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ACTIVE MAGNETIC BEARINGS - AS APPLIED TO CENTRIFUGAL PUMPS

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ABSTRACT

Application of magnetic bearings to boiler feed pumps presents various attractive features, such as longer bearing life, lower maintenance costs, and improved operability through control of the rotordynamics.
Magnetic bearings have been fitted to an eight-stage, 600 hp boiler feed pump, which generates 2600 ft of head at 680 gpm and 3560 rpm. In addition to the varied and severe operating environment in steady state operation of this pump in a power plant, it is also subjected to transient loads during frequent starts and stops. These loads can now be measured by the in-built instrumentation of the magnetic bearings. The pump was factory tested, including the adjustment (tuning) of magnetic bearings. Following site installation, a follow-up bearing tune-up was performed, and pump transient response testing was conducted. The bearing response was completely satisfactory, ensuring trouble-free pump operation even in the range of reduced load. The experience gained so far through design and testing proves feasibility of magnetic bearings for boiler feed pumps, which sets the stage for application of even higher energy centrifugal pumps equipped with magnetic bearings.

INTRODUCTION

Utilization of magnetic bearings in pumps is a natural development considering the successes of these bearings in large (35,000 Hp) centrifugal compressors. Some notable capabilities of magnetic bearings, namely (1),

- o Theoretically infinite bearing life potential,
- o Control of rotordynamical behavior,
- o Rapid response to load variations so as to maintain shaft position,
- o Elimination of oil system,
- o Measurement of bearing static and dynamic loads as a diagnostic monitoring tool

are becoming more sought after for various types of heavy machinery. In particular, for pumps of significant power levels, the elimination of lubrication systems that would otherwise be needed for conventional bearings and the attendant reduction of maintenance are benefits that arise from claiming these capabilities.

It is the purpose of this paper to show how these benefits of magnetic bearings were obtained for a 600 HP boiler feed pump. Such pumps often operate with close internal clearances and flexible shafts, and the ability to adjust the stiffness and damping of magnetic bearings can improve rotordynamics. Increased availability and reduction in operating and maintenance costs were projected, leading to a favorable evaluation (2).

The design of a multistage boiler feed pump allows sufficient clearance for shaft flexure and thermal distortion and minimizes clearances to increase the efficiency of the impeller rings and the shaft sealing system.

Ideally the bearings would be mounted closer together than current practice dictates, allowing a smaller tolerance on shaft movement. If submerged magnetic bearings were used, this could be accomplished by eliminating the outboard (nondrive end) seal and putting the inboard bearing between the first-stage impeller and the inboard seals. The shaft could be held within 0.003 in (0.076 mm) at all times by the magnetic bearings (3).

There are phenomena unique to pumps that could affect magnetic bearing performance. These include stall effects caused by nonuniform flow interruptions or reversals that occur in a pump impeller and diffuser when operating at reduced flow rates (4,5). These effects introduce unsteady, random, radial and axial loads that are often accompanied and exacerbated by cavitating flow; wear of the impeller neck rings (or adjacent casing rings) which leads to loss of stabilization and, therefore, to adverse changes in rotordynamical behavior. The above phenomena and the corresponding mechanical responses become more evident and critical in pumps of high energy level; i.e., those of more concentrated and higher power. It was therefore important to gain further insight into magnetic bearing behavior in such pumps in general, to access their ability to handle these otherwise detrimental situations, and to measure the actual instantaneous loads that might be encountered.

Since the current in a magnetic bearing is related to the load, readings of current vs time, speed and pump flow rate provided the needed insight into the stall effects. Scaling of the loads involved is fairly reliable and enables one to make a more informed estimate of the loads for a still-higher-power submerged-magnetic-bearing (canned) pump. Also, the ability of the magnetic bearings to cope with the rotordynamic uncertainties mentioned above has been revealed and could be analyzed for the future submerged-bearing application.

MAGNETIC BEARINGS

Figure 1 shows both active magnetic bearing assemblies and their relation to the boiler feed pump. The magnetic bearings for the boiler feed pump require a radial load capacity of 280 lbs (127 kg) steady state and an additional 280 lbs (127 kg) for transient conditions. For the magnetic thrust bearings, the required load capacities are 1000 lbs (454 kg) steady state and an additional 1000 lbs (454 kg) for transient conditions. In the design of active magnetic bearing systems, it is prudent to include load capacity margins to accommodate process-induced surge loads, abnormal imbalance or hydrodynamic loads.

