. This says that for a 6g load in the X-direction the top wheel will see a change in gap of 30.6 µm and the lower wheel will see a change in gap of 23.6 µm. This again demonstrates that the motion of the wheels is dependent on one and another. The strain gages see a different picture as when the load is in the X-direction there is no strain on the webs of the upper wheel. All of the displacement is due to the fact that the reflector is a certain distance from the center of rotation about the lower wheel, and it sweeps through an arc. The result of this is that the gage results must be put through a calibration matrix, and both the upper and lower wheel data must be known to resolve the forces from the fiber-optic gap data.
Figure 75: Top View with Displacements Measure by Fibers (6g Load on X-Axis)

Figure 76 shows the displacements in the X-direction and Figure 77 shows the displacements in the Y-direction. These figures are useful to view to get a notion of how much motion a load in the X-direction creates in each direction for the internal components of this gage. This will be the displacement measured by the fiber-optic system.
Figure 76: X-Displacement (meters) (6g Load on X-Axis)

Figure 77: Y-Displacement (meters) (6g Load on X-Axis)

Figure 78 shows the total displacements of the internal gage components for a 6g load in the positive Y-direction. The load vector, applied in ANSYS, is shown in red and the nodal reaction forces are shown in magenta. The center portion of the lower wheel is
constrained against all motion. This is the same condition used for the loads in the direction of the positive X-axis. All of the numbers in the figures are shown in meters of displacement from their original position. Figure 79 shows the predicted displacements that the fiber-optic system measures.

Figure 78: Total Displacements (meters) (6g Load on Y-Axis)

Figure 79: Top View of Total Displacements (meters) (6g Load in Y-Direction)
4.5.2 Modal Analysis

In addition to the static analysis performed in ANSYS, a modal analysis was performed. This information can be used to confirm the results of actual vibration tests of the gage and to make sure that the flow testing environment doesn’t have modes which coincide with those of the gage.

The analysis used the same Solid 92 element as the static analysis and the same mesh. When using ANSYS the analysis options were set to solve for the first five mode shapes of the gage between 0 and 10,000 Hertz. The elemental results were calculated, and the default solution method was used. The mode shapes were examined, and there are at least five nodes for every half wavelength for the modes in question. The results of this analysis are shown in Table 10.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (Hertz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>22</td>
</tr>
<tr>
<td>2</td>
<td>69</td>
</tr>
<tr>
<td>3</td>
<td>442</td>
</tr>
<tr>
<td>4</td>
<td>506</td>
</tr>
<tr>
<td>5</td>
<td>1138</td>
</tr>
</tbody>
</table>

Table 10: Modal Analysis Results

In order to ascertain the response of the structure throughout a specified vibration regime to check survival through the possible wind tunnel and flight test regimes the information from the modal analysis was used in a spectrum analysis.

4.5.3 Spectrum Analysis

A spectrum analysis was performed to estimate the effect of a known vibration range on the gage. This is needed to assess the survivability of the gage in the NASA flight test B-curve shown in Figure 19. This vibration requirement is to survive from 15-2000 Hertz with RMS acceleration 12 times the acceleration due to gravity. It is
desirable to be able to accurately predict the displacements of the structure due to any vibration loads.

The first step in a spectrum analysis is a modal analysis. The mode shapes and elemental results must be calculated in ANSYS from the modal analysis. Once this analysis is complete the finish command is used. The next step is selecting spectrum analysis in the new analysis portion of the solution menu. The analysis options should be set to solve for a P.S.D. spectrum with the same number of modes as the modal analysis and to expand these mode shapes as well as calculating the elemental results. A SOLID 92 element was used for the spectral analysis as it remains unchanged after the modal analysis. For this random vibration analysis the structure is assumed to be linear elastic and isotropic. This means that the results should scale linearly with the load; therefore, only one load case is sufficient to provide an estimate of the structure’s response.

The P.S.D. settings are changed in the load step option menu to show a spectrum with units of g²/Hz and a GVALUE of 386.4. A table defining the range that the random vibration load will be applied over is defined next, and this is shown in Figure 80. This figure shows the flexure assembly constrained about the center post of the lower wheel. That point will be held fixed in relation to the housing. For a vibration analysis all loads transmitted through the vehicle or wind tunnel walls will the pass through that point. The vibrations due to the flow would be transmitted through the head of the sensor. The focus of this investigation is the vibration through the walls as these are easily quantified.
Figure 80: Description of P.S.D. Base Excitation Problem

The next step in the solution process is to specify the P.S.D. table which defines the frequency range and accelerations that the gage will experience during the analysis. The table used to evaluate the flexure has a similar shape to the NASA B-curve but a much lower magnitude. This is so that the finite element model can be tested against experiment with the actual gage without risking damage to the internal gage components. The P.S.D. range used is shown in Figure 81. The peak accelerations are only about 1/3 of a “G” instead of the roughly 12 “G’s” on the NASA curve.

Figure 81: P.S.D. Range for ANSYS Analysis
Once the P.S.D. curve is defined, the damping of the structure must be specified. A 2% damping value was used in the initial computational analysis as it was found to be a typical value for metal structures. This is the key quantity, which must be determined experimentally, for an accurate computational analysis.

The nodes, attached to the center post of the lower wheel, have the base P.S.D. excitation applied. The first direction the load was applied is the X direction in the coordinate system shown. The participation factors are solved for using the base excitation and the default partial correlation form. Next in line is to set all of the calculation controls to calculate the displacement, velocity, and acceleration solution results relative to the base of the flexure. The random vibration solution is then obtained. The following step is to combine all of the modes using the mode combination command and then to solve this final step in the spectral analysis.

The results of the spectral analysis give the modal results in the first load step. The unit static solutions are in the second load step. The third through fifth load steps show the 1-σ displacements, velocities, and accelerations, respectively. The first load case examined was exciting the structure in the X direction with 2% damping. This 2% damping value is the nodal damping value and is assumed constant. The value of 2% damping was found in other studies and was assumed to be typical for metal structures. This can be improved with actual experimental test data but this is the best approximation for a structure with unknown properties.

Figure 82 shows the 1-σ displacement results for the structure with the load applied from Figure 81. The 1-σ displacement means that the displacement will be at or below this value 68% of the time. The 2-σ displacements are double the 1-σ displacements. The actual displacements should be at or below the 2-σ values 95% of the time. The displacements shown in the 1-σ results are only 183 nanometers and 366 nanometers for the 2-σ displacements. This means that the gage will not hit any mechanical stops, except for the head to housing gap and shaking it at this low level should be safe for experimental tests. This test data could be used to infer the damping ratio of the structure and refine the FEA results.

Figure 83 shows the nodal stress results for the same analysis. The maximum stress is experienced in the bottom webs. The maximum value for the 1-σ stress is 8918
Pascal. The lower wheel will be easier to excite than the upper wheel. This effect could be mitigated if the webs on the lower wheel were made stiffer than the ones on the upper wheel or if the upper wheel were lightened and the correspondingly large counter-weight could be reduced in mass. The lightening holes are omitted from the finite element model and should help reduce the mass moment of inertia about the center point of the actual gage.

Figure 82: 1-σ Displacements Low Curve-X $\zeta=2\%$

Figure 83: 1-σ Stress Low Curve-X $\zeta=2\%$

Figure 84: 1-σ Strain Low Curve-X $\zeta=2\%$

Figure 85, Figure 86, and Figure 87 show the displacement, stress, and strain results of the flexure when the excitation is applied in a direction perpendicular to the X excitation and is known as the Y excitation. The results show the same values as those calculated for the X axis calculations. This makes sense as the main vibration modes are rotational for the flexure. If there is a wheel with a pivot and a mass and one grasps the
pivot, the same result should be obtained by shaking the axle toward the body as from one side to another because of the symmetry of the problem. This calculation confirms this intuitive result numerically for an object that doesn’t appear symmetric to the eye but whose moments of inertia of the various components and counter balances are equal.

**Figure 85: 1-σ Displacements Low Curve ζ=2%-Y**

**Figure 86: 1-σ Stress Low Curve ζ=2%-Y**

**Figure 87: 1-σ Strain Low Curve ζ=2%-Y**
4.5.4 Thermal Analysis

A transient thermal analysis was performed with ANSYS to examine the effect of a heat flux on the flexure. A heat flux of 100,000 Watts/m² on the exposed gage head was chosen as it is representative of the heat flux experienced inside of a supersonic combustion ramjet.

The Solid 87 thermal element was chosen for the thermal-only portion of the analysis. This is a tetrahedron with ten nodes and is similar to the Solid 92 structural element. It is only applicable to thermal analysis. The mesh used in the structural analysis is shown in Figure 88. A preliminary thermal analysis was run with this mesh. The mesh needed to be changed from the structural analysis as a simple mesh refinement wasn’t possible. The ANSYS version available for student use has a limit if 128,000 nodes. This artificially limits the level of mesh refinement available for this study. Therefore, the meshes shown in Figure 89, and Figure 90 were created to concentrate the largest number of elements near the head of the sensor. Figure 90 shows the final mesh used in the thermal analysis. This mesh was chosen to have a large number of elements from the head and top wheel which heat quicker than the rest of the gage. This is where the largest heat flux should come from. This mesh used 64,021 elements and was as fine as could be reasonably approached with this ANSYS version. Temperature dependent material properties were used.

Figure 88: Mesh 1 from Structural Analysis
Figure 89: Mesh 2 for Thermal Analysis
The boundary conditions used for this analysis consist of starting the sensor at 300 K with all sides of the sensor insulated (ANSYS default boundary conditions). A uniform heat flux of 100,000 Watts/m² is applied to the surface of the head. The duration of the computation is set by the user. Ten minutes (600 seconds) of exposure to this heat flux heated the head of the sensor to approximately 1500 K. See Figure 91. This is considered a good practical limit as the Haynes 230 material melts at approximately 1600
Temperature dependent material properties were used in this model. A time step of 10 seconds was used as it was determined to be a good compromise between computational time and being able to plot smooth temperature curves of specific elements versus time. Shorter time steps were attempted but the additional information wasn’t thought to be worth the additional computational expense as this would require running on a mainframe instead of a PC.

