Design a German Equatorial Mount

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INTRODUCTION

After a question from a planetary observer/telescope maker was posed to me concerning telescope mounts a reference came to mind that may interest other telescope makers. Fred Tretta published a paper in the Riverside Telescope Makers Conference at 1980 that detailed the mathematical treatment of this fascinating subject [***Tretta,*** 1980]. Several other related articles will be referred in this paper. Many of our colleagues build their own observing equipment or redesign and modify existing telescopes to make them more stable for high powered observing. There are several caveats and debatable aspects of telescope mount design that will be discussed in the next section.

There is nothing more frustrating than a shaky mount that poorly tracks the planet you are studying. For those who photograph or use CCD cameras the tracking and alignment of the telescope mount is as important as good optics. To keep a magnified image of an extended object in place during exposures the planetary telescope must be at least ten times more stable than one used for deep sky photography and hundreds of time better than for visual observing.

BASIC CONSIDERATIONS

Generally, a planetary observer desires a permanent mounted or roll around telescope where portability is not paramount. To keep your telescope from moving about at the slightest touch or oscillating in the wind the mount must be massive and very strong. Rigidity of the mounting parts of the telescope, as well as materials that can take a lot of stress is desirable for mounts. Also, good dampening is important to quickly absorb and dissipate oscillations that may be induced into the mount.

The only materials available to the amateur for practical use are aluminum and steel. Other types of materials, such as titanium, stainless steel, hardened aluminum, and special hardened polymers are expensive and hard to work with. Generally, ordinary steel is more desirable than aluminum, even though aluminum is lighter than steel. Steel is stronger than and nearly as stiff as aluminum, an important consideration for dampening. However, if one uses aluminum then the diameter of the shaft must be increased because the *modulus of elasticity* of steel is three times that of aluminum. This also requires larger bearings and must be added into the cost of the system.

If one desires the same strength using a material other than steel they must calculate the equivalent diameter and find the nearest bearing size. Also, if you don't have a couple of steel shafts lying around then you must consider the difference in cost in the materials you wish to use.

TYPICAL INSTRUMENT FOR IMAGING

A typical planetary telescope might be a 12.5" (31.75cm) f/7 Newtonian and the combined weight of the optical tube assembly (OTA) and saddle weigh 100 pounds. This example will be discussed here as our instrument for CCD imaging and photography. The telescope is 8′ (2438.4mm) long and focuser is 53″ (1346mm) from the axis point of gravity along the tube assembly. CCD technology is used in this discussion instead regular photography because the image scale is smaller and CCD chips are usually much smaller than a frame of film. Exposures are shorter so a longer effective focal length can be used; however, due to the small area the CCD chip is more susceptible to telescope movement than film.

Let's say you have a couple of two-foot long, 1.5-inch steel shafts lying around and are curious as to the equivalent strength and weight of using a hollow shaft or even an aluminum shaft instead of steel. To find how much the 1.5-inch steel shaft will bend or flex under a load and compare it to a hollow or aluminum shafts we first dig out some mechanical equations. Two dimensions are involved here, bending and torque. Engineers of often refer to the two quantities to describe these two as: linear deflection and tortional deflection [***Albrecht***, 1989]. The ***moment of inertia*** use to calculate **linear deflection**:

$$Is = \pi D^4 / 64 \text{ (solid shaft)} \qquad (1)$$

$$Ih = \pi(D^4 - d^4) / 64 \text{ (hollow shaft)} \qquad (1a)$$

where π is 3.14159 and **D** is outside diameter and **d** is the inside diameter of the shaft.

For example, ***Is*** for a 1.5-inch solid shaft will be: $\pi(1.5^4)/64 = 0.2485$. If one is comparing a solid shaft with a hollow shaft then the outside diameter (**OD**) and inside diameters (**ID**) should be selected to give the same moment of inertia as the solid shaft. Some suggested the OD increased to 1.32D with a D/16 ratio for the wall thickness [***Brooks***, 1976]; however, I found in this case the OD will be **1.2468X** larger than the solid shaft with the same wall thickness ratio. So, the hollow shaft should be: 1.2468D or 1.8702″. OD then becomes D = 1.8702″ and ID becomes D – (2D/16) = 1.8702″ – 0.2338″ = 1.6364″. The hollow is then: ***Ih*** = $\pi(1.8702^4 - 1.6364^4) / 64 = 0.2485$; the same as the solid shaft.

The simple equation for linear deflection (Δ) of a cantilever is: $\Delta = WL^3 / 3E\mathbf{\mathit{Is}}$, where **W** is the weight of the load, **L** the length of the shaft, and **E** is the modulus of elasticity [***Tretta***, 1980]. At 70° F the **E** for Carbon-moly and Chrome-moly steel is ~ 30×10^6 and aluminum is 10×10^6 (See Figure 1). **NOTE**: In his 1980 paper Table-I the equation $I = 0.098D^4$ ($\pi D^4 / 32$) is for the tortional deflection.

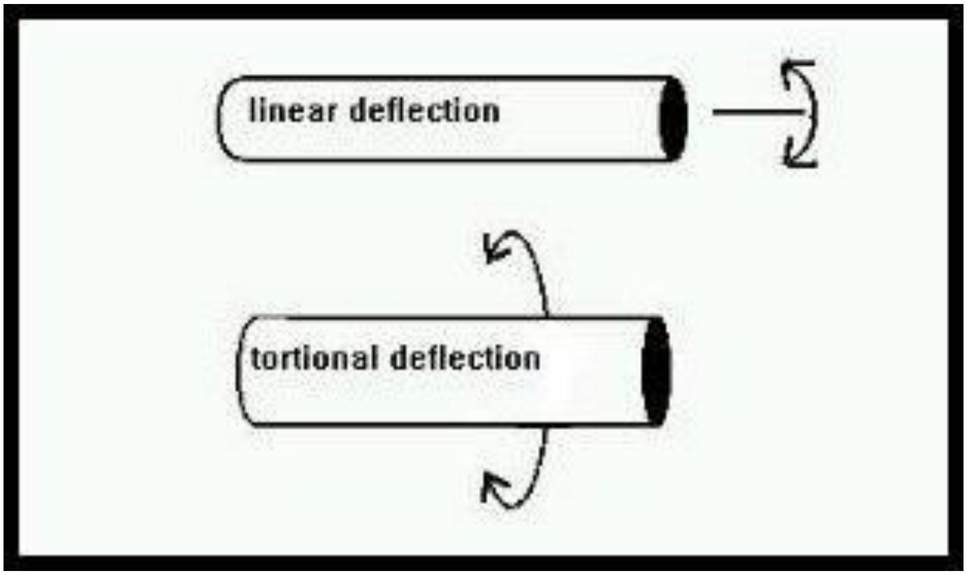


Figure 1. Simple illustration of the two elements to consider when selecting shaft sizes, linear deflection and tortional deflection.