Through the use of standardized magnetic bearing designs, which provided further margins over conventional practice, the resulting magnetic radial bearings provide a maximum steady state load capacity of 800 lbs (364 kg) and will modulate a force of 400 lbs (181 kg) at a frequency of 60 Hz (boiler feed pump running speed). The magnetic thrust bearing provides a maximum steady state load capacity of 4000 lbs (1814 kg) and will modulate a force of 1000 lbs (454) kg) at a frequency of 60 Hz.

The magnetic bearing assembly for the nondrive end of the boiler feed pump is shown on Figure 2. The completed stator assembly consists of two magnetic thrust bearing stators and an axial position sensor, a magnetic radial bearing stator (consisting of four electromagnets) and radial position sensor, a speed pick-up, and an auxiliary (emergency back-up) bearing. Additionally, the bearing windings are equipped with resistive temperature devices (RTDs) which can be used to monitor operating temperatures. The entire assembly provides the radial/thrust forces for the nondrive end of the pump. The drive end radial bearing is similar to that of the non-drive end.

The radial forces are transferred from the bearing assembly to the shaft via two 2.5 in (63.5 mm) by 6 in. (152.4 mm) diameter laminated silicon-steel rotors fitted as sleeves at each end of he shaft. The thrust forces are transmitted to the shaft through an interference-fitted 1 in (25.4 mm) by 11.25 in (285.8 mm) diameter disc. The shaft position is controlled by the magnetic bearing system which uses two inductive radial position sensors and one inductive axial position sensor. The radial position sensors (referred to as collocated sensors) are adjacent to the radial bearing lamination stacks and the axial position sensor is at the end of the machine. The radial sensors are designed to minimize rotor runout or other imperfections that would cause position sensor signal noise.

The control cabinet is self-contained with power conversion, signal processing, operation logic and alarm/trip monitoring of critical functions necessary for the active magnetic system. The air-cooled cabinet measures 52 in. (1321 mm) high, 21 in (533 mm) wide, and 20 in 508 mm) deep. The cabinet power supply requires 208 VAC, 3 phase (1.4 KVA) as input and is equipped with a battery backup which will power the system for ten minutes.

After ten minutes, the system will maintain operation, but the bearing load capacity may be diminished. The magnetic radial bearings are powered with eight 120V/15A power amplifiers and the magnetic thrust bearing is powered with two 120V/30A power amplifiers. Should the power to the magnetic bearings be lost, the reserve power is provided by the battery backup system. The auxiliary bearing at the nondrive end of the machine is a duplexed pair of angular-contact ball bearings. The bearing provides radial and axial load capability if the pump were required to coast to a stop because of a loss of both primary and backup power to the magnetic bearings. The auxiliary bearing at the drive end of the machine is a Conrad-type single radial bearing. The magnetic air gap for the radial bearings is 0.020 in (0.51mm), and the thrust bearing magnetic air gap is 0.028 in (0.71 mm). The auxiliary bearings are designed to prevent the shaft from moving past half the magnetic air gap clearance, hence the auxiliary bearing air gap for the radial and axial bearings is 0.010 in (0.25 mm) and 0.014 (0.36 mm) respectively.

With the use of auxiliary bearings, protection is assured against sustained loss of magnetic levitation as well as against any upset condition that might occur as a consequence of pump operation in the power plant environment. In addition, rotor bushings and impeller rings provide additional rotor damping, and further enhance the overall system reliability.

ROTOR DYNAMICS

The pump response to unbalance excitation was investigated using the recently developed AMB forced response computer program (6). This new computer code can include the influence of sensor axial location in addition to impeller and seal ring cross-coupling and direct stiffness and damping. The response of the pump rotor for a 0.5 oz-in unbalance placed at midspan is given in Table 1 for normal design conditions considering both dry and wet conditions. This level of unbalance corresponds to the API 4W/N balance specification. The results for dry running conditions indicate little influence of sensor position when operating at or near design speed. A greater influence of sensor position is indicated when operation is near the critical speed. The largest midspan response is for outboard sensor, smallest for inboard sensor. The predicted vibration levels at design speed are insensitive to sensor position for the normal wet conditions. The midspan response is indicated to be approximately 3-4 times the sensor location vibration. The forced response analysis predicts that the pump rotor could withstand just over 150 times this unbalance force and still have a response amplitude of less than 1.2 mil at the sensors and only 3.3 mils at midspan. For reference, the 0.5 oz-in unbalance corresponds to a force of 11.5 lbf for a rotor speed of 3560 rpm.

