An Experimental Study of Thrust Pad Flutter

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Tilting pad thrust bearings are commonly used on both the active and inactive (loaded and unloaded) sides of thrust collars of turbomachinery rotors. Fatigue failure of temperature sensor leadwires to the inactive side thrust pads and abnormal wear at contact points of these thrust pads, in several field applications, led to an investigation as to the cause. Since these problems occurred only on "unloaded" thrust bearings, flutter (vibration) of these inactive side thrust pads was suspected. Tests were run in the laboratory on a double thrust bearing under conditions simulating a turbomachine application. Pad flutter was found at two distinct conditions; low oil flows (as suspected), and also at high oil flows. The paper presents the results of these tests with data on the influence of the parameters studied, and the findings with respect to elimination of this flutter.

Introduction

Oil lubricated hydrodynamic tilting pad thrust bearings are commonly used on both the active and inactive (loaded and unloaded) sides of thrust collars of turbomachinery rotors, particularly in centrifugal compressors. The high load capacity of a tilting pad thrust bearing is desirable, even on the normally unloaded side, to accommodate thrust loads that may develop at off-design conditions, during surge conditions, during run-up and run-down, or simply due to the inability to accurately calculate the magnitude and direction of the thrust loads at all of the operating conditions. Axial end clearance (float) is used to prevent the inactive thrust bearing from hydrodynamically adding additional load to the active thrust bearing, and generating additional heat and power loss. Because of this end clearance, the thrust pads on the inactive side operate with relatively large, built-in, film thicknesses.

Fatigue failure of temperature sensor leadwires to unloaded thrust pads, and abnormal wear at contact points of these pads, in several field applications, prompted an investigation into the cause of these problems. Since this occurred only on unloaded thrust bearings, flutter (vibration) of these inactive side thrust pads was suspected. These problems occurred inconsistently even within a particular machine model, and no specific cause could be determined from an analysis of field data.

A literature search produced very limited information on the specific topic of flutter (vibration) of pads in tilting pad thrust bearings. In view of this limited published information, and since the exact nature of the action causing these problems was unclear, as well as what parameters were significant, it was determined that some laboratory tests were appropriate.

The primary goals of these tests were:

1. To reproduce the action causing the field problems.
2. To determine what parameters were significant.
3. To determine how to design to avoid this problem.

Test Equipment

The facility used for these tests was described by Gardner (1985). Briefly, this is a DC motor driven, hydraulically loaded thrust bearing test stand capable of accommodating thrust bearings in the range of 250 mm to 500 mm (10 to 20 in.) outside diameter. The 750 kw (1000 hp) variable speed geared drive has a maximum speed of about 10,000 rpm. The test and slave thrust bearings are enclosed in separate housings, and each thus operates against its own thrust collar, integral with the shafts. In the interest of covering a wide speed range within the power and cooling capabilities of this test stand, a 267 mm (10.5 in.) double thrust bearing was selected for these tests. The slave bearing in this setup was a 381 mm (15 in.) single thrust bearing. The oil used was an ISO VG32 turbine oil.

To determine if the thrust pads were vibrating, noncontact proximity probes were used to "view" the back face of two pads in the unloaded bearing and one pad in the loaded bearing. The loaded bearing was monitored for pad vibration although it was not anticipated that any significant pad movement would be found, except possibly when no load was applied. Vibration amplitude was of primary interest but vibration frequency was also monitored. The setup for the test bearing in its housing with the proximity probes is shown in Fig. 1. The tilting pad thrust bearings were eight pad, self-equalizing types. Oil was supplied separately to each thrust bearing but was discharged through a common tangential passage, centered axially in the housing.

In addition to the proximity probe information, the following was recorded for each test point:

1. Shaft rpm
2. Oil flow rate to the unloaded side bearing
3. Oil flow rate to the loaded side bearing
4. Oil inlet temperature
5. Oil discharge temperature
6. Thrust load (loaded side bearing)
7. Pad temperatures (loaded side bearing)

After a series of initial tests, two strain gage diaphragm type pressure probes were added in the oil discharge annulus and these are identified in Fig. 1. These probes were mounted with their sensing surfaces flush with the housing bore to measure pressures in this discharge annulus.

