CHARA TECHNICAL REPORT

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OPLE Drive Issues

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1. INTRODUCTION TO THE OPLE DRIVE WHEEL ISSUE

The OPLE design adopted for the CHARA Array is the JPL design used successfully at several facilities, most recently at the Palomar Test Bed Interferometer. CHARA is considering several minor design changes for purposes of economy and simplification of fabrication.

The JPL design uses a direct friction drive, consisting of a metal drive wheel which is pressed firmly against a metal bar. The JPL specifications of this bar are given in the final section of this report, and the specifications of the drive and roller wheels are discussed below.

The first concern here is that the drive rail stock specified is relatively expensive (on order $12/ft), that the grinding operations are expensive, and that the combination of grinding and hardening requested results in warpage of the rails.

One of the changes under discussion is the substitution of a softer and less expensive material for the metal drive rails. Concern for drive lifetime then leads to additional possible changes to the drive wheels. Some consideration of these design aspects is given below followed by some potential alternatives. Recommendations and some cost factors are given in the last sections.

The JPL drive design concept under consideration here is straightforward: a hardened traction roller wheel attached to a torque source will act against an even harder flat rail surface while a constant preload is applied through two other rollers (hardened cam followers) to maintain contact between the drive wheel and the rail. Hard drive rollers acting on a hard rail or journal surface is a standard design strategy although one that requires careful consideration of materials, heat treatment, and machining sequences. At issue here is whether CHARA needs to adopt this design as is or whether some simplification can be introduced, particularly with respect to the 1200' of rail that will be required for the delay line assemblies.

An additional consideration is the joint configuration between rails. The JPL specification is to grind the rail ends "flat and square. Do not break edges" (i.e., the corners should be left square, not rounded off as would normally be done). This operation is required to obtain tight fitting joints at the rail ends, but other configurations may offer somewhat easier installation with similar performance.

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2. GENERAL DESIGN ISSUES

As noted above, the drive wheels and the preloading cam rollers in the JPL design are intended to be quite hard (48 Rc) with the rail made even harder. The rail is specified to be made of 4340 steel hardened to 52 Rc which requires an oil quench hardening process and leads to the problems with rail warpage and mismatch at the rail joints mentioned by JPL staff. Long slender parts such as a 12' long rail with a 1" x 2" cross-section would be particularly susceptible to some warpage. Corrective measures for the warped condition would involve bending the rail back into a near- straight condition and performing a final grinding operation on both sides of the rail. Grinding would have to be done with the rail securely clamped and should be done in the same manner on both sides it order to leave the rail in a balanced stress condition. Grinding tends to leave a residual compressive stress in the ground surface which, if not equalized on both sides, will cause warpage also.

The drive wheel diameter (1.16") and the cam roller diameter (0.75") are small which leads to high contact stresses under the 80 lb. preload applied to keep the drive wheel pressed against the rail. This aspect of the design is unfortunate since these small diameters cannot easily be increased without design changes in other parts of the drive assembly which would greatly enlarge the scope of the job.

The drive wheel is specified to have a slight crown (24" radius) which effectively makes the wheel rim into a double-convex surface. The cam followers are also specified to have a crowned outer surface. Theoretically, a crowned surface would produce point contact between the wheel and the rail which would avoid skidding between the surfaces if the roller axis is misaligned relative to the rail surface. In practice, the contact stresses at the point of contact are always high enough to cause some flattening of the convex surface and some deflection of the rail surface so a finite contact area is created to carry the load. In that case, some skidding will occur as the wheel rotates if misalignment exists. This condition cannot be avoided although hardening the parts will increase the yield strength and helps to reduce the permanent deformation.

To illustrate the parameters governing contact stress, consider maximum contact stress(s) for a sphere acting on a flat surface (Note 1) is given by:

$$\sigma = 0.918\sqrt{\frac{P}{D^2 - C^2}} \tag{1}$$

D = Diameter of the sphere

P = Load applied to the sphere to press it against the flat

$$C = \frac{1 - n_1^2}{E_1} + \frac{1 - n_2^2}{E_2} \tag{2}$$

n_1, n_2 = Poisson's ratio for the respective materials

E_1, E_2 = Modulus of Elasticity (Young's Modulus)

for the respective materials

A more complex procedure for calculating contact stress for a convex surface formed by two radii acting on a flat surface is also available2 but will not be presented here to avoid adding unnecessary complexity to this report. In its final format, the parameters for the more complex case vary in a similar way to the spherical case given above.

