A FLEXIBLE PAD BEARING SYSTEM FOR A HIGH SPEED CENTRIFUGAL COMPRESSOR

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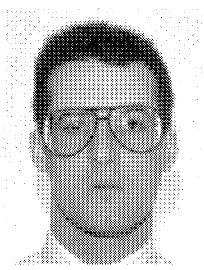

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ABSTRACT

The application of a new radial and thrust bearing for use in high speed integrally geared centrifugal compressors is presented. The bearing design is of the flexible pad type. Bearing design analyses were conducted to assure a stable rotor bearing system, along with possible improvement in bearing performance relative to the five shoe tilting pad journal bearing and taperland thrust bearing system.

Performance characteristics of both the radial and thrust bearings were measured at different bearing loads, and compared with the predicted results. Comparison of the bearing performance parameters were also made with the five shoe tilting pad journal and taperland thrust bearing system. Performance parameters included lubricant temperature rise between the oil inlet and discharge, oil flow, frictional power loss, and bearing pad metal temperature.

Rotor bearing stability characteristics were evaluated at different loads to assure and verify a stable rotor bearing configuration. Predicted rotor response was also compared with those obtained during the test.

The test results showed that the flexible pad bearing system resulted in a lower oil temperature rise, and frictional power loss over the five shoe tilting pad journal and taperland thrust bearing system. The flexible pad rotor bearing system was also found to be

stable, and that the observed peak response correlated well with prediction.

INTRODUCTION

High speed centrifugal compressors use different types of bearings to carry the radial and axial loads associated with geared compressors. One of the most stable journal bearings that is capable of operating over a wide load range is the tilt pad design. To carry the axial loads at these high speeds, the taperland thrust design has been utilized for many years. Engineered into one compact package, referred to herein as the tilt pad/taperland bearing, the design has supported and positioned high speed rotors for years.

A relatively new bearing applicable to the high speed turbomachinery industry is the flexible pad design for journal and thrust bearings. Designed properly, the bearing offers lateral stability [1] and load carrying capacity comparable to the tilt pad journal and taperland thrust design with lower mechanical losses [2]. It can also be packaged into a journal and thrust combination bearing similar to the tiltpad/taperland design described above.

The results are presented of a comprehensive test program comparing the flexible pad design with the tilt pad/taperland design applied to a high speed geared centrifugal compressor. The comparisons are limited to the mechanical losses and rotordynamic characteristics associated with these bearings. Also presented is a comparison of the analytical data with actual test results for the flexible pad design.

FLEXIBLE PAD BEARING

Journal Pad

The flexible pad journal bearing as shown in Figure 1, is a one-piece design that can provide the rotordynamic characteristics similar to that of the tilt pad bearing without the complexities of a multipiece design. The bearing consists of centrally-pivoted pads machined from a solid blank through an electrical discharge machining (EDM) process. The subject bearings utilized a bearing grade bronze material.

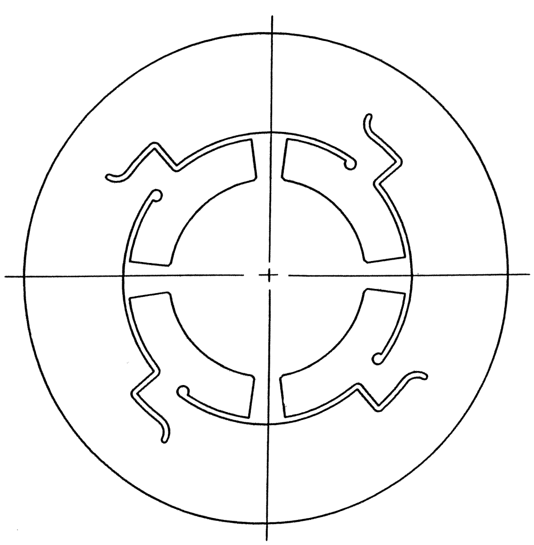

Figure 1. Tested Flexible Pad Journal Design.

These pads are supported by webs designed to carry the radial shaft loads, while also providing rotational (tilt) flexibility similar to a tilt pad. It is this freedom of the pads to tilt about an axis parallel to the rotor on individual pad pivots that is fundamental to the design. Pad pivot motion is illustrated in Figure 2 in both conventional tilt pad and flexible pivot bearings.

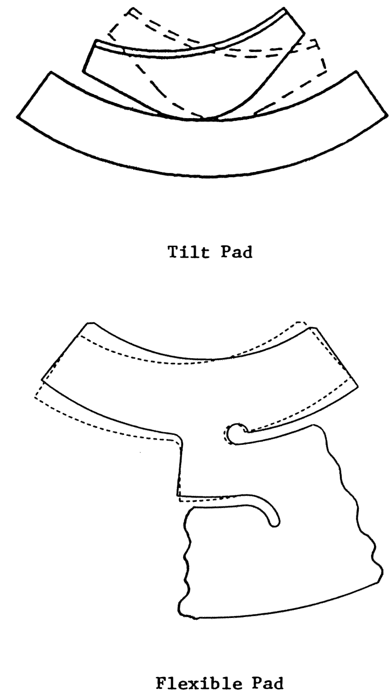

Figure 2. Pivot Geometry of Tilt Pad and Flexible Pivot Journal.

