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THE EFFECT OF STARVATION ON THE DYNAMIC PROPERTIES OF TILTING PAD JOURNAL BEARINGS

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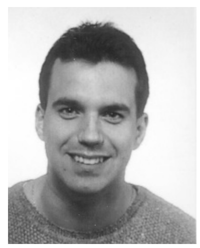

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ABSTRACT

A new 46 MW steam turbine became unstable as the load on the turbine was increased. The turbine had a relatively long slender shaft and the subsynchronous frequency was near the first critical speed of 1800 rpm. In an effort to allow the machine to reach full load a bearing redesign was undertaken. The bearing design was changed from the original 5 pad tilting pad journal (TPJ) bearings, orientated with the load on the bottom pad (LOP), to three pad TPJ's also in the LOP

configuration. Analysis indicated that the additional asymmetry in the dynamic coefficients would significantly increase the system logarithmic decrement (log dec) and allow the machine to run at full load without going unstable.

With the three pad bearings installed the machine was brought up in speed and experienced subsynchronous vibration before reaching full speed. A design review of the bearings indicated that the top two pads of the three pad bearing were operating in the fully starved condition; that is they were not receiving sufficient oil to develop a hydrodynamic oil film. As such the horizontal stiffness went to near zero.

This paper will present the results of the analysis work performed, including the starved bearing analysis, and the resulting redesign of the three pad bearing to run flooded. Operation as a flooded bearing allowed the machine to run fully loaded with no indications of unstable operation.

INTRODUCTION

Running hydrodynamic bearings in an evacuated condition can reduce power loss and pad operating temperatures. Evacuated, directed or leading edge lubrication implies similar lubrication mechanisms where oil is introduced to the pad leading edge via some sort of groove or spray nozzle arrangement, and excess oil (oil not going into the pad leading edge) is encouraged to leave the bearing to minimize churning losses.

Bielec and Leopard (1970) describe the results of testing performed with tilting pad thrust bearings and stated: "The use of directed lubrication in high-speed thrust assemblies achieves substantial reductions in power absorbed, at the same time reducing the pad surface temperature and increasing the film thickness."

Most bearing vendors employ one or more of the evacuated lubrication schemes described above for some of their thrust bearing offerings. As acceptance of this concept increased researchers investigated utilizing similar lubrication schemes in tilting pad journal bearings.

Tilting pad journal bearings that run evacuated are commonly designed without end seals and therefore the bearing shell is not flooded with oil. By severely limiting the flow of oil to a journal bearing unloaded pads in the bearing can run starved; that is there is insufficient oil available for every pad to run with a full film of oil from the pad leading to trailing edge. This is in stark contrast to thrust bearings where all pads theoretically are loaded evenly.

Tanaka (1991) was one of the first to discuss running TPJ's evacuated and presents test and analytical work to introduce the concept. He found that pad operating temperatures could be reduced and loaded pad oil film thicknesses increased with these lubrication schemes and that was - and has been - an important consideration with high speed turbomachinery. Tanaka did not address the starvation of unloaded pads in his work but did get good correlation with his pad temperatures, especially on the unloaded pads. His rig was static only and as such could not extract stiffness and damping coefficients and his theoretical results did not present dynamic coefficients.

Harangozo, et al. (1991) tested a TPJ with various lubrication schemes but again did not test or discuss dynamics. They reported pad temperature and power loss reductions are possible by utilizing evacuated designs. They also concluded that the benefits of running evacuated are not nearly as great as those achieved with thrust bearings.

Dmochowski, et al. (1993) and Brockwell et al, (1994) presented the first of several papers on the leading edge groove (LEG) journal bearing design. Here significant reductions in pad temperature were reported, attributed to the reduction in hot oil carryover. Also reported were reductions in power loss and the amount of oil required to be supplied to the bearing. They did mention: "Furthermore, there was no noticeable difference in the dynamic characteristics of the rotor between the flooded and leading edge groove bearing designs" but did not present any dynamic bearing data in the paper.

Edney, et al. (1996) and Edney, et al. (1998) discuss issues found with lightly loaded LEG journal bearings in high speed, light load applications. Steam turbines experienced unstable operation and higher than predicated first mode amplification factors when fitted with LEG TPJ's. They theorized that the low eccentricity ratios caused the pads to become unstable and solved the problem by introducing a profile after the groove. As with other work in evacuated TPJ's; they also reported lower pad temperatures, lower power loss and less oil required.

Nicholas (2003) presented a bearing design utilizing directed lubrication to reduce pad temperatures in high speed, heavily loaded bearings. These bearing designs utilize several design techniques to reduce pad temperatures, including the use of:

- Copper alloy pad materials
- Offset pivots
- Directed lubrication (running evacuated)
- "Spray Bar Blockers" (to reduce hot oil carryover)
- By-pass cooling where extra oil is introduced behind the pad to increase heat transfer.

Whereas a lot of static data was presented illustrating the successful application of these features in very high speed/high load applications, no specific bearing dynamic testing was performed.