Axial thrust end clearance and oil flow rate were the two parameters believed to have the most influence on any vibrations of the unloaded side pads, based on a simple theory of pad flutter described below in the Discussion section. Thus the setup included shims between the unloaded bearing and the housing to allow the end clearance to be a variable from test to test. The actual end clearance obtained in each test setup was determined by using dial indicators between the shaft and the housing to measure the total axial travel of the shaft as it was moved from hard against one thrust bearing to hard against the opposite bearing. Oil flow rates were readily varied with the normal test facility controls, as were shaft speed, oil supply

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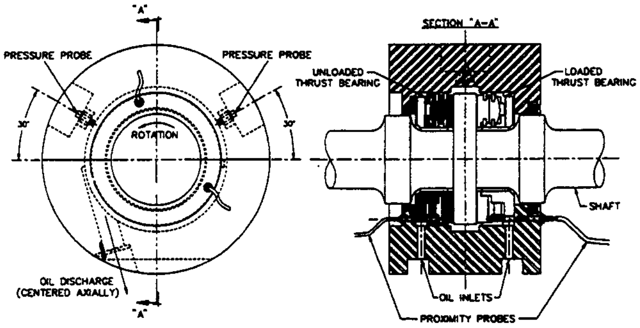

Fig. 1 Arrangement of double thrust bearing for pad flutter tests showing proximity and pressure probe locations

temperature, and the thrust load. Temperatures of the loaded side thrust pads were monitored. A loading of only 1.4 MPa (200 psi) was typically used, simply to insure that the unloaded side thrust bearing was operating with a full end clearance, thus the pad temperatures were relatively low.

All instrumentation was calibrated prior to these tests. Oil flow rates were accurate within 2 percent over the range of flows used, and shaft speed was accurate within 0.5 percent. The proximity probes were calibrated against the actual thrust pads (both steel and chrome-copper backed pads with babbitt facings) with resulting amplitude accuracies within 0.5 percent. The pressure probes were accurate within 3 percent.

Discussion

Inactive side thrust pad flutter is not an unknown phenomenon. Mikula et al. (1983) comments on the occurrence of this in thrust bearing tests comparing lubricant supply methods. In those tests, pad vibration was evident from the fatigue failure of instrumentation wiring to the unloaded pads, and also was manifested as an audible clicking sound emanating from the housing. These problems were eliminated in those tests by the use of an oil control ring around the thrust collar, generating a backpressure in the discharge annulus. It was noted, however, that this oil control ring (discharge restriction) adversely affected the thrust bearing power loss.

Thus, the test setup used here (Fig. 1) incorporated means to vary the discharge restriction by use of an orifice plate in the tangential oil discharge passage. This plate could also be removed to provide an essentially "open" discharge.

A plausible explanation of how unloaded side pad flutter may occur is that sufficient oil is not available to completely fill the large "film thickness," resulting in a full hydrodynamic film for only a portion of the pad circumferential length (Fig. 2).

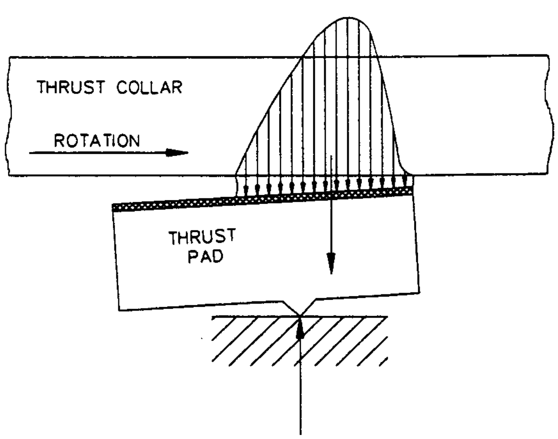

Fig. 2 Unloaded thrust pad with insufficient oil supply, resulting in unstable condition

