UNIVERSITY OF ARIZONA

Optimal Support Solution for a Meniscus Mirror Blank

Final Design Review

Edgar Madril

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Opti 523 Independent Project

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Scope

For this problem an optimal solution for a mirror support is to be found for a relatively thin mirror blank with an aspect ratio of about 20:1. The project will incorporate the same mirror blank dimension as in a previous design problem but the thickness will be reduced by half (~.6"). The symmetry of the support location shall be determined and varied to create and optimal solution. Below are the mirror blank parameters.

Develop a mount concept for a variable orientation meniscus mirror mirror:

12" diameter,
.6" center thickness
Spherical surface: Concave radius of curvature of 48"
Material: ULE™

The scope of this project is to determine best possible support location that limits RMS surface error at various orientations (Zenith or Horizontal). This is a design that offers a complete support system that has an interface to some plane. The project does not include motion control of the mirrors positional orientation.

Requirements

Top Level

Mirror rms surface error < 20 nm rms surface for zenith or horizon pointing, including

- Surface irregularity from specification
- Nominal self weight deflection
- Mount induced deflections (from flexures, tolerances,…)

This implies that the total of all three of these factors must be less that 20nm rms.

Operational

- The surface budget above is taken AFTER removal of power.
- Power tolerance comes from ROC = 48" +/- 0.5"
- Lowest resonant frequency > 80 Hz.
- Operational environment:
  20C +/- 10C

Operational position stability requirement:

- 200 urad tip/tilt
- 0.008" decenter
- 0.020" axial position

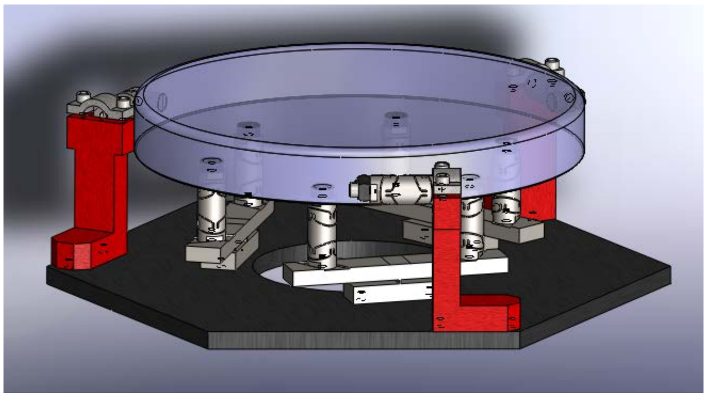
This stability requirement was doubled for the requirement set in problem 5. This is because this model is probably much more sensitive to loading from flexures. In order to mitigate this the flexures are to be made less stiff. This in turn will result in higher self weight deflection when orientated horizontally.
Survival:

- -10C to 50C
- 20 G shock

The main concern for these two parameters would be the flexures. The flexures would have to be able to be compliant enough so that the temperature variations would allow compliant motion and loading form shock would not yield the material. The mirror so far way about 8 lbs, if there is nine point contact each point will carry a .9lb load. This implies a shock loading of about 18lbs (80N) per support……

Design Concept

The initial model proposed for this design was based off of a previous design created. Initially it was assumed that the weight of the mirror would limit the ability to use a six point axial support and a 12 to 18 point support would be needed to better distribute the load of the mirror. But after an inspection of the rms surface error induced from self weight deflection, it was found that a six point support would offer an acceptable solution in terms of self weight deflection. A finite element analysis (FEA) was conducted and then post processed using Saguaro Software with the Zernike piston, tilt, and power term removed leading to a 8.2nm rms change in surface irregularity. This initial analysis was not based off an optimal support location so it was believed that a better result could be found by varying the support location to find an optimal solution. Below is the original proposed design form.

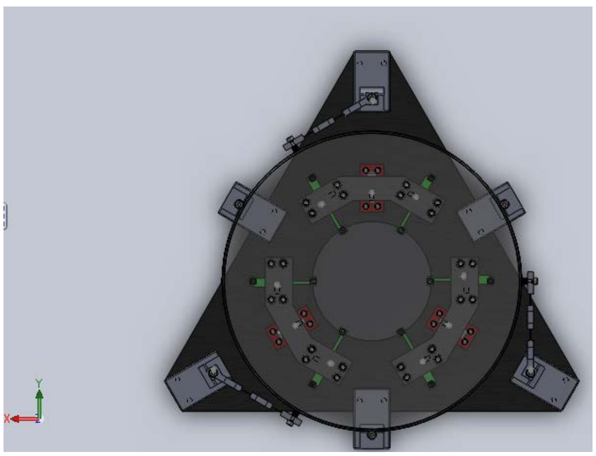
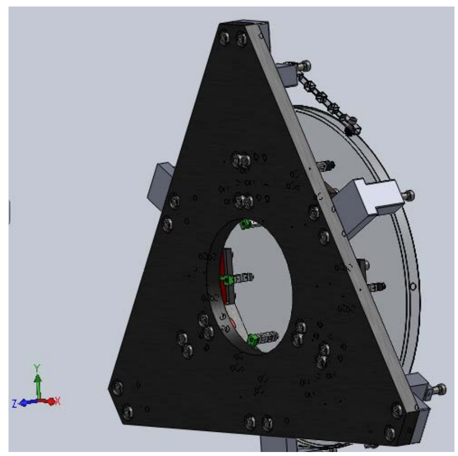


While the six point solution presented a promising and simple design path, it was found that the mirror blank was so sensitive parasitic moment and forces that it would not be a suitable design form. Since the weight of mirror was only distributed about six point the ability to compensate alignment and thermal variations was limited. The reaction moment created drastic surface distortions. The flexures needed to support the loads while enabling compliance in the orthogonal direction to accommodate tolerances and thermal variations. In the end the reaction moments and shear forces upon misalignment or thermal variation where too large to meet surface requirements. Flexure geometry was changed to accommodate the six point support but stability became a limiting factor (Buckling…..). So the design circled back to the starting point of the design, which was to go with a 12 or 18 point support solution.

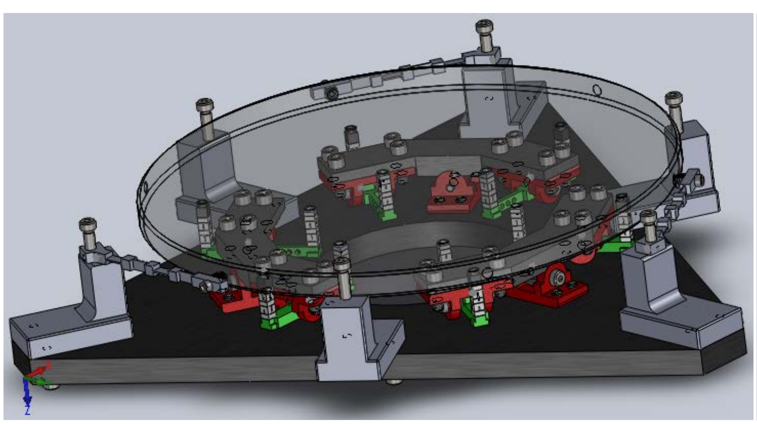
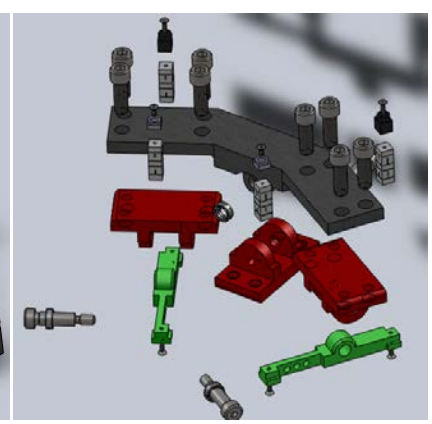
The final design form was a twelve support point support. The twelve point support was a simpler design form than an 18 point support and also offered a smaller part count. The support efficiency, based off of a paper written by “Nelson” indicated that a twelve point support offered support efficiency very comparable to an 18 point support. So the gain in going to an 18 point support was not large compared to the complications in design details.

The next decision made was the lateral support system. The axial support of course constrains motion in the axial direction but is compliant in the lateral direction. Usually, the lateral support system is trivial in comparison to the axial support system, but in the case of the thin meniscus mirror the loading sensitivities become large. A three point tangent support was chosen to support the mirror while horizon pointing. Below is a picture of the final design form.

Final Design

This design uses rockers with ball bearings in both the axial and lateral supports to create an approximate kinematic model. The support system becomes kinematic by introducing new degrees of freedom to create defined plane. In the axial support each rocker assembly enables sufficient degrees of freedom to emulate a three point support. The rockers are designed to minimize moments and reaction forces. Each rocker pivot has a ball bearing to negate the affects of frictional forces and corrosive effects of metal to metal contact.

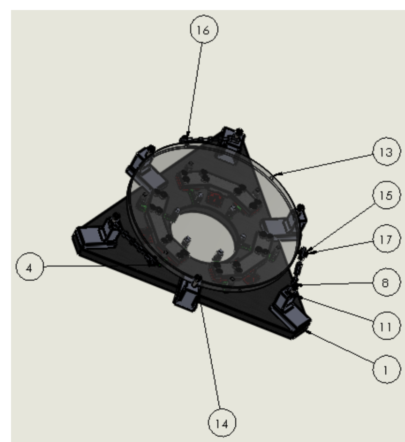
Material Properties

The material selected for almost all components is Stainless Steel 17-4PH. This material offers very good corrosive properties, low CTE (in comparison to Aluminum and conventional steels), and is very structurally sound. The parts that are not made of 17-4PH are the axial flexures and the arms that hold the tangent flexures in place. The material selected for these components is 6061 Aluminum. This was chosen because it was softer then steel enabling smaller parasitic loading from the flexure on the mirror blank.

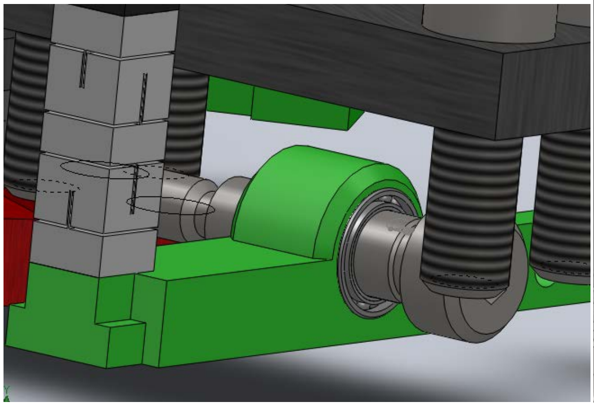
The other component not made of steel is what is called the thermal blocks. These thermal blocks are clamped down at the outer edge of the base plate and compensate radial thermal expansion by using the CTE of Al.

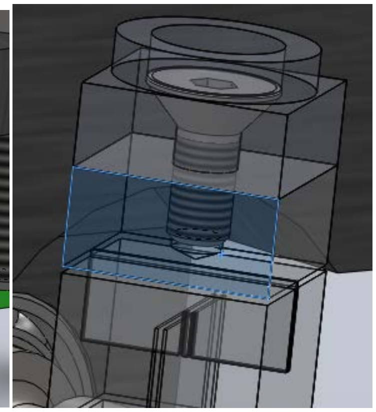
Part List

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	**BasePlate**	Stainless 17-4PH	1
2	**Double Pivot Arm**	Stainless 17-4PH	3
3	**PivotJoint**	Stainless 17-4PH	9
4	**swing arm**	Stainless 17-4PH	6
5	**UglyFlexure**		12
6	**OuterAxialPuck**	Invar	6
7	**InnerAxialPuck**	Invar	6
8	**SHSSCREW 0.25x0.75-N**	Cast Steel Alloy	15
9	**SCHCSCREW 0.073-72x0.188x0.188-HX-C**	Cast Steel Alloy	6
10	**HX-SHCS 0.25-20x0.75x0.75-C**	Cast Steel Alloy	48
11	**57155K355 (ball bearing)**	Unknown (steel)	12
12	**SCHCSCREW 0.073-72x0.3125x0.3125-HX-C**	Cast Steel Alloy	18
13	**ThinMirror2**	ULE	1
14	**ThermalBlock**	Aluminum 6061 T6	6
15	**TangentArm**	Stainless 17-4PH	3
16	**TangentPuck**	Invar	3
17	**HX-SHCS 0.164-32x0.4375x0.4375-C**	Cast Steel Alloy	3



Rocker Arm Assembly



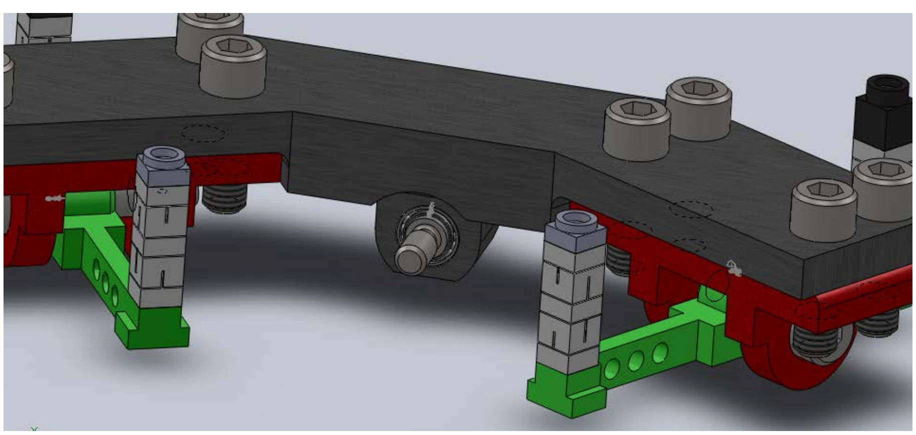


The rocker arms use precision shoulder screw to couple to the ball bearings. The ball bearings have a slight interference fit with the rocker arm. This securely holds the bearing in position after being pressed in. The pucks are pre-mounted to the flexures prior to being glued into position. This offers a relatively easy way to ensure flexures are in line with the pucks.

The pucks are made of Invar which is necessary to negate thermal stresses due to thermal variation between the flexures and mirror blank. Invar has a CTE of about 1 and ULE has a CTE that is close to zero. This interface enables the design to mitigate the need for in-depth analysis of the thermal stresses imposed on the mirror substrate.

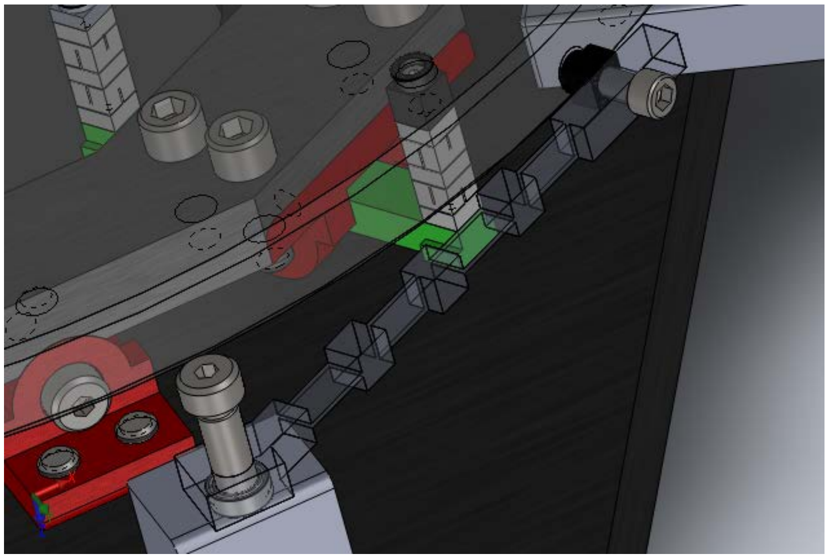
The mirror blank will be glued to the pucks using 3M-2216 epoxy. This epoxy is an excellent choice in adhesive because it has unique characteristics that enable shear flow.

The lower rocker arm is composed of the same parameters as the upper rocker arms. A ball bearing is used to create low friction forces. This in turn creates a very small moment that couples back into the mirror. The bearing has a slight interference fit. A precision shoulder screw is used to couple the bearing to the pivoting joint.



The rocker arms couple to the mirror blank via these two axis pivot flexures made of 6061 Al. The flexures are carefully designed to enable cross sectional compliance but relatively stiff in the axial direction of the flexure. The flexure axial stiffness is essential to meet system requirements.

Tangent Arm

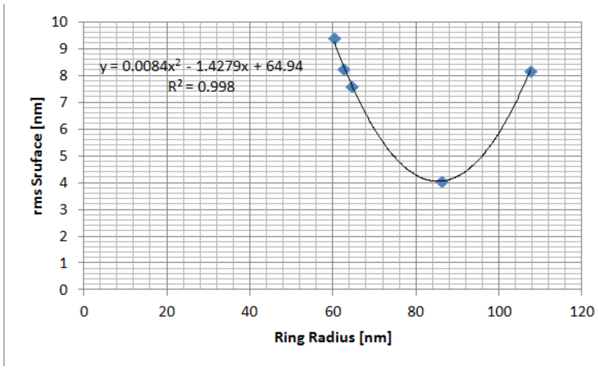


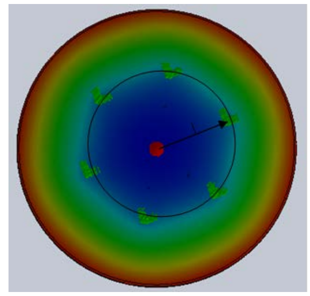
The tangent arm offers radial compliance and stiffness in lateral direction. The pivot arm is held in place by a shoulder screw that pivots about a bearing. The bearing was found necessary to this design form to minimize the frictional moment created by the pivot. The leaves in the flexure were designed such that they were thin enough to support the mirror but compliant enough to create small moments imposed on the mirror blank.

