APPLYING MODAL ANALYSIS TO IMPROVE MACHINERY RELIABILITY

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Abstract: This paper discusses the benefits of using experimental modal analysis (EMA) and operating deflection shape (ODS) analysis to better understand rotating machinery vibration problems. We explore structural dynamics that are difficult to visualize using standard vibration analysis techniques, and show how EMA & ODS analysis can help resolve those problems. We also show how accurate and detailed models are needed to fully capture a complex structure's dynamic activity, and how combining ODS and transient speed vibration analysis can be used as an effective modal analysis substitute on turbo-machinery. The author also introduces the new concept of a Non-Operating ODS.

Keywords: Modal analysis; operating deflection shape; ODS; vibration; reliability.

Introduction: Experimental modal analysis (EMA) and operating deflection shape (ODS) analysis have been used for many years to help solve vibration problems with both rotating equipment and static structural elements. Today's compact, light weight data collection hardware coupled with efficient modeling software allows analysts to easily gather large amounts of data and accurately model complex deflection patterns. These dynamic models provide a visual representation of structural motion that analysts can use to determine how to modify a structure to control vibration, reduce stress and fatigue, and to improve reliability.

This paper specifically discusses some benefits of EMA & ODS analysis as applied to rotating machinery to better understand the dynamics of machinery problems that were difficult to visualize using standard vibration analysis techniques. We show how accurate and detailed models helped to fully capture complex dynamic activity and can be used to determine the proper corrective actions, and how combining ODS and transient speed vibration analysis can be used as an effective modal analysis substitute on turbo-machinery. The author also introduces the new concept of a Non-Operating ODS, whereby effective ODS analyses can be performed using background vibration from adjacent machinery as an excitation source.

Case History #1 – An AC Generator Vibration Problem: A paper mill's cogeneration plant was experiencing excessive bearing cap seismic vibration on a 100 MW, 3,600 rpm AC generator. Vibration velocity levels on the inboard (drive-end) bearing ranged from 0.7 to 1.1 in/sec-pk in the axial direction during normal operation, and posed a serious reliability and safety concern for the plant. Power production was also compromised at times, as operators would vary or reduce load while trying to keep vibration to lower levels. In addition to the high bearing cap vibration, there was considerable flooring vibration in various areas, and structural vibration of the generator enclosure's side panels.

Plant engineers were concerned about the vibration severity, but due to plant electrical generation requirements they were unable to have the machine shutdown for inspection. Our company was contacted for consultation, and a ODS study was suggested as a first step to better understanding the unit's dynamic characteristics.

ODS Model Construction: The ODS study was conducted during February 2010. A detailed, scaled structural model was built using Vibrant Technology's ME'scopeVES software as shown in Figure 1.

