An Evaluation of the 64-Meter Antenna Radial Bearing for Use on the 70-Meter Antenna

H. D. McGinness
Ground Antennas and Facilities Engineering Section

An analysis of the 64-m antenna radial bearing shows that it will be satisfactory for the increased loads brought about by increasing the antenna dish to a diameter of 70-m.

I. Introduction

During the autumn and winter of 1983 and 1984 a new radial bearing runner assembly was installed at DSS-14. The runner was made in accordance with JPL drawing 9474446G. The problems associated with the old runner and their causes have been reported in Ref. 1. Figure 1 shows a cross section of the installed runner, its alignment bolts, anchor nuts, and grout. Before the old runner was removed it was not known whether the anchor nuts embedded in the concrete would be reusable for the new runner; that is, it was uncertain whether the old alignment bolts could be removed from the anchor nuts. Fortunately all of the bolts were removed without damaging the anchor nuts, thereby eliminating the need for the laborious task of chipping out concrete and embedding new anchor nuts.

Reference 1 describes the reference ring which was used for monitoring the changes in the old runner which had become distorted. This same reference ring was used for adjusting the new runner for roundness. Figure 2 shows the method of aligning the runner to the reference ring.

In late 1983 the radial bearing wheel assemblies were removed and thoroughly inspected. It was decided to regrind the rolling surfaces true to the bearing journals, and this was accomplished to concentricity and cylindricity accuracies of approximately 0.025 mm (0.001 in.). For the convenience of subsequent alignment, the upper outer edges of all wheels were made perpendicular to the rolling surfaces to within 0.025 mm (0.001 in.). This permits the wheel alignment to be made or checked simply by placing a precision level on the upper outer edge of the wheel. Figure 3 shows the plan view of the radial bearing and the three truck assemblies which are attached to the alidade base structure. The individual wheel bearing housings can be shimmed to adjust dimensions b and t as shown in Fig. 3. Since the bearings are self-aligning spherical rollers, such shimming can achieve near perfect alignment of the wheels. Figure 4 shows the wheel and bearing assembly. Even when the wheels are preloaded against the runner, a precision level can be placed on the upper edge of a wheel, aligned either in a radial or tangential direction. Since the runner surface forms a vertical cylinder, such level measurements detect any slight misalignment of the wheel, and corrective shim thickness can be calculated promptly.

The critical loads on the wheels are determined by the wind direction and magnitude, the preload in the azimuth drive links, imperfections in the runner (bumps or indentations), and the initial preload applied to the truck assemblies.

II. Truck Load From Non-Circular Track

The wheel truck load caused by an imperfectly round runner depends upon the stiffnesses of the several components constituting the elastic system. These components are the alidade base triangle, the girder connecting the truck assembly to the corner weldment of the base triangle, the pedestal supporting the runner, the runner, the wheels, the wheel bearings, the truck frame, the stem, and the pivot pin. Most of these components are identified in Fig. 5. The details of the truck assembly can be obtained from JPL drawing 9437995.

An attempt was made to calculate the above mentioned stiffness components. The corresponding deflections are listed in Table 1. Since there are three trucks involved, the deflections per truck must be multiplied by 3 to obtain the amount to be combined with the single alidade base triangle. The calculated stiffness, k_c, is the reciprocal of the total deflection per unit load; hence the calculated stiffness is

$$\frac{10^{10}}{17.91} = 0.0558 \times 10^{10} \text{ N/m } (3.19 \times 10^6 \text{ lb/in.})$$

An experiment was made wherein the deflection δ was measured for a series of hydraulic jack forces, applied as indicated in Fig. 5. The resulting measured stiffness, k_m, was 0.0606×10^{10} N/m (3.46×10^6 lb/in.). This value of the stiffness will be used in estimating the effect of an out-of-round runner on the wheel load.

The amount that the runner is out of round is determined by making a series of radial measurements when there is no preload in the trucks. From a pivot point inside the instrument tower, measurements can be made to the inside of the reference ring at every 15°. Then measurements are made from the reference ring to the runner, as shown in Fig. 2, and the sums are the desired radial measurements which can be plotted as shown in Fig. 6.

The variation in truck preload can be established as follows:

(1) Determine at what antenna azimuth angle the wheels are to be preloaded and locate the six wheel positions on Fig. 6(a). Denote the ordinates of the wheels of truck No. 1 as Y_{1B} and Y_{1A} in accordance with the wheel identification per Fig. 3. Denote the ordinates of the other wheels as Y_{2F}, Y_{2E}, Y_{3D}, Y_{3C}.

(2) Form the sum of the ordinates of all six wheels at the preload position and denote it as Y_P.

(3) Let the wheels traverse a distance of 120°, as on Fig. 6(a) and denote the sum of all six wheel ordinates as Y_θ.

(4) Plot Y_θ versus azimuth angle as per Fig. 6(b).

(5) Determine Y_θ maximum and Y_θ minimum.

(6) Denote the initial truck preload by P_i.

(7) The maximum increase from P_i is $+\Delta P_i$,

$$+\Delta P_i = (Y_{\theta \text{MAX}} - Y_P)\, k_m \tag{1}$$

(8) The maximum decrease from P_i is $-\Delta P_i$

$$-\Delta P_i = (Y_P - Y_{\theta \text{MIN}})\, k_m \tag{2}$$

(9) The total excursion of the preload, ΔP, is

$$\Delta P = (Y_{\theta \text{MAX}} - Y_{\theta \text{MIN}})\, k_m \tag{3}$$

III. Truck Load Induced by Azimuth Drive

The azimuth axis wind torque is resisted by four azimuth drive units, which effectively are four links connecting the corners of the alidade base triangle to tangent points on the azimuth bullgear. In Fig. 7 these four links are represented by the dashed vectors labeled P_D, where P_D is the preloaded value of the link load and exists at this value only when the azimuth torque is zero. When the wind torque is finite, the link loads are of different magnitudes such that the torque produced equilibrates the applied wind torque. Since the maximum and minimum truck loads occur when the wind blows directly into the dish at a 5° elevation angle, a condition for which the azimuth torque is zero, the effect of the link loads on the trucks will be analyzed only for the preloads, P_D.

