VIBRATION REDUCTION OF LARGE STEAM TURBINE GENERATOR THROUGH BEARING ANALYSIS/MODIFICATION

by

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ABSTRACT

Large utility steam turbines have experienced very high vibration levels on their generator exciter bearings. Units where the exciter rotor is supported by a single bearing are particularly susceptible. This synchronous vibration, measured as high as 14 mils (pk/pk), is typically unresponsive to balancing attempts. Vibration levels of this magnitude can lead to bearing or bearing support damage and higher scrutiny by insurers. Exciter and generator rotors are direct coupled, but differ significantly in mass. As the generator is loaded, it grows vertically, drawing the smaller exciter rotor with it. If adequate loading of the exciter journal bearing has not been established during generator to exciter alignment, the bearing becomes unloaded, affecting its stiffness and damping properties. These changes affect the exciter's vibration response and balance characteristics. Bearing modifications can enhance bearing stiffness and damping properties, reduce the effect of the exciter rotor's radial growth, and improve balancing characteristics.

INTRODUCTION

Duke Power operates several large nuclear and fossil powered electrical generation stations. Large steam turbine generator sets, such as the McGuire Nuclear Station unit depicted in Figure 1, provide base loaded service for the company. The generator, driven by a high pressure turbine and three low pressure turbines, is initially energized by an exciter that is rigidly coupled to the generator. Following major generator maintenance in the late 1980s, a high vibration condition was observed in the McGuire Unit 2 generator exciter. This exciter design (Figure 2) provides for only one support bearing; located on the outboard end of the exciter rotor. The inboard end is rigidly coupled to the generator, supported vertically by the generator rotor only. Exciter and generator rotors differ significantly in mass. The generator rotor weighs approximately 195 tons and has 32 in journals; the exciter weighs only 15 tons and has 12 in journals. As the unit is started, vertical movement of the generator rotor (due to oil wedge development, along with thermal growth) draws the smaller exciter rotor up with it. If adequate loading of the exciter journal bearing is not established during generator to exciter alignment, the bearing can easily lose its static load. Traditionally, exciter bearing static load is provided by an intentional elevation of the exciter bearing relative to the generator bearings. Ideally, this provides for increased static load at startup and adequate load at full speed and load.

![Figure 1. 1125 MW Turbine/Generator Set.](image)
Maintenance on the exciter or generator may change the relative positions of the generator and exciter bearings, affecting the static load on the exciter bearing. If the exciter becomes unloaded or lightly loaded, its stiffness and damping properties change. These changes affect the exciter rotor's vibration response and sensitivity to imbalance. Influence coefficients (balance sensitivity measurements) calculated from previous runs may no longer be valid. Vibration levels at the exciter bearing may vary greatly with slight turbine-generator process variations (pressures, flows, temperatures, load, etc.).

While adequate exciter bearing load may typically be addressed by alignment procedures, this method is somewhat unforgiving. Bearing modifications that would provide improved stiffness and damping properties, even in unloaded or lightly loaded bearing conditions, could potentially "desensitize" the exciter to slight variations in exciter to generator alignment. Analysis of rotor vibration characteristics and journal movement in the bearing motion can confirm the unloading of the bearing. Rotordynamic modelling of the exciter rotor and bearing can be used to assess the effect of various bearing modifications (clearance or preload variations, etc.). Appropriate bearing modifications should provide more stable bearing stiffness and damping characteristics and improve balancing efforts.

FIELD ANALYSIS

Vibration Characteristics

Outage work performed in 1987 on the 1125 MW nuclear turbine/generator unit, required the replacement of the generator rotor. The exciter rotor, directly coupled to the generator rotor, was not replaced. This smaller rotor is supported on its outboard end by a four pad, tilting pad bearing (load between pad) with no pad preload. The nominal bearing clearance on the 12 in journal, is 15 mils (diametral). Vibration sensors on the exciter consist of displacement probes indicating shaft vibration relative to the bearing housing and bearing housing absolute vibration. The exciter bearing and probe arrangement are shown in Figure 3. The turbine/generator supplier's standard generator to exciter rotor alignment procedure was followed during reassembly. This procedure requires that the exciter bedplate be slightly elevated relative to the generator, deflecting the exciter rotor and providing additional load on the exciter bearing. The alignment procedure was performed and exciter reassembly was completed.