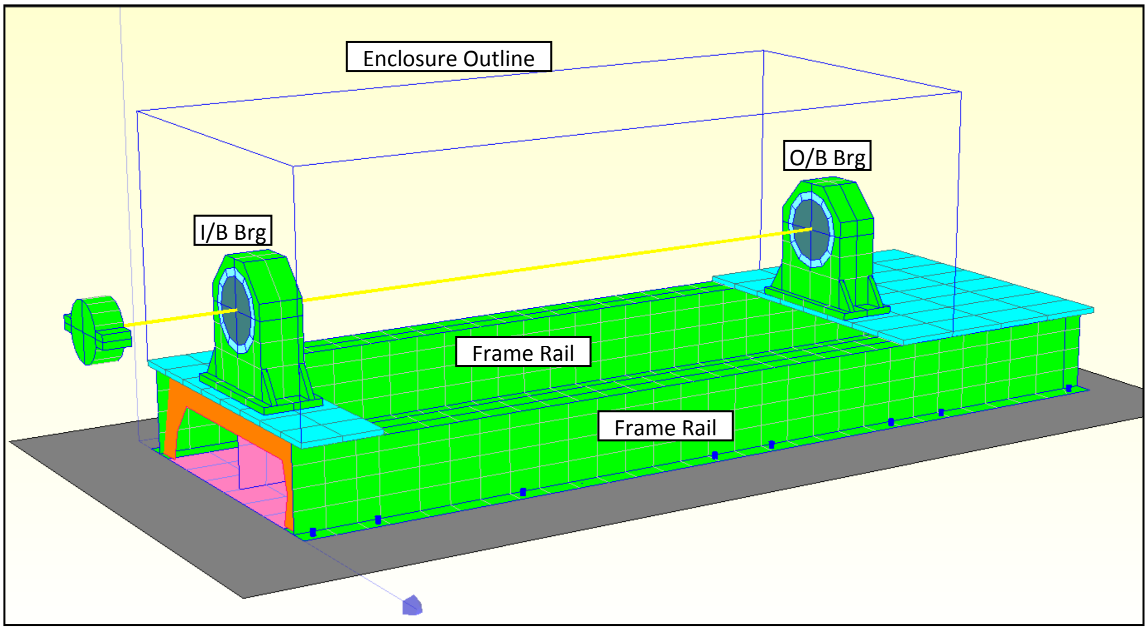

Figure 1: Generator ODS Structural Model.

Two supporting frame rails form the generator's enclosure base and run the full length of each side. Each frame rail is a large I-beam section, approximately 30" tall. A series of hold-down bolts are used along the length of the rail to fasten the generator to the concrete floor. Shim packs are used beneath each hold-down bolt to adjust the generator shaft alignment to the turbine, and to adjust frame loading at each bolt location to account for variations in flooring height.

A truss-shaped steel cross-plate connects the frame rails at each end and provides vertical support for the bearings pedestals. The span between bearing pedestal is 245". A large horizontal steel plate rests on top of the rails and the cross-plate each end of the frame, providing a mounting surface for the bearing pedestals. A vertical plate mounted directly beneath each bearing pedestals provides additional vertical stiffness.

On the left side of Figure 1 the turbine's inboard bearing housing is shown; it was used to provide an indication of motion across the coupling. On the right side there is also an exciter assembly (not shown) for the generator. The generator stator enclosure (outlined) rests on top of the frame rails, and is approximately 8'6" tall by 10' wide.

Modeling Considerations: During model construction, each major structure or surface is broken into a mesh pattern, as seen in Figures 1 and 2. Vibration measurement points are chosen at key mesh intersections on the structure, and provide the basis for the final animations. Each measurement point is assigned a location number that identifies all data for that point. At points where no vibration is sampled, the point's characteristics are derived by interpolating the data from the two or more adjacent sampled points. Figure 2 shows the generator inboard bearing area and the various points used in that area.