Nicholas does however mention that "The low housing pressures created by evacuated housings can lead to oil starvation and subsynchronous vibration if improperly applied.” Here he discusses that not properly distributing the oil along the pad axial length may lead to select portions of the pad being starved, resulting in a dry friction rub and subsynchronous vibration.

Running evacuated bearings in at-speed balance facilities under vacuum conditions is also discussed. Here the vacuum can cause the oil to atomize as it leaves the orifices, thereby starving the entire bearing. This potential problem can be addressed by utilizing dummy end seals to ensure the bearing is flooded with oil when in a vacuum.

He (2003) and He, et al. (2005) describe how pad starvation can be accounted for in Thermohydrodynamic (THD) modeling of evacuated TPJ’s. Here not only are pad temperatures and bearing power loss addressed but bearing dynamic coefficients when running with some pads starved or partially starved are calculated. The THD calculations had very good correlation with pad temperature and bearing power loss but unfortunately there was no test data available to validate the model’s dynamic coefficients with bearings running partially starved.

Nicholas, et al. (2008) goes into great detail on starved bearing design and operation. In this paper there is a discussion of two case studies where a machine exhibited vibration problems that may or may not have been attributable to some pads running starved or partially starved of oil. The authors describe a methodology for calculating minimum oil flow requirements for evacuated bearings and methods to analyze the effect of starved operation on bearing dynamic properties.

Based upon investigation of two cases where vibration problems were experienced the authors determined that despite the fact that some pads ran starved, starvation was not the cause of the problems observed.

DeCamillo, et al. (2008) presents a tremendous amount of data of a low frequency, low amplitude, broadband subsynchronous vibration that they termed “Subsynchronous Vibration Hash” or “SSV Hash.” Based upon this work one could conclude that the SSV Hash phenomena is tied to evacuated bearing operation since flooding the bearing can eliminate the problem and the use of special grooves in the pads (to direct pad side leakage to the leading end of the next pad) was also found to solve the problem. Utilizing these special (patented) SSV Grooves allowed successful evacuated operation and the associated benefits of lower pad temperatures, lower power loss and reduced oil flow requirements.

He et al. (2012) show very good correlation with static test data for three different evacuated TPJ’s. The code utilized is a Thermoelastohydrodynamic (TEHD) tool that incorporates models for spray and groove lubrication schemes. The properties compared consist of pad temperatures and shaft centerline positions. They do not address dynamics but do state that “The confidence in the dynamic coefficients is intrinsically dependent on the accuracy of its corresponding steady state predictions.” As such accuracy in the static data is required before accuracy of the dynamic data can be achieved.

Since the loaded pads require the least amount of oil, due to their thin oil films, they typically run with full films while the unloaded top pads can run with various degrees of starvation. This can have a significant impact on the power loss in the bearing and is one of the main reasons bearings are designed to run evacuated.

Unfortunately most journal bearing computer programs operate under the assumption that each pad is supplied with enough oil to ensure the pad has a full lubricating film and the associated contribution to the overall bearing dynamics. Under most conditions this is actually a safe assumption as the top pads typically have a minimal impact on the coefficients.

The subject three pad bearings however, being in a load on pad configuration, had one loaded pad and two unloaded pads. As analyzed should the top pads not receive enough oil to develop a hydrodynamic oil film then only the one bottom pad will control the bearing coefficients. Of course being a tilting pad bearing the bottom pad can tilt and therefore cannot support horizontal forces and as such the horizontal stiffness and damping terms can go to zero. In reality nonlinearity comes into play as the journal vibrates within the bearing.

OPERATION WITH 5 PAD BEARINGS

Figure 1 is a drawing of the steam turbine. Note this turbine has two journal bearings and utilizes a flexible coupling to connect to a speed reducing gearbox which drives a generator at 1500 rpm. The turbine operates at 5,500 rpm and is rated at 46 MW. The “front” bearing is a 200 mm (7.874 in) diameter bearing located at the steam end of the turbine. This is a combination bearing that also contains the active and inactive thrust bearings.

The “rear” bearing has a 220 mm (8.661 in) bore and is located at the exhaust end of the turbine. The bearing span is 4160 mm (163.8 in) and the shaft diameter at the rotor center

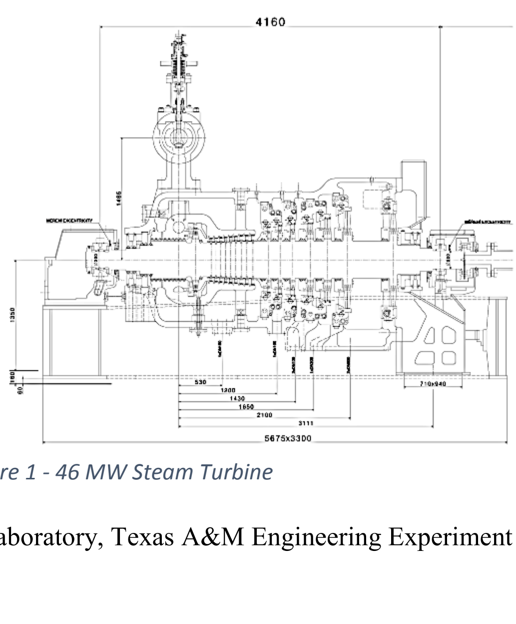

Figure 1 - 46 MW Steam Turbine

is about 435 mm (17.1 in) making the rotor L/D ratio about 9.5. Oil was ISO 46 nominally supplied at 1.5 barg and 43 ºC.