The link preloads, P_D, are exactly statically equivalent to their resultant components shown in Fig. 7, namely the force $0.816\ P_D$ applied at joint 3, and the forces $0.995\ P_D$ and $0.103\ P_D$ applied at joints 1 and 2. The elastic structure composed of the base triangle and the truck assemblies is statically indeterminate under this loading; hence, the truck loads will involve a consideration of the stiffnesses of the members. Denote the truck load at joint number 3 as βP_D and denote the load in truck number 1 as F_1. The load in each of the triangle legs meeting at joint 3 is $[(0.816 - \beta)/(2 \cos 30°)]\ P_D$.

Consider the horizontal equilibrium of joint 1 and obtain the following equation:

$$\left(\frac{0.816 - \beta}{2 \cos 30°}\right) P_D \cos 30° + F_1 \cos 60° = 0.995\ P_D \tag{4}$$

Solve Eq. (4) for F_1 and obtain:

$$F_1 = \frac{P_D \left[0.995 - \left(\dfrac{0.816 - \beta}{2}\right)\right]}{\cos 60°} = P_D\,(1.174 + \beta) \tag{5}$$

Consider the vertical equilibrium of joint 1 and obtain:

$$F_V + P_D (1.174 + \beta) \cos 30^\circ$$

$$+ P_D \frac{(0.816 - \beta)}{2 \cos 30^\circ} \cos 60^\circ + 0.103 P_D = 0 \qquad (6)$$

Solve Eq. (6) for F_v and obtain:

$$F_v = - P_D \left[0.103 + 1.174 (0.866) + 0.866 \beta + 0.236 - 0.289 \beta \right]$$

$$= - P_D \left[1.355 + 0.577 \beta \right] \qquad (7)$$

The loads in terms of β are now known for all the members of the elastic system and are marked in Fig. 7.

The strain energy, V, of a prismatic member loaded axially is:

$$V = \frac{1}{2} \left(\frac{F^2 L}{EA} \right) \qquad (8)$$

where F is the axial load; E is the elastic modulus; and L/A is the ratio of member length to member cross sectional area. For the base triangle legs, $L/A = 797/272 = 2.93$ 1/in. (115.35 1/m). Based upon the measured k_m of 3.46×10^6 lb/in. (0.0606×10^{10} N/m), the effective L/A for the truck assemblies is 2.05 1/in. (80.7 1/m). Using these L/A values together with the member loads marked on Fig. 7, sum the strain energy in the entire elastic system and obtain:

$$V = \frac{P_D^2}{2E} \Big\{ (1.355 + 0.577 \beta)^2 \, 2.93 + 2 (0.471 - 0.577 \beta)^2 \, 2.93$$

$$+ \beta^2 (2.05) + 2 (1.174 + \beta)^2 \, 2.05 \Big\} \qquad (9)$$

Simplifying there is obtained:

$$V = \frac{P_D^2}{2E} \left[12.329 + 11.0232 \beta + 9.076 \beta^2 \right] \qquad (10)$$

By the principle of minimum strain energy, the derivative of V with respect to the parameter β must be zero; hence,

$$\frac{\partial V}{\partial \beta} = \frac{P^2}{2E} \left[11.0232 + 18.152 \beta \right] = 0 \qquad (11)$$

$$\beta = -0.6187 \qquad (12)$$

For this value of β the member loads are marked in Fig. 8.

The value of the link preload, P_D, is proportional to the hydraulic motor supply pressure, p, and can be adjusted over a large range. The value of P_D is:

$$P_D = \frac{DR\eta \mathrm{p}}{2\pi \mathrm{r}} \qquad (13)$$

where

D = the displacement of the hydraulic motor per turn

$R > 1$ = the speed ratio of the gear box drive

η = the gear box efficiency

r = the pitch radius of the output pinion

Substituting the appropriate values into Eq. (13) there is obtained:

$$P_D = \frac{2.41 (610) (0.90) \mathrm{p}}{2\pi \, 8.912} = 23.63 (\mathrm{p_{psi}}) \text{ lb} \qquad (14)$$

If the current value of 550 psi for P is used, the value of P_D is:

$$P_D = 23.63 (550) = 12996 \text{ lb } (57804 \text{ N}) \qquad (15)$$

This value of P_D produces 8045 lb (35782 N) of tension in truck 3 and 7213 lb (32082 N) of compression in trucks 1 and 2.

The preceeding analysis is based on the assumption that the link preloads are tension loads. It is possible to reverse the hydraulic motor connections and obtain compressive link preloads, in which case the senses of the truck loads would reverse.

IV. Truck Loads From Wind

The maximum and minimum truck loads occur when the wind blows directly into the dish at an elevation angle of 5°. Figure 9 shows the symmetrical loading on the alidade base triangle in terms of the unknown parameter α. It is assumed that the truck compressive preload is sufficient to enable truck number 3 to carry the tension load induced by the wind.

The unknown truck forces are denoted as Q for truck number 3 and as P for trucks 1 and 2. Static equilibrium requires that:

$$W = Q + 2P \sin 30^\circ = Q + P \qquad (16)$$

In Fig. 9 the forces in the six members of the elastic structure are shown in terms of α, W, and Q.

The strain energy, V, of a prismatic member axially loaded is:

$$V = \frac{1}{2}\left(\frac{F^2 L}{EA}\right)$$

where

F = the axial load

E = the elastic modulus

L/A = the ratio of length to cross sectional area

The total strain energy in the elastic structure is:

$$V = \frac{1}{2E}\left\{\frac{Q^2 L_s}{A_s} + 2(W-Q)^2 \frac{L_s}{A_s} + \frac{2(Q-\alpha W)^2}{3}\frac{L}{A} + \left[(3-\alpha)\frac{W}{2} - Q\right]^2 \frac{1}{3}\left(\frac{L}{A}\right)\right\} \tag{17}$$

where L_s/A_s pertains to the equivalent truck member and L/A pertains to the triangle base members.

