Preliminary Design Review

Optimal Support Solution for a Meniscus Mirror Blank

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 problem 5 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 peramters.

Develop a mount concept for a variable orientation 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). 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

- The project requirement will be the same as the project requirements set for problem five, which include the following:

This must achieve < 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

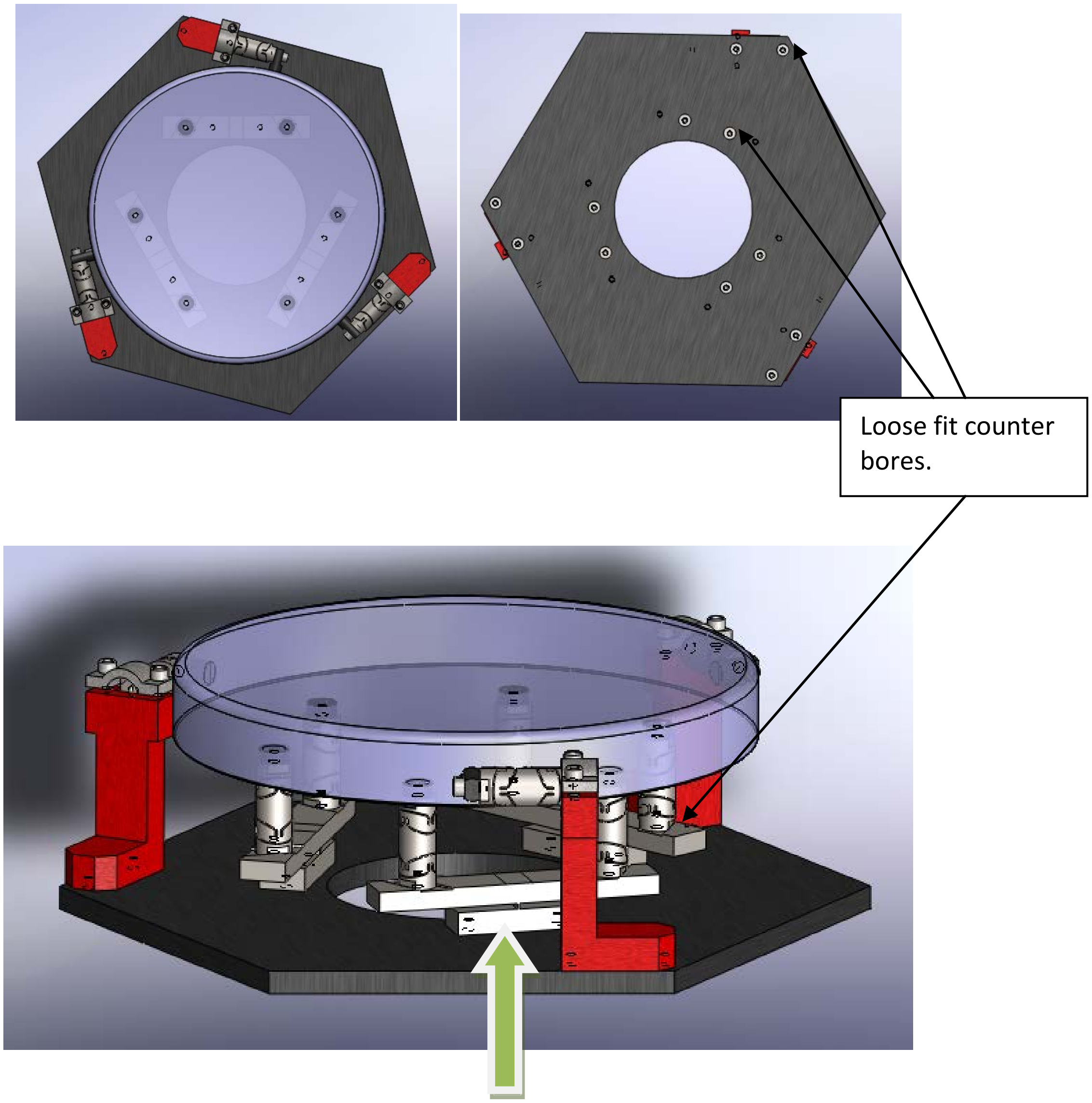
The main concern for these two parameters would be the flexures. The flexures would have to be able to be compliant enough so that the temperature variations would allow compliant motion and loading form shock would not yield the material. The mirror so far way about 8 lbs, if there is nine point contact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

