ADDENDUM TO MATERIAL SELECTION GUIDELINES
FOR GEOTHERMAL ENERGY UTILIZATION SYSTEMS

Part I: Extension of the Field Experience Data Base
Part II: Proceedings of the Geothermal Engineering
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ABSTRACT

The exploitation of geothermal resources often requires that naturally heated subterranean brines be pumped to the surface from depths of up to 6000 feet underground while minimizing heat losses and maintaining sufficient fluid pressure to prevent boiling. To accomplish this requires the use of downhole brine pumps capable of months of uninterrupted operation.

Significant problems have occurred with pump lineshaft bearings in the geothermal wells.

The objective of this research program was to determine the nature of the problems associated with commonly reported premature failures of downhole lineshaft pump bearings. Using the information gathered, a series of bearing endurance tests was performed on a variety of candidate bearing materials. These tests were accomplished using test rigs specially developed to simulate actual geothermal field conditions and to isolate specific bearing wear problems.

INTRODUCTION

Current geothermal brine pump technology has lagged behind other aspects of geothermal energy exploitation technology. Significant problems have occurred with the bearings used for downhole lineshaft in geothermal wells, particularly in the Imperial Valley area of Southern California. Pumps must be capable of months of uninterrupted operation to supply the unflashed, uncooled brine to power conversion plants. Field testing has so far provided the only research and development capability for improving bearings for geothermal applications. Unfortunately, such field installations provide simultaneously all facets of the overall problem: thermal expansion, shaft elongation, bearing side loads, material inadequacies, misalignments, vibrations, etc. Field tests do not allow isolation of bearing problems. In addition, field tests are often stopped for reasons other than bearing failure so that thorough testing of bearings can rarely be accomplished.

Discussions with personnel from several firms involved in geothermal energy extraction indicated that the life of current SAE 660 bronze bearings (as shown in Fig. 1) is, on the average, less than 45 days. Replacement of bearings is very time consuming and costly (upwards of $30,000 in labor costs alone to replace bearings to the 400-800 foot level). Consequently, it is becoming increasingly important to extend bearing life to, say, 12 or ideally as long as 60 months. This would require materials of extreme resistance to corrosion, erosion and wear.

Fig. 1. Bronze Bearings in Steel Retainers (Upper) and With Bronze Spider (Lower)

Two approaches to the materials issue have been considered, both historically and in this program. The first involves hard, corrosion-resistant materials. It has been held that hard surface bearing materials minimize bearing wear under high loads that can result from shaft misalignment due to machining eccentricities, drilling deviations, or seismic activity. Suspended solid particles, such as the silica sand often present in moderate quantities in open shaft pumps, would be pulverized by the harder surfaces and thus allowed to pass through the thin lubricant film without damaging bearing or shaft surfaces.
The other approach involves the use of softer, more pliable elastomeric bearings with axial gaps to allow passage of foreign particles while at the same time the resiliency of the elastomer would provide a self-aligning bearing, self-aligning in that it would tend to deform slightly in such a way as to decrease the effective misalignment-induced side loads. An example of the familiar "Cutless" bearing is shown in Fig. 2.

The objective of this research program was to determine the nature of the problems associated with commonly reported premature failures of downhole lineshaft pump bearings, especially those operated in geothermal wells in the Imperial Valley area of Southern California. Using the information gathered, a series of bearing endurance tests was performed on a variety of candidate bearing materials in addition to the baseline bronze. These tests were accomplished using test rigs specially developed to simulate actual geothermal field conditions and to isolate specific bearing wear problems. The program proceeded in three phases.

Phase I - Problem Definition, Materials Selection, Bearing Design and Rig Construction

A literature search was carried out to identify state-of-the-art geothermal pump technology. Also, a field survey was undertaken of pump manufacturers and geothermal facilities.

There are two basic lineshaft bearing systems: open shaft or enclosed shaft (Fig. 3). In the open design, the shaft bearings are supported by brass "spider" retainers which center the bearings in the outer column piping at 5- to 10-foot intervals along the length of the shaft. The upflowing pumped liquid acts as the bearing lubricant with no attempt made to seal the bearings from particles suspended in the fluid. The enclosed shaft design overcomes the problem of suspended particles by using an inner tube assembly to enclose the lineshaft and support the bearings.

The pumped fluid travels up the outer annulus. The water or oil lubricant is injected at the top of the shaft enclosure to lubricate all the bearings. Any leakage is of the lubricant into the pumped fluid to prevent contamination.