Figure 92 shows the choices of nodes selected for detailed study for the time dependent thermal analysis. Figure 93 shows the results of various internal gage components. Figure 93 has two lines labeled. The first is where the top wheel flexures will experience a 10% drop in strength. This marks where the flexures will begin to be significantly affected by the heat flux. This is predicted to occur after an exposure of 200 seconds. The second line shows where the head of the gage would begin to melt. This occurs after about 550 seconds of exposure. It should be noted that the top wheel heats up quicker than the lower wheel as it is closer to the heat source and the temperature at the junction between the wheels only rises 100 K. The low thermal expansion characteristics of the Haynes 230 should limit the thermal effects. If a calibration system can be arranged for these high heat fluxes the behavior can then be quantified. Once the response is known it will be simply reduced out of the final data.
Figure 92: Node Location for Time Dependent Data

Figure 93: Temperature of Various Internal Haynes 230 Gage Components vs. Time
Figure 94 shows the same computation in Figure 93 but with a gage made of 310 SS instead of Haynes 230. A 10% loss in web strength is noted at 150 seconds, a full 25% decrease in usable time when exposed to the same high heat flux value. The melting temperature limit is reached at 430 seconds for the stainless steel instead of 550 for the Haynes. This corresponds to a 22% reduction in duration when gage survivability is factored in. This check was performed to confirm that Haynes 230 was the best choice and to characterize another candidate material if fabrication with the Haynes proved impractical.

![Graph showing temperature of various internal 310 stainless gage components vs. time]

**Figure 94: Temperature of Various Internal 310 Stainless Gage Components vs. Time**

Due to the complex shape involved and the difficulty of verifying the grid independence of this solution, a model problem was chosen to examine the effects of mesh density. The problem chosen is a circular cylinder 14 mm in diameter and 350 mm. This size was chosen as it is similar in diameter to the floating head, and sufficiently long so that the unheated end of the cylinder experiences no temperature change during the computation. A heat flux of 100,000 Watts/m² one the top and test duration of 600 seconds with 10 second time-steps was chosen to approximate the conditions used for the
Haynes 230 gage. The rest of the bar is insulated and the initial temperature is 300 K. The ANSYS Solid 87 element was again chosen.

Material properties with constant values were used for the validation as the analytic solution for the temperature distribution in a semi-infinite solid assumes constant properties, and we wish to compare our numerical results against the available exact solution for that case. The length of the solid was increased until no temperature change was noted in the insulated end which is required for the semi-infinite solid approximation to be valid. A schematic of the model problem is shown below in Figure 95.

![Figure 95: Schematic of Model Heat Flux Problem](image)

An analytic solution for a semi-infinite solid was found in Ref. [75]. The boundary conditions and heat flux are assumed to be the same as those used in the ANSYS model. The resulting expressions for the temperature as a function of distance from the top are shown below in Equation 20 and Equation 21. The resulting surface temperature is 668.73 K after 600 seconds of exposure to the heat flux.

\[ \alpha = \frac{k}{\rho C_p} \]

Equation 20
\[ T(x,t) = T_i + 2q \left( \frac{\alpha t}{\pi} \right)^{1/2} e^{-\frac{x^2}{\alpha t}} - \frac{q_x}{k} \text{Erfc} \left( \frac{x}{2\sqrt{\alpha t}} \right) \]

Equation 21

The first mesh used in the thermal analysis is shown in Figure 96; it has 194 elements. This mesh corresponds to the approximate mesh density used on the lower wheel rim and on the counterweights. The second mesh is shown in Figure 97 and consists of 488 elements. This corresponds to the final mesh density used on the flexures. The third mesh is shown by Figure 98, and it has 2,444 elements. The third mesh is similar in density to that used on the gage head. The fourth verification mesh shown in Figure 99 has 56,358 elements.
Table 11 shows a summary of the thermal analysis verification results. The error for each case is quite low. This demonstrates that even a coarse mesh will provide a good approximation for simple geometries. All of the finer meshes did better than the coarsest mesh, but there was only a very slight change in the error when the number of elements was increased from mesh 2 to mesh 3. Since all of the errors are less than 1% the results are considered acceptable for the purposes of this engineering investigation. Figure 100 shows the results of the thermal analysis with the finest mesh used at the 600 second final time value. This shows that the results from the lowest mesh density used here will provide sufficient resolution for the thermal analysis of the gage.

<table>
<thead>
<tr>
<th>Mesh</th>
<th>Elements</th>
<th>( T_s ) (Kelvin)</th>
<th>( \Delta T ) (K)</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>194</td>
<td>666.96</td>
<td>-1.77</td>
<td>0.26%</td>
</tr>
<tr>
<td>2</td>
<td>488</td>
<td>668.03</td>
<td>-0.70</td>
<td>0.10%</td>
</tr>
<tr>
<td>3</td>
<td>2,444</td>
<td>668.00</td>
<td>-0.73</td>
<td>0.11%</td>
</tr>
<tr>
<td>4</td>
<td>56,358</td>
<td>667.97</td>
<td>-0.76</td>
<td>0.11%</td>
</tr>
</tbody>
</table>

**Table 11: Thermal Analysis Verification Results**
Figure 100: Thermal Verification Results for Mesh 4
5 Dual Wheel Gage Construction and Static Calibration

This section details the construction and assembly methods used to bring the components into reality. More detail will be shown on this process than in other investigations. When creating a delicate device the details can make or break the project. Finally, static calibration results will be presented.

5.1 Gage Construction

The gage was constructed at Damon Company in Salem, Virginia. Conventional machining techniques were used, wherever possible, in the manufacturing process. Haynes 230 is a difficult material to work with, and many of the more critical or difficult machining operations were performed using Electron Discharge Machining (EDM).

Conventional machining techniques worked reasonably well with standard high-speed steel cutting tools. Very low tool speeds and special tapping fluids were required for satisfactory results. The Haynes 230 alloy tended to work harden rapidly and heat up when machined. The use of powdered metal bits, end mills, and taps significantly reduced the machining problems. This was discovered later in the manufacturing process.

EDM machining uses an electrical charged passed from an electrode to the part being created. High tolerances are possible. The thin webs of the flexures were created using a wire EDM machine. The process consisted of roughing out the shape of the wheel on a lathe and then drilling pilot holes for the wire to pass through the material. The wire was then fed through the hole and an electrical charge is passed through the wire. The wire moved continuously and the part was moved slowly, as the desired material was eroded away. The beams are 140 microns (0.0055 in.) in thickness with a machining tolerance of approximately ± 5 microns.

Another technique that was used in the construction of the gage was a down-burn EDM machine to cut the clearances out of the housing. This allows extremely tight clearances between the moving parts of the gage and the fixed housing. A wire EDM could not be used here as it requires the shape to be an extruded plane. The housing required the largest amount of down-burning as the clearances between all of the
important features were tight. An emphasis on small size was placed on the housing of the gage at the expense of machining ease.

If the gage housing was approximately 20% larger internally, then a significant savings in time and expense could be realized. The most complicated mechanical portions of the gage, the wheels, were actually some of the easiest parts to create. They were designed for wire EDM machining from the beginning of the project.

Figure 101 and Figure 102 show the upper and lower wheels, respectively. The upper wheel has a square peg which fits into a square hole on the bottom wheel. The upper wheel peg is threaded and a 316 stainless steel screw affixes the two wheels to each other. This arrangement was the most difficult feature of the wheels. The holes in the rim of the lower wheel are simply to reduce the total mass of the flexure assembly to reduce the out-of-plane loads on the webs.

Two webs on each wheel are equipped with the strain gages, on each side of the web for a total of four gages per wheel. This creates a full bridge and gives the maximum sensitivity possible. The gages are mounted near the hub of the wheel where the maximum strain values will occur. Figure 72 shows the peak values of strain occurring near the hub in the computational model of the flexures, which was used to design the gage. Figure 101 shows the completed upper wheel with strain gages affixed to the two webs nearest the scale and Figure 102 shows the same for the lower wheel.

The strain gages used for this gage are Micron Instruments EA-06-031CE-350. Each gage has a resistance of 350 Ohms. They were attached to the Haynes 230 using
the methods suggested by Micron Instruments. The M-Bond 600 adhesive was used to attach both the strain gages and the CPF-38C bondable terminal strips to the wheels. This adhesive was cured for several hours in a convection oven to assure a uniform bond. Clamps were placed on each side of the webs to produce a uniform pressure on each gage as it cured and to reduce the possibilities of bubbles or problems with the bond. The upper and lower wheels are shown in Figure 103, next to each other, for a sense of scale and in Figure 104 assembled together.

Figure 103: Lower and Upper Wheels with Strain Gages and Scale (Inches)

Figure 104: Both Wheels with Strain Gages (Scale in Inches)

Figure 105 shows the upper wheel with all of the wiring and wire strain relief completed, and Figure 106 shows the same for the bottom wheel. The copper wires going from the strain gages to the solder terminals are 36 gage wire with a thin enamel type of insulation. Every effort was made to reduce the possibility of the wires interfering with the movement of the flexures. 38 gage wire with Teflon insulation was used to go from the bondable terminals to the connector on the housing.

Figure 105: Upper Wheel with Strain Gages Wired

Figure 106: Lower Wheel with Strain Gages Wired
5.2 Gage Assembly

All of the gage components are displayed in Figure 107 with a scale. The wheels, counterweights, housing, gaskets, top bossing, connectors, and all of the associated hardware are shown in this figure. A brass spacer ring was created so that the strain gage connector would require fewer turns to be tight and provide additional space for the wires. The 0.38mm (0.015 in.) thick gaskets can be seen in the picture for the housing and the top bossing. The gaskets for the strain gage spacer are not shown, but they were made from the same material.

![Figure 107: All Gage Parts (Scale 2.5 in. Long)](image)

The housing had an additional pocket machined in it to provide clearance for the strain gage wires. The wires exit the solder pad for the lower wheel vertically and the housing, as originally designed, did not provide enough clearance for the wires. The new pocket is a semicircle and was cut with a 0.25 in. end-mill. The pocket was then coating with Micro Measurements Group M-Coat A so that the wires will not short out on the housing if the insulation wears off due to use or vibration.