This design was the result of numerous attempts of suitable subsystems that where designed were analyzed. The final design became or at least in my mind the best way to support such a thin mirror. The system can be decomposed into two sub systems, the lateral support and the axial support. Normally the axial support is a main design issue. But in this design both the tangential and axial system provide difficult sensitivity to parasitic loads. This design was tedious to create but the main area of work was concentrated around the analysis. This was a completely analysis driven project so the line of reasoning the led to the design shall be presented.

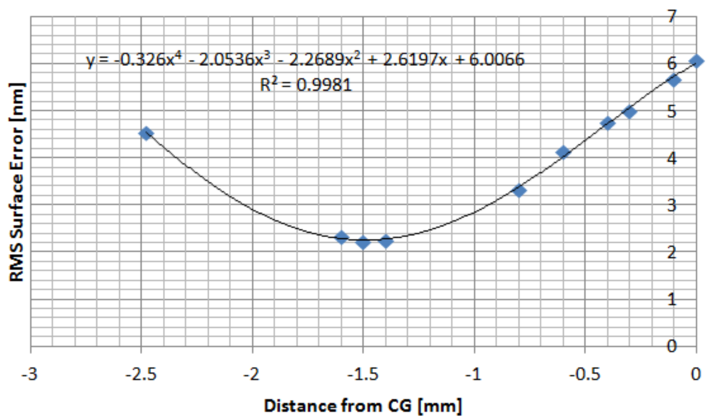
Design Analysis Formulation

The design began with optimizing a six point support solution. The Nelson model predicted that this would be a sufficient support to mitigate self weight deflection. Since the six point support could be thought of as a ring support, the only variable to optimize was the location of the ring radius. This was evaluated.



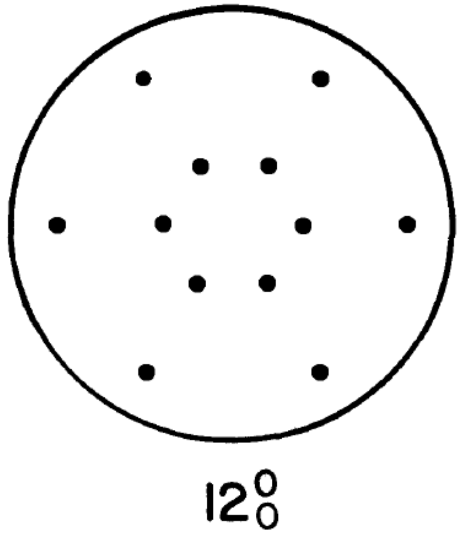


Next the position of the lateral support location with respect to the axial location of the center of gravity was determined; similarly only one variable was needed to find the optimal location, the z location with respect to the CG. Below are the evaluated results.



Next a sensitivity analysis was initiated to determine the allowable loads on the mirror blank to maintain performance less than 20nm surface error. As stated previously, the mirror proved to be too sensitive to the reaction moments and forces due to the flexures. The one thing that was established in this analysis was the location of the tangential support. Interestingly it was found that the surface error incurred from the support location dropped by a factor of 50% by moving the support location negatively from the CG.

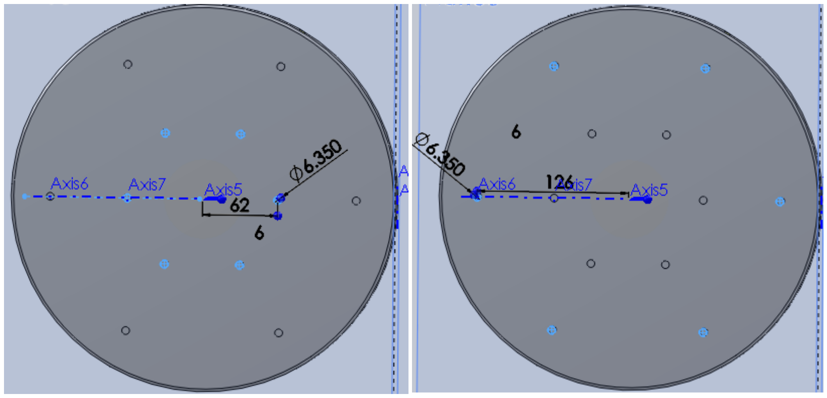
The analysis on the twelve point axial support system was then established. The support on this case consisted of twelve points that created 2 rings with six points at each ring respectively. As displayed below.



The support was manually optimized to determine an approximate location. It was found that the location was not very difficult to find because the minima had a very low gradient. The results showed that the solutions almost converged at a large range of radial locations. This is an approximation in that the analysis could only cover so many data points without the help some stronger computing power.

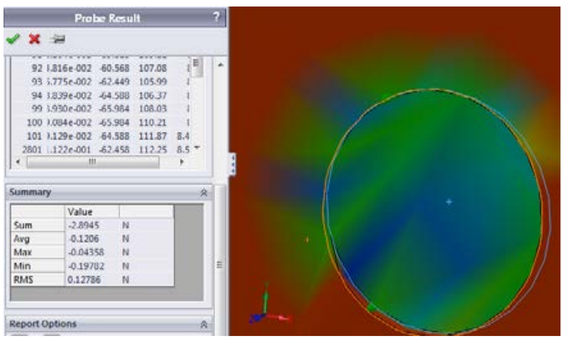
	Ro [mm]	Ri [mm]	RMS Surf [nm]	% Radius difference
1	111.125	47.625	4.96	0.4
2	116	52	4.6	0.403149606
3	121	57	4.4	0.403149606
4	126	62	4.4	0.403149606
5	128	67	4.5	0.384251969
6	130	65	4.4	0.409448819
7	135	40	4.4	0.598425197
8	140	35	8.6	0.661417323
9	145	60	6.2	0.535433071
10	133	57	5.6	0.478740157

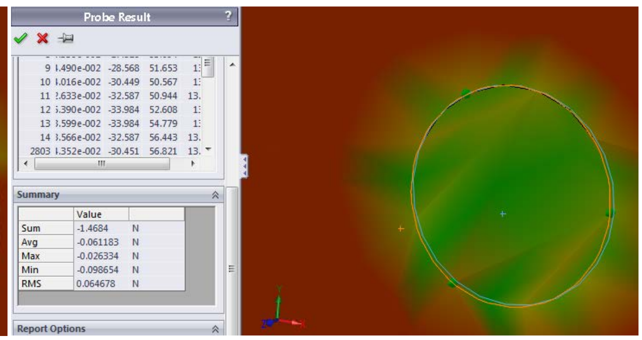
The final radii location where selected off of this analysis.



Next the reaction forces due to the weight of mirror were determined using FEA.

Outer Ring Reaction Force **Inner Ring Reaction Force**





Each point in the outer ring sees a reaction force of -2.8945N. Each point in the inner ring sees a reaction force of -1.4684N. The total weight of the mirror is 26.17 N. The sum of the reaction moments adds to the weight of the mirror.

Adhesive Analysis

For the adhesive analysis the bounds between the axial support and the mirror were investigated. The lateral support was ignored because the guess was that the bond would be strong enough to sufficiently hold the mirror under shock loading. The affects of shear loading where determined because this would be the point at which the glue is weakest.

Adhesive Loading	Shear Strength [Mpa]	Shear Force [N]	Area [mm^2]	Shear Stress [Pa]	Factor of Safety
Self Weight Outer Pucks	1.40E+07	-2.8945	1.67E+01	-173136.7388	8.09E+01
20G Shock Loaded Outer Pucks	1.40E+07	-57.89	1.67E+01	-3462734.777	4.04E+00
Self Weight Inner puck	1.40E+07	-1.4684	1.67E+01	-87833.4729	1.59E+02
20G Sock Loaded Inner Pucks	1.40E+07	-29.368	1.67E+01	-1756669.458	7.97E+00

Flexure Analysis

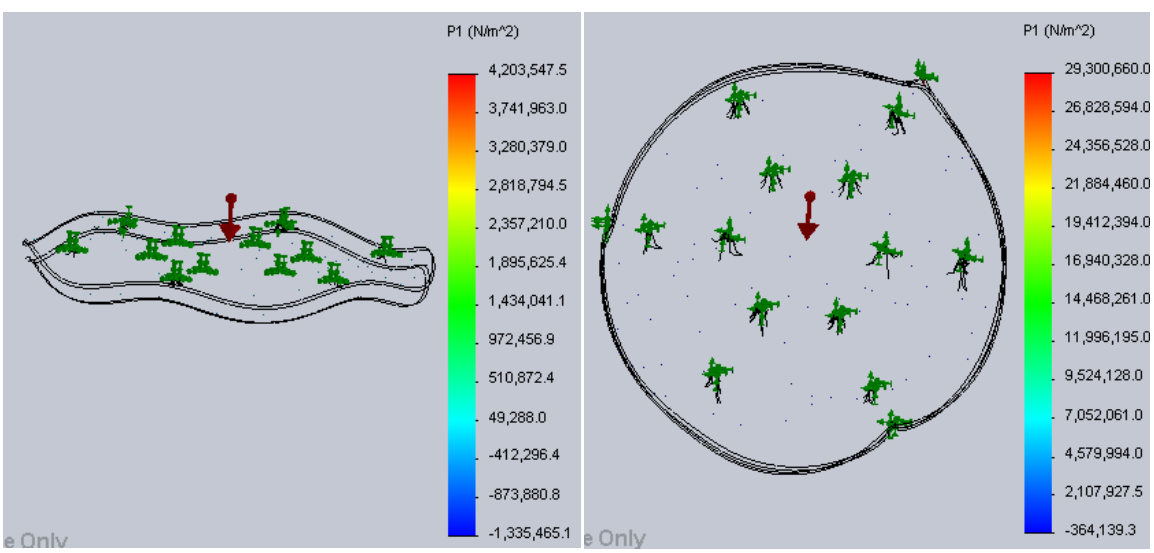
Next the flexure stiffness's were determined to continue on to the sensitivity analysis. The reactions forces were determined for a given displacement to create allowable tolerance and thermal variation.

Can the Mirror Survive a 20G Shock Load?

The mirror needs to be able to withstand a 20G shock load without failing. To determine if the mirror will fail Weibull statistics are used. The mirror is susceptible to crack propagation under tensile loading. Therefore, an analysis of principal stresses is needed to determine the maximum tensile load the mirror induces under shock loading. The stress can then be correlated to Weibull characteristic strength. For ULE glass the characteristic strength is 40.4 Mpa.

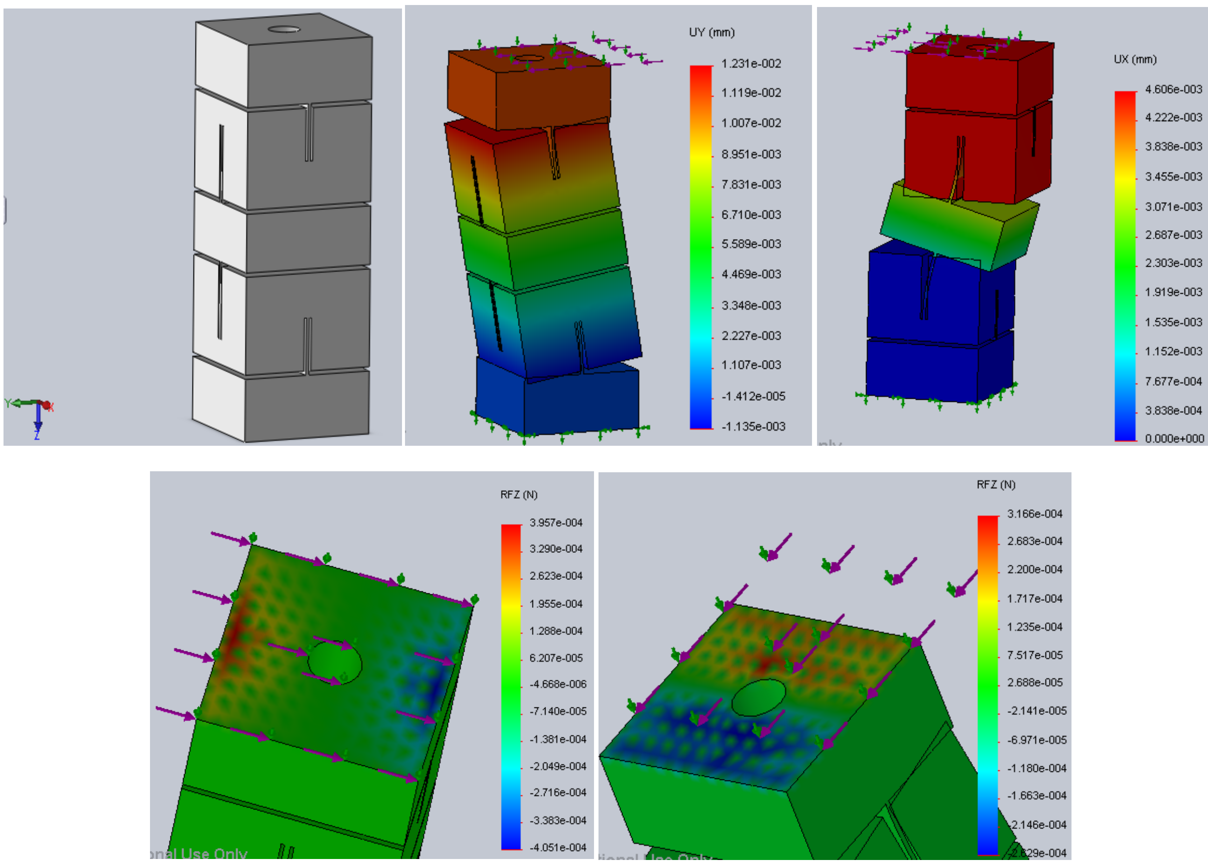
Probability of Failure at 6.9MPa (1,000 psi) for common glasses

Material	Weibull Modulus (m)	Characteristic Strength (MPa)	Probability of failure
N-BK7	30.4	70.6	0
F2	25.0	57.1	0
SF6	21.9	57.3	0
Silicon	4.5	346.5	2.2×10^{-8}
Germanium	3.4	119.8	6.1×10^{-5}
ZnSe	6.0	54.9	3.9×10^{-6}
Sapphire	4.0	485	4.1×10^{-8}
Calcium Fluoride	3.0	5.0	0.93
Zerodur	5.3	293.8	2.5×10^{-9}
Corning ULE	4.5	40.4	3.75×10^{-4}



The figures about show the tangent support with 20G shock load FEA of first principal stress and axial support with a 20G shock load FEA of first principal stress. The mirror is most sensitive the lateral shock loading but will sufficiently support the mirror without failing at a factor of safety of about 1.4.

