CHARA TECHNICAL REPORT

NO. 20 4 MAR 1995

Miscellaneous Telescope Drive Considerations

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1. ESTIMATE OF WIND FORCES ACTING ON THE TELESCOPES

The enclosure style and provisions for ventilation are unknown at this time. CHARA specs indicate that operation in a 30 mph (44.1 ft/sec or 48 kph) wind is possible.

The telescope "tube" is presently configured with the altitude axis about midway between the primary and secondary.

- Assume wind blows only on structure above the altitude axis and is uniform.
- Assume the simplified structure shown in Figure 1.

$$
\begin{aligned}
F_{\text{wind}} &= 1/2 C_D \cdot \sigma_{\text{air}} \cdot V^2 \cdot A \\
C_D &= \text{drag coefficient} \equiv 1.0 \text{ for this estimate} \\
\sigma_{\text{air}} &= \text{average density} \equiv 0.063\,\text{lb/ft}^3 \ \text{ at } 6500\,\text{ft} \\
V &= \text{wind velocity} \equiv 44.1\,\text{ft/sec}
\end{aligned}
$$

$$
\begin{aligned}
F_{\text{wind}} &= \frac{0.5 \cdot 1.0 \cdot 0.063\,\text{lb/ft}^3 \ \cdot (44.1\,\text{ft/sec}\,)^2 \cdot \text{A}}{32.2\,\text{ft/sec}^2 \ \cdot 144\,\text{in}^2/\text{ft}^2} \\
&= 0.0132 \cdot A
\end{aligned}
$$

for A in square inches and F_{wind} in lb/in^2.

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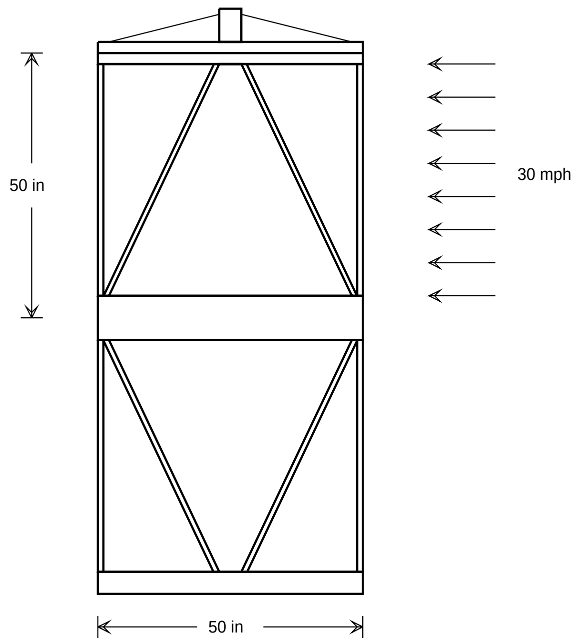

FIGURE 1. Schematic of telescope tube assembly.

Item	Areas Exposed to Wind	Wind Cross Section (in^2)	Area c.g. Distance from Alt. Axis (in)	Wind Force (lb)	Wind Torque (in lb)
Top frames (2)	2 in × 50 in (two sides on both frames)	400	50	5.3	265
Struts	2 in × 25 in × 4 struts	200	50	2.7	133
Sec. housing	5 in × 20 in	100	50	1.3	65
Trusses	2 in × 50 in × 2 × 4 trusses	800	25	10.6	265
Metering rods	2 in × 50 in × 4	400	25	5.3	133
Total					861 in lb (∼ 72 ft lb)

2. CRUDE ESTIMATE OF TUBE ASSEMBLY WEIGHT

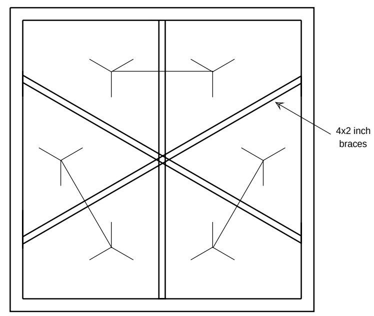

FIGURE 2. Primary mirror support.