Since our shafts are mounted in a housing separated between at least two bearings, as show in Figure 2, it is subject to two deflections; 1) in the length of the shaft extending out from the top bearing, or over hang, and 2) in the distance between the bearings. So, we must conjure up a little more complex equation [***Tretta***, 1980]:

$$\Delta = WCp^2 (2Cp + 3Lp) / 6E\mathbf{\mathit{I}} \qquad \textbf{(2)}$$

where **W** = OTA + saddle weight, **Cp** is the shaft extended beyond the top bearing, **Lp** is he distance between bearings, $E = 30 \times 10^6$ and ***I*** = moment of inertia

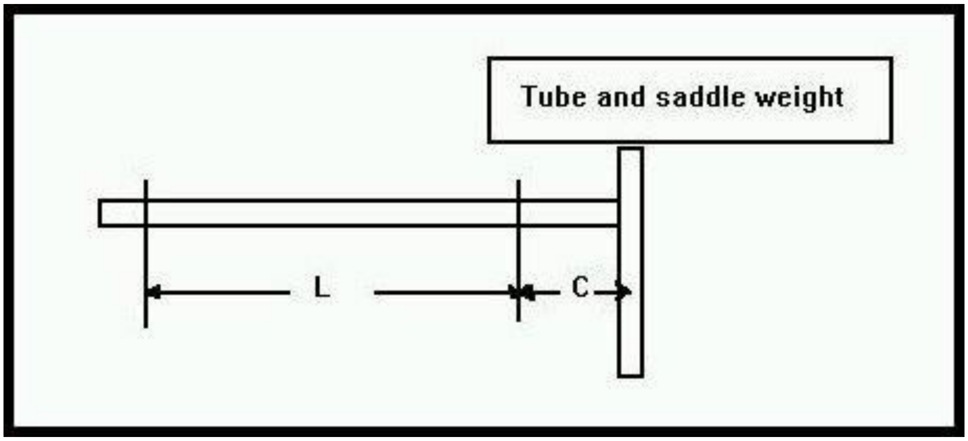


Figure 2. Simplified drawing of polar axis shaft with declination axis and telescope tube and saddle weight loading the shaft. Linear deflection results in shaft between bearings (Lp) and overhanging shaft (Cp).

If your OTA and saddle weigh 100 pounds and the bearings are separated by 12 inches, with 4 inches extended beyond the top bearing of the polar axis and the center of weight is at the end of the 4 inch extension (**Cp**). Now:

$$\Delta = [100 \times 4^{2} (2 \times 4 + 3 \times 12)] / (6 \times 30 \times 10^{6} \times 0.2485) = 0.001574'' \text{ (0.04mm)}$$

If a solid aluminum shaft is used then the diameter for the same linear deflection of 0.001574" we manipulate the above equations (2) then (1) we find a diameter of 1.974": $I = WCp^2 (2Cp + 3Lp) / 6E\Delta = 70400 / 60 \times 10^{6} (0.001574) = 0.7454$, then as a check:

$$\Delta = [100 \times 4^2 (2 \times 4 + 3 \times 12)] / (6 \times 10 \times 10^{6} \times 0.7454) = 0.001574'', \text{ therefore;}$$

$$I = \pi (D^4)/64 = 0.7454$$

$$D^4 = (0.7454 * 64) / \pi$$

$$D = \sqrt[4]{15.185164} = 1.974''.$$

EFFECTS OF WIND ON OUR TELESCOPE

So far the discussion has been about an unbalanced telescope tube and it would certainly not operate properly without counterweights. The design problem we need to solve is what effect an outside force, such a wind or thump from the observer's hand, has on our balanced telescope tube. The effect of wind on our telescope is discussed in this section. Wind is probably the one element in our design criteria that we should concentrate on most. Richard Berry gave a good talk at R.T.M.C. in 1980 on the effects of wind on Dobsonian telescopes and was a stating point for this discussion [***Berry***, 1980].

Equation (3) enables you to figure the amount of force on the exposed portion of your telescope tube, say 3 feet of the upper end of the 15-inch (1.25-feet) diameter tube, where the square area is 11.8 square feet (**2πrh**), where r = 0.635′, h = 3′. Then apply this force as the weight (Wf) in the calculations for deflections in the above. In the equation the constant **S** is 0.7 for a cylinder and 2.0 for a square tube:

$$Wf = A \times P \times Cd \qquad \textbf{(3)}$$

Where **A** = area (2πrh), **P** = 0.00256 V^2 and **Cd** (Drag coefficient) = 0.7 for cylinder

For a typical South Florida sea breeze or lake breeze of 5 to 10 MPH, find Wf from equation **3**: $P = 0.00256\ (10^2) = 0.256$, $A = 2\pi(1.5 * 3) = 11.8$ sq/ft;

$$\mathbf{Wf} = 11.8 \times 0.064 \times 0.7 = 0.53\text{-lbs. force (5 MPH)}$$

$$\Delta = [0.53 \times 4^2\ (2 \times 4 + 3 \times 12)] \ / \ (6 \times 30 \times 10^6 \times 0.2485) = 0.0000083'' \ (0.00021 \text{ mm})$$

We now have to consider what the shaft deflection of 0.0000083″ (0.00021 mm) translates to movement at the focuser, a long way down the telescope tube from the center of axis! If the distance (**Df**) from the center of the axis, or center of gravity, is 4.4 feet (1346 mm), α = 0.00021 mm then the focal plane at the camera chip will move:

$$CCD = \mathbf{Df} \tan \alpha = 1346 \tan 0.00021 = 0.005 \text{ mm}$$

What would this small deflection have on a magnified planetary image on your brand new CCD camera chip? Using a 3.97mm x 4.60mm rectangular CCD chip with 494 x 659 pixels, each pixel would be 5.6 microns (millionths of a meter) in size. A typical eyepiece projection for CCD images of planets we usually use f/26.5, so the projection magnification for this setup is 3.77 x 2235.2mm = 8,427mm or an image scale of 206,265 / 8,427mm = 24.5 mm/arcsec. The entire chip spans 5.33 seconds of arc (24.5/4.60) and each pixel covers 0.0081 arcsec (5.33/659). How many pixels will an image cover on the CCD chip?

$$Pix = efl\ (i) \ / \ 206.265\ \mu m,$$

where ***Pix*** image pixels, ***efl*** effective focal length, ***i*** image size in seconds of arc, ***µm*** = pixel size

Note: the ratio between the focal length and the linear dimension at the focal plane, called the image scale, is expressed in arc-seconds per inch or millimeter, using the formula 206,265 / EFL, where 206,265 is the seconds of arc in one radian (57.3 degrees) and EFL is the effective focal length. Image Size = θ (EFL) / 206,265, where θ is diameter of the planet in seconds of arc.