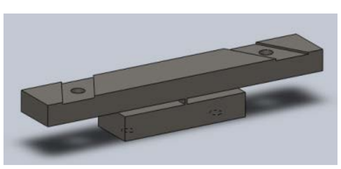
Design Concept

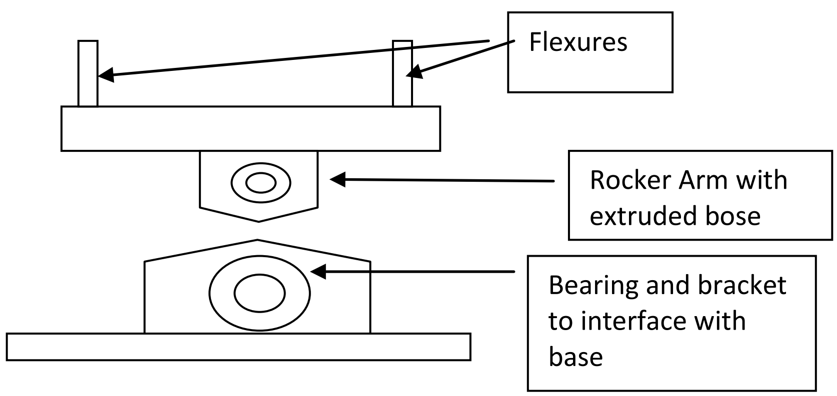
The model proposed for this design is based off of a previous design created (problem 5). Initially it was assumed that the thickness of the mirror would have extreme effects on a six point axial support and a 12 or 18 point support would be needed. But after an inspection of the rms surface error induced from self weight deflection it was found that a six point support would offer an acceptable solution. Originally when this model was analyzed the Zernike power term was not removed, but the system requirements were met. The new meniscus type mirror was analyzed with the Zernike power term removed and it was found that a nominal 8.2nm rms change in wave front was created by self weight deflection. This implied that since the support was randomly located, a better solution could be found by optimization.

Since this model is half the thickness and a sixth of the weight than that of the previous mirror analyzed, the flexure system will have to be changed significantly. The rocker arm will now incorporate a bearing to minimize moments induced by rocker arms. The thickness of the flexure leaves in the compact pivot flexure will be decreased so the reaction moment created by misalignments and thermal expansion shall be mitigated.

The tangential support shall also be maintained in its functional form but will have modifications to decrease stiffness of flexures. This will mitigate large reaction moment that distorts the mirrors surface. The nominal self weight deflection is higher but at the same time the weight of the mirror has significantly decreased. Below is the design form of the previous model.







Manufacturing issues

Manufacturing issues are going to be decreased by increasing the possible tolerance on the axial support by converting the rocker arms into a roller bearing rockers. The previous rocker arm required wire EDM. It is still to be determined whether or not the Evil Flexure model will be incorporated into the design. The Evil Flexure offer great efficiency in stiffness and reduce the need for long flexure element but are expensive to have built. To build the evil flexure the part must first be shaped using a mill and lather, and then wire EDM'd. This requires the budget of the design be increased significantly. While there is no requirement for costs, it makes the model somewhat un practical.

Design path and Decisions

Initially the assumption made were based off of a paper written by Lubliner and Nelson[1`]. The paper documented the deflections of plates do to self weight. Since the meniscus mirror could be thought of a plate, the anlaysis was initially going to be driven by these equations.

