Antenna Azimuth Bearing Model Experiment

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A reduced scale model of an antenna wheel and track azimuth bearing indicates that its prototype would have a long life and require little maintenance.

I. Introduction

During the past few years various wheel and track configurations have been studied in order to appraise their suitability for use as azimuth bearings of large antennas. A particular design comprising a single circular track and a novel wheel suspension system appeared to have unique advantages, not only for the antenna application, but for many applications involving a flat roller on a flat track. An analysis of the suspension system is given in Ref. 1.

A reduced scale model of an antenna azimuth bearing employing this concept was constructed. The following describes the model and the test results obtained so far.

II. Purpose of the Model

The model was designed to test several independent things pertaining to an antenna azimuth bearing wherein the supporting wheels are attached to the alidade base and the track is grouted onto a concrete foundation. Among these are the wheel and track resistance to contact surface failure, grout strength, circumferential movement of track on its supporting grout, effect of a large temperature difference between the wheel supporting structure and the track, and the amount of the rolling resistance torque. The model was designed to produce a wheel cyclic loading approximately 3200 times as fast as that of an antenna prototype.

The failure mode of a wheel and track usually is spalling of the contact surface caused by the propagation of fatigue cracks. The maximum shear stress in the neighborhood of the contact surface varies as the square root of the applied load, therefore it is efficient to use high strength materials for both the roller and track. Commercial roller bearings use extremely strong steel of Rockwell C hardness 60 or more. Since the high stress regions exist only near the surface, it would be prudent to design large wheels and tracks to have hard surfaces on relatively soft mild steel in order to minimize material costs. If the track is composed of a continuous ring made of mild steel and covered with replaceable hardened wear strip segments, advantage can be taken of the high material strength in the high stress region. This permits the ring or track runner to be made of much less expensive mild steel, and makes possible a fast replacement of a damaged or badly worn wear strip. For large wear strips it seems best from an economical standpoint to use medium hardness values, say, from 35 to 40 Rockwell C. Although there is a great amount of information on the fatigue failure of both very hard and very soft steels, such as commercial bearing steel and mild carbon steel respectively, there is a paucity of fatigue information on steels in the hardness range of 35 to 40 Rockwell C, when subjected to the cyclical loading experienced by a wheel rolling on a flat track. Ordinary rotating beam cyclical loading tests are deemed inappropriate for the determination of an endurance limit for roller bearing stresses, since in the bearing the peak tensile and compressive stresses are of entirely different absolute values.

The conclusions reached by two independent studies conducted for JPL by bearing engineers, Refs. 2 and 3, were that it is highly desirable that model tests be made at the nominal Hertz stress level of 86184 (10^4) N/m^2 (125,000 psi) using surface hardnesses between 35 and 40 Rockwell C.

The track ring or runner must be made sufficiently flat and level and then grouted to its concrete foundation. The loads imposed on the grout and concrete by a rotating antenna are unusual in that the load at a point is completely on when a wheel is directly above it, and is completely off when the wheel is a short distance away from this point. In most civil engineering structures, a large percentage of the total load is steady. A further complication is that the grout or concrete under the runner will likely experience some tensile stress and this is difficult to evaluate by analysis. The existence of tensile stress may be understood by considering the problem of a small loaded area on the surface of a semi-infinite body, the solution of which gives surface tension just outside the loaded area. The analogous semi-infinite plate problem solution does not give any surface tensile stresses. The actual problem at hand lies somewhere between these two cases. The model tests should establish whether a cement grout is adequate for the proposed loads.

A phenomenon that has been observed with wheel-track systems is that of circumferential displacement of the track with respect to the grout or foundation. This will hereinafter be called track walking. At first glance it might seem that the high friction between the bottom of the track and the grout would be sufficient to resist the torque applied to the runner, and indeed it is. However, if the track is considered as a beam on an elastic foundation and without any shear connection to the grout, other than friction, it will be seen that the beam is displaced upward on both sides of a loaded wheel. If there is no tensile adherence between beam and grout there will be a space between the two. As the wheel advances there can be a rolling or ironing action of the track on the grout, causing the track to advance circumferentially. Whether this action occurs probably depends upon the shear strength between the track and grout away from the immediate vicinity of the wheel. It is important to determine if this action occurs, and if so, to develop adequate auxiliary restraints to prevent it.