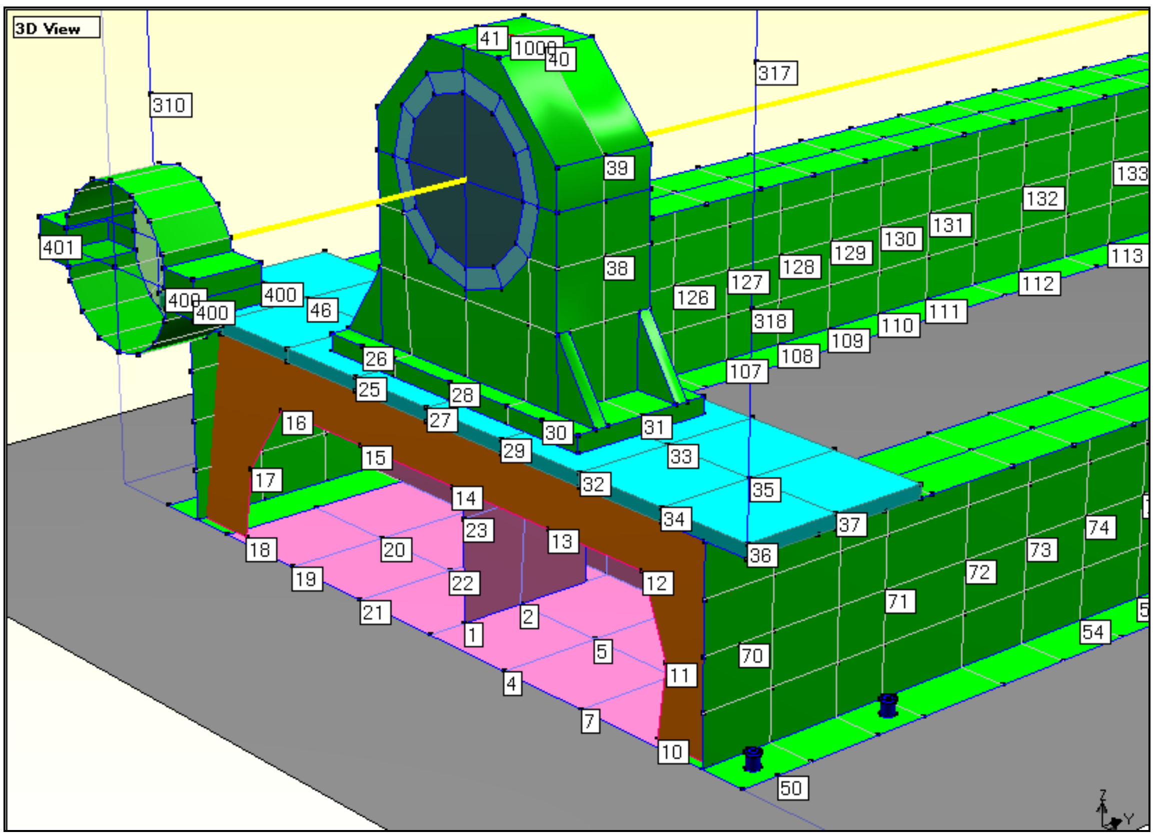

Figure 2: Generator Inboard Bearing Sampling Locations.

Determining the proper mesh density and selecting vibration measurement points requires experience and an understanding of structural vibration. If a coarse pattern with relatively sample few points is used, the model may not sufficiently capture the various bending modes of interest and it may be difficult to properly identify corrective actions. For example, if a structure is too flexible and additional members are needed to stiffen it, a coarse model may miss a significant bending mode is missed, and extra supports may actually amplify the vibration or they may have little to no effect, resulting in a trial and error solution.

We can see in Figure 2 that relatively fewer points were sampled on the bearing pedestal, while a significant number of points were used on the supporting framework. Since the pedestal is essentially rigid in comparison to the framework, it will move in rigid-body manner and fewer points need to be sampled to accurately portray the motion. Conversely, the steel plates in the supporting structures have relatively thin cross-sections in relation to their width and will display a variety of bending modes. It can be difficult to predict the resulting mode shapes of thin or flexible sections prior to conducting an initial ODS model. Because of the significant time involved in collecting data for even a coarse model on a large structure such as this, we find it more productive to construct our initial models with relatively high density. By sampling vibration from the flexible sections at fairly close physical distances, we can produce an accurate ODS animation using the first set of data with relatively little increase in total project time. If we suspect there are closely coupled modes or repeated roots that require further definition, it is easy to sample a few additional points and add them to the model.

After constructing a model we must first establish a reliable “driving point.” The driving point, or reference point, is a vibration measurement locations that remains fixed throughout the ODS data collection process. It is used as the cross-reference to calculate the Frequency Response Function (FRF) between it and every response location sampled on the structure, also known as the Degrees of Freedom, or DOFs; a DOF is simply a measurement point & direction.

Since the driving point DOF is used as the cross-reference for every FRF, proper driving point selection is important if modes are to be properly captured. One primary decision is choosing whether to use a single driving point or multiple driving points. In general, using single a reference will yield mode shapes with better definition (more DOFS) than if multiple references are used [1]. So, from the standpoint of defining mode shapes with the maximum number of DOFs, single reference testing should be preferred. However, if closely coupled modes or repeated roots are suspected, multiple driving points may be required for accurate mode shape identification.

Selecting a single driving point can be a challenge because we must identify all modes in the desired frequency range from a single point. On large or complex structures it may be difficult to find a point that reasonably shows activity from the entire structure. To help with this task, we can measure driving point FRFs, which have the source and excitation at the same DOF. We then examine and select the driving point FRF with the maximum number of strong (large magnitude) resonance peaks as potential references [2]. If no single driving point FRF contains all of the peaks, we must find two or more driving point FRFs that contain all of the peaks. These FRFs then determine the number & location of the references. As a final consideration, it is important that the driving point reference DOFs are not at or near nodal points of the mode shapes of interest, as this will result in that mode being poorly captured or missed entirely.

