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T E C H N I C A L P A P E R

Design and Testing of New NT Tilt Pad Ring Oil Lubricated Liner with Emphasis on Application Testing and Performance Evaluation as Compared to the Conventional DIN Ring Oil Lubricated Liner

By

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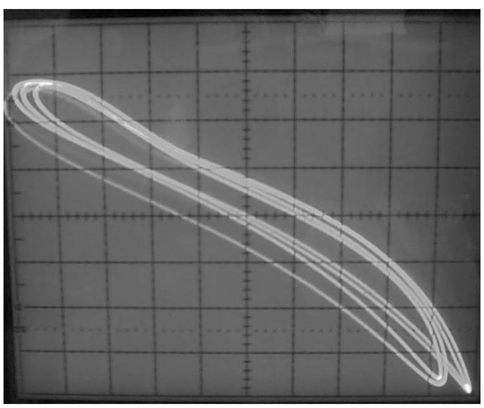
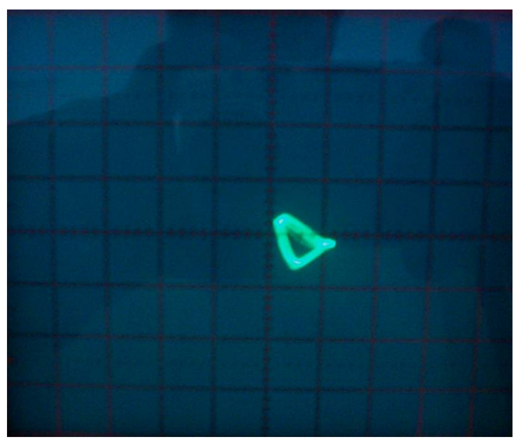


Standard DIN Liner



NT DIN Liner

Abstract

This paper discusses the design, development and test results for a new Oil Ring Lubricated bearing suitable for replacing the conventional DIN liner. The new bearing design can also be applied to OEM Oil Ring Lubricated bearing designs. The new design utilizes a tilting pad with an embedded pocket. The embedded pocket begins at the leading edge of the pad and extends about one half of the pad arc length. The pocket depth, width and arc length are design variables, which in addition to the usual tilt pad design variables can be adjusted by the bearing designer to optimize the performance of the rotor bearing system. The development program included manufacture and testing of a conventional DIN liner so that a direct comparison between the conventional DIN liner bearing with the new NT Tilt Pad bearing liner is presented.

The test results show that the conventional DIN liner exhibits a low frequency pulsation, on the order of 0.5 sec to 1.0 sec in frequency, which appears to be linked with the degree of starvation of the DIN liner. Photographs of the journal orbit clearly show the pulsing orbit of the conventional DIN liner. The conventional DIN liner orbit photographs also show the large elliptical orbit produced by the DIN liner bearing. The conventional DIN liner, due to the low horizontal stiffness and damping, develops this large elliptical orbit. The new NT Tilt pad bearing was built to have the exact same length, and diameter as the conventional DIN liner. The NT bearing was tested in the same test machine, at the same speeds and load. By comparison, the new NT Tilt pad bearing journal orbit does not exhibit the pulsating orbit as can be seen by examining the journal orbit photographs. The new NT bearing produces a very small "nearly" circular orbit that indicates that a substantial amount of damping is produced in the bearing fluid film. Furthermore, response to unbalance tests have shown that increasing the residual unbalance by a factor of three times, as compared to the conventional DIN liner residual unbalance level, produced no measurable increase in the rotor orbit size or orientation. This remarkable performance is attributed to the design and performance characteristics inherent in the new NT bearing technology.

INTRODUCTION

There are many types of rotating machines such as pumps, electric motors, fans, generators, blowers, turbines, and gearboxes, which operate using two axial grooves, Oil Ring Lubricated bearings. These bearings receive their lubrication by means of an "oil ring" which surrounds the shaft journal and hangs into the oil reservoir. When the shaft is rotated, the oil ring rotates and drags oil up from the sump and deposits a portion of the adhering oil from the ring onto the journal to lubricate the bearing. Little development and analysis have been conducted and published on this type of bearing, especially in view of the huge number of Oil Ring Lubricated bearings that are in operation today.

The published data agrees with and confirms work done by this author several years earlier, and it is reproduced as part of this developmental test program. The testing, conducted by the author's company and published data and analysis, confirm that the conventional DIN Oil Ring Lubricated bearing operates in a starved condition, i.e. the amount of oil delivered to the journal for lubricating the bearing by the oil ring is less than the amount of oil required to develop and maintain a full oil film over the entire babbitted bearing surface. The testing has shown that the size, shape and cross section of the oil ring produce varying degrees of oil delivery. However, no oil ring design tested to date has been shown to deliver sufficient oil to enable the bearing to develop a full oil film and eliminate bearing oil starvation.

The new NT Tilt pad bearing liner is designed using a tilting pad with an embedded pocket. The test bearing used a constant depth pocket that began at the leading edge of the pad and terminated with a trailing edge abrupt step. The pocket width was 0.13 % of the pad axial length, 50% of the arc length of the pad and about three film thickness' deep.