As originally supplied the turbine was fitted with 5 pad tilting pad journal bearings orientated in a load on pad (LOP) configuration. Ertas, et al. (2014) describes this turbine and the original run with the five pad bearings, discussing the sub synchronous vibration presenting itself as the machine is loaded.

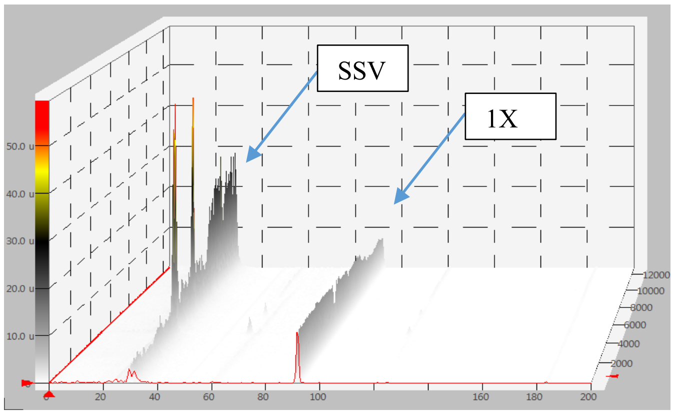

Figure 2 - DE Waterfall plot with 5 pad bearings

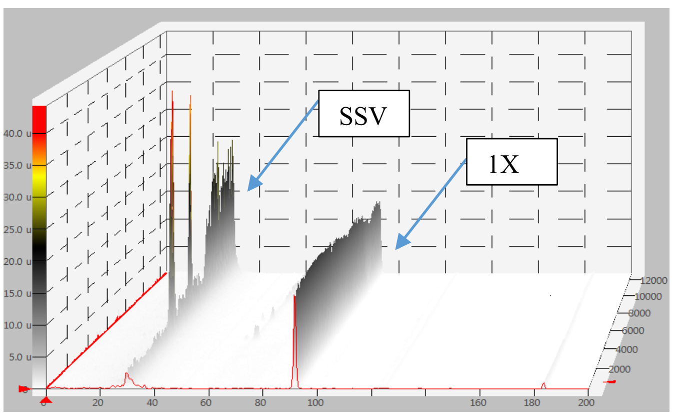

Figure 3 - NDE Waterfall plot with 5 pad bearings

Figures 2 and 3 are waterfall plots from these initial runs. These plots have vibration frequency on the horizontal axis, vibration levels (microns p-p) on the vertical axis and time going into the page. They are both at the synchronous running speed of 5500 rpm (92 Hz). Note the relatively constant synchronous vibration through the time period. Also note the low sub synchronous vibration for the first third or so of the plots, this represents the machine being loaded prior to the onset of the subsynchronous vibration.

As the load is applied the vibration levels are acceptable until about the 35 MW point. That is the first peak observed in the waterfall plots, with this sudden increase in vibration the load was reduced and the vibration subsided. The machine was then loaded again and once again at 35 MW the vibration spiked, and once again the vibration reduced when the load was reduced. Further attempts to slowly load the machine reintroduced the sub synchronous vibration. It became apparent that cross coupled forces, associated with the loading of the turbine, where driving the machine unstable.

As detailed in Ertas, et al. (2014) the cross coupled forces generated in the steam packing was driving the machine unstable, vibrating at its first natural frequency (1800 rpm with the 5 pad bearings). The work done here was completed independently of Ertas, indeed it was done in parallel to fast track to a solution, and we found that an API 617 level II (API, 2014) analysis predicts a log dec of -0.199 at 1826 cpm utilizing full bearing and aerodynamic coefficients (figure 4).