Various empirical formulas have been employed to calculate the torque required to rotate a large wheel supported structure. The question arises whether the methods used to derive the torque resistance of a small roller thrust bearing would apply to the wheel and track system. The effect of misaligned wheels on the azimuth torque has been calculated. A model could confirm these theories.

During periods of maintenance or repair, antennas usually are not rotating in azimuth. If during this parked condition a differential temperature occurs between the track and the alidade base structure, which supports the wheels, there likely will be a radial displacement of the wheel with respect to the track. It was not known how much wheel and track damage would result from this scuffing action.

III. Description of the Model

The model being tested is as shown in Figs. 1 and 2. The track runner is a mild steel circular ring of 1.77-m mean radius. Its rectangular cross section is 30.48 by 10.16 mm. Each of eighteen hardened steel wear strips of 2.54-mm thickness and 30.48-mm width is fastened to the ring with 14 screws. The ring is then centered and leveled, and dry-pack cement grout placed in the 9.50-mm space between the ring and concrete pedestal.

The alidade base is simulated by the H-beam frame shown in Fig. 1. Each wheel truck is pivoted to the frame by two ball joints, one of which is adjustable by opposing set screws. This adjustment allows the truck to be rotated about a vertical axis for best wheel-to-track alignment.

The center pivot is a spherical roller bearing. Its inner race is attached to a steel pedestal. Its outer race housing is fastened to a diaphragm that joins a cylindrical shell fastened to the H-beam frame. The flexible diaphragm insures that all the ballast weight loading goes onto the wheels and not onto the center pivot bearing. A sprocket of pitch diameter 0.869 m is attached to the lower edge of the diaphragm-cylinder structure and a roller chain connects it to the drive unit mounted on the floor. Four screw jacks in the corners of the frame are connected by a chain so that one person can operate them in unison, thus allowing convenient wheel alignment and inspection. The flexible diaphragm also allows the frame to be jacked without the necessity of removing the center bearing.

The drive unit output sprocket has a pitch diameter of 0.1317 m and is driven by a 43:1 ratio reducer powered by a 0.75 kW, three-phase electric motor running at 1750 rpm. This gives the H-beam frame an angular speed of approximately 6 rpm. The motor can be driven in either direction.

The wheel loads come from the H-beam frame weight and from the ballast weights stacked onto the frame. Originally it was planned to have a wheel truck in each of the four corners of the H-beam frame. It was thought that the torsional flexibility of the frame would ensure a uniform distribution of loading among the four wheel trucks, provided the flatness tolerance of the track ring was met. Although each cross section of the track is level when viewed as a vertical radial plane, there are some slopes in circumferential directions. Therefore it was decided to employ three trucks only, making the reactions statically determinate, and adjust the ballast weights so as to give equal truck loadings when rotating at 6 rpm. The effects of centrifugal force on the reactions was considered in adjusting the ballast weights.

When two wheels were used on one truck, the proper angle between the two wheels was established in the metrology laboratory by an optical method. The truck assembly was subsequently aligned to the frame. The outer end of each wheel axle was made square with respect to the wheel axis. A special mirror with parallel surfaces could be spring loaded against the axle end by running a small coil spring through a hole in the axle. One end of the spring was attached to the back side of the mirror and the other end was pinned against the inner end of the axle. An autocollimating jig transit was positioned approximately one meter outside the track and at the elevation of the wheel. The instrument was aligned to a vertical pin extending upward from the center bearing pedestal. Then the frame was rotated until a wheel with mirror attached intercepted the line of sight. The truck was rotated about a vertical axis, by adjusting the opposing set screws on the ball pivot joint, until autocollimation on the wheel mirror was obtained. It is believed that by this method the wheel alignment can be known to within 1 arc minute.