For an extreme example let's say we desire to image Jupiter's largest satellite, Ganymede, that is around 1.5 arcsec in apparent diameter and would occupy 11 pixels (8,427 * 1.5) / (206.265 * 5.6) on the PC screen or (4.6 * 11) / 659 = 0.077mm. For a movement of 0.021mm would scatter the image only: Pix = (0.005 * 659) / 4.6 = 1 pixels. This is an acceptable error; however, we still have to deal with the tortional deflection in the shaft. This deflection produces more movement at the camera and can be found by:

$$\delta = 584\ TL\ /\ 0.038XE \qquad (\mathbf{4})$$

where **T = W * C** , or twisting force measured in inch-pounds. **W** is the weight, **C** is the distance from the center of the torque axis to the weight, **L** is the distance from the drive gear or restrained end of the shaft to the weight, **E** modulus of elasticity, and **X =** D^4 for the solid shaft (See Figure 3).

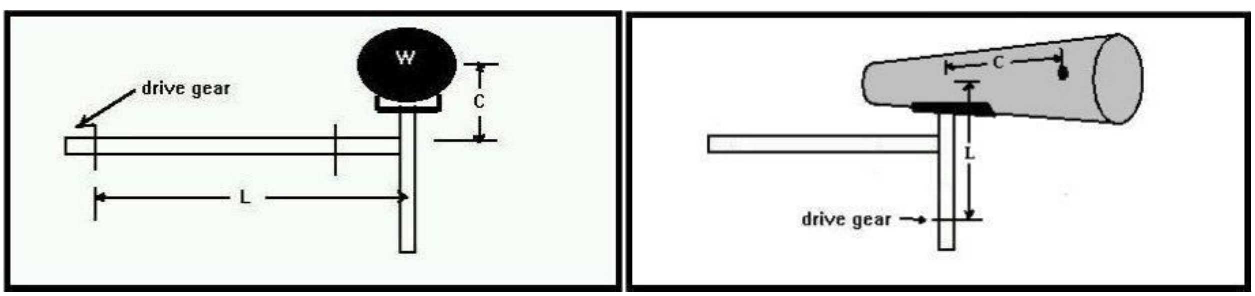


Figure 3. LEFT: Simplified drawing of polar axis shaft with declination axis and telescope tube pointed up. Counterweight (not shown). Tortional deflection results when external force causes unbalance to tube and saddle and twists polar shaft (L). Overhang of tube and saddle (C) from center if polar axis. RIGHT: Declination axis with telescope tube pointed up.

The polar shaft distance (L) is 16 inches to the weight, C is from the shaft to the center of the telescope tube weight, say 14.5 inches, T becomes 0.53-pound x 14.5 = 7.67 inch-pounds, and X = D^4 = 5.0625. The declination shaft distance (L) is 24 inches, C is 31 inches (92" – 30") / 2, so T becomes 0.53-pound x 31 = 16.4 inch-pounds:

$$\delta = (584 \times 7.67 \times 16) / (0.038 \times 5.0625 \times 30 \times 10^6) = 0.0124 \text{ radians} \quad \text{(RA Axis)}$$

$$\delta = (584 \times 16.4 \times 24) / (0.038 \times 5.0625 \times 30 \times 10^6) = 0.04 \text{ radians} \quad \text{(Dec Axis)}$$

For the RA axis a 0.53-pound wind force now translates to 0.0124 x 1346 mm or 16.7mm, but the real angular displacement is 0.7 arcsec arc (arctan 0.0125), or nearly 60 pixels (0.7/0.012) on the CCD chip. The same wind force applied the declination axis translates to 0.04 x 1346 mm or 54mm with the real angular displacement is 2.3 arcsec arc (arctan 0.04), or nearly 190 pixels (2.3 /0.012) on the CCD chip. Quite a force from a fairly mild breeze.

Magnifying the image of this little 1.5 arcsec Ganymede to 0.5-mm an effective focal length would need to be around 69 meters! A 41-cm telescope with an effective focal ratio of f/169 would project Ganymede to a half millimeter on the typical CCD camera chip. Well, if you don't believe this is possible, then Don Parker published an image of Ganymede he made using his 41-cm Newtonian and LYNXX-PC camera and he lives on the windy coast line of South Florida [***Parker,*** 1995].

Also, the frequency of the vibrations caused by wind flowing in the front and down the tube can be approximated by: 0.2 x velocity (mi/hr) / D (diameter in feet) [***Bely***, 2004]. In the example for the 12.5-inch f/7 telescope above it would try to oscillate at: (0.2 x 5) / 1.25 = 0.8 Hz

OTHER MECHANICAL CONSIDERATIONS

Shaft size is not the only important factor in mount stability. Distance between bearings play an important role in both linear and torsional deflections. After the telescope tube design and saddle have been optimized for strength and weight, one must begin design on the mount and acquire the materials. If you are like most telescope makers you have a favorite junk yard or other TM friends with plenty of junk laying around not in use.

If one were to consider temperature effects of steel and aluminum they will find that aluminum expands and contracts about twice as much as steel. An aluminum shaft in steel bearings will tighten and loosen as the ambient temperatures change, causing some slop in the system at times.

From the discussions above we can now concentrate on the elements the designer has control over; 1) distance between bearings of each axis, and 2) distance from the top bearing of each axis to the point where the weight loads that particular shaft. That is, the distance **C** , as can be seen in figure 2, and distance **C** in figure 3. The overhang from the top bearing of the polar axis shaft to the declination axis housing should be as short as possible. The telescope tube should be as close to the polar axis as possible without hitting the drive gear or its housing.

Using the discussion above a mount was designed for a 12.5-inch f/7 Newtonian with an eight foot tube. The tube is a rolled aluminum tube 0.0625-inch thick and

15 inches in diameter. The weight of the tube, mirror and cell and other components weigh in at 85 pounds. The requirement is to provide a stable instrument for CCD imaging in 10 to 15 MPH winds with a calculated error in position of 0.24 to 0.54 arcsec!