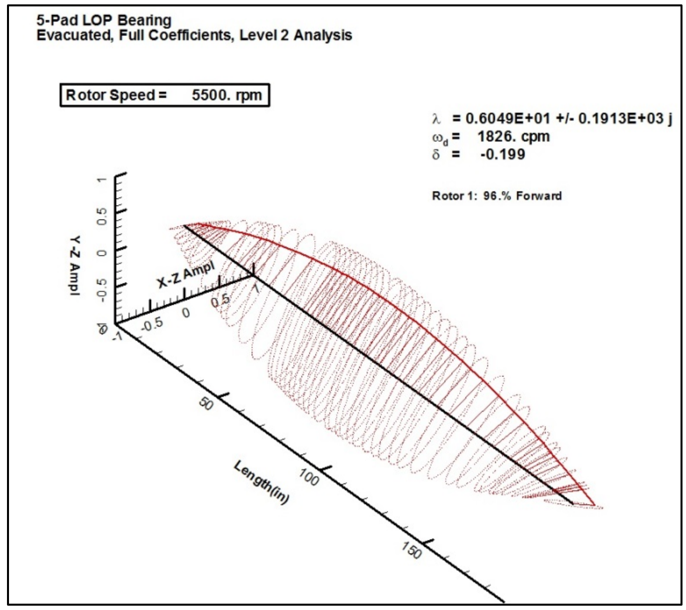

Figure 4 - Stability analysis with 5 pad bearings

REDESIGN WITH THREE PAD BEARINGS

Based upon these findings it was decided to investigate other bearing geometries that could possibly increase the system stability to avoid this sub synchronous operation. During this investigation it was determined that a three pad load on pad bearing would increase the log dec to acceptable levels throughout the load range (from -0.2 with the 5 pad bearings to +0.2 with the three pad bearings). This could be accomplished due to the very high stiffness asymmetry in a bearing design such as this.

Figures 5 and 6 are drawings of the two, three pad journal bearings and tables 1 and 2 contain details of the bearing geometries.

As the drawings indicate these two bearings utilize spray bars in between the pads to supply oil directly at the leading edge of each pad. The bearings also have open end seals allowing the bearing to run evacuated (not flooded).

The bearings were installed and the machine started. During the runs the turbine experienced sub synchronous vibration before even reaching full (synchronous) speed of 5,500 rpm.

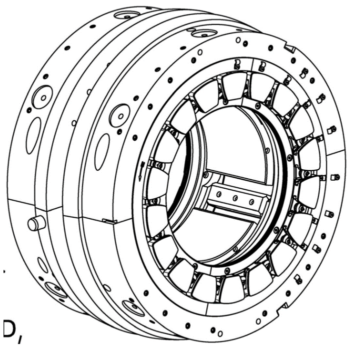

Figure 5 - Thrust end (front, Steam or NDE) three pad TPJ

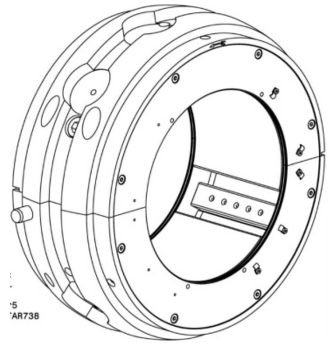

Figure 6 - Plain end (rear, Exhaust or DE) three pad TPJ

- 200 mm bore (7.87")
- 108 mm Pad Length (4.25")
- 410 mm fit OD (16.14")
- 28.3 kN load (6360 lbs)
  - 1.3 MPa unit load (190 psi)
- 57.7 m/s (190 fps) at 5500 rpm
- Clearance 0.445 mm (.0175") on diameter
- Pad arc length: 90 degrees
- Preload:
  - 0.12 bottom pad
  - 0.35 top pads

Table 1 Thrust (Steam) end bearing details

- 220 mm bore (8.68")
- 134 mm Pad Length (5.25")
- 374 mm fit OD (14.72")
- 35.2 kN load (7910 lbs)
  1.2 MPa unit load (173 psi)
- 63.5 m/s (208 fps) at 5500 rpm
- Clearance 0.490 mm (.0190")
- Pad arc length: 90 degrees
- Preload:
  - 0.12 bottom pad
  - 0.34 top pads

Table 2 Plain (Exhaust) end bearing details

Figure 7 is a plot of data from the run-up and subsequent run-down. As shown the machine was run at 3000 rpm for half an hour and then was ramped up in speed. Shortly after increasing the speed the overall vibration at both bearing locations increased significantly. However note that the subsynchronous vibration is not growing and the levels are of the same order as the synchronous vibration.

The magnitude of the vibration was such that the operators tripped the machine at 5300 rpm. The coast down was uneventful and the vibration levels returned to acceptable

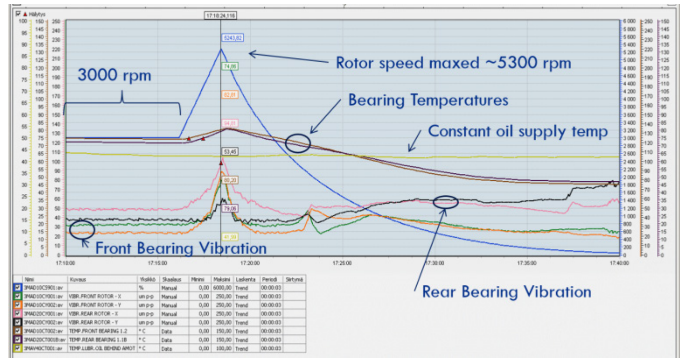

Figure 7 - Run up and run down with evacuated three pad bearings

levels shortly after the trip. Several attempts to start the machine resulted in similar issues.

Analysis of the vibration signature indicated that the vibration consisted of both synchronous and sub synchronous vibration. Note that there is no load on the machine so aerodynamic cross coupled forces are near zero.