By the principle of minimum strain energy, the derivative of V with respect to Q must be zero; hence,

$$\frac{\partial V}{\partial Q} = \frac{1}{2E}\left\{2\frac{L_s}{A_s}Q + (4Q - 4W)\frac{L_s}{A_s} + \left(\frac{4}{3}Q - \frac{4}{3}\alpha W\right)\frac{L}{A} + \left[\frac{2}{3}Q + \left(\frac{\alpha}{3} - 1\right)W\right]\frac{L}{A}\right\} = 0 \tag{18}$$

Simplifying Eq. (18) there is obtained:

$$\frac{Q}{W} = \frac{4\frac{L_s}{A_s} + (1+\alpha)\frac{L}{A}}{6\frac{L_s}{A_s} + 2\frac{L}{A}} \tag{19}$$

For the particular case of $\alpha = 1/3$, when the wind load is applied equally to the three corners of the base triangle, Eq. (19) becomes:

$$\frac{Q}{W} = \frac{4\frac{L_s}{A_s} + \frac{4}{3}\frac{L}{A}}{6\frac{L_s}{A_s} + \frac{6}{3}\frac{L}{A}} = \frac{4}{6}\left(\frac{\frac{L_s}{A_s} + \frac{1}{3}\frac{L}{A}}{\frac{L_s}{A_s} + \frac{1}{3}\frac{L}{A}}\right) = \frac{2}{3} \tag{20}$$

which shows that for the particular $\alpha = 1/3$, Q/w is independent of the relative stiffnesses of the members.

Estimates for L/A and L_s/A_s are respectively 2.93 and 2.05 in.$^{-1}$

Substituting these into Eq. (19), there is obtained:

$$\frac{Q}{W} = \frac{4(2.05) + 2.93 + 2.93\alpha}{6(2.05) + 2(2.93)} = 0.613 + 0.161\alpha \tag{21}$$

Even if α is substantially different from 1/3, the value of Q/W is not strongly affected; therefore the value of 1/3 for α shall be assumed, thus producing:

$$Q = (2/3)W \tag{22}$$

$$P = (1/3)W \tag{23}$$

V. Truck Preload

The minimum truck preload must be enough to insure that truck No. 3 is in compression at a wind speed of 70 mph (31.3 m/s) when the antenna is at the critical attitude of zero wind azimuth and 5° elevation angle. Denote the truck loads as F_1, F_2, and F_3. Each of these will be the algebraic sum of the loadings caused by preload, drive link load, wind, and out of roundness of the track. The values for these components are set out in Table 2. The values of F_3 and F_1 for the 64-m and 70-m antennas in terms of the truck preload P and the track out of roundness factors are:

$$F_{3_{64}} = 8044 + 285866 + (Y_{\theta MAX} - Y_{\theta MIN})\, 3.46 \times 10^6 - P \tag{24}$$

$$F_{3_{72}} = 8044 + 386667 + (Y_{\theta MAX} - Y_{\theta MIN})\, 3.46 \times 10^6 - P \tag{25}$$

$$F_{1_{64}} = -7212 - 142933 + (Y_{\theta MAX} - Y_{\theta MIN})\, 3.46 \times 10^6 - P \tag{26}$$

$$F_{1_{72}} = -7212 - 193333 + (Y_{\theta MAX} - Y_{\theta MIN})\, 3.46 \times 10^6 - P \tag{27}$$

Equations (24), (25), (26), and (27) apply for the critical condition of a 70-mph (31.3-m/s) wind at zero wind azimuth and an elevation angle of 5°. For these conditions it is necessary that F_3 be greater than zero. From Eqs. (24) and (25) the required values of the truck preloads P are:

$$P_{64} > 8044 + 258866 + (Y_{\theta MAX} - Y_{\theta MIN})\, 3.46 \times 10^6 \text{ lb} \tag{28}$$

$$P_{72} > 8044 + 386667 + (Y_{\theta MAX} - Y_{\theta MIN})\, 3.46 \times 10^6 \text{ lb} \quad (29)$$

where the factor $(Y_{\theta MAX} - Y_{\theta MIN})$ must be measured in accordance with the instructions previously given.

The substitution of Eqs. (28) and (29) into Eqs. (26) and (27), respectively, yields the following minimum values for F_1 at the critical configuration:

$$F_{1_{64}} = -7212 - 142933 - 8044 - 285866$$

$$= -444055 \text{ lb } (-1975255 \text{ N}) \quad (30)$$

$$F_{1_{72}} = -7212 - 193333 - 8044 - 386667$$

$$= -595256 \text{ lb } (-2647830 \text{ N}) \quad (31)$$

The value of the preload, P, should exceed the minimum value by some amount, ΔP, to be determined. Then the value of F_1 would be increased by ΔP.

VI. Truck and Wheel Alignment

In the past it has been difficult to align the trucks and wheels so that each has a lateral runout hysteresis loop within the desired amount. Past practice has been to jack the trucks into the correct vertical position, align the wheels to be parallel to the runner, and then apply the preload. Oftentimes one truck would have a satisfactory hysteresis loop, whereas others would not. It might be that vertical equilibrium of forces on the wheel required a vertical friction force; that is, at the time the preload was applied, the temporary forces from the vertical alignment jacks were not zero. A cylindrical roller, in general, does not have one definite rolling radius over the entire width of the roller when the elastic roller is pressed against an elastic track. The result is that over most of the roller width there is a finite amount of slippage as the rolling motion progresses. Small geometrical imperfections in the roller or track can cause the narrow no-slip regions to change. The effect can be that the vertical frictional resistance reduces drastically, thus allowing the wheel to seek a lower equilibrium position.