The first step in the assembly process is affixing the counterweights to the lower and then upper wheels with the 316 stainless steel #4-40 Allen-head, cap-screws, while the wheels are held by their rims in a vice. These screws are locked in place with a
medium strength thread-locker to avoid them loosening during testing or handling. The next step is the attachment of the upper wheel to the lower wheel. This very delicate operation involved inserting the square center-hub of the upper wheel into the corresponding square slot in the lower wheel. Extreme care must be taken to avoid putting a torque on either wheel. A 316 stainless #2-56 socket-head, cap-screw was then used to pull the upper and lower wheels together. Thread-locker was used on this screw as well. This arrangement provides a very simple and tight joint between the two wheels. The wheel assembly is shown in Figure 108 below.

![Figure 108: Housing Ready to Accept Wheels](image)

![Figure 109: Wheels in Housing](image)

The next stage of the assembly is to insert the wheel assembly into the starboard half of the housing. This is shown in Figure 109. The 36-gage wires for the upper wheel strain gages were routed between the webs of the lower wheel. Once below the lower wheel, they follow the wire clearance notch in the housing along with the lower wheel wires until reaching the connector. The exit of the wires from the housing can be seen in Figure 110.
Figure 111 shows the strain gage wires attached to the connector. The small 38-gage wire presented a number of difficulties. The Teflon insulation was difficult to strip without breaking the wire and the bare wire, and would break easily with repeated flexing. The connectors used are the Lemo EGG.1K.310.CLL receptacle and Lemo FGG.1K.310.CLAC60 connector. These 10-pin connectors are sealed from the environment. Eight of the pins were used for the strain gages: four pins for the upper wheel and four pins for the lower wheel. The wires go clockwise from a little circle with the following color pattern: red, white, black, white, green, for the top wheel and then a repeat of the same pattern for the lower wheel.

The final step of assembling the basic gage is the attachment of the sensing head to the upper wheel. The head used a 316 stainless steel set-screw whose head is specially sized to fit through an access hole in the starboard housing half. Due to some play in the fit of the sensing head to the upper wheel this operation was repeated several times in order to make the top bossing line up with the head. A perfect fit was not obtained and the head diameter was reduced by 152 µm (0.006 in.) in order to provide a sufficient clearance between the head and top bossing. Figure 112 shows the attaching of the head to the top wheel and shows how the screw is accessed. A #10-32 set-screw with Teflon tape was used to plug this hole once the head was attached and seal the housing.
The optical fibers were then attached to the sensor. Gold coated optical-fiber was used as it has superior properties at elevated temperatures versus the standard polyamide coated fibers. The gold fiber was reinforced for handling purposes for all of the low temperature tests. The higher temperature tests require special care as the fiber is quite fragile without protective material.

The completed gage is shown in Figure 113 below. This figure shows the top bossing in place around the sensing head and the optical fiber assemblies installed. The corresponding channel and axis are shown for each of the fibers in this figure.
The optical-fiber assembly was set in the housing with a set-screw to hold it in place. A conical point #4-40 thread set-screw was used as it would cause less movement than a conventional set-screw while tightening. Small notches were machined into the 316 stainless steel ferrule holders. These notches allowed the set screws to be used without fear of gouging the ferrule holder. Oxidation on the ferrule holders was removed with 3M Scotch-Brite pads. This thin layer of oxidation was caused by the firing process used to create the ferrule holder/ferrule/fiber assembly. The ideal location for the end of each fiber is about 200 microns from the end of the reflector for this gage. The actual distances from the reflector to each fiber are shown in Table 12 below.

<table>
<thead>
<tr>
<th>Gap (µm)</th>
<th>Channel-1</th>
<th>Channel-2</th>
<th>Channel-3</th>
<th>Channel-4</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>219</td>
<td>226</td>
<td>222</td>
<td>246</td>
</tr>
</tbody>
</table>

Table 12: Fiber and Reflector Gap for Each Channel

5.3 Balancing and Calibration

Carefully balancing the gage greatly reduces its sensitivity to vibration by providing the same moment of inertia about the center of rotation of each wheel. In order to provide accurate wall shear data and to confirm that the flexures are working in the desired manner, the gage requires calibration of both the strain gages and fiber-optic displacement measuring systems.

5.3.1 Balancing Procedure

The first step in the balancing process involved creating a fixture to hold the flexure assembly, since that contains all the moveable parts. This assembly is shown in Figure 114. This simple fixture is made of 6061-T6 aluminum. A simple aluminum clamp fastens the wheels to the fixture using the square end of the lower wheel’s center post. The upper wheel can be constrained in a similar manner by itself. It only has flats on one side of its center post, so a screw is passed through the open slot on the fixture to hold it in place.
The balancing simply involves using the strain gages to see if mass needs to be added or removed from the counterweight of each wheel. The upper wheel is hung so that gravity acts equally on the sensing head and the upper counterweight, and then the assembly is rotated 90°. Material is then removed (or added) from the counterweight to minimize the change in strain measured by the strain gages during the rotation of the assembly. This procedure must be completed before the lower wheel can be balanced as any change in mass to the upper wheel will require more or less of the bottom counterweight to be removed.

The same procedure is used on the lower wheel when the balanced upper wheel is affixed. See Figure 108. This procedure only needs to be done once to each set of wheels and counterbalances. If a change is made which has an effect on the mass of the system everything must be rebalanced. When another head size is used it must be exactly the same mass as the head it replaces, or the unit must be rebalanced.
Figure 115: Balanced Wheels in Rig

The experimental setup is shown in Figure 116 below. The strain gages of each wheel were attached to a Micro Measurements Model 2310 signal conditioning amplifiers. These units provide the excitation used to provide an electrical power to the gages. This setup used a 5 Volt Direct Current excitation for the balancing. The gain of the amplifier was between 1000 and 11,000. The value used depended on how close the wheels were to being balanced. The strain gage bridge completion feature of the amplifier was used as it allows the strain gages to be balanced to zero. A Fluke 1608169 digital multimeter was used to read the voltage. A recording device was not required for this procedure, but a log of the material removed and the corresponding change in voltage was kept.
5.3.2 Calibration Procedure

This static calibration involves hanging known masses from the head of the sensor and measuring the resulting displacements and strains. Figure 117 shows a schematic of the calibration process. Essentially a paper cup of a known mass is hung from a string. This is affixed to the floating head and this creates a force on the head. A seven point calibration from 0-6g, in 1g increments. Repetition of the calibration 5 times yielded data on the basic errors in the gage and measurement systems.
The actual X-axis calibration arrangement is shown Figure 118. This shows the paper cup, string, and tape used to hole the weights to the head of the sensor. The mass of the cup assembly is 0.5017 grams. The setup for the Y-axis calibration is shown in Figure 119. The cup assembly weighed 0.5015 grams for this experiment. Figure 120 shows the setup for the 45° calibration case and the cup weighed in at 0.5016g. The zero point is taken with this mass as in an actual wind tunnel experiment one only looks at the change in gap due to the effect of the shear force. The absolute positions were recorded but are not as important as the change in gap to determine the slopes of the response.

The calibration results are shown in Figure 121 for the fiber-optic measurements. The Fast Fiber Scan (FFS) system was used as it has a higher sample rate than the Fiber-
Pro USB and can accept four simultaneous channels of data. The wall shear gage requires two channels for each axis of shear measured. The throughput of this system is approximately 960 Hertz compared with about 100 Hertz for the FiberPro USB even though the spectrometers sample at the same rate. The slower USB connection just uses less of the available information and records fewer data points. The masses were applied in the X-Axis direction of the gage. This figure shows five calibrations superimposed over each other. The calibrations proved linear and repeatable in all cases. A linear fit was applied to each channel to determine the corresponding slope. Figure 122 shows the Y-Axis data for all four channels of the FFS system.

The strain gage output was recorded at the same time as the FFS output. The strain gage results are shown in Figure 123 and Figure 124 for the X and Y-Axes, respectively. There are only two separate plots with five runs as the strain gage output comes from one set of gages on the upper wheel and one set of gages on the lower wheels. For reference, the resolution of Smith’s single component gage with an aluminum flexure, was 0.165 Volts/Gram.\(^5\) The output of the lower wheel of the current gage is 0.156 V/g, and the upper wheel is 0.191 V/g. The wheels are a coupled system and the components of shear cannot be separated without the use of both upper wheel and lower wheel data at the same point in time.
Figure 121: FFS X-Axis Calibration (5 Runs)

Figure 122: FFS Y-Axis Calibration (5 Runs)
Figure 123: Strain Gage X-Axis Calibration (5 Runs)

Figure 124: Strain Gage Y-Axis Calibration (5 Runs)
A comparison of ANSYS displacement predictions with the fiber-optic displacement results for a 6 gram load is shown in Table 13. This shows that there is agreement in the general motion of the wheels but that the magnitude of that motion is off by a significant amount. The last row shows the ratio of upper and lower wheel displacements and the agreement for the X-axis load is quite good. This could mean that the load for the Y-axis calibration may not have been perfectly aligned or the model isn’t as flexible as it should have been. The joint between the two wheels was modeled in ANSYS as perfect with no motion allowed. There are also likely small differences in the geometry of the actual gage and that assumed for the calculations.

<table>
<thead>
<tr>
<th></th>
<th>X-ANSYS</th>
<th>X-Calibration</th>
<th>Error</th>
<th>Y-ANSYS</th>
<th>Y-Calibration</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper Wheel</td>
<td>30.6</td>
<td>41.3</td>
<td>-26%</td>
<td>38.3</td>
<td>73.9</td>
<td>-48%</td>
</tr>
<tr>
<td>Lower Wheel</td>
<td>23.6</td>
<td>32.5</td>
<td>-27%</td>
<td>15.5</td>
<td>23.7</td>
<td>-35%</td>
</tr>
<tr>
<td>Upper/Lower Ratio</td>
<td>1.30</td>
<td>1.27</td>
<td>-2%</td>
<td>2.47</td>
<td>.3.12</td>
<td>-21%</td>
</tr>
</tbody>
</table>

Table 13: 6g Fiber-Optic Displacement Calibration Data Compared with ANSYS Displacement Results ($\mu$m)

Table 14 displays the slopes predicted by the ANSYS static model and compared with the calibration results for the fiber-optic system. This shows that the prediction for motion in the X-direction is significantly more accurate than that for motion in the Y-direction.