At our favorite junk yard, near the air port, we found a two 3-inch hardened steel shafts, four 3-inch I.D. tapered bearings, and a length of 5-inch steel water pipe to use as the axis housings. Two 14-inch lengths of the pipe were cut and the inside of each end was machined to fit the bearing race and were pressed in to place. The polar shaft and a steel block is threaded then the end of the shaft was welded in place to the block. Holes were made in the block to attach the polar shaft to the declination housing. The block also provided the stop to hold the tapered bearing in place. A steel plate and side supports were welded the declination housing.

The declination shaft assembly was made the same except the block was larger and was used to attach to the saddle. Each previously cut and machined shaft was snugly fitted in the bearing inside races and placed with bearings in the housings. Each shaft had been threaded to accept a ring to tighten or snug up the bearings with the housing. Both shafts were machined down to fit a 2-inch drive gear clutch assembly. The gear end of the declination shaft was threaded and a longer shaft was attached for the counterweight (See Figures 4 and 5). The next step is to weld plates and brackets to the polar housing to attach this "equatorial head" to a pier of some sorts.

NOTE: periodic error cannot be completely eliminated in worm gear drives due to the inherent inaccuracy in the gears; however, it can be reduced by "lapping-in" the gears using a hand drill and applying lapping compound to the gear surfaces. After removing the motor and reduction spur gears attach a hand drill to the worm shaft and spin the gears at a moderate speed for 10 or 15 minutes. Afterward clean the gears thoroughly to remove any lapping compound, metal debris and dirt so the gears will not continue to lap during normal use.

THE PIER

The pier is the foundation to the entire telescope system and can be a primary source of instability in the telescope mounts. It is often over looked by many and they use a simple rule is that the larger and heavier the better. However, two weak points mounts can be the point at which the equatorial head meets the pedestal (wedge) and the point at which the wedge is the coupled to the pier can nullify the advantages of a large pier.

The pier must attach to the equatorial head with a wedge that aligned with the latitude the particular observer is located. While is may be possible to weld together a steel wedge at the exact angle, it should never the less have some way to make small adjustments in azimuth and polar angle. Even the best made mounts will move about from seasonal temperature changes or just medal fatigue after years of use. As seen in figure 4, the pedestal or wedge is made from two, 1/2-inch 9" x 12" steel plates welded together with triangular wedges. The wedge assembly is attached to the pier using three welded steel tabs and 3/8-inch bolts. The wedge is closely aligned to the North, but can be fine adjusted with an eccentric screw for polar adjustment.

Figure 4. Telescope pier and wedge is made from two, 1/2-inch 9" x 12" steel plates welded together.

For a stable base for the pier and equatorial mount a concrete base was installed. Step one was to dig another 3-foot deep, 2x2-foot truncated pyramid shaped hole. After the concrete cured I set in the steel pier and bolted it down. A yard of concrete filled the hole.

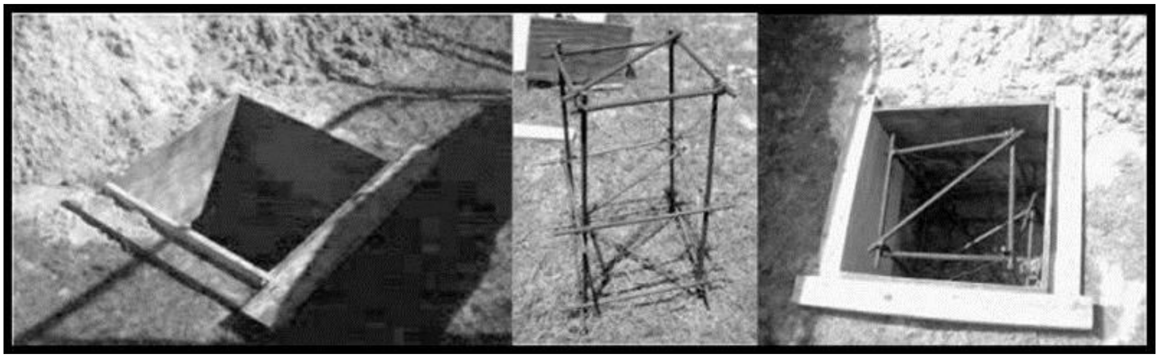


Figure 4a. LEFT: A hole dug in sand 3.5-feet deep Bottom is 3 x 3-feet square and top is 2 x 2-feet square (truncated pyramid). Frame in place is ¾"x24"x18" CDX plywood. CENTER: Rebar (steel reinforcing rods) welded into box frame to be used installed in hole and frame. RIGHT: Hole now is framed with Rebar cage in place and ready for pouring concrete.

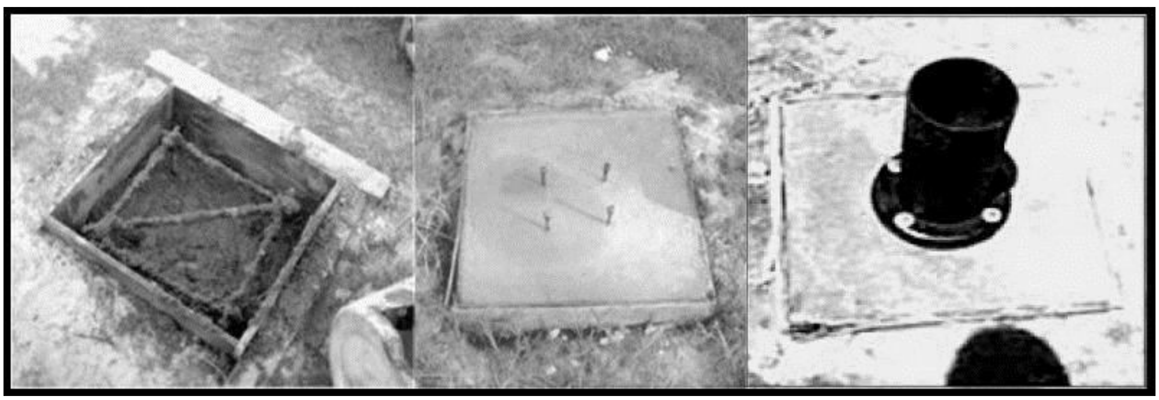


Figure 4b. LEFT: 5000-PSI fast setting concrete pour in progress. Constant mixing with shovel eliminates air pockets. CENTER: Concrete poured and smoothed, with anchor bolts in place. Bolts were inserted into plywood frame that had been marked with proper layout for bolting steel pier in place. RIGHT: 18" high, 6-inch I.D. steel water pipe pier welded to 12" diameter automobile engine flywheel. Holes were drilled in flywheel before welding to pipe.