One basic difference between the designs is that the tilt pad has essentially zero pivot angular stiffness, while the flexible pad has an angular stiffness determined by the support web. The web geometry, optimized for individual applications, must be capable of handling the stresses due to the radial load, while flexible enough to permit the pad to pivot similar to a tilt pad design. The one-piece design offers better reliability due to the absence of relative part movement that could lead to fretting corrosion due to high contact stresses.

Another variable that can be used to tune the bearing, is the pad geometry. For example, the pad can be modified by changing the offset and preload similar to a fixed bore design. This geometry along with the web geometry can extend the performance over a wide operating range.

Thrust Pad

The flexible pad thrust bearing consists of a flexible pad and support structure as shown in Figure 3. Pad movement is provided by minute deflection of the pad on its support structure. This system provides varying hydrodynamic wedge shapes, rather than relying exclusively on elastic behavior of the pad surface.

The design, similar to the journal, also takes advantage of the elastic behavior of the support structure. Due to the flexible support, the design can support a moment. This capability assists in shifting the pressure profile so that the resultant load vector need not pass through the pivot as it does for the traditional pivot pad bearing. As the load leads the pivot, it can help deflect the structure so that a converging wedge is formed. On the other hand, a flexible structure will resist the formation of a diverging wedge if the load trails the pivot. It has been shown [2] that this design allows for a significant decrease in the coefficient of friction, by optimizing the tilt thus resulting in lower power losses for thrust and journal applications.

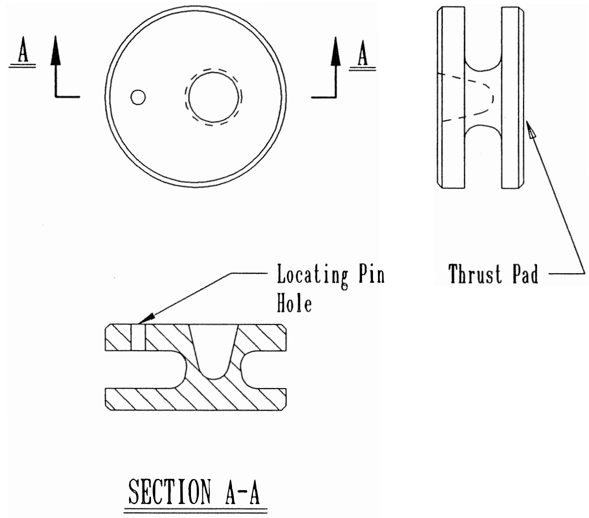

Figure 3. Thrust Bearing Pad with Flexible Support Structure.

TEST VEHICLE

Both bearing designs, the tilt pad with integral taperland thrust and flexible pad, were tested in a medium flow high speed centrifugal air compressor driven by a steam turbine. The compressor was capable of operating at various load conditions by throttling the inlet valve, and opening or closing the discharge backpressure valve.

As illustrated in Figure 4, the compressor consists of two compression stages, each of the single overhung rotor design on separate pinions driven by a common bullgear rotating at a nominal design speed of 3570 rpm. Due to the gear ratio between the bullgear and pinions, the low speed pinion operates at a speed of 30,900 rpm, while the high speed pinion operates at a speed of 40,500 rpm.

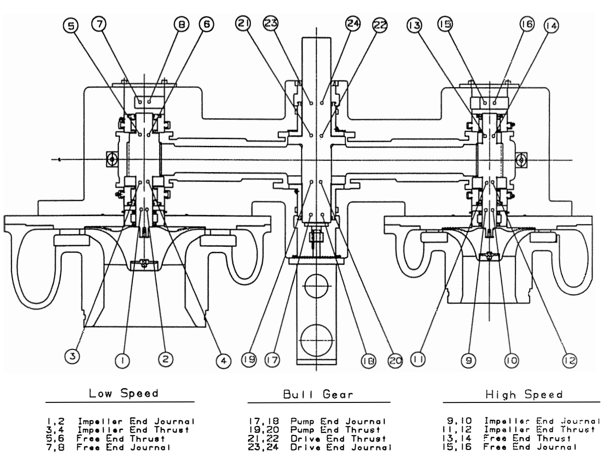

Figure 4. Bearing Throwoff Thermocouple Location.

The initial bearing design for the pinions is a five shoe tilt pad journal with an integral taperland thrust face. The tilt pad low speed journal is 1.75 in in diameter with a pad length of 1.25 in, while the high speed journal is 1.5 in in diameter with a pad length of 1.0 in. Each of these journal bearings was oriented in the gearcase to allow loading between the pads.