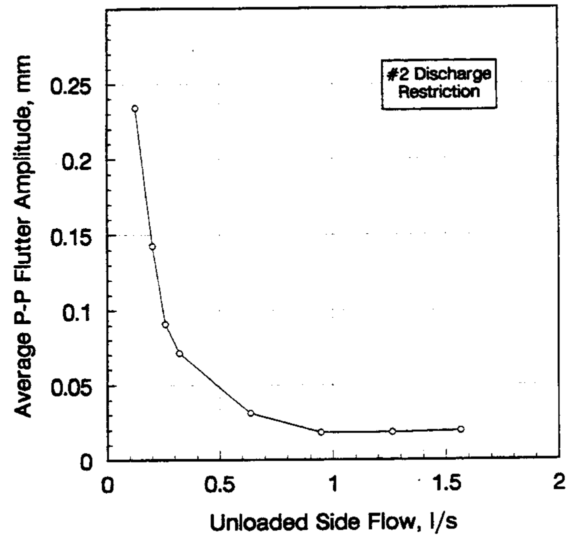

Fig. 3 Pad flutter vibration amplitude resulting from low oil flow rates. 9000 rpm, 0.86 mm end clearance.

The axially directed hydrodynamic force from this partial film may then be circumferentially downstream of the pad pivot. This force results in a moment about the pivot which causes the pad to tilt such that the pad leading edge comes closer to the thrust collar. This "dumps" the hydrodynamic oil pressure. The oil being carried by the collar then attempts to re-establish a hydrodynamic oil film, but insufficient flow results in only the partial film being formed, this again being centered downstream of the pivot. The action thus repeats itself, resulting in the pad vibration known as flutter. With this being the action involved, it would seem reasonable that this could be controlled by a discharge restriction to insure that the thrust chamber operates flooded, or by increasing the oil flow to insure the same action. Also, a reduction in end clearance would appear logical to reduce the unloaded side film thickness and thus reduce the amount of oil needed to insure a full oil film on these pads.

Test Results

The initial test work thus concentrated on reducing the oil flow to the unloaded side of the test bearing, using a nominal end clearance, to determine if pad vibrations would develop. At low flows significant vibrations did develop. Figure 3 is a plot of typical vibration amplitude data obtained in these first tests. The amplitudes and frequencies of these pad flutter vibrations were not steady, as might be seen from an unbalanced rotor for example, but varied over a limited range at an otherwise steady operating condition. Thus, the amplitude and frequency data from both the proximity probes and the pressure probes were recorded as time averaged values.

The extent of the discharge restriction is noted on the vibration data plots and is described later. Although these initial tests were made with only the one level of discharge restriction (#2), later tests showed that these low flow vibrations were affected by the degree of restriction in the oil discharge path; the greater the restriction the lower the flow to generate flutter. This agrees with the experience reported by Mikula (1983).

In addition to checking the influence of reduced flow rates, these initial tests also included different shaft speeds, oil inlet temperatures, and end clearances. In summary, the results of these initial tests were:

1. At low flows, pad vibrations developed.
2. These vibrations occurred at all shaft speeds tested (3000 to 9000 rpm). Amplitudes were generally larger at higher speeds.

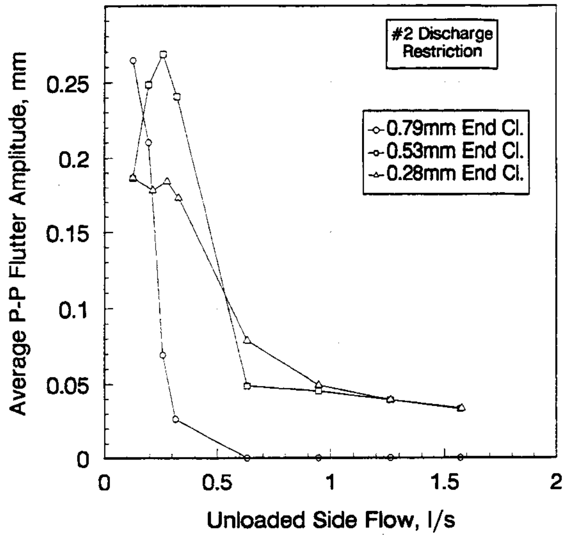

Fig. 4 Pad flutter vibration amplitude at low oil flows and three end clearance values. 9000 rpm.