Data Analysis: After completing modeling some preliminary vibration readings were acquired around the machine to identify the specific frequencies that were present and their relative vibration amplitudes. It was quickly apparent the dominant vibration response was occurring at the 1st order of the generator's operating speed, 3,600 cpm (i.e., 1X). There were almost equally strong responses at many locations at the 2X frequency, 7,200 cpm. Additional running speed harmonics were also seen at 3X, 4X, 5X and 6X, with 4X being the dominant component of this group. In terms of relative vibration amplitudes the generator inboard bearing cap at DOF #1000 (see Figure 2) showed the strongest 1X and 2X responses. Based on this preliminary data, driving point FRFs were calculated for DOF #1000 and several other locations, and DOF #1000 was chosen as the best overall driving point.

A tri-axial accelerometer was then used to acquire cross-channel measurements by roving around the machine and referencing each acquired DOF to the driving point accelerometer at DOF #1000; Figure 3 shows a typical cross-spectra. To minimize the effects of background noise and extraneous vibration, 8 linear sample averages were taken at each DOF.

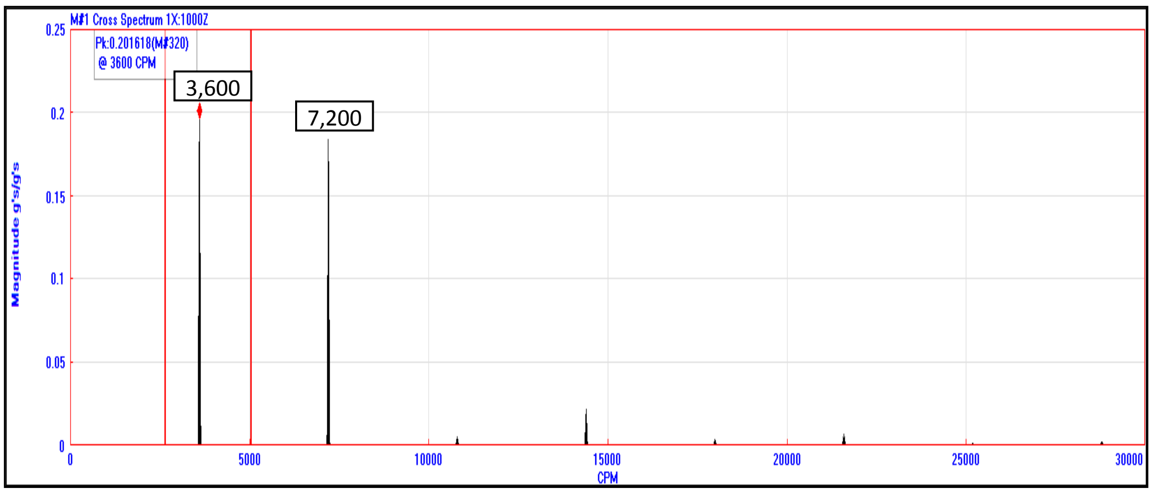

Figure 3. Generator Cross-Spectra Results.

Cross-spectra from the entire machine showed the two predominant ODS responses were at 1X and 2X (3,600 & 7,200 cpm). Relevant activity was also seen at 4X (14,400 cpm). After identifying these modes interpolation equations were created to populate the non-sampled structure mesh points with calculated data. Interpolation was extended across the nearest six points to ensure a reasonably smooth response shape while still capturing any acute localized vibration. The model was then animated to determine the ODS at each specific frequency.

1X Response Mode: The ODS animations indicated that 1X motion was significant on the frame rails, bearing pedestals, and enclosure, with several interesting characteristics.

The high axial vibration on the inboard bearing noted by the customer was distinctly visible. The ODS model showed the pedestal moving in a combined axial and side-to-side pattern as shown in Figure 4. The inboard and outboard bearing pedestals were also noted to be moving out-of-phase with each other in the axial direction, which can be seen in Figure 6. (Please note that the ODS amplitudes shown in the graphics are amplified for animation purposes.)