The dry first critical speed was predicted to be 1605 rpm, with the second critical at a speed of 5798 rpm. When the wet condition is considered, the forced response versus speed indicates that the first critical speed (5900) is critically damped as shown in Figure (3).

The stability of the pump was calculated considering various conditions of pre-swirl at the entrance to the front rings, center and outer bushings. A summary of the results are given in Table 2. For a zero preswirl, the pump 1st critical speed was predicted to be stable with a log decrement of 9.13. For 50% preswirl, the rotor first critical was also predicted to be stable with a log decrement of 15.5. If the impeller coefficients are not considered the 50% preswirl case log decrement reduces to 14.2 . Actual test stand operation gave no indication of instability. The calculation of the actual seal inlet pre-swirl and the eccentric dynamic seal coefficients is not a proven capability for actual pump operating conditions. The results for this pump are not very sensitive to this influence.

The conclusions of the rotor dynamics analysis indicate that the pump design is acceptable for forced response excitation. The pump design, as built, can resist strong unbalance caused by hydraulic excitation forces which is much more important in pump design than the possibility of small self-excited vibration. The pump is predicted to have very good stability margins and may only respond to non-synchronous hydraulic excitation during off-design operation. The pump will not experience self-excited subsynchronous vibration during operation.

FACTORY AND ON-SITE TESTING

Prior to factory testing, the magnetic bearing controls had to be tuned to the characteristics of the shaft, with the pump in the wet condition. This took significantly less time than had been experienced on other projects, and was completed in under a week, by a team of two.

The factory tests, with the exception of an unplanned shutdown onto the battery backup as a result of a thunderstorm, were uneventful. It was noticed, however, that one of the motor-end journal bearing poles was showing a fairly high load. This was attributed to misalignment due to the temporary nature of the baseplate alignment for the test, and it was agreed to monitor this further on-site.

Figure 4 shows the pump installed in the power plant. The site start-up was uneventful, with the exception that the high bearing load noted above reappeared on the same motor-end bearing, but on a different pole. This seemed to confirm that it was an alignment load. By adjusting the coupling to be a little more flexible, the load was significantly reduced.

High loads were noted during the run up to full speed which took only about 2 seconds. These levels needed to be understood better,and a second retuning with test was performed, monitoring bearing loads over all representative running conditions. This second tuning took place 4 months later together with testing of the pump through start-up, shutdown and part load operations. During this testing, no indications of hydraulic and/or rotordynamic instability were observed. Bearing currents and shaft displacements were stored on a high speed "Adre" recorder for detailed analysis. The key results of this operation were that after some tuning adjustments, the pump rotor was held both axially and radially within about 1 mil during all transient and steady state operations.

Figures 5 (before tuning) and 6 (after tuning) show two traces of the shaft radial displacement as a function of time during the run up to full speed. Figure 5 shows displacements of the order of 5 mil, where the bearing did not fully control the shaft displacement. Figure 6 shows that the shaft is held geometrically to within 1 mil during the whole transient. Provided that the operating transients are all covered in the specification, this approach should ensure satisfactory lifetime operation.

Conventional bearings can often absorb higher than design loads, accompanied with increased wear rate, but not necessary leading to immediate failure. However, a magnetic bearing that reaches its design point where the electromagnet is saturated, will not resist any additional load and the shaft will make contact with whatever backup bearing device is provided. Any additional load will now be transferred to this auxiliary support system, and the shaft will be about 10 mil out of alignment on one or more axes.

For the reasons stated above it is absolutely essential for any prototype equipment running on magnetic bearings that this procedure of running the machine through all its operation transients and documenting the shaft response is carried out before putting the equipment into service.

Checking of the power consumption of the magnetic bearing feed pump against a conventional bearing machine purchased at the same time, and with the same rotor design showed a slight reduction in power consumption for the same feedwater flow. This was fairly consistent over a long time period.

Operator Acceptance - After an initial period of enthusiastic but cautious use of the pump, the operators' confidence has increased to the point where it is being operated as confidently as any other similar equipment.