SADDLE

The telescope tube assembly is usually attached to the mount by the saddle and can be a source of stability problems that are hard to find. Since this interfaces the tube to the mount it must be made to dampen out vibrations from both eternal and internal forces, and to provide a strong platform for the tube. In the examples above, the Newtonian telescope has long tube and requires a fairly long saddle. The saddle base plate in Figure 4 was too long, so it was replaced with a shorter 24″ x 14″ x 4″plywood structure, glued and bolted from end to end with 5/8th drill rod, steel end plates. The saddle cradles are made from a 24″x 20″x 0.375″ aluminum tube from a junk yard that was cut and shapes then sandwiched between ¾-in plywood layers and with an edge support by a 2″ wide by 1/8th inch aluminum strip. A 1″x1″ round Teflon spacer is screwed to the saddle cradles that holds the primary tube in place and allows it to rotate when necessary.

Although steel and aluminum are stiff, they tend to dampen out slowly as compared to wood or fiberglass. A saddle made mostly of wood provides excellent dampening and is durable when preserved with paint or other coatings. A saddle with a steel or aluminum base with wooden cradle is a very good combination for a saddle. Below is a simple but efficient saddle with circular cut ¾-inch laminated plywood ends that was glued then bolted in place using short carriage bolts. The two ends were then screwed to a piece of 2″ oak board. The restraint is made form a strip of 1/8th-inch x 2″ aluminum material and attached to the saddle board on one end and the other using a turn buckle with a hook on the end.

Figure 5. A simple saddle arrangement descried in the above.

The weak link in the chain here is the connection between the declination shaft and the saddle. Here is where you can use a large steel or aluminum block to provide a solid support for the saddle and surface for the shaft.

WEIGHING THE MOUNT

To find the weight of your steel shaft you first find the square area of the shaft and multiply that times the density of material. In the example, a steel shaft (**L**) 24 inches long and 1.5-inches in diameter (**r** = 0.75") will have a square area of and the density of steel is 0.283 lb/in^3:

$$We = \pi r^2 L = 3.14159 \times 0.75^2 \times 24$$
$$= 42.4 \text{ in}^3 \times 0.283 \text{ lb/in}^3 = 12 \text{ pounds} \quad \text{(Dec Axis)} \qquad \mathbf{(5)}$$
$$We = \pi r^2 L = 3.14159 \times 0.75^2 \times 16$$
$$= 28.3 \text{ in}^3 \times 0.283 \text{ lb/in}^3 = 8 \text{ pounds} \quad \text{(RA Axis)} \qquad \mathbf{(5)}$$

The density of steel is 0.283 lb/in ^{3}, so: 0.283 x 42.4 = 12 pounds

How about those lead counterweights? The density of lead is 0.411 lb/in ^{3} . Most counterweights are either cast iron or molded lead and are usually thin cylinders. Use equation (**5**) for predicting the weight of the counterweights.

For example, you need to balance the one hundred pound telescope and saddle, so you might place a hundred pound lead weight at the same distance at the opposite end of the declination shaft. You have a discarded two pound coffee can so that would be a good mold to pour the lead in. Before you begin remember to leave a hole in the weight for the shaft! A wood dowel with the same diameter as the shaft works fine. If the can is 6 inches in diameter and 9 inches tall, find the weight if the can if filled with lead:

$$Vcan = 3.14159 \times 3^2 \times 9 = 254.5 \text{ cu. in.} \times 0.411 \text{ lb/in}^3 = 104.6 \text{ lb}$$

$$Vhole = 3.14159 \times 0.75^2 \times 9 = 15.9 \text{ cu. in} \times 0.411 \text{ lb/in}^3 = 6.5 \text{ lb}$$

or, 104.6 - 6.5 = 98 pounds!

In practice the bearing and shaft housing for the declination axis is usually 12 to 14 inches long, so the counterweight has to be further down the shaft to clear the housing and have enough room for small adjustments. We do not want too far away from the center of axis, so, we could use two thirds of the counterweight (66 lb) at 18 inches down the declination shaft to produce the same inch/pounds.