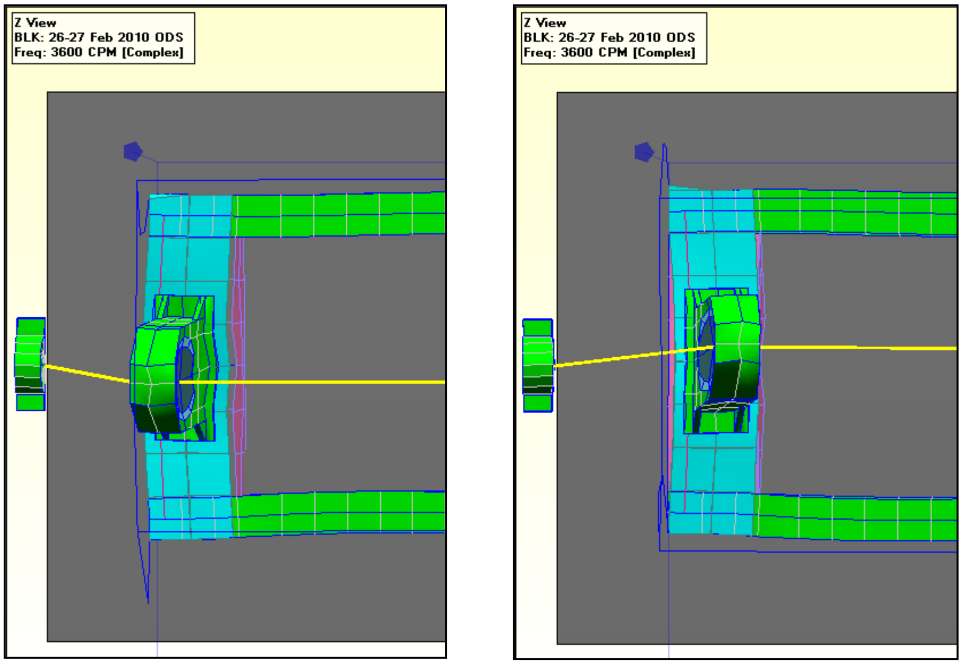

Figure 4. Generator Inboard Bearing, 1X ODS, Top View.

Figure 5 shows the inboard bearing as viewed looking from the coupling toward the generator. The 1X ODS responses showed the right-side frame rail having considerable vertical movement at the corner indicating either a loose hold-down bolt or a soft-foot condition. In conjunction with the frame rail movement, the supporting cross-plates experienced significant vertical deflection. The combination of the frame rail movement and cross-plate deflection resulted in the high 1X vibration seen at the top of the bearing pedestal.

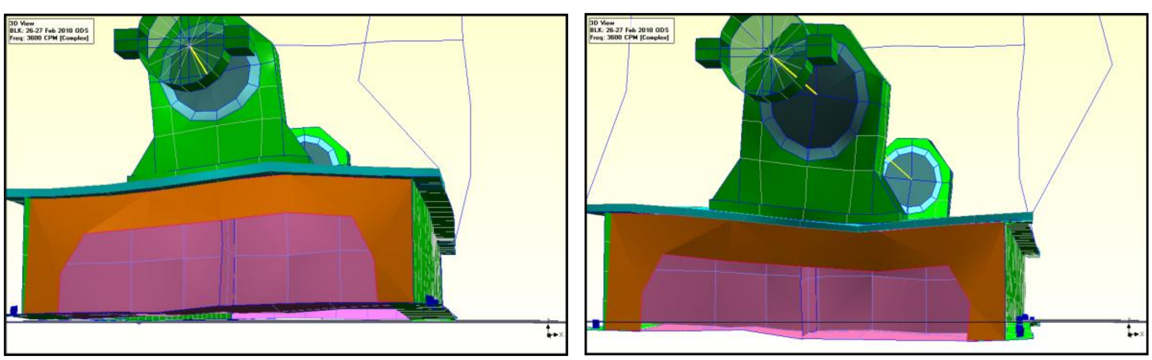

Figure 5. Generator Inboard Bearing & Support Frame, 1X ODS.