Figure 8 is a plot of the vibration spectrum of the startup as observed by one of the radial probes. Note that at the maximum attained speed of 5300 rpm (88 Hz) the sub synchronous component is at about 40% of running speed (or about 35 Hz, 2100 cpm). Also note that the sub synchronous vibration initiated at about 4800 rpm (80 Hz) and subsided on coast down again at about 4800 rpm. Lastly note that at 5300

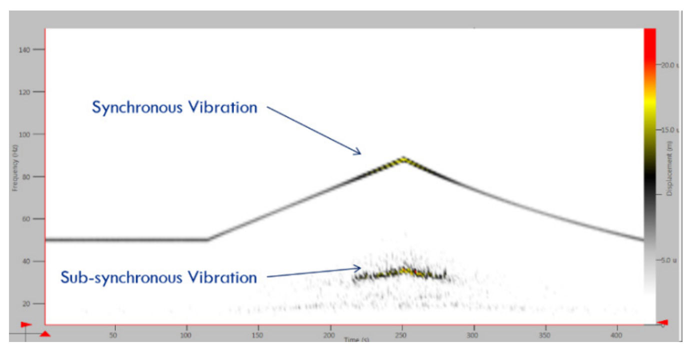

Figure 8 - Vibration spectrum of data in figure 7

rpm the sub synchronous component is of the same magnitude as the synchronous component.

DESIGN AUDIT OF THE THREE PAD BEARINGS

As the rotordynamic analysis indicated that the turbine should not be going unstable with the three pad bearings the accuracy of the coefficients used in the analysis was investigated. It was found that standard industry accepted analysis tools all supplied coefficients that when applied to the analysis indicated that the machine should not be going unstable.

Detailed analysis of the bearing design and oil flow was then undertaken. It is widely known that bearings running in an evacuated situation function satisfactorily with some pads starved or partially starved of oil. That is; there is not enough oil being supplied to these pads to allow them to fully develop an oil film. As such their contribution to the overall bearing stiffness and damping is diminished.

The starved pads are typically the upper or unloaded pads as these have larger oil film thicknesses and therefore require more oil to satisfy the full film requirement. Loaded pads have much thinner hydrodynamic oil films, require less oil flowing into the pad leading edge and therefore almost always run with a full film of oil.

Normally with 4-pad load between pad bearings or 5-pad bearings (either load on pad or load between pads) having some of the top pads partially starved has a minimal impact on the overall dynamic coefficients. This is because at least two pads are supporting the load and therefore have thinner films and the associated reduced oil flow requirement. This would also hold true for a three pad load between pad bearing as the bottom two pads would share the load.

A three pad load on pad bearing is different due to the fact that only the one bottom pad is loaded; the top two pads are unloaded. As such the top pads require a significant amount of oil to be fully lubricated.

One of the bearing analysis tools, as detailed in He (2003), has a model to analyze bearings that are potentially running with some or all pads partially or fully starved.

Starved Bearing Analysis

The modelling described in this section comes from He (2003) further described in He et al. (2005), the program utilized is MAXBRG as found in He et al. (2003). Figure 9 represents the film pressure profile for a typical sleeve bearing that is supplied with sufficient oil to satisfy the oil film requirements. Pressure begins to develop at the zero degree location and builds up to the maximum pressure, dropping to zero at the point of minimum oil film thickness. The widely used Reynolds boundary condition assumes the pressure past the minimum oil film thickness is zero for analysis purposes.

However, when the bearing is not supplied enough oil to fill the gap between the bearing and the journal, there is a starved condition.

Figure 10 illustrates the model used (rotation is left to right). On the right side of this figure we see streamers that are assumed to exist downstream of the point of minimum oil film thickness. Here, since the gap between the pad and journal is diverging, the program assumes “air” is pulled in and the pressure is assumed to be ambient.

To the left side of figure 10 a similar assumption is made when there is not enough oil supplied to fill the converging gap. In the example shown there comes a point where the oil supplied does fill the gap and a continuous film is developed. This is the only portion of the bearing that is assumed to generate hydrodynamic pressure. The analysis performs an iterative calculation to determine when the onset of this continuous film exists.

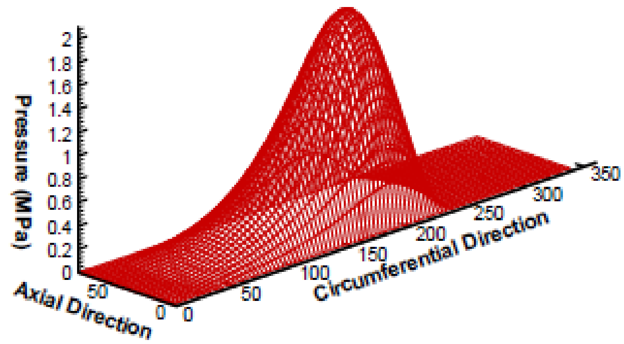

Figure 9 - Representative film pressure for flooded bearing - Courtesy He (2003)