TEST RIG DESIGN

A test rig, shown by the CAD drawing of Figure 1, below, was designed that would enable the test bearings to be operated in usual application conditions of load and speed and rigid mounting support. A solid rotor was machined from a bar of ASTM 4340 material; with the weight being distributed about 65% to the non-drive end journal and 35% to the drive end journal of the test machine. The rotor was precision dynamically balanced to the limit of the sensitivity of the balance machine so the rotor had almost "zero" residual unbalance. Two different journal bearings were used in a single test rig.

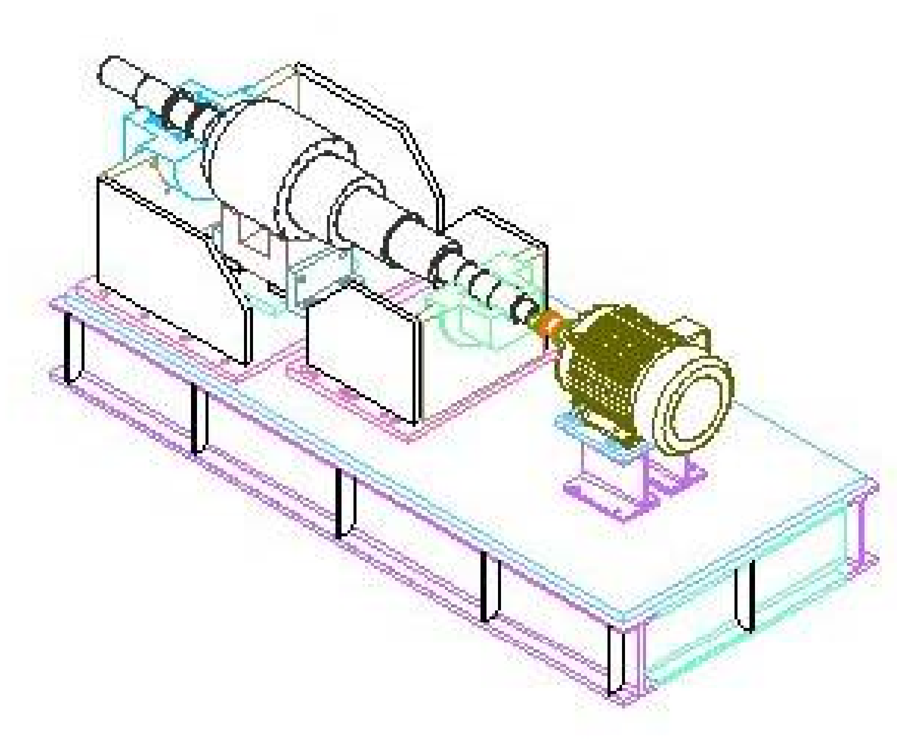


Figure 1 – CAD drawing showing a solid model rendering of the test rig.

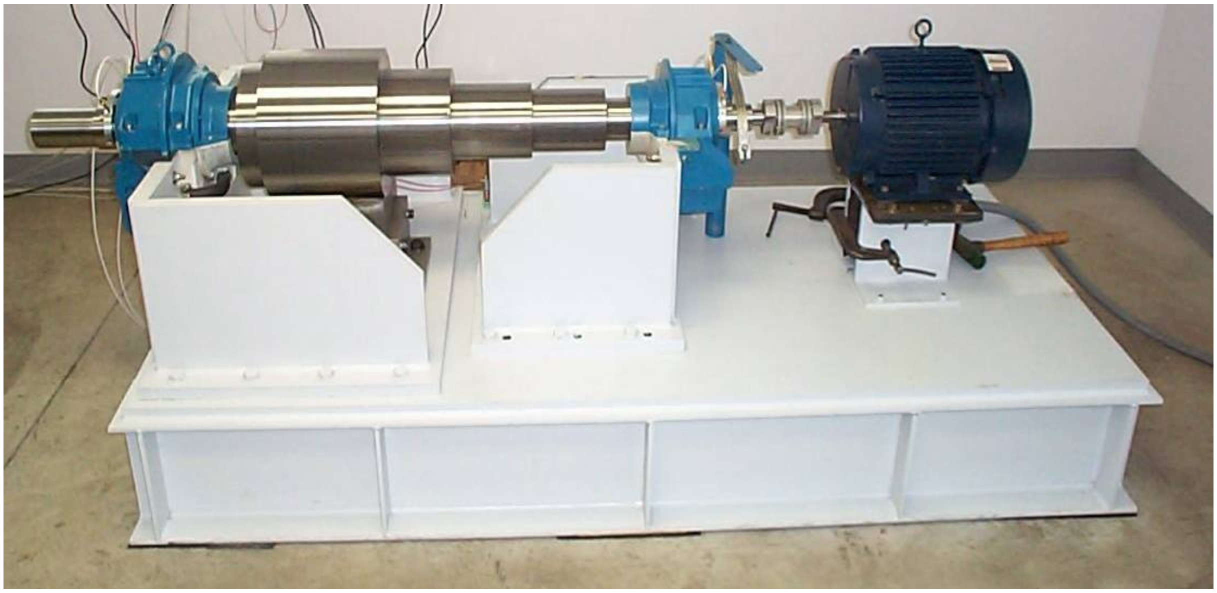


Figure 2 – Photograph of the test rig being assembled.