Figure 6 shows the 1X ODS responses looking from the right-side of the generator. The axial movement of both bearing pedestals is clearly seen to be out-of-phase, with significantly more movement on the inboard bearing. We also see the significant frame rail movement at the coupling-end (left side of Figure 6). The center portion of the frame rail appeared firmly affixed to the floor, with very little vertical movement. Some vertical movement of the frame rail was also noted in the plane of the outboard bearing. Overall, the combined 1X ODS effect on the frame rail was to produce a vertical bending mode with little movement at the center of the beam, and higher motion on both ends. Interestingly, the turbine inboard bearing showed very little motion in any direction. This indicated the generator vibration was not being driven by the turbine, but was due to conditions specific to the generator.

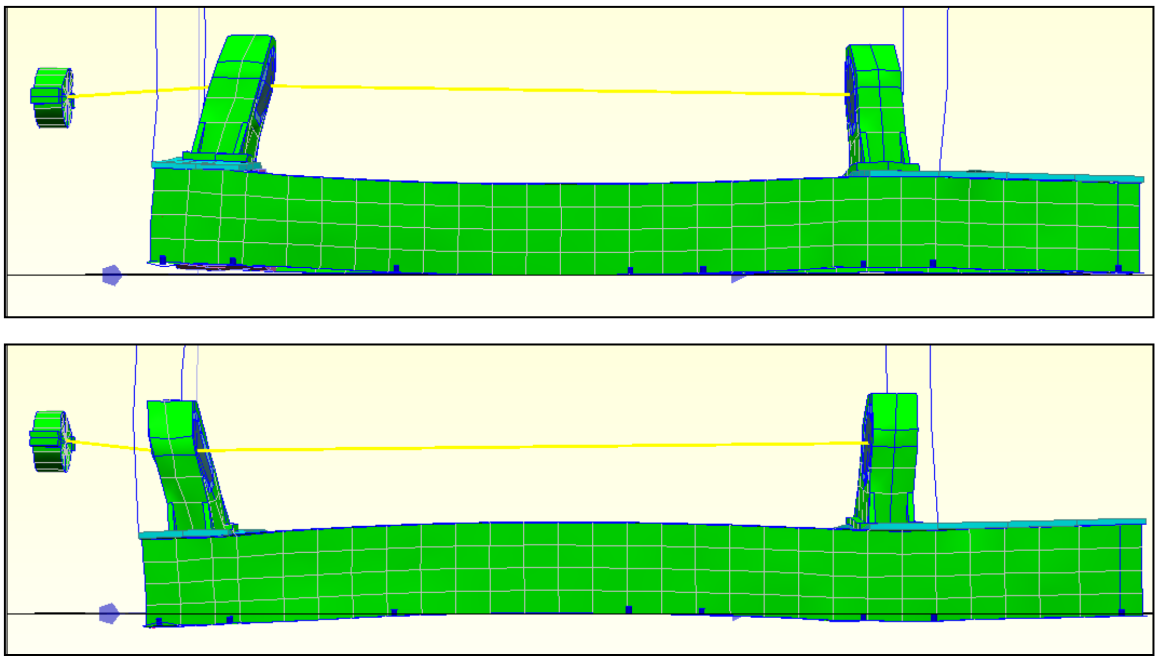

Figure 6. Generator Right-side Frame Rail, 1X ODS.

2X Response Mode: The 2X ODS responses at 7,200 cpm revealed a distinct lateral flexure of both frame rails, as shown in Figure 7, with a maximum deflection at the center of the rails. This motion was directly related to the generator stator vibration occurring at 2X-line frequency.

Stator vibration and winding insulation condition had been an ongoing concern at the plant, and the 2X ODS indicated the frame was allowing significant flexure at this frequency, aggravating the typical 2X-line frequency vibration commonly seen on 2-pole generators.

