National Radio Astronomy Observatory Socorro, New Mexico

VLBA Test Memo No. # 54

Investigation of VLBA Azimuth Wheel Bearing Failures

J. E. Thunborg July, 1997

I. Introduction

Several VLBA antennas have had wheel bearing failures. Since we are still in the early stages of the antenna service life, we would expect very few bearings to fail. This test memo presents analysis and inspection data that suggest that the bearings are failing due to fatigue. If this is the case, we would expect our failure rate to increase drastically over the next few years. Since it is very expensive to change wheel bearings, we need to understand why these bearings failed and make changes that will extend their life. This test memorandum analyzes the forces on the bearings and calculates their projected service life as well as lists possible reasons for early failures.

II. Bearing Failures

Repetitive failures can be divided into three regions: early-life failures (infant mortality), mid-life (random) failures and wear out. Early life failures are normally caused by faulty materials, installation and handling errors. Mid-life failures are caused by random effects external to the component, e.g. operating changes, lightning strikes, hurricanes. Wear-out can be the result of mechanical wear, fatigue, corrosion, etc. If we examine our latest bearing failures we will see that they are beginning to wear out. If this is the case, then the time between bearing changes will become more frequent. We can assume that all of the bearings are starting to wear out and will eventually need replacement.

If we look at each failure individually we can find a pattern that provides the clues necessary to find the cause. Our first wheel bearing failure occurred at Los Alamos. This bearing was replaced by RSI. It was evident that this bearing was never greased. This would clearly fall into the infant failure category. The next failure occurred at Kitt Peak. On this antenna, metal chunks chipped off from the edge of the outer race on an outside drive bearing. A careful inspection of this bearing showed signs of fatigue. The fatigue flakes and isolated pieces of bearing material occurred on the outside edge of the bearing. This suggests that the bearing was subjected to an excessive thrust load. This is very interesting because the telescope is designed so that the outside bearing is free to float axially within the pillow block. Therefore, this bearing should never see a thrust load.

Recently St. Croix had a similar problem with an outside drive wheel bearing. The inside wheel bearing on the same axle was also replaced because metal particles were found in the grease. Inspection of the outside bearing showed that it failed in fatigue due to excessive dynamic thrust loading just like the Kitt Peak bearing. Inspection of the inside bearing showed that it also failed in fatigue, but the one-sided fatigue flaking due to thrust was less evident. It is also interesting to note that this drive axle was worked on the previous year because the axle had slipped out of its pressed on coupling. Subsequent testing revealed that the coupling was not assembled according to specification. The coupling had only 0.003" interference and began slipping with 80,000 lb. of force. The force required to pull a correctly assembled axle with 0.008" interference from its coupling should exceed 270,000 lbs as calculated in Appendix B. Even though the coupling failed because it was not assembled correctly, it demonstrated that the wheel was subjected to a thrust load of 80,000 lbs.

Metal particles were found in both outside drive wheel bearings on the Fort Davis antenna. Metal particles were also found in an outside bearing on an idler wheel. This is the first time we have seen problems with an idler bearing. These bearings have not been removed or inspected yet. However, metal flakes are usually caused by fatigue. If this is the case, then we have a pattern developing and we can see that we are rapidly approaching the service life of the bearings on all of the antennas.

III Bearing Life

In order to estimate the service life of the bearings, we must first understand the loads in the bearings. The dead weight (ignoring wind, track alignment, etc.) on the azimuth drive wheels is 153,300 lbs. In the ideal case this load is distributed as shown in Figure 1. The bearings at the gearbox are very stiff, tapered roller bearings. These bearings rigidly fix the drive shaft in all directions except rotation about the X-axis, which is constrained by the drive motors. The inside pillow block supports a Torrington number 23038 BR spherical roller bearing. This bearing constrains the shaft from moving along the Y and Z axes. A spacer ring was installed in this pillow block to constrain the axle from movement along the X-axis as well. The outside pillow block supports the same Torrington bearing, but the spacer is left out permitting motion along the X axis.

The axis of the axle is 3.5 degrees off horizontal. Therefore, a thrust load develops along the axle and the radial load is not divided equally between the bearings. The antennas were designed so that the thrust is taken by the inside bearing, which has the lighter radial load. The bearing life calculation in Appendix A shows that if the inside bearing bears the thrust load, then the equivalent load in the bearings is balanced and both bearings should have approximately equal life. However, I have noticed that a large percentage of the outside bearing to bear a portion of the thrust load, which added to its already high radial load greatly shortens its life. This does however extend the life of the inside bearing, which is much harder to replace.