It is recommended that the truck stems be aligned so that when the wheels are barely clearing the track, and not supported by vertical jacks, the wheels are centered on the track wear strip and are parallel to it. This means that the structure supporting the truck stems be shimmed or modified so that the natural cantilevered position of the wheels is at the desired elevation. When such a position is obtained, the clearances between the track and wheel edges can be measured. This is best done by measuring the level of the wheel upper edge in the radial direction. The wheel can then be shimmed at the spherical bearing housings until uniformity of clearance has been obtained. Then the wheel level can be measured in the track tangential direction and the bearing housings shimmed appropriately. Note that there is a small interaction between the radial and tangential alignments (the shimming surfaces are not quite perpendicular to radial and tangential lines), and this should be taken into account when striving for best alignment. When shimming for radial alignment, if dimension t of Fig. 3 is increased by a unit amount, dimension b must be decreased by 0.135 times the unit amount. When shimming for tangential alignment, if dimension b is increased by a unit amount, dimension t must be increased by 0.135 times the unit amount.

The hysteresis loop tests have been run over a ±120° azimuth range. The usual result is that the lateral runout is more or less linear for a certain distance and then remains approximately constant for the rest of the distance. Upon direction reversal, the linear lateral motion commences and is followed by a constant displacement, the sum of which approximately equals the first half of the cycle.

It is believed that the linear portion of the lateral runout obtains until the induced side force caused by tangential misalignment is equilibrated by the elastic restoring force of the truck considered as a cantilevered beam. If this is true, the amount of lateral runout is proportional to the truck load since the induced side force is proportional to the truck load. Therefore, it is desirable to keep the truck preload no higher than is required by previous considerations. Furthermore, the contact stresses increase with the square root of the truck load. The minimum value of the truck preload, P, is obtained for the condition of a perfectly circular track for which the $(Y_{\theta MAX} - Y_{\theta MIN})$ factor of Eqs. (28) and (29) would be zero. For the case of the 70-m antenna Eq. (29) may be written as follows:

$$P_{72} = 394711 + \Delta P + (Y_{\theta MAX} - Y_{\theta MIN})\, 3.46 \times 10^6 \text{ lb} \quad (32)$$

where ΔP is the increment above zero at the critical condition. If ΔP is arbitrarily set at 5% of the first right hand term of Eq. (32), then P_{72} becomes:

$$P_{72} = 1.05\,(394711) + (Y_{\theta MAX} - Y_{\theta MIN})\, 3.46 \times 10^6 \text{ lb} \quad (33)$$

If the second right hand term of Eq. (33) is set at, say, 15% of the first right hand term, then the value of $(Y_{\theta MAX} - Y_{\theta MIN})$ becomes:

$$(Y_{\theta MAX} - Y_{\theta MIN}) = \frac{1.05\,(394711)\,(0.15)}{3.46 \times 10^6}$$

$$= 0.018 \text{ in. } (0.456 \text{ mm}) \qquad (34)$$

This apparently is a practical track alignment goal to strive for when setting the circularity of the runner, since the initial measurements made at DSS-14 on the new track gave approximately this result. It seems remarkable that such a large diameter bearing can be set to this accuracy.

If the value of Eq. (34) is substituted into Eq. (33) there is obtained:

$$P_{72} = 1.05\,(394711) + 0.018\,(3.46 \times 10^6)$$

$$= 476727 \text{ lb } (2120585 \text{ N}) \qquad (35)$$

$$P_{64} = 1.05\,(293910) + 0.018\,(3.46 \times 10^6)$$

$$= 370885 \text{ lb } (1649781 \text{ N}) \qquad (36)$$

Taking $\Delta P_{64} = 0.05\ (293910) = 14695$ lb and $\Delta P_{72} = 0.05\ (394711) = 19736$ lb and adding these respectively to Eqs. (30) and (31) there are obtained:

$$F_{1_{64}} = 444055 + 14695 = 458750 \text{ lb } (2040622 \text{ N}) \qquad (37)$$

$$F_{1_{72}} = 595256 + 19736 = 614992 \text{ lb } (2735621 \text{ N}) \qquad (38)$$

These can be taken as tentative values, but they should be revised if final track roundness values differ from the initial ones used here.

VII. Contact Stresses in Wheels and Wear Strip

For the case of contact between a uniformly loaded cylindrical wheel and a cylindrical track of ten times the wheel diameter, the Hertz stress, σ_H, and the maximum shear stress, τ, are, from Ref. 2:

$$\sigma_H = 0.605\sqrt{\frac{FE}{DL}} \qquad (39)$$

$$\tau = 0.183\sqrt{\frac{FE}{DL}} \qquad (40)$$

where

F = the normal force between cylinder and plane

E = the common modulus of elasticity of the materials

D = the roller diameter

L = the roller width

For the case of a triangularly distributed load across the width of the wheel, the load intensity per unit length at the heavily loaded edge would be twice as much as for the uniformly loaded case. Since the stresses of Eqs. (39) and (40) vary as the square root of the load per unit length, the Hertz stress, σ_H, and maximum shear stress, τ, for this loading will be $\sqrt{2}$ times those for the uniformly distributed case, thus,

$$\sigma_{HT} = 0.855\sqrt{\frac{FE}{DL}} \qquad (41)$$

$$\tau_T = 0.259\sqrt{\frac{FE}{DL}} \qquad (42)$$

In Table 3 are listed the maximum shear stresses corresponding to truck preloads and truck maximum loads for both the 64-m and 70-m antennas. It is assumed that the contact wearstrip has a minimum tensile yield strength of 68950 N/cm^2 (100000 psi) which corresponds to a shear yield strength of 39991 N/cm^2 (58000 psi). From Table 3 it may be seen that all shear stresses are well below this shear yield value. The wheels are case hardened and have considerably more strength than the wear strips.