Two axis flexure



Kx=2154.2N/m

Ky=813 N/m

Kz=301841N/m

For 100 micron displacement in + radial Direction

Fy=.08N

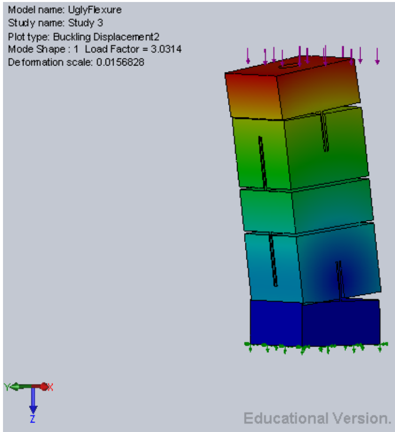
Mx=0.000014 N-m

Displacement in Tangential Direction

F=.2N

My=.00003N-m

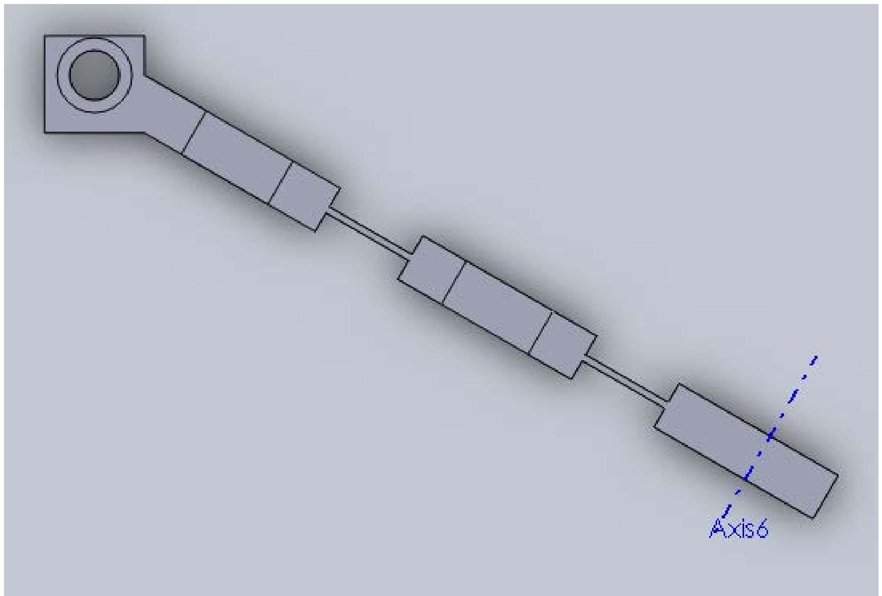
Buckling

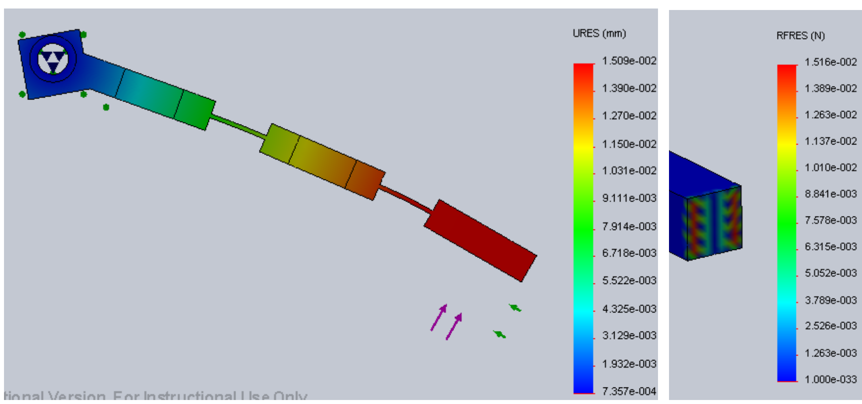


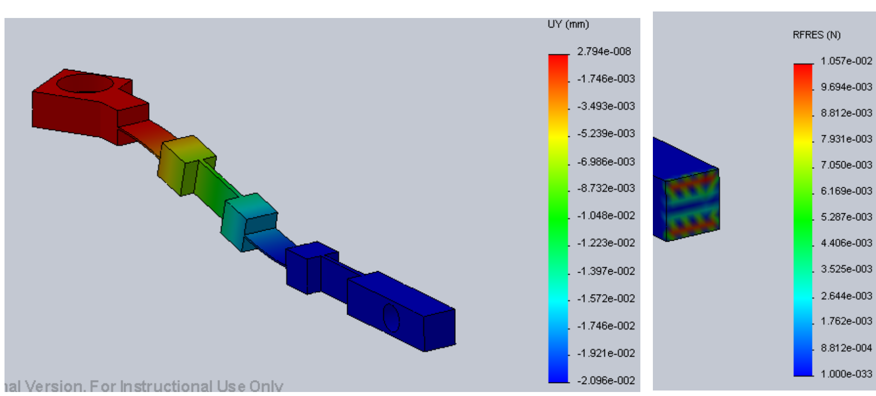
Here the first buckling mode shows that the flexure will buckle at a slightly higher load that the reaction forces induced under self weight. But since the flexure is constrained so that the top face cannot rotate, this mode of buckling will not be present. So a higher order buckling mode will occur under loading. Since the flexure leaves are formed with thin slits the design will mitigate failure under shock load by limiting motion by the width of the slits. Under high accelerations, the flexures will not be able to create the desired compliance. But the flexure will create desired performance specification at standard earth gravitational acceleration.

Tangent Arm Flexure

Similarly the critical stiffness's were determined to calculate the reaction forces imposed on the mirror for affects of tolerance alignment and thermal variations.







K_{radial}=659.63 N/m

K_z=500N/m

@ 100micron displacement

F_{radial}=.065 N

M_z=0.000096 N-m

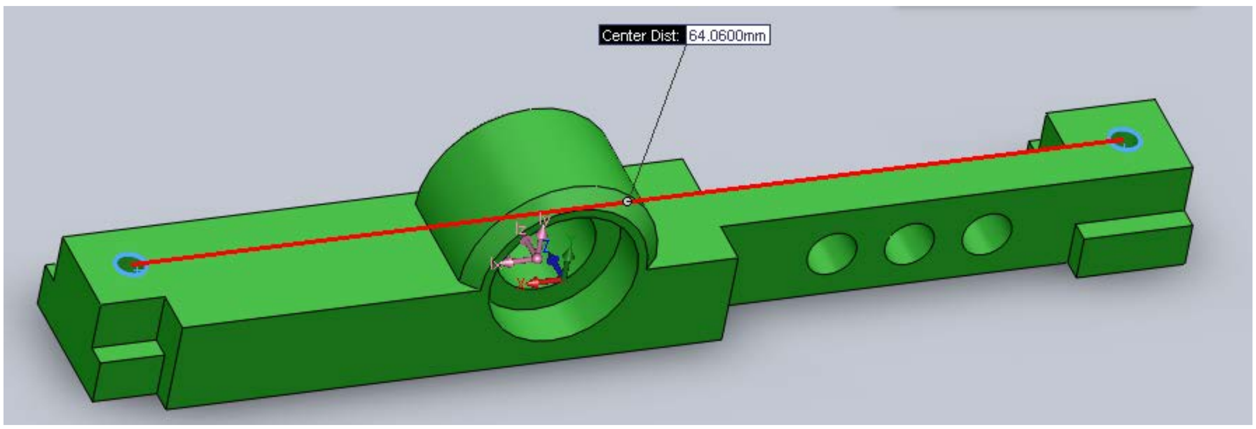
F_z=.05N

M_{radial}=.000073N-m

Rocker Reaction Frictional Reaction Moments

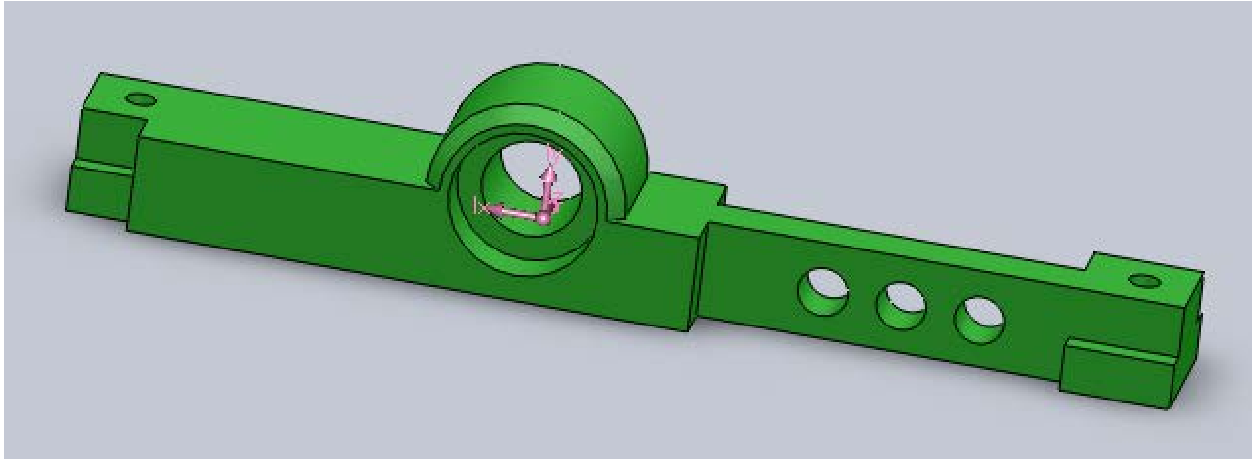
Top Rocker

The top rocker was designed to balance the load difference seen from the inner and outer ring contact forces. This was done be uses statics. Below are the derive lengths from the center hole to the outer holes.



L_1=38.4741mm, L_2=25.5259mm

The rocker was also deigned to balance self weight such that the center of gravity was located very close to the center of the hole. So that unwanted moments would be placed on the mirror from the weight of the rocker arm.



The rocker arm has a pressed ball bearing in it. The friction coefficient of a ball bearing can be approximated to be:

$$f=0.0015$$

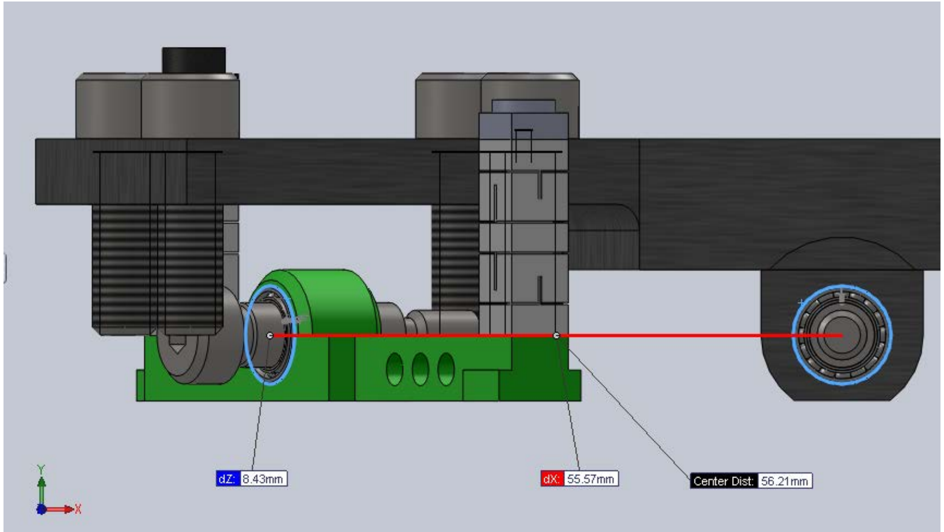
$$Moment_{friction}=F_{load}*f*r_{pin}$$

The Forces seen on each arm where then calculated.

F_1=0.000743N, F_2=-0.000112N

Lower Rocker frictional Moment

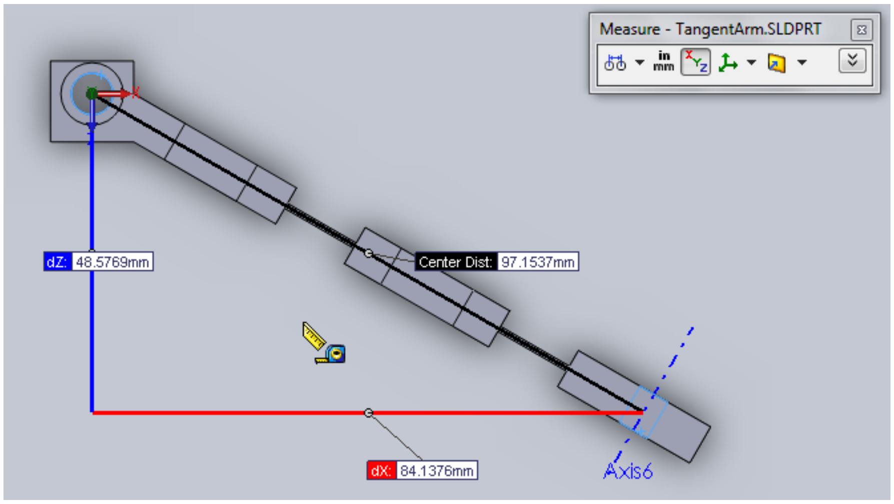
Similarly the forces seen at the ends of the bottom rocker that transfer to the top rocker were calculated. Then the force transferred to the top rocker was transferred to the forces seen on the mirror blank.



F_1=0.000842N, F_2=-0.00055N

Tangent Arm Rocker frictional Moment

Similarly the forces were found that would be imposed on the mirror blank caused by the frictional moment.



F_1=0.00045N-m, F_2=-0.00045N-m

Met Requirements

Sensitivity Analysis

The parasitic reaction forces were all calculated and then the change in rms surface error was determined based off of these effects. Below are the results.

Derived Values

Nominal Horizontal Self Weight Deflection	Nominal Vertical Self Weight Deflection	Surface Form Error λ/50 (546nm)	Vertical Loaded Deflections	Horizontal Loaded Deflections
1.37E-09	4.41E-09	1.09E-08	1.17E-08	1.45598E-10

	Horizontal Pointing	Zenith Pointing
Total [m]	1.60998E-08	1.66366E-08

Margin [m]	3.90016E-09	3.36341E-09

Error Budget

Reaction Shear Force [N]	Reaction Moment [N-m]	Nominal Vertical δ-rms	Nominal Horizontal δ-rms	Δ Vertical δ-rms	Δ Horizontal δ-rms
Tangential Arm Radial Moment and Shear Force					
0.0659	0.000096	1.37E-09	4.41E-09	4.77E-10	
0.0659	0.000096	1.37E-09	4.41E-09	4.77E-10	
0.0659	0.000096	1.37E-09	4.41E-09	4.77E-10	
Tangential Arm Radial Twist and Axial Shear Force					
0.013	0.000064	1.37E-09	4.41E-09	2.64E-09	
0.013	0.000064	1.37E-09	4.41E-09	2.64E-09	
0.013	0.000064	1.37E-09	4.41E-09	2.64E-09	
Tangential Arm Induced Frictional Forces Due to Axial Motion					
0.013	0.00127	1.37E-09	4.41E-09	6.23E-09	
0.013	0.00127	1.37E-09	4.41E-09	6.23E-09	
0.013	0.00127	1.37E-09	4.41E-09	6.23E-09	
Top Rocker Induced Frictional Forces					
0.00075		1.37E-09	4.41E-09		5.00E-11
-0.0012					
0.00075		1.37E-09	4.41E-09		5.00E-11
-0.0012					
0.00075		1.37E-09	4.41E-09		5.00E-11
-0.0012					
Bottom Rocker Induced Frictional Forces					
0.00085		1.37E-09	4.41E-09		5.00E-11
-0.00055					
0.00085		1.37E-09	4.41E-09		5.00E-11
-0.00055					
0.00085		1.37E-09	4.41E-09		5.00E-11
-0.00055					
Axial Support Flexure Radial Moment					
0.04	0.000014	1.37E-09	4.41E-09		2.00E-11
0.04	0.000014	1.37E-09	4.41E-09		2.00E-11
0.04	0.000014	1.37E-09	4.41E-09		2.00E-11

0.04	0.000014	1.37E-09	4.41E-09		2.00E-11
0.04	0.000014	1.37E-09	4.41E-09		2.00E-11
0.04	0.000014	1.37E-09	4.41E-09		2.00E-11
		1.37E-09	4.41E-09		2.00E-11
Axial Support Flexure Tangential Moment					
0.2	0.000028	1.37E-09	4.41E-09		2.38E-11
0.2	0.000028	1.37E-09	4.41E-09		2.38E-11
0.2	0.000028	1.37E-09	4.41E-09		2.38E-11
0.2	0.000028	1.37E-09	4.41E-09		2.38E-11
0.2	0.000028	1.37E-09	4.41E-09		2.38E-11
0.2	0.000028	1.37E-09	4.41E-09		2.38E-11

The surface form irregularity requirement has been met……

System Level Requirements

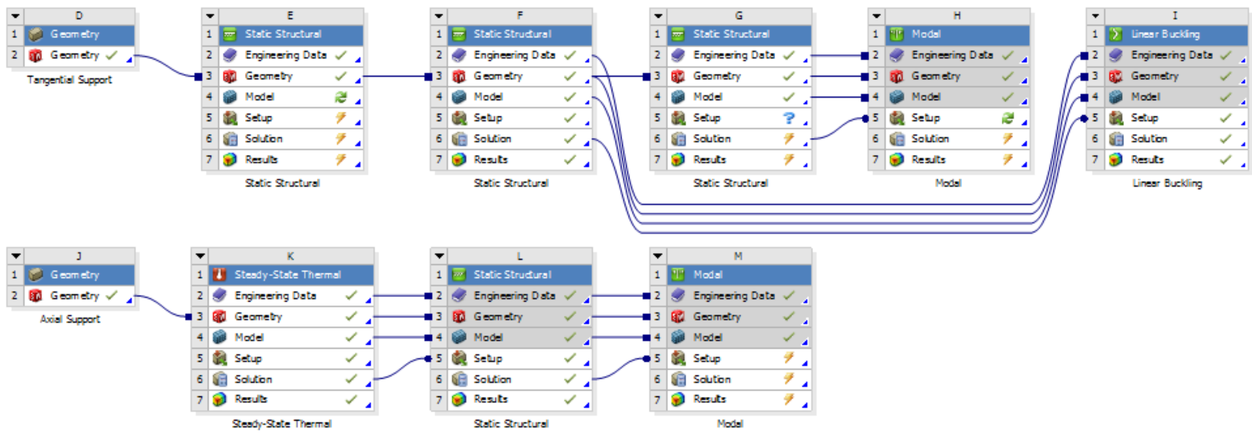
1. **Operational Requirements**

   a.) Lowest Resonant Frequency > 80Hz

   b.) Operational environment: 20C +/- 10C
   c.) Operational position stability requirement:
      I.) 200 urad tip/tilt
      II.) 0.008” decenter
      III.) 0.020” axial position

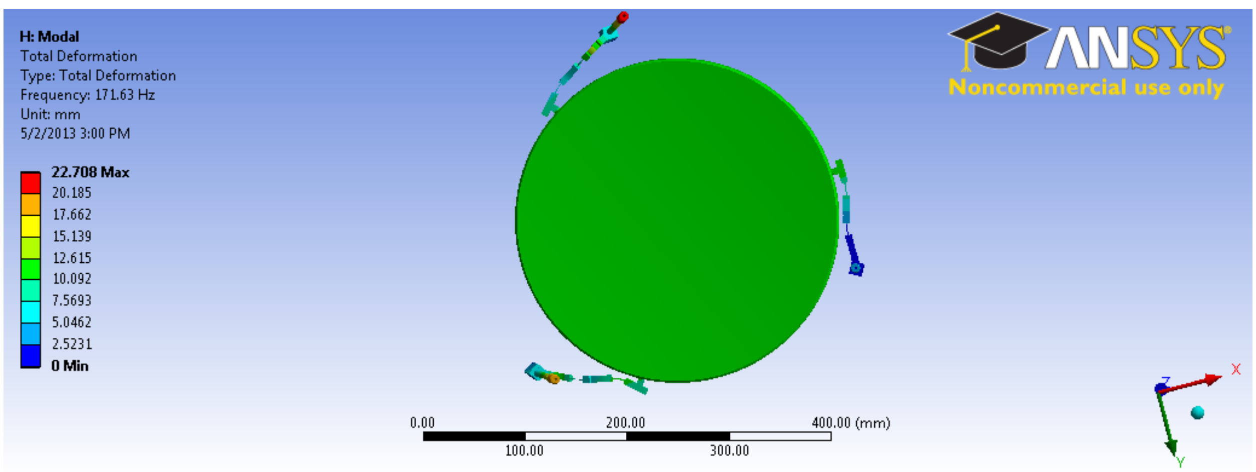
This analysis was done by separating the model into subsystems, the tangential support and the axial support. In reality the system would be even stiffer because it would be coupled to the other sub system. While this should not be a huge difference because the corresponding sub system is designed to be compliant in the direction the system being analyzed is stiff in. So in short this is a suitable method of approximation for a solution that should prove to be better than the results. ANSYS FEA was used to determine the system level requirements.