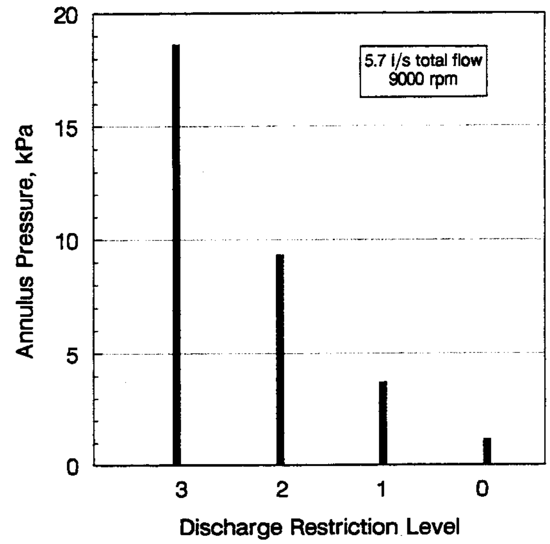

Fig. 6 Static pressure levels in the annulus around the thrust collar at the four discharge restriction levels tested

3. Oil supply temperature had no discernible influence over the range tested (43 to 60°C).
4. The effect of end clearance was not well defined (see Fig. 4).

Although pad flutter (vibrations) was reproduced, it only developed at flow rates that were well below what would reasonably be used in practice, and at relative flow rates much lower than those in use in the thrust bearings where field problems had occurred. The test scope was then expanded to include higher flow rates. Figure 5 is a plot of pad vibration amplitude vs oil flow rate over a wider range of flows, showing the large vibration amplitudes found at the higher flows, in addition to the start of those that develop at the low end of the flow range.

It appeared that these "high flow" vibrations resulted from a different phenomenon. They had the nature of a resonant vibration resulting from some exciting force in tune with a natural oscillating frequency of the pad/oil film system. This prompted the installation of the pressure probes in the annulus surrounding the thrust collar. These probes were mounted flush with the bore of this annulus and thus measured static oil pressures, and changes in these static oil pressures, but not velocity head pressures due to circumferential rotation of the oil in this annulus. Magnitude and frequency from these pressure probes, as from the proximity probes, were recorded. The interest here was to determine if pressure pulsations were present, and if these had some relationship to the pad vibrations (flutter).

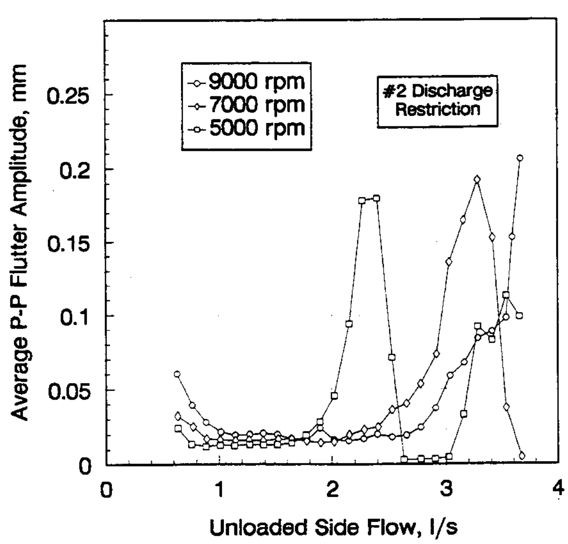

Fig. 5 Pad flutter vibration amplitude over a wider range of oil flows, and at three shaft speeds. 0.28 mm end clearance.

Due to the number of variables investigated, an extensive array of data was accumulated. Figures 3 through 12 present only a small portion of this data but give representative results, showing the effects of individual variables. The emphasis here is on the high flow flutter.