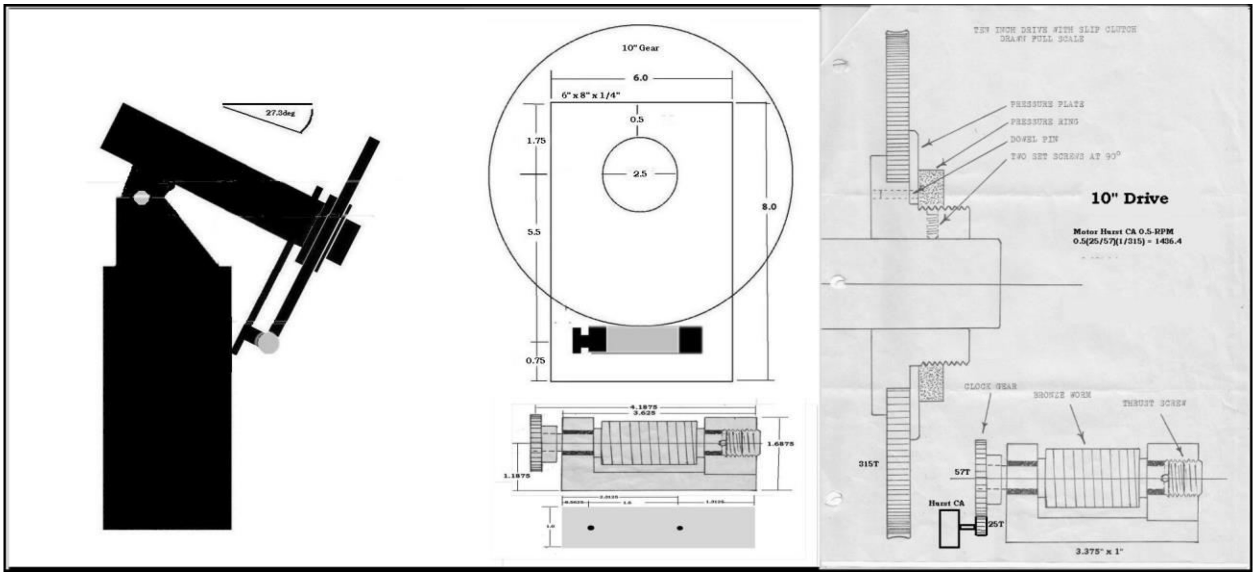
Finished Telescope Equatorial Mounts

Using a Parks German equatorial mount that was purchased back in the late 1970's each axis provides stability and weighs approximately 150 pounds. This mount was cast from aluminum and has 1.5-inch chromium steel shafts with two bearing in the polar assembly and 0.5"x 6" flange that separated the polar assembly from the declination assembly. I added a thin disk of Teflon (0.03" thick) to separate these surfaces to further stabilize the assembly. The declination shaft did not have bearings so a 3/8th-inch aluminum disk was machined to fit the shaft in the 0.5"x 6" flange at the top of the declination shaft and bolted to the aluminum saddle plate. Also, a thin disk of Teflon separated the flange and aluminum disk. Some sanding to the inside of the top and bottom housing allowed a thin sheet Teflon to be inserted around the shaft to act as a bearing. The Teflon surface provided roller bearing smoothness anyway.

Also, a 14-inch radius tangent arm declination drive was machined using scarp metal. The tangent arm slips into the adjustable aluminum/cork clutch assembly and attached to the declination shaft. A ½"/ 13 screw was machined from brass stock to fit inside a threaded aluminum block and associated hardware to drive the arm back and forth to adjust the declination axis. Using lapping compound the brass screw was attached to a hand drill and ran in and out several times to insure a close and smooth fit. To avoid backlash and to move the axis north or south a 3/8th-inch hardened bolt is tightly fitted into a slot milled in the tangent arm end. A DC motor and homemade reduction gear box for designed for a 7-arcsec/minute drive rate. **NOTE:** The mount with counterweights is shown **here**.

The polar dive is a Mathis 10" worm drive and fitted to the polar assembly using home made 3/8" thick plate and attachment hardware to the end of the polar axis. The polar drive turns the axis as a sidereal rate of 1/1436.4 revolutions per day (RPD) and an accompanying drive corrector it can also be set for solar, lunar or planetary rates. To achieve a sidereal rate a 315-tooth worm gear, bronze worm shaft combination is driven by a 57-tooth spur gear and 25-tooth spur gear on the shaft of a ½-RPM motor that produces gear ratios of: 0.5/(25/57 * 1/315).

Figure 6. LEFT: Parks German equatorial mount with UPPER RIGHT Thomas Mathis 10-inch worm gear drive using a 1/2-RPM Hurst motor [Transmission ratio: (25/57) (1/315) = 0.5 / 718.2 = 1 / 1436.4]. CENTER RIGHT: A 14-inch tangent arm declination drive with DC motor and a machined clutch assembly. The drive gear is a 6-inch, ½″O.D. brass gear and threaded aluminum block was machined for smooth and precise fit. BOTTOM RIGHT: Another view of equatorial head and Dec drive. (Note spider webs, other bug debris and corrosion from years of use).



LEFT: A silhouette of the mount pier, worm mounting plate and gear. CENTER: Layout plan for the worm gears and motor placement. RIGHT: Mathis 10" worm drive schematic.

In the image below is a heavy duty German equatorial mount was built by Brad Dischner of Riverside, California in November 1976. [See Sky & Telescope Magazine article "Gleanings for ATM's: A Portable 16-Inch of Great Steadiness," Vol. 57, No. 1, January 1979, p 87–91] and purchased from Virginia Capen by me in 1987. The wedge is made from 1/2-inch boilerplate steel with adjustment mechanisms included for up to 15 degrees in both polar and azimuth directions. This mount features 3-inch aluminum shafts with two Timken Roller Bearings per axis. The 125-pound equatorial head (RA and Dec axis) was attached to a wedge. The saddle cradles are made from a 24"x20"x0.5" aluminum tube from a junk yard that was cut and shapes then sandwiched between ¾-in plywood layers and with an edge support by a 2" wide by 1/8th inch aluminum strip. A 1"x1" round Teflon spacer is screwed to the saddle cradles that holds the primary tube in place and allows it to rotate when necessary. **NOTE:** The mount with counterweights is shown here.

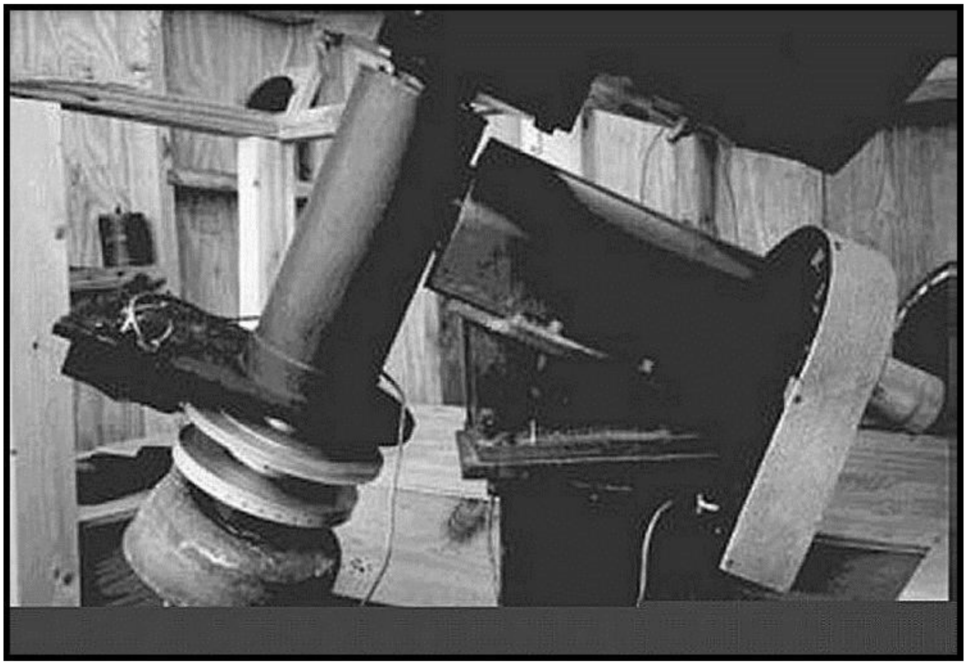


Figure 7. Photograph of author's German equatorial mount used for either a 12.5-inch f/7 or 16-inch f/7 telescope. Mount features 3.125-inch shaft diameters with two Timken Roller Bearings per axis. Saddle is made from 3/4-inch AC plywood lined with 1/8-inch aluminum strips and screwed into place.