The low speed taperland active thrust face has an outside diameter of 3.4 in, while the inactive face diameter was 3.0 in. The high speed active thrust face was 3.25 in in diameter with a 2.5 in diameter inactive face. Both thrust faces had eight taperland pads.

The flexible pad bearings have the same 1.75 in journal diameter with a journal pad length of 1.4 and 1.58 in for the low speed impeller and free end, respectively. The high speed impeller and free end have journal pad lengths of 1.16 and 1.35 in, respectively, with the same 1.5 in diameter journal as the tilt pad design.

The thrust faces consisted of six 0.75 in diameter pads for the low speed active face and sixteen 0.3 in diameter pads for the inactive face. The high speed active thrust face had ten 0.75 in diameter pads, and sixteen 0.3 in diameter pads for the inactive face. A typical picture of both the journal and thrust deflection pad designs used for these tests can be seen in Figure 5.

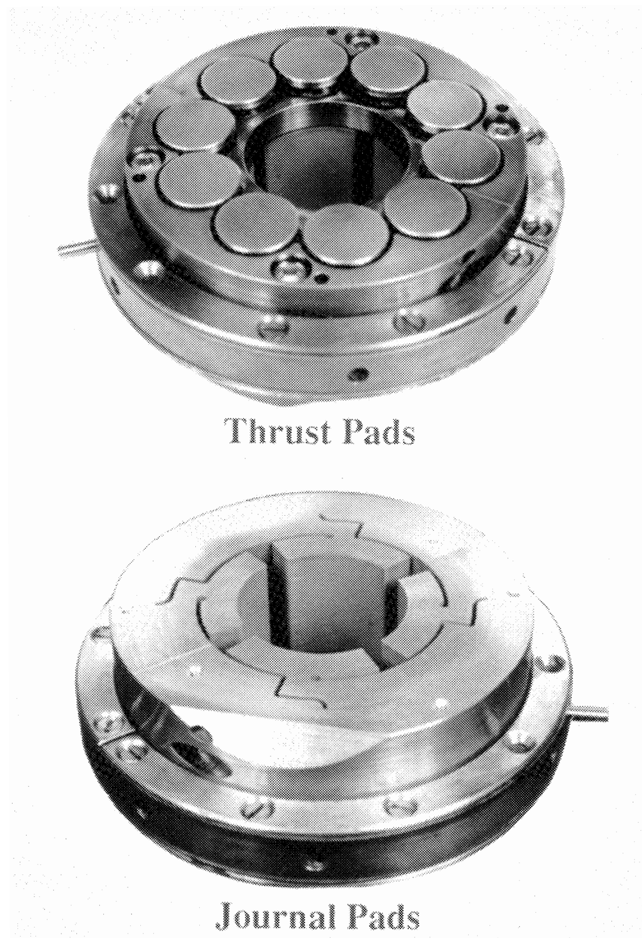

Figure 5. Flexible Pad Test Bearing High Speed Impeller End.

The bearings are supplied with oil through an annulus in the gearcase around the bearing housing shoulder. The oil is then distributed to the journal and thrust pads through a series of feed holes and clearance spaces in the housing.

The bullgear utilized the same sleeve type bearings for all of the tests. Therefore, the only discussions pertaining to these bearings will be associated with the measurement of the overall gearcase mechanical losses.

TEST ARRANGEMENT AND PROCEDURE

Mechanical Loss

As shown in Figure 4, the bullgear and each pinion had a total of 24 thermocouples located in the drains of each bearing. Two thermocouples were positioned in the drain adjacent to the thrust face, and two in the drain on the opposite end of the bearing which more represents the journal conditions. This was done for the active and inactive bearing of the bullgear, low speed, and high speed rotors. A separate flow meter was located at the oil feed to each rotor so that the oil flow could be measured to that rotor bearing system.

As a further check, the total oil flow and average total temperature rise across the entire unit was also measured. Two thermocouples and a flowmeter were located in the gearcase main oil feed along with two thermocouples in the drain of the compressor gearcase which contains the bullgear and both the low and high speed rotor bearings. These data were used to calculate the total gearcase mechanical loss, and compare the results with the summation of losses for each rotor and bullgear.

The mechanical loss data was taken at two load conditions which represented a typical compressor power operating range for each pinion. A tabulation of the bearing journal and thrust loads for each bearing at the two load conditions is presented in Table 1. The load was varied by throttling the compressor inlet and controlling the discharge valve. At each load condition, the oil temperature was set at 120°F while data were taken for various oil flowrates. This was to establish a profile of the mechanical loss and oil throwoff temperatures for a wide flow range, which enabled a minimum required oil flow to be selected thus minimizing the mechanical losses.

Table 1. Bearing Loads For Loaded and Unloaded Conditions.