The wheel suspension system is shown in Fig. 2. Each wheel bearing housing frame is connected to the truck frame by a pair of flexure struts at each side of the wheel. The flexure struts extended would intersect at the top of the track surface, thus they act as truss members and are capable of resisting both vertical and horizontal forces applied at the wheel-track contact point. The resistance to a moment, however, is very small. This means that the interface moment between the wheel and track will be very small thereby precluding any high edge loading of the wheels even though the wheels and track may have a relative misalignment. This is an important advantage because in practice it is difficult to prevent a relative angular displacement between the wheel and track. Such displacement, for example, may come from needed manufacturing tolerances, or from a warping of the track foundation. The interface moment magnitude is discussed in Refs. 1 and 9.

There are disadvantages associated with this suspension system. Its intrinsic complexity is tempered somewhat by the fact that it can use smaller wheels because of the nearly uniform loading achieved across the width of the wheel. Perhaps its principal disadvantage compared to conventional designs is its smaller stiffness in the vertical direction. This is due to the small cross-sectional area of the flexure struts. In many instances, however, this may reduce a vibrational frequency by less than 10 percent.

The wheels of the model are tapered so as to be true rolling elements. The mean diameter of the wheel is 50 mm and its width is 12 mm. As originally designed, it was a 1/20 scale model of a 100-m diameter antenna, but can be considered also as a larger scale model for smaller antennas.

IV. Scale Effects

A model geometrically similar to the prototype and properly loaded in the same manner will have the same stresses as the prototype. If the model length is λ times the prototype length, the proper loads to be applied to this model, in order to match the prototype stresses, are λ^2 times the prototype loads. Much significant information can be obtained from a reduced scale structural model. However, it must be realized that the scaling does not apply to the material grain size and probably not to surface finishes. Therefore, the statistical fatigue failure performance of models may differ appreciably from that of their full-scale counterparts. This is known to be true for the spalling failure of commercial rolling bearings. The maximum shear stress, τ_{MAX}, in the neighborhood of the cylindrical roller and track contact is: (from Ref. 4)

$$\tau_{MAX} = 0.179\sqrt{\frac{PE}{DL}} \tag{1}$$

where

P is the normal force between roller and track

E is the common modulus of elasticity of the wheel and track

D is the roller diameter

L is the roller length

From this equation it is clear that the model stress will match the prototype stress if the model is geometrically similar, has the same elastic modulus, and is loaded so that:

$$P_{MODEL} = \lambda^2 P_{PROTOTYPE} \tag{2}$$

Rearranging Eq. (1), we obtain:

$$P = \frac{DL}{E} \frac{\tau_{MAX}^2}{(0.179)^2} \tag{3}$$

Reference 5 presents an empirical formula that establishes the dynamic capacity of commercial roller bearings. The dynamic capacity is defined as that load which at least 90 percent of a large sample of bearings will endure for 1,000,000 revolutions without exhibiting any fatigue-induced spalling of the rollers or races. As applied to a roller thrust bearing, which corresponds to the case at hand, the equation is:

$$P_1 = L^{7/9} D^{29/27} f_c \tag{4}$$

where f_c is a dimensional factor having the dimensions of force/(length)$^{50/27}$. For a geometrically similar model made of the same material as its prototype, the relationship between P_1 model and P prototype, derived from Eq. (4) is:

$$P_{1\ \mathrm{MODEL}} = \lambda^{50/27} P_{\mathrm{PROTOTYPE}} \tag{5}$$

Dividing Eq. (5) by Eq. (2), we obtain:

$$\frac{P_1}{P} = \frac{1}{\lambda^{4/27}} \tag{6}$$

which is the factor by which the reduced scale model load must be increased in order to be a fatigue model of the contact area. Since Eq. (4) pertains to commercial bearing steels that are very hard, and the subject model is made of medium hardness steel, it is not known how accurate the load factor of Eq. (6) would be when applied to the subject model. Equation (4) as applied to commercial roller bearings has been experimentally verified by a large number of tests conducted by various experimenters (Ref. 6). However, there is a theoretical rationale supporting it. This is based upon Weibull's premise that as material volume is increased, the number of potential crack sources is also increased. Lundberg modified this concept to apply to the highly stressed region near the roller-track contact. It would appear that the factor represented by Eq. (6) would not be unconservative when applied to steels of reduced hardness.