The analysis begin with the tangential support and moves onto the axial support.

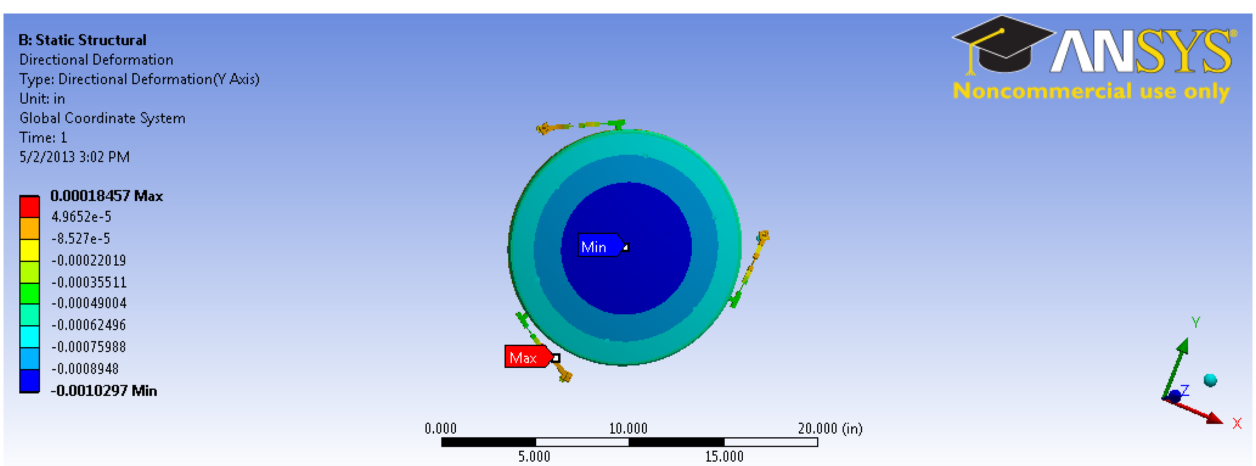


Tangential Support

The frequency analysis showed that the 4th order mode moved in the lateral direction with a frequency of 171Hz.



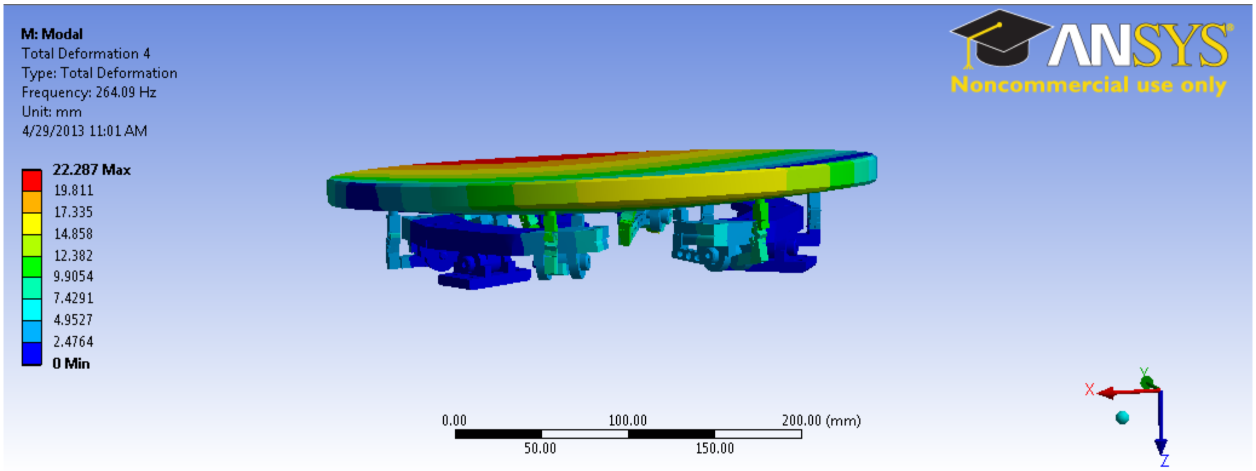
Next the lateral decenter was examined to see if the requirement was met.



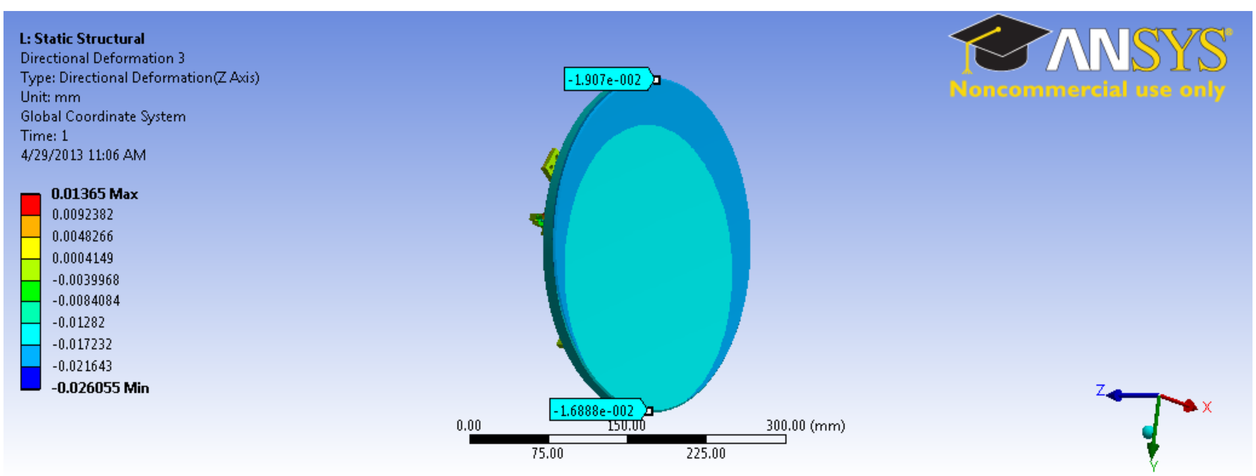
Decenter=.001"

Axial Support

Now, the requirements for the axial support is analyzed. The 4^{th} order mode shows rocking motion about the edges of the mirror in the z direction. The frequency at which this occurs is 264Hz.



Next the piston and tilt offset was observed. The only tilt in the system is a function of the axial support since this maintains axial position. To determine the tilt the gravitational load was placed vertically and the tilt was observed by probing the axial position of the top and bottom of the mirror.



Similarly the axial displacem was determined by the maximum offset in the Z direction.

Axial Displacement =-.026mm=0.001"

ΔZ=(1.907e-5m-1.688e-5m)=3micron

Tilt=3micron/317.5mm=9 micro radians

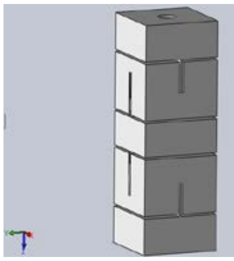
Operational requirements all Met…….

Survival

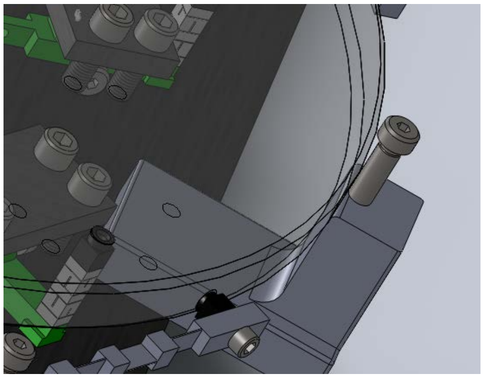
-10C to 50C
20 G shock
All flexural components are design to have limiting range of motion in the range of 1mm. This implies that the model can endure high temperatures. The temperature survival was not analyzed in detail because it presents itself as no major concern.

The 20 G shock load is a problem on the other hand. The flexures leaves on all flexures are very thin. They operate in such a fashion that they are just below the buckling limits of the analysis. To mitigate this design has safety system the prevent failure.

First area of concern is the axial flexures. The leaves range from 100 to 300 micron thickness. To mitigate failure the flexures have a 100 micron spaces to prevent failure.



Next the tangent arms are susceptible to buckling failure. To insure this does not occur a multiple stops are positioned a distance away the edge of the mirror blank. This will insure failure does not occur under higher accelerations.



Preliminary Fabrication Plan

The axial flexures will need to be created with wire EDM. As stated earlier the slits in the flexures are essential to meet system requirements. All other components are custom made and will require normal machining operations with the mill and the lathe. It is possible all these parts can be made without a CNC.

Assembly and alignment

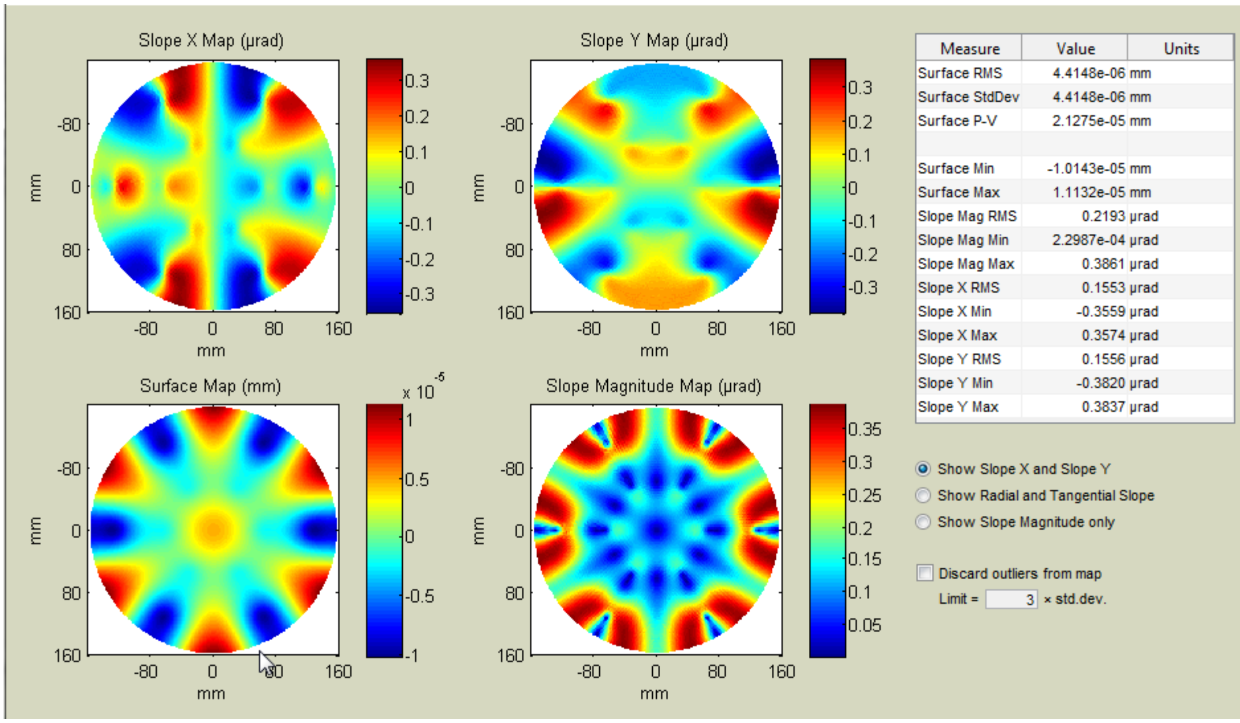
The assembly begins by gluing the Invar pucks to the mirror blank. A tooling device will need to be fabricated to properly position the pucks. This is to be modeled later in the design. Important, the axial pucks will need to be glue with the flexures bolted on. This will require tedious alignment of the flexures so they align to the rocker arm. This will require a detailed time and attention.

Once the pucks are secured the mirror shall be assembly from the axial flexures down. When assembling it is recommended that the flexures are positioned correctly by inserting 100micron tabs between flexure leaves. This will insure the flexures are at the most stress free state. The base plate has loose fit counter bores that will compensate for any tolerance offset. Once the axial support system is tightened to the base plate the 100micron tabs should be removed.

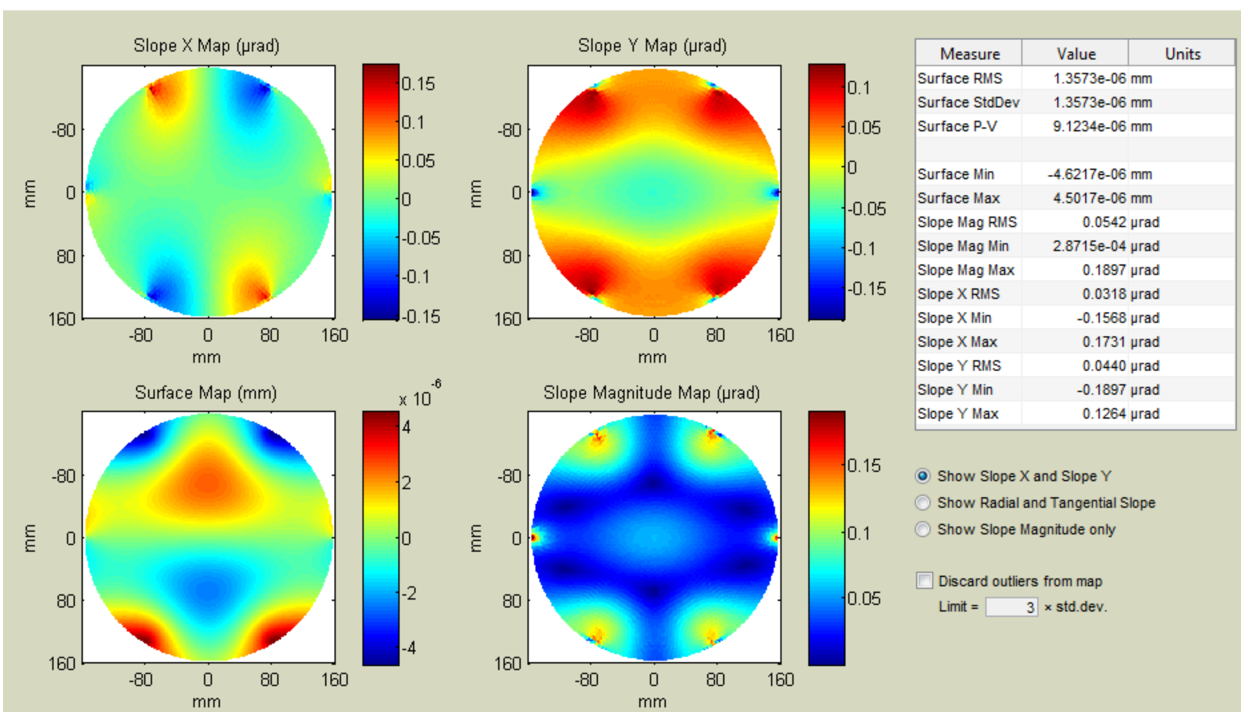
Next the tangential arms are to be put together from the pucks outward. Once again the base plate has loose fit counter bores to compensate any tolerance offset. Similarly it is recommended that the tangent arms are to be clamped in there stress free position. This should be done very carefully as the mirror is very sensitive to external loads.