	Condition 1 (Loaded)		Condition 2 (Unloaded)	
	Radial lbf	Thrust lbf	Radial lbf	Thrust lbf
Low Speed Rotor:				
Impeller Bearing	375	800	90	650
Free End Bearing	425	0	105	0
High Speed Rotor:				
Impeller Bearing	405	600	120	235
Free End Bearing	460	0	135	0

The mechanical loss of the bearings was determined by measuring the oil flow and average temperature rise across the bearings for each pinion and bullgear, and evaluated by using the following expression:

$$\text{MECHANICAL LOSS (hp)} = 0.0804\ (\dot{m} \cdot C_p \cdot \Delta T) \quad (1)$$

0.0804 Conversion constant (lbm-hp-min/Btu/gal)

$\dot{m}$ Flow rate of lubricating oil (gal/min)

C_p Specific heat of lube oil (BTU/lbm °F)

ΔT Avg. temperature rise of lube oil (°F)

Finally, due to this being the first test of a flexible pad bearing on this type of equipment, two thermocouples were embedded in the free end bearing loaded and unloaded pads to monitor and record the journal pad metal temperature.

Vibration Measurement

The rotor vibration was measured using noncontacting proximity probes. A total of eight probes were used for the low and high speed rotors. Each rotor had horizontal and vertical probes located just inboard of each bearing. Two additional axial noncontacting probes, one for each rotor, were used as a keyphaser by sensing a groove in the end of each pinion. These vibration amplitude and key phase signals along with a speed signal was recorded on a high speed recorder.

The nosepiece of each rotor was designed to accommodate individual setscrews at various angles. These setscrews were then used to accurately induce specific amounts of unbalance to the rotors for the unbalance response tests.

The rotor vibration data were recorded from zero to the normal compressor operating speed of 3570 rpm. Some of the tests were conducted up to a speed beyond nominal design. These tests were conducted with 120°F inlet oil along with the two load conditions for each pinion to establish the rotor bearing system sensitivity through a normal operating range.

TEST RESULTS AND ANALYTICAL COMPARISONS

Mechanical Loss

The data herein are shown as a comparison of the flexible pad test with the original five shoe tilt pad test. The test results were also compared with the calculated mechanical loss and oil film temperatures for the flexible pad design. The following discussions describe the findings of these tests.

Plots of the mechanical loss as a function of oil flow are shown in Figures 6, 7, and 8. The plots show that the mechanical loss increases as the flow increases for both the tilt pad and flexible pad designs. Also shown are the power savings of the flexible pad design compared to the tilt pad/taperland design at the various oil flowrates.

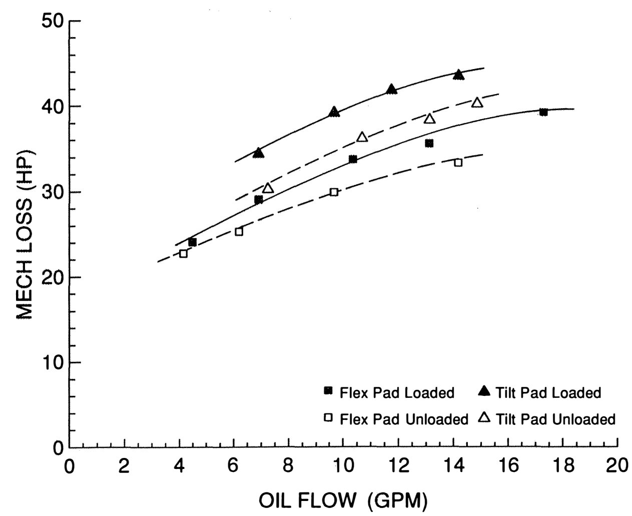

Figure 6. Low Speed Rotor (1.75 In Diameter) Mechanical Loss vs Oil Flow for the Loaded and Unloaded Conditions at 30,900 RPM Using ISO 32 Oil at 120°F Inlet Conditions.

The low speed test data presented in Figure 6, show that at an oil flowrate of 9.0 gpm the mechanical loss is reduced by 16 percent and 15 percent for the loaded and unloaded conditions, respectively, for the flexible pad design over the tilt pad/taperland design.

The high speed rotor results (Figure 7), illustrate that the reduction in mechanical losses, for the flexible pad design relative to the tilt pad/taperland bearing, is better than the low speed rotor results. At an oil flowrate of 14.5 gpm, there is a 21 percent savings for the loaded condition and a 19 percent savings for the unloaded condition.