Contact stress for a 1.16" diameter steel sphere on a flat steel surface will be about 232,000 psi if a load of 80 lbs applied at the contact. This value reduces to about 177,000 psi if one of the materials is bronze instead of steel. If one follows the procedure in Roark & Young, Table 33, Case 4 to calculate contact stress for a convex surface formed by two radii, (0.58" wheel radius and 24" crown radius) a value of 86,000 psi is obtained for steel-on-steel and 68,000 psi for bronze-on-steel.

Contact stresses are largely compressive in nature and cannot be compared directly to tensile yield strengths. There is no simple way to ascertain a safe stress although some empirical rules have been established based on tests (Note 3). From these one may infer a reasonably safe contact stress for a cylindrical roller to be about 1.5× Tensile Yield Stress. A safe tensile yield strength thus would be about 57,000 psi for steel-on-steel and about 45,000 psi for bronze-on-steel.

The weaker material suffers most and, if it is the rail, will eventually develop a concave wear groove along the line of travel. The presence of a groove is not necessarily a problem if the wheels/rollers track perfectly along the groove. The risk to performance arises if the drive wheel can wander in and out of the groove and produce a change in the driving rate. If the design has provisions for ensuring that the roller can always track in the concave groove it creates, the risk to performance would be largely removed.

The drive wheel is the most critical element since it is the one connected to the torque source and must act predictably. The preloading cam rollers are somewhat less critical since they only follow. As long as the cam rollers act smoothly without adding abruptly to the drive torque requirement, it is not important whether they run in or out of their respective wear grooves.

3. OPTIONS TO CONSIDER

Since the rail material has not yet been purchased, other options than the JPL design specifications may be considered. Three options are discussed below.

3.1. Mild steel Rails (Option A)

A low-carbon or mild steel such as AISI grades 1010 through 1020 would be less expensive than the 4340 steel specified by JPL. Further savings would result if the steel could be used without heat treatment, although some uncertainty would exist concerning the mechanical properties of the rail if its thermal history was unknown. AISI 1010 steel has insufficient carbon content to warrant heat treatment, but other low- carbon grades can be significantly improved through annealing or normalizing. These heat treatments involve heating the steel to a temperature in the range 1600-1700F and then cooling in the furnace (annealing) or air cooled (normalizing). . A significant difference in properties also exists between hot-rolled and cold-rolled or cold-drawn steel. Properties of several low-carbon steel grades are listed below (Note 4).

AISI Grade	Min.Yield Strength psi	Hardness HB
1010 Hot Rolled	26,000	95
1010 Cold Rolled	44,000	105
1015 Hot Rolled	27,000	101
1015 Cold Drawn	47,000	111
1015 Annealed	41,000	111
1015 Normalized	47,000	121
1018 Hot Rolled	32,000	116
1018 Cold Drawn	54,000	126
1020 Hot Rolled	30,000	111
1020 Cold Rolled	51,000	121
1020 Annealed	52,000	111
1020 Normalized	50,000	131

The yield strength values may be compared to the previously estimated safe yield strength values of 45,000 psi for bronze-on-steel and 57,000 for steel-on-steel. If the bronze-on-steel option was adopted, all of the low-carbon steels in cold-drawn condition would be satisfactory. The AISI 1015 grade in a normalized state also satisfies the yield strength criterion as would all others with a higher carbon content. However, one must consider the drive wheel to be mated with the rail before choosing any of these materials. That will be done next. Since the drive wheel may be made weaker or stronger than the rail both of these options will be considered in sections to follow.