Figure 10 - Assumed streamers at leading edge due to inadequate oil flow (streamer downstream of minimum film also shown) - Courtesy He (2003)

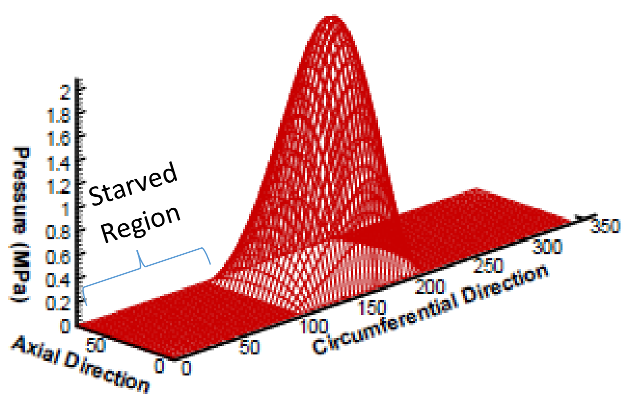

Figure 11 - Oil film pressure distribution for partially starved pad - (Courtesy He (2003)

Figure 11 is the resulting pressure profile for this example case where the continuous film develops downstream of the leading edge of the pad (at around the 100 degree location).

When there is insufficient oil to fill the gap the pressure is assumed ambient and when the gap is small enough to be filled with the available oil then the hydrodynamic pressure begins to develop.

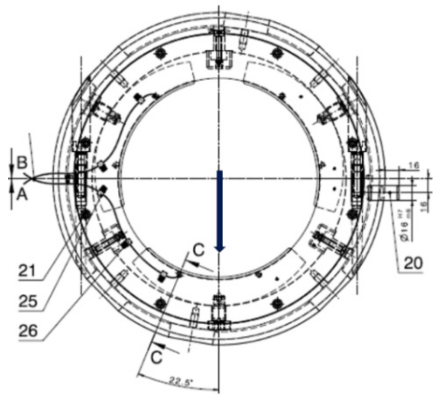

Figure 12 - Three pad bearing

Should the required film be such that there is not enough oil supplied to fill any portion of the pad, the pad is assumed to be fully starved and does not contribute at all to the bearing dynamics (zero stiffness and damping contribution).

When the above analysis was applied to the subject three pad bearings it was found that the two top pads in both bearings were not supplied enough oil to generate film pressure. As such only the one pad at bottom dead center contributed to the bearing dynamics.

As shown in figure 12 these three pad bearings have only a single pad with the pivot point in line with the load on the bearing. As such when the top two pads are fully starved then the bearing horizontal stiffness and damping go to zero.

Figures 13 and 14 summarize results from three different bearing analyses. The green lines are the horizontal stiffness and damping coefficients originally used in the rotor dynamic analysis, they assume all pads have a fully developed hydrodynamic oil film (flooded). The blue lines are from MAXBRG when configured to assume the bearing is flooded, and the red lines are MAXBRG results with the starvation model mentioned previously.

First, the original or supplied analysis presents nice smooth data through the speed range. The discontinuity in the MAXBRG flooded analysis (blue lines) is due to the fact that even flooded the top pads don't contribute to the dynamics until the speed is high enough for them to generate a hydrodynamic film.

The difference between original and flooded MAXBRG is attributed to differences in solution algorithms. Analysis of these differences is beyond the scope of this paper. The important thing to note in figures 13 and 14 are the results from the starved analysis, where the code is predicting the pads never get enough oil to generate a continuous film and therefore contribute virtually nothing to the overall bearing stiffness and damping terms.