The overall unit measured mechanical loss, based on the gearcase drain, comparison between the two bearing designs is illustrated in Figure 8. The flexible pad bearings showed a 16 percent

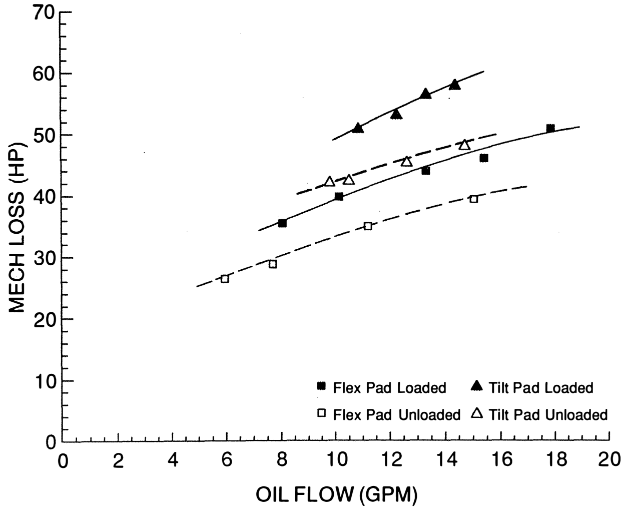

Figure 7. High Speed Rotor (1.5 In Diameter) Mechanical Loss vs *Oil Flow for the Loaded and Unloaded Conditions at 40,500 RPM using ISO 32 Oil at 120°F Inlet Conditions.*

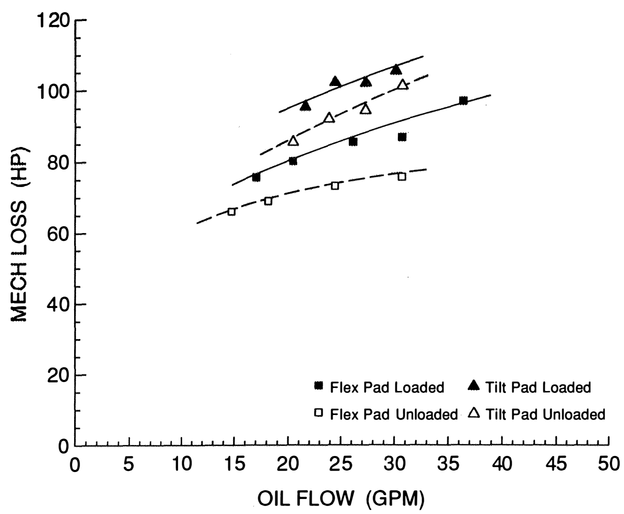

Figure 8. Total Unit Mechanical Loss (Based on Unit Drain) vs *Oil Flow for the Loaded and Unloaded Conditions Using ISO 32 Oil at 120°F Inlet Conditions.*

overall mechanical loss reduction for the loaded condition and a 22 percent reduction for the unloaded condition when compared to the tilt pad/taperland design for an oil flowrate of 26 gpm. Note that these measurements include the bullgear losses.

In Figures 9 and 10 are shown the maximum oil throwoff temperatures as a function of oil flow for the loaded conditions of the low and high speed rotors, respectively.

The flexible pad design resulted in lower throwoff temperatures when compared to the tilt pad/taperland design for the low speed rotor, as shown in Figure 9. This allowed the oil flow for the flexible pad design to be reduced from 9.0 gpm to 6.5 gpm without exceeding the maximum oil throwoff temperature experienced by the tilt pad design. As shown in Figure 6, this reduction in oil flow resulted in a better low speed rotor power savings of 26 percent for the loaded condition and 23 percent for the unloaded condition.

In Figure 10 is shown that, similar to the low speed rotor, the high speed rotor flexible pad design resulted in lower throwoff

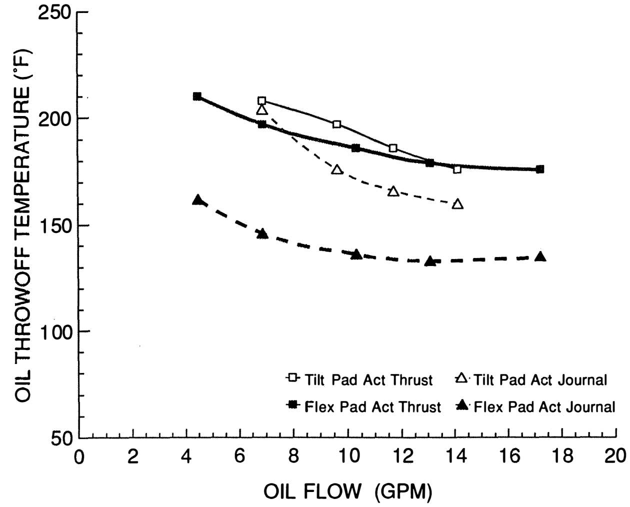

Figure 9. Low Speed Rotor (1.75 In Diameter) Max Oil Throwoff Temperature vs *Oil Flow for the Loaded Condition at 30,900 RPM Using ISO 32 Oil at 120°F Inlet Conditions.*

temperatures relative to the tilt pad/taperland design. Thus, at a reduced oil flowrate of 11.5 gpm and a comparable oil throwoff temperature relative to the tilt pad/taperland design, the flexible pad mechanical power savings was further improved to 27 percent for both of the load conditions, shown in Figure 7, for the high speed rotor.