The wheel loading discussed in the previous paragraph assumes that the axle is perfectly aligned with the track. The wheels are machined as a frustum of a cone with the nominal tip at the azimuth axis and in the plane of the top track surface. If the axle of the wheel does not point through the tip of the cone, the wheel will try to roll on either a larger or smaller diameter circle. The pintle bearing and structure will not allow the wheel to follow a different path so forces develop until the wheel slides along the track. This sliding can be demonstrated by marking a line across the wheel and the track at a point where the wheel rests on the track. If the antenna is then rotated a short distance and then brought back to the same point the lines will not line up. This sliding takes place on all of the VLBA antennas. The force required to make a drive wheel slide across the track was calculated to be approximately 50,000 lbs. (NOTE: St. Croix, which is subject to rusty rail and wheels can see much greater loads.) This thrust load must be reacted against by the azimuth bearings.

Another circumstance that can increase the load on the bearings is a misalignment between the bearings and the gearbox. Since the drive axle is rigidly coupled to the drive gearbox, the axle is over constrained and any misalignment must be countered by deflection of the bearings and drive shaft. This misalignment can create very large forces within the bearings. The bearings on the gearbox output shaft are very stiff tapered roller bearings that should be able to handle the forces. These forces, however, will drastically decrease the life of the azimuth drive bearings.

Table 1 lists the expected minimum life for 90% of the bearings if they are subjected to the previously mentioned loads. These lifetimes were calculated as a comparison of different load profiles and are not indicative of the actual bearing life. The calculations are based on 24 hour per day operation at 1 rpm.

High winds will also decrease the life of the bearings. I have not evaluated the effects of wind loading on the bearings, but I would expect that the bearings will not last as long at sites with high winds or constant moderate winds.

VLBA - AZIMUTH DRIVE WHEEL





Up to this point I have only discussed azimuth drive wheels. The idler wheels are supported by the same Torrington 23038 BR bearings as the drive wheels. A spacer is also placed in the inside pillow block to ensure that the thrust will be reacted against by the inside bearing. The dead weight (ignoring wind, track alignment, etc.) on the idler wheels is 75,900 lb.. The reaction forces within the bearings are 43,700 lb. for the outside bearing and 32,000 lb. for the inside. The inside bearing also takes a 4,600 lb. thrust load. Under this relatively light loading, the idler bearings should greatly outlast the drive wheels.

Table 1.	L ₁₀ Lif	e Com	parison
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Description of load case	L ₁₀ Life (Years)	
	Inside Bearing	Outside Bearing
Drive wheel, Perfectly aligned with both gearbox and track. Inside bearing takes thrust load.	15.5	15.0
Drive wheel, Perfectly aligned with both gearbox and track. Outside bearing takes thrust load.	42.5	6.9
Drive wheel, Perfectly aligned with gearbox, 50,000 lb. outward load at track. Inside bearing takes thrust load.	1.0	4.5
Drive wheel, Perfectly aligned with gearbox, 50,000 lb. inward load at track. Inside bearing takes thrust load.	0.4	435
Drive wheel, Perfectly aligned with gearbox, 50,000 lb. outward load at track. Outside bearing takes thrust load.	870	0.4
Drive wheel, Perfectly aligned with gearbox, 50,000 lb. inward load at track. Outside bearing takes thrust load.	5.3	0.9
Drive wheel, Perfectly aligned with track. 0.031" misalignment at gear box. Inside bearing takes thrust load.	3.8	42
Drive wheel, Perfectly aligned with track. 0.031" misalignment at gear box. Outside bearing takes thrust load.	11.4	5.4

IV. What Now

The analysis in the previous sections indicates that the bearings we are currently using are failing because they are not adequately robust for the high loads to which they are subjected. A thrust load is extremely detrimental if it is reacted against by the outside bearing. We can increase the lifetime of the outside bearings by centering them within the pillow blocks so that they do not see the thrust component of the load. This, however, shortens the life of the harder to replace inside bearing.

All of the bearings have failed at the top of the outside race. This is because the outer race is fixed within the pillow block and the load is applied at the same place regardless of the wheel position. The bottom part of the outer race never comes in contact with the rollers and therefore shows no wear. We can therefore greatly extend the life of the bearings by rotating the outer races 180 degrees. This can be accomplished without removing the wheel.

The above solutions are "band aids" to the larger problem of not having large enough bearings. Using bearings with larger outside diameters would require moving the pintle bearing. This should be avoided if at all possible. Torrington's 24038 spherical roller bearings have the same inside and outside diameter

but are 25% longer. These bearings greater load capacity should give us 3 times the life as the current bearings. There are two problems with this bearing. The first is that it will not fit in our current pillow blocks, So we would have to fabricate our own steel pillow blocks. The second problem is that if we don't deal with the over-constraint problem that comes from the rigid coupling between the axle and the gearbox, the stiffer bearings may cause the bearings on the gearbox output shaft to fail. These bearings would be even harder to fix.