Two frames made of $4\,\text{in} \times 2\,\text{in} \times 1/4\,\text{in}$ tube ~ 4 ft square			
	$\Rightarrow 32\,\text{ft} \cdot 8.8\,\text{lb/ft}$	282 lb	
+	secondary assembly	40	
+	struts and braces	60	
+	counterweight (to offset primary assembly and tertiary pole)	~418	
Sum			800 lb
Trusses & Metering Rods: assume 2 in dia $\times$ 1/4 in, $\sim$ 4 ft long (4.7 lb/ft)			
	$\Rightarrow 12 \cdot 4\,\text{ft} \cdot 4.7\,\text{lb/ft}$	226	
$\times$ 2	(upper and lower)		
Sum			512
Altitude Journal (this should be lightweighted)			
	$\pi/4 \cdot (48\,\text{in})^2 \cdot 2 \cdot 0.283$		1025
Mid-section Frame: made of 8 in standard channel (11.5 lb/ft)			
	$\Rightarrow 16\,\text{ft} \cdot 2 \cdot 11.5\,\text{lb/ft}$	368	
+	$2\,\text{in} \times 2\,\text{in} \times 1/4\,\text{in}$ damper tubes (5.4 lb/ft)		
	$\Rightarrow 4 \cdot 2 \cdot 4\,\text{ft} \cdot 5.4\,\text{lb/ft}$	173	
+	1 in diameter center rods		
	$\Rightarrow \pi/4 \cdot (1\,\text{in})^2 \cdot 48\,\text{in} \cdot 0.283 \cdot 8$	85	
+	epoxy	~24	
Sum			650
Tertiary Pole: $2.5\,\text{in} \times 1/4\,\text{in} \times 5\,\text{ft}$ long			
	$\Rightarrow \pi/4 \cdot (2.5^2 - 2^2) \cdot 60 \cdot 0.283$	30	
+	tertiary assembly	25	
+	struts and brackets	20	
Sum			75
Primary Mirror		310	
+	$4\,\text{in} \times 4\,\text{in} \times 1/4\,\text{in}$ frame (12.0 lb/ft) = $16\,\text{ft} \cdot 12\,\text{lb/ft}$	192	
+	$4\,\text{in} \times 2\,\text{in} \times 1/4\,\text{in}$ braces 5 ft long @ 8.8 lb/ft (3 req'd)	132	
+	3 whiffletrees @ 30 lb	90	
Sum			724
TOTAL WEIGHT $\sim 724 + 75 + 650 + 1025 + 512 + 800$		3,786 lb	
+	20%	758	
GRAND TOTAL			4,544 lb

3. ALTITUDE AXIS DRIVE TORQUE ESTIMATE

To obtain the acceleration torque requirement, an estimate of the "tube" moment of inertia will be made. The weights estimated earlier will be treated as concentrated masses at the average radius. The exception will be the Alt drive journal, which will be treated as a solid disc.

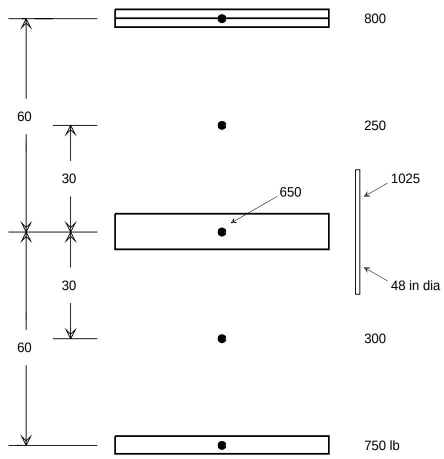

FIGURE 3. Weight distribution assumed for torque estimate.

$$\begin{aligned} I_{k-x} &= \sum W k^2 \qquad (\text{k} = \text{radius of gyration}) \\ &= 800 \cdot (60)^2 + 250 \cdot (30)^2 + 1025 \cdot \frac{(24)^2}{2} + 300 \cdot (30)^2 + 750 \cdot (60)^2 \\ &= (2.88 + 0.225 + 0.295 + 0.270 + 2.70) \times 10^6 \,\text{lb in}^2 \\ &\approx 6.37 \times 10^6 \,\text{lb in}^2 \end{aligned}$$