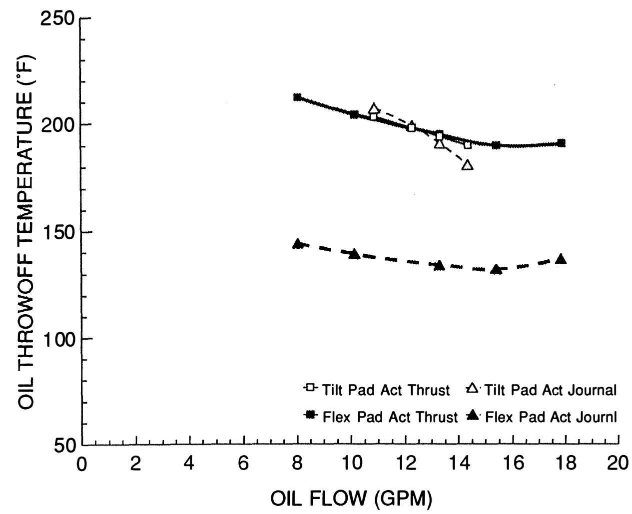

Figure 10. High Speed Rotor (1.5 In Diameter) Max Oil Throwoff Temperature vs *Oil Flow for the Loaded Condition at 40,500 RPM Using ISO 32 Oil at 120°F Inlet Conditions.*

Both the low and high speed calculated mechanical losses for the flexible pad design were higher than the actual test results shown in Figures 6 and 7. The low speed calculated power loss of 32.5 hp at 5.5 gpm was 21 percent higher than the measured power loss. The high speed calculated loss of 48 hp at 10.0 gpm was 18 percent higher than the actual test results at the same oil flow.

The measured journal pad metal temperatures of the low and high speed free end (opposite from the impeller) bearing for the loaded condition are shown in Figures 11 and 12. The embedded thermocouples are located at a distance of 75 percent of the pad length from the leading edge for the trailing pad, and 50 percent for the leading pad. A comparison of the calculated oil film temperatures with the measured pad metal temperatures for the trailing and leading pads showed some deviation. The low speed calculated oil film temperatures, for both the trailing and leading pads, were 15°F higher than the measured pad metal temperatures. The high speed calculated leading pad oil film temperature was similar to the low speed at 18°F higher than the measured pad metal temperature, but the high speed calculated trailing pad oil film temperature was considerably different at 40°F lower than the measured pad metal temperature.

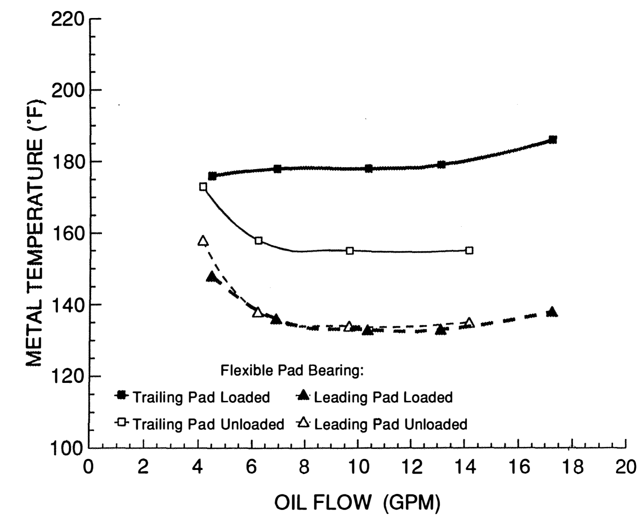

Figure 11. Low Speed Rotor (1.75 In Diameter) Free End Journal Pad Metal Temperature vs *Oil Flow for the Loaded and Unloaded Conditions at 30,900 RPM Using ISO 32 Oil at 120°F Inlet Conditions.*

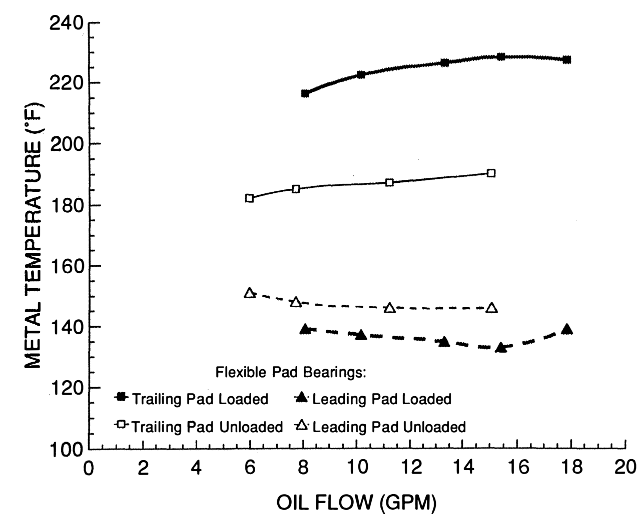

Figure 12. High Speed Rotor (1.5 In Diameter) Free End Journal Pad Metal Temperature vs *Oil Flow for Loaded and Unloaded Conditions at 40,500 RPM Using ISO 32 Oil at 120°F Inlet Conditions.*

Rotor Bearing Dynamics

The low and high speed five shoe tilt pad rotor bearing system proved to be stable over a wide range of operating loads at these high speeds. For the flexible pad design to be successful, it had to provide similar stability characteristics.