Appendix

Optimal Axial Support



Nominal tangential Support

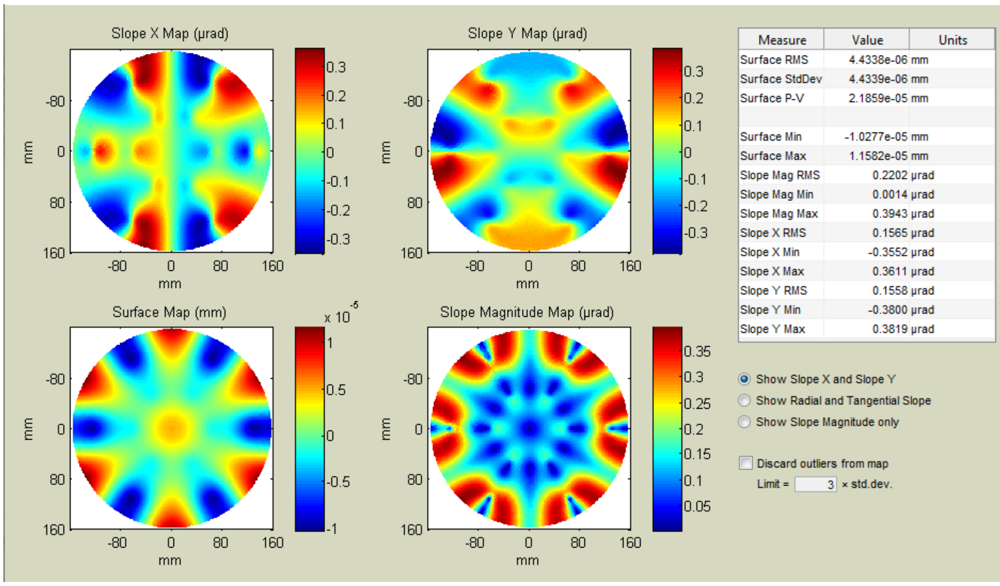


Flexure Axial Load at outer Ring

Kx=2154.2N/m,   Ky=813 N/m

For 100 micron displacement in + radial Direction

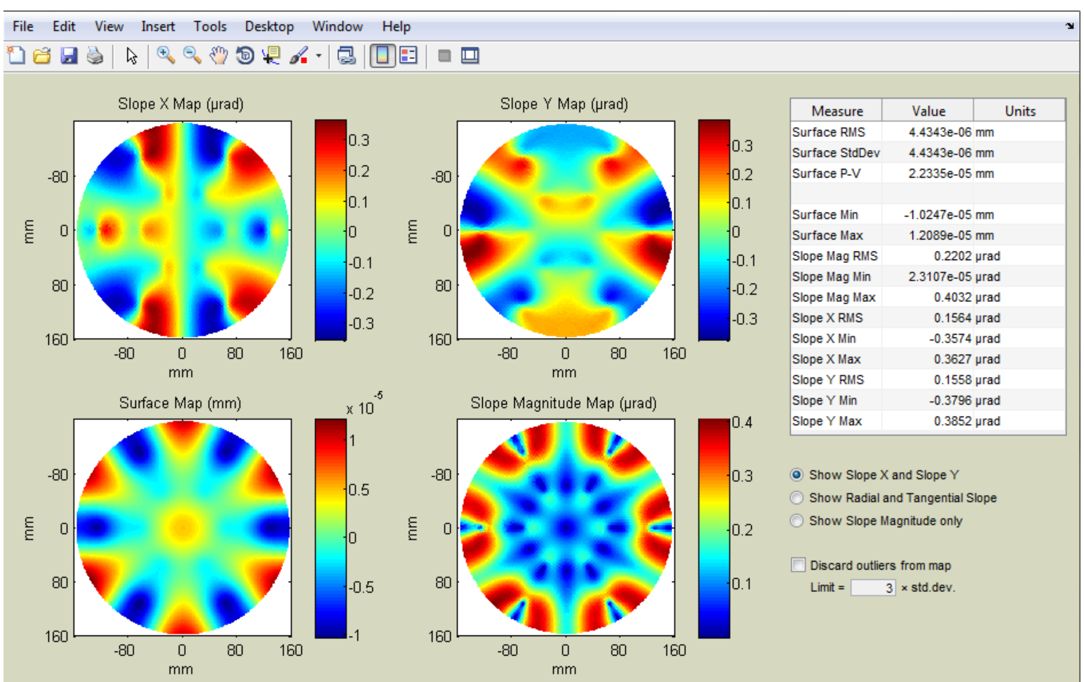
Fy=.08N,   Mx=0.000014 N-m



Displacement in Tangential Direction

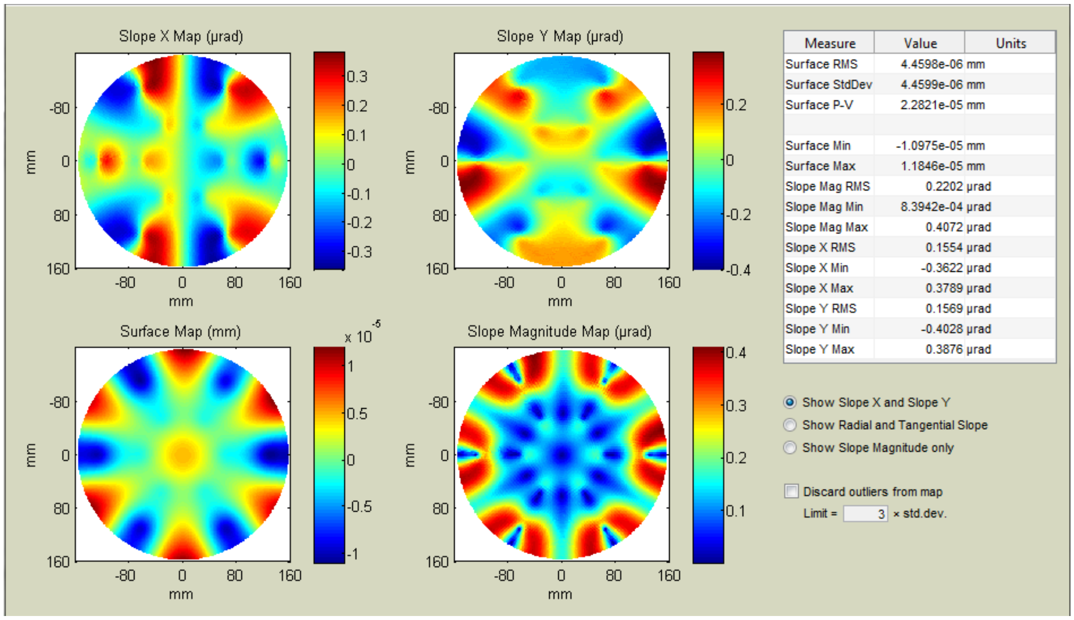
F=.2N

My=.00003N-m



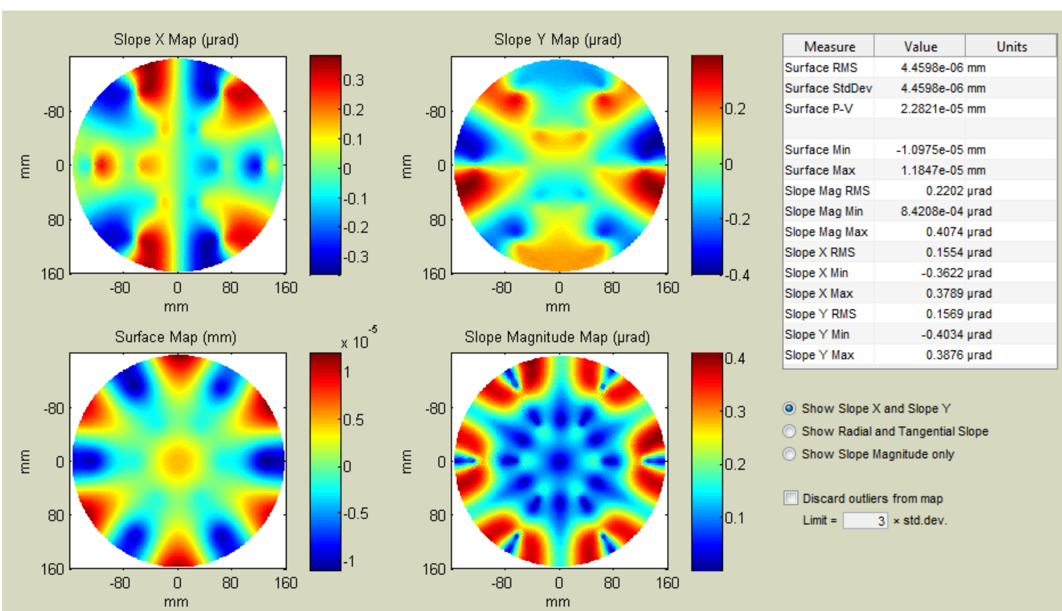
Top Rocker

F1(Outer Ring)=.000743N, F2(Inner Ring)=-.000112N



Torque Transferred from bottom Rocker to top Rocker

F1(Outer Ring)=.000842N, F2(Inner Ring)=-.00055

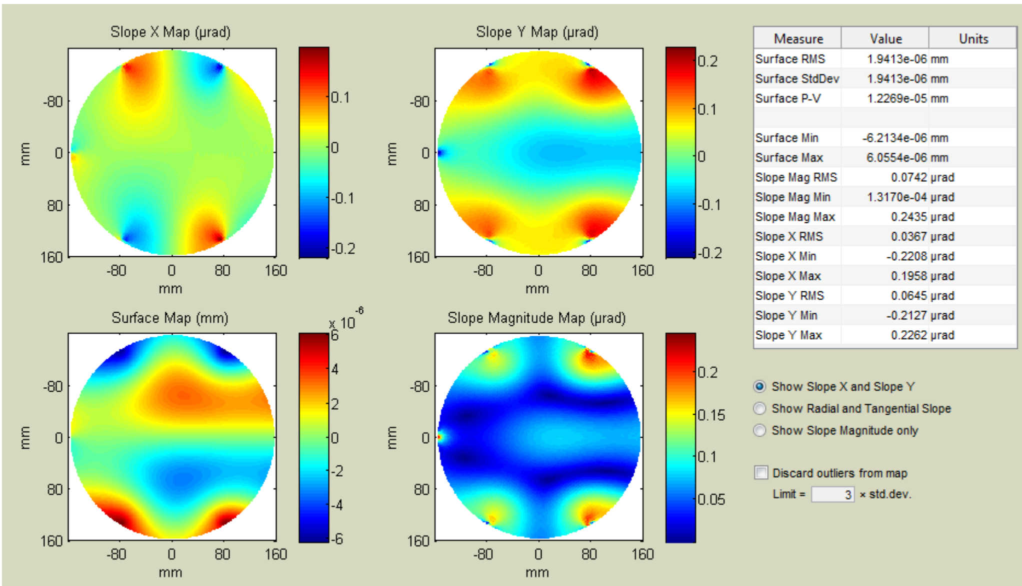


Tangential Support

Radial Arm Displacement of 100micron

Radial Direction

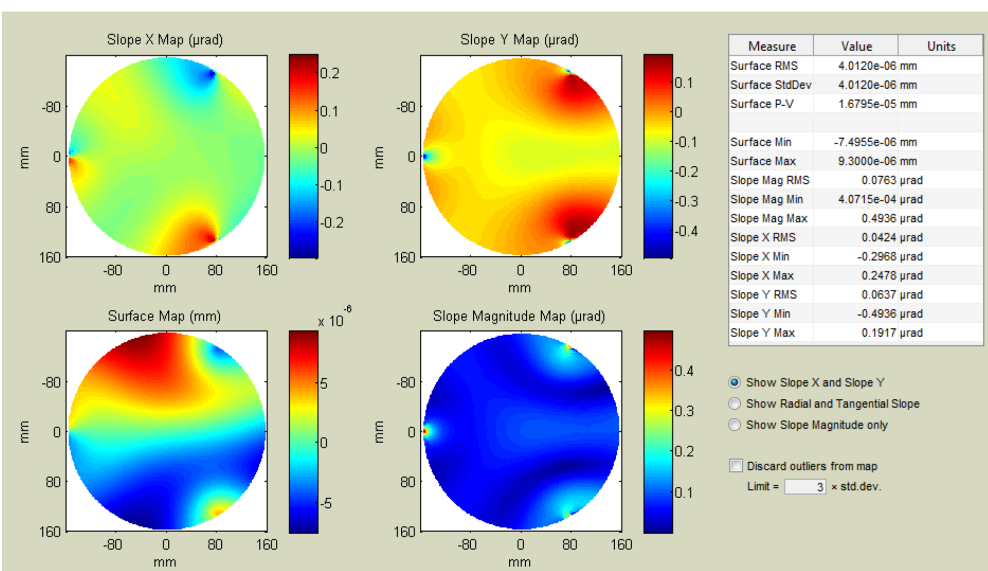
F=.0659N, M=.000096 N-m



Axial Direction

F=.013N

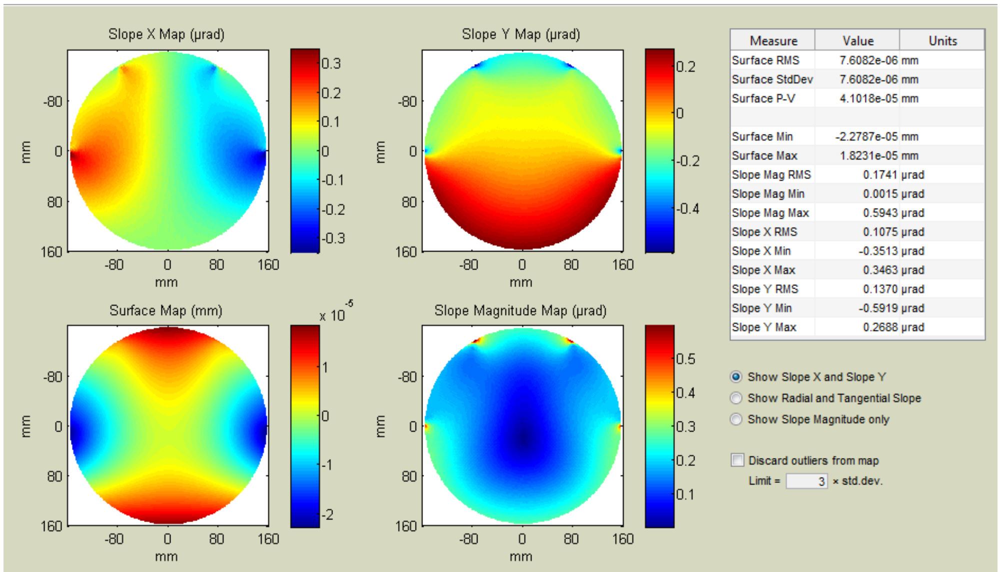
M=.001271 N-m



Tangential Rocker Moment

F1=.000425N

F2=-.000425



$$\delta_{rms} = \gamma_N \frac{q}{D}\left(\frac{A}{N}\right)^2$$

$$D = \frac{EH^3}{12(1-\nu^2)}$$

ϒN = Support Constant

q = the applied force per area

E = elastic modulus

v= Poisson Ration

h = plate thickness

r = plate radius

A = plate area

N = number of support

D = flexural rigidity

$\delta_{Vmax\text{-}rms}$ = rms surface deflection

arms and the bearings in the rocker arms. The rocker arms reaction moment can easily be determined by knowing the frictional constant of the bearings. Below is an approximate value of bearings frictional coefficients without oil.