Structural stiffness in the radial direction is provided in the open shaft case by rigid bearing retainers. Radial support in the enclosed shaft design is provided by the low radial stiffness enclosing tubing which is held in place by stabilizing spiders, either rigid or self-aligning types.

Both bronze and graphite-filled teflon bearings, in combination with both mild and stainless steel shafts from 1" through 1-11/16" diameters, have proven successful over many years in irrigation and cold water well lift pumping. In geothermal wells, notably in Iceland, teflon bearings have been refitted after 10 years' service but at temperatures ranging from about 150° to 285°F (67°-140°C).

In Imperial Valley wells at temperatures ranging from about 355° to 415°F (180°-215°C) bearing life has varied from dismal to a few months (continuously) at most, even with such refinements as clean water lubrication and enclosed shafting with 3-foot bearing spacing. This latter figure did, however, provide some improvement over 5-ft prior spacing. In other instances systems intended to run at speeds of 3600 rpm or more were only able to reach 2200 rpm.
The only failed bearing which was made available for this investigation was one of the teflon (80%)/graphite (20%) type. A set of such bearings was tested for approximately 30 days in an enclosed shaft pump using ambient temperature canal water lubrication. Investigation of the surface deterioration and 0.050”-0.080” diametral wear was accomplished by scanning electron microscopy and spot energy dispersive X-ray analysis. This revealed decomposition of the teflon and foreign particles from 15 to 200 microns in size of iron oxide, zinc, silica and a calcium-rich material in combination with iron and silicon. Since melting of the zinc was noted, it was concluded that bearing surface temperatures of about 800°F had been reached during frictional heating, leading to extrusion of the teflon and closing of the lubrication grooves—a self-worsening type of failure. Bronze bearings that were examined visually displayed severe wear—almost totally through the bearing in one case.

Detailed examination was carried out on shaft journal sections acquired from Imperial Valley well operations. Many of the most badly worn journals exhibited obvious evidence of one-sided wear in the bearing contact area (see lower shaft in Fig. 4). Such wear patterns point to conditions of resonant shaft "whip" during pump operation at full speed. Shaft whip of this type—a precessing motion in addition to rotation—would also account for significant localized bearing overloads as well as the reports of numerous broken shaft couplings and excessive drive power requirements to maintain normal operating speeds.

The top shaft in Fig. 4 was run against a teflon/graphite bearing. Besides the one-sided wear pattern indicating shaft whip, there was also pronounced scoring all around the shaft, suggesting the presence of large, hard abrasive particles probably imbedded, but only partially; in the soft teflon matrix.

An analysis was made to determine if any rotor dynamic problems could be identified which could result in scoring and wear problems. Two cases were investigated: a 1.000” diameter shaft with bearings and supports every 60” and speeds up to 3500 rpm and a 1.687” shaft supported every 120” at 1800 rpm. Shaft critical speeds were calculated for each system and are noted in Table I. These critical speeds assume infinitely (or at least comparatively) stiff bearings. For "soft" bearings the speeds would be reduced.

<table>
<thead>
<tr>
<th>Mode of</th>
<th>1” Dia. Critical Speed (5-ft Span)</th>
<th>1.687” Dia. Critical Speed (10-ft Span)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1330</td>
<td>560</td>
</tr>
<tr>
<td>2</td>
<td>5320</td>
<td>2240</td>
</tr>
<tr>
<td>3</td>
<td>11940</td>
<td>5040</td>
</tr>
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</table>

For the closed shaft system, a natural frequency calculation was made for the 3” IPS piping surrounding and supporting 1” or 1.687” diameter shafts. This showed that the natural frequency of the lube string tubing would be 434 rpm for 10-ft "spider" spacing and 1737 rpm for 5-ft "spider" spacing.

With sleeve bearings shaft "whip" can occur in lightly loaded systems under the influence of well understood principles. It most commonly sets in at and persists above a shaft speed twice the system first critical speed. Severe vibrations can result when operation is at speeds which are at or near integer multiples of natural frequencies and critical speeds. They can be especially bad if the latter two are exact multiples of each other.

Since the operating speed for each shaft design is above the respective first critical, the critical must be encountered during startup and shutdown. How severe the resultant vibration is would depend on balance and damping and how
quickly the critical speed is traversed. Most likely these criticals are severe in that the damping of the sleeve bearing is low, at least because of the low viscosity of hot water. High imbalance from eccentric and/or "kinked" shaft sections would add to the severity.