Several bearing geometries were analyzed. Results of some of the early designs for the low and high speed rotors compared with the final configuration, cases 5 and 6, and the tilt pad can be seen in Tables 2 and 3. Shown are the log decrements associated with the first forward even mode of vibration at the maximum operating speed and various load conditions. The results demonstrate that there could be a wide range of stability characteristics associated with the geometry of the flexible pad design. The bearing must be customized for individual applications.

Table 2. Low Speed Rotor (1.75 Diameter) Log Decrements. (All cases loaded between pads).

	Unloaded	Avg. Load	Loaded
Tilt Pad Reference: No. Pads 5 m = 0.2923 C_b = 0.0023 C_p = 0.00325 L/D = 0.7	0.615	0.602	0.557
Case 1: No. Pads 6 m = 0.582 C_b = 0.0021 C_p = 0.005 IE L/D = 0.50 FE L/D = 0.5714	-0.306	-0.0447	0.366
Case 2: Same as case 1 except m = 0.500 C_b = 0.0025 C_p = 0.005	-0.5810	-0.0174	0.338
Case 3: Same as case 1 except m = 0.584 C_b = 0.0016 C_p = 0.00361	0.0222	0.0725	0.223
Case 4: Same as case 1 except m = 0.510 C_b = 0.002 C_p = 0.00408	-0.305	-0.0715	0.372
Case 5: No. Pads 4 m =0.380 C_b = 0.0016 C_p = 0.00258 IE L/D = 0.8 FE L/D = 0.9	0.517	0.517	0.501
Case 6: Same as Case 5 except m = 0.30 C_b = 0.002 C_p = 0.00286	0.266	0.253	0.3970

Note: IE = Impeller end & FE = Free end

The tables illustrate that the six pad geometry for this particular application is relatively unstable at the lower load conditions for both the low and high speed rotors. The stability somewhat improves with a reduction in clearance, but not to the degree required for the operating range.

However, the log decrements associated with the four pad geometry, with a lower preload, higher L/D ratio, and an acceptable clearance ratio, shown in cases 5 and 6, provides stable operation over the entire load range. It can also be seen that thelog decrement for this configuration compares favorably with the five shoe tilt pad journal design.

Table 3. High Speed Rotor (1.50 Diameter) Log Decrements. (All cases loaded between pads).

	Unloaded	Avg. Load	Loaded
Tilt Pad Reference: No. Pads 5 m = 0.1305 C_b = 0.0028 C_p = 0.00325 L/D = 0.71	0.189	0.504	0.422
Case 1: No. Pads 6 m = 0.510 C_b = 0.0022 C_p = 0.00449 IE L/D = 0.75 FE L/D = 0.75	-0.239	-0.0357	0.236
Case 2: Same as case 1 except m = 0.58 C_b = 0.0018 C_p = 0.00429	-0.0049	0.0465	0.148
Case 3: Same as case 1 except m = 0.58 C_b = 0.0015 C_p = 0.00357	0.109	0.120	0.148
Case 4: Same as case 1 except m = 0.510 C_b = 0.0019 C_p = 0.00388	-0.0766	0.0133	0.142
Case 5: No. Pads 4 m =0.480 C_b = 0.0015 C_p = 0.00288 IE L/D = 0.7713 FE L/D = 0.898	0.282	0.295	0.303
Case 6: Same as Case 5 except m = 0.397 C_b = 0.0019 C_p = 0.00317	0.105	0.162	0.296

Note: IE = Impeller end & FE = Free end

Frequency plots are shown in Figure 13 for the low and high speed rotors at the impeller end in the vertical direction with the unloaded condition, and an inlet oil temperature of 130°F. The low speed rotor which has an operating speed of 30,900 rpm successfully ran with 1.9 g's of unbalance and a running speed vibration component of 0.14 mils. The high speed rotor operating at 40,500 rpm also ran successfully with 2.8 g's of unbalance and a running speed vibration component of 0.10 mils. Both rotors were stable with no discernable subsynchronous vibration component which is characteristic of rotor instability. Similar results for the loaded condition are shown in Figure 14, which indicate that both rotors are stable over the entire operating range.