<table>
<thead>
<tr>
<th></th>
<th>X-ANSYS</th>
<th>X-Calibration</th>
<th>Error</th>
<th>Y-ANSYS</th>
<th>Y-Calibration</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper Wheel</td>
<td>5.10</td>
<td>5.49</td>
<td>-7%</td>
<td>6.38</td>
<td>12.2</td>
<td>-48%</td>
</tr>
<tr>
<td>Lower Wheel</td>
<td>3.93</td>
<td>3.95</td>
<td>-0.5%</td>
<td>2.58</td>
<td>6.99</td>
<td>-63%</td>
</tr>
<tr>
<td>Slope Ratio</td>
<td>1.30</td>
<td>1.39</td>
<td>-6%</td>
<td>2.47</td>
<td>1.75</td>
<td>+41%</td>
</tr>
</tbody>
</table>

Table 14: Fiber-Optic Calibration Slopes Compared with ANSYS Predicted Slopes ($\mu$m/g)
The general form of the system is shown in Equation 22 below. The letter “d”
denotes the displacement or strain from some applied force “f”, and the subscript denotes
the direction. The coordinate system is the previously defined gage coordinate system
unless otherwise stated. Coefficients \( c_1 \) to \( c_4 \) are unknown coefficients. There will be
two matrices of coefficients. One for the fiber-optic system and one for the strain gage
system.

\[
\begin{pmatrix}
  d_y \\
  d_y
\end{pmatrix} =
\begin{bmatrix}
  c_1 & c_2 \\
  c_3 & c_4
\end{bmatrix}
\begin{pmatrix}
  f_y \\
  f_y
\end{pmatrix}
\]

Equation 22

Equation 22 is simply a system of linear equations and can be re-written in the form of
Equation 23 and Equation 24.

\[
d_x = f_x c_1 + c_2 f_y
\]

Equation 23

\[
d_y = f_x c_3 + c_4 f_y
\]

Equation 24

When the force is applied only in the “X” direction, as is the case during the X-
calibration, the system can be written as Equation 25. A similar expression can be
written for the Y-calibration, and it is shown by Equation 26.

\[
\begin{pmatrix}
  d_x \\
  d_y
\end{pmatrix} =
\begin{bmatrix}
  c_1 & c_2 \\
  c_3 & c_4
\end{bmatrix}
\begin{pmatrix}
  f_x \\
  0
\end{pmatrix}
\]

Equation 25

\[
\begin{pmatrix}
  d_x \\
  d_y
\end{pmatrix} =
\begin{bmatrix}
  c_1 & c_2 \\
  c_3 & c_4
\end{bmatrix}
\begin{pmatrix}
  0 \\
  f_y
\end{pmatrix}
\]

Equation 26

Equation 25 can be expanded and written as Equation 27 which relates two of the
unknown coefficients to the slope values found in the calibrations. Equation 26 must be
solved in the same manner to produce Equation 28 which determines the values of the
final two unknown coefficients.
\[ d_X = c_1 f_X \]
\[ c_1 = \frac{d_X}{f_X} \]
\[ d_Y = c_3 f_Y \]
\[ c_3 = \frac{d_Y}{f_X} \]
\[ d_X = c_2 f_Y \]
\[ c_2 = \frac{d_X}{f_Y} \]
\[ d_Y = c_4 f_Y \]
\[ c_4 = \frac{d_Y}{f_Y} \]

**Equation 27**

**Equation 28**

The result of the calibration is shown in Equation 29 for the fiber-optic system and in Equation 30 for the strain gage system on the gage. The resulting coefficient values are for the average of five calibrations. In order to find the force based on the displacement, each calibration matrix must have an inverse and therefore be nonsingular. The determinants of both of the matrices are non-zero which proves they are nonsingular. Equation 31 simply states that the inverse of the calibration matrix multiplied by the displacements (or strains) recorded by the gage will lead to the applied force.

\[
C_{FO} = \begin{bmatrix} 5.48761 & 3.95073 \\ 6.99156 & 12.1822 \end{bmatrix} \quad |C_{FO}| = 39.229
\]

**Equation 29**

\[
C_{SG} = \begin{bmatrix} -0.15622 & -0.11224 \\ -0.00046 & -0.19058 \end{bmatrix} \quad |C_{SG}| = 0.29824
\]

**Equation 30**

\[
\vec{f} = C^{-1} \vec{d}
\]

**Equation 31**

To check the validity of the calibration matrix, the angular calibration data was used. Five calibrations were performed, and the average values were input into the calibration matrix. The output results for the fiber-optic and the strain gage measurement systems are shown in Figure 125 below. This shows excellent agreement between the X
and Y data for both measurement systems. This condition is where the error should be a maximum as the coefficients are computed when only pure X or Y force is applied. The maximum error noted for this figure is a 12% over-estimate of the shear force, for small forces. This decreases to a maximum error of 7% at a load of 6 grams, which corresponds to a shear of 300 Pa. This will be roughly the maximum shear expected during tests in the Virginia Tech supersonic wind tunnel. At the nominal wind tunnel value of 200 Pa the maximum error, at 45°, is 7.7% of the full scale value. These errors will decrease for smaller angles, and when the forces are perfectly aligned with the X and Y gage axes the errors approach zero as the calibration coefficients were created with these cases. It should be kept in mind that the exact solution assumed an angle of 45°, and there is a measurement uncertainty of about 2° in the actual angle.

A detailed angular calibration could reduce the magnitude of the gage error for flow with two components of shear not aligned with the X or Y axis of the gage. The flow situations where this gage is to be used have the majority of shear in one direction or

![Figure 125: Reduced Strain Gage and FFS Angular Calibration Data](image)

Figure 125: Reduced Strain Gage and FFS Angular Calibration Data
the other and this is known before the test, so maximum accuracy can be obtained through proper gage orientation and a detailed angular calibration was not needed. For unknown flow situations, a detailed angular calibration and an iterative table lookup system for data reduction would minimize this error and, if properly done, should bring it close to zero.
6 Thermal Testing

6.1 Cold Thermal Tests

Cold thermal tests were performed by utilizing a Teaney Environmental (Tenn II) environmental test chamber. The floor plate in the Virginia Tech supersonic wind tunnel experiences approximately a -15°C temperature change during each wind tunnel run at Mach 2.4. A test of the gage’s temperature sensitivity without a flow lets the magnitude of gage response to a change in temperature be quantified. This test is more severe than actual flow conditions as the change in temperature is experienced by the entire gage instead of just the head and indicates the possible thermal effects of a long exposure to a change in temperature.

The thermal test chamber is shown in Figure 126 below. Since the last static calibration was performed at a 45° angle, the gage was inserted in the test chamber at that angle. A Type-K thermocouple was inserted through an opening in the chamber and set near the gage so that the air temperature could be monitored. The location of the thermocouple is shown in Figure 127.

The operating characteristics of the chamber made it necessary to cool the gage and then heat it. In actual supersonic tunnel operation, the gage will be at room temperature.
temperature and then be rather suddenly cooled down. The chamber, however, took 15-20 minutes to cool 15 K, while it took only a minute to heat the same $\Delta T$. Since the cooling in the wind tunnel (~10 seconds) is rapid, it is felt that this yields a more useful comparison. Since the wind tunnels heat or cool much more rapidly than possible in the test chamber, this isn’t a direct comparison or a thermal calibration. It simply gives an idea of the sensitivity of the gage to a quasi-steady-state change in temperature.

Figure 128 shows the change in strain gage output versus the change in air temperature of the test chamber. It takes approximately 50 seconds to heat the chamber from 283 K to 298 K. The output of the gage lags significantly behind the change in temperature. This indicates that a short wind tunnel test of 5-10 seconds may not have significant temperature effects during the run, but rather that the gage might drift after the flow stops. Neglecting the cross-talk between the wheels, the output of 0.045 V, shown by the lower wheel, would roughly correspond to a change in shear force of -0.00703 g or -0.1092 Pa in the $\sim$X-direction. The upper wheel has a change of -0.065 V which would lead to an approximate error of 0.01239 g or 0.1924 Pa. If these two errors are multiplied through the calibration matrix an error of -0.534g (-26 Pa) in the X-direction and 0.342 g (17 Pa) in the Y-direction would be experienced. This demonstrates that the dependency of one axis on another may magnify any thermal errors in the system.

![Figure 128: Strain Gage Data vs. Change in Temperature (+15 K $\Delta T$) Run-2](image)
Figure 129 shows the lower wheel fiber-optic data versus time which are Channels 1 and 2 on the FFS. The heating begins at around 0 on this time scale instead of around 50 seconds on the time scale in Figure 128. The change in gap of the lower wheel counterweight is $-2.5 \mu m$ for channel-1 and $3.2 \mu m$ for channel-2. The average change reckoning the sign of channel 2 is then $2.85 \mu m$. Since the dependence is greater between the upper and lower wheels than for the strain gage data, the upper wheel data must be known before any estimate of the thermal effect can be made. The upper wheel data is shown in Figure 130 below and is channels 3&4 on the FFS. The time scale on this figure is the same as that on Figure 129. The change in gap for channel-3 is $6 \mu m$ and $-6 \mu m$ for channel-4. When multiplied through the calibration matrix this yields a change in shear of 0.2808g (13.7 Pa) in the x-direction and 0.3314g (16.2 Pa) in the y-direction. It should be noted that the change in Y-shear due to temperature with the fiber-optic measurement system is less than that of the strain gage measurements and that the change in X-shear is half that of the strain gages. Hot thermal tests were not performed as they will destroy the strain gages and the redundant measurement system would be lost.
Figure 129: Channel 1&2 Fiber-Optic Data vs. Time (+15 K ΔT) Run-2

Figure 130: Channel 3&4 Fiber-Optic Data vs. Time (+15 K ΔT) Run-2
7 Supersonic Wind Tunnel Testing

7.1 Test Setup

The supersonic wind tunnel test phase of this project was designed to evaluate the performance of the gage in a known flow. A number of prior wall shear gages have been tested at Mach 2.4 so this was the first condition tested. This facility allows the gage to be exposed to a shear of approximately 150-300 Pa at Mach 2.4, depending on the total pressure of the tunnel. The setup and equipment is essentially the same for the Mach 4.0 testing, so what follows is generally applicable. The nozzle block is changed but the same floor plate and instrumentation are used when switching to Mach 4.0.