Availability Experience - From studies referenced above, expectations for availability have been developed. To date, there have been no events on site that would indicate that these targets cannot be met even with the prototype unit. However, a longer time frame of several years will be needed to fully justify such high availability claims.

CONCLUSIONS

Boiler feed pumps in an electric utility may benefit from the expected life and low maintenance advantages of magnetic bearings. These pumps can provide a comprehensive platform for evaluating the feasibility of magnet bearing technology under a variety of severe operating conditions that can impose strong and random loads on the rotating elements.

Successful in-plant and on-site demonstrations of a 600 hp, magnetic-bearing-equipped, eight-stage volute-type boiler feed pump were conducted. No effects on pump operation were experienced that could be attributed to hydraulic and/or rotordynamic instability. From the standpoint of the user, experience gained so far has demonstrated that a pump equipped with magnetic bearings can be characterized as follows:

1. The bearings operate satisfactorily.
2. Equipment is reliable.
3. The pump is operable.
4. Provides new data, not available in the past: bearing loads and positions.
5. Requires less attention.
6. Requires less maintenance.

By monitoring the operation of the 600 hp pump, which has the magnetic bearings mounted external to the pump casing and running in air, it should be possible to gain the rotordynamical insights needed to design improved, higher-power machines with these bearings submerged in fluid within the interior of the casing. An order of magnitude more power, e.g. 10,000 hp, should be the next step.

A one-year test, monitoring and benefits evaluation program on two side-by-side new pumps - one with magnetic bearings; the other, with conventional bearings - is continuing through the remainder of 1991. To date the pump is fulfilling all the expectations that accompanied the initiation of the project, and provides a sound basis for future developments.

Specifically, we expect to develop availability estimates, spares requirements, running costs, and design specification requirements for future utility applications. We expect to have all but the availability estimates completed at the end of the first year of operation, that is by November 1991. The availability data will need to be acquired over several years of operation to be statistically significant, and until we have that data the experience in other industries is the only yardstick that we have. But that experience appears to be very encouraging and could provide an alternate technology to the utility industry resulting in lower forced outages, higher availability and more efficient operations.

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TABLE 1 RESPONSE TO 0.5 oz-in at MIDSPAN (at design speed of 3560 RPM)

Condition	Thrust end (mil-pp)	Midspan (mil-pp)	Coupling End (mil-pp)
a. dry			
collocated sensor	.010	.384 (21.0 at N-cr)	.056
outboard sensor	.062	.380 (42.0 at N-cr)	.056
inboard sensor	.064	.392 (14.4 at N-cr)	.038
b. wet (c = 0.010 in.- wear rings and bushings)			
collocated sensor	.006	.022	.008
outboard sensor	.006	.022	.008

TABLE 2 STABILITY RESULTS SUMMARY for PUMP ROTOR DAMPED CRITICAL SPEEDS

Condition	Eigenvalue real part (sec^{-1})	Eigenvalue damped critical (cpm)	log decrement
a. dry No influence of liquid seals	-0.76	1830	0.03
	-145.	3807	1.49
	-19.5	6462	0.18
b. wet (zero preswirl)	-323.5	2126	9.13
	-118.7	5549	1.28
	-198.7	6276	1.89
c. wet (50% preswirl)	-349.1	1350	15.5
	-118.2	5552	1.28
	-197.2	6256	1.89
d. wet (50% preswirl, no impeller characteristics)	-677.3	2882	14.2
	-120.2	5541	1.30
	-197.3	6323	1.88