It is unlikely that accelerations will ever exceed 1 deg/s^2.

$$\begin{aligned} T_{\text{acc}} = I\alpha &= \frac{6.37 \times 10^6 \,\text{lb in}^2}{386 \,\text{in/sec}^2} \left(1 \,\text{deg/sec}^2\right) \left(\frac{2\pi \,\text{rad}}{360 \,\text{deg}}\right) \\ &= 288 \,\text{in lb} \\ &= 24 \,\text{ft lb} \end{aligned}$$

This is a conservative number.

$$\begin{aligned} \text{Total torque} &= T_{\text{wind}} + T_{\text{acc}} \\ &= (72 + 24) \,\text{ft lb} \; = 96 \,\text{ft lb} \end{aligned}$$

4. AZIMUTH YOKE WEIGHT ESTIMATE

Yoke arms: 12 in dia × 0.5 in × 8.5 ft

$$\begin{aligned} \text{Weight} &\approx \pi \cdot 12 \cdot 0.5 \cdot 100 \cdot 0.283 \approx 533\,\text{lb} \\ I &\approx 533 \cdot (42\,\text{in})^2 \times 2 \text{ tines} \approx 1.9 \times 10^6\,\text{lb in}^2 \end{aligned}$$

Yoke base: 30 in × 0.5 in × 8 ft

$$\begin{aligned} \text{Weight} &\approx \pi \cdot 30 \cdot 0.5 \cdot 96 \cdot 0.283 \approx 1280\,\text{lb} \\ I &\approx W\left(\frac{L^2}{12} + \frac{D^2}{16}\right) \\ &\approx 1280\left(\frac{96^2}{12} + \frac{30^2}{16}\right) \approx 1.1 \times 10^6\,\text{lb in}^2 \end{aligned}$$

Base tube: 36 in × 0.5 in × 8 ft

$$\begin{aligned} \text{Weight} &\approx \pi \cdot 36 \cdot 0.5 \cdot 96 \cdot 0.283 \approx 1540\,\text{lb} \\ I &\approx Wr^2 \\ &\approx 1540\,\text{lb} \cdot (18\,\text{in})^2 \approx 0.5 \times 10^6\,\text{lb in}^2 \end{aligned}$$

Base:

$$\begin{aligned} \text{Weight} &\approx \pi/4 \cdot (72^2 - 36^2) \cdot 2 \cdot 0.283 \approx 864\,\text{lb} \\ I &\approx W\left(\frac{r_o^2}{2} - \frac{r_1^2}{2}\right) \approx 1.7 \times 10^6\,\text{lb in}^2 \end{aligned}$$

Totals are as follows:

$$\begin{aligned} \text{Total I value} &= (1.9 + 1.1 + 0.5 + 1.7) \times 10^6 \\ &= 5.2 \times 10^6\,\text{lb in}^2 \\ \text{Total yoke weight} &= 533 + 1280 + 1540 + 864 = 4217\,\text{lb} \\ &+ \text{ 20\% for optical bench, \#4, \#5, \#6, etc.)} \\ &= 5060\,\text{lb} \end{aligned}$$

The total rotating weight on the lower Az bearing is

$$\begin{aligned} \text{Total weight} &= \text{yoke weight} + \text{tube weight} \\ &= 5060 + 4544\,\text{lb} \\ &= 9604\,\text{lb} \qquad \text{(assume 10,000 lb)} \end{aligned}$$

