Re-Design of a Shear-Stress Sensor Load Cell for Liquid Flows

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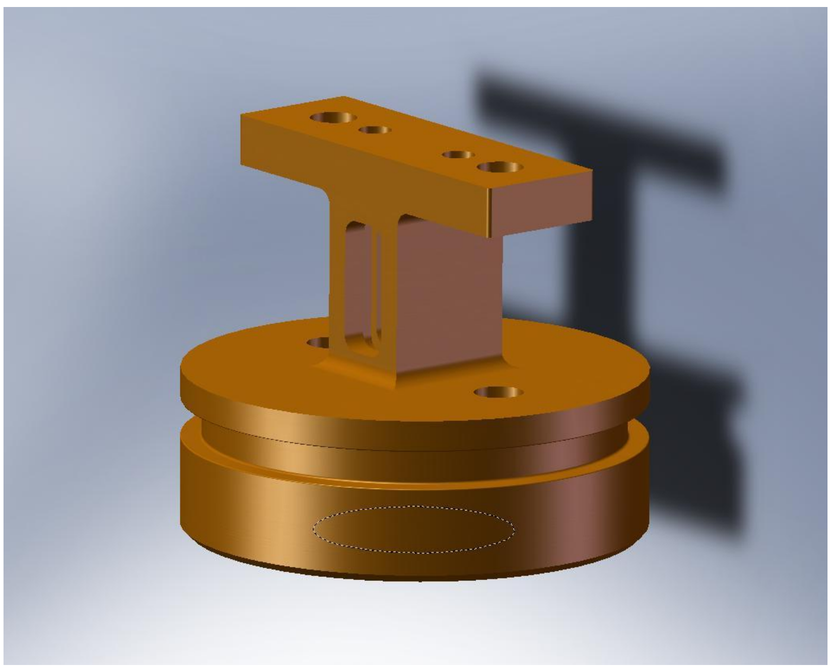

ABSTRACT

The measurement of friction drag produced by flowing liquids can be accomplished with a floating plate balance. A flush-mounted plate will experience the flow-induced shear stress, and this stress is measured by a load cell. Precision mounting of the plate, and the stiffness and sensitivity of the load cell are key elements to the design of the shear stress sensor. Moreover, these elements must be submersed in liquid if they are to be used for water flows. A shear stress sensor and load cell assembly have been developed which is sufficient for these measurements, but improvements should be made in the following areas:

- Replace the semiconductor strain gauges on the cell with foil strain gauges without decreasing the sensitivity of the cell. This will require a re-design of the load cell itself.
- Redesign of the waterproofing of the cell.
- Consider optical strain gauges to replace the resistive gauges.

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INTRODUCTION

For many engineering purposes, particularly in the naval field, it is useful to be able to measure the friction drag imparted on a surface by a liquid flow. Practical applications include ship hull design and the evaluation of friction reducing coatings, among many others. One method of determining the friction drag is by measuring the shear stress induced by the liquid on the surface of interest. Our project sponsor, Professor Steven Ceccio of the Department of Mechanical Engineering at the University of Michigan, uses such a method.

Current Design

The current design, shown in Fig. 1 and Fig. 2 below, uses a floating plate attached to a beryllium copper load cell. The load cell is mounted in a housing such that the plate is flush with the surface being studied. When the liquid, usually seawater, flows over the plate, the resulting shear stress causes a small deformation in the load cell. A semi-conductor strain gauge mounted inside the load cell records this deformation. The friction force of the liquid can then be found using this deformation and the material properties of the load cell.

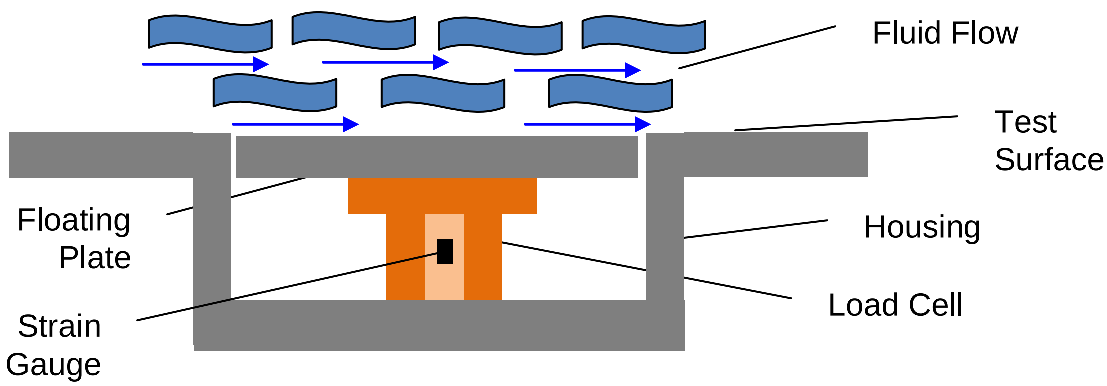


Fig. 1. Cross-section of the current design configuration

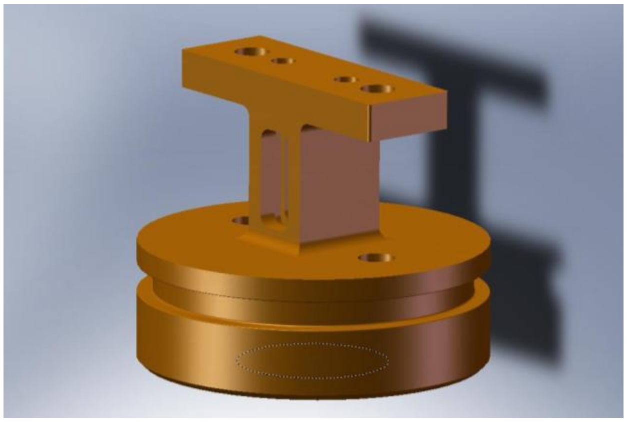


Fig. 2. CAD drawing of the current load cell. Strain gauge not shown.

Current Design Issues. The current load cell in service has several problems associated with it. First, the load cell must be sent to France for installation of the semi-conductor strain gauge. This is both time consuming and costly; each load cell costs several thousand dollars and takes three to four months to manufacture. Next, the strain gauge is extremely susceptible to failure when in contact with water. Consequently, water intrusion into the load cell is the dominant failure mode of this design, which has an average lifetime of about six months. Therefore, Professor Ceccio has requested the design be improved in the following ways:

- Replace the semiconductor strain gauges on the cell with foil strain gauges without decreasing the sensitivity of the cell. This will require a re-design of the load cell itself.
- Redesign of the waterproofing of the cell.
- Consider optical strain gauges to replace the resistive gauges.

NOMENCLATURE

Gauge Factor: A measure of resistive gauge sensitivity. The higher the gauge factor, the more sensitive the strain gauge is.

$$GF = \frac{\Delta R / R}{Strain}$$

INFORMATION SEARCH

Because the primary customer request is replacement of the strain gauge, we began our web information search by investigating several different types of strain gauges that are currently available. The basic principle of all strain gauges is universal. In a strain gauge, the force applied changes a physical property of the gauge in predictable, quantitative way. Since the change in the physical property is directly related to the force applied, one is able to determine the force applied.

Semi-conductor Strain Gauges

Most semi-conductor gauges are silicon based. In a semi-conductor strain gauge, the resistance, resistivity, and physical dimensions of the gauge change with strain. This type of strain gauge has a high gauge factor, typically from 50 to 200; however this number is neither constant nor linear. The change in resistance with strain is also nonlinear. Thus, when using a semi-conductor strain gauge it is necessary to linearize the output signal to compensate for the non-linear relationship between resistance and strain. A correction for temperature is needed as well, as these gauges “drift” with temperature change. As previously mentioned these expensive gauges are fragile and fail when exposed to water. [1]

MEMS Glass-bonded Semi-conductor Strain Gauges

This new special type of semi-conductor strain gauge fuses micro machined silicon semi-conductor strain gauges with high-temperature glass to a stainless-steel substrate. The fusing occurs at approximately 550°C, resulting in bonds that are less prone to aging or breakdown and creep than epoxy-joined systems. These gauges can operate at very low strains and can achieve gauge factors greater than 100. They also have a long life expectancy. These gauges can be manufactured in large quantities and thus are relatively inexpensive; however they are prone to failure when in contact with water. [2]

Foil Strain Gauges

Similar to semi-conductor strain gauges, foil strain gauges operate by using a change in resistance due to deformation to quantify the strain. Foil strain gauges are much less sensitive than semi-conductor strain gauges. Their gauge factor of approximately 2 is 25 to 100 times lower than that of semi-conductor gauges. An advantage of foil strain gauges it that they can be constructed to be self-compensating for temperature changes. If constructed correctly, the change in resistance due to temperature is cancelled out by the change in resistance due to thermal expansion of the object under test. Another advantage to foil strain gauges is they are readily available, come in wide range of styles, and are relatively inexpensive. [3]

Fiber Bragg Grating (FGB) Strain Gauges

In an FGB strain gauge a grating is recorded on an optical fiber. This grating reflects a narrow bandwidth of light and transmits all others. When the fiber is strained, it causes a shift in the wavelength of the reflected light, which can then be used to calculate the strain. FBGs are rugged and can be used in harsh environments. They are highly sensitive to strain and do not need to be recalibrated over the course of their long lifetime. They are also relatively easy to install and potentially low in cost; however, the signal processing equipment is expensive. [4]

Fabry-Perot Strain Gauges

To calculate strain, this type of optical strain gauge uses inteferometery to assess the change in length of a Fabry-Perot cavity contained inside the gauge. They are able to measure displacements on the micro-strain scale and can operate in harsh environments. These gauges and their signal processing equipment cost less than the FBG systems, but significantly more than foil gauges. They are not affected by electromagnetic or radiofrequency interference and have a long life expectancy. [5]

CUSTOMER REQUIREMENTS AND ENGINEERING SPECIFICATIONS

Before beginning the actual design process, we began by clearly defining the customer requirements, translating them to engineering specifications, and prioritizing them using Quality Function Deployment (QFD).

Customer Requirements

To define our customer requirements we started with the requirements given to us directly from our sponsor, and expanded them. Our list is as follows:

<u>No sensitivity loss compared to the current design</u>. This is a top priority given to us directly from our sponsor.

<u>Withstand seawater environment</u>. The new design must be able to survive in the corrosive environment in which it primarily operates.

<u>Shorter manufacturing time</u>. The current turnaround time of three to four months is both inconvenient and undesirable.

<u>Reduce cost.</u> The current cost of several thousand dollars per load cell is very high, especially when taking into consideration the short life of the strain gauge.

<u>Longer lifetime than current design.</u> The lifetime of the current design is only approximately six

months. The sponsor would like to improve this considerably.

Usable output signal. Additional signal processing equipment will need to be purchased if the output signal is not a voltage or current signal.

Geometry fits in current housing. If the new load cell can be used with the current housing, it will the save time and money that would be spent designing a new housing.

Engineering Specifications

After defining our customer requirements we proceeded to define our engineering specifications. We began by translating the voice of the customer into attributes that could be quantified. We then set targets for these attributes.

Time to manufacture. We set the target for this at four weeks, approximately 25% of the time required for the current design. This is reasonable if the load cells do not need to be sent out of the country for strain gauge installation.

Corrosion resistant material. Because the load cell operates in a saltwater environment, it needs to have a high resistance to corrosion. On a scale from 1 to 10 (1 being most resistant) we set the target at 2.

Height, width and depth. In order to fit in the current housing, the load cell needs to be sized similarly to the existing design: less than 38mm high and less than 45mm in width or depth.

Output signal type. Anything other than a voltage or current signal will necessitate the purchase of additional hardware, thus the target is voltage or current.

Average life to failure. The target of 24 months is four times longer than the current lifetime.

Water intrusion. Because water destroys the semi-conductor gauge, we set this target at $0mm^3$.

Sensitivity. The design must be at least as sensitive as the current design: +/- 5%

Cost. Our budget is $5,000, thus this is our target value. It is also much less than the cost of the current load cell.

Quality Function Deployment (QFD)

We used QFD to help guide the development of out design by prioritizing our customer and engineering requirements. We constructed a "house of quality" to do this both visually and numerically.

Weights To determine which of the customer requirements were most important we used a matrix to compare each customer requirement to every other customer requirement. We assigned a 1 to the more important of the two, and a zero to the other. We then totaled up the number of 1s for each requirement. The higher the point score, the more important the requirement is. This can be seen in Fig. 3 below.

Requirement																													Points	% of Total
Water Proof	1	0	0	1	1	0	1																						4	14.29%
Geometry fits in current mount	0							0	0	1	1	0	1																3	10.71%
Usable output signal		1						1						1	1	1	1	1											7	**25.00%**
Longer life (guage)than current design			1						1					0					1	1	0	1							5	17.86%
Reduce Cost of Single unit				0						0					0				0				1	0	1				2	7.14%
Shorter Manufacturing time					0						0					0				0			0			0	1		1	3.57%
No loss of current Sensitivity						1						1					0				1			1		1		1	6	**21.43%**
Material withstands environment							0						0					0				0			0		0	0	0	0.00%
																												Total	**28**	**100.00%**

Fig.3. Comparative weighting shows signal type and sensitivity to be the most important customer requirements.

The weighting revealed that output signal type and sensitivity were the most important requirements, followed by lifetime.
This process was then repeated for the engineering specifications. The weighting revealed that sensitivity and lifetime were the most important requirements, followed by signal type.

