THE POWER

LUBRICANT BOOSTS BEA

A highly sophisticated and unique journal bearing technology, only very recently developed, has been demonstrated to operate up to 30,000 rpm, or 3 million DN. That's 50 percent faster than today's conventional journal bearings, which rarely push beyond 2 million DN. (DN is an expression of rotating speed, calculated by multiplying the rotating shaft's diameter in millimeters by its speed in rpm. So a 100 mm shaft rotating at 30,000 rpm would be moving at 3 million DN.)

What accounts for this enormous leap in speed? It's a revolutionary design called a Powder-Lubricated, Quasi-Hydrodynamic (PLQH) Journal Bearing. And, as its name reflects, it depends on a completely new form of lubrication: powder.

It is based on 15 years of fundamental research conducted by Dr. Hooshang Heshmat, president and technical director of Mohawk Innovative Technology Inc., who introduced the Quasi-Hydrodynamic Theory. This theory states that powders can be caused to behave like liquid lubricants under appropriate geometries and pressures, thus generating a hydrodynamic-like pressure. This causes the bearing to be lubricated and creates a load capacity.

This is a truly revolutionary concept that has only recently been demonstrated, and shown to have exceptional performance capabilities. There is no other such technology in the world; it is one of a kind, Heshmat says.

The currently targeted application of this technology is an auxiliary back-up bearing, especially with magnetic bearing supported systems, such as ground- and space-based flywheel energy storage systems, gas turbine engines and advanced auxiliary and integrated power unit systems.

Although magnetic bearings have many advantages, they require self-contained back-up or auxiliary bearings for protection during a failure or severe transient. Most conventional auxiliary back-up bearings have been of the rolling element type. Rotors dropped onto these bearings are susceptible to violent backward whirl, which generates large centrifugal bearing loads, and leads to severe bearing wear and radio deterioration.

The uniqueness of the PLQH bearing lies in its application of dry triboparticulate powders to provide a long-life, low-power-loss

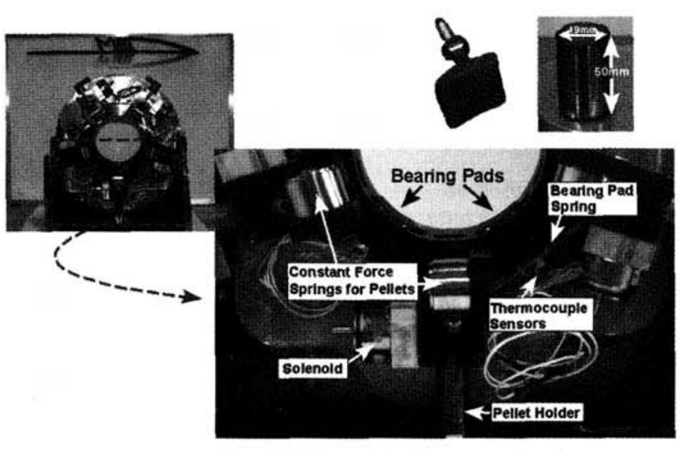

Fig. 1 Inside the bearing, MoS_2 pellets (above right) supply lubrication to the pads (above center).

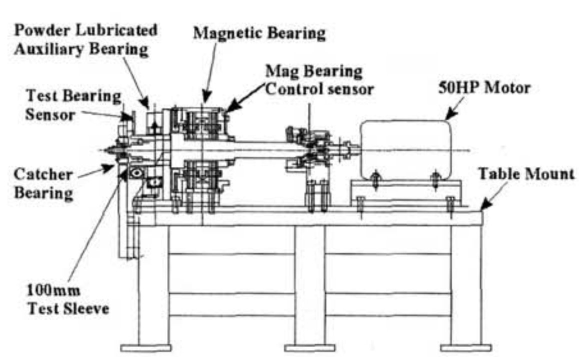

Fig. 2 Schematic of the PLQH test bearing set-up

OF POWDER

RING SPEED 50 PERCENT

auxiliary bearing. The introduction of powders between the bearing pad and rotating shaft generates a quasi-hydrodynamic film that separates the shaft and bearing, minimizing wear and transferring a significant portion of heat away from the contact zone. And since no moving parts are involved, the PLQH bearing design approach also eliminates the problems — backward whirl, instability, and extremely short life — associated with conventional back-up bearings.

A series of experiments conducted by Mohawk Innovative Technology Inc. (MiTi) has demonstrated the basic feasibility of developing a PLQH bearing for advanced rotating machinery. Based on the demonstrated operation of a powder-lubricated journal bearing at speeds to 3 million DN, these may be the only bearings capable of meeting and completing the ever-demanding tribological goals of a solid lubrication system for extreme environments.