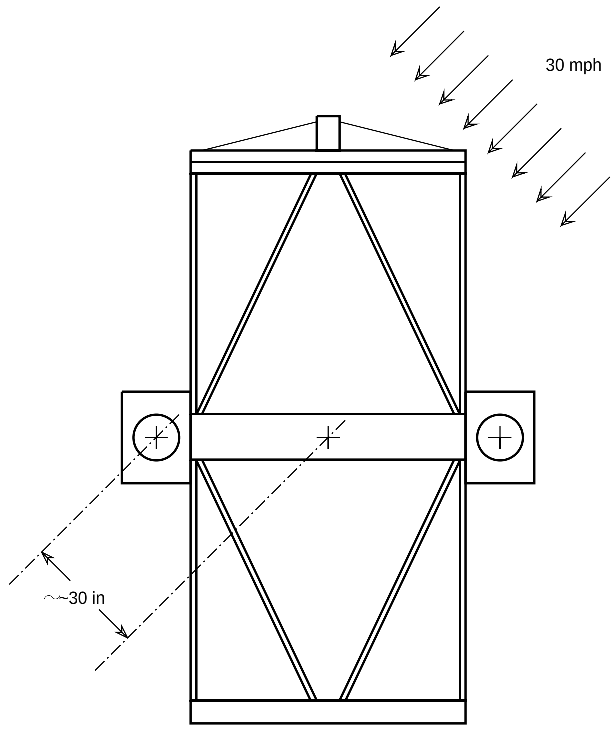

FIGURE 4. Wind torque on the azimuth drive.

5. AZIMUTH AXIS TORQUE REQUIREMENT

It is more difficult to estimate wind torque on the azimuth drive because assumptions have to be made about the way wind enters the enclosure. The worst case might be with the tube pointed to the horizon with the wind able to act more or less at 45° to the optical axis. Also, the wind might act on one yoke arm and not the other. A reasonable "worst case" is to use the maximum wind torque for the Alt axis case and add the effect of wind blowing on one yoke arm.

Note: The greater cross-section "seen" by a 45° wind is offset by the reduction in moment arm about the Az axis compared to the Alt axis case.

For the yoke arm, 8′ long × 12″ diameter,

$$\begin{aligned} A &= 12 \cdot 96 = 1152\,\text{in}^2 \\ F_{\text{yoke}} &= 0.0132 \cdot A = 15.2\,\text{lb} \\ T_{\text{yoke}} &\approx 15.2 \cdot 30\,\text{in lb} \\ &= 496\,\text{in lb} \;\approx 38\,\text{ft lb} \end{aligned}$$

Note: The lower yoke cross-arm and "handle" can be shielded.

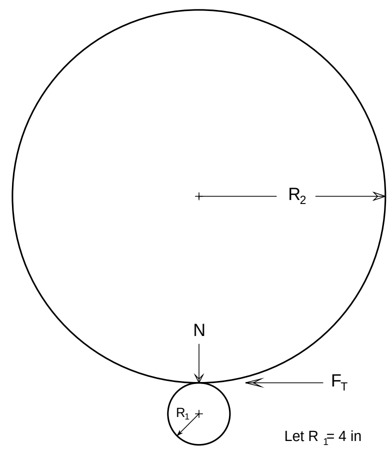

FIGURE 5. Friction on roller interface.

Total potential wind torque on the azimuth drive may be:

$$\begin{aligned} T_{\rm Az}(\text{wind}) &= 72\,\text{ft lb (from Alt axis)} + 38\,\text{ft lb (from yoke arm)} \\ &= 110\,\text{ft lb} \end{aligned}$$

The Az axis acceleration torque will be maximized when the tube is horizon-pointing. Then the torque will be about equal to the maximum torque needed for the Alt axis plus that needed to accelerate the yoke. Use $\alpha = 1\,\text{deg/sec}^2$.

$$\begin{aligned} \text{Max Az torque} &\approx 110\,\text{ft lb} + 24\,\text{ft lb} + \text{I}_{\rm Az}\cdot\alpha \\ &\approx 134 + \left(\frac{5.2\times10^6\,\text{lb in}^2}{386\,\text{in/sec}^2}\right)\cdot\left(1\,\text{deg/sec}^2\right)\cdot\left(\frac{2\pi\,\text{rad}}{360\,\text{deg}}\right)\cdot\left(\frac{1\,\text{ft}}{12\,\text{in}}\right) \\ &\approx 134 + 19.6\,\text{ft lb} \\ &\approx 154\,\text{ft lb} \end{aligned}$$