The mounting plate for the tests is shown in Figure 131. A Haynes 230 insert is used so that the thermal properties of the surrounding wall match the head and housing of the sensor. This will help eliminate any error created when multiple materials are used where heat transfer changes can affect the skin friction coefficient by introducing wall temperature variations. The insert is 76.2x 207.9x 4.8 mm and is secured at the corners by #1/4-20 steel pan-head bolts. Every effort was made to create a smooth surface, and before installation the plate was polished to a near mirror finish with a die-grinder. The top surface of the insert is mounted flush with the aluminum tunnel floor plate. The bottom of the plate is sealed with a silicon gasket to prevent the leakage of air into the test section. The silicon is applied to the mating surface before the floor plate is installed and acts to seal the tunnel from the outside environment. Since the static pressure in the tunnel is around 3 psia and atmospheric is around 14 there is about an 11 psia vacuum and this would create random jets and disturb the test section flow field if the floor plate was to be left unsealed.
Figure 132 shows the shear gage assembled and all of the fiber-optic components attached. The ferrule holders were glued to the housing of the gage with a five-minute epoxy. Bonding them in place assures that they will not move during testing, even if the set screws loosen slightly. This was simply an extra precaution and isn’t absolutely needed.

The gaps between the fibers and reflectors were reset before the tests. A fiber had broken when the gage was transported and an additional fiber was moved to facilitate access. A slightly closer gap was used to provide the best signal quality possible. The actual gaps used in the first series of tests are shown in Table 15 below. It should be
noted that these measurements were taken with a Luna Innovations FiberPro USB instead of the Luna Innovations Fast Fiber Scan System used for all of the experiments and calibrations. The differences in operating properties are believed to be insignificant for setting the gaps.

<table>
<thead>
<tr>
<th>Gap (µm)</th>
<th>Channel-1</th>
<th>Channel-2</th>
<th>Channel-3</th>
<th>Channel-4</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>171</td>
<td>168</td>
<td>169</td>
<td>171</td>
</tr>
</tbody>
</table>

**Table 15: Gap Readings Before 10-21-03 Tests in Supersonic Wind Tunnel**

Figure 133 shows the wall shear gage in the tunnel with the X-axis of the gage aligned with the flow. This view is looking up at the bottom outside of the tunnel to the floor plate. The connector for the strain gage wires can be seen as well as the fiber-optic cables. The coordinate system of the gage is shown in the figure as a reference. Figure 134 shows the same view of the gage with a 25° angle between the X-axis of the gage and the streamwise flow. This test was performed to test the response of the second axis of the gage in a controlled manner by creating an artificially 3D flow relative to the gage coordinate system.
Figure 135 shows the Haynes 230 floor plate insert and the aluminum floor plate mounted in the test section of the Virginia Tech Supersonic Wind Tunnel. The locations of the static pressure port and wall-mounted Medtherm thermocouple are shown in Figure 136. These devices are upstream of the gage head and off to one side, so that they do not disturb the flow over the gage head. This necessitates them being near the edge of the Haynes 230 floor plate insert. A 2.5 in. scale is shown behind the gage head to provide a reference. The gage coordinate system is again superimposed on this photograph for clarity and so the reader has a reference when the data is displayed.

### 7.2 Accelerometer Results

Some tests were conducted with accelerometers mounted on the gage as well as on the tunnel to document the actual vibration environment. The locations of the accelerometers, on the gage, are shown in Figure 137 below. The accelerometers are labeled, based on how they are oriented with respect to the gage coordinate system. This view is looking up, at the bottom of the wind tunnel floor plate, from beneath the tunnel. The floor plate, of the tunnel, is 12.25 mm thick (0.5 in.) aluminum. Figure 138 shows the X-axis accelerometer on the nozzle block of the wind tunnel. This structure is massive, compared with the floor plate, and is made of solid steel. Figure 139 shows the Z-axis tunnel accelerometer mounted on the nozzle block/ test section junction. The tape is to prevent movement, and although it looks quite crude, it gets the job done.
Typical accelerometer data in the time domain is shown in Figure 140. As long as the tunnel started properly the acceleration data was similar between runs. This figure simply shows the accelerations versus time. The first subplot in the figure is the Z-axis for the wind tunnel, the second is the X-axis on the gage, the third is the gage Y-axis, the fourth is the gage Z-axis, and the fifth is the tunnel X-axis. The largest magnitude of vibration can be noted on the gage Z-axis. This is seems to indicate that the harshest loads during the starting and unstarting processes occur out-of-the-plane of the wind tunnel floor plate. This is similar to the result found by Remington at Mach 2.4.\textsuperscript{60}

Another piece of information that can be noted from the figure is that the gage mounted to the floor-plate experiences much more vibration than is transmitted through the larger structural members of the tunnel. The X-axis tunnel accelerometer measured
vibration quite a bit lower in amplitude than that measured by the X-axis accelerometer on the gage. A frequency domain view of the data is need for further analysis and interpretation.

![Figure 140: Channels 1-5 in Time Domain](image)

Figure 140: Channels 1-5 in Time Domain

The signal was transformed into the frequency domain using the “fft” command in Matlab. Care was taken so that the magnitudes are scaled correctly. Since the data was sampled at 50,000 Hz, the Nyquist frequency was 25,000 Hz and usable information is available up to that frequency. The figures only show up to 5,000 Hz for readability. Evidence of the ability to disregard the higher frequency content is quite apparent in the response of the shear sensor which simply uses all of the input energy on its first and second bending modes.

Figure 141 and Figure 142 give the frequency domain representations of the Z-axis accelerometer measurements on the wind tunnel. These two figures show that what was measured by this accelerometer is primarily noise. Figure 141 shows a DC component of about 0.05 G, which is small, compared with the accelerations measured on the gage itself. The frequency domain information is also shown in Power Spectral
Density (PSD) form, in Figure 142, as the energy is spread out over a large frequency with random vibration and one must be able to add it all up in order to get a real idea of the magnitudes.

Figure 141: Channel 1 (Tunnel Z-Axis) –FFT

Figure 142: Channel 1 (Tunnel Z-Axis)-PSD

Figure 143 and Figure 144 show the frequency domain information for the X-axis of acceleration measured on the gage itself. The DC component of the signal is 0.30 G and there are significant accelerations between 500 and 3000 Hertz, as shown in Figure 143. Again the data is plotted in PSD form in Figure 144 with the magnitude in dB/Hz. In the tunnel Z-axis data, the signal was about -90dB and this signal is about -40dB, which is quite an increase in magnitude especially on a logarithmic scale.

Figure 143: Channel 2 (Gage X-Axis)-FFT

Figure 144: Channel 2 (Gage X-Axis)-PSD
Figure 145 and Figure 146 show the frequency domain information for the Y-axis of acceleration measured on the gage. There are significant accelerations between 500 and 3000 Hertz, as shown in Figure 145. A peak is quite noticeable at about 850 Hz with amplitude of about 0.16 G, and this peak is also seen in Figure 143, albeit at a lower amplitude of 0.08 G. Again the data is plotted in PSD form in Figure 146 with the magnitude in dB/Hz, and it is around -35 dB.

Figure 147 and Figure 148 display the Z-axis data, the final component of acceleration measured on the gage. The DC component of this signal is about 1.25 G, and it is the highest of any of the signals measured in this study. At 851.5 Hz, there is a peak of 0.3 G which should be avoided in future shear gage design for Mach 4.0. The PSD representation of this signal is shown in Figure 148.
The final accelerometer data set is shown in Figure 149 and Figure 150 for the X-axis acceleration measured on the tunnel. It should be noted that the maximum amplitude of vibration is 0.05 G, and it occurs at 1702 Hz. 0.08 G at 850 Hertz is the maximum measured acceleration for the X-axis on the gage. The fact that the results on the gage and on the tunnel are different is to be expected. One is experiencing the effects of the flow first hand through a comparatively thin aluminum plate, and the other is mounted to a massive steel structure. The fact they respond differently makes sense. Figure 150 shows the frequency domain information for this case in PSD form.
7.3 **Supersonic Wind Tunnel Data Reduction**

7.3.1 Strain Gage Data Reduction

Data was reduced using the Matlab 6.1 software package with the addition of the signal processing toolbox. A sample of the lower wheel raw strain gage voltage data is shown in Figure 151, and the upper wheel voltage data is shown in Figure 152. There is a significant vibration in the signal, but it is simply the result of the first mode of the structure being excited by the operation of the wind tunnel. This first mode was predicted at 22 Hz in the idealized *ANSYS* model of the structure.

![Figure 151: 10-22-03 Run-1 Raw SG Lower-Wheel Voltage Data](image)

![Figure 152: 10-22-03 Run-1 Raw SG Upper-Wheel Voltage Data](image)
The frequency of the first mode was checked by hand and then numerically with a discrete Fourier transform (DFT). Matlab plots of the DFT’s are shown in Figure 153 and Figure 154. They are scaled to display the frequency composition of the signals. The lower wheel data shows the 18.9 Hertz first mode, and the upper wheel data confirms this and also shows a second mode at 47.5 Hertz. The first mode is 14% low from the predicted value of the first flexure mode of 22 Hertz. The FEA prediction of the second mode was 69 Hertz, therefore the error is 31% for the second mode. This large of a discrepancy is not, however, unexpected. The 140 µm (0.00551 in.) thick webs had a machining tolerance of about ± 10 µm (0.0004 in.). The webs also have strain gages affixed with all of the associated glue, wires, and solder pads. Modeling all of these parts would be important if a high degree of precision was required, but this level of detail was decided to be impractical with the computing resources available.