Requirement																																					Total Points	% of Total
Reduce manufacturing time (<=1mo)	1	0	0	0	0	0	0	1																													2	5.56%
Non-corroding material (long life in water/saltwater)	0								1	1	0	0	0	0	1																						3	8.33%
height (1.48in)		1							0							1	0	0	0	0	0																2	5.56%
width/depth			1							0						0						0	0	0	0	1											2	5.56%
Signal type				1							1						1					1					1	1	0	1							7	19.44%
MTBF (6-12mo)					1							1						1					1				0				1	0	1				6	**16.67%**
Water Intrusion (mm^3)						1							1						1					1				0			0			0	1		5	13.89%
Sensitivity (+/-5%)							1							1						1					1				1			1		1		1	8	**22.22%**
Cost								0							0						1					0				0			0		0	0	1	2.78%
																																		Total			**36**	**100.00%**

Fig. 4. Comparative weighting shows sensitivity and lifetime are the most important engineering specifications.

Benchmarking Next, we used the current load cell design as a benchmark, and evaluated it on a scale of 1-10, based on the customer requirements. It scored very low in lifetime, cost and manufacturing time. It did not score above a five in any area. This is because the customer requirements were based mostly on improvements that need to be made to this design.

Correlation The center of the house of quality contains the correlation between the customer requirements and the engineering specifications. Requirements that correlated strongly were rated a 9. Requirements that were somewhat related received a 3, and requirements that were weakly related received a 1. Unrelated requirements received no rating.

Cross Correlation of Engineering Requirements The house of quality roof is where we identified engineering requirements that were interrelated. For example, we identified time to manufacture as being strongly related to cost because generally as the machining time needed to make a part increases, so does the cost.

Targets The floor of the house of quality contains the engineering targets previously mentioned. The unit of measurement is in the top row, and the magnitude of the target is directly underneath the unit. The completed house of quality is shown below in Fig. 5.

Quality Function Development (QFD)

Key:

9 => Strong Relationship

3 => Medium Relationship

1 => Small Relationship

(blank) => Not Related

***Weights are figured on a scale of 1 to 10**

(ten being most important)

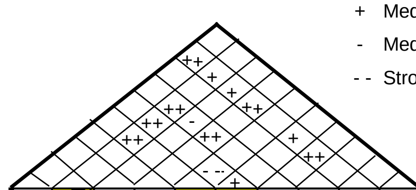
Relationships

++ Strong Positive

\+ Medium Positive

\- Medium Negative

\- - Strong Negative



Benchmark

	Weight*	Time to manufacture	Corosiveness of materia	Height	Output signal type	Avg. life to failure	Water intrusion	Sensitifity	Width / Depth	Cost		*Current Design*
Water Proof	7	1	9			9	9			1		3
Geometry fits in current mount	6			9				1	9			5
Usable output signal	10				9					3		5
Longer life than current design	8		9			9	9			1		1
Reduce cost	5	1	1	1	3	3	1	3	1	9		1
Shorter manufacturing time	4	9	1				1			3		1
No sensitivity loss compared to current device	9				3			9		3		5
Withstands seawater environment	3		9			9	9					5
Measurement Unit		mo	1-10	mm		mo	mm^3	%	mm	$		
Target Value		1.5	2	38		24	0	5	45	**5K**		
Importance Rating		3	6	5	9	8	7	10	**5**	2		
Total		48	171	59	132	177	171	102	59	129		
Normalized		0.05	0.16	0.06	0.13	0.17	0.16	0.10	0.06	0.12		

Fig. 5. Normalized values in the house of quality identify the most important engineering specifications.

According to the normalized values in the house of quality, the lifetime, water intrusion, and material corrosiveness are the most important requirements. This is not surprising because each of these requirements heavily influences three of the customer requirements.

CONCEPT GENERATION

We used several techniques to help generate design concepts, including a FAST diagram, a morphological chart, and a Pugh chart. Specific major areas we targeted for brainstorming included load cell geometry, sensor type, and waterproofing techniques.

FAST Diagram

Since concept follows function, we developed a FAST diagram to help us decompose and identify the basic functions of the load cell. This helped us visualize exactly how it operates and also prevented us from prematurely locking in our favorite concepts or “reinventing the wheel.” We began by first identifying the primary basic functions and primary supporting functions. We then expanded each function into its sub-functions. This diagram is below in Fig.6.

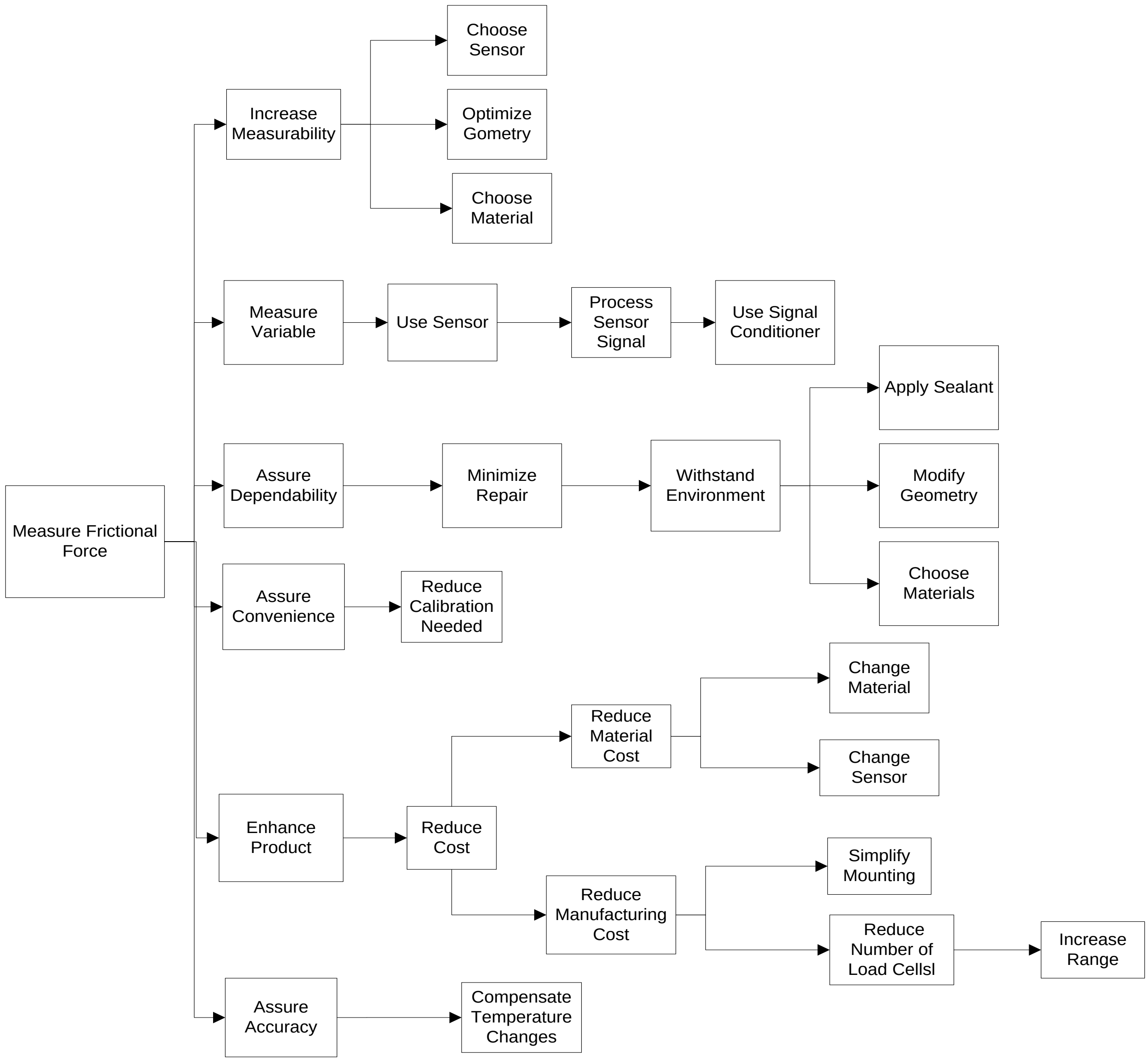


Fig. 6 The FAST diagram decomposes and identifies the basic functions of the load cell.

This diagram helped us to realize that measuring shear strain was not the only way to determine the friction force. We could also try to analyze other force dependent variables such as plate displacement.

Morphological Chart

Generating the FAST diagram resulted in a breakdown of the functions of the load cell. We organized the key functions into a morphological chart to help us brainstorm concepts for each function. First, on the left side of the chart we listed our functions. Next, we filled the chart horizontally with multiple concepts for each function. Finally, we looked vertically along the chart to identify concepts that could be feasible to combine. The morphological chart is shown below in Fig. 7.

Morphological Chart

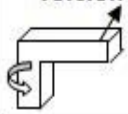
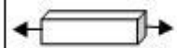
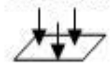
Functives						
Load Member	Torsion 	Bending 	Axial Loading 			
Use Sensor	Optical 	Foil 	MEMS 	Semi-conductor 	Magnetic Flux Detector 	Force Gauge and Spring 
Attach Sensor	Glue 	Mechanical 	Chemical 			
Improve Water Proofing	Surface Finish 	Modify Current Design 	Change Cavity Filling 	Spray Sealant 		

Fig. 7. The morphological chart organizes our concepts and helps identify possible design combinations.

Overview of Concepts

Load Member The first area we developed concepts for was how to load the flat plate and load cell. The current design uses bending. We developed concepts to load the load cell axially and in torsion.

Use Sensor We next brainstormed different sensor types we could use. We considered optical, foil, MEMS, and semi-conductor strain gauges. We also considered using magnets and a flux detector to directly calculate plate displacement. A final thought was using a laser displacement sensor which would also allow displacement to be calculated directly.

Attach Sensor Next we considered different ways to attach the sensor to the load cell. We thought of using glue or adhesive, mechanical means such as screws or bolts (although probably not feasible), or chemical methods such as soldering.

Improve Water Proofing We developed concepts for improving the waterproofing of the load cell. We could improve the surface finish for the epoxy bond, modify the geometry of the current design to better hold the epoxy, change the epoxy material itself, or add a waterproof coating.

CONCEPT EVALUATION AND SELECTION

To evaluate and narrow down our concepts we first eliminated any designs that were obviously infeasible. We then continued by choosing our sensor type. This was necessary because the sensor type, size, and its sensitivity greatly influence the load cell geometry. Once the sensor was chosen we moved on to the geometry of the load cell and the waterproofing design. We developed a Pugh chart to analyze the most promising designs and gauge how well they satisfied the customer requirements.

Sensor Selection

Our previous research had indicated that if the project budget allowed, the FBG sensor system would most likely be the best option. These sensors were the most sensitive and eliminated the need for waterproofing. However, the sensor size (2+cm) and cost of the system were two fixed constraints that eliminated this option.

The MEMS semi-conductor strain gauge was initially promising – it had a relatively high sensitivity, was very small, and was most likely affordable. Unfortunately, we would not have been able to acquire one from the manufacturer within the fixed time constraints of this project; therefore it too was quickly eliminated.

The gauge we ultimately selected is the Fabry-Perot optical strain gauge. This gauge is able to measure deformation on the micro-strain scale and is relatively small (less than 1cm long). The cost of several gauges and the signal processing equipment was also within our budget. This gauge can operate in harsh environments; however, it does require waterproofing.

In addition to the Fabry-Perot gauge, our sponsor also requested we choose a foil strain gauge. Although this type of gauge is much less sensitive than the optical gauge, it is small and very inexpensive. Additionally, our sponsor already has the signal processing equipment necessary to utilize such a gauge. After a considerable amount of research, we selected a Platinum-Tungsten alloy foil gauge. This gauge has the highest gauge factor we could find – just under 5. It was a bit more expensive than many foil gauges, but approximately twice as sensitive.

Geometry Selection

We considered several different geometric options for applying the force to the load cell. First we considered applying a torsional load. To do this, we would need to modify the housing that the load cell sits in, in order to incorporate a moment arm. Recently we were informed by our sponsor that the housing design was fixed, and we cannot modify it, therefore this concept was eliminated. This design is shown below.

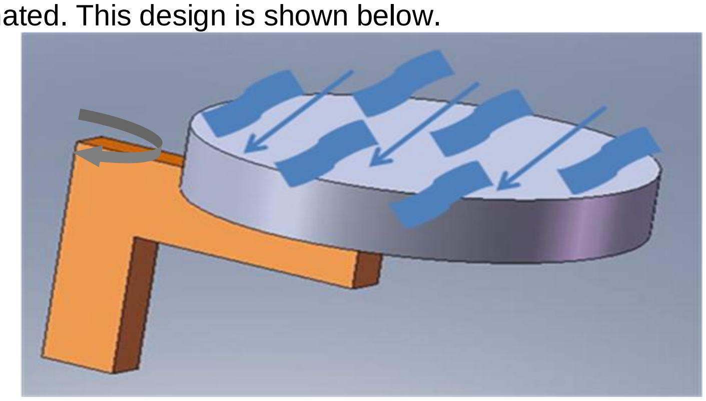


Fig. 8. The torsion based load cell is not feasible due to the housing constraints.

Next, we analyzed applying the force in such a way that it would induce a bending stress on the load cell that we could then analyze to determine the friction force. Preliminary analysis indicated that this stress would be even smaller than the shear stress in the current design. The only sensor available that could possibly measure a stress of this magnitude accurately was the FBG system. Since we had already ruled out this sensor type, we were forced to eliminate this concept as well. A sketch of this concept is shown in Fig. 9 below.