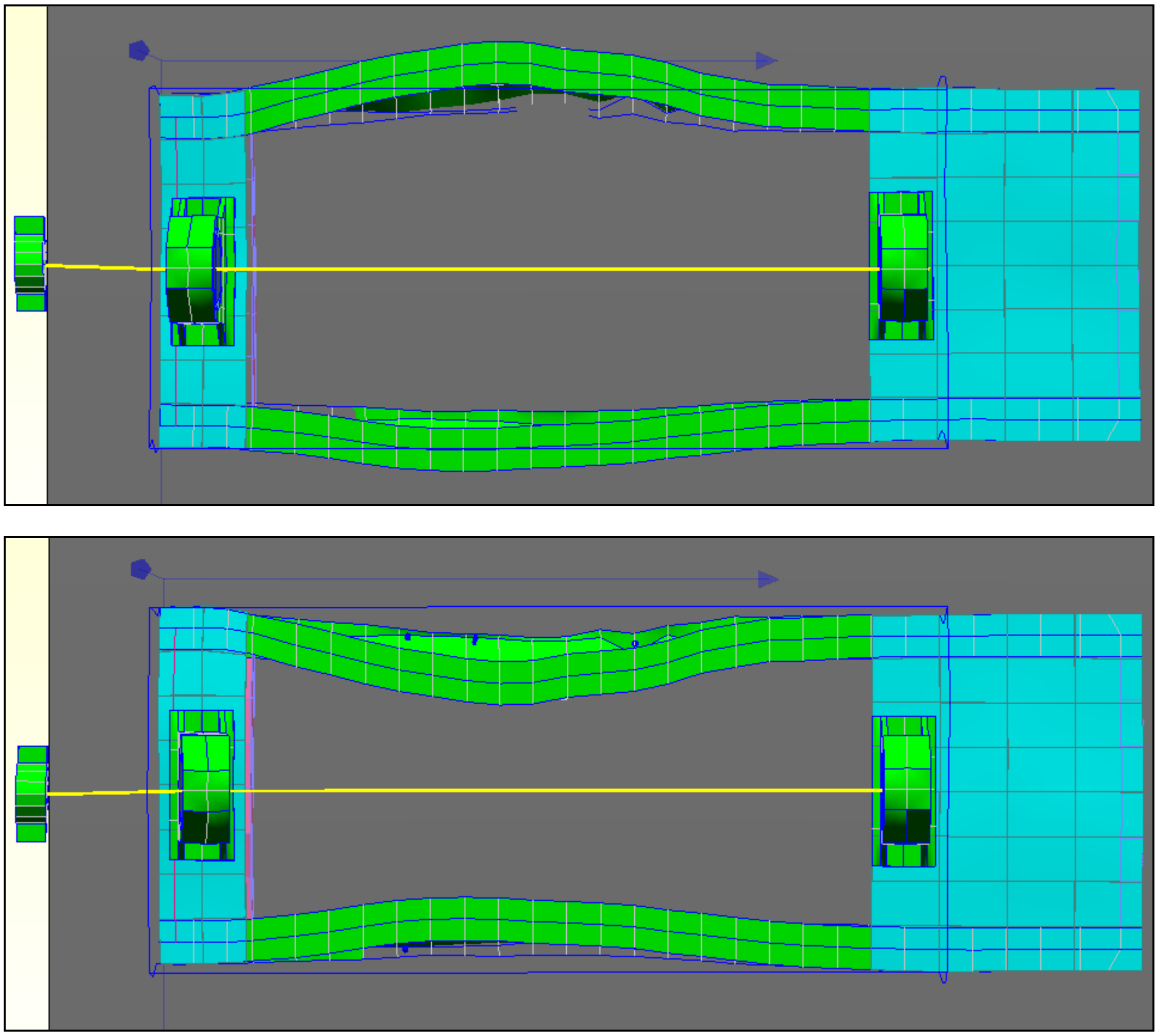

Figure 7. Generator 2X ODS, Top View.

Transient Speed Vibration Analysis: Based on the ODS results, it was apparent the significant generator frame rail movement was creating excessive vibration on the inboard bearing pedestal, and was likely contributing to stator vibration as well. The next question that needed to be answered was whether or not the frame was actually resonant at or near either 3,600 or 7,200 cpm.

To begin answering this question the next course of action recommended to the customer was to acquire transient speed vibration data during the unit shutdown. Due to operational requirements this did not occur for several months. During this time plant personnel tried several adjustments to the generator hold-down bolting, but the inboard generator bearing cap vibration remained in the 0.7 to 1.1 in/s-pk range.