Upon restart, vibration levels on the exciter were considerably higher than before the generator rotor replacement. Attempts to balance the exciter using earlier "effective" data were unsuccessful. Levels remained higher during the next several fuel cycles. During 1991, exciter vibration levels reached as high as 14 mils (pk/pk).
The vibration orbit of the exciter bearing journal is indicated in Figure 6. As depicted by this orbit, the X probe on the Unit 2 exciter has traditionally exhibited 40 percent to 50 percent higher levels than the Y probe.

![Figure 6. Exciter Rotor Orbit at Full Speed and Full Load.](image)

**Bearing Characteristics**

In addition to vibration measurements, the displacement probes provide “gap” information that indicates journal travel in the bearings. This measurement is useful in determining the relative loading of the bearing. If the journal remains relatively low within the bearing clearance, a more heavily loaded bearing is indicated. Journal travel into the center or upper half of the bearing indicates light loading or upward loading of the bearing.

The rotor centerline positions for the generator rotor “turbine end” bearing and “exciter end” bearing during startup are indicated in Figures 7 and 8. These bearings are of elliptical design. The position of the journal centerline is indicated at slow roll (on turning gear) and at several intermediate speeds as it approaches a normal operating speed of 1800 rpm. Moderate loading of the generator bearings is indicated, per Figures 7 and 8, as the journal center moves up within the clearance, but not as far as the center. In contrast, the journal in the exciter bearing (Figure 9) moves completely up through the center of the bearing, actually into the upper half. Note, also, that the journal moves mostly vertically in the exciter bearing, whereas the generator journal moves vertically and horizontally. This is primarily the result of “cross-coupling” forces generated in the elliptical bearings that are, by design, absent in the tilting pad exciter bearing. The potential for low loads is the most likely reason for the manufacturer’s use of a tilting pad bearing on the exciter. All other bearings on the turbine/generator train are of elliptical design.

Analysis of displacement probe data from the exciter bearing indicates a propensity for the exciter bearing to become unloaded as it reaches full running speed. The manufacturer concurs that unloading of the bearing can occur, resulting in higher vibration levels and associated balancing difficulties. Their position on this problem has been that it is primarily the result of misalignment, soft foot, or poor bearing fit.

**ANALYTICAL MODELLING**

**Reproducing Field Conditions**

The tilting pad exciter bearing was modelled using different clearance, loading, and lubricant conditions. As discussed by Nicholas and Wygant [1], adequate dynamic characterization of a

![Figure 7. Shaft Centerline Plots for Generator Rotor Bearings (Turbine Side).](image)

![Figure 8. Shaft Centerline Plot for Generator Bearings (Exciter Side).](image)

![Figure 9. Shaft Centerline Plot for Exciter Bearing.](image)

tilting pad journal bearing should include pad pivot stiffness along with pad thermal and elastic deformation effects. Bearing pedestal
stiffness should also be estimated. This stiffness along with damping (from the oil film) must be adequately combined to determine the effective stiffness and damping of the bearing [2]. Pad pivot stiffness estimation is a somewhat complex issue, as it is determined by the contact of a cylindrical surface (pad back) with a spherical surface (pad pivot). Deformations (and associated stiffness), governed by Hertzian contract stress theory, can be estimated for various contact conditions [3, 4]. Stiffness for this analysis was calculated for high and low load conditions, per manufacturer’s design data. Pivot stiffness of 5.5E6 lb/in for high loads (15,000 lb), and 3.4E6 lb/in for low loads (3000 lb), were used.

Exciter bearing pedestal stiffness (bearing shell to ground stiffness) was calculated from experimentally derived modal data. Stiffness values of 3.2 E6 lb/in (horizontal) and 4.3 E6 lb/in (vertical) were used, having been calculated at 30 Hz (running speed) from the experimental data.

Stiffness and damping coefficients of the generator and exciter bearings, along with bearing housing stiffness, were used as input into a synchronous response rotordynamic analysis [5]. This type analysis generates predicted responses of the rotor to specified imbalance forces. Varying bearing conditions were assessed through their effect on the stiffness and damping properties. Indicated in Table 1 are the effects of bearing load reduction on bearing dynamic properties and on the predicted synchronous vibration. Loads are shown at a nominal 15,000 lb condition and for a 3000 lb (unloaded) condition. A 750 oz-in static imbalance was used for all cases. Horizontal vibration is presented, as it represented the maximum of the two directions calculated.