Inside the Bearing

The two major technology components for this system are the powder pelletized lubricant delivery system and the compliantly mounted slider type journal bearing (Figure 1).

The self-contained PLQH journal bearing consists of five equally spaced bearing pads with a bearing diameter of 100 mm (3.939 inches) and projected pad area of 682 sq. mm (1.058 sq. inches) with bearing diametrical clearances of 0.1 and 0.2 mm (0.004 and 0.008 inch). The pads were preloaded at 60 percent from the leading edge and were attached to the bearing cartridge via sets of adjustable compliant pad mounts. These were designed to provide radial, pitch and roll stiffness.

The test bearing cartridge and compliant elements were made of Inconel 718, a nickel-base alloy. The 100 mm test journal was made from M50, and the bearing pads were made from titanium carbide cermet. The test journal and pad surfaces were ground after proper heat treatment and then lapped to obtain a surface finish better than 0.1 micron Root Mean Square.

The molybdenum disulfide (MoS_2) powder pellets, made with a proprietary MiTi binder, were placed in cylindrical sleeves attached to constant force springs. These were further controlled by

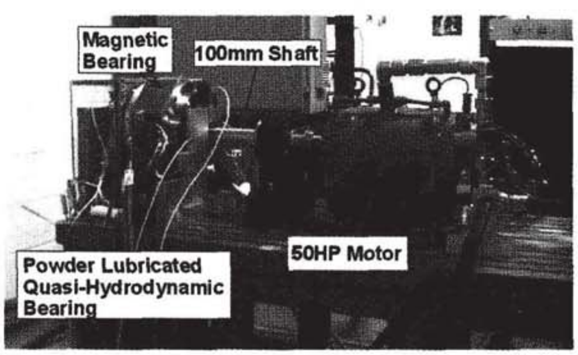

Fig. 3 Final set-up of the PLQH Journal Bearing

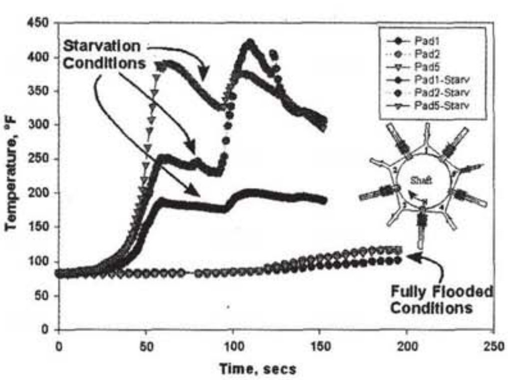

Fig. 4 Temperature change of bearing pads 1, 2 and 5 under fully flooded and starvation test conditions at 30,000 rpm

an electromechanical system employing solenoids. A test rig was built with a rotor supported at one end by an active magnetic bearing (AMB) with the PLQH journal bearing, and at the other end by a deep groove ball bearing. The rig incorporates an electric drive motor, shaft, magnetic bearing and the powder-lubricated quasi-hydrodynamic bearing. (Figure 2 shows a schematic of the set-up.)

The rotor is 556 mm (21.9 inches) long. The active magnetic bearing has a 25.4 mm (1 inch)-thick laminated stack, a 102 mm (4 inch) rotor OD and a typical 2-axis homopolar bearing with a total of 8 pole face configurations. The rotor weight at the active magnetic bearing was 15.9 kilograms (35 pounds). (Figure 3 shows the test rig hardware assembly mounted on a 25.4 mm (1 inch)-thick steel base plate.)

The working principle of the PLQH system was that as the shaft contacted the bearing pads due to AMB deactivation, an electrical signal caused the solenoids to deactivate. This releases the powder pellets to the shaft and bearing pad interface. Conversely, as the shaft was again levitated there was no longer an electrical signal and this reactivated the solenoids thus causing the powder pellets to retract from the bearing pad interface. As a result, the MoS_2 lubricant was only applied at the bearing interface as and when required.

A total of 10 thermocouples were used to continuously measure the temperature change of the pads during testing. The thermocouples were installed in the leading and trailing edges of the five axially centered pads. Two displacement sensors were located behind the magnetic bearing and these were used to control the shaft position. Two displacement sensors were also located near the PLQH bearing and these were used to continuously monitor and measure the shaft motion.

The signals from these sensors were output to both a dual channel oscilloscope, that measured the motion of the shaft during testing, and a dynamic signal analyzer (Fast Fourier Transformer, or FFT) used to measure the frequency content.

The rotor was driven at the ball bearing end by a variable speed motor. The maximum rotor speed was more than 30,000 rpm and the PLQH test model was tested to this high speed several times without failure.