- Figure 5 shows the influence of flow rate and RPM. Note that these high flow vibrations develop at higher flow rates as the RPM is increased. And, that there is evidence of amplitude peaks at more than one flow value (at 5000 rpm in this plot), not considering the "low flow" vibrations. This data is for a partially restricted (#2 restriction) discharge. Four discharge path restriction levels were tested. The #0 condition is with the orifice plate in the tangential discharge path completely removed and represents the most "open" discharge practical in this test setup. The #3 condition is with a 15.9 mm (0.625 in.) diameter orifice (the most restricted discharge tested), the #1 condition is with a 22.2 mm (0.875 in.) diameter orifice, and the #2 condition is with a right angle turn in the flow path downstream of the tangential discharge (within a portion of the drain passage not seen in Fig. 1), but with no orifice plate.
- Figure 6 shows the static pressure in the annulus around the thrust collar for a specified flow rate and shaft rpm, for comparison of these four discharge conditions. Note the extent to which the right angle turn in the discharge flow path (#2 condition) creates back pressure. Discharge restriction was found to be a significant parameter influencing the development of both the high flow and low flow flutter.
- Figure 7 shows the influence of end clearance; again not well defined at the low flows, but changing the flow at which the amplitude peaks are reached when higher flows are used.
- Figure 8 shows the significant influence of the discharge restriction and identifies the design change with the most influence on eliminating the high flow flutter.
- Figure 9 shows the influence of total oil flow (sum of the flows to the loaded and unloaded sides). In the earlier

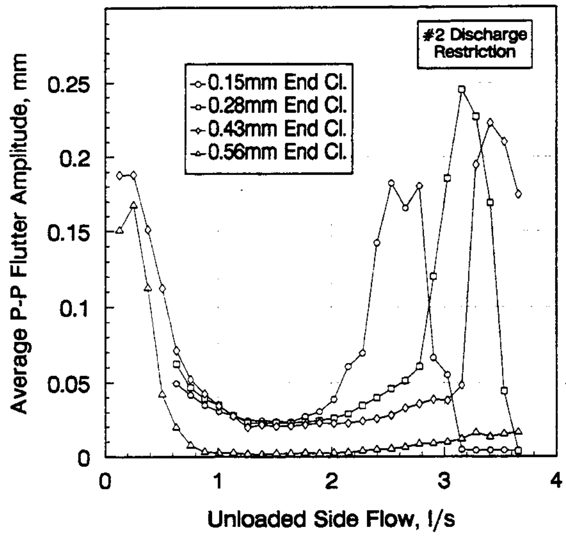

Fig. 7 Pad flutter vibration amplitudes over a wide range of flows, and four end clearance values. 7000 rpm.

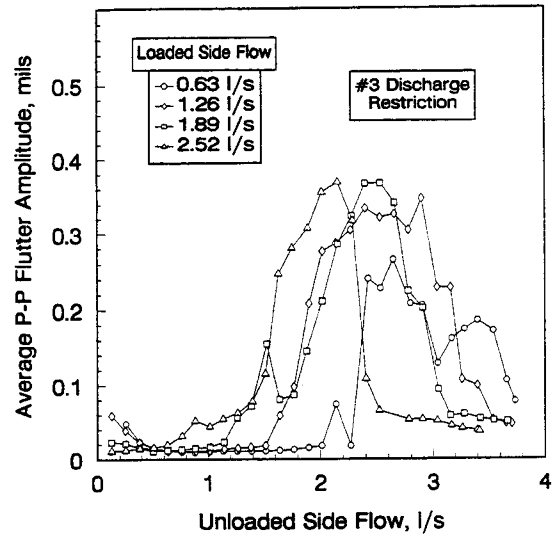

Fig. 9 Influence of loaded side flow on unloaded side pad flutter. 7000 rpm, 0.35 mm end clearance.

tests a constant flow of 1.89 l/s (30 gpm) was supplied to the loaded side thrust bearing. When it was shown that the degree of restriction in the oil discharge path was a significant parameter, tests were then run with different flows to the loaded side also, since the total oil flow was discharged through the one drain (Fig. 1). As shown here, changing the flow rate to the loaded side affects the flutter of the pads on the unloaded side.