**Table 1: Effects of Bearing Load Reduction on Exciter Vibration for Various Clearances Conditions (Nominal Clearance Is 15 Mils/Diametral).**

<table>
<thead>
<tr>
<th>Load (lb)</th>
<th>Assembled Clearance (mil) (mil-diametral)</th>
<th>Preload (oz)</th>
<th>Stiffness (horiz.) (lb/in)</th>
<th>Damping (horiz.) (lb-s/in)</th>
<th>Vibration (horiz.) (mil/pk-pk)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15,000</td>
<td>15</td>
<td>0</td>
<td>4.5E6</td>
<td>1.2E5</td>
<td>3.0</td>
</tr>
<tr>
<td>3,000</td>
<td>15</td>
<td>0</td>
<td>7.2E5</td>
<td>7.8E3</td>
<td>5.0</td>
</tr>
<tr>
<td>15,000</td>
<td>20</td>
<td>0</td>
<td>3.7E5</td>
<td>9.5E3</td>
<td>5.1</td>
</tr>
<tr>
<td>3,000</td>
<td>20</td>
<td>0</td>
<td>5.5E5</td>
<td>5.4E5</td>
<td>10.0</td>
</tr>
<tr>
<td>15,000</td>
<td>24</td>
<td>0</td>
<td>2.1E6</td>
<td>8.7E3</td>
<td>6.0</td>
</tr>
<tr>
<td>3,000</td>
<td>24</td>
<td>0</td>
<td>2.7E5</td>
<td>3.5E3</td>
<td>11.0</td>
</tr>
</tbody>
</table>

The effects of these variations on exciter rotor synchronous vibration are displayed in Figures 10 through 15. Horizontal and vertical exciter journal vibration for standard bearing conditions (15 mil clearance and a 15,000 lb load) and a static exciter imbalance (750 oz-in) are indicated in Figures 10 and 11. The predicted response at 1800 rpm is approximately three mils (pk/pk) at the bearing. As static load is dropped to 3000 lb, the vibration level increases at running speed increase to nearly five mils (pk/pk) per Figures 12 and 13. Vibration levels of nine mils (pk/pk) are predicted, per Figures 14 and 15, if excessive clearance (two times nominal) and low loads are included.

As mentioned previously, a resonant mode of the exciter rotor exists near 2100 rpm. This mode, with a standard load and tight clearance, is predicted in the 2200 to 2300 rpm. The mode shape is shown in Figure 16. With reduced bearing stiffness (caused by lower bearing load and increased clearance) the mode shape changes considerably, taking more of a cantilever shape (Figure 17).

The synchronous response predictions indicate that for a given imbalance, the exciter journal vibration will increase considerably, as bearing load is reduced and bearing clearance is enlarged. This appears to be a combination of two effects:

- The proximity of running speed to an exciter resonant mode
- The general sensitivity of the mode to bearing stiffness and damping changes
Figure 13. Predicted Vertical Exciter Vibration (1X) For Standard Bearing at Low Load MNS Steam Turbine-Generator Probe and #11 Bearing x Relative Displacement.

Figure 14. Predicted Horizontal Exciter Vibration (1X) For High Bearing Clearance and Low Bearing Load MNS Steam Turbine-Generator Probe and #11 Bearing x Relative Displacement.

Figure 15. Predicted Vertical Exciter Vibration (1X) For High Bearing Clearance and Low Bearing Load MNS Steam Turbine-Generator Probe and #11 Bearing x Relative Displacement.

Figure 16. Exciter Mode at 2100 RPM for Nominal Exciter Bearing.

Figure 17. Exciter Mode at 1900 RPM for Bearing with High Clearance and Low Load.

Identifying Potential Solutions

To investigate potential improvements, bearing geometry variations were analyzed. Of primary interest were the effects of bearing clearance and pad preload on bearing properties, since these parameters are relatively easy to change on the actual bearing. The "assembled" clearance is the actual minimum clearance of the journal and bearing as it exists in the machine. Pad "machined" clearance is the difference between the pad bore diameter and the journal diameter. Pad preload is an indication of the difference in pad curvature and journal curvature and is calculated as:

\[ m = 1 - \frac{c}{c_b} \]  

where:
- \( m \) is the pad preload (dimensionless),
- \( c \) is the pad machined clearance (mils), and
- \( c_b \) is the bearing assembled clearance (mils).

Indicated in Table 2 are the predicted effects of diametral clearance (and pad preload) on bearing properties and predicted vibration levels. Assembled clearance \( (c_b) \) is varied from 8 to 24 mils. Pad machined clearance \( (c) \) is held constant at 15 mils (diametral). In general, tighter clearance and increased preload produced improved stiffness and damping properties, even at low bearing loads. This reduction in assembled bearing clearance resulted in a predicted vibration reduction of more than two mils, when compared to standard clearance case; along with a nine mil reduction, compared to the high clearance case. Of course, bearing film and babbitt temperatures increase with reducing clearances, as do bearing power loss. These effects had to be adequately assessed as well.