V. Model Tests

The tests conducted so far have employed six wheels each loaded to 5560 N (1250 lb) which produced Hertz stresses of 86184 (10^4) N/m^2. The first test ran a total of 40.60 hours, half of which was in the clockwise direction. Since the model turns 375.55 turns per hour, this was a total of 15,247 turns. It is estimated that the DSN antennas make approximately 1000 turns per year, thus the first test corresponds to about 15 years of antenna operation as far as the wheels are concerned. Regarding the antenna track, the test corresponds also to 15 years of operation provided the antenna has six wheels, and to a greater or smaller amount depending upon the number of antenna wheels. Most of the first test was run with no lubricant on the track.

Very small pits soon occurred on the wear strip and wheel surfaces. It was judged that these were not from fatigue but resulted from small particles being broken from the sharp edges of the wear strip mitered ends. The particles were distributed around the track by the wheels. When this action was understood, the wear strip ends were stoned to a small radius and the pitting action seemed to cease.

The wear strips were reground slightly in preparation for the second extended run. Upon removal of the wear strips, it was observed that there was fretting corrosion between the wear strips and their steel support ring. Various lubricants, namely, SAE No. 30 engine oil, bearing grease, chassis grease, thread lubricant, MoS_2, dry graphite, and a commercial product called Cortec were placed on the bottom surfaces of the wear strips. During the second extended run, some of these lubricants were extruded through the wear strip mitered joints and smeared over the upper surface of the wear strips, thus producing an inadvertent lubrication on the wheel track surface. This second test was run for a total of 81.54 hours, approximately half of which was in the clockwise direction. The total time for the two tests was 122.14 hours or 45,869 revolutions. There was no evidence of fatigue failure of wheels or wear strips. Of the several lubricants used between the wear strips and their support ring, only the thread lubricant (Silver Goop) and the Cortec VCJ-309 were completely effective in preventing fretting. Various platings will be tried on future tests.

A. Track Walking

After three hours of counterclockwise rotation of the model, it was observed that there was no circumferential displacement of the track with respect to the runner, i.e., there had been no track walking. After an additional 3.75 hours in the same direction, it was observed that the track had moved 17.8 mm in the counterclockwise direction. After 1.13 hours more of rotation in the same direction, the walking displacement had reached 25.4 mm. In the 4.88 hour period in which the 25.4-mm displacement occurred, the model made 1833 turns, and the average displacement per turn was 0.0139 mm. Initially there was a tensile bonding between the track ring and the cement grout. A short distance from the loaded wheels there was a tensile stress at the bond, per Winkler beam theory. After three hours of cyclical loading the bond failed in tension, thus allowing the track to separate slightly from the grout. The loaded wheels then produced an ironing or smoothing action on the track, which caused the "wrinkle" to advance in the direction of motion.

Four mild steel tangent links, each having a cross-sectional area of approximately 21 square millimeters, were then installed to connect the track ring to the concrete foundation. Each link was attached to the track ring with one 4.8-mm diameter steel screw in shear. After 1.5 hours of running, one of these screws had yielded and was on the verge of breaking. It is probable that this one screw was resisting the entire walking force since the anchor nuts in the concrete at the other links were loose. Four larger tangent links having a cross sectional area of 91 square millimeters were installed using 6.35-mm diameter screws. These have not failed after many more hours of testing.