As noted previously, one proximity probe was mounted to "view" one of the pads in the loaded side bearing. With no load applied, pad flutter could develop in this bearing also. However, with even small loads applied, dynamic motions disappeared.

Proximity probes were used at two pad locations in the unloaded bearing, and two pressure probes were mounted in the oil discharge annulus around the thrust collar. In both cases the outputs from each of the two pressure or proximity probes were similar, but often with some difference in amplitude. No particular significance was attached to these differences.

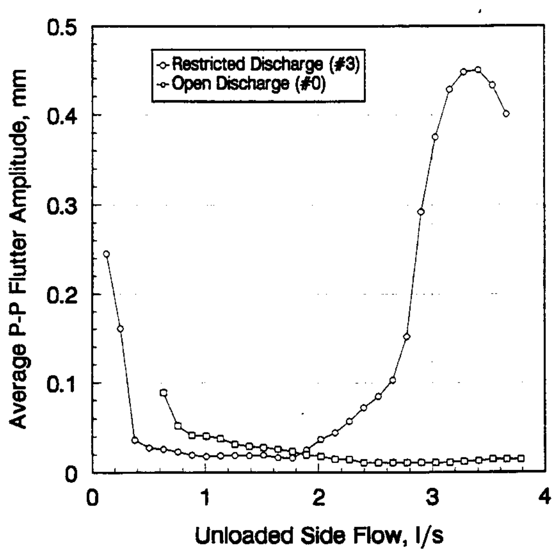

Fig. 8 Effect of discharge restriction on pad flutter amplitudes. 9000 rpm, 0.35 mm end clearance.

One other variable included in this test work was the type of thrust bearing. These bearing construction variations included:

- Center and offset pivoted pads.
- Pads with point and line type pivot contacts.
- Conventional, direct lubricated and pocket feed type thrust pads.
- Steel backed and copper-chrome backed thrust pads (all pads were babbitt faced).

None of these construction variables had any significant influence on the pad flutter.

The flutter amplitude data presented in Figs. 5, 7, 8, and 9 indicated that the high flow pad vibrations were possibly being driven by periodic pressure variations in the flow in the discharge annulus, and when the frequency of these variations (pulses) coincides with the natural frequency of the pad/oil film system, the high flutter amplitudes result.

A review of basic fluid flow technology, and specifically Blevins (1994) and Karassik et al. (1986), indicated that vortices were possibly being formed, within the oil flowing in the annular chamber surrounding the thrust collar, at the throat (cutwater) where this annular chamber connects to the tangential discharge. These vortices would be embedded within the oil being carried around by the rotation of the thrust collar and would be reflected as pressure pulsations at the pressure probes. These pressure pulsations would represent an oscillating force on the thrust pads as these vortices traveled around the annular chamber. If the oil flow rate was such that this annular chamber was not operating full, and all of the oil was essentially discharging through the tangential opening on its first passage, then the development of vortices would be curtailed.

Figure 10 is a plot of frequency spectrums from one of the proximity probes and one of the pressure probes, at the same test condition. The pad flutter frequency of about 9 Hz coincides with one of the two primary pressure pulsation frequencies. The other pressure frequency is about 75 Hz, which coincides with $\frac{1}{2}$ the shaft rotational speed.

Figure 11 is a second comparison of pad flutter and pressure pulsation frequencies, at a different set of operating conditions. The prominent frequencies are different from those in Fig. 10, but the patterns are much the same. The 58 Hz frequency is $\frac{1}{2}$ of the shaft rotational speed and the 4 Hz pressure frequency coincides with the pad flutter frequency.