The shaft journals and non-contact probe targets were ground in a single setup to be concentric to within 0.0001 inches. The ground surfaces were equal to or better than a 16 finish. The lighter journal end of the machine is driven by a two-pole induction motor through a flexible disc spool coupling by an adjustable speed inverter. Both bearing sumps were filled with DTE Light Turbine oil. A series of 12 equally spaced holes, drilled axially in each of two planes, can be seen in the upper right corner of Figure 3 below. These drilled holes were used to add unbalance weights during testing.

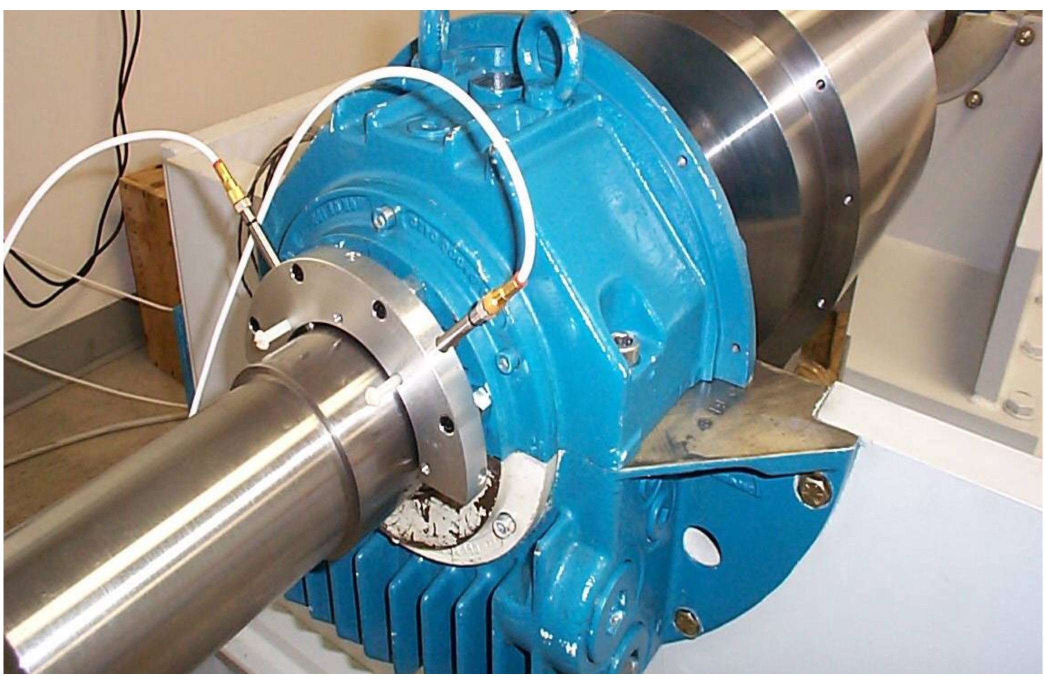


Figure 3 – Test bearing housing, non-contact capacitance probes, and unbalance weight holes.

The journal orbit was measured using non-contacting capacitance probes arranged in the usual API placement of 45 degrees off of the horizontal, as shown in Figure 3 above. Capacitance probes were selected because they are inherently very sensitive, easily able to measure a 0.01 of a thousandth of an inch.

Most importantly however, capacitance probes are not affected by the magnetic permeability variation of the target shaft material, commonly known as electrical runout. Hence, they are ideal for making fluid film thickness and vibration measurements. The capacitance probe will measure the mechanical runout of the probe target area so roundness and concentricity are important and must be controlled.

Low pass filters conditioned the signals from the capacitance probes, with an upper cut off frequency of 250Hz. The probe signals were plotted against each other on the same time base to produce a rotor orbit on the oscilloscope. The amplitude setting was selected to capture the entire journal orbit and fill the oscilloscope screen.

ELECTROMAGNET LOADER DESIGN

A "C" shaped electromagnet was designed such that by varying the current to the magnet, the load applied to the bearings can be selected. Figure 4 below is a photograph of the electromagnet installed under the test rotor in the test rig.

Figure 4 – Electromagnet loader installation.

A load cell (center of photograph) was fitted in the larger test bearing liner and a range of currents applied to the electromagnet and the resulting journal load (LBF) was measured.