VIII. Wheel Bearings Loads

As Fig. 4 shows, each wheel is supported by two spherical roller bearings which are spaced 0.546 m (21.5 in.) apart. When the wheel is uniformly loaded across its width, the bearing radial loads are the same and each bearing radial load is equal to one-fourth the truck load since there are two wheels on each truck. For the case of triangularly distributed loading across the wheel width, the radial load on the more highly loaded bearing is two-thirds the wheel load or one-third the truck load. As Fig. 4 shows, only the upper wheel bearing can resist axial load. With the tangential wheel alignment required to achieve the desired hysteresis loop value, it is believed that the axial bearing load will not exceed 10% of the bearing radial load. From Ref. 3 the equivalent bearing load, P_E is:

$$P_E = F_r + 2.2\,F_a \qquad (43)$$

where F_r is the radial load and F_a is the axial load. For $F_a = 0.10\,F_r$, Eq. (43) becomes:

$$P_E = F_r + 2.2\,(0.10)\,F_r = 1.22\,F_r \tag{44}$$

In Table 4 there are set out the equivalent bearing loads for the various loading conditions mentioned above. In all cases it is assumed that the axial load is 10% of the radial load. From Ref. 3 the bearing static and dynamic load ratings are respectively 2068320 N (465000 lb) and 1845920 N (415000 lb). These are substantially more than any of the entries of Table 4.

IX. Conclusion

This report has discussed the derivation of the proper truck preload and has shown that it is dependent upon the azimuth drive preload direction, the wind direction and magnitude, and the out of roundness of the track. The resulting wheel and track contact stresses have been evaluated and found to be satisfactory for either the 64-m or 70-m antenna. The wheel bearing loads have been found to be quite satisfactory for the larger 70-m antenna.

Methods for evaluating the effects of track out of roundness and for aligning the wheel and truck assemblies have been presented. It is believed that these methods are practical and that the increased loadings on the parts will not affect the expected long life of the azimuth bearing.

References

1. McGinness, H., and G. Gale, Rehabilitation of 64-m antenna radial bearing, *TDA Progress Report 42-65*, July 1981, pp. 151-161, Jet Propulsion Laboratory, Pasadena, Calif.
2. Timoshenko, S., *Theory of Elasticity*, McGraw Hill, 1934, First Edition, p. 349.
3. SKF Industries, Catalogue Reg. No. 47122, 1981, p. 30.

Table 1. Stiffness components and corresponding deflections

Component	Deflection per Truck, 10^{-10} m/N	Deflection per Set of Three Trucks, 10^{-10} m/N
Pedestal	0.354	1.062
Girder to Corner Weldment	1.598	4.796
Truck Pivot Pin	0.513	1.542
Truck Stem	0.194	0.582
Truck Frame	0.268	0.805
Wheel Contact	0.600	1.799
Wheel Bearings	0.519	1.557
Alidade Triangle Base		5.767
Total Deflection		17.910

Table 2. Truck load components

Load Source	F_3		F_1	
	N	lb	N	lb
Drive link load from Fig. 8 for p = 550 psi	35,781 TENSION	8,044	32,080 COMP	7,212
Wind Load from Eqs. (22), (23) and Fig. 9 for W_{64} = 428800 lb[a]	1,271,595	285,866	635,798	142,933
W_{72} = 580000 lb[a]	1,719,980 TENSION	386,667	859,988 COMP	193,333
Track out of roundness from Eq. (3) $\Delta P = (Y_{\theta MAX} - Y_{\theta MIN})\ 3.46 \times 10^6$	$(Y_{\theta MAX} - Y_{\theta MIN})\ 3.46 \times 10^6$ TENSION			
Preload	P COMP	P	P COMP	P

[a]From *The effects of wind loading on the bearings and drives of the 64-m and 72-m antennas* (unpublished), by H. D. McGinness, 1984, Table VI, Reorder No. 84-2, Jet Propulsion Laboratory, Pasadena, CA.

Table 3. Wheel and track contact stresses

	64-m Antenna		70-m Antenna	
Truck preload	1,649,781 N	(370,885 lb)	2,120,585 N	(476,727 lb)
Maximum shear stress from wheel preload uniform distribution (Eq. [40])	$14{,}704 \times 10^4$ N/m^2	(21,326 psi)	$16{,}671 \times 10^4$ N/m^2	(24,178 psi)
Truck maximum load	2,040,622 N	(458,750 lb)	2,735,621 N	(614,992 lb)
Maximum shear stress from wheel maximum load uniform distribution (Eq. [40])	$16{,}354 \times 10^4$ N/m^2	(23,718 psi)	$18{,}936 \times 10^4$ N/m^2	(27,461 psi)
Maximum shear stress from wheel maximum load triangular distribution (Eq. [42])	$23{,}145 \times 10^4$ N/m^2	(33,568 psi)	$26{,}798 \times 10^4$ N/m^2	(38,866 psi)

Table 4. Equivalent wheel bearing loads for axial load = 10% radial load

Loading Condition (See Table 3 for truck load values)	64-m Antenna		70-m Antenna	
	N	lb	N	lb
Preload uniformly distributed across wheel	503,157	113,120	646,747	145,402
Preload triangularly distributed across wheel	670,877	150,827	862,329	193,869
Maximum truck load uniformly distributed across wheel width	622,359	139,919	834,323	187,573
Maximum truck load triangularly distributed across wheel width	829,811	186,558	1,112,430	250,097