6. CONTACT STRESSES AT DRIVE ROLLERS

The coefficient of friction (μ) at the roller interface sets the amount of compressive load, N, needed to transmit the torque force F_T without slippage. A reasonable value is $\mu = 0.1$. Then the governing equation for contact stress at the interface is:

$$\begin{aligned} S &= 0.591\left[\left(\frac{N}{L}\right)\left(\frac{E_1\cdot E_2}{E_1+E_2}\right)\left(\frac{D_1+D_2}{D_1\cdot D_2}\right)\right]^{1/2} \\ L &= \text{length of the contact line between rollers} \equiv 0.5\,\text{in} \end{aligned}$$

$$
\begin{aligned}
E_1, E_2 &= \text{Young's modulus for the roller materials} \\
&= 15 \times 10^6\,\text{lb/in}^2 \quad \text{for bronze material to be specified for the drive roller} \\
&= 30 \times 10^6\,\text{lb/in}^2 \quad \text{for the driven steel journal}
\end{aligned}
$$

$$
N = \frac{\text{Required Torque}}{\mu \cdot R_2}
$$

$$
\begin{aligned}
R_2 &\approx 24'' \text{ for Alt drive} \\
&\approx 36'' \text{ for Az drive}
\end{aligned}
$$

$$
\begin{aligned}
N_{\text{Alt}} &= \frac{96\,\text{ft lb}}{(0.1)\cdot(24/12)\,\text{ft}} = 480\,\text{lb} \quad \text{for Alt axis} \\
N_{\text{Az}} &= \frac{154\,\text{ft lb}}{(0.1)\cdot(36/12)\,\text{ft}} = 513\,\text{lb} \quad \text{for Az axis}
\end{aligned}
$$

$$
\begin{aligned}
S_{\text{Alt}} &= 0.591 \left[\frac{480\,\text{lb}}{0.5\,\text{in}} \left(\frac{15\cdot 30}{15+30}\right)\left(10^6\,\text{lb in}^2\right)\left(\frac{8+48}{8\cdot 48}\right)\left(\frac{1}{\text{in}}\right)\right]^{\frac{1}{2}} \\
&= 21,715\,\text{lb/in}^2 \quad \text{for Alt axis} \\
S_{\text{Az}} &= 0.591 \left[\frac{513\,\text{lb}}{0.5\,\text{in}} \left(\frac{15\cdot 30}{15+30}\right)\left(10^6\,\text{lb in}^2\right)\left(\frac{8+72}{8\cdot 72}\right)\left(\frac{1}{\text{in}}\right)\right]^{\frac{1}{2}} \\
&= 22,310\,\text{lb/in}^2 \quad \text{for Az axis}
\end{aligned}
$$

The contact stress just calculated is a compressive "Hertz" stress and at 22,000 lb/in^2 is modest. There are no absolute guidelines for choosing a "safe" stress[2]; however, 1.5 × Tensile Yield Stress would be considered a conservative value.

For manganese bronzes (a/k/a "High-Strength Yellow Brass"[3]:

UNS# (ASTM-SAE)	Tensile Yield Strength (lb/in^2)	Hardness (HB)
86100	48,000	180
86200	48,000	180
86300	67,000	225
86400	25,000	105
86500	28,000	130

Any of these alloys would satisfy the present requirement; however, the hardness should be kept below that of the driven steel journal.

Virtually all grades of structural steel have tensile yield strength above 24,000 lb/in^2 and thus would be strong enough. However, the hardness properties vary, e.g.

[2]see, for example, Chapter 13 in Roark & Young, "Formulas for Stress and Strain", 5th Edition

[3]see "Metals Handbook", 9th ed., ASM vol. 1, 2, & 4)

AIST Grad	Minimum Yield Strength (lb/in^2)	Hardness (HB) as rolled	annealed	normalized
1015	41,000	126	111	121
1020	43,000	131	111	143
1030	50,000	179	126	149

Only by heat treating or going to an alloy steel will it be possible to make the journal significantly harder than a manganese bronze roller. Considering other bronzes, I find:

UNS#	Tensile Yield Strength (lb/in^2) (sand castings)	Hardness (HB) (sand castings)
C90700 (Phosphor Gem Bronze)	22,000	80
C91700 (Nickel Gem Bronze)	22,000	85
C92200 (Navy “M” Bronze)	20,000	65

C90700 and C91700 bronzes are often used in worm gears and bearings under heavy loads at low speed. It would appear that either would be a good choice for the drive roller. This would:

- Enable usage of low-carbon rolled steel plate for the driven journal. Any welding required would not significantly change the hardness properties. No complicated heat treating would be necessary.
- Require sand casting of drive roller blanks. Since we will need at least ten blanks, this is a reasonable procedure. (The pattern will be simple)
- Ensure that the steel journal will be harder than the drive roller, and thus easier to replace in case of unexpected damage.