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

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

ϒN = Support Constant

q = the applied force per area

E = elastic modulus

ν= Poisson Ration

h = plate thickness

r = plate radius

A = plate area

N = number of support

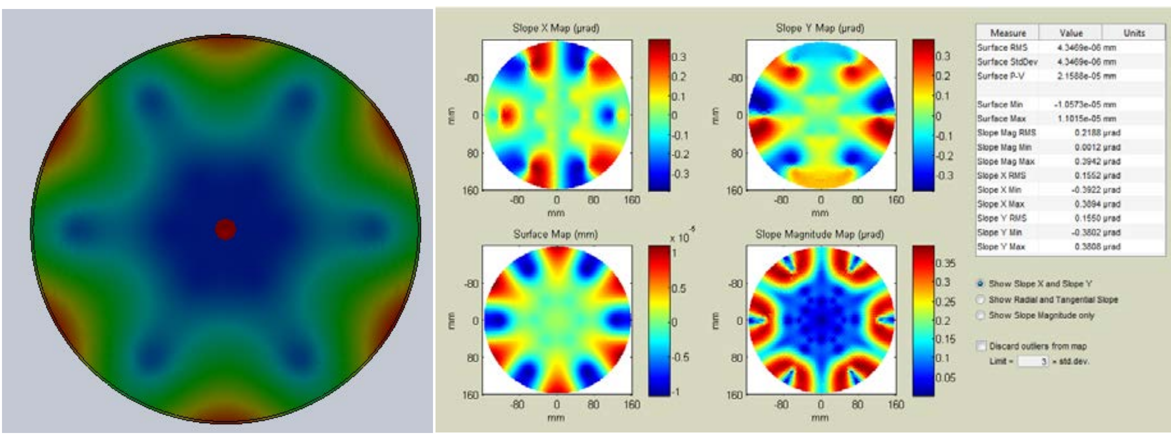
D = flexural rigidity

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

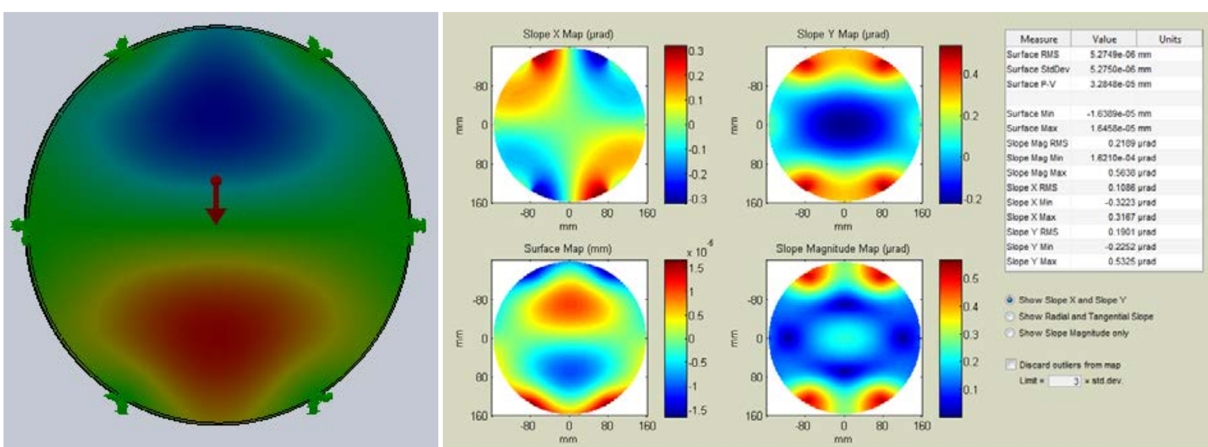
ϒN	q	E	ν	h	r	A	N	D	$\delta_{Vmax\text{-}rms}$	λ/50	RMS	Req	Margin
3	2.19	7E+10	0.17	0.015	0.318	0.07951	6	20533	5.61888E-08	1.266E-08	5.75974E-08	2.00E-08	-3.76E-08
2.4	2.19	7E+10	0.17	0.015	0.318	0.07951	7	20533	3.30253E-08	1.266E-08	3.53687E-08	2.00E-08	-1.54E-08
3.8	2.19	7E+10	0.17	0.015	0.318	0.07951	9	20533	3.16322E-08	1.266E-08	3.40716E-08	2.00E-08	-1.41E-08
1.94	2.19	7E+10	0.17	0.015	0.318	0.07951	12	20533	9.08386E-09	1.266E-08	1.55818E-08	2.00E-08	4.42E-09