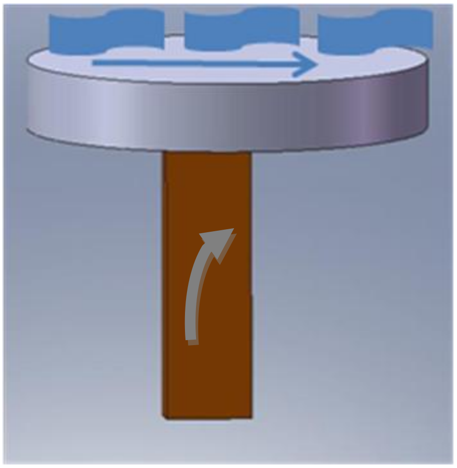


Fig. 9. The bending stress based load cell is not feasible due to our inability to measure the small magnitude of the stress.

The next concept we evaluated was an axial tension design. By changing the orientation of the load cell we would then measure axial stress instead of shear stress. One potential problem was that the weight of the plate would induce a bending moment on the load cell that would be large in comparison to the axial stress. Additionally, this design would involve modifying the load cell housing. As previously mentioned, our sponsor had decided that this was not an option, thus we eliminated this concept. This design is illustrated below in Fig. 10.

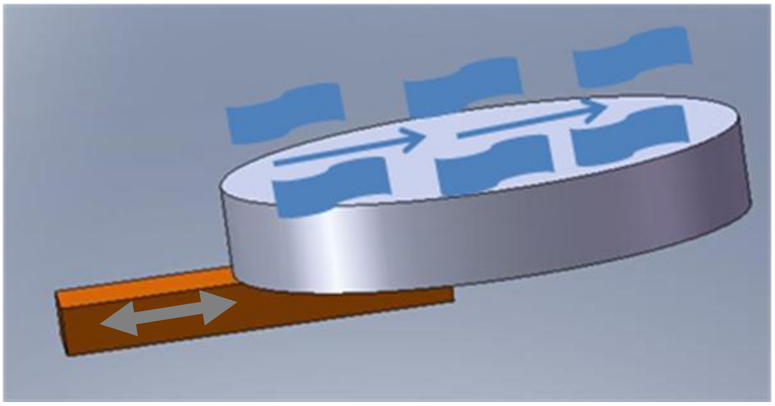


Fig. 10. The axial tension based load cell is not feasible due to the housing constraints.

The final geometry we evaluated was an optimization of the current design. This was strongly recommended to us by our sponsor because it is already proven to work. We can modify this basic geometry to maximize the shear stress in the foil web as well as accommodate the size of a foil or Fabry-Perot strain gauge. Of course, this design meets the requirement of fitting into the current housing. Because of these reasons, and the recommendation by our sponsor, we chose this geometry. The existing geometry is shown below in Fig. 11.

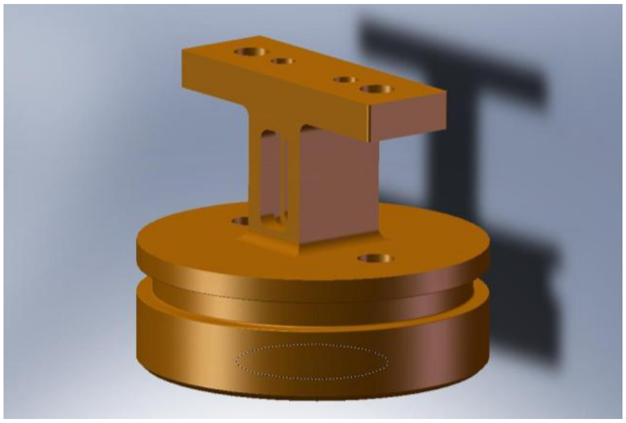


Fig. 11. Modifying the existing geometry is the most feasible solution.

Waterproofing Design

To improve the waterproofing of the design we could improve the surface finish for the epoxy bond. This could be accomplished by simple washing, a short acid bath, or improved machining. All of these are viable options and at least one will be included in the final design.

Another concept was to alter the geometry of the current epoxy cavity to better hold the epoxy in place. This could be accomplished by making the cavity more conical or sandwiching the epoxy in place with a small plate. It is possible that these changes could alter the stiffness of the overall design, thus they would have to be further analyzed with FEA to determine if they could be implemented.

Additionally, we could change the epoxy material itself. It is possible we could utilize an epoxy that would form a better bond with the beryllium copper, or be more flexible and less prone to cracking or dislocating. Also, another approach is to be sure the potting material we use expands rather than contracts when it hardens.

Finally, another concept is spraying the epoxy cavity with a waterproof coating. This is a simple and inexpensive way to increase the waterproofing. It would also have a negligible effect on the stiffness of the load cell.

Pugh Chart

We developed a Pugh chart to further analyze the strength of these concepts. We judged each of the concepts listed based on whether or not they could meet the customer requirements and be feasible with the available time and resources. A strong positive was given a +1, a strong negative a –1. The customer requirements were weighted based on importance, and the two concepts with the highest positive score were selected. The Pugh chart is below in Fig. 12.

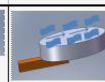
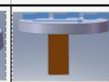
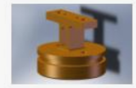
Customer Requirement	Weight	 Torsion	 Axial	 Bending	 Current Geometry (optics)	 CurrentGeometry (foil)
Water Proof	7	-1	-1	-1	1	0
Geometry fits in current mount	6	-1	-1	0	0	0
Usable output signal	10	0	0	0	0	0
Longer life than current design	8	0	0	0	1	1
Reduce cost	5	1	1	1	1	1
Shorter manufacturing time	4	1	1	1	-1	-1
No sensitivity loss compared to current device	9	0	0	0	1	-1
Withstands seawater environment	3	-1	-1	-1	1	0
	Total +	2	3	3	4	3
	Total-	3	3	4	1	1
	Total	-1	0	-1	3	2
	Weighted Total	-7	-7	-1	28	0

Fig. 12. The Pugh chart shows the best concepts are optimizing the current geometry and using the Fabry-Perot or foil strain gauges.

SELECTED CONCEPTS

Based on the Pugh chart analysis, we chose to optimize the current design geometry; doing this should allow us to meet all of our customer requirements.

The first customer requirement is that the new load cell is waterproof. We determined this could be accomplished by re-designing the groove for the potting material and by changing the potting material itself. Adding a waterproof spray coating may help as well. The second customer requirement is that the load cell fits inside the current housing. By keeping the load cell base and height the same and altering only the pillar structure, we knew we would guarantee this. The third customer requirement is a usable output signal. Our sponsor currently possesses the equipment necessary to analyze the signal from a foil strain gauge. A signal conditioning box is part of the Fabry-Perot package; therefore we will have the ability to analyze that signal as well. Because we will improve the waterproofing of the load cell, we will satisfy the next customer requirement by increasing the lifetime of the cell. Changing the gauge from the semi-conductor based type to either the foil or Fabry-Perot gauge both reduces cost and shortens manufacturing time. Changing the geometry to maximize the strain in the web should offset the reduced sensitivity of the new gauges, resulting in a new load cell that is as sensitive as the existing load cell. Finally, using a corrosion resistant material for the load cell body and potting material will allow the load cell to withstand the seawater environment. We believe our single prototype load cell is capable of operating over the entire range of water flows, and can utilize either the Fabry-Perot or foil strain gauge.

ENGINEERING ANALYSIS

To further develop and refine our design we conducted both quantitative and qualitative engineering analysis. Our quantitative analysis consisted mainly of MATLAB and Finite Element Analysis (FEA). Our qualitative analysis consisted of Failure Modes and Effects Analysis (FMEA), Design for Manufacturing and Assembly (DFMA), and Design for the Environment (DFE).

MATLAB Fluid Analysis

We began analyzing our design by finding the magnitude of the fluid forces on the load cell. We used MATLAB software to solve the fluids equations necessary to estimate these values. The MATLAB code we used to calculate this, as well as the assumptions we made can be seen in Appendix E. Figure 13 below shows the forces we anticipate vs. the water flow speed.

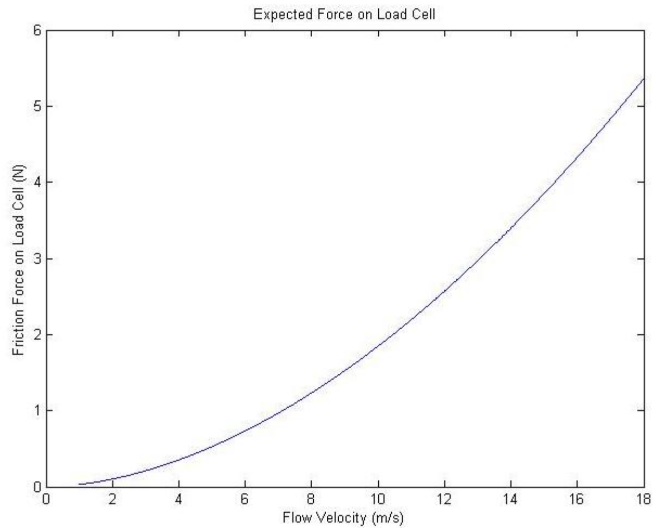
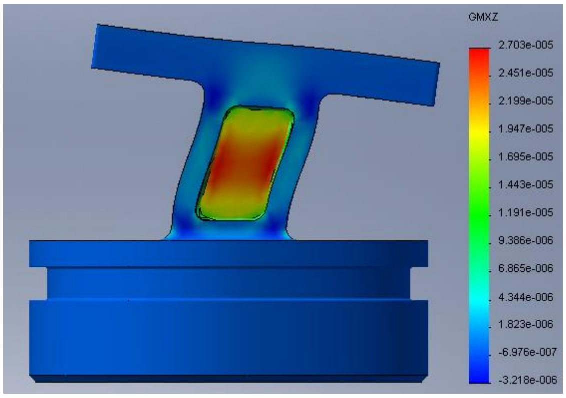


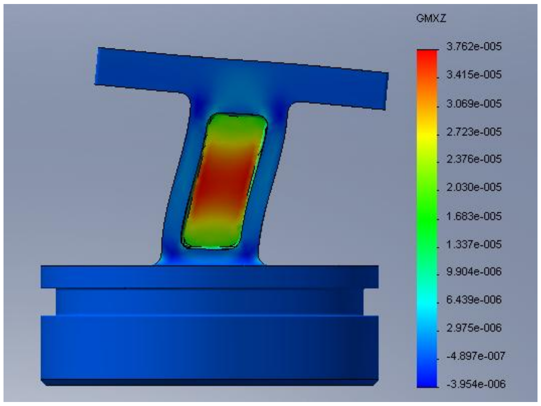
Fig. 13. At a flow speed of 18 m/s, the fluid to exerts a force of ~5.5N on the load cell

Finite Element Analysis

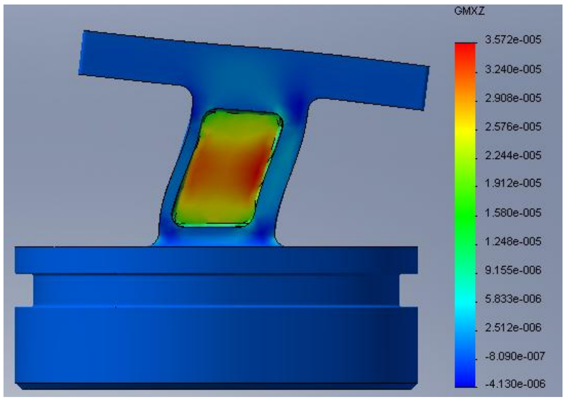
We next studied the effects of small geometry changes on shear strain using COSMOSWorks. Not surprisingly, the FEA showed that increasing the pillar height and thinning the pillars increased the shear stress in the foil. Widening the pillars decreased the shear stress in the foil. Asymmetrical pillars did not induce much bending stress in the pillars, contrary to what we had expected.



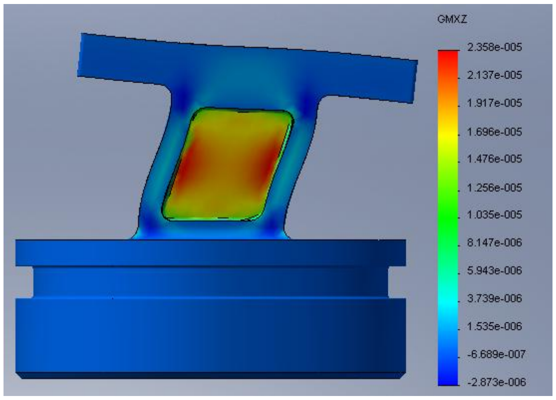
Existing Load Cell: Max shear strain = 27με



Taller Load Cell: Max shear strain = 38με



Asymmetrical Load Cell: Max shear strain = 36με

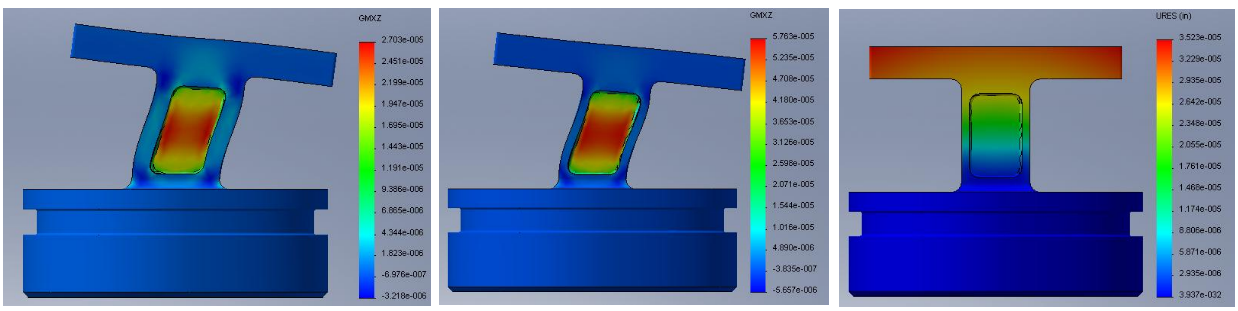


Wider Load Cell: Max shear strain = 24με

Fig. 14. COSMOS analysis of different load cell geometries. Shown are the effects of making the foil region wider and altering the pillar dimensions.