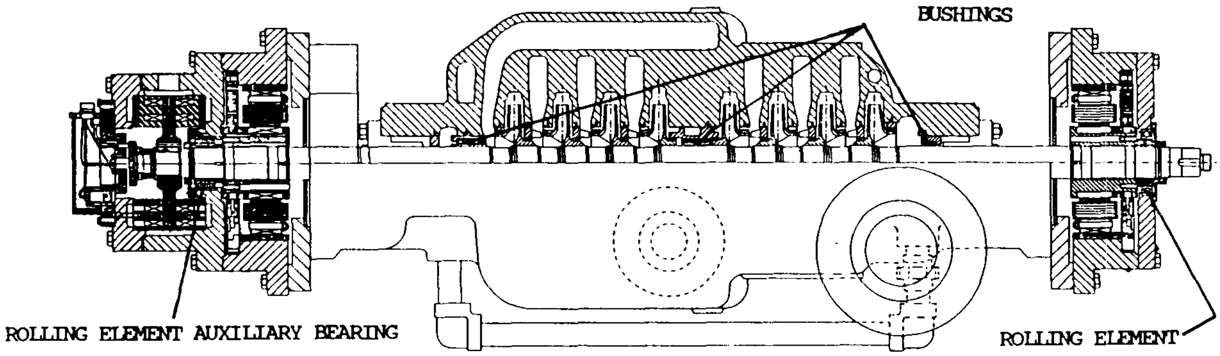

FIGURE 1 MULTISTAGE VOLUTE PUMP FITTED WITH MAGNETIC BEARINGS. I-R PUMP MODEL 4X10DA-8 WITH RADIAL BEARING ON DRIVE END AND RADIAL AND AXIAL BEARINGS ON NONDRIVE END.

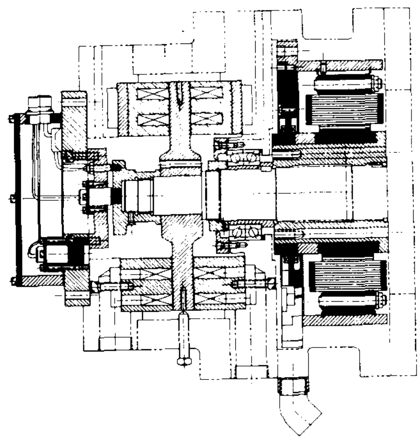

FIGURE 2 RADIAL AND AXIAL MAGNETIC BEARING ON NONDRIVE END OF PUMP.

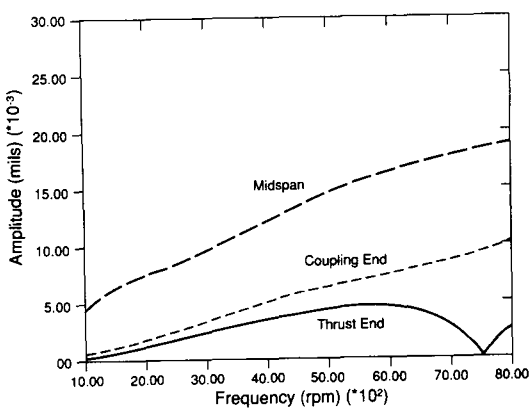

FIGURE 3 RESPONSE FOR 0.5 OZ-IN AT MIDSPAN INCLUDING WEAR RINGS, BUSHINGS, AND NON-COLLOCATED AMB SENSORS.

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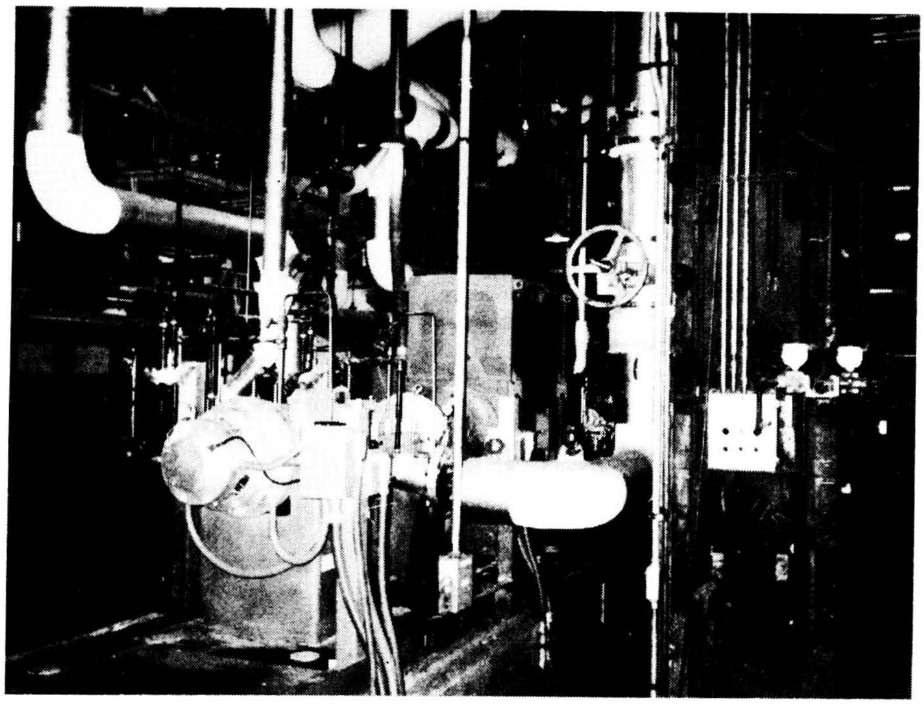