Figure 153: FFT of Lower Wheel Strain Gage Data
The damping ratio of the structure can be computed using Equation 32. Figure 155 shows how the needed information was taken from the upper and lower wheel raw strain gage data. The resulting damping ratio, found with the data in Figure 151 is $\zeta_L = 0.00109$ for the lower wheel. The data used from this figure was the portion after the tunnel stops and the gage is free to vibrate. The data in Figure 152 for the upper wheel...
Equation 33 shows the expression to calculate the un-damped natural frequency $\omega_n$ from the damping ratio and the damped natural frequency $\omega_d$. Due to the very small values of $\zeta$, for this structure $\omega_n$ and $\omega_d$ are essentially identical. A summary of the results for the frequency and damping data is shown in Table 16.

$$\zeta = \frac{\frac{1}{\pi} \left( \ln \frac{4}{\pi} \right)}{\sqrt{4\pi^2 + \left( \frac{1}{e-1} \left( \ln \frac{4}{\pi} \right) \right)^2}}$$

Equation 32

$$\omega_n = \omega_d \sqrt{1 - \zeta^2}$$

Equation 33

<table>
<thead>
<tr>
<th></th>
<th>Calculated $\omega_n$ (Hz)</th>
<th>$\omega_d$ (Hz)</th>
<th>$\omega_n$ (Hz)</th>
<th>% Error</th>
<th>$\zeta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lower Wheel</td>
<td>22</td>
<td>18.9</td>
<td>18.9</td>
<td>16.4</td>
<td>0.00109</td>
</tr>
<tr>
<td>Upper Wheel</td>
<td>69</td>
<td>47.5</td>
<td>47.5</td>
<td>31.2</td>
<td>0.00144</td>
</tr>
</tbody>
</table>

Table 16: Summary of Frequency Results

The next step in the data reduction process is the use of a low-pass digital filter in Matlab. The filter chosen is the Matlab “fir1” which is linear in phase. A 200 order filter was used, and this shifts the signal 200 steps forward in time. This was corrected for when plotting and comparing the filtered results to the wind tunnel data. Figure 156 shows the filtered raw data for the lower wheel. Figure 157 shows the effect of filtering on the upper wheel data. A notch filter could be designed to filter out only the natural frequencies of the structure and preserve the frequency response of the gage.

![Figure 156: Lower Wheel Filtered Strain Gage Data](image-url)
7.3.2 Fiber-Optic Data Reduction

The same procedures for reducing the data were used for the fiber-optic data with two important exceptions. When the data from the FFS system was first examined there was a phase correction applied to the reduced gap data. This changed the results for the large displacements seen for this sensor. The solution to this problem was to reprocess the gap data without the phase correction. This required the use of the raw spectrometer data. All data presented for the fiber-optic data acquisition system is from this reprocessed data. If the gage is to be used with maximum displacements less than a couple of microns this phase correction must be turned on.

The second way in which the FFS data is processed differently arises from lost data points. While the strain gage system continuously outputs a voltage, the fiber-optic system calculates a gap based on the reflected light. If the change happens too rapidly, smearing occurs and the gap cannot be calculated. When there is no calculated gap, the value output by the algorithm the value is filled with a number. This creates an obviously bad point as it is offset from the rest of the data by a significant amount. This is shown by the data displayed in Figure 158. There are a few points which fall a great distance away from the rest. The solution to this problem is simple. The “bad” points are thrown out, and an average of the previous 50 points is used in its place. This can be done with
reasonable confidence as there only seem to be a few flyers from over 80,000 other data points. This prevents the flyers from skewing the average value. The time scale for this figure is the time scale recorded for the FFS and is not corrected to match the wind tunnel data.

Figure 158: 10-22-03 Run-1 Fiber-Optic Channel-2
7.4 Mach 2.4 Wall Shear Results

7.4.1 Gage X-Axis Aligned with Flow

Typical results are shown in Figure 159 through Figure 162 for the case where the gage X-axis is aligned with the streamwise flow. The flow conditions in the wind tunnel are shown in Figure 159. There is a “test window” labeled in the figure and in all subsequent figures for the run. This is the region considered with “steady-state” test conditions. This yields an approximate 6 seconds of steady state time for each 8 second supersonic wind tunnel test. The computed Mach number for this case is 2.34, with the nominal Mach number for the tunnel being 2.4.

Figure 159: 10-22-03 Run-1 M=2.4 Tunnel Conditions

Figure 160 shows the components of wall shear for both the strain gage and fiber-optic output of the gage. Since the flow is aligned with the X-axis and nominally 2D one would expect that the Y-component would be zero or close to it. This is true in the data for both measurement systems. It takes about 3 seconds for the total pressure to stabilize.
and once this occurs the Y-component of shear is close to zero. When the tunnel starts and unstarts the gage receives a sudden load. This can be seen in the data set for both the X and Y-axis data. The X-component of shear for both the strain gages and fiber-optic system smoothly follows the same shape as the total pressure plot in Figure 159. The two measurement systems agree with each other within a few Pascal of shear. After the tunnel shuts off the gage gradually returns to zero. The fiber-optic data returns to zero faster than the strain gage data. The fiber-optic system should be less sensitive to changes in temperature than the strain gages so temperature may be a portion of the cause.

Figure 160: 10-22-03 Run-1 M=2.4 Wall Shear Components

Figure 161 shows the magnitude of the total shear measured at Mach 2.4 by both the strain gages and fiber-optics. This again shows good agreement between the two measurement systems of the gage and the shape of the total pressure curve. This data is shown in nondimensional form in Figure 162. The data is for the “steady-state” portion of the experiment and the approximate skin friction value measured is 0.0013. The fiber-
optic system measured a slightly higher value of wall shear than the strain gages for all Mach 2.4 cases.

Figure 161: 10-22-03 Run-1 M=2.4 Total Wall Shear

Figure 162: 10-22-03 Run-1 M=2.4 $C_f$
7.4.2 Gage X-axis at 25° Angle

The actual flow in the supersonic wind tunnel is nominally 2D, so the gage axis was purposefully misaligned by 25° to create a flow that appeared 3D to the gage. Figure 163 through Figure 166 show the flow results at Mach 2.4 with the x-axis of the gage tilted at a 25° angle from the streamwise direction of the flow. Figure 163 shows the tunnel conditions. Note the change in wall temperature is about 6 K. Figure 164 shows the X and Y axis shear data. Figure 165 shows the total shear value computed from the components shown in Figure 164. Figure 166 shows the value of $C_f$ computed from the total wall shear and nondimensionalized by the local flow values.

Figure 163: 10-23-03 Run-1 M=2.4 25° Tunnel Conditions
Figure 164: 10-23-03 Run-1 M=2.4 25° Wall Shear Components

Figure 165: 10-23-03 Run-1 M=2.4 25° Total Wall Shear
Table 17 shows the error in flow angle, relative to the gage coordinate system for both the strain gage and fiber-optic data. The flow angle is calculated from the X and Y shear components, with the assumption that the flow is essentially 2D and the shear force is down the X-direction of the tunnel coordinate system. There is a bigger error in the fiber-optic flow angle, from the ideal 25°, than the strain gage calculated flow angle. The larger error in the fiber-optic angle is likely due to the higher dependence between the wheels for displacement based shear component computation. There is also some uncertainty in the “25° angle.”

<table>
<thead>
<tr>
<th></th>
<th>Total Shear</th>
<th>X-Shear</th>
<th>Y-Shear</th>
<th>Angle</th>
<th>Error from 25°</th>
</tr>
</thead>
<tbody>
<tr>
<td>Strain Gage</td>
<td>163 Pa</td>
<td>152 Pa</td>
<td>66 Pa</td>
<td>23.5°</td>
<td>-6%</td>
</tr>
<tr>
<td>Fiber-Optic</td>
<td>174 Pa</td>
<td>164 Pa</td>
<td>62 Pa</td>
<td>20.7°</td>
<td>-17%</td>
</tr>
</tbody>
</table>

Table 17: Error in Angular Shear Data
7.5 Mach 4.0 Results

The results for the Mach 4.0 case are shown in Figure 167 through Figure 170, and they follow the same format as the Mach 2.4 results. The main things to note are that the gage was kept aligned with the flow for all of the Mach 4.0 cases and that 32 runs were performed. The total pressures at Mach 4.0 are about 13 atmospheres, as opposed to around 4.5 atmospheres at Mach 2.4. The Mach number calculated from the total pressure and static pressure was 3.62. The tunnel control was somewhat more inconsistent at Mach 4.0 and more sensitive to changes in settings than at Mach 2.4. This likely contributes to the uncertainty in the measurements.

![Figure 167: 12-13-03 Run-13 Tunnel Conditions](image)

Figure 167 shows the X and Y components of the wall shear for both measurement systems. The magnitude of the wall shear is lower at Mach 4.0 than at Mach 2.4 with a peak shear value of approximately 85 Pa compared with about 200 Pa in Figure 160 for Mach 2.4. The starting and unstarting of the tunnel is considerably more violent at Mach 4.0 with a larger change in shear value. The steady-state portion of the run shows good agreement between the strain gages and fiber-optics. The Y-component
is slightly below 0 at -15 Pa. The shear values follow the total pressure for both measurement systems. The return to zero is slower at Mach 4.0 than at Mach 2.4 with a larger offset immediately after the run. The fiber-optic system returns to zero slightly quicker than the strain gages.

Figure 168: 12-13-03 Run-13 Wall Shear Components

Figure 169 shows the magnitude of the total wall shear measured. The strain gages output a slightly higher shear value than the fiber-optic system at Mach 4.0. This was true for every test performed. This is the opposite result noted at Mach 2.4. Figure 170 shows the $C_f$ values for this test. The values are about 0.0007 for this condition compared with 0.0013 for Mach 2.4.
Figure 169: 12-13-03 Run-13 Total Wall Shear

Figure 170: 12-13-03 Run-13 $C_f$
7.6 Boundary Layer Survey

The results of the boundary layer survey yielded the thickness of the wind tunnel boundary layer. The raw data from a typical traverse is shown in Figure 171. The circle around the $P_{o2}$ pressure indicates the location chosen for the boundary layer edge. This was chosen as the point where the slope was going towards zero. The other circle is the corresponding traverse position, in this case about 22.9 mm (~0.903 in.). This is used to calculate an estimate of the wall shear. All of the other boundary layer survey data is available from the author.