To better define the optimized geometry, we conducted further FEA analysis into the effects of changing the section shape and angle of the pillars, and aspect ratio of the web. This included the geometry of the epoxy cavity and size and shape of the foil. We analyzed the effects of changing the groove shape from rectangular to a dovetail shape, to help improve the waterproofing of the cell. As expected, this had little effect on the strain in the foil. We also investigated angling the pillar walls toward each other so that the entire cavity was shaped like a dovetail. While possibly better for waterproofing, this change lowered the strain in the foil. In addition, we used Hypermesh to identify the location and direction of the principal stresses, as well as confirm our COSMOS results. Because shear is the dominant mode of deformation here, we found that the principal stresses act on a 45°angle, as expected. This is important to know for proper foil strain gauge selection and mounting of the strain gauge. The full results of this iterative analysis are located in Appendix A.

FEA Results Based on our analysis, we determined that the simplest and most effective way to increase the strain in the foil (while staying within our height constraints) was to thin the pillar walls. In addition to being simple and effective, this solution allowed us to re-manufacture existing load cells, rather than begin from scratch. Our initial plan was to machine the pillars to half of their original thickness from the center out, increasing the size of the web. This was not the most desirable solution because it involves very difficult, precise machining. After further consideration, we realized we could achieve the same result by machining the pillars from the outside in; this is a much simpler process. According to our FEA, this geometry change will more than double the amount of strain in the foil, but the load cell will still be well within the displacement constraints.



Existing Load Cell:
Max shear strain = 27με

New Load Cell:
Max shear strain = 58με

New Load Cell:
Max displacement = 3.5e^{-5}in

Fig. 15. COSMOS analysis shows the new load cell has twice the shear strain of the current load cell. Despite this, the displacement is well below the maximum allowed.

Because the resulting displacement is so much smaller than what is allowed by the housing constraints, we considered thinning the pillars even more. However, due to the groove on the inside of the pillar walls, we were prevented from machining any further from the outside. As previously mentioned, machining from the inside out and creating a larger foil would be very difficult. We used COSMOS yet again to analyze the benefit of thinning them further. Decreasing the pillar thickness to ¼ of the original thickness only resulted in a shear strain increase from 58με to 65με. Thus we concluded the small benefit did not justify the extra machining time and risk of damage to the part.

Material Analysis

There were three major material choices we made for this load cell: body of the load cell, gauge adhesives, and potting material.

Load Cell Body Because we were able to re-manufacture the existing load cells for our new design, this choice was actually made for us. However, the Beryllium Copper alloy currently used satisfies our customer requirements. It is corrosion resistant and has satisfactory material properties. It is relatively easy to machine and can be recycled.

Gauge Adhesives The Fabry-Perot strain gauge adhesive was another choice that was not completely our own. Adhesive was included in the package we purchased from the supplier. It bonds well with metal and is designed to be used in harsh environments. Since it was specifically recommended by the manufacturer and suitable for our needs, we chose this adhesive.

For the foil strain gauge we knew we needed an adhesive that would not add a lot of stiffness to the web, could be easily applied in a thin coat, and would bond well with metal. We found a two part epoxy that fits this description, and is moisture and chemical resistant. In addition, it can be cured at room temperature, although post-curing is an option.

Potting Material This was perhaps the most important material choice because it has been the source of failure in the existing load cell. We knew to prevent water intrusion we needed a waterproof material that would bond with the metal and not shrink excessively as it dried. We also wanted it to be as compliant as possible after it dried so it didn't add to the stiffness of the load cell. We found a polyurethane material with these characteristics. When dry, the stiffness is similar to that of a kneadable art eraser.

Suppliers A list of the suppliers for each of these materials is shown in the table below.

Material	Supplier
Fabry-Perot Strain Gauges and Adhesive	Fiso
Tooling and Solvents	McMaster-Carr
Foil Gauges and Adhesive	Vishay
Potting Material	Epoxies, Etc…

Table 1. Load cell material and suppliers

Failure Modes and Effects Analysis (FMEA)

We developed a FMEA chart to help identify potential failures in our design, their effects, and ways to prevent them. Our design has very few parts which fall into three categories: the load cell body, gauges and adhesives, and potting material.

Since the failure mode of the current design has consistently been water intrusion, it came as no surprise that the highest Risk Priority Numbers we calculated (90 and 100) were associated with water intrusion into the load cell. This indicated that we needed to pay particular attention to our waterproofing methods in the new design. We were able to reduce this failure risk by improving the surface of the load cell before adding the potting material, changing the potting material, and visually inspecting the potting material after it sets. The full FMEA chart is located in Appendix B.

Design for Manufacturing and Assembly (DFMA)

We began this analysis by identifying high cost production processes associated with the manufacturing and assembly of the original load cell. The original load cell has a very high production cost associated with having the semi-conductor strain gauge installed by the manufacturer (unknown) in France. To address this problem we are changing the type of strain gauge used in the load cell. The new gauges can be installed by technicians at the University much cheaper than having the work sent overseas. An added benefit is that the turn-around time for installation is also greatly reduced.

Another high cost in the production of the load cell is the machining work that is necessary to produce the cell. The cell is produced from round bar stock and the geometry requires a great deal of material be removed. This results in a relatively large amount of waste material per load cell and a great deal of time and money to produce (due to labor costs). We have been working on cleaning the load cells by using solvents to remove the existing potting materials and gauge adhesives. If these are successful it will no longer be necessary to produce a new load cell body each time a strain gauge quits functioning. Instead the dead cell can be cleaned and a new gauge installed. With the cost of solvents being significantly less than the labor costs associated with building a new cell, this has an enormous impact on cost due to recycling.

Another important aspect of our load cell is that it is a specialty item with a small market. This means that there are very few end users and mass production is not likely in the future. For completeness, we will briefly look at a solution to producing this load cell design in bulk. It is our opinion that the best solution to mass production would be to cast a blank and then finish machining the cell. Machining would be necessary because of the tight tolerances of the web thickness, a tolerance that simply could not be produced reliably with a casting process. In addition to that, surface roughness, or lack thereof, in the gauge region is also an important consideration, and beyond the means of many casting operations. The casting process would certainly reduce the amount of waste material. However, it also has its drawbacks. Measures to eliminate porosity in the web region would be essential as a hole in the web may render it useless. This of course requires strict quality control measures and increases the cost per casting. Additionally, although these measures would reduce the production time for the load cell body, the real bottleneck in the production operation would be the precision installation of the strain gauges. This is a process that cannot be automated with current production techniques, therefore estimating the labor involved is difficult. Therefore, we are unable to calculate an accurate cost estimate comparing the machining operation to the theoretical casting operation.

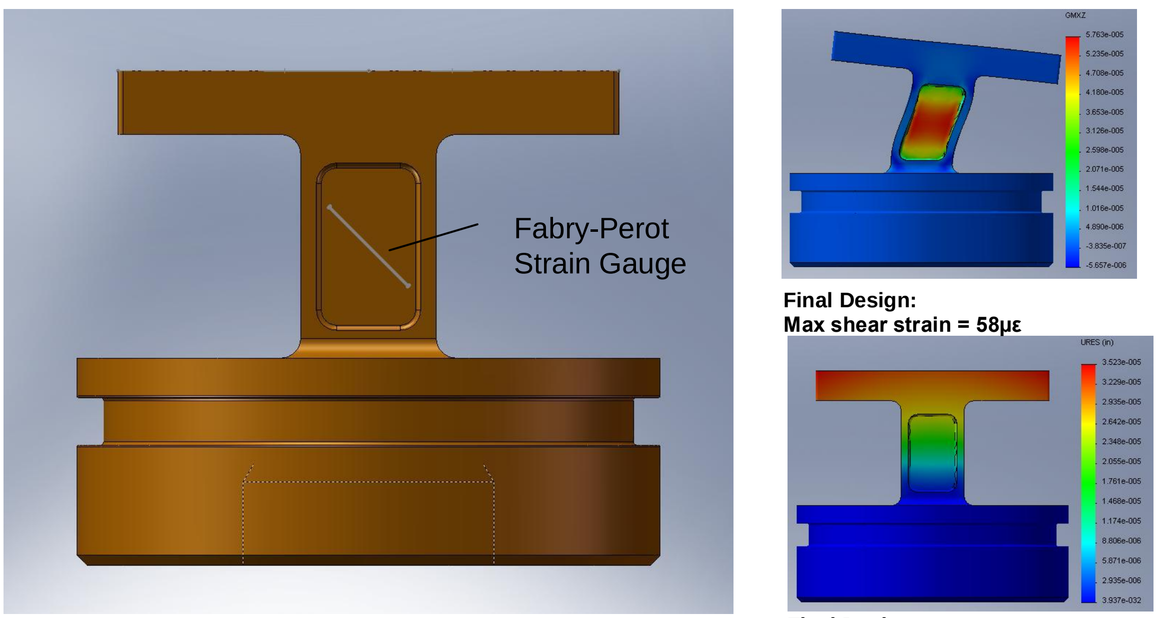
Due to the many fixed constraints of the geometry of the load cell and the small number of parts, Design for Assembly System (DFAS), Design for Part Handling (DFPH), Design for Part Insertion (DFPI), and Design For Joining (DFJ) are irrelevant to this project.

Design for Environment (DFE)

Because of the very low production volume of our load cell (less than 10 per year), our environmental impact is very small. Additionally, the new design recycles scrap load cell bodies from the previous design, eliminating the need for procurement of metal stock. At the end of our product's gauge life, the gauges can be replaced and the load cell reused. The only waste is the old gauge, potting material, and solvents needed to remove them. Should the load cell body itself fail, it can easily be recycled. Despite all of this, we used "Eco-indicator for Designers 99' [8] to determine the environmental impact of our product. Specifically we were interested in which life cycle phase impacted the environment the most so we could identify areas where we could improve. For our product, the benchmark for ecological improvement was the existing design. When comparing the two designs, we found that the life cycle phase for each load cell was the same with the exception of assembly, and end of product life. This analysis is shown in Appendix C. The new design has a smaller ecological impact mainly due to the recycling and reuse of load cells. Other design changes had a negligible effect on the ecological impact of the product. Considering the case where mass production would be necessary, modifications would need to be made for ecological considerations in our manufacturing process. Optimization of production techniques, end of life systems techniques and material use techniques would have the biggest impact in a large volume production. Physical optimization can be done by further increasing the water resistance of the potting material, using better sealing techniques or switching to a strain gauge that is insensitive to aqueous environments. Production and material optimization could be accomplished by switching to casting and finish machining methods for the load cell body. This would greatly decrease the amount metal wasted.

FINAL DESIGN

Our final design is a modified version of the current load cell, produced by re-manufacturing some of the existing load cell bodies. The new load cells will be fitted with either Fabry-Perot optical gauges or Platinum-Tungsten alloy foil gauges. They will be sealed with a polyurethane potting material. Our design meets all customer requirements.



Final Design:
Max shear strain = 58με

Final Design:
Max displacement = 3.5e^{-5}in

Fig. 16 SolidWorks and COSMOSWorks models of final design

Detailed dimensions of our load cell can be found in the engineering drawing in Appendix D. The Bill of Materials (BOM) for our load cell is shown below.

Qty	Part Description	Purchased From	Part Number	Cost (each)
1	End Mill, 0.500” 0.060R, Carbide	McMaster-Carr[1]	2851A264	$64.70[5]
1	End Mill, 0.125” 0.020R, Carbide	McMaster-Carr[1]	2851A212	$17.19[5]
1	Epoxy Solvent, 1 pint	McMaster-Carr[1]	7532A12	$17.21[5]
10	Platinum-Tungsten strain gauge, Dual Shear pattern, polyimide back, encapsulated	Vishay[2]	J5E-NC-S4217-350/S	$26.55
1	Gauge Installation Kit, M Bond AE-10 epoxy	Vishay[2]	GAK-2-AE-10	$287.00
1	Fiber Optic Signal Conditioner, Single Channel	Fiso Technologies[3]	FTI-10	$2995.00
3	Strain Sensor, Fiber Optic, ±1000με	Fiso Technologies[3]	FOS-N-BA-C6-F1-M2-R1-ST	$195.00
1	Polyurethane, Resin, Catalyst, 1 quart each	Epoxies, Etc…[4]	20-2355	$152.70
			Total=	$4384.30

[1]http://www.mcmaster.com
[2]http://www.vishay.com
[3]http://www.fiso.com
[4]http://www.epoxies.com
[5]Price Includes shipping charges

Table 2: Bill of Materials for final design.

MANUFACTURING

A fixture already existed to aid in the manufacture of the load cell body. This fixture helps hold the part in place while machining. The redesigned load cell geometry is very similar to the existing load cell geometry. Because of their similarities we were able to use load cells from the old design and augment their geometry to fit the new design. This step saves considerable time and cost as our design does not need to be created from raw material. To thin the pillar walls of the existing load cell, a single operation was performed (per pillar). Only one tool was required to remanufacture our part. The operation was performed on a milling machine. (A 0.500” carbide end-mill was used with the appropriate corner radius.)

The next step in the manufacture of our load cell was to apply the strain gauge. As previously mentioned, our design can accommodate both a Fabry-Perot strain gauge and a standard foil gauge. In order to apply our new gauges, we first had to remove the old gauges still attached to the newly modified load cells.