2.32	2.19	7E+10	0.17	0.015	0.318	0.07951	15	20533	6.95243E-09	1.266E-08	1.44434E-08	2.00E-08	5.56E-09
1.89	2.19	7E+10	0.17	0.015	0.318	0.07951	18	20533	3.93322E-09	1.266E-08	1.32569E-08	2.00E-08	6.74E-09

From this analysis it was concluded that the support system would at least need to be 12 point. From there the mirror was modeled and then simulated to determine deflections. The analysis began by inspection of a 12 point support. It was found that the performance would be work accpetionally without even attempting to optimize it. Because the weight of the mirror had significantly decreased it became evident that the need for such a support might not be necessary. Similarly it was assumed that a six point lateral support would be needed. After analyzing the nominal deflections of the six point support type it was noted that the need for a six point support would not be necessary.



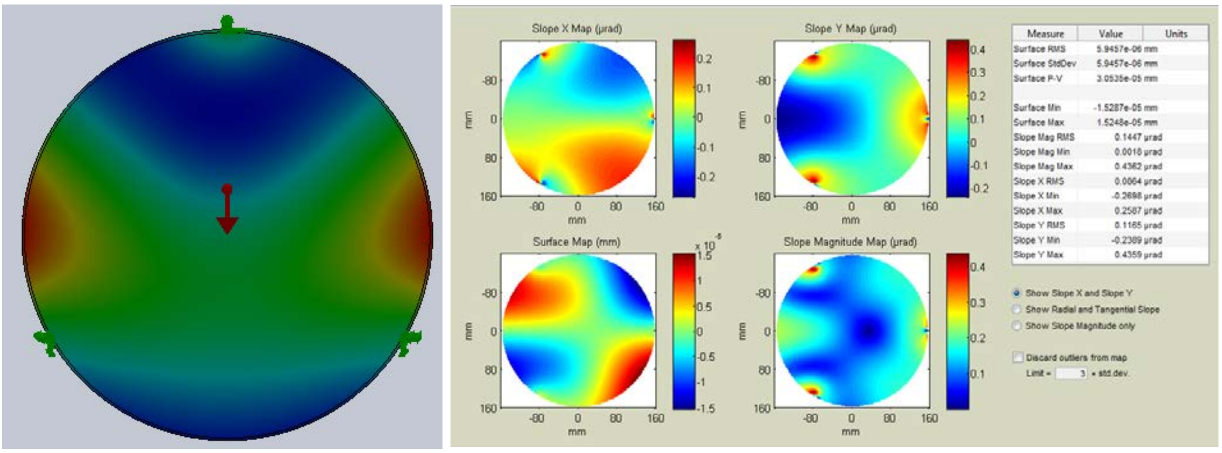
12 point axial support creates a nominal 4.3nm rms surface error