3.2. Drive Wheels Weaker Than the Rail

In operation, a crowned drive wheel constructed from a weaker material than the rail would flatten out and become a cylinder. The JPL drive wheel is 1.16" diameter x 0.375" thick which sets the maximum width for such flattening action to produce a cylinder (unless a design change is undertaken). The contact stress for this case is given by (Note 5):

$$\sigma = 0.798\sqrt{\frac{P}{L_c D_c C}} \tag{3}$$

Where

L_c = Length of the cylinder (i.e., flattened width of the wheel)

D_c = Diameter of the cylinder

C as defined earlier

Using Lc = 0.375" as an upper limit on width, one obtains a contact stress of 45,000 psi for an applied load of 80 lbs. Using the rule-of-thumb that maximum contact stress should not exceed 1.5 x Yield Strength for the given material, it is evident that the wheel material must have, at least, 30,000 psi yield strength.

Considering the drive wheel size, it is likely that a wrought or drawn length of round bar stock would be purchased from a metals supplier as raw material for the wheels. A variety of hardness and strength conditions are likely to be available. Most wrought or drawn bronze/brass materials in a soft condition have yield strengths below 20,000 psi. However, copper-based materials experience large increases in yield strength with mechanical working of the material. For example, a common free-cutting leaded brass, UNS # C36000, has a yield strength of 18,000 psi in the annealed condition which increases to 45,000 psi in the H02 temper condition (Note 6). If one assumes that flattening action would work-harden the wheel to an H02 condition, the required contact width would be about 0.19", well within the 0.375" width available. Desirably, one would purchase raw material in the H02 condition if it was available. However, softer material could be purchased and an actual test performed to determine the true effects of mechanical working on the chosen material.

There are candidate materials for the drive wheels with yield strengths above 30,000 psi, but these are normally obtained in the form of castings. Properties of these materials are listed below (Note 7).

UNS ASTM-SAE	Min. Yield Strength psi	Hardness HB
86100	48,000	180
86200	48,000	180
86300	67,000	225

Notably, all of these materials are harder than the low-carbon steels listed earlier and have yield strengths higher than most of them. It is possible that a drive wheel made of these bronze materials would actually be stronger than a low-carbon steel rail. However, a flat rail is a sturdier configuration than a convex crowned roller so it is not obvious which element would yield the most. Again, an actual test would be required.

3.3. Drive Wheels Stronger Than the Rail

A harder, stronger, convex crowned drive wheel applied to the rail would produce a wear groove which could result in the wheel wandering in and out of the groove as mentioned earlier. However, one could consider shaping the drive wheel so that it can create a well-defined wear groove which will improve the ability of the wheel to track within the groove. This could be done by re-shaping the drive wheel to incorporate a vee instead of the more gently contoured convex surface produced in the crowned roller.

In effect, the vee wheel approach does not try to avoid a wear groove but, instead, intentionally forms the groove in a way to help keep the roller engaged in the groove. The drive wheel would be hardened as specified in the JPL drawing then ground to form the vee. The required depth of the vee is very small, since the depth of the groove would be small. A flat, about .010" wide, would be left on the tip of the vee to enable the machinist to form and measure the shape accurately. If a normal preload of 80 lb. was applied, the resultant contact stress at the tip of the vee would be 270,000 psi, well above the yield strength of mild steel. Under these conditions, a groove will be formed in the rail to a depth that increases the contact area enough to reduce the compressive contact stress to the yield value or slightly below.

At initial assembly, the drive assembly would be mounted in its intended final configuration. The drive wheel would then be lightly loaded (i.e., to about 25% of the intended preload) and tracked along the rail from end to end. Next, the preload would be increased to about 50% of the intended value and again tracked from end to end. This procedure could then be done again at 75% and 100% preload values. The groove thus formed by the wheel would be in the correct location for the drive assembly.

In order for the vee wheel method to work well, a small amount of axial direction compliance will be needed to compensate for the radial runout errors in the drive carriage wheels (i.e., radial runout errors will cause the drive wheel to move up/down relative to the rail which the groove will tend to resist). Enough compliance may already exist in the bearings that support the drive wheel, but this should be verified. In the event the bearings prevent any axial compliance, alternative wheel configurations may be considered to accomplish the task, but with some loss of stiffness.

3.4. Hardenable Rails (Option B)

The 4340 steel presently specified for the rails could be used in an unhardened condition which would avoid some of the warpage problems caused by heat treatment. Even in its softest state (annealed) this material will be harder than about 215 HB which would enable using low- carbon steel or one of the stronger manganese bronze materials for the drive wheels. If problems developed, the rails could subsequently be hardened to comply with the JPL specifications.