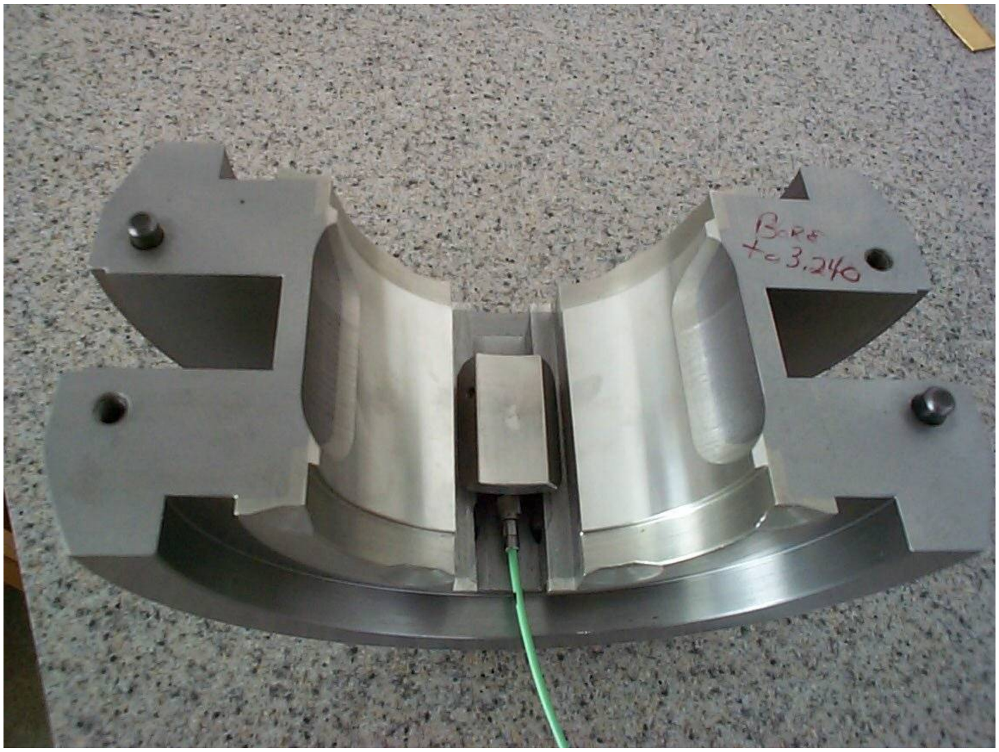


Figure 5 – Load cell setup for measuring electromagnetic force.

The measured load data was compared to the design calculations for the electromagnet load (LBF) and a calibration curve and equation were developed for use in selecting an applied journal load during bearing testing.

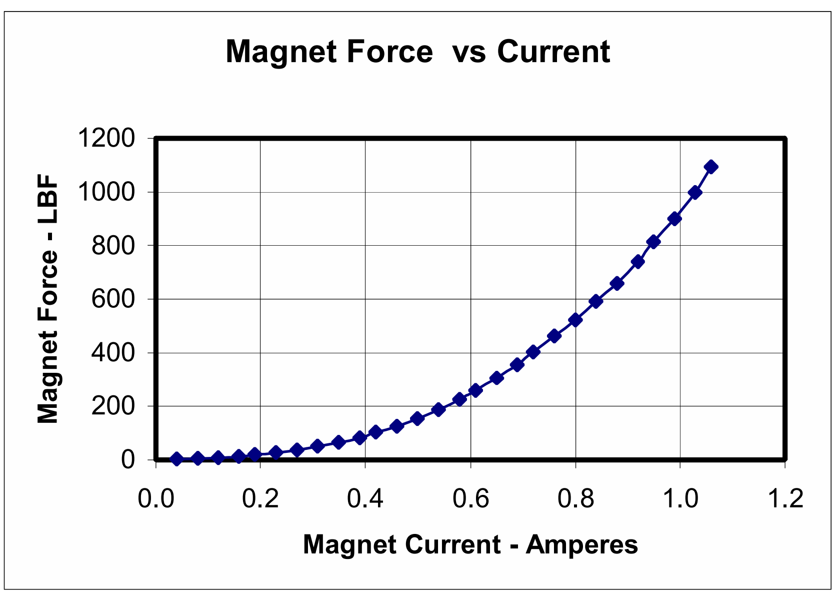


Figure 6 – Plot of test data showing magnetic force (LBF) as a function of current.

Temperature Measurements

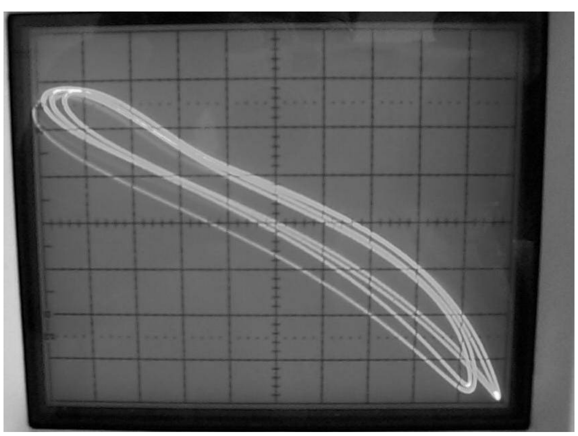
Temperature measurements were automatically logged, about every 20 seconds, by a multi-channel data logger into the lab PC. Temperatures measured included the oil sump temperature, the room ambient temperature, and the pad "hot spot" temperature for both the conventional and NT liner tests. In the case of the NT liner test, the tilt pad inlet temperature and the oil temperature between the pads was also measured and recorded. The test rig was operated at steady speed and load until three equal "hot spot" temperature measurements at 15-minute intervals were obtained. All bearing configurations were tested at 3000, 3600, and 4300 RPM using three different oil ring cross sections.