B. Grout

After 40.6 hours of operation the track ring was raised from the grout enough to make its surface visible over a distance of approximately 0.30 m. There were no loose particles and no evidence of the track ring having slid over the grout. This verified the concept that the circumferential displacement of the track was of the nature of a rolling action on the grout, rather than sliding.

After an additional 49.13 hours of model rotation, it was observed that the track ring over an angular distance of 80 degrees had been displaced outward, with respect to the grout by as much as 1.4 mm. The outer shoulder of the grout was broken away over this 80 degrees; however, the grout beneath the ring appeared to be sound. The reason for this is not presently understood. Misaligned wheels can exert a large radial force on the track, but it would seem that the frictional resistance between grout and track would be sufficient to resist the induced force from a misaligned wheel. It may be that when circumferential walking is prevented by tangent links, the ironing action of the loaded wheels tends to produce a radial displacement of the track ring. It is important that this effect be understood.

The maximum grout compressive stress, σ_{BR}, has been calculated by the following equation, derived from Refs. 7 and 8:

$$\sigma_{BR} = \frac{0.306\,P}{(1-\nu^2)^{0.2775}\,h^{5/6}\,b^{7/6}}\left(\frac{E_F}{E}\right)^{0.2775} \tag{7}$$

where

P is the wheel vertical load

ν is Poisson's ratio of the foundation

h is the beam (track runner) depth

b is the half width of the beam

E_F is the elastic modulus of the foundation

E is the elastic modulus of the beam

This equation may be used for comparing the calculated compressive stress on the grout with various standards such as building codes, etc. As stated before, the nature of the cyclical loading is uncommon and it is difficult to correlate these calculated compressive stresses to endurance stresses as determined by other methods of cyclical loading.

If there should be a horizontal force applied at the wheel-track contact surface as a result of a misaligned wheel (Ref. 9), the maximum compressive stress on the grout will be more than that given by Eq. (7). This is the primary reason for maintaining good wheel alignment in the plan view.

C. Frictional Torque Resistance

For the case of properly tapered wheels, the frictional torque resistance, T, can be considered as composed of three parts, namely, the rolling resistance of the wheels on the track, the resistance of the wheel bearings, and the sliding resistance caused by misaligned wheels. Each of these components is proportional to the product of total weight, W, and the mean wheel radius, R; therefore, the total frictional torque is:

$$T = WR\left[f_1 + f_2\frac{d}{D} + \mu \sin\theta\right] \tag{8}$$

where

f_1 is the factor for roller thrust bearings

f_2 is the factor for radial ball bearings on the wheel axles

μ is the coefficient of sliding friction

θ is the misalignment angle of the wheels in the plan view

d is the wheel bearing bore diameter

D is the wheel diameter

Using values from Ref. 5 for f_1 and f_2 and setting d/D = 0.30, μ = 0.30, and $\sin\theta$ = 0.00015, which is the expected value of the wheel misalignment, we obtain from Eq. (8):

$$T = WR\ [0.0011 + 0.0015(0.30) + (0.30)\,0.00015]$$

$$= WR\ (0.0016) \tag{9}$$

The drive chain was temporarily removed and the model rotated at a uniform angular velocity of approximately 6 rpm by applying a tangential force at a radius of 1.84 m. The force was applied through a force scale and the values varied between 53 and 71 N. Using the average of 62 N, the torque was 114 N · m. The total weight, W, was 33188 N and the mean wheel radius was 1.775 m. From these values, the measured friction coefficient, f_m, is:

$$f_m = T/WR = 114/33188(1.775) = 0.0019 \tag{10}$$

which is to be compared with the factor 0.0016 of Eq. (9).