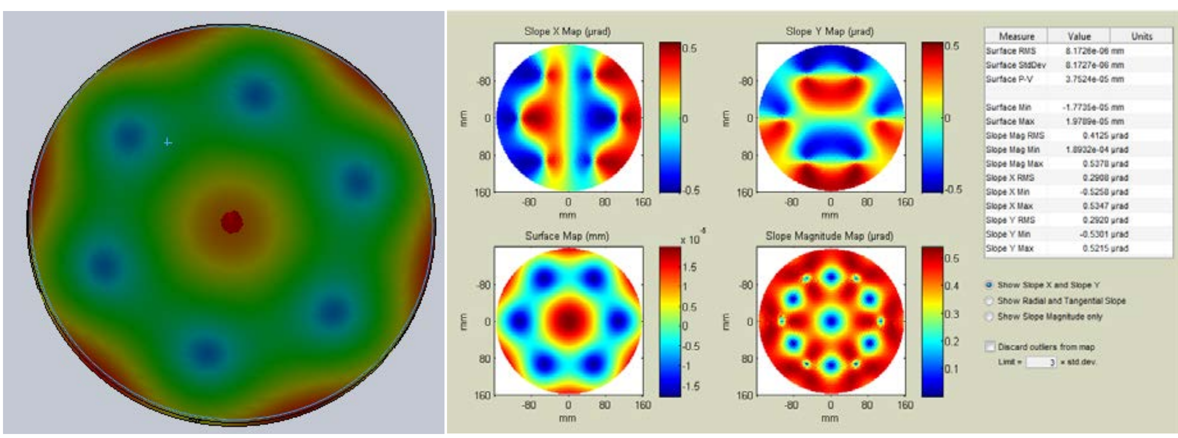
6 point lateral support creates a nominal 5.3nm rms surface error

Based on the data above the analysis was driven to determine new possibilities that would offer easier, less expensive, and possible to manually optimize. So a six point axial support was analyzed followed by a three point lateral support.

Three Point lateral Support (removing power, tilt, and piston)



3 Point lateral support to have ~5.9nm rms surface distortion.



6 point axial support gives a nominal 8.2nm rms surface distortion

By deciding to go with a six point support the model is greatly simplified. Simplicity is imperative to create a feasible method to creating an optimal solution. For a twelve or eighteen support, the optimization to do manually would be nearly impossible. If two variable were designated to determine the optimal solution, the square of the sampling points would be needed to determine the tendency of the deflections. The point at which the solution converges to local minima would dictate the number of sampling points.

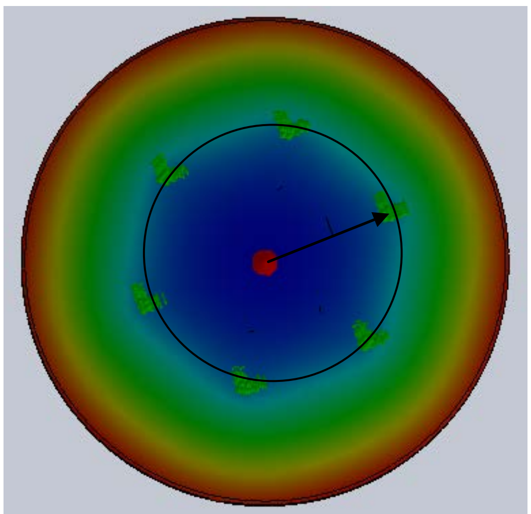
With this design choice the requirements will dictate the outcome of the final design by properly choosing flexure and bearings. The thin meniscus mirror will be susceptible to large distortions with induced reaction moments. For this reason the stiffness of the flexures is a key parameter. The model will be constructed such that reaction forces are greatly minimized to the point that the real reaction moments and forces shall only originate from affects of thermal expansion rather than misalignment due to tolerances.