Cleaning the cells was critical for us to successfully apply the strain gauge to the new load cells. Cleaning consisted of two important steps, adhesive/epoxy removal and surface preparation. The first step was to remove the existing materials using an epoxy solvent. The solvent we selected was effective at removing the adhesive used to install the strain gauges that were currently mounted on the load cell; however it was not effective at removing the polyurethane potting material. This was accomplished with needle-nose pliers, and a dremel tool. Once this was done satisfactorily, additional scraping and sanding was necessary to remove residual deposits on the gauge surface, as well as to prepare it for mounting our strain gauges. Finally, a cleaner and degreaser was used to prepare the surface for the adhesive that was used to mount the strain gauge to the load cell. Proper cleaning methods were absolutely necessary for a good bond between the strain gauge and load cell to be made.

Once the surfaces were properly cleaned and prepared, we were ready to mount our strain gauge. The Fabry-Perot strain gauge documentation indicated that the fiber optic cable should not be bent to a radius smaller than 10mm. To avoid this situation, we drilled an additional hole in the load cell that allowed for a more favorable cable route. This design change is indicated in ECN1 in Appendix F. Finally, the adhesive was applied and the strain gauge was mounted according to the manufacturer's instructions.

After finishing the manufacturing for our Fabry-Perot prototype, our sponsored indicated that he was pleased with the results and would rather us spend time carefully calibrating the Fabry-Perot prototype rather than assembling a prototype using the foil strain gauges we had purchased.

TESTING

In order to test and calibrate our load cell, we built a small fixture on which to mount the gauge. This fixture was then clamped securely to a table. A small hook was fashioned from a paperclip from which to hang various masses. We chose masses of 20, 50, 100, 200 and 500g to cover the entire range of forces we expected to see from our earlier analysis. The load cell exhibited an approximately linear relationship between force and strain. Our signal conditioner also provides a voltage signal output. We used a voltmeter to measure this output as well in relation to applied mass (force). Plots of both can be seen below.

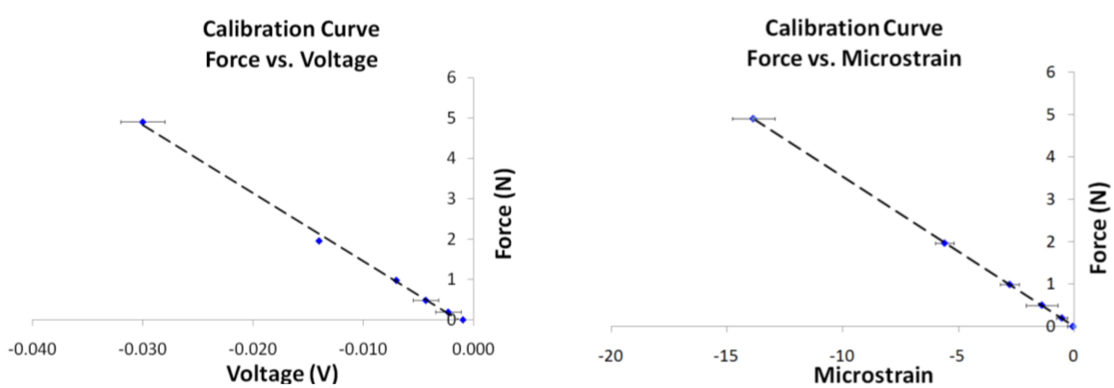


Fig. 17. Calibration curves showing linear relationships exist between Force and Microstrain, and Voltage and Microstrain.

From this testing, we believe our new load cell can accurately measure down to a force of 50g. This corresponds to a force of 0.49N, which, from our graph in Fig. 13 on pg. 15, corresponds to a water velocity of just over 5m/s. Our sponsor requires that this gauge be accurate for water velocities from 6-18 m/s, thus this new design should be acceptable for his uses.

Professor Ceccio also asked us to look for a hysteresis effect on the load cell when loads were applied and then removed. In order to do this, we first did set our output to zero on the signal conditioner and then performed our tests as described above. After unloading the load cell we checked the reading on the display. Over multiple tests the load cell returned to its initial reading ±0.2με. This leads us to conclude that there is no appreciable hysteresis effect on our load cell to be accounted for.

One more matter of interest is the manner in which the load cell was tested. In use, the load cell is mounted in a housing with a flat place attached to the top of it. The loads are actually applied to that plate, which has the effect of applying a larger moment to the web, where the strain gauge is located. Thus, the results shown above are useful for showing a general relationship between force and selected output, but that our results should not be used to accurately infer a load from a given value of voltage or microstrain. Additional calibration to account for the moment is needed.

The final area of testing is to use our design in actual experiments to see if the water proofing of the design is adequate. This testing may not be completed before the project end date. If our design works extremely well it would take several weeks of testing to determine that our gauge is very safe from the ravages of water.

DISCUSSION FOR FUTURE IMPROVEMENTS

Although our design has succeeded in achieving the engineering and customer requirements we set for it, we have identified a few key areas for further improvements in the future.

Geometry Improvements

One improvement that could be made would be to widen the distance between the pillars. Although we originally believed we had enough space on the foil web to mount our gauge at a 45° angle, we found we did not leave enough workroom to do this effectively. Our resulting angle was closer to 50-55°. Widening the distance between the pillars would allow the fiber optic cable to be more easily mounted at the optimal angle, maximizing sensitivity. This change would change the aspect ratio of the web and decrease the amount of strain in the foil, but only to a small extent (see appendix A).

An additional improvement that could be made would be to modify the shape of the groove that holds the epoxy in place. We suspect that changing the groove to a half dovetail shape could better hold the epoxy in place, and thus better stop the leakage of water into the load cell. This would then increase the life of the load cell. Changing the shape of the groove would have a minimal impact on the shear strain in the web, as seen in the FEA in Appendix A.

Sensor Improvements

To increase our ability to sense the total shear we would use two optical sensors to mimic the set-up of a foil shear strain gauge. This could be done by mounting a Fabry-Perot gauge on each side of the foil web; the two gauges should be perpendicular to each other and at a 45° angle from the horizontal. Using the two signals should give a more accurate result. Additionally, the two gauges could still be used individually. Thus, if one gauge failed, the load cell could still be used until the second failed as well. The expected cost of implementing such a setup would be approximately $7000.00 for the multi-channel signal conditioner.

CONCLUSIONS

During the course of this project, we have redesigned the shear-stress load cell provided to us by Professor Steven Ceccio. With a simple modification to the geometry of an existing load cell we doubled the shear strain in the foil web. This allowed us to replace the highly sensitive yet costly semi-conductor based strain gauge with a Fabry-Perot strain gauge. This new gauge is less expensive and can be installed relatively easily in-house. We have also identified a new waterproofing material that is very compliant and will chemically bond with the load cell, hopefully increasing the water resistance. Our prototype meets the sensitivity needs of our sponsor and should have a significantly longer lifetime – key requirements identified by our QFD analysis. The new design also greatly reduces cost and shortens manufacturing time.

ACKNOWLEDGEMENTS

The authors would like to thank all professors and students who contributed to this project and paper. We are appreciative to Dr Stephen Ceccio for his sponsorship, assistance and guidance on this project, and a special thanks to his graduate student Brian Elbing for his assistance in testing and prototype assembly. We would also like to thank Dr. Katsuo Kurabayashi for his insight and direction concerning MEMS strain gauge technology. We would also like to thank Kent Pruss for his assistance, suggestions and tools for the manufacturing of the prototype. Additionally, we would like to thank Mr. Josh Alden for the use of his machine shop facilities. Finally, we would like to thank Professor Bogdan Epureanu for his thought provoking questions, guidance, suggestions and constructive criticisms.

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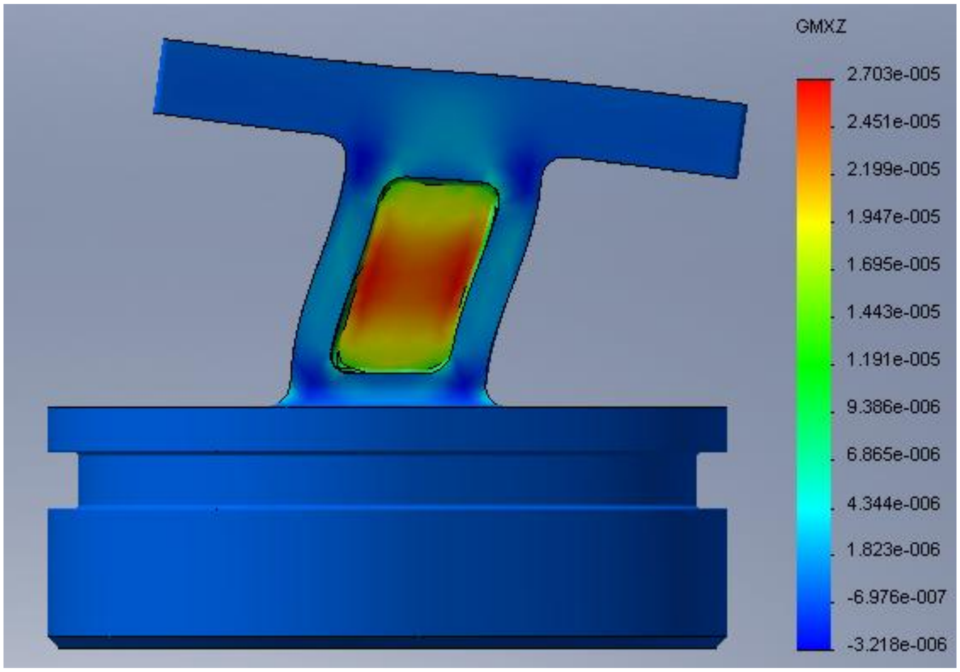
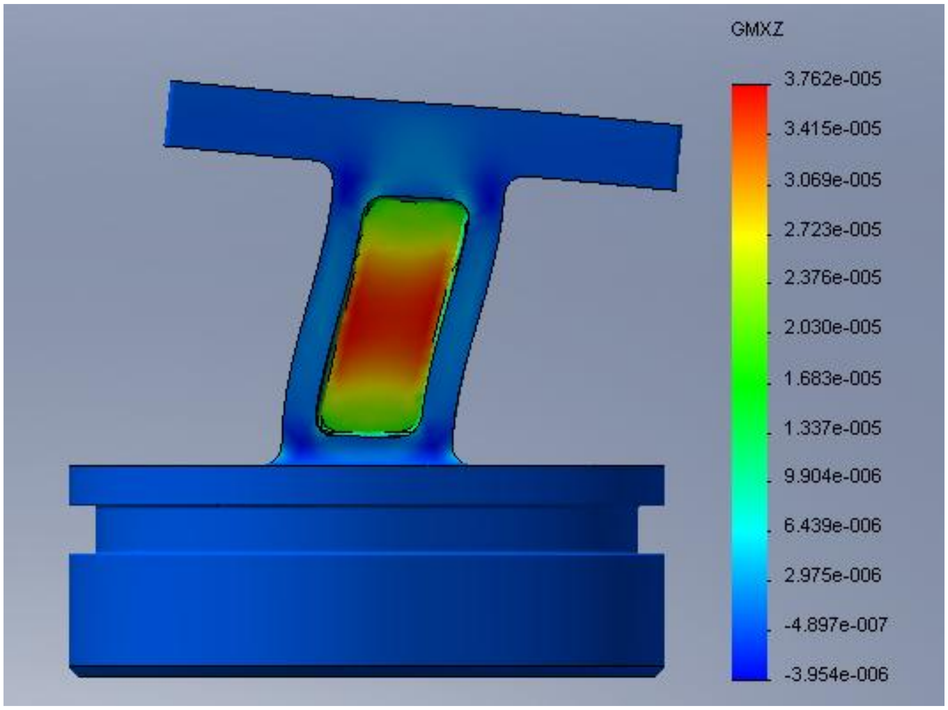
APPENDIX A: Results of Detailed CAD Analysis of Geometry Changes

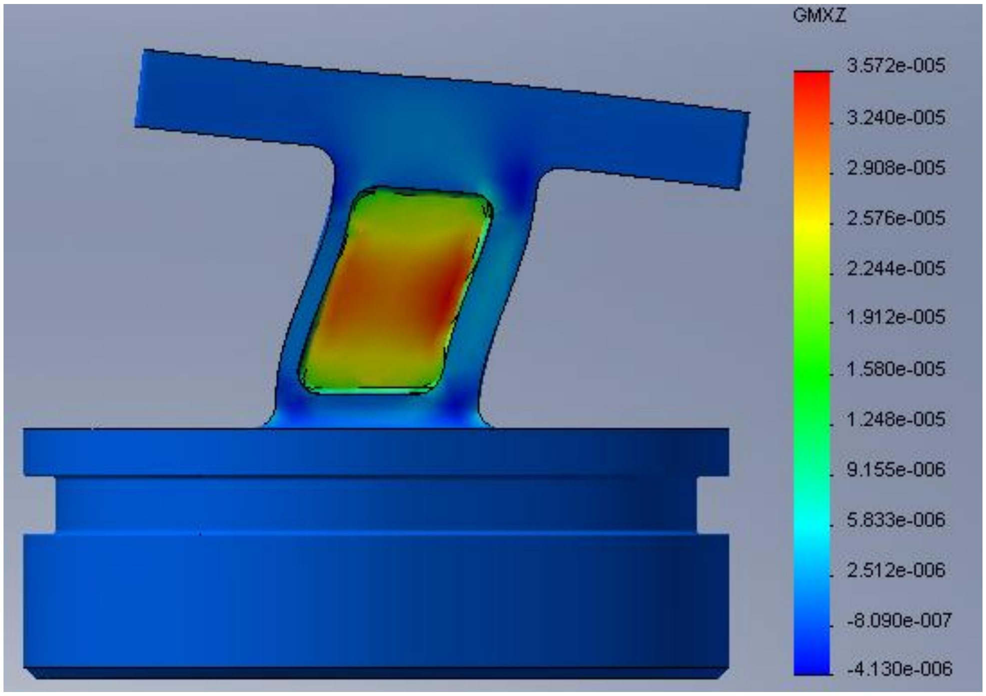
All CAD analysis was done using a horizontal force of 1.2lbf applied to the top surface of the load cell. This amount of force was used because it is the maximum force we expect our load cell to experience in testing. The same restraints were used for all cases.