Subsequently all the wheels were misaligned by approximately 13.7 arc minutes. This was accomplished by turning each alignment set screw one half turn. The torque required to rotate the model at approximately 6 rpm varied between 158 and 190 N · m. If we use the average of 174 N · m, Eq. (10) yields:

$$f_m = T/WR = 174/33188(1.775) = 0.0030 \tag{11}$$

Calculating the torque T from Eq. (8) by using the new value of $\theta = 13.7$ arc seconds, we obtain:

$$T = WR\ [0.0011 + 0.0015(0.30) + (0.30)0.00398]$$

$$= WR\ (0.0027) \tag{12}$$

The value 0.0030 of Eq. (11) is to be compared with the factor 0.0027 of Eq. (12).

Therefore it would appear that the methods of estimating the frictional torque of small bearings can be applied to large bearings and reasonably accurate results obtained.

D. Traction Capacity

The traction capacity was measured by tying the model turntable to the building structure and applying torque to a wheel until it slipped. The ballast weights were adjusted until the wheel reaction was 1214 N. A 6.35-mm diameter bolt was put through the hole in the wheel axle and a nut tightened onto it. The hexagon head of the bolt was then torqued with a torque measuring wrench until slippage occurred. The values recorded were the maximum ones before slippage and correspond to the static capacity. The static coefficient of friction was calculated from the following equation:

$$\mu_s = t/rw \tag{13}$$

where

t is the measured torque

r is the radius of the wheel

w is the vertical load on the wheel

First the wheel and track were wiped clean with acetone and several torque measurements made. The calculated value of μ_s ranged from 0.351 to 0.395.

Second the wheel was lubricated with SAE No. 30 non-detergent engine oil. The calculated value of μ_s was 0.153.

The wheel and track were wiped with paper, thus removing most of the oil. Then several tests were made. The calculated values of μ_s ranged from 0.242 to 0.286.

It was recommended in Ref. 3 that a small amount of lubrication be provided by means of rubbing a piece of oil impregnated plastic against the track. It is believed that this kind of lubrication would approximate the lubrication obtained when the SAE No. 30 oil was "wiped off".

E. Thermal Expansion

If a heavily loaded nonrotating wheel is forced to slide transversely across the track, some damage will be incurred. For the loads and expected displacements under consideration, it was not known how serious the damage would be. The model turntable was restrained from rotation and the H-beam members of the turntable base were heated by playing torches over their surfaces. A dial indicator was mounted between the track and the end of a wheel axle. At this time all the wheel loads were 5560 N, corresponding to a Hertz stress of 86184 (10^4) N/m^2. The heating continued until the dial indicator read 1.40 mm. During the heating period, the displacement of the dial indicator was not continuous but incremental and accompanied by popping sounds. Two and a half hours after the heating ceased, the indicator read 0.63 mm. Eighteen hours later, when the H-beam members surely were back to their original temperatures, the dial indicator read 0.51 mm. The model was then jacked up exposing the smear marks at all six wheel positions. The width of the marks was approximately equal to the calculated contact area width and the length was approximately equal to the wheel width plus 1.40 mm. Although the marks on both wheels and wear strips were clearly visible, they could not be felt with a finger nail. Examination with a 10X magnifier revealed little damage, which could be polished out easily. Therefore it can be concluded that infrequent transverse sliding of nonrotating wheels across the track will produce only negligible damage.

F. Conclusions

From the tests conducted so far the following conclusions can be made:

(1) The track must be restrained circumferentially if portland-cement-type grout is used. Four tangential links of moderate size are adequate to resist the walking action.

(2) The torque required to overcome the rolling resistance of the wheels can be estimated fairly well by Eq. (8). Since the rolling resistance is usually small in comparison to other resistances, the accuracy of Eq. (8) should be satisfactory.

(3) The traction capacity has been measured and found to be consistent with values obtained from other sources.

(4) The infrequent transverse sliding of a nonrotating wheel across the track produces only negligible damage when the contact stresses and material properties are as herein stipulated.

(5) The wheels, wear strips, and grout have survived 45,869 model turntable revolutions with no sign of fatigue failure. Additional tests must be run at higher wheel loadings in order to compensate for scale effects.