TEST RESULTS – CONVENTIONAL DIN LINER

The conventional DIN Liner was an 80 mm bore, with a length of pad in the axial direction of 48.6 mm, giving an L/D ratio of 0.61. The diametral clearance was 0.008 inches. The bearing was a conventional bore design having a lower pad of 150 degrees in arc length. The upper half pads are also 150 degrees in arc length, and the upper half split into two narrow pads with a large relief groove in the center to accept the oil ring. All pads are fixed.

The conventional DIN liner was fitted with three RTD's at 210, 220, and 230 degrees from the top vertical in the direction of rotation, and on the axial centerline of the bearing pad. Fluid film thickness measurements were made using a digital voltmeter to measure the voltage of the non-contact capacitance probe. The non-contact capacitance probes have a sensitivity of 1.0 volt per 0.001 inch over a 10.0 mil range.

One of the significant findings of the conventional DIN sleeve oil ring lubricated test program was the low frequency pulsation of the rotor bearing system. During all tests, for all three oil ring cross section configurations, and all three speeds, a low frequency of about 0.5 sec to 1.0 sec, pulsation of the journal orbit was observed as shown in the photographs below.



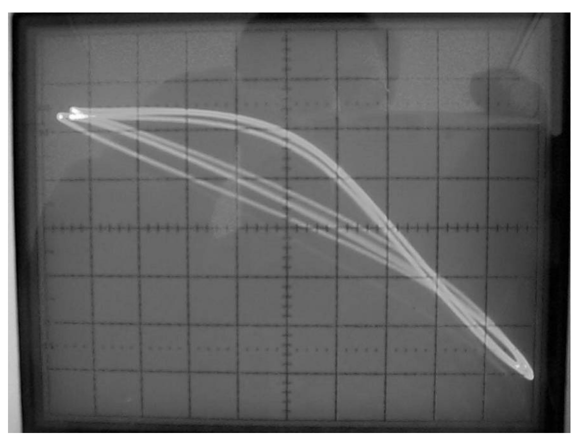


Figure 7 - Conventional DIN Liner 3600 rpm (left) and 4300 rpm (right).
Each major division is equal to 0.0002 inches.

The pulsation appears to be linked to the degree of starvation of the bearing oil film. This pulsation causes the rotor orbit to "throb", oscillating from a smaller amplitude to a larger amplitude, and back to a smaller amplitude, repeatedly. The photographs above clearly show the "throbbing" journal orbit. The variation in journal amplitude as a function of time can be seen in the photographs. The highly elliptical orbit is also clearly visible. The rotor orbit has a nominal ellipticity ratio of 3.5. The rotor unbalance used for all conventional DIN liner tests was 3.37-ounce inches.

TEST RESULTS – NEW NT TILT PAD LINER

The new NT tilt pad was fitted with an RTD embedded from the backside of the pad, on the centerline of the pad, 75% of the arc distance from the leading edge in the direction of rotation, and an RTD was fitted in the backside of the tilt pad on the leading edge.

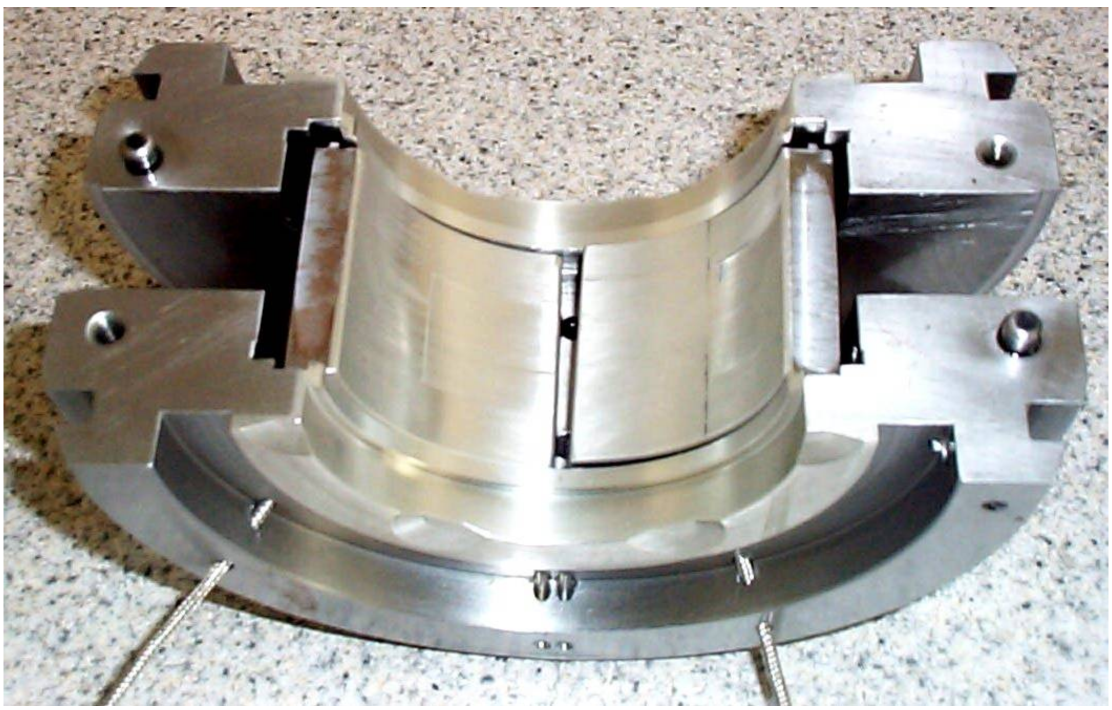


Figure 8 – NT Tilt Pad with Embedded Pocket.