In both of these figures, as in other data taken during these tests, the $\frac{1}{2}$ shaft rotation frequency is present in the pressure

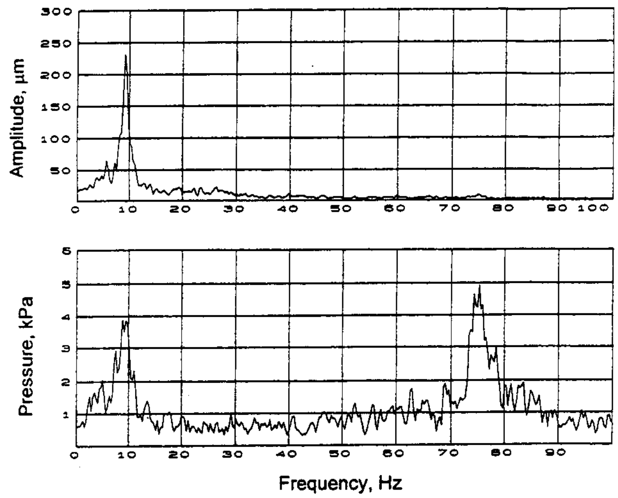

Fig. 10 Comparison of frequency spectrums from one pressure probe and one proximity probe. 9000 rpm, 3.5 l/s flow, 0.30 mm end clearance, #3 discharge restriction.

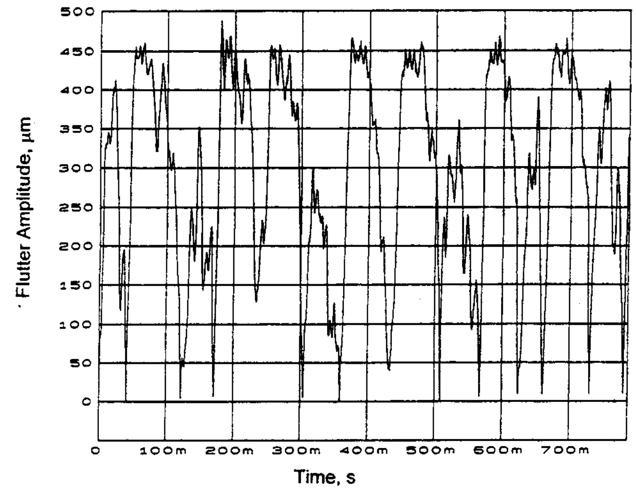

Fig. 12 Time trace of typical proximity probe flutter data. 9000 rpm, 0.30 mm end clearance, 3.5 l/s oil flow, #3 discharge restriction.

probe signals but does not result in pad vibrations at these same frequencies.

Figure 12 is a time plot of the pad flutter vibration (frequency spectrum shown in Fig. 10) and shows the unsteady nature of both the amplitude and the frequency.

As often is the case with test work of this nature, interesting questions are raised as more detailed data is obtained. Some understanding of these details is of value even though the basic question may have been answered. The correlation between the flutter frequency and the pressure pulsation frequency raises the question as to which is the fundamental action. That is: are the pads vibrating and causing the pressure pulsations or, are the pressure pulsations causing the pads to vibrate? The latter is believed to be the action taking place considering the following:

- Changing the oil discharge conditions downstream from the pads is the single most influential factor in the presence or absence of the high flow pad flutter. That is, with all else constant, a change remote from and downstream of the pads and "connected" to the pads only by the oil, changes the pad vibration.
- The other significant influence is the oil flow rate. Relatively small changes in the flow rate can result in large changes in the pad vibration amplitudes. Since the pads may be quite stable at lower and higher flows, then this specific flow rate must result in a disturbing force in tune with the pad/oil film natural frequency. This premise leads to the subject of vortex generation as mentioned above, and the Strouhal number, discussed below.
- If the pad vibrations were, in turn, causing the pressure pulsations, then it could be expected that the pressure pulsation frequency would reflect the number of pads in the bearing (8), or at least not be in a 1:1 ratio. (It might be assumed from Fig. 10 that this is the case, since the two predominant pressure frequencies are in a ratio of about 8. This does not hold true in Fig. 11, however, where this ratio is about 14.)
- A phase comparison of the low frequency pressure signals from the two pressure probes shows the 120 deg relationship that would be expected. This indicates the presence of a rotating pressure disturbance.