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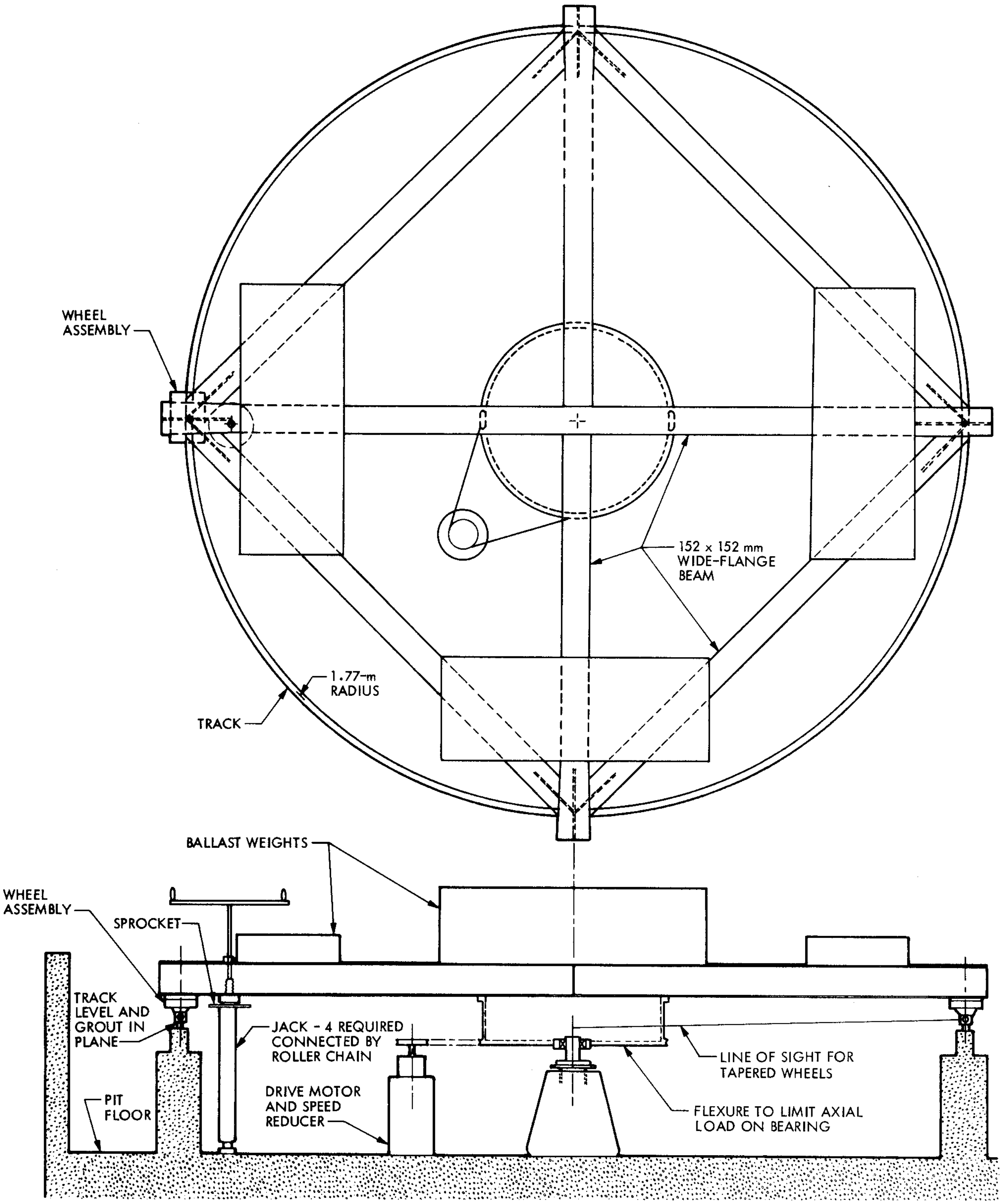