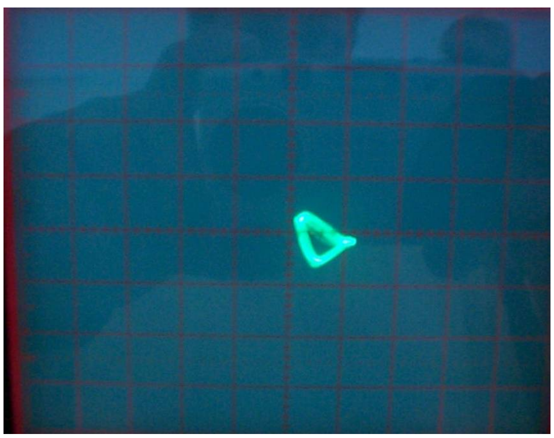
The new NT Tilt pad oil ring lubricated bearing never exhibited a pulsating journal orbit. The rotor orbit was very nearly a circle, with a nominal ellipticity ratio of about 1.15. The amplitude of the journal orbit is about 14 times smaller than the journal orbit produced by the conventional DIN liner, using the same residual unbalance magnitude and rotor location.

The amplitude of the journal orbit is of the same order of magnitude as the mechanical runout of the shaft. The residual unbalance of the rotor was tripled and the test repeated with no measurable difference in rotor orbit size, shape, or orientation. An additional test was conducted where the electromagnet was used to increase the journal load by about 20%, with no significant observable change in temperature, film thickness, or orbit size, shape, or orientation.

This result suggest that the new NT Tilt pad with the embedded pocket design produces a significant amount of damping as compared to the conventional DIN liner in the exact same application and operating conditions.

Film thickness measurements made during testing showed that the trailing pad oil film thickness (0.00068 inches) is slightly smaller than the leading pad oil film thickness (0.00076 inches). The trailing pad "hot spot" temperature is slightly warmer than the leading pad hot spot temperature, by about two degrees Fahrenheit.

An interesting condition was observed during the NT liner testing. The RTD placed in the oil film and between the leading and trailing pads measured a temperature very nearly equal to the leading edge pad temperature for the trailing tilt pad. The leading pad hot spot temperature was only slightly less, by about two degrees Fahrenheit, as compared to the trailing pad hot spot. These measurements tend to suggest that the bearing oil flow is very low, with just a minimum of oil exchange from the sump and virtually no side leakage from the bearing; a flow condition we refer to as bearing starvation.



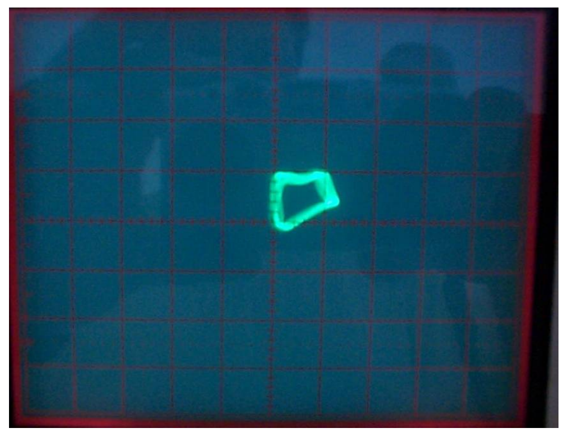


Figure 9 – New NT DIN Liner at 3600 RPM (left) and 4300 RPM (right).
Each major division is equal to 0.0002 inches.

Work done in the past, by this author and others, suggests that a thin film of hot oil clings to the rotating shaft and that modest amounts of cooler oil from the oil sump are mixed with this thin film by the action of the shaft rotation and the oil ring such that the bearing operation is approximately isothermal. Hot oil carryover in a starved condition describes the operating mode for the oil ring lubricated bearing. Since the bearing operates starved, the hot spot temperature is the governing parameter for avoiding bearing failure.

OIL RING CROSS-SECTION

Three oil ring cross-sections were tested with both the conventional DIN liner and the new NT Tilt pad liner. The cross-section of oil rings tested were all trapezoidal with different bore designs, which included a smooth bore, a five groove bore, and a single large central groove. While differences are measurable between the various combinations of liner and oil ring cross-section, it can be said that all cases were operating in a starved condition of lubrication. None of the oil rings tested provided sufficient lubrication such that either bearing was able to develop a full oil film.

CONVENTIONAL PAD - DIN LINER LIMITATIONS

As part of the test program, we attempted to operate the new NT DIN liner configuration, i.e. two tilt pads with the embedded pocket, at three test speeds, while holding all other test variables constant. In this way we could evaluate the contribution of the embedded pocket. Variables such as film thickness, oil temperature rise and vibration levels were measured and compared to the New NT DIN Liner having two tilt pads with the embedded pocket.