The polar dive is a 12" worm drive and fitted to the polar assembly by aluminum plate and attachment hardware to the end of the polar axis. The polar drive turns the axis as a sidereal rate of 1 / 1,440 revolutions per day (RPD) and an accompanying driver corrector it can also be set for lunar or planetary rates. To achieve a sidereal rate a 320-tooth worm gear, bronze worm gear and shaft combination is driven by a 54-tooth spur gear and 24-tooth spur gear on the shaft of a Hurst "CA" ½-RPM synchronous motor, The gear ratios are: (24/54) * (1/320).

The declination drive uses a 10" aluminum worm dive, a reduction gear box and 1-RPM Hurst synchronous motor for a 7-arcsec/minute drive rate. In the late 1970's I designed and build a crystal controlled dive corrector that is still in use today. The drive housings are 5/8th-inch thick aluminum castings.

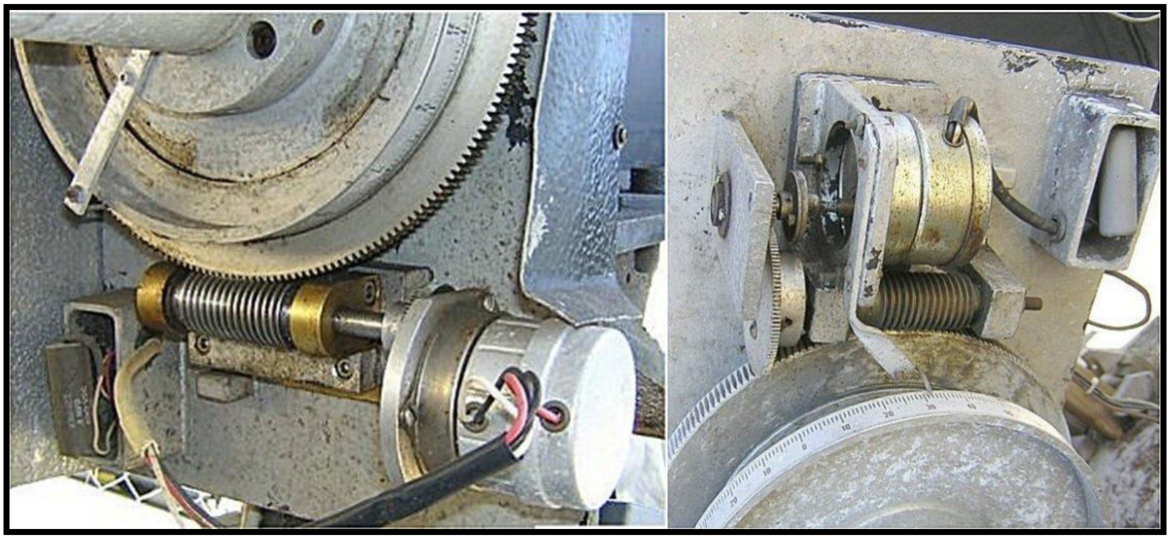


Figure 8. Close-up images of the right ascension and declination axis drive systems. LEFT: RA uses a 12″ worm gear combination and 1/2-RPM Hurst (5-watt) motor [Transmission ratio: (24/54) (1/320) = 0.5 / 720 = 1/1440]. The declination drive uses a 10″ aluminum worm dive combination with a 25 : 1 gear reduction and Hurst 1-RPM motor.

A simple hand paddle with either four switches or “joy stick” switch arrangement works fine for R.A. slow and high and Dec north or south. I have used several versions of clock drive correctors and now use an old Orion Accutrack 4404 corrector for my telescopes. The rates for each setting are as follows:

Drive Rate	RPD	Freq (Hz)	Period (msec)
Sidereal	1436.07	60.164	16.621
Solar:	1440.00	60.000	16.667
Lunar	1490.48	57.969	17.251

Figure 9. Accutrack 4404 drive corrector.

Some two axis drive correctors, including the 4402, outputs +/- 12VDC to drive the declination axis that has a DC motor. Some use a 120VAC motor in the declination axis, so Figure 10 shows a simple design to switch 120 VAC to the AC motor for directional control using two 12VDC relays and two diodes:

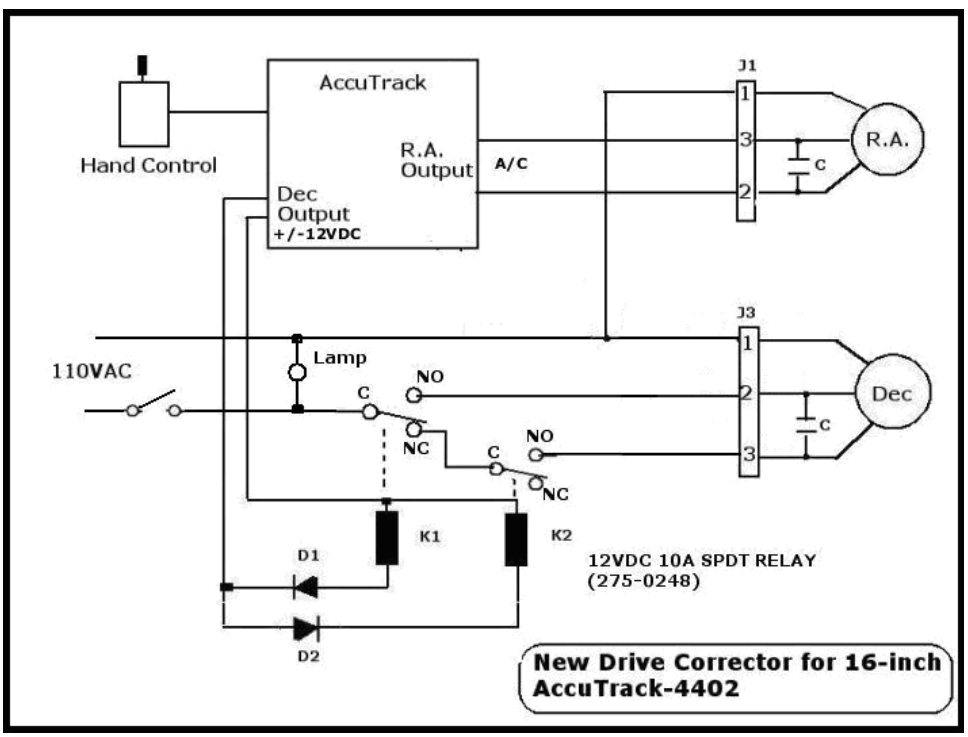


Figure 10. Declination drive with AC motor needs relays for directional control using 120VAC power.