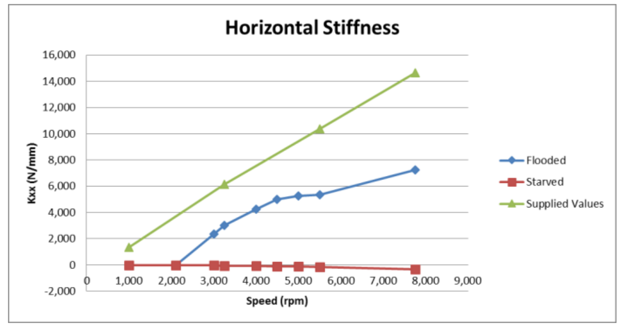

Figure 13 - Three pad bearing Horizontal Stiffness

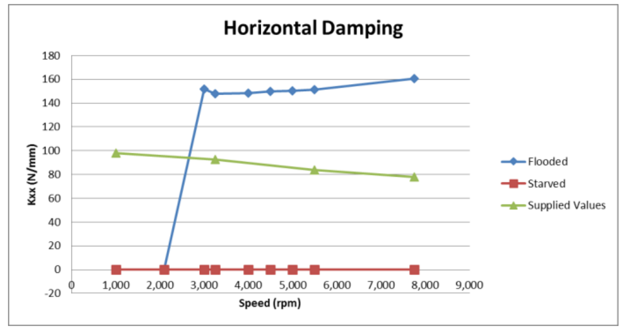

Figure 14 - Three pad bearing horizontal damping

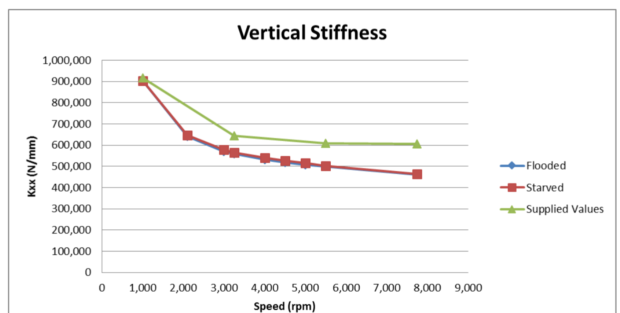

Figure 15 – Three pad bearing vertical stiffness

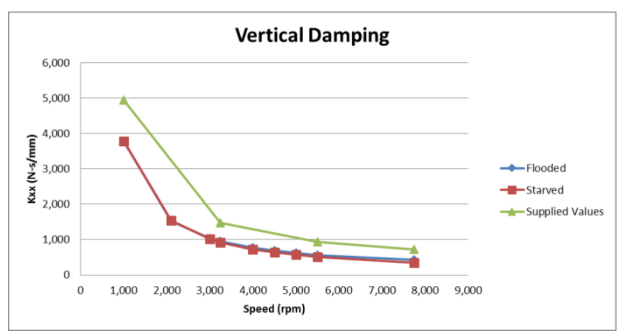

Figure 16 - Three pad bearing vertical damping

Figures 15 and 16 are the vertical coefficients. Here the original data is very close to the MAXBRG data and the MAXBRG flooded and starved analyses are virtually identical. For these coefficients the codes are correctly assuming that the one bottom pad has a full continuous oil film and therefore is not running starved. The vertical

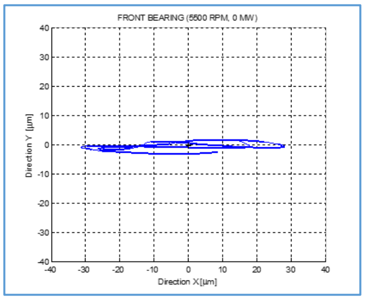

Figure 17 – 0.4X Filtered Orbit running at 5500 rpm (92 Hz)

coefficients are dominated by the only pad supporting any load at bottom dead center.

Figure 17 is an orbit filtered at 0.4X from the front bearing at 5500 rpm (92 Hz), it is nearly flat indicating that the horizontal coefficients are much smaller than the vertical.

To further confirm the starved bearing analysis bearing temperatures were recorded and compared to calculated values as detailed in table 3:

Speed	Top Pad Actual/MAXBRG	Bottom Pad Actual/MAXBRG
3000 rpm	—	74 ^{o}C/71 ^{o}C
5500 rpm	63 ^{o}C/64 ^{o}C	83 ^{o}C/83 ^{o}C

Table 3- Pad temperatures actual and as predicted

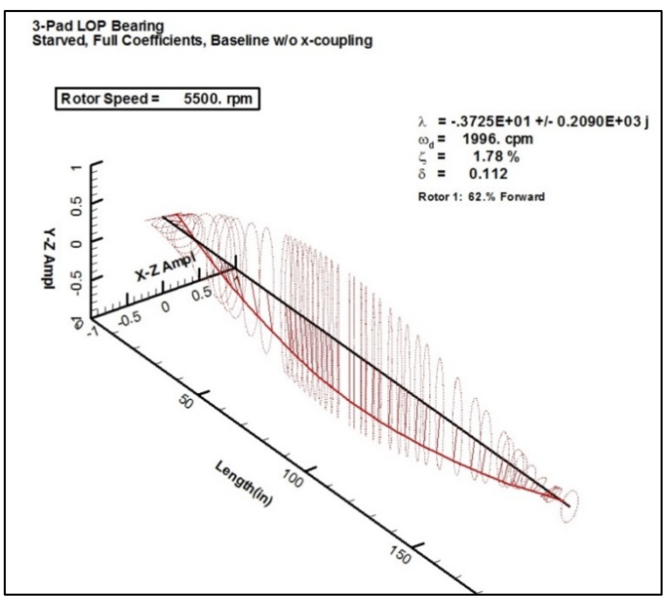

Figure 18- Stability analysis with three pad evacuated bearings

With the rotor-bearing system analyzed with three pad evacuated coefficients (accounting for the top two pads in the fully starved condition) and no aerodynamic cross coupling, the Log Dec is predicted to be +0.112 (figure 18).

Redesign to flooded operation

Based upon the determination that the top pads were fully starved the decision was made to modify the bearings to run in a flooded condition. This involved passing more oil flow to the bearings and adding end seals to trap oil in the bearing shell. As such, since the bearing is fully flooded with oil, the pad leading edges will have sufficient oil available to ensure the pads do not run starved.