3.5. Use a Different Rail (Option C)

The JPL design specifies a rectangular rail, 1.000" thick, with both faces machined flat. This would appear to have the advantage of good stress distribution, but the contact stresses are principally governed by the shape of the crowned rollers which largely negate this advantage. Other rail shapes may reasonably be considered. One feasible alternative would be a round rail that is harder than the rollers. Specifically, a 1.000" diameter rail shaft hardened to the range 55-60 Rc is commercially available from Thomson Industries, Inc. Thomson also sells a shaft support. The potential disadvantage of this approach is the cost. Prices, depending on quantity and material choice, will be in the range \$3-\$4/inch for "60 Case" steel shafts including supports and \$4-\$5/inch for stainless steel shafts.

In the event this option was adopted, the drive wheels and cam followers would not be specified with crowned surfaces, but would instead, be left cylindrical. The wear grooves would be created in the rollers which would be advantageous considering the relative costs of replacement if that became necessary.

4. RAIL JOINT ISSUES

The joint between rails represents a potential source of drive error if the drive wheel is able to drop into a gap between rails or experiences a misalignment between levels of the rail surfaces.

The drive wheel side of the rails is critical while the opposite side is not for reasons mentioned earlier; i.e., as long as the cam followers can roll smoothly without major perturbations on the driving torque requirement, it is not important whether they experience irregularities along the way. Thus, the two principal requirements for the rail joint are:

1. Maintain the rail end surfaces in tight contact.

2. Make the two critical drive-wheel-side surfaces coplanar at the joint.

This could be accomplished by using a diagonal joint configuration. At assembly, both drive-wheel-side rail surfaces would be clamped against a stiff, flat surface, then bolted together with an initial snug fit at the tapered end surfaces, using counter-sunk bolts. A pin would be installed, then the bolts would be tightened to the final condition. The bolts would force the tapered end surfaces tightly together while the assembly procedure just described would make the critical surfaces coplanar. The pin would prevent lateral slippage of the joint and also enable re-assembly in the correct condition if the joint had to be separated for any purpose.

The principal advantage of this joint configuration is the ability to clamp the end surfaces tightly which cannot - this cannot easily be done with squared-off end surfaces. The end surfaces must still be machined to a near-flat condition with no edge breaks at the critical surfaces in order to avoid unwanted gaps. However, ground surfaces are not required since the clamping pressure will help overcome the effect of high spots. With care in fitting the joints, it may be possible to use saw-cut surfaces which will greatly simplify fabrication. The critical locations for the gap-free condition are where the drive wheels roll over the joint line. Gaps elsewhere are unimportant which should make the assembly process somewhat easier and more specific.

5. RECOMMENDATIONS

The mild steel rails are the lowest cost option since the material costs will be less and any heat treatment will be simpler. The mild steel option is recommended. If available, cold drawn steel could be used with a possible saving in both heat treating and machining costs. The JPL thickness specification for the rail is 1.000/.997" which is an accuracy achievable with cold drawn shapes. This specification seems more stringent than necessary since there must be compliance in the spring preload applied to the cam followers. Such compliance should tolerate more than 0.003" thickness variation. If cold drawn steel is unavailable, AISI 1018 or 1020 steel in normalized condition should be used.

The critical requirement is that the rail surfaces be smooth where the rollers ride. Cold drawn steel should be smooth enough to be used as-is if the required dimensions can be obtained. If normalized steel is used, it is likely that surface scale from heat treatment will be present. Both of the major faces of the rail should then be ground equally to avoid stress-induced warpage. Material thickness allowance for grinding will be needed.

The tapered joint configuration described in the previous section is recommended since it may cost less than the JPL design and should perform better.

There is enough uncertainty regarding the best drive wheel configuration to warrant some experimental verification. The brass wheel will produce the least effect on the low-carbon rails and should be evaluated first. It is recommended that test wheels be fabricated from brass (UNS #36000 or equivalent) and tested against a low-carbon steel rail to determine the effect of mechanical working (i.e., flattening) of the wheel. A simple test unit consisting of a 2-wheel carriage carrying 160 lbs and driven back and forth on a short section of rail material would soon provide useful results. It would be useful to include a joint in the test rail to evaluate its performance as well. It is not necessary to test the exact drive wheel configuration (i.e., with shaft and bearing journals, etc.), only the portion that will actually contact the rail.