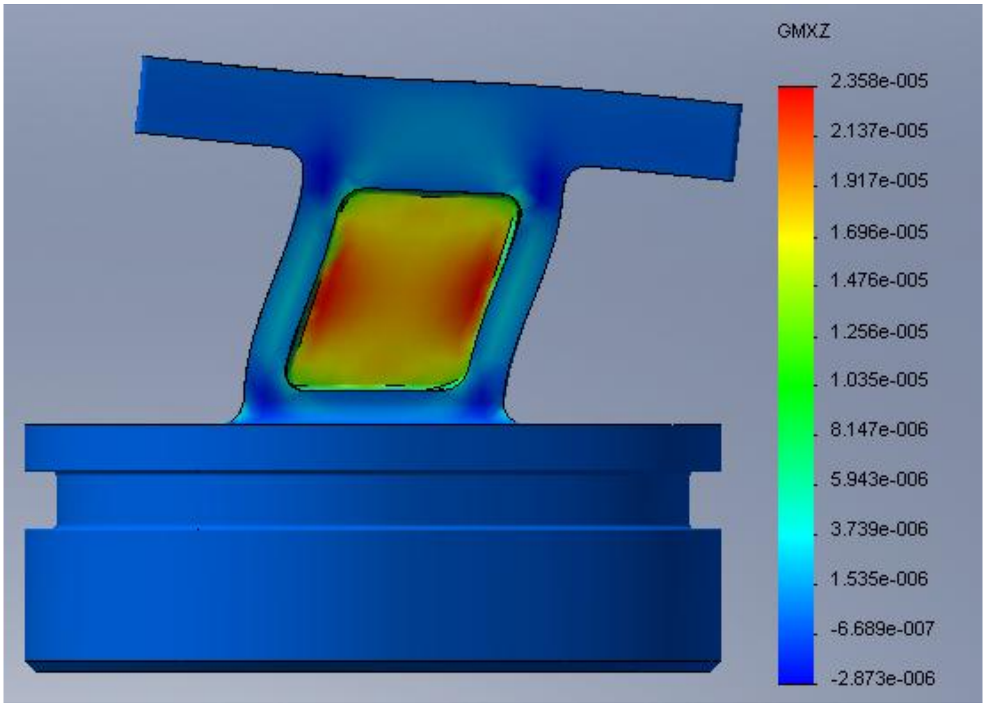
Existing Load Cell: Max shear strain = 27με



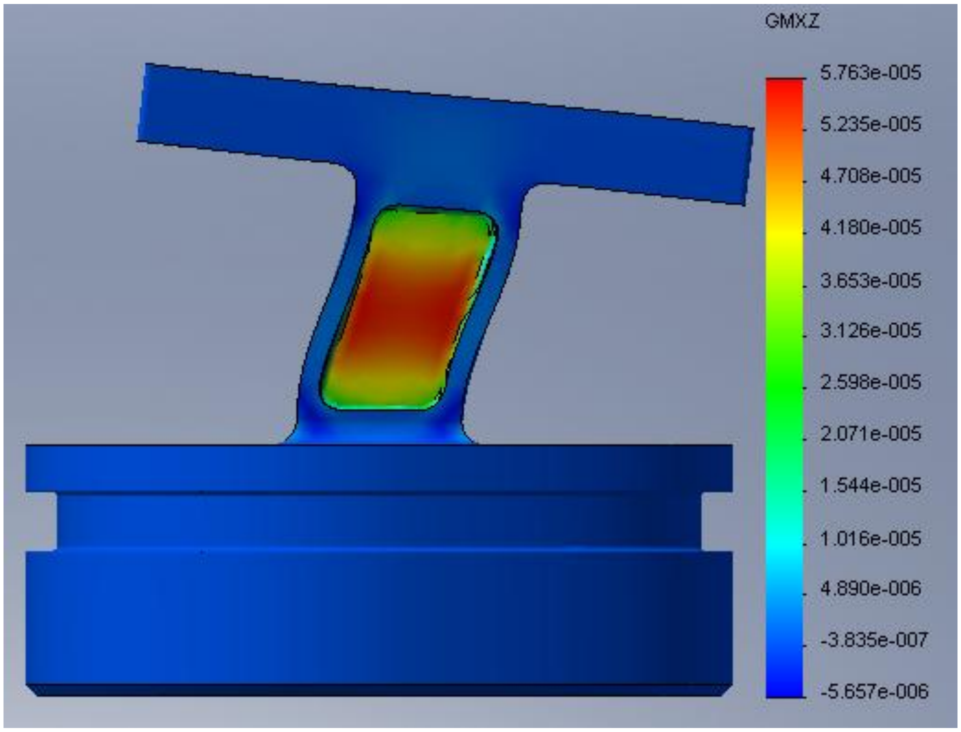
Taller Load Cell: Max shear strain = 37με



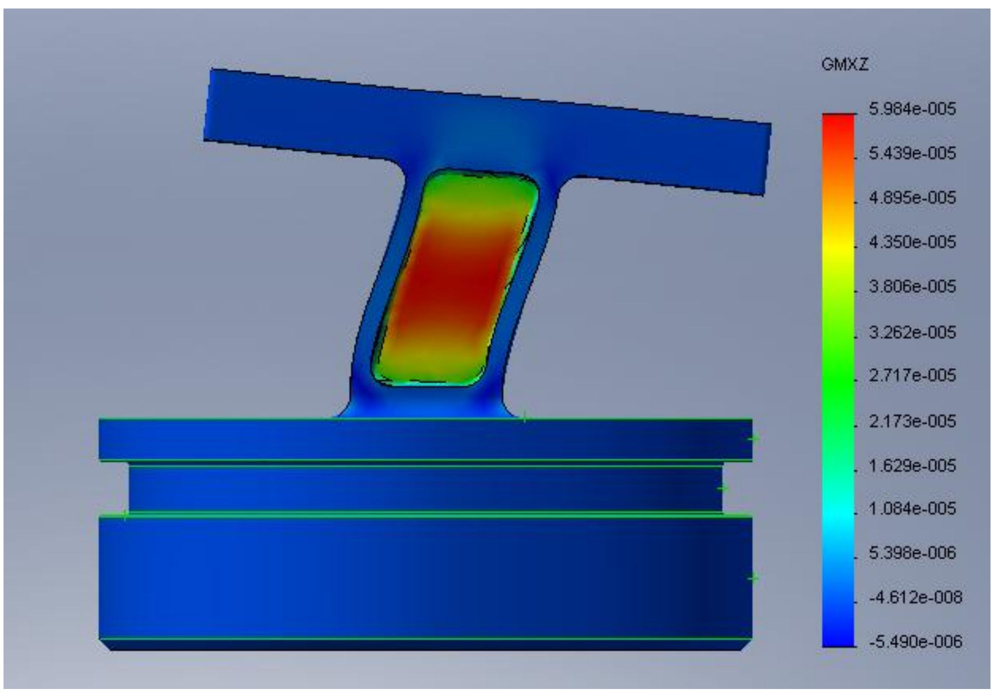
Asymmetrical Load Cell: Max shear strain = 36με



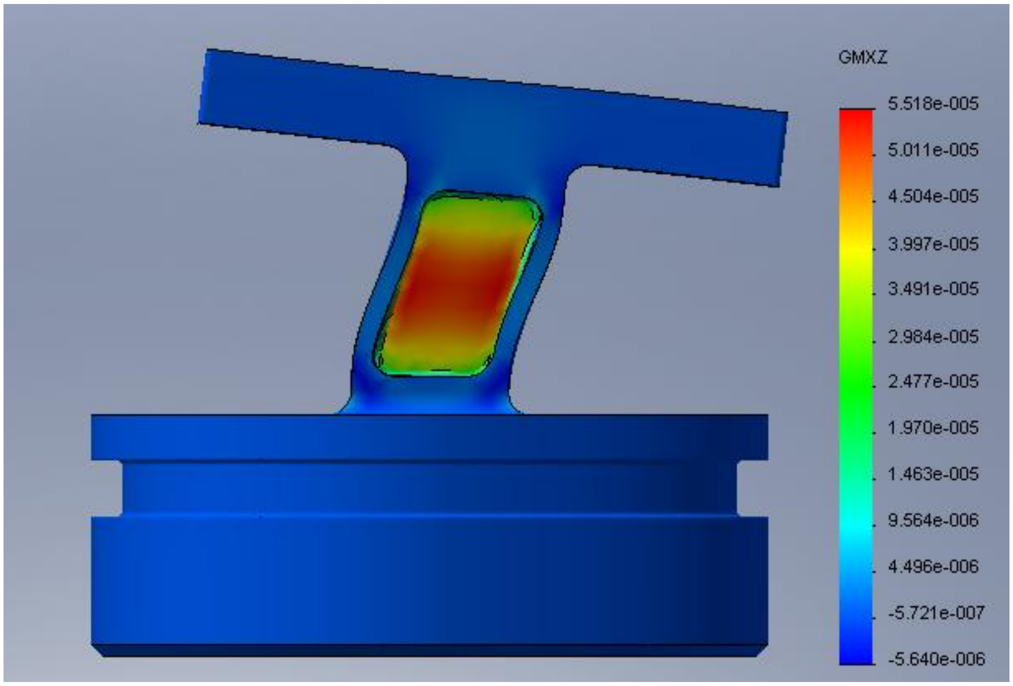
Wider Load Cell: Max shear strain = 24με



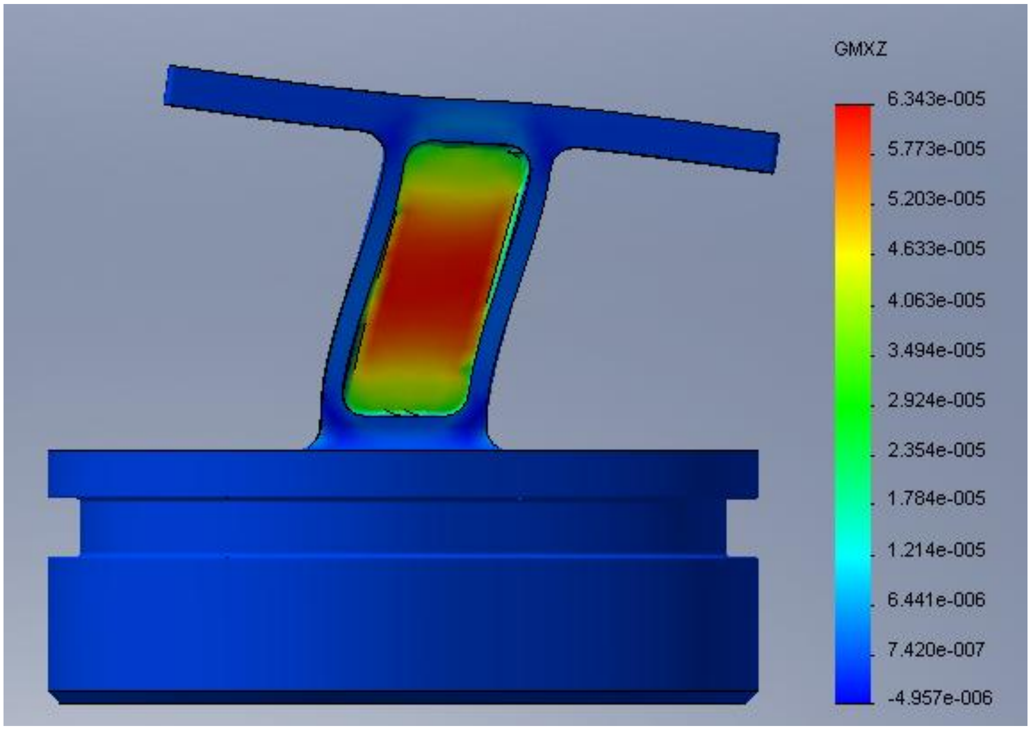
New Load Cell (Thinned Walls): Max shear strain = 58με
This geometry had the highest shear strain thus far. From this point on, all changes were made to this new geometry.