Figure 171: 12/17/03 Run-9 M=4.0 Boundary Layer Traverse Data
The boundary layer thickness is used to calculate a Reynolds number based on the boundary layer thickness. The formula is shown in Equation 34 below and is simply the standard formula for Reynolds number calculation with the boundary layer thickness as the chosen characteristic length.

\[
\text{Re}_\delta = \frac{\rho U \delta}{\mu}
\]

Equation 34

This information is first used in the incompressible, flat-plate skin friction formula in Equation 35\(^1\). The calculation was performed in Mathematica with the experimental data and a skin friction coefficient obtained for each boundary layer run. The total pressure was varied, in order to get a \(C_f\) estimate for as wide a range of total pressures as possible. This estimate yields value for incompressible, turbulent flow.

\[
\sqrt{\frac{2}{C_f}} = A \log \left( \text{Re}_\delta \sqrt{\frac{C_f}{2}} \right) + C - B
\]

\[
A = 5.6 \\
B = -2.5 \\
C = 4.9
\]

Equation 35

Equation 36 shows the Van Driest II incompressible-to-compressible correction factors for \(C_f\), taken from Figure 10-7 in Ref. [1].

\[
\frac{C_{f, \text{compr}}}{C_{f, \text{incomp}}} = 0.66 \rightarrow \text{(Mach 2.4)}
\]

\[
\frac{C_{f, \text{compr}}}{C_{f, \text{incomp}}} = 0.50 \rightarrow \text{(Mach 3.6)}
\]

Equation 36
There are many methods of computing the incompressible-to-compressible correction factor and another technique is the reference temperature method. Equation 37 shows the needed expressions from Reference 78. The Prandtl number is assumed to be 0.7 for air. For Mach 2.4 this yields a correction factor of 0.686 and for Mach 4.0 the factor is 0.477. These agree quite well when compared to the results shown in Equation 36. Both results are plotted when the results are displayed.

\[
\frac{C_{f,\text{compr}}}{C_{f,\text{incompr}}} = \left(1 + r \frac{\gamma - 1}{4} M^2 \right)^{-1}
\]

\[r = \sqrt{Pr} \]

Equation 37

The results for \(C_f\) predicted using the of the boundary layer survey data for Mach 2.4 are shown in Figure 172 along with the experimentally determined values of \(C_f\) for both the strain gages and the fiber-optic data acquisition system. The results are plotted as a function of the total pressure. It should be noted that the fiber-optic system tends to predict a higher skin friction coefficient at Mach 2.4 than the strain gage system. The Van Driest II incompressible-to-compressible factor predicts a slightly lower skin friction coefficient value when compared with the reference temperature method at Mach 2.4. The gage values fall in a range which encompasses both values. The agreement between the direct \(C_f\) measurement and the semi-empirical \(C_f\) prediction is very good, as might be expected for a simple, flat-plate flow. This represents a critical step in the validation process for the gage.
Figure 172: Boundary Layer Survey Predicted $C_f$ & Gage Values at Mach 2.4

Figure 173: Boundary Layer Survey Predicted $C_f$ & Gage Values at Mach 4.0
The results of the boundary layer survey at Mach 3.62 (Mach 4.0 nominal) are shown in Figure 173. The skin friction coefficient is noticeably lower at Mach 4, with a value of about 0.0007, compared with about 0.0013 at Mach 2.4. The strain gages predicted a consistently higher $C_f$ value than the fiber-optic system at Mach 4.0. Both measurements were lower than the predicted value. This could mean that the prediction doesn’t work as well at Mach 4.0. There appears to be a larger deviation in the data from the mean. This is likely due to the smaller shear values at Mach 4.0 as the noise and therefore uncertainty would become a larger proportion of the signal. A comparison of the semi-empirical results to the values measured with the gage is shown in Table 18. The Van Driest II incompressible-to-compressible factor predicts a higher lower skin friction coefficient value when compared with the reference temperature method at Mach 4.0. At Mach 4.0 the spread between the two semi-empirical prediction methods is greater and the gage data agrees with more closely with the reference temperature method. The direct $C_f$ measurements are in agreement with semi-empirical prediction methods at Mach 4.0. This shows that the gage works well at both Mach 2.4 and at Mach 4.0 which validates the gage design at two test conditions.

<table>
<thead>
<tr>
<th>Method</th>
<th>Mach 2.4</th>
<th>Mach 4.0</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Van Driest II</td>
<td>Reference Temperature</td>
</tr>
<tr>
<td>Fiber optic</td>
<td>±12%</td>
<td>±8%</td>
</tr>
<tr>
<td>Strain Gage</td>
<td>±7%</td>
<td>±10%</td>
</tr>
</tbody>
</table>

*Table 18: Semi-Empirical Results Compared with Gage Results*
8 Gage Error and Uncertainty

Errors in wall shear gages can be quite difficult to accurately quantify. The possible sources of error include the calibration of the gage, misalignment between the outer housing and sensing head, thermal errors, angular errors, and random errors in acquiring the data. Every attempt was made to minimize these errors. A careful calibration was performed, and the same data acquisition equipment was used whenever possible. This section breaks down the errors into their sources and examines the error in the measurements from the multiple tests performed in the wind tunnel. To assess errors under conditions different from those tested, multiple measurements should be done to provide a large number of samples for a statistical analysis of the gage.

8.1 Calibration

The calibration of the gage was performed 6 times each for the X-axis, Y-axis, and on a 45° angle. One of these calibrations was performed as a check during wind tunnel experiments at Mach 2.4. A statistical analysis was performed on the data, and all of the statistical quantities were computed. Twice the standard deviation was used as an error for the calibration analysis. With a Gaussian distribution of samples, this means that the values should fall within $2\sigma$ of the mean 95% of the time. The highest percent error for each condition was used here to estimate the worst case scenario. This information is made on only 6 samples so it boarders on marginal statistical ground.

8.2 Head Misalignment/Gap/Geometry

A known source of errors for wall shear gages is the misalignment of the floating head with the outer housing. Even small misalignments or geometrical differences can create forces on a similar order of magnitude to shear forces and yield error in the measurements. The geometry of the head is shown in Figure 174 below.
The gap between the head and the outer housing was designed at 100 \( \mu m \) (0.005 in.) and the actual gap is approximately 175 \( \mu m \) (0.008 in.) as the head wasn’t perfectly centered on its outer housing. With the 16 mm (0.63 in.) head diameter this yields a 1.09\% head to gap ratio as shown in Equation 38 below ratio is defined by MacLean.\textsuperscript{79,80} MacLean found that this would produce an error of 1.9\% in the strain measured. While this finding is for the style of skin friction gage which used a single flexible beam this information is felt to be applicable to this gage.

\[
G / D_{\text{Head}} = \frac{\text{gap}}{\text{Diameter}} 100 = 100 \frac{175 \times 10^{-6}}{16 \times 10^{-3}} = 1.09\%
\]

Equation 38

The next misalignment error source investigated is recession or protrusion of the floating head above or below the surface. The misalignment of the floating head with surrounding surfaces is believed to be less than 25 \( \mu m \) as there is no visible or tactile misalignment noted when the gage is inspected. This yields a 0.16\% factor when nondimensionalized in the same manner as Equation 38. This should yield a maximum error of less than 1\%.\textsuperscript{79}

Another source of possible error is the thickness of the lip. The lip on this head is 1.5 mm thick, which yields a c/D ratio of 9.4\%. The thinner this part of the gage is the less sensitive the gage will be to pressure forces. An estimate of the possible effect on
gage output would be 1% due to the lip geometry. This is dependent on the pressure gradient across the head of the gage and should be reevaluated if the gage is to be used in flow with a strong pressure gradient.

The method of Allen in references [81] and [82] was investigated as a possible secondary method of estimating errors due to misalignment of the floating head. Allen used a method of assuming a total force coefficient shown in Equation 39. The value for “a” is defined as the distance between the surface of the floating head and the moment center of a single-pivot balance. This distance has no direct relation to the current design geometry. The $C_N$ value was assumed zero as the pressure gradient across the head should be small and this design should be insensitive to it.

$$C_l = C_l + (1 - \frac{c}{2a})C_l + \frac{b}{a}C_N$$

Equation 39

The parameters $G/D=0.0109$, $c/D=0.094$, and $z/\delta=0.0018$ at $M=2.4$ and $z/\delta=0.0025$ at Mach 4.0 yield a $C_l$ with essentially no effect on the lip force. The parameter $z/\delta$ is the possible recession divided by the head diameter. The recession is assumed as 25.4 µm or less. According to Figure 10 in Ref. [81] this small of a recession error should have no measurable effect. The lip force cannot be accurately estimated with Ref. [81] as the maximum $c/D$ ratio tested in that investigation is half of that used in this investigation.

### 8.3 Thermal

The thermal testing of the gage was shown in Section 6. This demonstrated the heating of the gage $+15^\circ$ C from a cooled condition over a few minutes. This is a more severe test than would be experienced in a flow facility, as the entire gage was heated. It is simply indicative of the magnitude of thermal error expected during wind tunnel experiments. The thermal error values of 10.4% for the strain gages and 6.5% for the fiber-optic system should be very pessimistic for the supersonic wind tunnel as only the head is exposed to the flow. Therefore, it should help to make the overall error estimate conservative.
8.4 Angular

The uncertainty of the measurement due to the angle that in forces make in relation to the head is another source for possible error in measurement. The angled calibration data was used in this case. The gage was set at a 45° to the ground and weights were hung as in the other calibrations. The resulting plot was shown as Figure 125, and the results are discussed in that section. For this discussion, the primary result is the +2.7° error in the strain gages and +3.4° error in the fiber-optic system from 45° at 6 grams of load. The angular error was a maximum at smaller load values with a maximum of +6.8° at 1-g for the fiber-optics and +6.4° for the strain gages.

8.5 Summary of Uncertainty Results

The results of the error and uncertainty computations are summarized in Table 19 below. The first four rows in the table show the effect of misalignment errors of the head on the final results. The total error, expected from the geometry, is only about 2.4%. When compared to the thermal errors and calibration uncertainty, it is clear which areas have the most room for improvement. A more careful calibration or one in a more controlled environment would likely reduce the magnitude of the total uncertainty. The small number of samples may also be skewing those results.