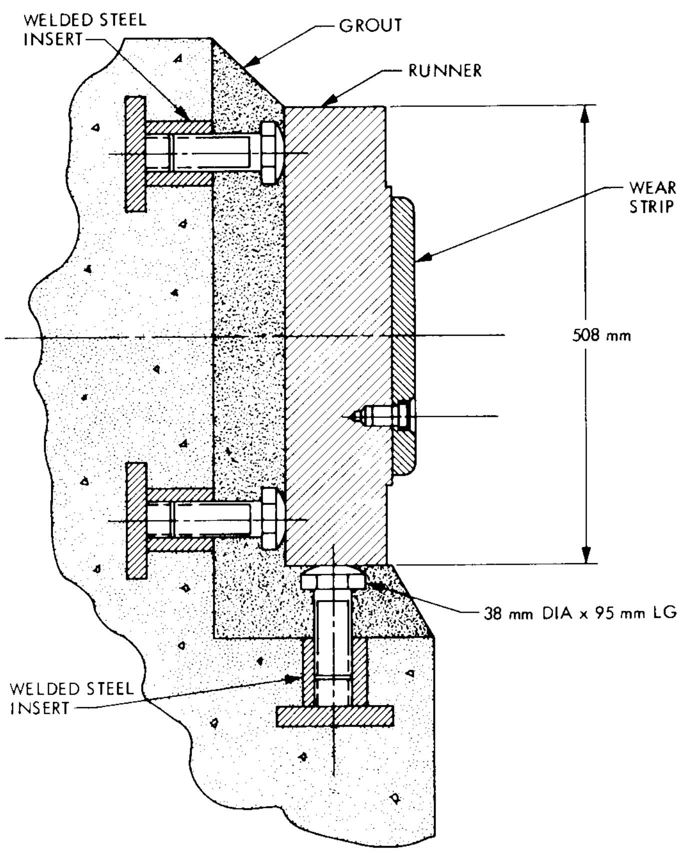

Fig. 1. Cross section of runner showing alignment bolts and grout

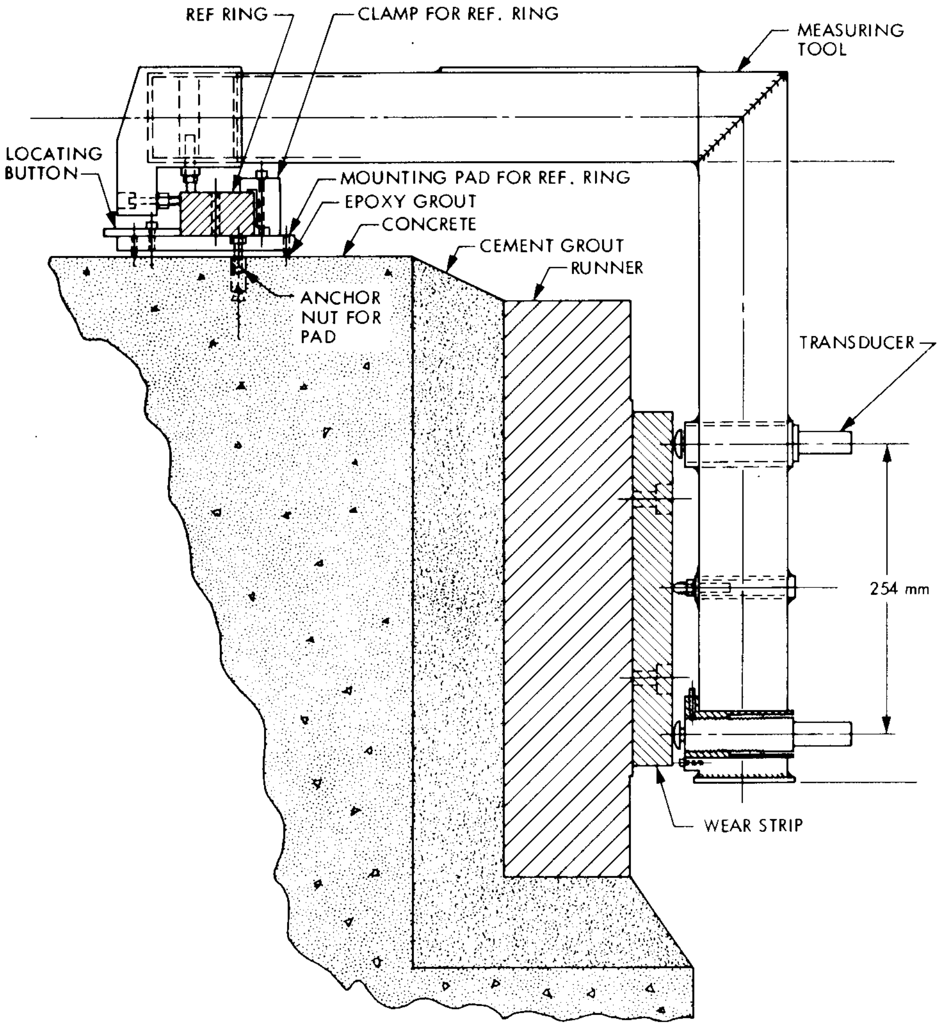

Fig. 2. Method of aligning runner to reference ring

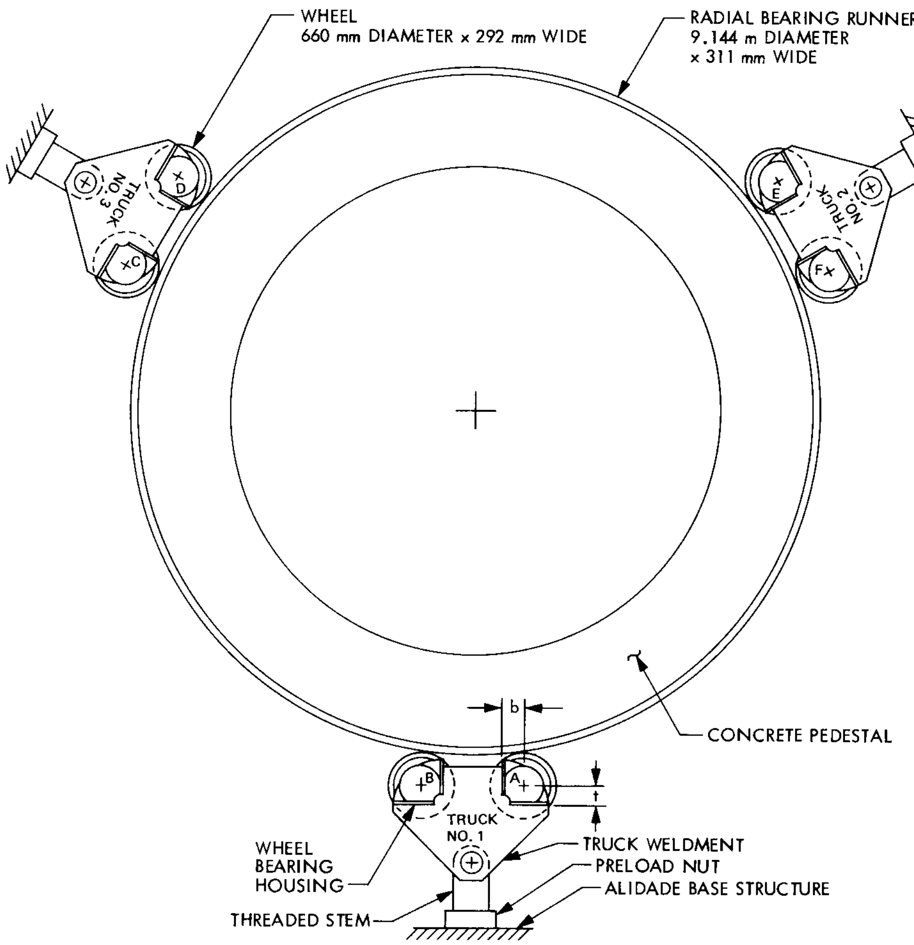