With the top pads contributing the system stability now follows the predicted results with the calculated log dec of +0.198 (figure 19). Note that these calculations were performed with no aerodynamic cross coupled coefficients, representing the unloaded case.

Figure 20 is a plot of the turbine being loaded to 46 MW with no indications of subsynchronous vibration with the three pad bearings installed to run in the flooded condition. The four turbine radial vibration probes are plotted as well as one vibration reading from the generator and the turbine power.

The four probes reported overall vibration levels at full load of about 30 microns peak to peak (1.2 mils P-P). With the flooded three pad bearings the turbine was able to run to full speed and load.

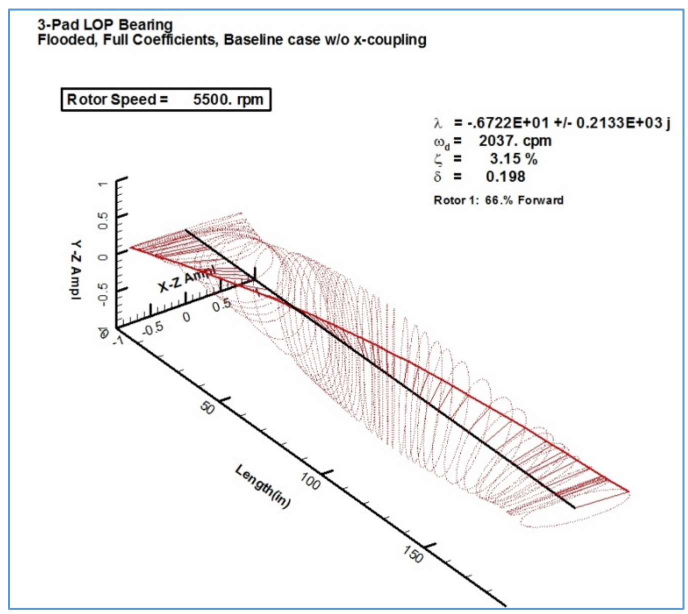

Figure 19- Stability at no-load with flooded three pad bearings

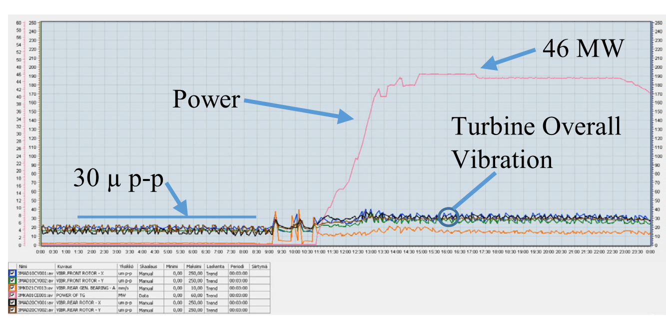

Figure 20 - Resulting full load run - no sub sync vibration

DISCUSSION

As mentioned earlier, when the three pad evacuated bearing experienced SSV, the turbine was unloaded. Therefore, it is likely little to no destabilizing forces were present from components such as the annular seals. This, combined with the fact that the SSV did not grow with time to very high levels approaching the bearing clearance, suggest the rotor mode was very lightly damped, but still "stable."

Rotordynamic predictions seem to support this stable, but lightly damped conclusion in two ways. First, the mode's base log decrement is predicted to be relatively low at +0.112 at 1996 cpm, a frequency which correlates well with the measured SSV frequency. Furthermore, the analysis indicates the machine is relatively robust to destabilizing forces, i.e., it takes a significant amount of cross-coupled stiffness to drive the mode from +0.1 log decrement to a negative, unstable value. Predictions indicate a similar robustness for the flooded bearing design, but more significantly the base log dec becomes +0.198, nearly a 100 percent increase from the evacuated level.

With the mode being so lightly damped with the evacuated bearings, it is susceptible to small forcing excitations. Such small excitations could be occurring within the bearings themselves due to the likely flow disturbances and pad vibrations created by the starved upper pads. DeCamillo et al. (2008) show that these flow induced pad vibrations occur in this subsynchronous region of concern.

It is also suspected that significant nonlinearity may exist within a starved bearing. As this rotor vibrates back and forth horizontally, some portions of the top pads may activate (create force on the rotor) and deactivate as the film goes from being starved to developed. Such nonlinear behavior is not accounted for in a conventional steady-state bearing analysis.

Note that pivot and support stiffness models were not used in these analyses. A check of the effect of support stiffness presented a reduction from +0.112 to +0.095 (a 15 percent reduction) when comparing infinitely stiff support structure to a five million lbf/in support stiffness model. Including pivot stiffness would present an even lower log dec.

CONCLUSIONS

Further work could be done to investigate starved pad influences on bearing dynamics but certainly design consideration should include ensuring the unloaded pads are receiving enough oil to at least partially develop a hydrodynamic oil film. The other option would be to design these journal bearings to run in the flooded condition to eliminate these issues all together (or course at the cost of increased pad temperature, increased power loss and the need for additional oil flow).

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