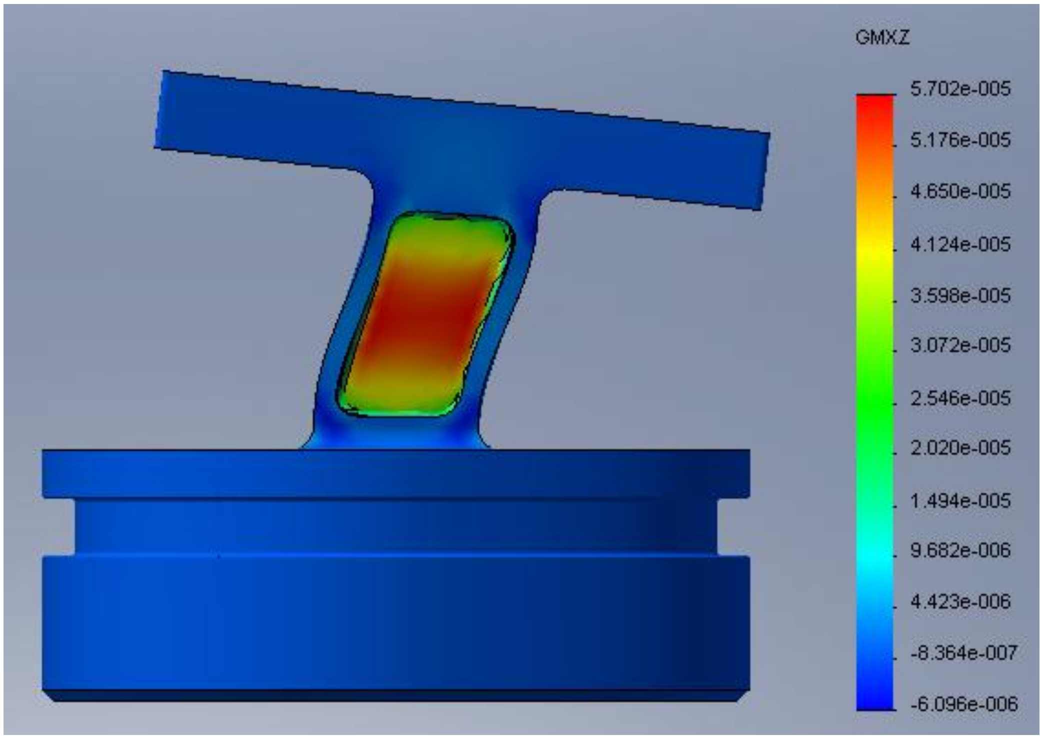
New Load Cell – Taller Web: Max shear strain = 60με
Making the web taller slightly increased the shear strain.



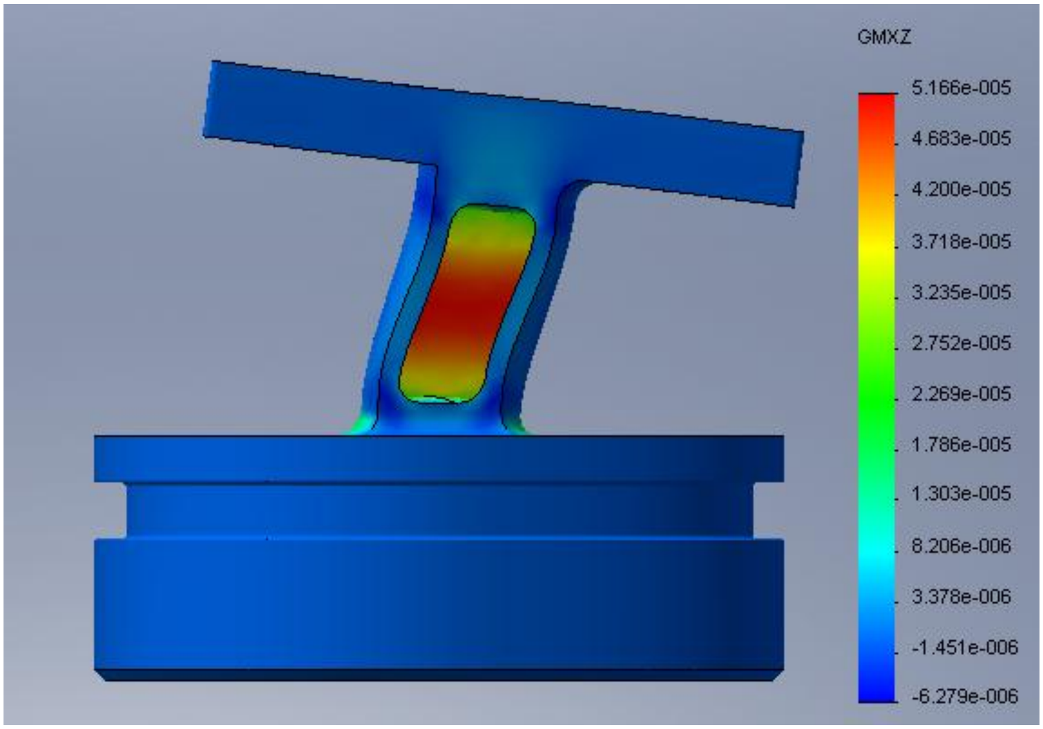
New Load Cell – Shorter Web: Max shear strain = 55με
Making the web shorter slightly increased the shear strain.



New Load Cell – Much Taller Web, Same Overall Height: Max shear strain = 63με
Making the web much taller didn't increase the shear strain enough to justify the additional machining.



New Load Cell – Dovetail Shaped Waterproofing Groove: Max shear strain = 52με
Changing the waterproofing groove to a dovetail shape did not significantly affect the shear strain.



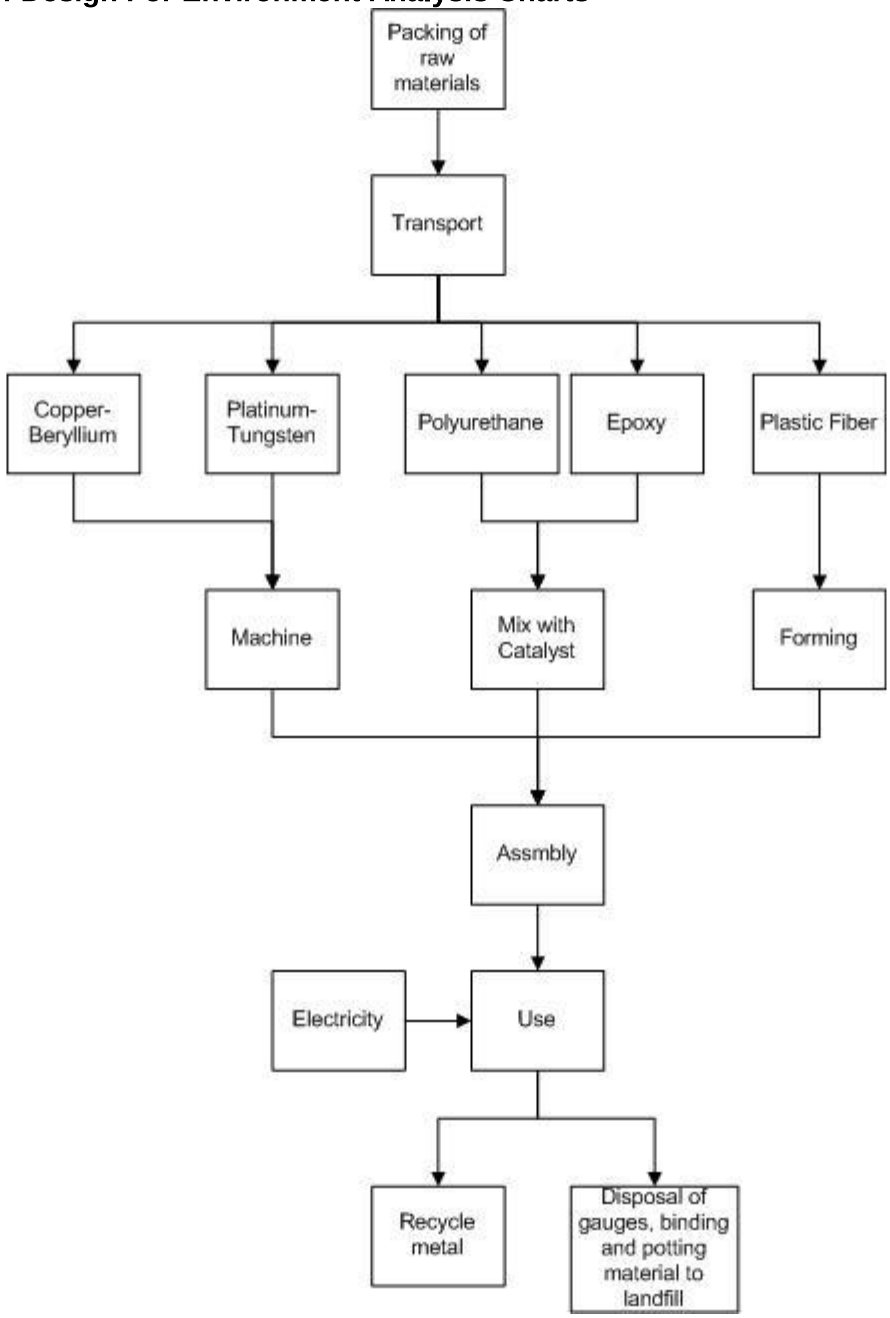
New Load Cell – No Waterproofing Groove – Angled Pillar Walls:
Max shear strain = 52με
Removing the waterproofing groove and angling the pillar walls in to help hold in the potting material decreased the shear strain.

APPENDIX B: Complete FMEA Chart

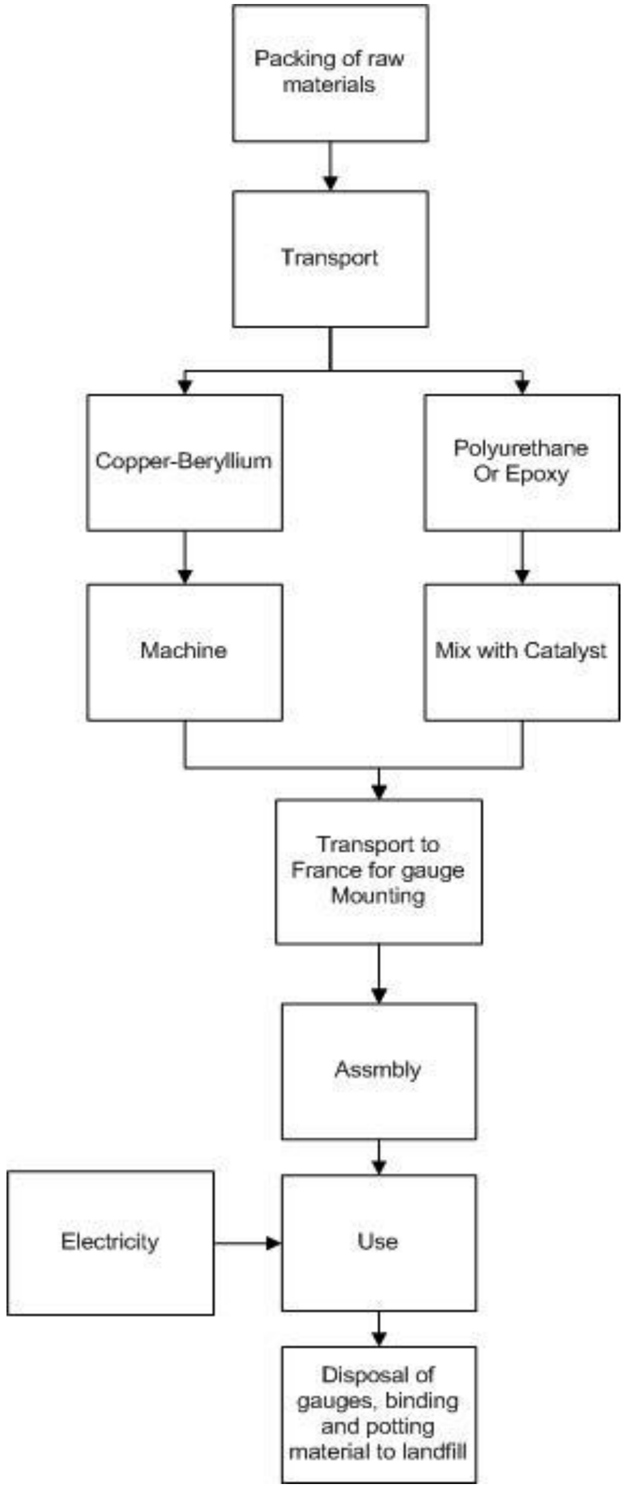
The FMEA chart referenced in the body of the paper is shown here in its entirety. It relates the different modes of failure to different parts in our design.

Product Name :Load Cell			Developing Team:				Page no. 1 of 1						
System			Anthony Bacon	Sean Cook			FMEA Number 1						
Subsystem			Corwin Holmes	Jennifer Hoskins			Date:11-20-07						
Part Number and Functions	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity (S)**	**Potential Causes/ Mechanism(s) of Failure**	**Occurrence (O)**	**Current Design Controls/ Test**	**Detection (D)**	**Recommended Actions**	**RPN**	**NEW S**	**NEW O**	**NEW D**	**NEW RPN**
Load Cell/ connects frictional force to strain within material	Corrosion of the material	Alters Geometry of Load Cell	7	Material reacting with environment	1	Emerse in water and inspect	2	Use Corrosion Resistant Material	14	7	1	2	14
Strain Guage/ measure strain	Short Circuit	Outputs no usable data	10	Exposure to water	10	Check data for reasonable values in an actual test	1	Alter geometry to prevent water from reaching gauge	100	10	8	1	80
	Temperature drift	Outputs inaccurate data	7	Changes in Temperature outside of range	5	Compare signal to expected values	1	Use a self-compensating gauge	35	7	1	1	7
Adhesive/ attatches Strain Gauge to Load Cell	Misalignment	Decreases Sensitivity	3	Wrong gauge orientation	3	Measure with protractor	2	Mark location and visually inspect	18	3	2	2	12
	Stiffens	Outputs no usable data	10	Excessive amounts of glue	3	Check by inspection	1	Be careful when applying glue to sensor test, may have to reglue several times	30	10	2	1	20
Pads/ out put signal of Strain Gauge to data interpreter	Open Circuit	No usable output signal	10	Bad Soldering Joints	1	Check for electrical conduction with an amp meter/inspection of solder joints with microscope	1	Add solder until joints are full and correctly bonded with materials	10	10	1	1	10
Wires/Connect Pads and Strain Gauge	Open Circuit	No usable output signal	10	Lack of electrical connection with pads	1	Check for electrical conduction with an amp meter	1	Check wire for conductivity after soldering, iterate if needed	10	10	1	1	10
Potting Material/prevent gauge from exposure to water	Bonding Failure	Strain gauge doesn't work	10	Water gets through smallest gaps between epoxy and metal	9	Check for existing output signal	1	Give material a good surface finish so there is no space between the gauge and the potting material	90	10	7	1	70
	Loose Fitting	Strain gauge doesn't work	10	Potting material shrinks too much after being installed	1	Inspection of potting and data	1	Use an epoxy that doesn't shrink as it sets/ visually inspect	10	10	1	1	10

APPENDIX C: Design For Environment Analysis Charts



Life-Cycle phase diagram of our new load cell design.