Fig. 3. Plan view of radial bearing runner and trucks

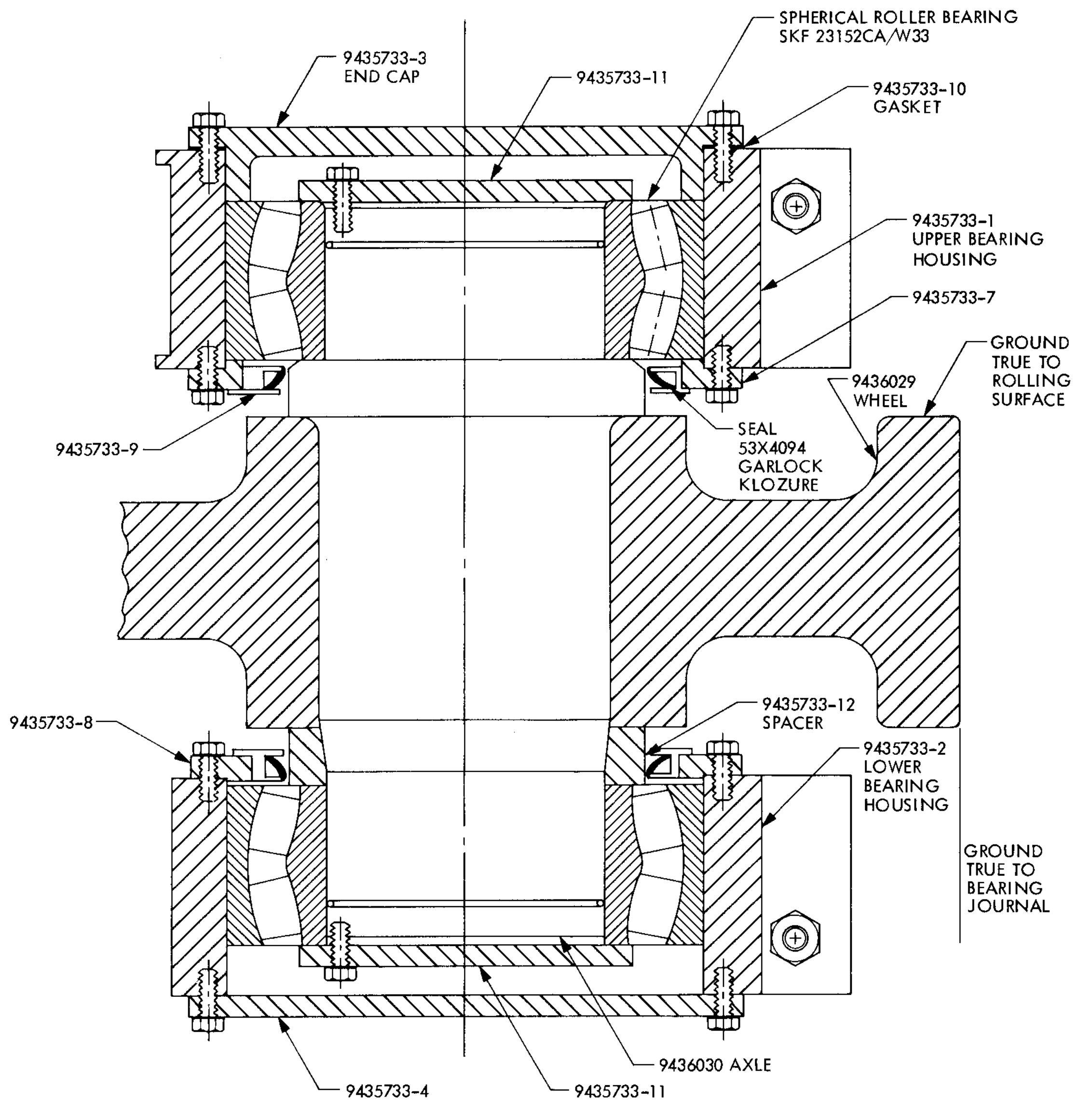

Fig. 4. Wheel and bearing assembly

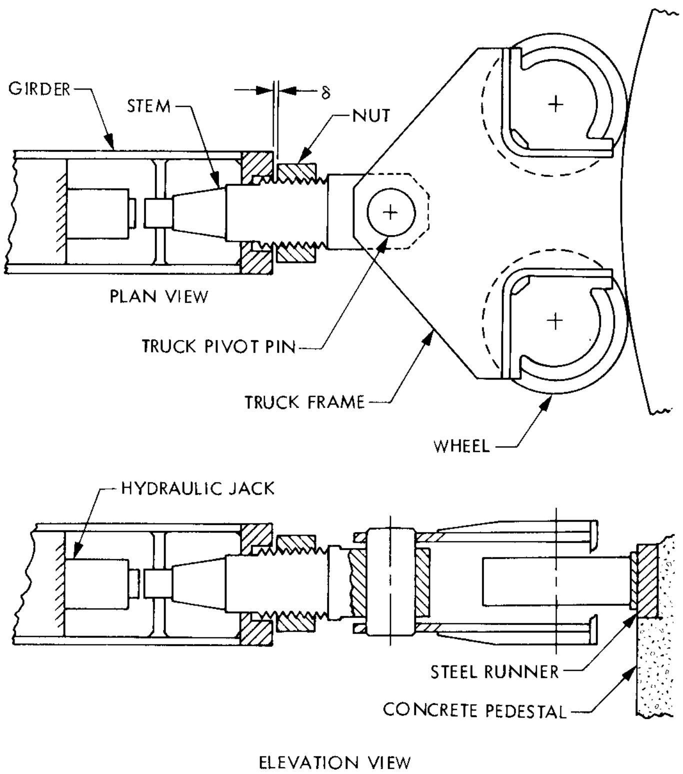

Fig. 5. Truck assembly, method of preloading

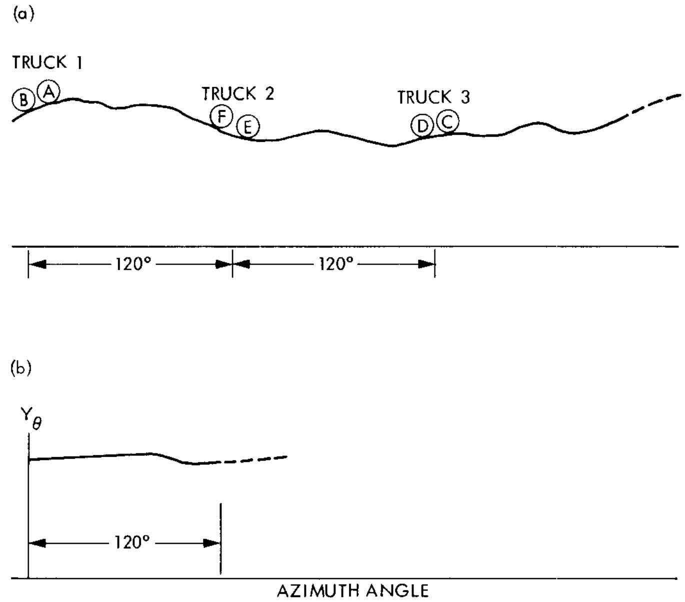

Fig. 6. Method of measuring runner roundness