CHOOSE: AISI 1015 Steel Plate – Structural Quality C90700 Bronze – Sand Cast Blanks

With these materials, only one drive roller is required for either the Azimuth or Altitude axis (assuming an 8 in diameter roller). However, two rollers would be advantageous on the Azimuth journal to balance the normal forces required.

The same would be true on the Alt axis if it were practical to install two opposed rollers.

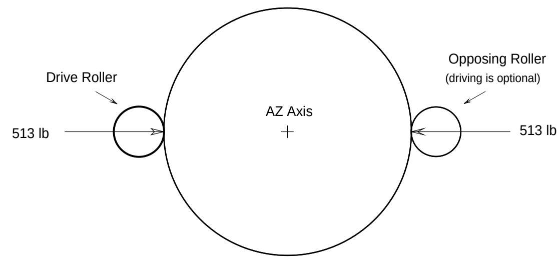

FIGURE 6. Azimuth axis with drive and opposing rollers.

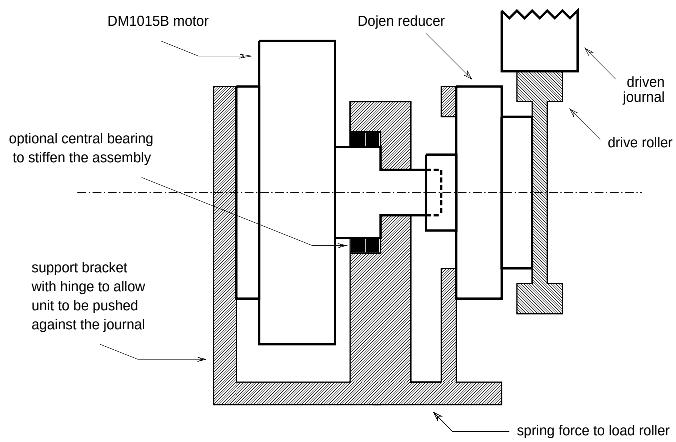

FIGURE 7. Schematic of drive unit configuration.

7. DRIVE UNIT CONSIDERATIONS

"Tracking jitter" shall be $<0\farcs02$. This requires a very small drive step or a substantial speed reduction between the drive motor and the driven journal.

Proposed (after looking at various options):

- **Motor:** – Compumotor DC brushless servo motor with an integral optical encoder. Model DM-1015B (see data sheets). This motor will have $\sim 2''$/step resolution and can produce $\sim$11 ft lb of torque.

- **Speed Reducer** – Dojen 04, 100:1 reducer

The general configuration is shown in Figure 7.

8. MICROSLIP BETWEEN ROLLERS VS. ENCODING

Poritzky[4] analyzed slippage occurring between cylindrical rollers that are transmitting torque. The applicable equation is:

$$\Delta e = a \cdot \mu \cdot \left(\frac{1}{R_1} + \frac{1}{R_2}\right) \cdot \left(1 - \sqrt{1 - \frac{F_T}{\mu \cdot N}}\right)$$

where e is the elongation in the tangential direction produced on the rollers by the torque force, and Δe is the difference between roller elongations (i.e. the slip).

$$a^2 = \frac{8N}{\pi}\left(\frac{1-\sigma_1^2}{E_1} + \frac{1-\sigma_2^2}{E_2}\right) \Big/ \left(\frac{1}{R_1} + \frac{1}{R_2}\right)$$

N = normal compression force between rollers per unit axial length
σ_1, σ_2 = Poisson's ratio for the roller materials
E_1, E_2 = Young's modulus for the roller materials
R_1, R_2 = roller radii ($\text{Roller}_1 \equiv$ small roller; $\text{Roller}_2 \equiv$ large roller)
$2a$ = width of contact region
F_T = tangential force between rollers (i.e., the torque force)