<table>
<thead>
<tr>
<th>Type of Error</th>
<th>Strain Gages</th>
<th>Fiber-Optic System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Head Lip Size</td>
<td>± 1%</td>
<td>± 1%</td>
</tr>
<tr>
<td>Head Protrusion</td>
<td>± 1%</td>
<td>± 1%</td>
</tr>
<tr>
<td>Head/Housing Gap</td>
<td>± 1.9%</td>
<td>± 1.9%</td>
</tr>
<tr>
<td><strong>Total Head Effect Uncertainty</strong></td>
<td>± 2.4%</td>
<td>± 2.4%</td>
</tr>
<tr>
<td>Maximum Error in 45° Calibration (6g)</td>
<td>+6.4°</td>
<td>+6.8°</td>
</tr>
<tr>
<td>15 K ΔT (On Whole Gage) Error</td>
<td>±10.4%</td>
<td>±6.5%</td>
</tr>
<tr>
<td>2σ Uncertainty in Calibration</td>
<td>±10.5%</td>
<td>±10.3%</td>
</tr>
<tr>
<td>2σ Uncertainty in M=2.4 C_f (28 Runs)</td>
<td>±8.0%</td>
<td>±3.2%</td>
</tr>
<tr>
<td>2σ Uncertainty in M=4.0 C_f (32 Runs)</td>
<td>±18%</td>
<td>±16%</td>
</tr>
</tbody>
</table>

Table 19: Summary of Gage Errors and Uncertainty
The uncertainty in $C_f$ measured in the supersonic wind tunnel at Mach 2.4 is less than the calibration uncertainty recorded. There are enough runs to have confidence in this statistical data. The higher uncertainty at Mach 4.0 is likely due to the higher vibration and noise environment. Part of the higher uncertainty could have been the lower magnitude of the shear force at Mach 4.0 and the signal to noise ratio will decrease yielding higher uncertainty, even if the noise didn’t increase from the Mach 2.4 level. This level of uncertainty in the final measurement compares well with the prior single wheel flexure. The prior gage had an 11-16% error for high enthalpy scramjet wind tunnel tests.\(^5\)

A systematic and detailed set of experiments to detail the floating-head misalignment error of this gage type would be of great benefit to future investigators. Much of the prior error analysis has been based on a nulling or non-nulling cantilever beam design. In these designs the head tilts slightly with any motion. With the dual wheel-flexure concept the head translates and the gap changes asymmetrically. This asymmetric gap change may be another source of gage error that hasn’t been investigated in the past.
9 Conclusions and Recommendations

9.1 Conclusions

The main goals of this research were to design, analyze, produce, and test a fiber-optic direct-measuring skin gage for high-temperature applications. Two entirely different sensor concepts were created to try and meet this objective.

The first skin friction gage design incorporated only a fiber-optic displacement measurement system for testing. Finite element analysis was successfully used to predict the gage’s behavior in calibration. The main objective was to raise the first mode so high that it would be unlikely to be excited in the test environments. The path chosen was to create a gage that could be made with high-temperature ceramics. Designing a head light enough and a flexure stiff enough to raise the first mode above any significant vibration turned out to be dead end in this study. An aluminum prototype gage was designed and tests its extreme sensitivity to vibration and resulting smearing of the fiber-optic system led to the abandonment of this concept. Also, the thermal sensitivity of this gage rendered it unusable in a flow environment.

The dual wheel-flexure shear gage design was the second and successful gage design. The design, analysis, calibration, and cold-flow testing of this shear gage have shown the validity of the two-wheel basic design concept. A workable two-component fiber-optic wall shear gage can be made without the use of viscous damping fluids.

High-temperature materials were chosen for the second gage. The flexures, head, and top bossing are made of Haynes® 230® which is a high-temperature nickel super-alloy that can survive up to 1600K before melting and retains its strength at high temperatures. The gage housing and fiber-optic housing components can survive to 1660K and are made from 316 stainless-steel. The limiting factor on survivability for the fiber-optic only gage is 1143K due to the silver paste bond between the fused-silica fiber-optic ferrules and the 316 SS ferrule holders. The strain gages have a temperature limit of 473K due to the solder and adhesives used. Thus, they are for lower temperature measurements and fiber-optic system validation only.
The use of both the strain gages and fiber-optic measurements systems in a single gage was instrumental to the success of this gage. Redundant measurement systems aided in troubleshooting and doubled the amount of information that could be gathered in each test performed. Dual measurement systems should be used whenever the application allows it, as it provides a much higher level of confidence in the results and provides a direct means to trouble-shoot either measurement system. The fiber-optic system works without having to instrument the flexures or touch the counterweights and doesn’t have wires that could interfere with the movement of the flexures. Gold-coated fiber optics are capable of operating at very high temperatures.

Finite element analysis proved a powerful design tool. The predictions of the slopes for displacement versus load on the lower wheel were within -0.5% of the experimentally determined value and within -7% for the upper wheel when the load was applied in the X-direction. The error in the predicted slope versus the experimentally measured slope increased for calibration in the Y-direction. The error was -63% for the lower wheel and -48% for the upper wheel. Statically, the basic behavior was as predicted even though there was some error in the magnitudes of the values. The geometry modeled and the actually geometry of the flexures are not identical. If the actual geometry could be measured to a high degree of precision the finite element model could be refined and more accurate results could be obtained.

The modal analysis proved essential as it was able to predict the modes of the upper and lower wheels in combination. This allowed comparison with known high vibration frequencies in the supersonic tunnel. It is highly desirable that the modes of the test facility do not coincide with those of the gage. Finite element modal analysis results were within 16.4% for the first mode and within 30% for the second mode when compared with the experimentally determined modes. Uncertainties in the finished geometry make this level of error acceptable.

The thermal analysis provided an estimate of how long the gage will be able to survive and operate effectively with a high heat flux on the floating head. The gage is capable of surviving up to 10 minutes of a very high heat flux. It should be able to operate for up to 3 minutes with only a 10% change in the temperature-dependent strength of the flexures of the upper wheel.
Repeated calibrations demonstrated the linear and repeatable output of the gage. Uncertainties in the calibration are about 10% for both the strain gage and the fiber optics systems. A 45° angular calibration showed a maximum gage error at 6g load of 6.4° for the strain gages and 6.8° for the fiber-optic measurement system. This is the worst-case angular error situation. The angular error can be reduced through a series of detailed angular calibrations.

Wind tunnel tests at Mach 2.4 and Mach 4.0 under high Reynolds number conditions showed the robustness and repeatability of wall shear measurements with the gage. Excellent agreement was shown between semi-empirical methods which used boundary layer survey data, and the gage output. This agreement held true for both the strain gage and fiber-optic measurement systems in all cases tested. The gage has also been shown to make measurements in 3D supersonic flows. Digital signal processing techniques were used to post process the wind tunnel data and extract the maximum amount of information possible.

Uncertainties at Mach 2.4 were very good at 3.2% for the fiber-optic data and 8% for the strain gage data. At Mach 4.0, the uncertainties are acceptably low with 16% for the fiber optics and 18% for the strain gages. The gage has been shown to operate with the vibrations present during wind tunnel runs. Successful operation at these two flow conditions was vitally important for validation of the dual-wheel-flexure concept.

This investigation has accomplished all of the goals set forth at the start. A skin friction gage was created that can survive up to 1073 K in pure fiber-optic form. High-temperature materials are integrated into the design. The gage uses redundant measurements systems with both fiber-optics and strain gages available up to 473 K. There is no need for viscous damping. A successful series of measurements have been made at Mach 2.4 and Mach 4.0.
9.2 Recommendations for Future Work

High-temperature, supersonic and/or hypersonic tests need to be undertaken to define the actual operational limits of this gage. The results of the high-temperature thermal analysis will be verified when these hot-flow tests are undertaken. Active cooling with heat sinks and/or water cooling could further increase the acceptable test duration of the gage but at the expense of added complexity. It may be worthwhile for very long duration tests and further expansion of the operating envelope of the gage.

An area that needs further development is reduction in the amount of uncertainty in wall shear measurement. A dedicated uncertainty analysis should be undertaken with the dual wheel-flexure gage design, since the head gap changes asymmetrically as it moves. The motion is different than the gage types used in previous skin friction gage error analysis experiments or computations. These have mainly focused on nulling style gages. An experimental uncertainty study which examined the effect of protrusion and recession error, floating head to housing gap size changes, and lip size effects should be undertaken for a non-nulling gage of the current configuration. This study could be performed the existing gage in the Virginia Tech supersonic wind tunnel. This would not be a simple task, but the ease of replacing the floating head of this design would make it possible. The top bossing can be adjusted to provide a known recession or protrusion error with the use of easily fabricated shims. A detailed series of angular and thermal calibrations should be useful in further reducing and/or quantifying the uncertainty of the present gage concept.

The frequency response of the gage needs to be quantified. Different post-processing techniques could increase the frequency response if that is needed. The current design uses a low-pass filter which limits the frequency response of the gage to ~18 Hz. The frequency response function of the gage could be measured and a notch filter designed that would allow the frequency response to be much higher. This would improve the gage performance with only a change in post-processing of the data.
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VITA

Matthew William Orr was born January 20, 1978 in Barnstable, MA. He entered George Mason University in 1995 in physics and graduated from James Madison High School in Vienna, Virginia in 1996. Upon completion of high school requirements he transferred to Virginia Polytechnic Institute and State University where he received his Bachelors degree in aerospace engineering in December of 1999. He continued study at Virginia Tech under Dr. Joseph A. Schetz and received his Masters of Science Degree in December of 2000 for research dealing with the aerodynamic testing of a twin-fuselage commercial transport. Research on sonic boom mitigation was undertaken and wind tunnel investigations were performed on burning ethylene in the wake of a keel. In 2002 the research turned to fiber-optic shear sensors for hot high-speed flows.

Personal interests are related to aviation. He is a private pilot with single engine land privileges. He has flown sailplanes, helicopters, and antique tailwheel airplanes. He is active in the design and construction of amateur-built aircraft and UAV’s. He is has been an active model aircraft designer and builder since age 10 and has completed over 100 aircraft. He has worked as a professional model/UAV builder and pilot and competed in three AIAA Design/Build/Fly competitions and in a DARPA micro-UAV competition.