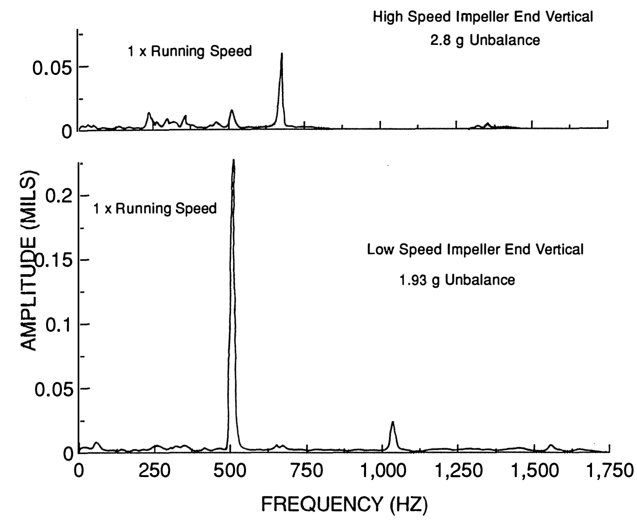

Figure 14. Frequency Spectrum During Operation for the Loaded Condition, at an Oil Temperature of 130°F.

A comparison is illustrated in Figure 15 of the observed and calculated peak response in the vertical direction for the free end and impeller end of the low speed rotor. The correlation between the observed and calculated critical speed is reasonably good with a 11 percent deviation relative to the observed value. No attempt was made to correlate the predicted peak response of the high speed rotor with the observed, because the observed vibration was very low with no discernable peaks.

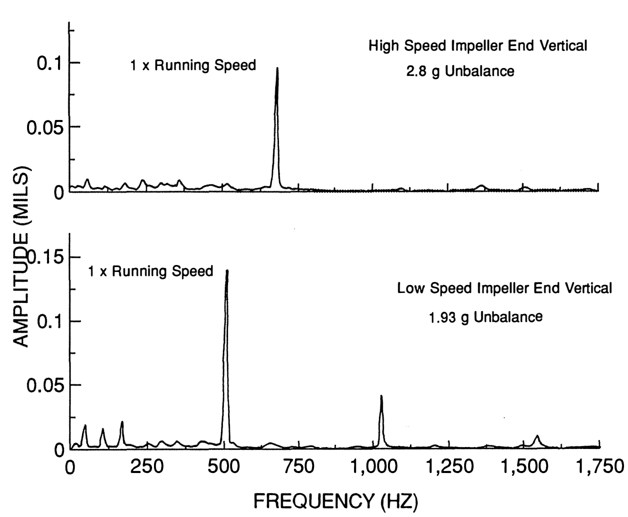

Figure 13. Frequency Spectrum During Operation for the Unloaded Condition, at an Oil Temperature of 130°F.

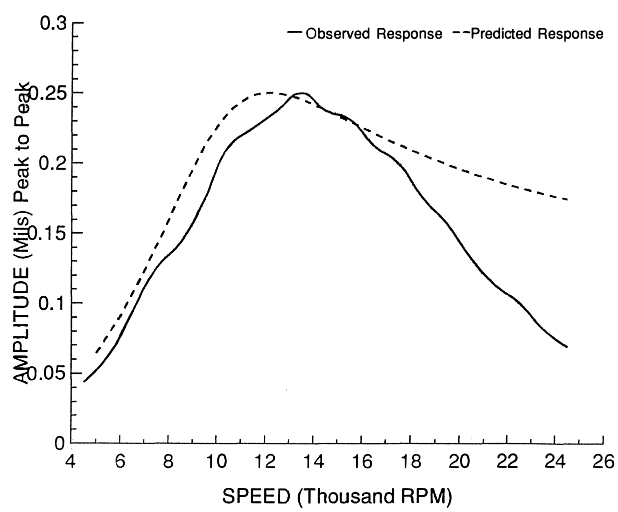

Figure 15. Comparison between the Observed and Predicted Response for the Low Speed Rotor. Impeller end vertical vibration amplitude vs *speed.*

CONCLUSIONS

• The flexible pad bearing tested resulted in lower oil throwoff temperature relative to the five shoe tilting pad journal bearing with an integral taperland thrust bearing for the same oil flow. As a result, the flexible pad design could be operated at a lower oil flow and still provide acceptable oil throwoff temperatures and temperature rise.

• Relative to the five shoe tilting pad/taperland bearing, the flexible pad bearing resulted in lower power losses over the flow range tested.

• A flexible pad journal bearing was designed to provide a stable rotor bearing system similar to a five shoe tilt pad journal bearing for a high load high speed application. The flexible pad journal bearing showed stable operation during the tests over the entire operating conditions as predicted. It is to be noted that the flexible pad bearing had to be customized appropriately for this specific application. The ability to customize a bearing to a specific application allows each application to be optimized.

• The flexible pad journal bearings tested proved to be able to provide an acceptable synchronous response with relatively high unbalance.

• The correlation of the predicted location of the first peak response with the test results was within 11 percent.

• The predicted power loss was 18 to 21 percent higher than the tested results.

• The predicted oil film temperature was 15°F higher than the test results for the low speed rotor, and 18°F higher for the high speed unloaded pad. The high speed loaded pad differed from the others with a 40°F lower than measured temperature.

• The mathematical model needs to be revised to more accurately predict the mechanical power loss and film temperatures.

NOMENCLATURE (For Tables 2 and 3)

m Bearing preload
C_p Pad machined clearance, inches
C_b Bearing clearance, inches

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