The tests indicated that the Conventional Pad DIN Liner operated at 194.9 F at 4300 RPM. However, the conventional pad DIN Liner, i.e. the tilt pad without the pocket, could not be operated long enough at 4300 RPM to stabilize the temperature. As the test machine was brought up to 4300 RPM the downstream hot spot temperature began to rise rapidly. Within a minute at 4300 RPM, the temperature was on a steep rate of climb and had passed through 195 F. The test was aborted.

The fact that the tilt pad without the pocket could not operate long enough to achieve an acceptable stable temperature at the down stream hot spot, validates the theory that the tilt pad with the embedded pocket operates at a lower film temperature. The New NT DIN Liner tilt pad with embedded pocket (Pkt width= 0.5 inches) operated at 4300 RPM with a stable downstream temperature of 187.7 F, well below our test threshold limit of 195 F.

DISCUSSION

This paper discusses the design, development and testing of a new NT Tilt Pad DIN Liner which utilizes a tilt pad with an embedded pocket. The new tilt pad was designed with a constant depth pocket and an abrupt trailing step. The new bearing was tested at three speeds and with three different oil ring cross-sections. The new NT Tilt pad bearing liner performed better than the conventional DIN liner since it did not exhibit any pulsation of the rotor orbit under any of the test conditions.

The NT Pad bearing orbit was nearly circular with a small ellipticity ratio of 1.15. The film thickness measured to be about 0.00068 to 0.00076 inches, which is acceptable. The centerline downstream hot spot temperature was about 20 degree F higher than the DIN Liner.

However, the rotor orbit of the NT Pad DIN Liner was about 14 times smaller than the orbit of the Conventional Din Liner. The large reduction in orbit is due to the significant damping produced by the NT DIN Liner. The viscous damping works to reduce rotor motion. The damping dissipates energy, which produces heat in the oil film. This heat adds to the heat produced by viscous shear of the oil film.

CONCLUSIONS

Four major advantages of the new NT Tilt pad bearing have been shown by the testing:

(1) The new NT Tilt pad bearing operates without exhibiting the pulsating journal orbit observed during operation of the conventional DIN liner bearing;
(2) The new NT Tilt pad bearing journal orbit is very small and nearly circular in shape. The journal orbit is about 14 times smaller than the orbit produced by the conventional DIN liner bearing under exactly the same test conditions;
(3) When the residual unbalance of the rotor was increased by a factor of three, no significant change was observed in the rotor orbit size, shape, or orientation. This result suggests that the new NT Tilt pad bearing develops a significant amount of damping in the oil film. The tilting pad feature works to all but eliminate the cross coupled stiffness and damping and therefore the bearing design is inherently more stable as compared to the conventional DIN liner design; and
(4) When the load was increased 20%, no significant change in rotor orbit, size, or orientation was noted. This test indicates that the new NT Tilt pad bearing produces significant damping and thereby develops a strong resistance to increased vibration due to increased load.

The test results given in Table 1 of Appendix I show that the new NT Tilt pad bearing can be utilized as a direct replacement for the conventional DIN liner. By replacing the conventional DIN liner with the new NT Tilt Pad liner, all of the above-discussed advantages would be afforded to the machine performance. In addition, bearing instability is all but eliminated due to the tilt pad arrangement of the pads and their inherent ability to minimize cross coupling.

Testing also showed that the NT DIN Liner configuration having tilt pads without the embedded pocket could not operate at an acceptable downstream “hot spot” temperature at 4300 RPM. While the tilt pad with the embedded 0.5 inch wide pocket operated successfully at 4300 RPM, achieving a downstream “hot spot” temperature of 187.7 F. This test result validates the theory that the tilt pad with the embedded pocket operates at a lower film temperature.