Table 2: Effects of Bearing Clearance and Pad Preload on Exciter Vibration for Low Load Condition (3000 Lb).

<table>
<thead>
<tr>
<th>Load (lb)</th>
<th>Assembled Clearance ((c_b)) (mils-diametral)</th>
<th>Preload ((m))</th>
<th>Stiffness ((\text{thrust}, \text{in}))</th>
<th>Damping ((\text{thrust}, \text{in-s}^{-1}))</th>
<th>Vibration ((\text{thrust}, \text{in-s}^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>3000</td>
<td>10</td>
<td>0.13</td>
<td>3.6E6</td>
<td>2.2E3</td>
<td>2.6</td>
</tr>
<tr>
<td>3000</td>
<td>15 (Nom)</td>
<td>0</td>
<td>7.2E5</td>
<td>7.8E3</td>
<td>5.8</td>
</tr>
<tr>
<td>3000</td>
<td>17</td>
<td>0</td>
<td>6.3E5</td>
<td>6.7E3</td>
<td>9.0</td>
</tr>
<tr>
<td>3000</td>
<td>20</td>
<td>0</td>
<td>5.7E5</td>
<td>5.4E3</td>
<td>10.0</td>
</tr>
<tr>
<td>3000</td>
<td>24</td>
<td>0</td>
<td>2.7E5</td>
<td>3.5E3</td>
<td>11.0</td>
</tr>
</tbody>
</table>

In addition to this evaluation, an independent assessment of bearing dynamic characteristics was obtained from a bearing consultant. Results were compared to these analyses findings prior to the rotodynamic evaluation.

EXCITER MODIFICATION

Based on the results of the analysis, the exciter bearing was modified to incorporate an 11 mil assembled clearance and a 0.27
preload. Pad curvature was assessed by "blue checking" its fit to a properly turned mandrel. Pads were scraped, as necessary, to get proper curvature. The assembled clearance of 11 mils was obtained by adjusting pad support pins. Film and babbit temperature estimates were acceptable for this clearance.

Effective bearing stiffness and damping are also dependent on having adequately stiff supports below the bearing (shell to ground interface). Improved film characteristics can be rendered ineffective if the bearing is mounted on an overly flexible pedestal. To improve bearing pedestal stiffness, bearing shell to pedestal fit was changed from a clearance fit to a slight interference fit. Also, the exciter baseplate was reground to improve support rigidity.

MODIFICATION RESULTS

Before the restart of the unit, all balance weights were removed from the exciter fan. The unit was brought up to speed with exciter levels quite low (2 mils pk/pk) at full running speed (no load). However, as the unit was loaded, the exciter reached vibration levels similar to those observed before the modification (12 mils pk/pk, as compared to 14 mils pk/pk). After reaching full load, however, a balance shot was calculated based on nearly forty year old effect data. Prior to the modification, the unit had been unresponsive to balance attempts. This time, a balance shot successfully reduced the vibration to four mils (pk/pk). This shot was nearly the same weight and location as that used to balance the exciter in 1988. The unit has run in the three to five mil range since 1992. Vibration trends are shown in Figure 18.

CONCLUSIONS

Steam turbine generators, with only one support bearing for the exciter rotor, present a particular challenge for establishing and maintaining an adequate exciter bearing load. The turbine generator manufacturer has developed and specified alignment procedures to provide for adequate bearing load under varying load conditions. Maintenance efforts on the exciter or generator, settling of the baseplate, or bearing wear can affect the exciter journal position and result in an unloading of the exciter bearing. This results in varying bearing stiffness and damping properties that affect the response of the unit to imbalance forces. Analytical methods indicate that by reducing the exciter bearing clearance and incorporating pad preload, more uniform bearing dynamic characteristics can be maintained over a wider range of bearing loads. This allows for more predictable exciter rotor balancing. Care must be taken to ensure that bearing babbitt temperatures are in acceptable ranges with the tighter clearances.

While the authors agree with the manufacturer that the unloaded bearing condition may be the result of some deviation in alignment or assembly process for the exciter, it is believed that the proposed bearing modifications provide an improved margin for acceptable alignment and assembly. The bearing modification has been used successfully on one 1125 MW generator exciter and a similar modification is planned for the sister unit in early 1997.

REFERENCES


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