In the event, the ordinary brass drive wheels do not perform satisfactorily, a similar test could be performed using the crowned drive wheels which have already been made from 4340 steel. Since the rails will be softer than 4340 in any condition, it will not be necessary to harden the wheels. These wheels will almost certainly produce shallow wear grooves in the rail. The ability of the wheels to track within the groove smoothly could be examined by mounting an accelerometer on the test carriage.

In the event the crowned drive wheels made from 4340 steel do not track well within the shallow grooves and cause irregularities in the drive motion, the wheels may be modified into a vee configuration and tested using the groove-forming procedure outlined above.

It is probable that one of the three drive wheel options will prove satisfactory in the short-term test and may then be adopted with considerable assurance of longer-term success.

The mild steel rails will be more susceptible to rusting than their hardened counterparts, but the critical surface will be along the wear groove which will be kept largely rust-free by the wheel itself. If protection from rusting is desired for other reasons (e.g., appearance, insurance against bad technical advice, etc.), cadmium plating in the thickness range 5-10 m will provide good protection against corrosion in an interior environment and could be applied to the rails before installation. The drive wheel will work through any such coating within a short time as it creates the wear groove, but should still roll smoothly.

6. COST FACTORS

Some basic cost information was assembled for two approaches to the rail specification.

For the recommended JPL option of hard, precision ground, high carbon steel rails, Thomson rails were considered. Thomson 60 Case LinearRace Shaft in 1" dimater costs $2.07/inch. For our application, the required 800 linear feet would cost enough to qualify for at least a 25% discount, for a net cost per foot of $18.60. The supports for this rail can be purchased or fabricated.

For the option of a soft steel rail, an extreme alternate strategy will be considered. The strategy would be to obtain cold rolled material with dimensional tolerances and surface finish considered suitable for the drive rail, without further machining or grinding of the working surfaces. Cold rolled bar in 1"x2" is available - it is important to obtain bar which is rolled directly to size, and not cut from wider stock, as the cutting my allow relaxation of internal stresses resulting in warpage. Furthermore, it is important that the surface finish be satisfactory. Any grinding operation would also release residual stress and create a warpage problem. This rail would still require mounting fixtures, but the straightness should be satisfactory, alleviating potential mounting problems. Cold rolled steel in 1" thickness will satisfy the thickness specification of (1.000/0.997).

Industrial Metal Supply in Burbank quotes 1018 in the required quantity at $3.81/ft, with a possible quantity discount with sufficient lead time. This would be in 12' bars.

The cost of the two approaches to provide 800' of drive rail would be:

Thomson rail: $14880

Cold rolled bar: $3048.

There could be a possible additional discount on both numbers. The cold rolled bar will require some machining on each piece to square the ends and cut to exact length. Any additional machining for joinery of the bar ends is not included for the bars, while it comes automatically with standard ways of mounting the shaft.

7. NOTES

1. Roark, R.J. and Young, W.C., "Formulas for Stress and Strain," 5th Ed., Table 33, Case 1, McGraw-Hill.
2. Ibid, Table 33, Case 4.
3. Ibid, Chap. 13.
4. "Engineering Properties of Steel," Am. Soc. for Metals, 1982.
5. Roark & Young, "Formulas for Stress and Strain, 5th Ed." Table 33, Case 2, McGraw-Hill.
6. "Metals Handbook," 9th Ed, Vol. 2, Table 61, Am Soc. for Metals. HO2 is a hardness condition formerly known as "half-hard" which would be produced by 20% cold drawing. The yield strength values given are for a 1" diameter rod.
7. "Metals Handbook," 9th Ed, Vol. 2, Am Soc. for Metals Pg 1

8. ADDENDUM: JPL RAIL SPECIFICATIONS

Dimensions (inches) 144.00 by 2.000 by (1.000/0.997)

Tolerances: .xx +- 0.030, .xxx+- 0.010

Material: 4340 steel

Harden to Rockwell C52-55

Machine ends flat and square, do not break edges

Grind 2.0" faces to 32

Chamfer long edges 45 degrees, 0.01-0.04"