Since e represents an elongation, the creep between rollers will be:

$$1 + \Delta e = \frac{(2\pi R_1) \cdot (\text{Number of rotations of Roller}_1)}{\text{Distance traveled on Roller}_2}$$

In the present case, roller creep would cause the driven journal to rotate through a smaller angle than it would if microslip did not occur.

$$1 + \Delta e = \frac{\text{Theoretical angular journal travel}}{\text{Actual angular journal travel}} = \frac{\theta_T}{\theta_A}$$

Therefore,

$$\Delta e \cdot \theta_A = \theta_T - \theta_A = \text{Angular Position Error}$$

For tracking purposes in the CHARA case, the tangential force F_T will be small, but not zero. Bearing friction will be present. If one assumes a total rotating weight = 10,000 lb acting on a 36 in diameter bearing with a coefficient of friction = 0.003,

$$\begin{aligned} T &= 0.003 \cdot 10,000\,\text{lb} \cdot \frac{36\,\text{in}}{2} \\ &= 540\,\text{in lb} \end{aligned}$$

[4] H. Portsky, "Stresses of Deflections of Cylindrical Bodies in Contact with Application to Contact of Gears and Locomotive Wheels," Journal of Applied Mechanics, June, 1950

Allowing for a small amount of friction from the smaller bearing, an estimate of 600 in lb is reasonable. This torque must be exerted at the journal rim which will be about 6 ft (72 in) in diameter. The torque force then will be:

$$F = \frac{600\,\text{in lb}}{36\,\text{in}} = 16.7\,\text{lb} \qquad (\text{use } 17\text{ lb})$$

These figures apply to the azimuth axis. The altitude axis case will be less severe because the bearings are much smaller and the weight is less. The journal size will be ~4 ft diameter but not small enough to offset the other factors.

If F is applied over 0.5 in of contact length, the tangential force/unit length = 34 lb/in = F_T. For CHARA,

$$\begin{aligned}
E_1 &= 15 \times 10^6\,\text{lb/in}^2 \text{ (bronze)} \\
E_2 &= 30 \times 10^6\,\text{lb/in}^2 \text{ (steel)} \\
\sigma_1 = \sigma_2 &= 0.3 \\
\mu &\equiv 0.1 \\
R_1 &\equiv 4\,\text{in} \\
R_2 &\equiv 36\,\text{in} \\
N &\approx 1000\,\text{lb/in} \quad \text{(from other estimates for maximum torque required)}
\end{aligned}$$

$$\begin{aligned}
a^2 &= \frac{8 \cdot 1000\,\text{lb/in}}{\pi}\left(\frac{1-0.3^2}{15\times10^6\,\text{lb/in}^2} + \frac{1-0.3^2}{30\times10^6\,\text{lb/in}^2}\right) \Big/ \left(\frac{1}{4\,\text{in}} + \frac{1}{36\,\text{in}}\right) \\
&= 834 \times 10^{-6}\,\text{in}^2 \\
a &= 28.9 \times 10^{-3}\,\text{in} \ \approx 0.030\,\text{in}
\end{aligned}$$

$$\begin{aligned}
\Delta e &= 0.03 \cdot 0.1 \cdot \left(\frac{1}{4} + \frac{1}{36}\right) \cdot \left(1 - \sqrt{1 - \frac{34\,\text{lb/in}}{0.1 \cdot 100\,\text{lb/in}}}\right) \\
&= 0.000156
\end{aligned}$$

Thus, for each 1° of Azimuth tracking, the error accumulated is:

$$\begin{aligned}
\epsilon &= 1\,\text{deg} \cdot \left(\frac{3600''}{1\,\text{deg}}\right) \cdot 0.000156 \\
&= 0''.563
\end{aligned}$$

This is a "worst-case" scenario, but clearly shows that absolute position encoding from the driving roller is impractical. The situation is greatly improved if the encoder is driven by an independent roller that transmits only enough torque to rotate the encoder (e.g., the method used to drive the incremental encoders on the 4-m Mayall Telescope). However, the need for positive fiducials still exists for absolute position encoding.