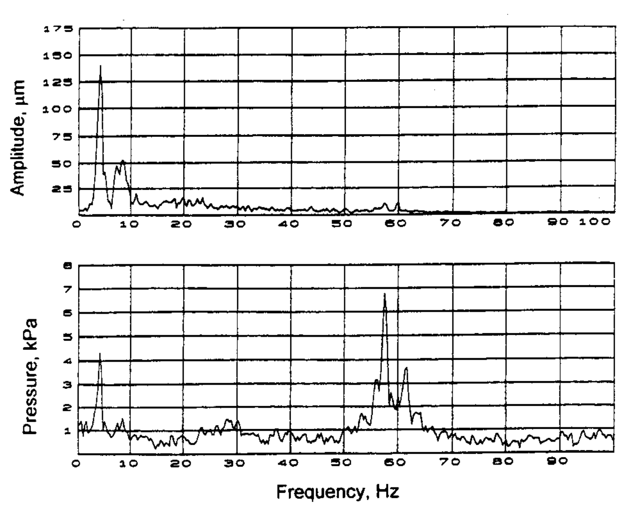

Fig. 11 Comparison of frequency spectrums from one pressure probe and one proximity probe. 7000 rpm, 3.8 l/s flow, 0.35 mm end clearance, #3 discharge restriction.

Blevins (1994) and Karassik et al. (1986) discuss vortex shedding frequency (f_s) as a function of the Strouhal number, the fluid velocity and a characteristic dimension (as a pipe diameter). The Strouhal number (S_n) is defined as:

$S_n = f_s \cdot D/V$, where;
f_s = predominant vortex shedding frequency, Hz
D = a characteristic dimension of the system, m (in)
V = velocity of the fluid, m/s (in/s)

Strouhal numbers are commonly in the range of 0.2 to 0.5, and are dependent on the geometry of the flow system, and to a lesser extent on the Reynolds number. Strouhal numbers have been determined for a number of the more common flow geometries (flow past a cylinder, for example). Although the geometry here is not a common one that has been investigated (as far as could be determined), it could be expected that a consistent Strouhal number would result if the proper values for V and D could be determined. It is not readily apparent what characteristic dimension should be used as the D value for calculating a Strouhal number for this geometry, and a determination of the correct oil velocity is even less apparent. Normally, the velocity value is that of the free stream approaching the obstruction (in this case the cutwater). Due to the interaction of the thrust collar with the oil flow being supplied by the external lubrication system, the range of oil velocities within these passages is high. No satisfactory set of D and V values could be found to provide consistent Strouhal numbers, but this is believed to be due to the highly three-dimensional nature of the flow field.

A second question of interest is the natural frequency of the pad/oil film system, and whether this agrees with the flutter frequencies found, particularly at the high flow amplitude peaks. The pad mass is readily determined but evaluating the stiffness of the oil film, as it would influence a rocking motion of the pad, was a task beyond the scope of the present work. A review of the pad flutter frequency data gives values from a low of about 3 Hz to a high of about 20 Hz. This covers the broad range of operating and geometry conditions tested.

Summary

The three primary goals outlined previously were met. Pad flutter was produced in the test thrust bearing, the significant influencing parameters were found, and means to eliminate the flutter were identified. The two primary conclusions from this work are:

1. An unrestricted, open discharge eliminated the high flow pad flutter, indicating that this phenomenon is related to the action of the oil in the discharge paths.
2. This unrestricted, open discharge did not eliminate the low flow pad flutter. Increased restriction of the discharge required lower flows for this low flow flutter to develop. This supports the concept of this being the result of incomplete (starved) oil films between the pad faces and the thrust collar.

A detailed analysis of all of the test data, in conjunction with an analytical study of the flow fields and pad characteristics, would be required to provide a complete, and possibly predictive, picture of this flutter phenomena. However, this work found two distinct modes of pad flutter, and means to control each were suggested and confirmed. Although these two means oppose each other, the unrestricted discharge to limit the high flow flutter is the choice since the oil flows to generate the low flow flutter were below those normally used in practice.

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