Setup Telescope Mount Using Drift Alignment

One of many fine articles on how to polar align an equatorial mount can be found at Astro-Tom.com Polar Alignment and Sam Brown's book, **All About Telescopes**, 14th Edition (1999), "Adjustment to Pole," p126 – p129, Edmund Scientifics, so this will be brief. After centering a star close to the celestial equator in the finder scope cross hairs watch the direction the star drifts. Assuming the drive corrector is setup for the correct polar drive rate the star should not drift fast or slow as indicated in the east-west cross hair. Also, if the star drifts north or south then the mount requires adjusting either to the west or east or north or south, as described in the articles sited.

First, consider the orientation of the eyepiece view in the telescope. One should then orient the telescope so a star would drift from left to right in the eyepiece field and assume the 'north' is at the bottom of the field.

In a Newtonian Reflector or a Refractor without a star diagonal the inverted view is south at the top, north at the bottom. When telescope drive is off then the left side is the Following Drift (East) and the right side is the Preceding Drift (West). The opposite view is rendered in a non-inverted in a Schmidt-Cassegrain (SCT) or Refractor with a star diagonal. In this section the "inverted view" will be discussed.

Next, choose your star near where the celestial equator (i.e. at or about 0^o in declination) and the meridian meet. The star should be approximately 1/2 hour of right ascension from the meridian and within five degrees in declination of the celestial equator. Center the star in the field of your telescope and monitor the drift in declination.

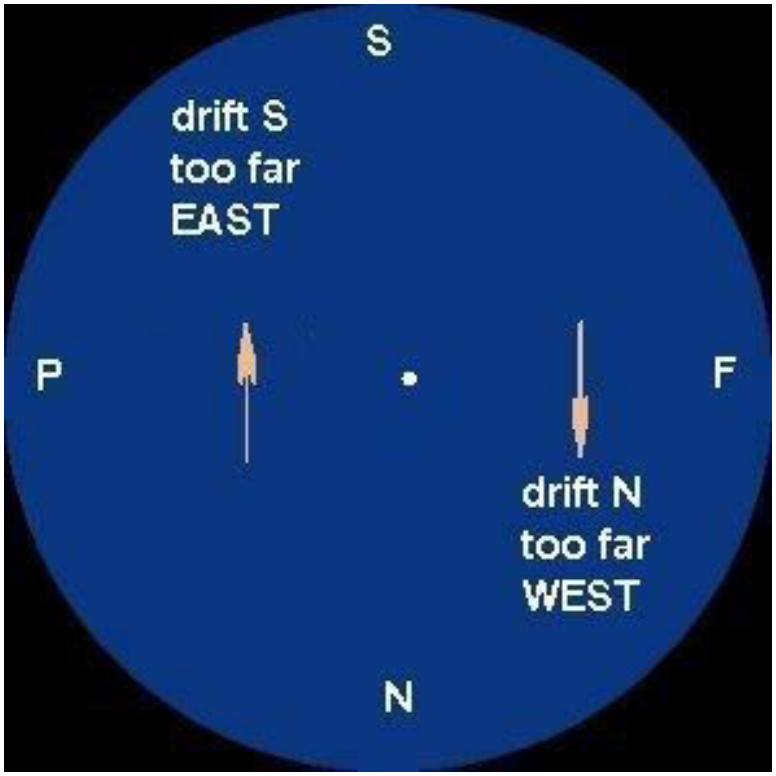


Figure 11. An inverted view of eyepiece field and drifting star. Orientation: S = south, N = north, P = preceding and F = following.

If star **drifts north**, the polar axis is **too far west;** Turn mount CW (eastward).
If star **drifts south**, the polar axis is **too far east;** Turn mount CCW (westward).

Using the telescope's azimuth adjustment knobs, make the appropriate adjustments to the polar axis to eliminate any drift. Once you have eliminated all the drift, move to the star near the eastern horizon. The star should be 20 degrees above the horizon and within five degrees of the celestial equator.

If the star **drifts north**, the polar axis is **too high**. Lower polar angle.
If the star **drifts south**, the polar axis is **too low**. Raise polar angle.

SUMMARY

1) The shafts must be of adequate cross-section to support the scope and guiding equipment without sagging or bending too much.

2) The mount should be "short coupled." That is, the distances from bearing supports to the payload should be as short as possible.

3) The RA and declination housings must be joined at an accurate right angle. Adjustment in preload is desirable.

4) The RA shaft should be in preloaded roller bearings to prevent any play.

5) The point at which the RA housing meets the pedestal should be thick and rigid.

6) The RA clock drive should be 70% to 100% the diameter of the mirror. Any less will normally result in an erratic drive, and any more will not substantially improve your system. Your money is better spent on other parts of the mount.

7) The declination shaft should have preload adjustments on it to eliminate play.

8) The declination system should be electrically driven to avoid laying your hands on the mount to make corrections. Try a drive resolution of at least twice that of the RA drive.

This author found a very interesting Internet web page that detail very nicely the engineering aspects of designing a telescope and I highly recommend Mechanical Design of Telescopes for the Amateur, by Bob Lombardi, at: http://www.tfn.net/~blombard/book/MechanicalDesign.htm

References

Albrecht, Richard E., "The Design of Telescope Structures - I," ***Sky and Telescope Magazine*** , Vol. 77, No. 1, pp. 97-101, January 1989

Albrecht, Richard E., "The Design of Telescope Structures - II," ***Sky and Telescope Magazine*** , Vol. 77, No. 2, pp. 210-214, February 1989

Bely, Pierre Y. [Editor], (2005), **The Design and Construction of Large Optical Telescopes**, Springer-Verlag New York, Inc., Chapter 7.6.1 "Effects of wind: Generalities," p290

Berry, Richard, "How to Control Friction in a Dobsonian Telescope," ***R.T.M.C. Proceedings*** , 1980.

Brooks, John J., "Mechanical Considerations of Telescope Makers," ***Sky and Telescope Magazine*** , Vol. 51, No. 6, pp. 423-428, June 1976.

Engineering ToolBox (http://www.engineeringtoolbox.com/)

Parker, D.C., "CCD Image of Jupiter's Moon Ganymede," ***Astronomy Magazine*** , Vol. 23, No. 7, p. 106, July 1995.

Tretta, Fred, "An Analysis of Telescope Mounts," ***R.T.M.C. Proceedings*** , 1980.