V Recommendations

Short term: The course of action I suggest is to replace the Fort Davis bearings during a double maintenance day. This should be done as soon as possible because the inspection of the St. Croix bearing demonstrated that bearings with metal particles in the grease could be in very poor condition. We will also gain additional information by inspecting these bearings. We should also rotate the outer races of the bearings and center the outside bearings within their pillow blocks. This should be done at all of the antennas during tiger team visits.

Long term: The above solutions will buy us some time, but we should see if it is possible to develop a long term solution. We should evaluate the possibility of designing a more robust bearing arrangement with a flexible gear box coupling. If we can come up with a cost effective design, we should fabricate a prototype and try it out at Pie Town for a couple of years. This way we will be ready, if our failure rate starts increasing exponentially.

APPENDIX A Bearing Life

e := .28	Load Test
X ₀ := 1 X ₁ := .67	Radial Load Factor
Y ₀ := 2.44Y ₁ := 3.63	Thrust Load Factor
C := 164100-lb	Dynamic Load Rating
Nr := 1	Operating Speed (rpm)

Inside Bearing:

R := 64640·lbRadial LoadT := 9340·lbThrust Loadi :=
$$\left(\frac{T}{R} > e\right)$$
i = 0P := $X_i \cdot R + Y_i \cdot T$ Equivalent LoadP = 8.743·10⁴ ·lbL_{10} := $\frac{16667 \cdot hours}{Nr} \cdot \left(\frac{C}{P}\right)^{3.3333}$ Expected minimum life for 90% of the bearings in a given population.L_{10} = 15.519 · years

Outside Bearing:

 $R := 88376 \cdot lb$ Radial Load $T := 0 \cdot lb$ Thrust Load

$$i := \left(\frac{T}{R} > e\right) \quad i = 0$$

$$P := X_i \cdot R + Y_i \cdot T$$

$$P = 8.838 \cdot 10^4 \cdot lb$$

$$L_{10} := \frac{16667 \cdot hours}{Nr} \cdot \left(\frac{C}{P}\right)^{3.3333}$$
Expected minimum life for 90% of the bearings in a given population.
$$L_{10} = 14.972 \cdot years$$

AFPENDIX B. SHRINK FITS

Shaft Hole Diameter ad := $0 \cdot in$ bd := 6.9325 · in Shaft Diameter Hub Outside Diameter $cd = 11 \cdot in$ $\mathbf{E} := 3000000 \cdot \frac{\mathrm{lb}}{\mathrm{in}^2}$ Modulus of Elastisity $\delta = .008 \cdot in$ Interference Length of Fit 1 := 6·in **f** := .1 **Coefficient of Friction** $\alpha := 0.0000065 \cdot \frac{1}{\text{deg}}$ Coefficient of Thermal Expansion Desired Assembly Clearance cl := .006 · in

$$\begin{array}{rcl} \mathbf{ft} &\coloneqq& 12 \cdot \mathbf{in} \\ \mathbf{tons} &\coloneqq& 2000 \cdot \mathbf{lb} \end{array} \quad \mathbf{a} &\coloneqq& \frac{\mathbf{ad}}{2} \quad \mathbf{b} &\coloneqq& \frac{\mathbf{bd}}{2} \quad \mathbf{c} &\coloneqq& \frac{\mathbf{cd}}{2} \end{array}$$

$$\mathbf{P} := \frac{\mathbf{E} \cdot \delta}{\mathbf{b}} \cdot \left[\frac{(\mathbf{c}^2 - \mathbf{b}^2) \cdot (\mathbf{b}^2 - \mathbf{a}^2)}{2 \cdot \mathbf{b}^2 \cdot (\mathbf{c}^2 - \mathbf{a}^2)} \right]$$
$$\mathbf{P} = 2.087 \cdot 10^4 \cdot \frac{\mathbf{lb}}{\mathbf{in}^2}$$

Pressure Between Shaft And Hub

$$F := \pi \cdot f \cdot P \cdot bd \cdot l$$
Pressure Required in Making Press Fit $F = 136.353 \cdot tons$

- $T = \frac{1}{2} \cdot \pi \cdot \mathbf{f} \cdot \mathbf{P} \cdot \mathbf{l} \cdot \mathbf{bd}^{2}$ Torsional Holding Ability $T = 7.877 \cdot 10^{4} \cdot \mathbf{ft} \cdot \mathbf{lb}$
- t = $\frac{(cl + \delta)}{bd \cdot \alpha}$ Hub Temperature Required For Assembly

 $t = 310.688 \cdot deg$