In a vertical bearing/shaft system no gravity load exists at the supports. Sleeve bearings exhibit no theoretical stiffness at all without load. Without any stiffness, however, the shaft is intuitively free to whirl within the sleeve as dictated by rotor dynamics, limited only by the physical constraint of the sleeve itself.

An analysis of a vertical shaft whirling within a sleeve bearing indeed showed that very high shaft eccentricities would result along with weailstically thin fluid-film thicknesses. In other words, realistic bearing load capacities and stiffnesses were not realized until film thicknesses were very low. Fig. 5 shows the computed load capacity of a 1" diameter shaft versus the minimum film thickness of the lubricating fluid.

![Diagram](image)

**Fig. 5. Load Capacity of 1" Diameter Shaft Whirling at High Eccentricity**

This figure depicts whirling load capacity as distinct from straightforward hydrodynamic load capacity. A comparison is shown for water at 70°F and 360°F and for oil at 360°F. Significantly thicker minimum films are noted for oil than for water at equal loads. It is interesting to note in Fig. 5 also that satisfactory operation with clean, cold water appears possible in contrast to the hot, gritty condition; with particles larger than the minimum film thickness the possibility of bearing scoring damage is very high.

Possible loadings at the bearing supports in the closed shaft system were estimated. For a 1" shaft to deflect at its midspan position enough to touch the inside surface of the supporting 3" Schedule 80 piping, the shaft would produce a load of 160 lbs at each bearing. Based on the data in Fig. 5, the minimum film thicknesses would become 0.0012" for oil at 360°F, 0.0009" for water at 70°F and 0.0002" for water at 360°F. The latter is extremely small compared to a 10 micron particle which requires a minimum film of 0.0004" to pass freely.

Another bearing design noted for its anti-whirl characteristics was evaluated as a potential substitute for the conventional sleeve bearing. The whirling load capacity for a 3-lobe, pre-load type sleeve bearing was also found to be extremely low until very small film thicknesses were reached. It was concluded from this that bearing load capacity maximization is a most important design factor.

The overall conclusions of this preliminary work are presented in Fig. 6. The open shaft design remains a possible solution to brine pump requirements if adequate bearings can be developed. In this program analysis and testing were concentrated on the four parameters circled: bearing materials, design, loads, and particles found in the brine.

![Diagram](image)

**Fig. 6. Problem Sources With Downhole Pump Bearings in Geothermal Wells**

Two test facilities were designed and constructed for the experimental work. The first was a brine immersion test bath, required to evaluate numerous candidate bearing and test rig materials in a static hot brine environment. Three 100-hour brine immersion tests were conducted on a total
of 26 separate material samples including various elastomers, teflons, silicon carbides, silicon nitrides, aluminum oxide, bronze, titanium diboride, and 303 stainless steel. The second facility, shown in Fig. 7, comprised two bearing test rigs set up to simulate the open shaft configuration. Both bearing housings were connected to a common brine circulation system. A pump circulated the brine at approximately 25 gpm through the bearing test sections each consisting of an 8" length of 1" diameter shaft rotated against a test bearing at a constant speed of 1750 rpm. In the wells a flow velocity of approximately 500 fpm produces a Reynolds number (Re) of about 700,000, which ensures fully turbulent flow. The flow velocity in the rigs was about 500 fpm. This yielded a Re of 130,000 - still fully turbulent.

The test bearing was free-floating within the housing, although rotation of the bearing and its retainer was prevented. An air cylinder was used to apply a calibrated side load (up to 200 lbs) to the bearing retainer. A 1.5 hp DC motor drove each shaft. Filters were incorporated to prevent system scale from entering the bearing housings and interfering with test accuracy. Air-buffered stainless steel packing glands and stainless steel bearing retainers were substituted during the course of the program.

Candidate bearing materials were selected, on the one hand, with high hardness (to resist scoring by sand particles), corrosion resistance (to resist the hot, harsh brine environment) as well as known good friction and wear properties. These materials were the datum SAE 660 bronze, titanium diboride coating on titanium, alumina, TC-30 cermets, silicon nitride, and tungsten carbide.

At the opposite extreme were the two experimental high-temperature EPDM elastomers nominated for evaluation in the form of "Cutless" bearings, as shown in Fig. 2. Used in some geothermal wells, 416 stainless steel shafts were nominated for all testing. Two plasma-sprayed coatings, Tribaloy T-700 and an Ni-Cr-Si-B alloy, were applied, however, to some shafts. The purpose was to assess the influence of journal surface hardness on bearing and shaft wear in general and to assess the performance of these two coatings in particular.