Life-Cycle diagram of the existing load cell.

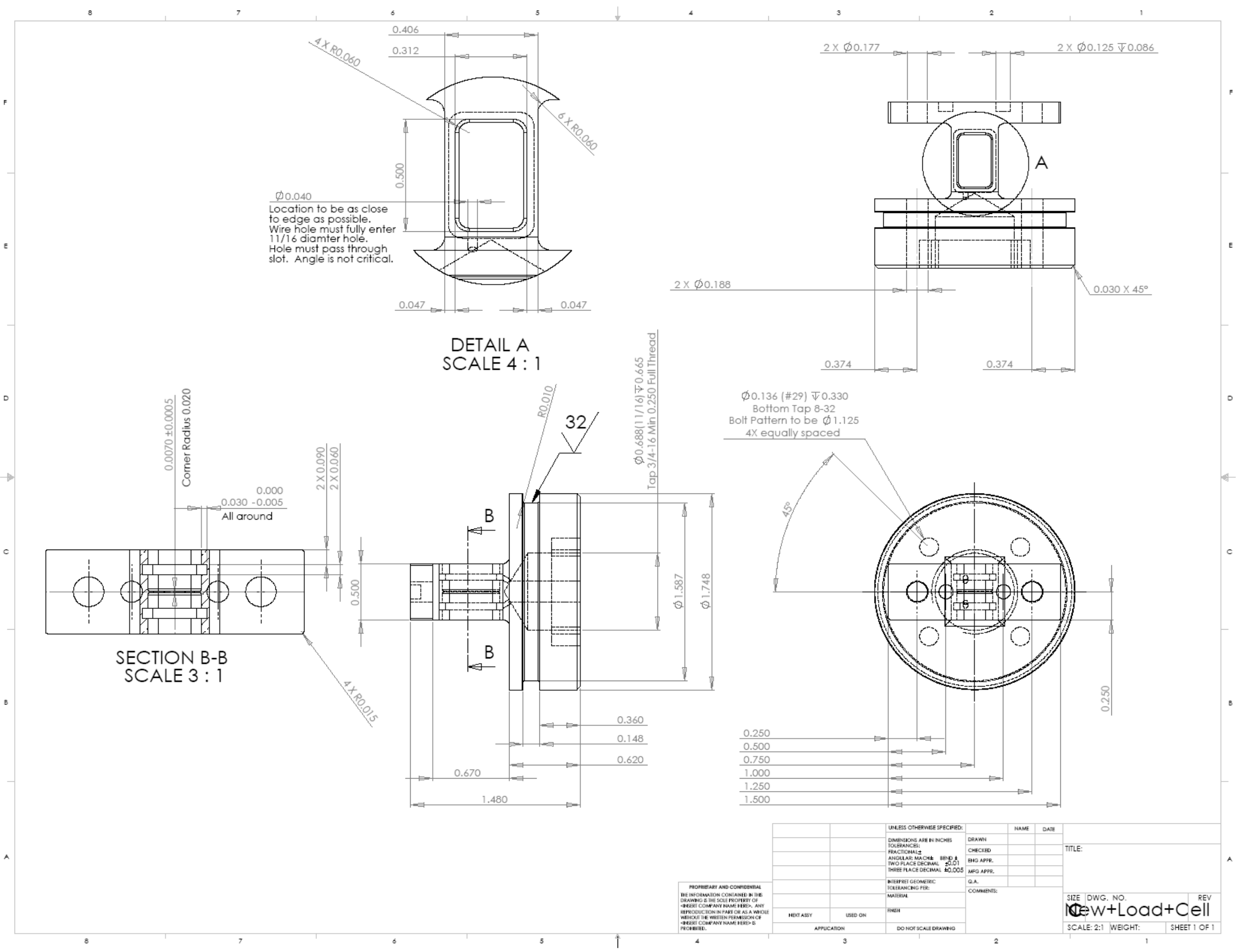
Product				
re-designed Strain sensing load cell				
Notes: Analysis of a foil guge load cell, assumtion: 6mo use at 12 hrs, electricty is average between France and Europe				
Analysis of opitical gauge assumed to be comprable to semi condortor gauge and foil gauge.				
Production	Amount(kg)	indicator	**result(mpts)**	
Copper-Beryllium	1	2500	2500	
Platinum-Tungsten (gauge)	0.00001	7005000	70.05	
Polyureathane	0.002	490	0.98	
			2571.03	
Assembly			**result(mpts)**	
Milling turning, drilling of Coppe	0.65	800	520	w/o prodution of lost material
PUR (energy absorbing)	0.002	490	0.98	
			520.98	
Use	Amount(kJ)	indicator	**resultm(pts)**	
electricity	0.144	20	2.88	
Disposal	Amount	indicator	**result(mpts)**	
municipal waste, PUR	0.002	2	0.004	
Recycle Copper	0.65	-70	-45.5	
municipal waste, platinum-tunst	0.00001	987885	9.87885	
insineration	0.002	2.8	0.0056	
			-35.6116	avoid primary copper prod.

Product			
old load cell			
Notes: Analysis of a semiconductor guge load cell			
Production	Amount(kg)	indicator	**result(mpts)**
Copper-Beryllium	1	2500	2500
semicondutor gauge	0.00002	53	0.00106
epoxy	0.002	490	0.98
			2500.981
Assembly			**result(mpts)**
Intercontinental transport Franc	1.95E-03	80	1.56E-01
Milling turning, drilling of Coppe	0.65	800	5.20E+02
PUR (energy absorbing)	0.002	490	9.80E-01
			521.1360
Use	Amount	indicator	**result(mpts)**
electricity	0.144	20	2.88
Disposal			
municipal waste,epoxy	0.002	2	0.004
municipal waste, semicondutor	0.00002	5	0.0001
inseniration epoxy	0.002	2.8	0.0056
			0.0097

The complete Eco-Indicator chart referenced in the paper is shown here. This chart shows the ecological impact for each life-cycle phase of the product. It compares the old design to the new design.

APPENDIX D: Engineering Drawing of New Load Cell

This is the blueprint we used when re-machining the old load cells so that they would have the altered geometries required for the new design.



APPENDIX E: MATLAB Fluid Analysis Code

The following MATLAB code was used to calculate the fluid force on the flat plate. These values were used in our COSMOSWorks analysis.

Main Program: Analysis.m

```
function [U,Fd,V,Re,Cd] = Analysis (E)

% Input units for Variables:
   %   rho = density of fluid (kg/m^3)
%   D = Diameter of disk (m)
%   mu = viscosity of the fluid (N*s/m^2)
%   U = upstream velocity (m/s)
%   Re = Reynolds Number ( )
%   Cd = Drag Coefficient ( )
%   A = Area perpendicular to stream flow of disk (m^2)
%   L = Length of leg of load cell (m)
%   W = Width of leg of load cell (m)
%   I = Moment of Inertia about y axis (m^4)
%   V = Displacement (m)
%   Fd = Drag Force on disk (N)
%   E = Modulus of Elasticity (Pa)
% Reference:
% Fundamentals of Fluid Mechanics, 5th edition Munson Young Okiishi,
%   Front page

 rho = 999;
 D = 4*.0254;
 A = pi*D^2/4;
 mu = 1.12*10^-3;
 L = .25*.0254;
 W = .094*.0254;
 I = W*L^3;


 U=[1:.01:18];

n=length (U);
 for y=[1 : 1 : n]
    Re (y) = Reynolds (rho, U (y), D, mu);
    Cd (y) = DragCoef ( Re (y));
    Fd (y)=Force (Cd (y), rho, U (y), A);
    Fd (y)=Fd (y)/4.448;
    V (y)= BeamDisp (Fd (y),L,W,I,E);
    U (y)=U (y)*3.2808399;
    V (y) = V (y)*39.3700787;
 end

% Output units for variables:
%   U = speed of flow (ft/s)
%   Fd = force on disk from flow (lbf)
%   V = displacement of disk (inches)
%   Re = Reynolds Number ( )
%   Cd = Coefficient of friction due to drag ( )
```

Sub Function: BeamDisp.m

```
function [V] = BeamDisp (F,L,W,I,E)

% Variables:
%   P = Load (force) on Plate (N)
%   F = Force felt on each individual support (N)
%   L = Length (m)
%   W = Width (m)
%   I = Moment of Intertia (m^4)
%   V = Strain (m)
%   E = Modulus of Elasticity (Pa)

P =F/2;
V=P*L^3/(E*3*I);
```

Sub Function: DragCoef.m

```
function [Cd] = DragCoef ( Re)

% Assumptions:
%   1) We can treat our material that is flush with the vessels
%      surface
%   meaning it is parallel to the flow
%   2) This material is smooth
%   3) Re<1
% Variables:
%   Re = Reynolds Number (distinguishes between laminar and turbulent
%   flow
%   Cd = Drag Coefficient

Cd = 0.455/(log10(Re))^2.58;

% Reference:
% Fundamentals of Fluid Mechanics, fifth edition, Munson Young Okiishi
%   pg .512, table 9.3
```

Sub Function: Force.m

```
function [Fd] = Force (Cd, rho, U, A)

% Variables:
%   Fd = Force on entire plate
%   Cd = drag coefficient ( )
%   rho = density of fluid (kg/m^3)
%   U = velocity of streamline (m/s)
%   A = Area of plate perpendicular to the flow

Fd = Cd*.5*rho*U^2*A;
```

Sub Function: Reynolds.m

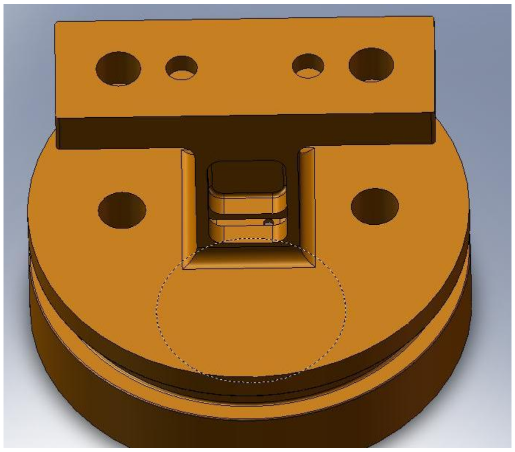
```
function [Re]= Reynolds (rho, U, D, mu)

% Variables:
%   Re = Reynolds Number ( )
%   U = upstream velocity (m/s)
%   rho = density of fluid (kg/m^3)
%   D = diameter of surface (m)
%   mu = viscosity of fluid (N*s/m^2)

Re = (rho*U*D)/mu ;
```

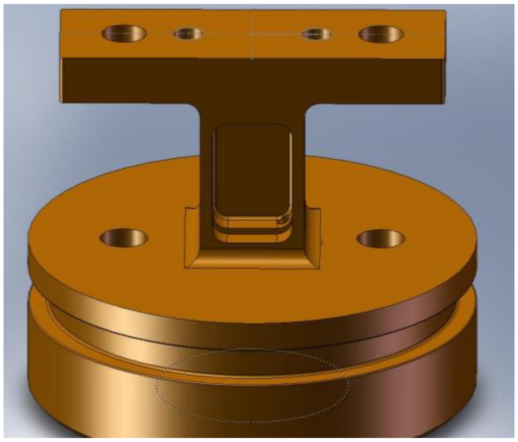
APPENDIX F: Engineering Change Notice

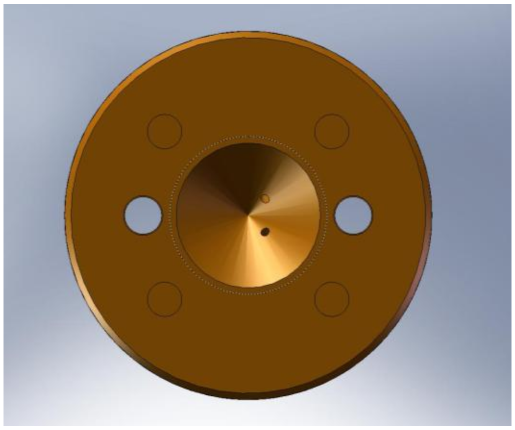
The ENC below depicts a hole added to the load cell to allow for better routing of the fiber-optic cable.

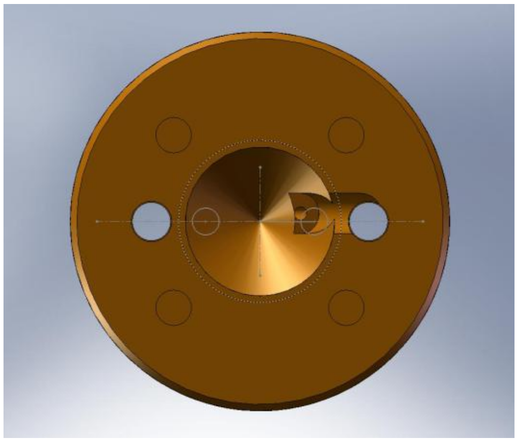
WAS:

IS:









Notes:

Needed to alter hole location to allow for better cable routing.

Necessary to prevent breakage of fiber-optic cable.

Team 14	
Project: Stress Sensor	
Ref Drawing: Load Cell	
Engineer: A. Bacon	11/28/2007
Sponsor: S. Ceccio	11/28/2007