- Single row ball bearing (radial Load) ..f = 0,0015
- Angular contact ball bearing (single row) ..f = 0,0020
- Angular contact ball bearing (double row) ..f = 0,0024
- Self aligning ball bearing (radial load) ..f = 0,0010
- Cylindrical roller bearings with cage ..f = 0,0011
- Cylindrical roller bearings full complement ..f = 0,0020
- Thrust ball bearing (axial load) ..f = 0,0013
- Spherical roller bearing (radial Load) ..f = 0,0018
- Taper roller bearings ..f = 0,0018
- Needle roller bearings-with cage ..f_m = 0,003
- Needle roller ball bearings-full Complement ..f_m = 0,005
- Combined needle roller bearings ..f_m = 0,004
- Axial Needle roller ball bearings ..f_m = 0,0035
- Axial Cylindrical roller bearings ..f_m = 0,0035

With this the bearing friction torque can be determined by knowing a few of the bearing parameters: the radial force being applied to the bearing, the friction coefficient, inside diameter, and the outside diameter of the bearing. This is the following equation that determines the torque created by the bearing

The bearing friction torque $M_r = F \cdot f \cdot (d/2)$

The bearing friction torque $M_r = F \cdot f_m \cdot (D_m/2)$

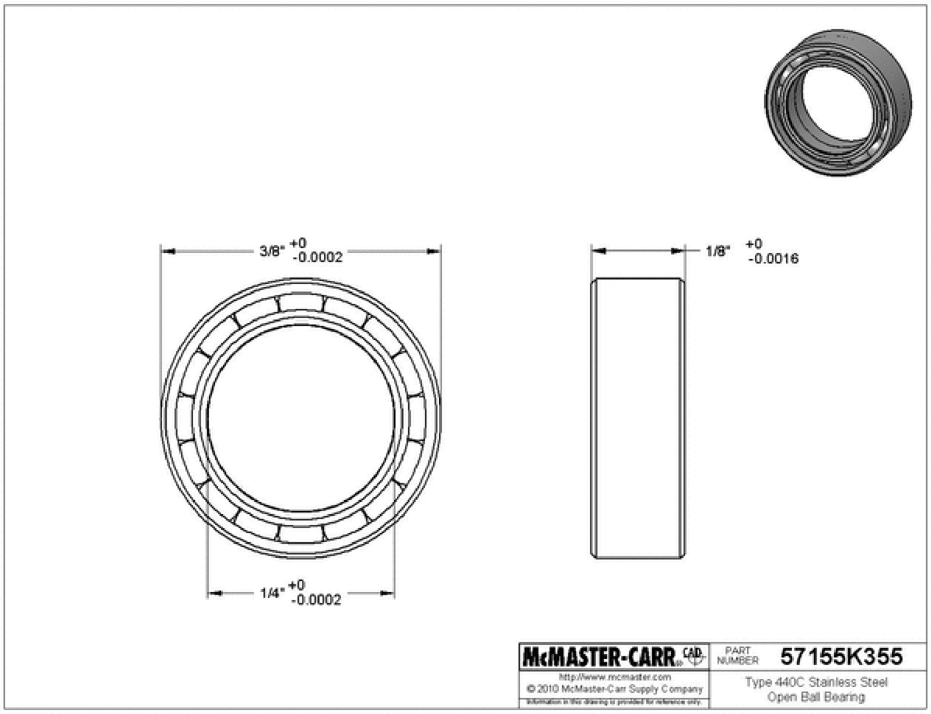
- M_r = Friction torque (Nmm)
- F = Radial (or axial load) (N)
- f = coefficient of friction of rolling bearing .
- f_m = coefficient of friction of rolling bearing based on mean diameter
- d = Diameter of the bore of the bearing (Shaft diameter)(mm)
- D = Outside diameter of the bearing (mm)
- D_m = (d+D)/2 (mm)

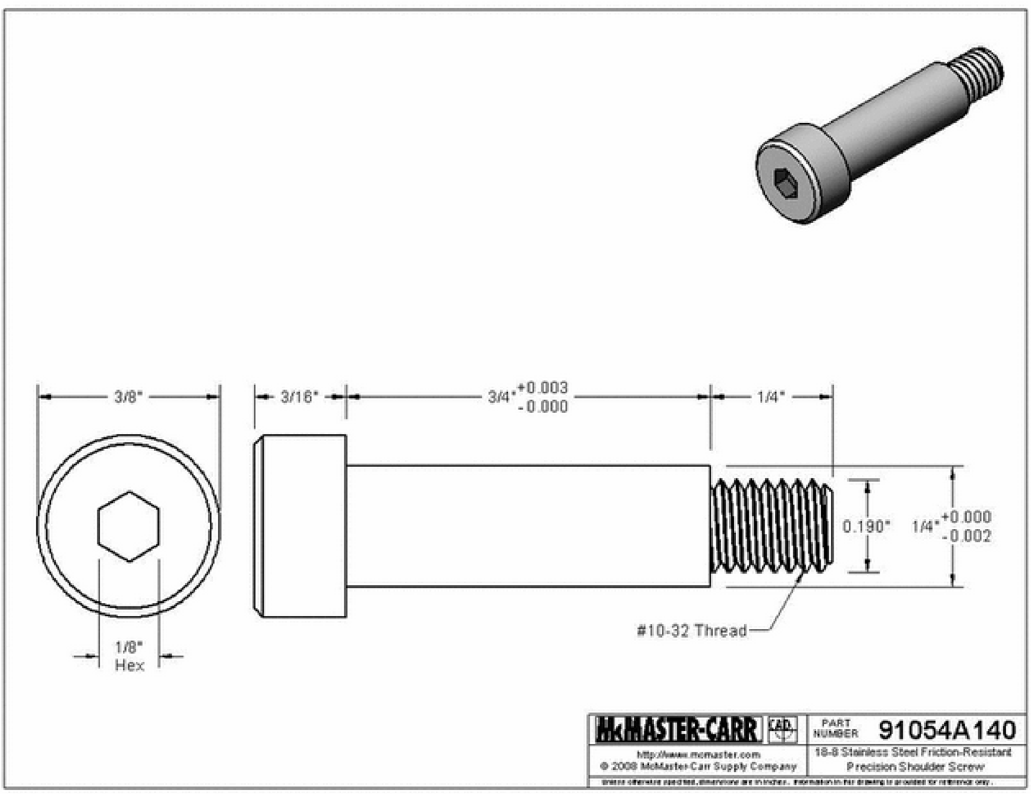
Material	Index of Refraction - n_d	Transmission Range (µm)	Young's Modulus –E (GPa)	CTE – α ($x10^{-6}$/°C)	Density – ρ (g/cm^3)	dn/dT (absolute) ($x10^{-6}$/°C)	Poisson Ratio – ν	Thermal Conductivity – λ (W/mK)	Stress Optic Coefficient – K_s (10^{-12}/Pa)
N-BK7	1.5168	0.35 – 2.5	82	7.1	2.51	1.1	0.206	1.11	2.77
Borofloat 33 Borosilicate	1.4714	0.35 – 2.7	64	3.25	2.2		0.20	1.2	4
Calcium Fluoride	1.4338	0.35 – 7	75.8	18.85	3.18	-10.6	0.26	9.71	2.15
Fused Silica	1.4584	0.18 – 2.5	72	0.5	2.2	8.1	0.17	1.31	3.4
Germanium	4.0026 (at 11µm)	2 – 14	102.7	6.1	5.33	396	0.28	58.61	-1.56
Magnesium Fluoride	1.413 N_{ord} (at 0.22	0.12 – 7	138	13.7 8.9	3.18	2.3 (//) 1.7 (_|_)	0.276	11.6?	
Sapphire	1.7545 N_{ord} (at 1.06	0.17 – 5.5	335	5.3	3.97	13.1	0.25	27.21	
SF57	1.8467	0.4 – 2.3	54	8.3	5.51	6.0	0.248	0.620	0.02
N-SF57	1.8467	0.4 – 2.3	96	8.5	3.53	-2.1	0.26	0.990	2.78
Silicon	3.4223 (at 5 µm)	1.2 – 15	131	2.6	2.33	160	0.266	163.3	
ULE (Corning	1.4828	0.3 – 2.3	67.6	0.03	2.21	10.68	0.17	1.31	4.15
Zerodur	1.5424	0.5 – 2.5	90.3	0.05	2.53	14.3	0.243	1.46	3.0
Zinc Selenide	2.403 (at 10.6	0.6 – 16	67.2	7.1	5.27	61	0.28	18	-1.60
Zinc Sulfide	2.2008 (at 10µm)	0.4 – 12	74.5	6.5	4.09	38.7	0.28	27.2	0.804

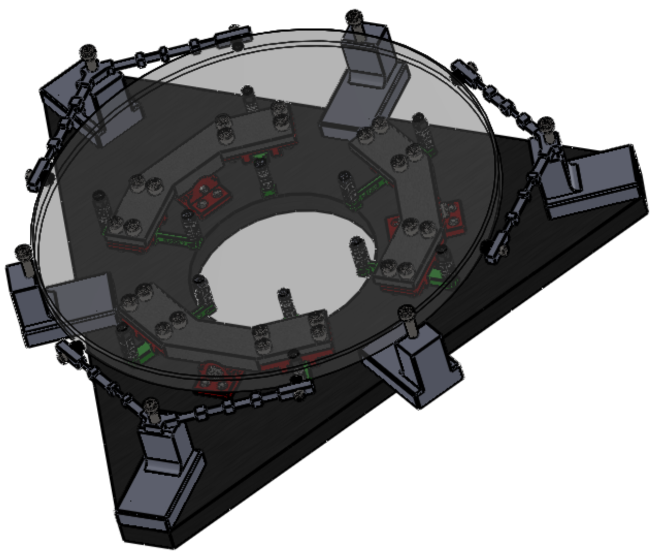
Material	Young's Modulus – E (GPa)	Density – ρ (g/cm^3)	Poisson Ratio – ν	CTE – α ($x10^{-6}$/°C)	Thermal Conductivity – λ (W/m/K)	Hardness
Aluminum (6061-T6)	68	2.7	0.33	24	167	Rockwell A – 40 Rockwell B – 60
Beryllium	303	1.84	0.29	11.5	216	Rockwell B - 80
Copper C260	110	8.53	0.38	20	120	Rockwell F – 54
Graphite epoxy (CFRP)	180	1.7		0.02	11.5	
Invar 36	148	8.0	0.29	1.3	14	Rockwell B -90
Molybdenum	320	10.2	0.31	5.0	138	Brinell 1500 MPa
Silicon Carbide	410	3.1	0.14	4.0	120	Knoop - 2800
Steel 1020	200	7.8	0.29	11.3	52	Rockwell B - 76
Stainless Steel CRES 17-4PH	198	7.8	0.27	11	18	Rockwell C - 35
Stainless Steel CRES 316	193	8	0.30	16	16	Rockwell B - 79
Titanium (6Al-4V)	108	4.5	0.31	8.6	7	Rockwell B - 80

Drawings

Ball Bearing


3/8" +0 -0.0002
1/8" +0 -0.0016
1/4" +0 -0.0002
McMASTER-CARR CAD
PART NUMBER 57155K355
http://www.mcmaster.com
© 2010 McMaster-Carr Supply Company
Type 440C Stainless Steel
Open Ball Bearing


3/8"
3/16"
3/4" +0.003 -0.000
1/4"
0.190"
1/4" +0.000 -0.002
#10-32 Thread
1/8" Hex
McMASTER-CARR CAD
PART NUMBER 91054A140
http://www.mcmaster.com
© 2008 McMaster-Carr Supply Company
18-8 Stainless Steel Friction-Resistant
Precision Shoulder Screw



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	BasePlate		1
2	Double Pivot Arm		3
3	PivotJoint		9
4	swing arm		6
5	UglyFlexure		12
6	OuterAxialPuck		6
7	InnerAxialPuck		6
8	SHSSCREW 0.25x0.75-N		15
9	SCHCSCREW 0.073-72x0.188x0.188-HX-C		3
10	HX-SHCS 0.25-20x0.75x0.75-C		48
11	ThinMirror2		1
12	ThermalBlock		6
13	TangentArm		3
14	TangentPuck		6
15	HX-SHCS 0.164-32x0.4375x0.4375-C		6

NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

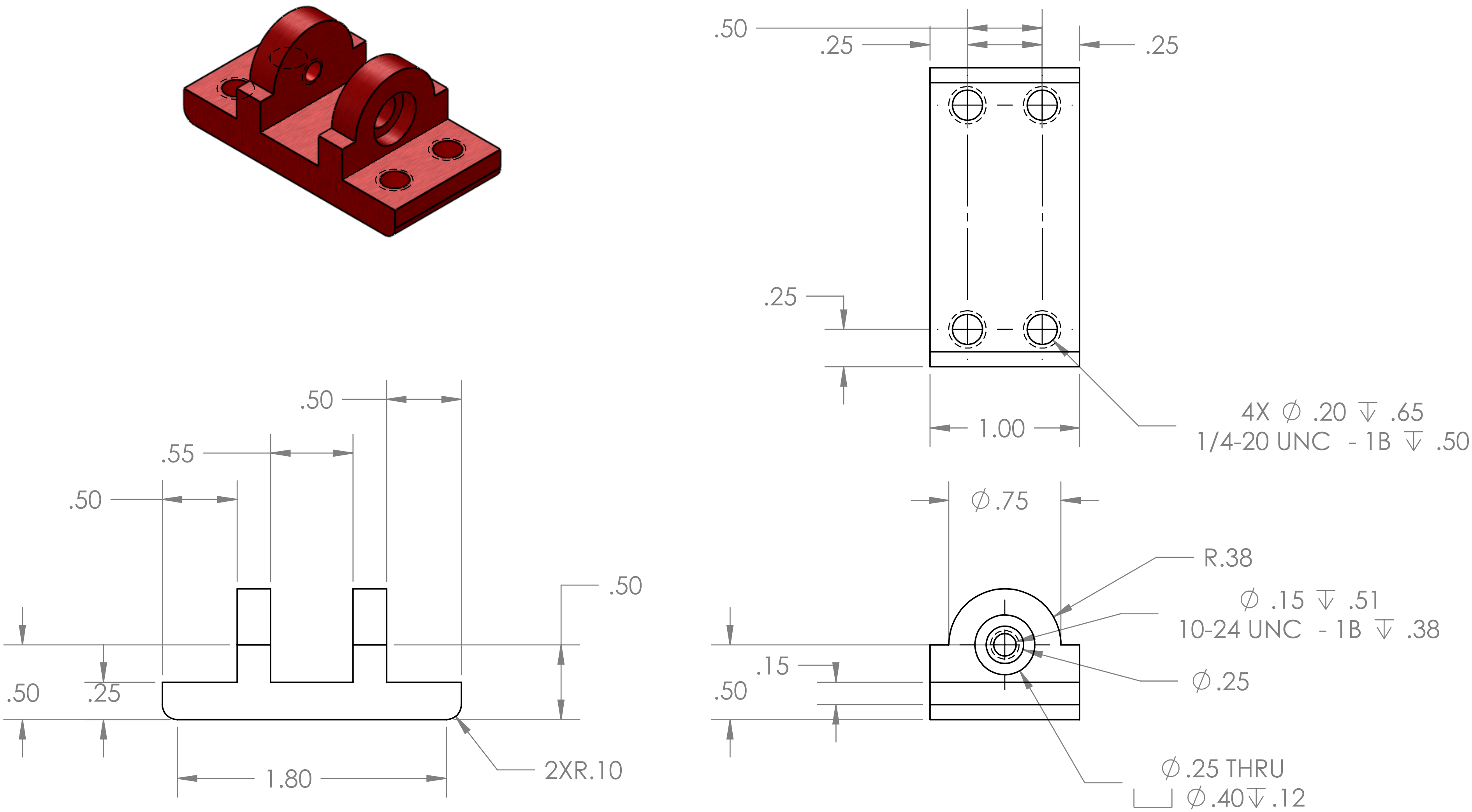
TITLE:

SIZE | DWG. NO. | REV

MountAssembly

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



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			CHECKED			
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS:			SIZE A DWG. NO. PivotJoint REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:1 WEIGHT: SHEET 1 OF 1

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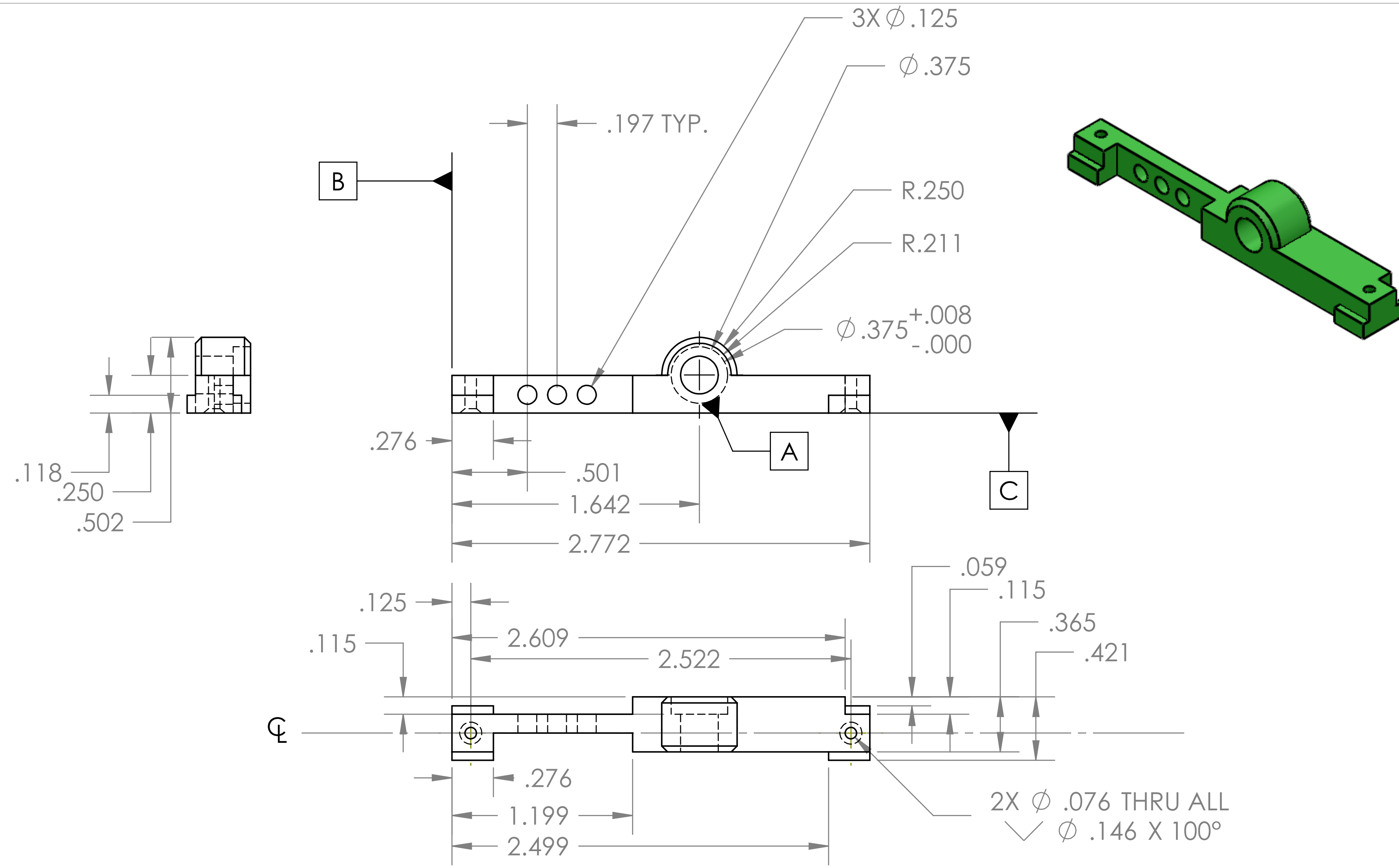
5 4 3 2 1



NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

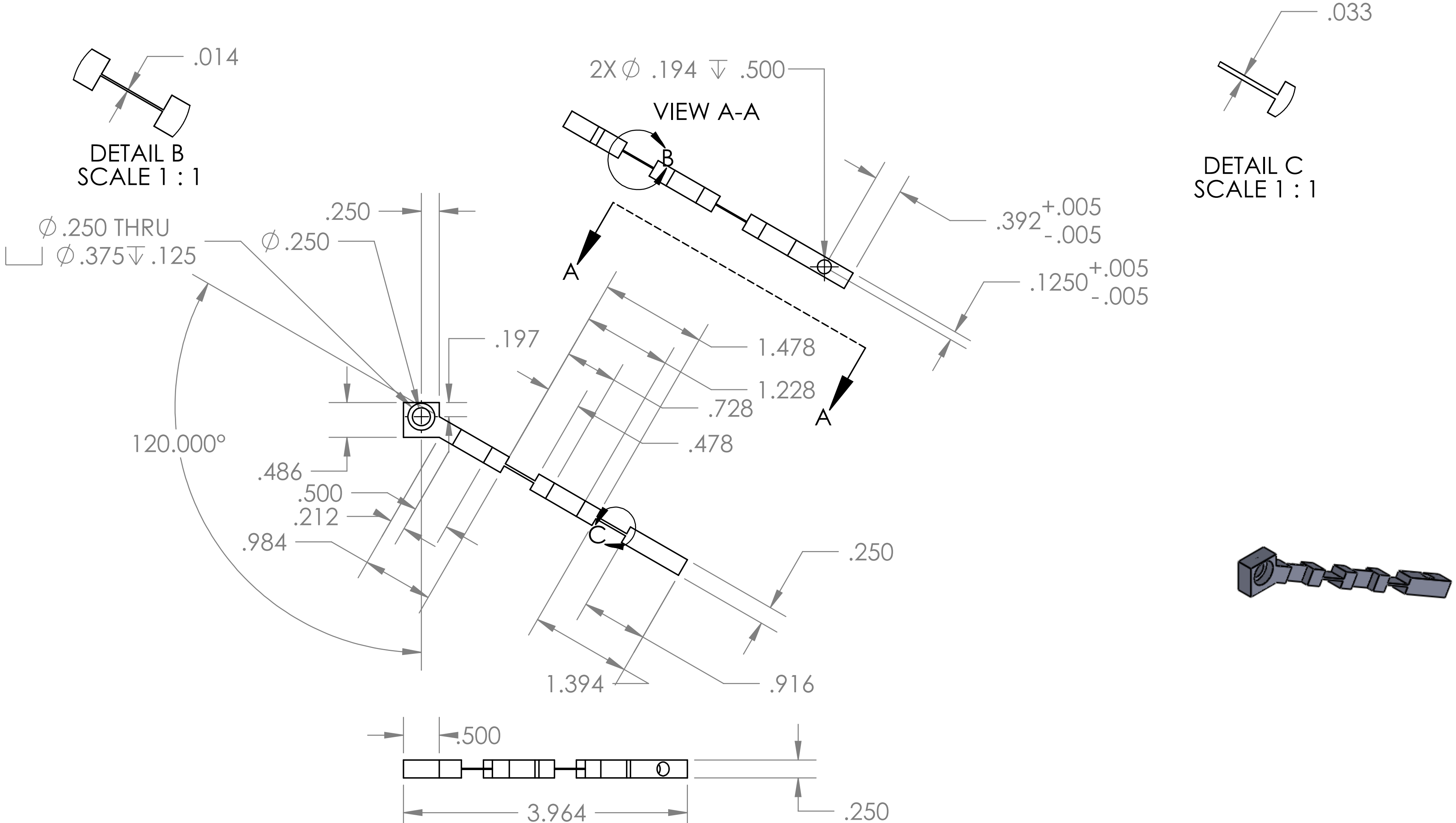
COMMENTS:

TITLE:

SIZE	DWG. NO.	REV
A	swing arm	

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

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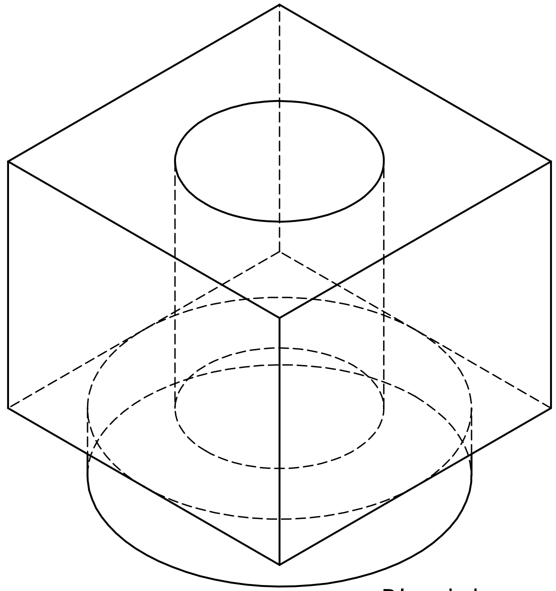
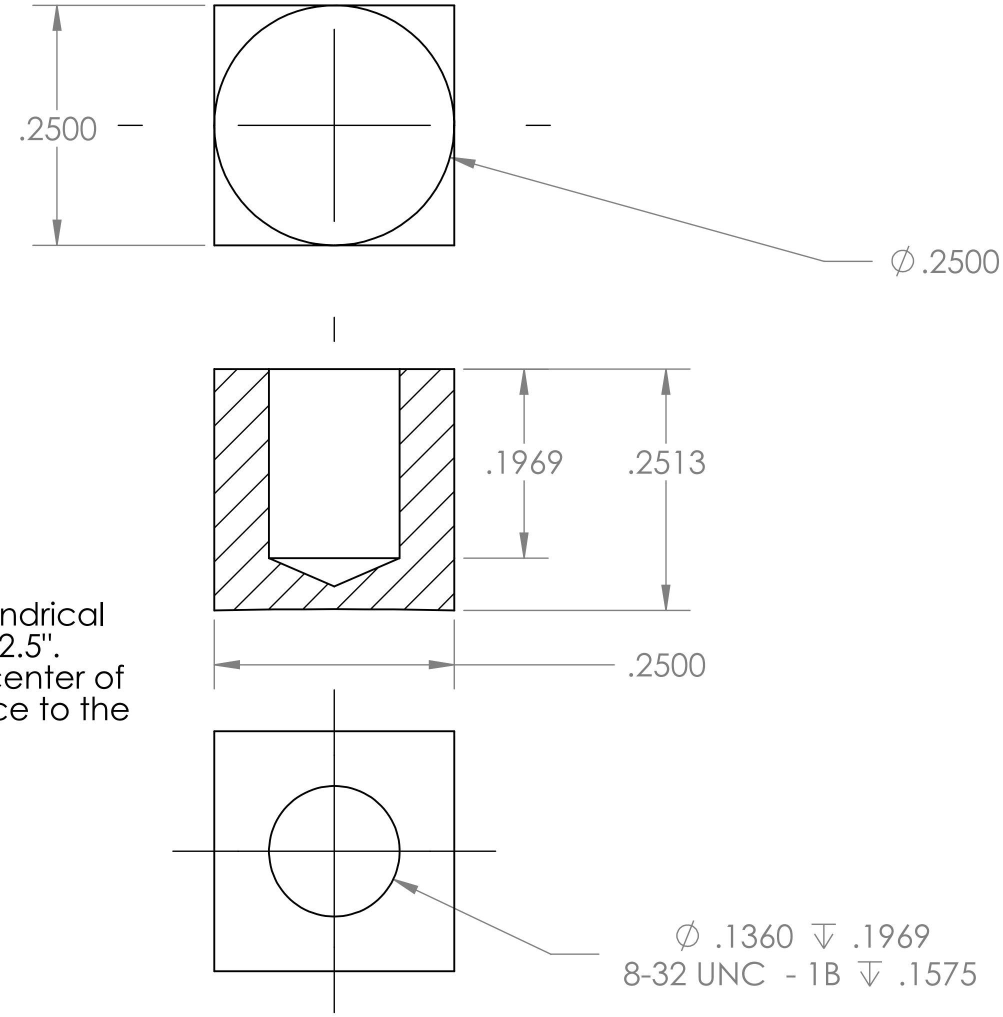
TITLE:

SIZE	DWG. NO.	REV
A	TangentArm	
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1





Block has a cocave cylindrical radius of curvature of 12.5".
The CC is aligned with center of the block.Vertex distance to the top of the black is .25in.

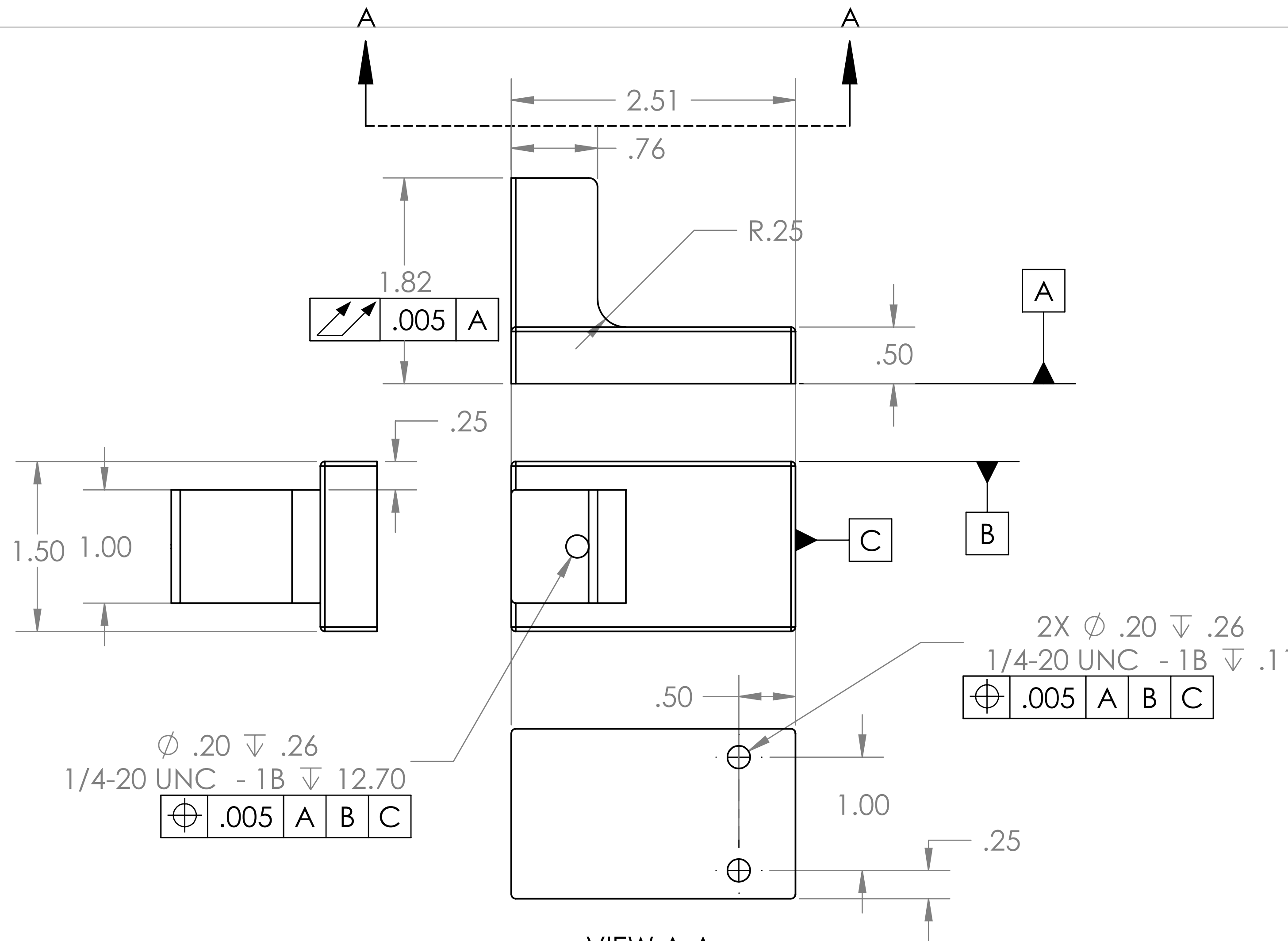
		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

SIZE	DWG. NO.	REV
A	TangentPuck	

SCALE: 5:1 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



VIEW A-A
SCALE 1 : 1.5

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

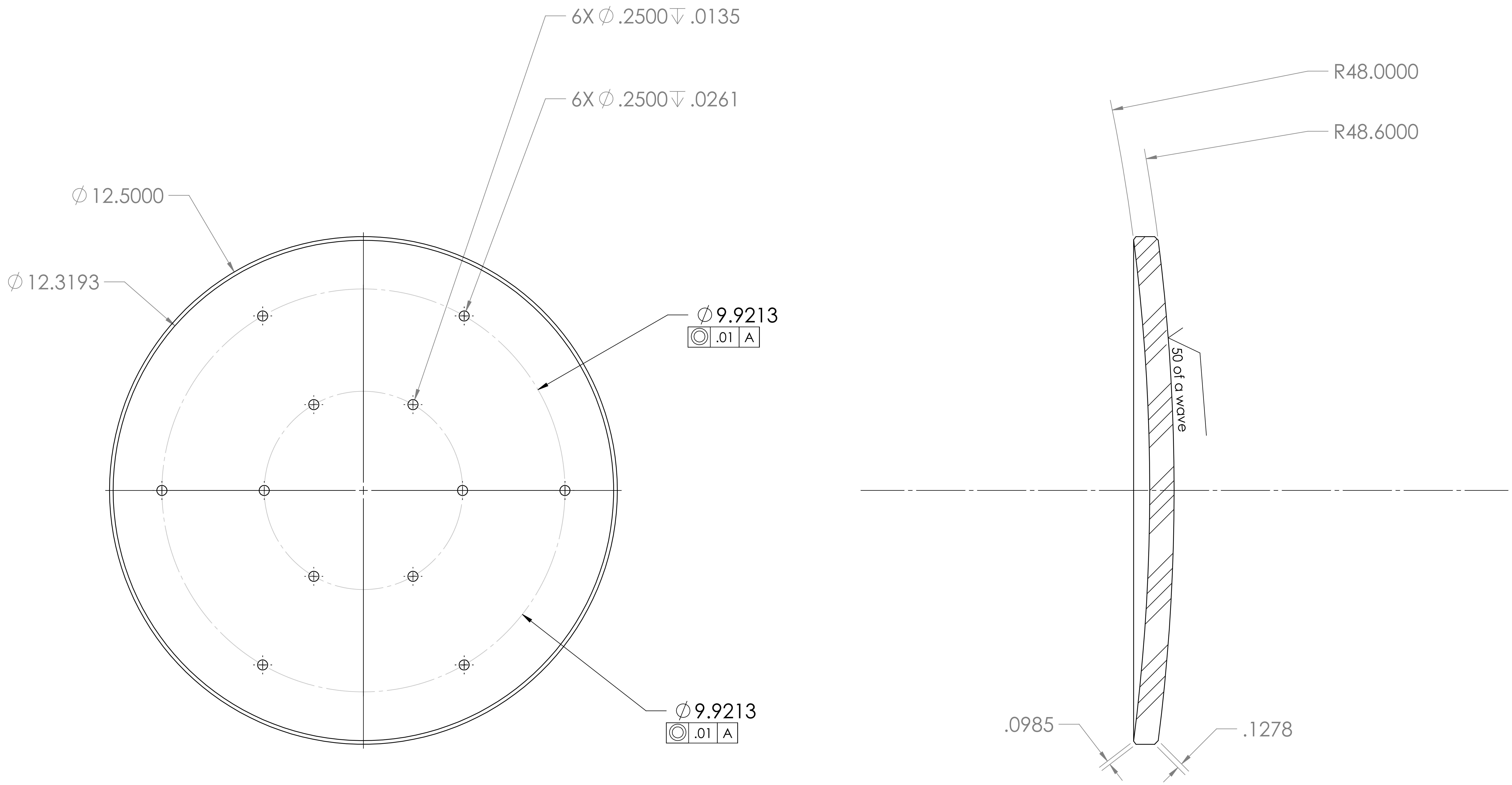
COMMENTS:

TITLE:

SIZE	DWG. NO.	REV
A	ThermalBlock	

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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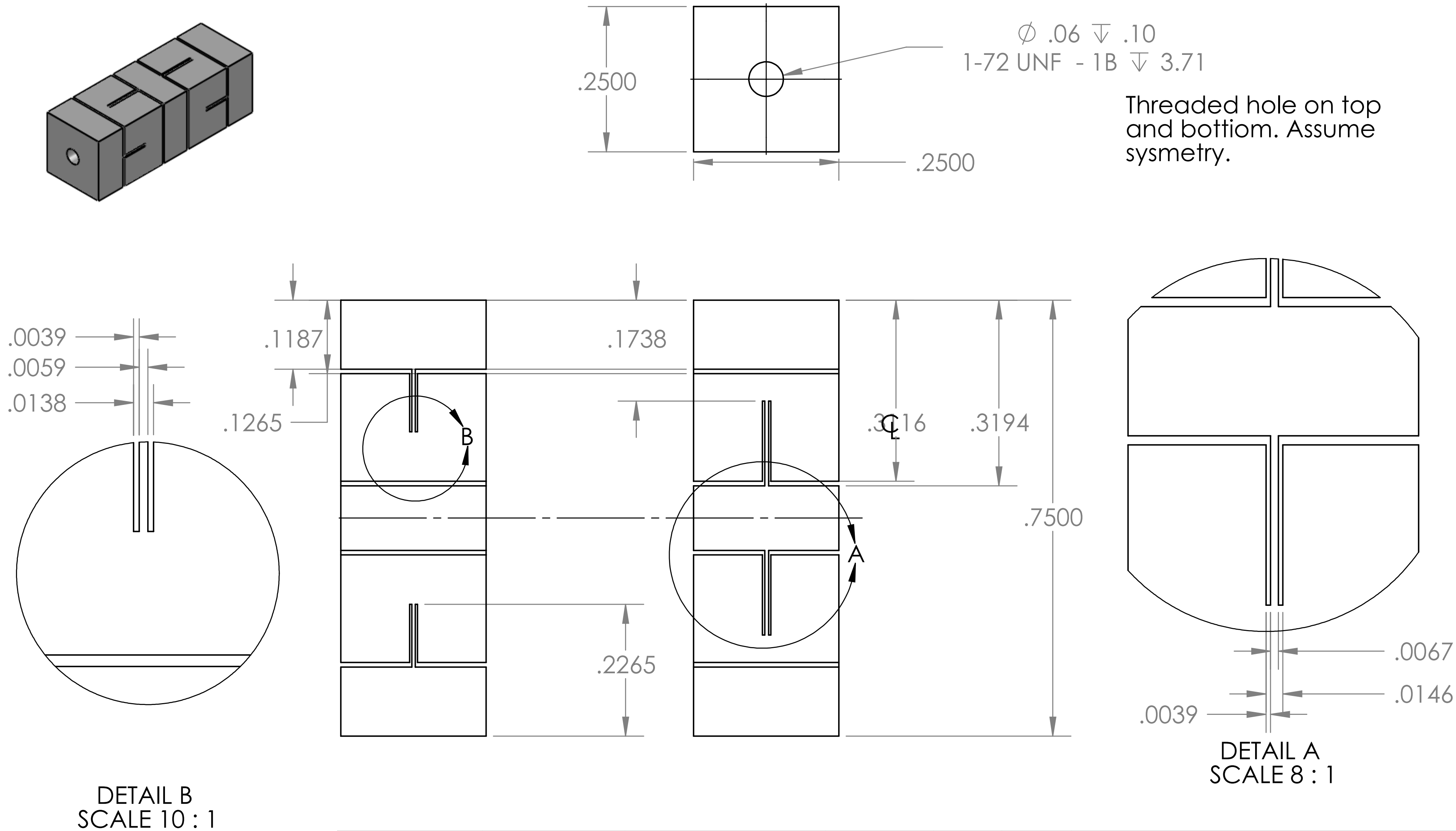
NEXT ASSY	USED ON	UNLESS OTHERWISE SPECIFIED:		NAME / DATE
APPLICATION		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	
			CHECKED	
			ENG APPR.	
		INTERPRET GEOMETRIC TOLERANCING PER:	MFG APPR.	
		MATERIAL	Q.A.	
		FINISH	COMMENTS:	
		DO NOT SCALE DRAWING		

TITLE:

SIZE C DWG. NO. ThinMirror2 REV

SCALE: 1:2 WEIGHT: SHEET 1 OF 1



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			CHECKED			
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS:			SIZE **A** DWG. NO. **UglyFlexure** REV
NEXT ASSY	USED ON	FINISH				
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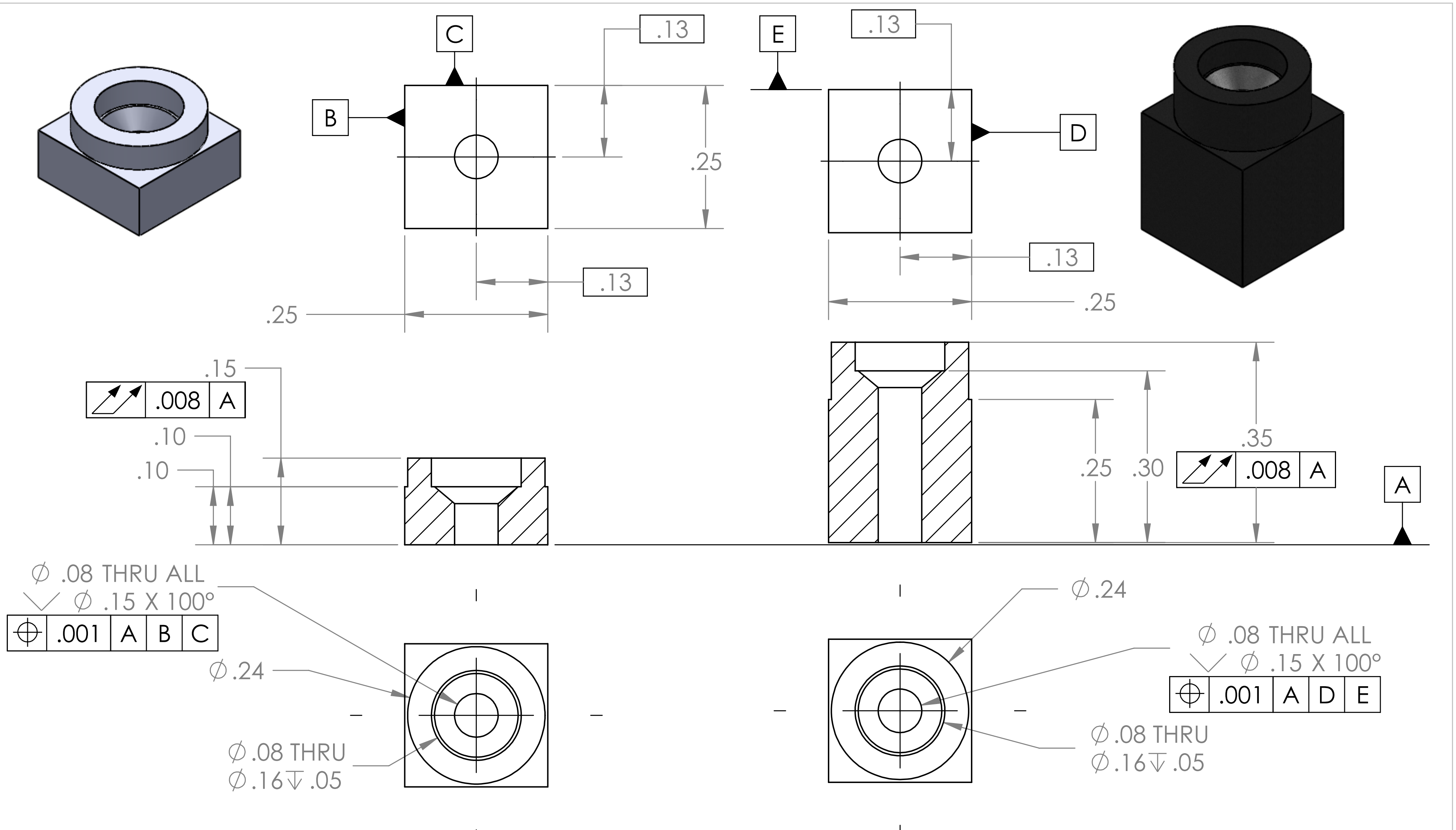

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			CHECKED			TITLE:
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
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NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 5:1 WEIGHT: SHEET 1 OF 1

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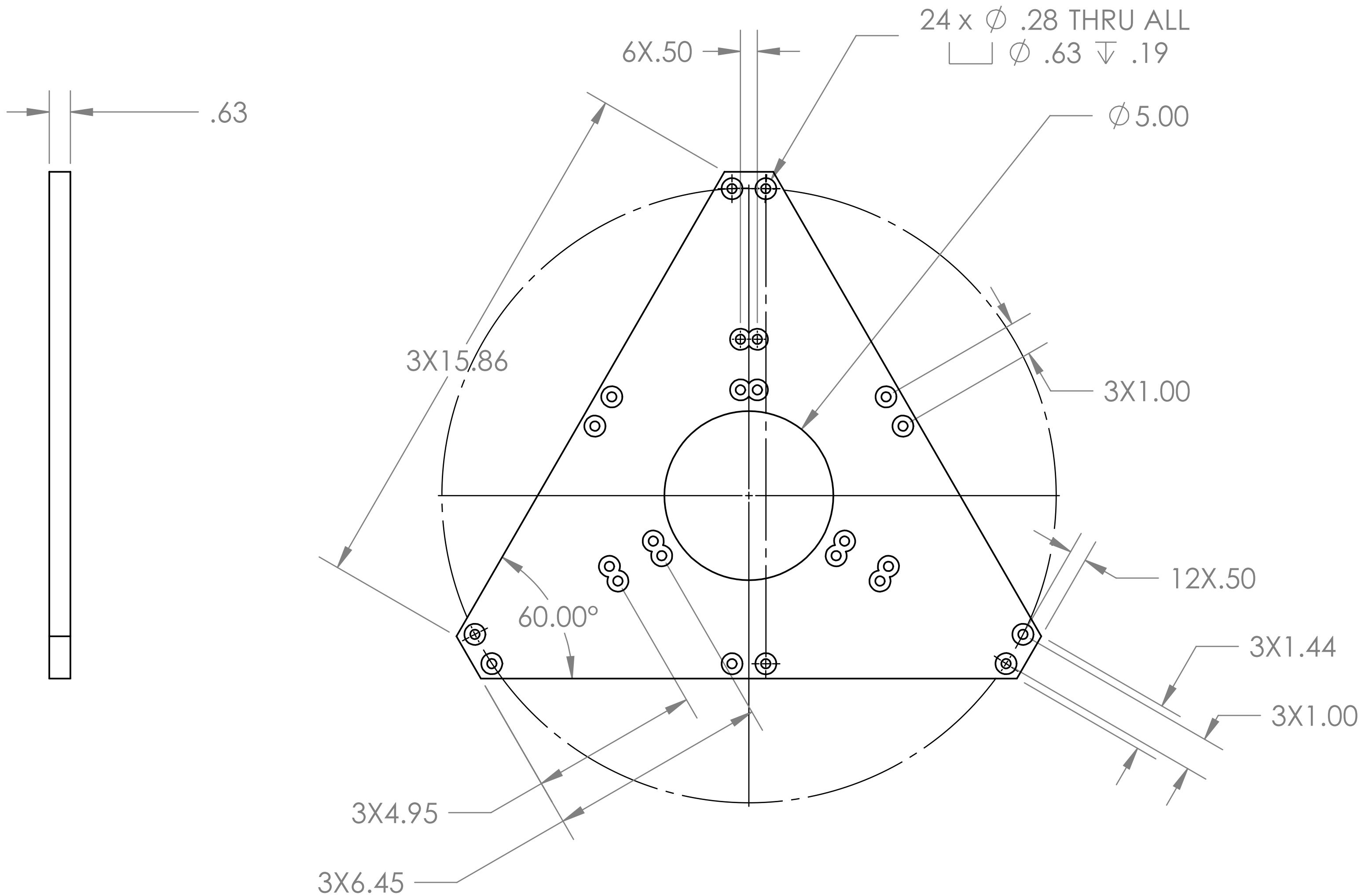


UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

NEXT ASSY	USED ON
APPLICATION	

TITLE:

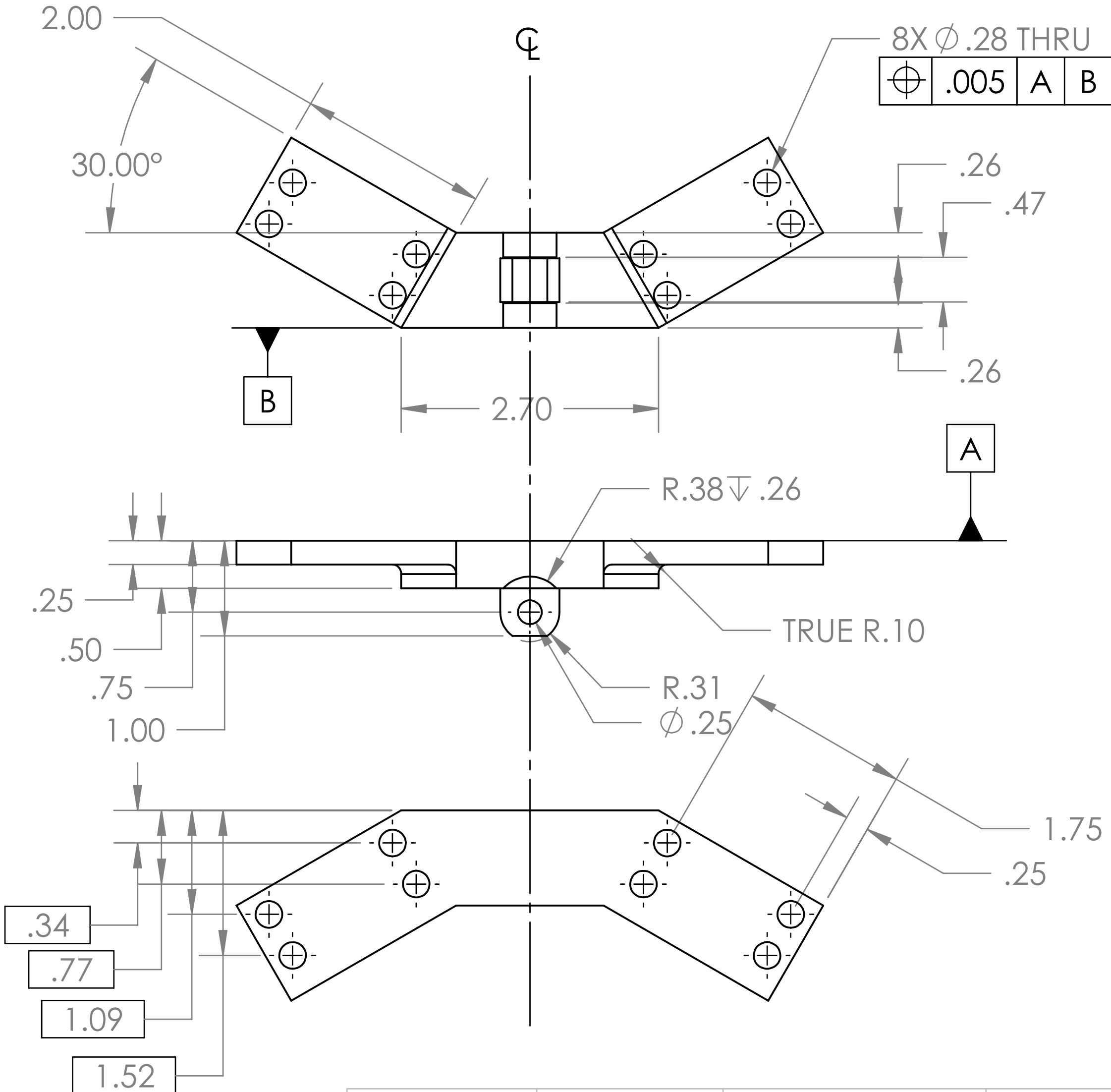
SIZE A

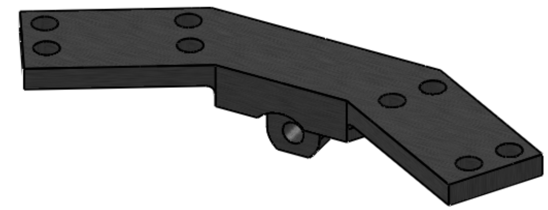
DWG. NO. BasePlate

REV

SCALE: 1:5 WEIGHT: SHEET 1 OF 1

5 4 3 2 1





NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

SIZE A DWG. NO. Double Pivot Arm REV

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1