Fig. 1. Antenna azimuth bearing model, general arrangement

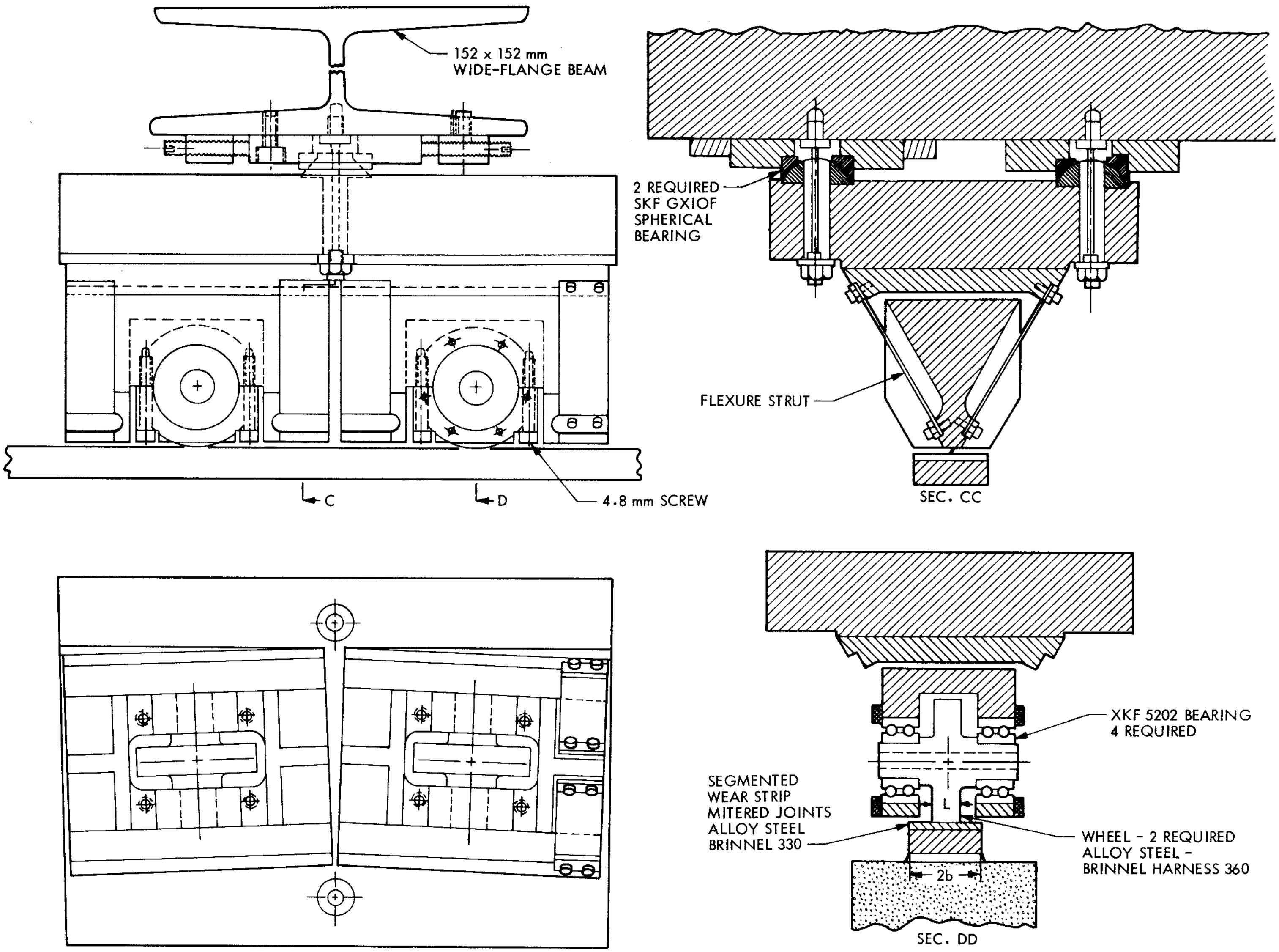

Fig. 2. Wheel suspension system