Important Choices Left to be made

- Flexure design
- Bearings

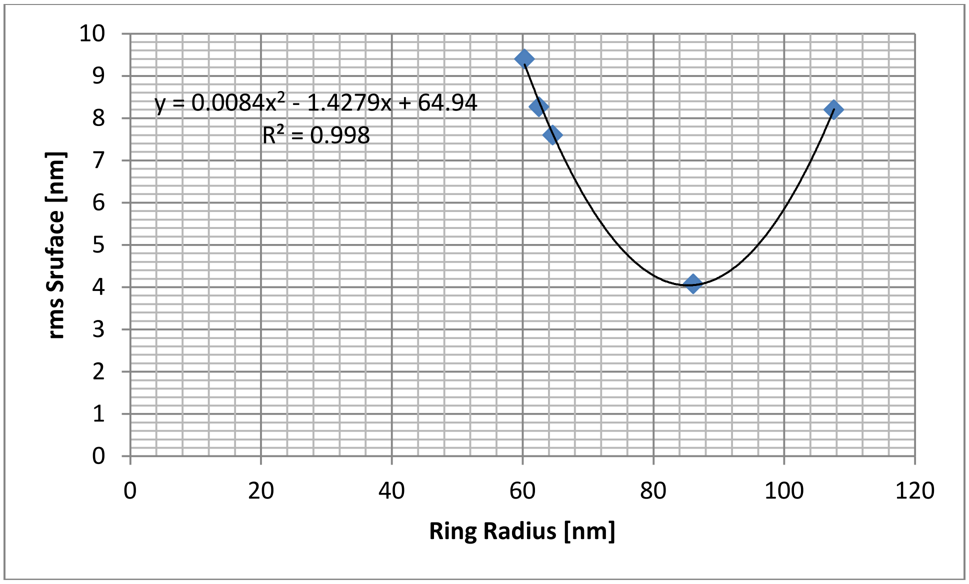
Analysis still needed to create Design

The main purpose of this project is to optimize a support location. The support locations emulate a ring. I ring would be the best support type but is hard to create because it over constrains the mirror blank. The kinematic mount type tries to simulate the affect of supporting the mirror at discrete points. As more points are added, the solution converges to that of the ring support. So to optimize only on variable will be needed, the radius of the ring. This will be varied and plotted to find the relative minima of the plot.



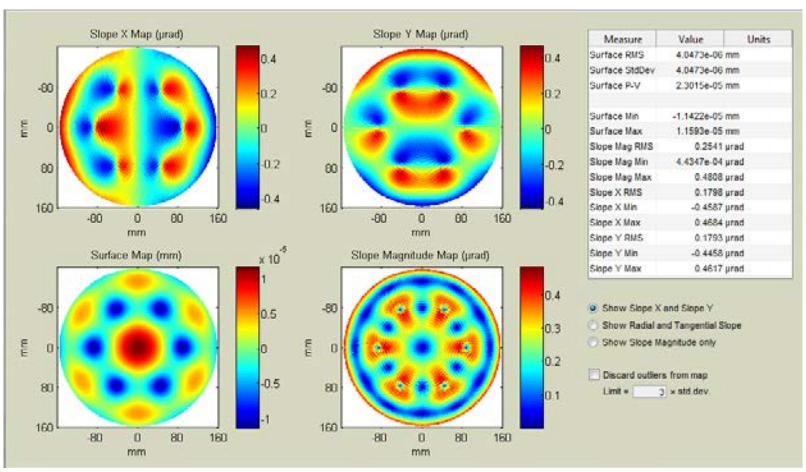
Geometry of Ring Radius

1. Find best nominal position to locate axial supports
    - Plot rms surface error as a function of ring location



$$\frac{dy}{dx} = 2 * 0.0084x - 1.427$$
$$x_{optimal} = 84.994mm$$

Results



Optimal axial support location results in a 4nm rms surface error

2. Sensitivity analysis due to reactions forces
3. Perturbation analysis of the puck
4. Flexure stiffness capabilities
    - Axial
    - Bending
    - shear
5. Tolerances needed to meet requirements
    - Variation in mechanical tolerances and thermal geometries

I fair amount of detailed analysis is still required to insure the system will meet requirements.

The analysis will need to determine the reaction moment induces by the flexure 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 $M_r = F \,.\, f \,.\, (d/2)$

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

- 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)