Test Sequences

The prototype bearing testing was accomplished in four distinct phases:

- short duration testing at low speeds ranging from 2,000 to 5,000 rpm;
- impact or rotor drop testing (transient shock simulating magnetic bearing failure) at 15,000 rpm;
- high-speed testing at 30,000 rpm; and
- lubricant starvation testing, also at 30,000 rpm.

For each of the tests the following data was recorded: pad temperatures, shaft speed, bearing load, motor power loss and the rotor vibrational displacement during coast-down via an FFT waterfall plot. The data was recorded on two data acquisition systems, a PC-based LabView system and a digital tape recorder. At the end of each test the bearing and journal surface were inspected for wear and film transfer. Surface roughness measurements were conducted on selected bearing pads and the shaft.

Results from Experimental Rig

Figure 4 shows the temperature increase of the bearing pads during the high-speed and starvation tests at 30,000 rpm on the fully lubricated PLQH bearing. For the fully flooded conditions the temperature increase for pad 1 was lowest and in the range of 90 degrees F. For pads 2 and 5 the temperature increase was in the range of 115 F. However, under starvation conditions the temperature increase for pad 1 was in the range of 180 to 190 F, pad 2 was in the range of 250 to 400 F, and pad 5 was in the range of 400 F.

Figure 5 shows a "waterfall plot" of a high-speed

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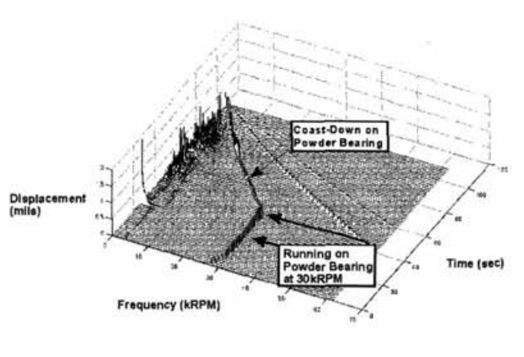

Fig. 5 Rotor displacement during operation at 30,000 rpm (high-speed test, front horizontal displacement)

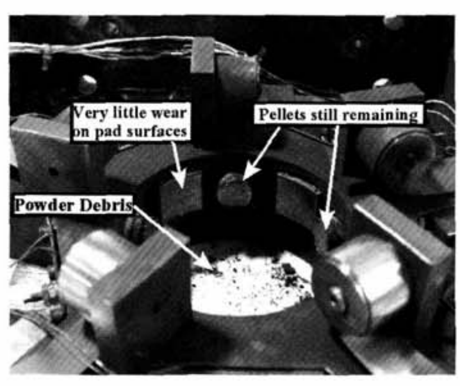

Fig. 6 Inspection of bearing test components after testing

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test at 30,000 rpm. This is a compilation of instantaneous FFTs (Frequency Spectra) taken at discrete time intervals, showing an overall view of the test process with respect to the front horizontal displacement of the PLQH bearing. The waterfall plot shows an initial smooth run-up on the magnetic bearing, and as the shaft was set down to run on the PLQH bearing only a small amount of low-frequency motion was observed. Under starvation conditions, once the shaft was running on the PLQH bearing, there was clearly some low-frequency motion, associated with the first rigid body mode. Coastdown was done on the PLQH bearing and there was a significant amount of low frequency motion.

Final evaluation was done on the bearing pads and pellets after six hours of accumulated testing (Figure 6). From initial surface observations it appeared that there was no significant wear on any of the compliantly mounted slider pads. The MoS_2 powder had formed a protective surface film at the contact interface. The MoS_2 powder pellets were also intact and showed good structural integrity throughout the duration of the tests.

Record Speeds Reached

The acquired test data spanned the range of expected operating test conditions including lubricant feed rate, bearing temperatures and operational dynamic performance. This has clearly been demonstrated by the resistance to wear following six hours of accumulated testing. Thermal stability was achieved at all load and speed combinations. An intermediate layer of powder adhered to the shaft and pad surfaces, resulting in wear protection to the bearing mating surfaces.

For the very first time a powder-lubricated journal bearing has been demonstrated to operate at speeds equivalent to 3 million DN. This is a one-of-a-kind bearing which has significant future implications on auxiliary bearing technology. ∎

ACKNOWLEDGMENTS

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Rita Kaur, PhD, is a materials scientist and tribologist with Mohawk Innovative Technology Inc. For more information about powder lubrication and the Quasi-Hydrodynamic Theory of Lubrication, she can be reached at MiTi, 1037 Watervliet-Shaker Road, Albany, NY 12205. Phone: (518) 862-4290, ext. 16. Fax: (518) 862-4293.