FIGURE 4 MULTI-STAGE BOILER FEED PUMP WITH MAGNETIC BEARINGS INSTALLED AT NEW YORK STATE ELECTRIC AND GAS POWER PLANT

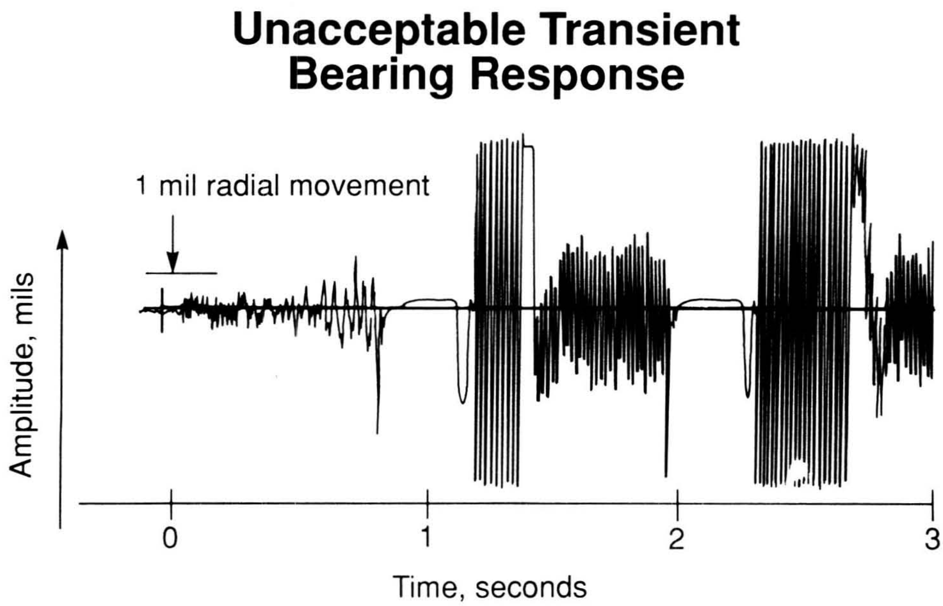

FIGURE 5 UNACCEPTABLE TRANSIENT BEARING RESPONSE

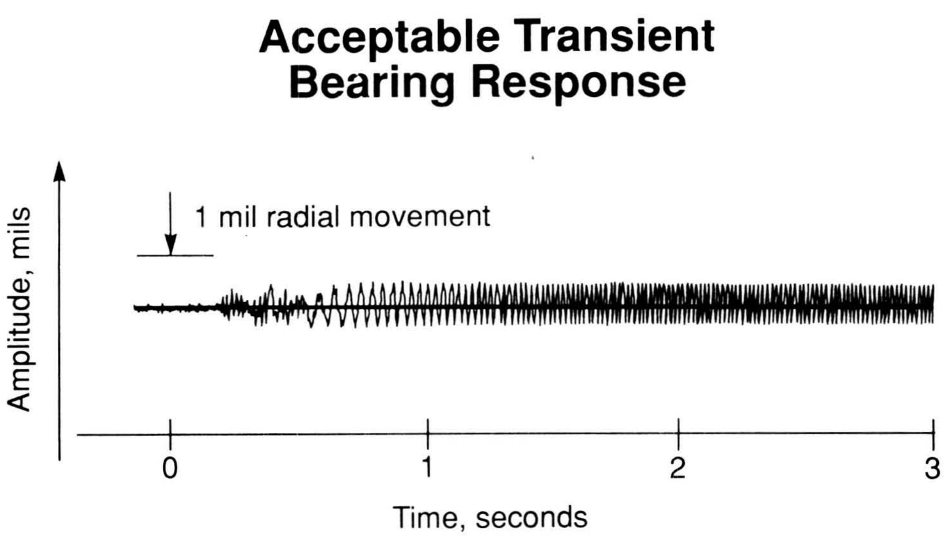

FIGURE 6 ACCEPTABLE TRANSIENT BEARING RESPONSE