Phase II - Short-Term Material Screening Tests

The purposes of this testing were primarily to establish a baseline wear rate for the datum SAE 660 bronze bearings on 416 stainless steel shafts and to develop comparative data on all other candidate pairs. The leading candidates would then be endurance tested in Phase III. A secondary purpose of the initial short-term testing was debugging of the rig and test procedures. Two sets of test conditions were, of necessity, established. The first simulated Imperial Valley well conditions. The second was derived to pro-
latter conditions produced identical excellent results. The two sets of parameters are listed in Table II. Flat specimens of the elastomers were subjected to 360°F brine statically for 1000 hours each in the immersion tester and, apart from a slight "set", were in excellent condition.

<table>
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<th>Table II. Brine Test Conditions</th>
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<tr>
<td>Salinity, ppm NaCl</td>
</tr>
<tr>
<td>Suspended</td>
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<tr>
<td>Temperature, °F</td>
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<tr>
<td>Brine Pressure, psig</td>
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<td>Bearing Load, lbf</td>
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The results of the matrix testing, nominally 100 hours on each candidate combination, are shown in Table III. On the basis of that performance the four combinations in Table IV were selected for long-term endurance testing in Phase III.

<table>
<thead>
<tr>
<th>Table III. Phase II - Test Matrix</th>
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<tbody>
<tr>
<td>Bearing/Shaft</td>
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<tr>
<td>Bronze</td>
</tr>
<tr>
<td>TiB₂</td>
</tr>
<tr>
<td>Alumina</td>
</tr>
<tr>
<td>SS</td>
</tr>
<tr>
<td>WC</td>
</tr>
<tr>
<td>EPDM Y267</td>
</tr>
<tr>
<td>EPDM 317-II</td>
</tr>
<tr>
<td>Si₃N₄</td>
</tr>
<tr>
<td>Si₃N₄</td>
</tr>
</tbody>
</table>

The test results for the four final combinations are shown in Figs. 8 and 9.

**CONCLUSIONS AND RECOMMENDATIONS**

It was concluded that maximized bearing load capacity and system rotor dynamics are perhaps the most important aspects of lineshaft design in addition to bearing material choices for service at temperatures above 300°F (150°C). It cannot be over-stressed that bearing design and system rotor dynamics are intimately interrelated. At least until patterns of satisfactory experience become apparent, each individual application should be engineered separately.

Since 1-11/16" shaft systems have operated at their design speeds of 1800 rpm whereas the 1" systems have had difficulty rising much above that speed, and since bearing load capacity varies directly with (diameter)³, it would seem wise to concentrate on the larger shaft size.

Since bearing load capacity varies inversely as (clearance)³ a smaller clearance is clearly beneficial. However, lower clearances also increase heat generation, reducing lubricant viscosity and load capacity. Careful optimization of clearance is therefore required. As load capacity increases linearly with bearing length, this may provide an additional approach.

For equal bearing support spacing, the rotor dynamics of the larger shaft system would be better than the smaller one. Ideally, the shaft...
George W. Hosang and Alvin R. Stetson

system first critical frequency should be raised above the design speed. This would seem easier to accomplish with the larger shaft size. Caution would suggest that the new critical not be exactly twice the operating speed.

From the materials standpoint the inescapable conclusion is that "harder is better". The tungsten carbide was the hardest of the bearing materials and significantly out-performed all others. Likewise, the Ni-Cr-Si-B shaft coating was harder and wore less than the Tribaloy T-700 uncoated 416.

While the experimental EPDM elastomers were unsuitable as bearing materials at the elevated temperatures, they endured long-term static exposure. Because the tungsten carbide apparently has, thin-walled though it was and in contrast to the sintered silicon nitride, sufficient tensile strength and coefficient of thermal expansion, an interesting new design possibility arises. This would comprise a thin cylindrical steel shell with the same outer diameter as the standard bearing, the thin-walled tungsten carbide bearing as the inner member and the space between the two filled with the high-temperature EPDM elastomer. A sketch is shown in Fig. 10.

The new bearing would offer two additional benefits: a misalignment capability and rotor dynamic damping. With the misalignment capability it might also be possible to reduce the bearing clearance slightly and enhance the load capacity.
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Fig. 10. Proposed New Lineshaft Bearing Design