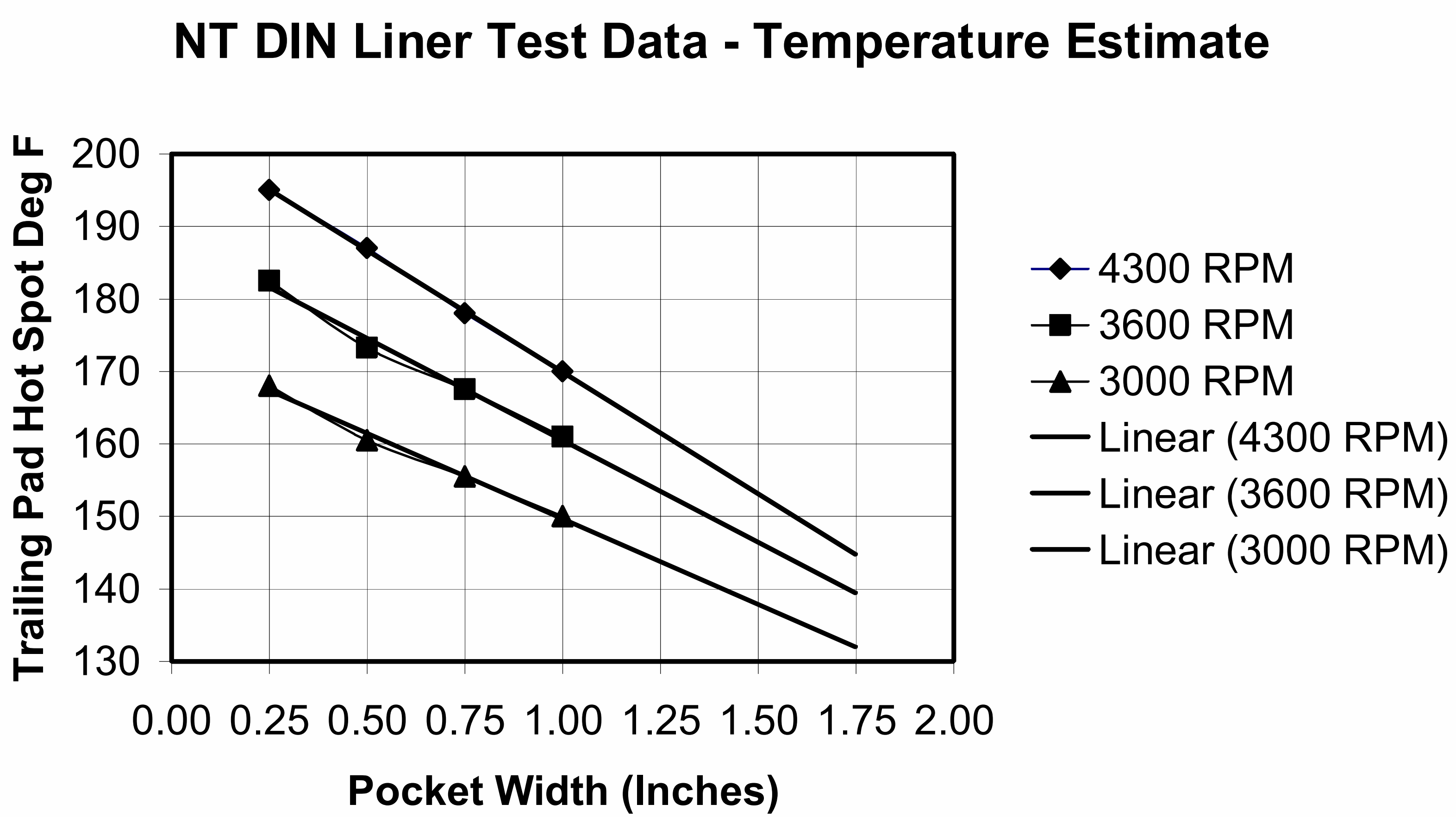
The new NT Tilt Pad bearing liner would benefit from a new oil ring design that could deliver increased oil flow. This is a topic of further research and development that the author’s company will be pursuing as the testing pushes forward in the future.

Appendix I

Table 1 - BBI 80 mm Oil Ring Lubricated Liner, Test Data Summary

All L/D = 0.61 — All Trapezoidal Oil Ring Cross Section
All Journal Loads at 412 LBF — All Unbalance at 3.67 oz*in total

	Conv Din Liner	80 Deg Arc Conv Pad	80 Deg Arc NT Pad 0.25 Pkt	80 Deg Arc NT Pad 0.5 Pkt	80 Deg Arc NT Pad 0.75 Pkt	80 Deg Arc NT Pad 1.0 Pkt
RPM = 3000						
Room Temp (F)	85.8	87.9	92.2	95.9	87.0	90.5
Test Sump (F)	116.6	122	124.2	120.4	113.0	112.2
Trailing Pad Hot Spot Temp (F)	144.7	166.8	169.2	160.5	156.0	150.0
Delta Temp (F)	28.1	44.8	45	39.8	43	37.8
Hot Spot Reduction (F)			-2.4	6.6	10.8	15
RPM = 3600						
Room Temp (F)	88.4	94.6	96.6	99.6	88.3	90.8
Test Sump (F)	120.5	130.3	132.3	126.2	118.7	115.6
Trailing Pad Hot Spot Temp (F)	155.9	181.2	182.5	173.3	167.5	160.8
Delta Temp (F)	35.4	50.9	50.2	47.1	51.2	45.2
Hot Spot Reduction (F)			-1.3	7.9	13.7	20.4
RPM = 4300						
Room Temp (F)	83.7	99.4	99.7	102.9	89.4	89.7
Test Sump (F)	123.3	138.6	139.1	133.5	124.0	119.4
Trailing Pad Hot Spot Temp (F)	160.5	194.9*	195.2	187.7	178	170.0
Delta Temp (F)	37.2	56.3	56.1	54.2	54	50.6
Hot Spot Reduction (F)			-0.3	7.2	16.9	24.9


NT DIN Liner Test Data - Temperature Estimate
Trailing Pad Hot Spot Deg F
200
190
180
170
160
150
140
130
0.00 0.25 0.50 0.75 1.00 1.25 1.50 1.75 2.00
Pocket Width (Inches)
4300 RPM
3600 RPM
3000 RPM
Linear (4300 RPM)
Linear (3600 RPM)
Linear (3000 RPM)