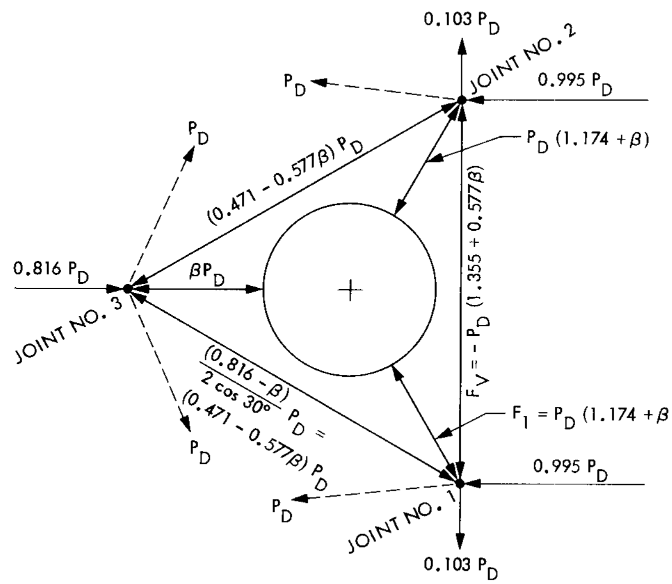

Fig. 7. Effect of azimuth drive preload

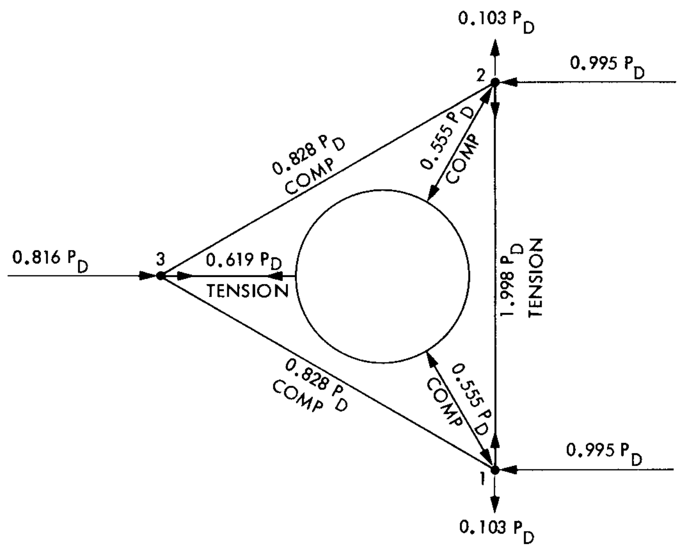

Fig. 8. Truck loads caused by azimuth drive preloads

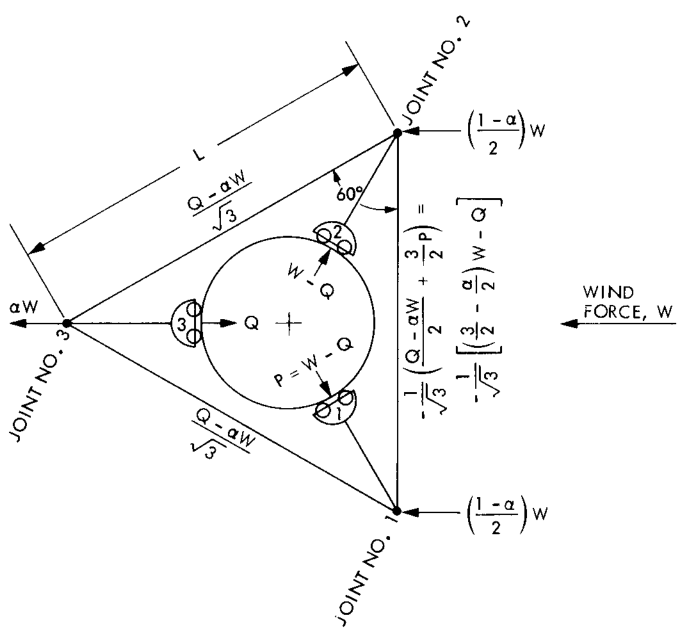

Fig. 9. Truck loads from wind