The unit was finally shutdown in October to inspect the generator bearings, check coupling alignment, and inspect the frame rail loading. For the shutdown we suggested that an over-speed trip be performed. This would provide us with transient speed vibration data from 3,960 rpm, well above the operating speed of 3,600 rpm, which often proves helpful in clarifying resonance activity near operating speed.

Vibration data was collected during the shutdown from the permanently installed proximity probes. The generator inboard bearing showed the rotor had a distinct 1st lateral balance resonance ("critical") at 1,410 rpm as shown in Figure 8.

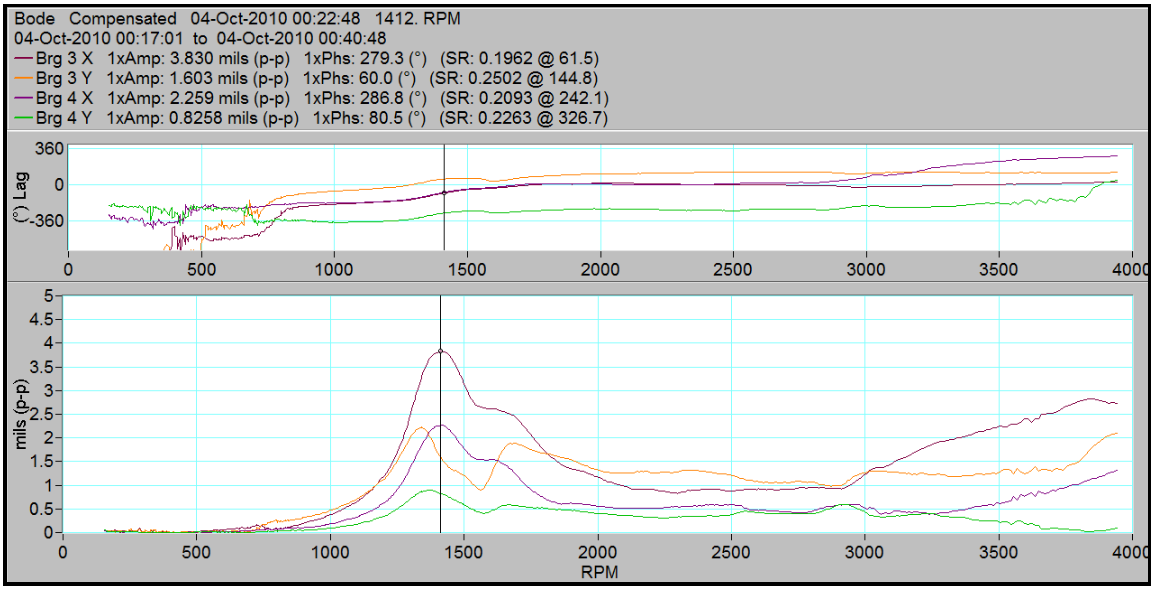

Figure 8. Generator Shaft – 1X Bode Plot during Shutdown.

Near operating speed the vibration amplitudes increased, but the 1X-filtered polar plot clearly showed there was only rotor critical, as shown in Figure 9.

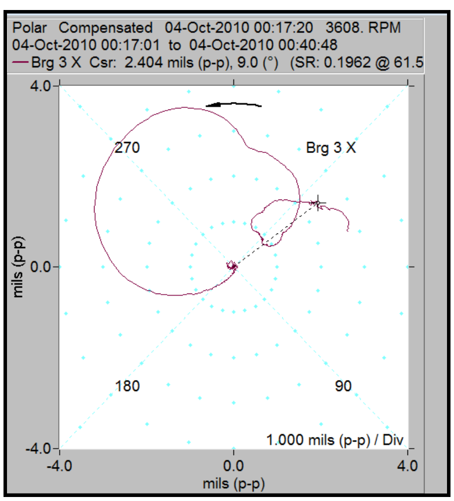

Figure 9. Generator Shaft – 1X Polar Plot.

Additional vibration data was gathered from the bearing cap using temporarily installed velomitors mounted in the horizontal, vertical & axial directions. The 1X-filtered bode plot showed a distinct resonance in the axial direction occurring at 3,810 rpm, as shown in Figure 10. The presence of this structural resonance at 3,810 rpm results in the bearing pedestal being subjected to high vibration at the operating speed of 3,600 rpm.

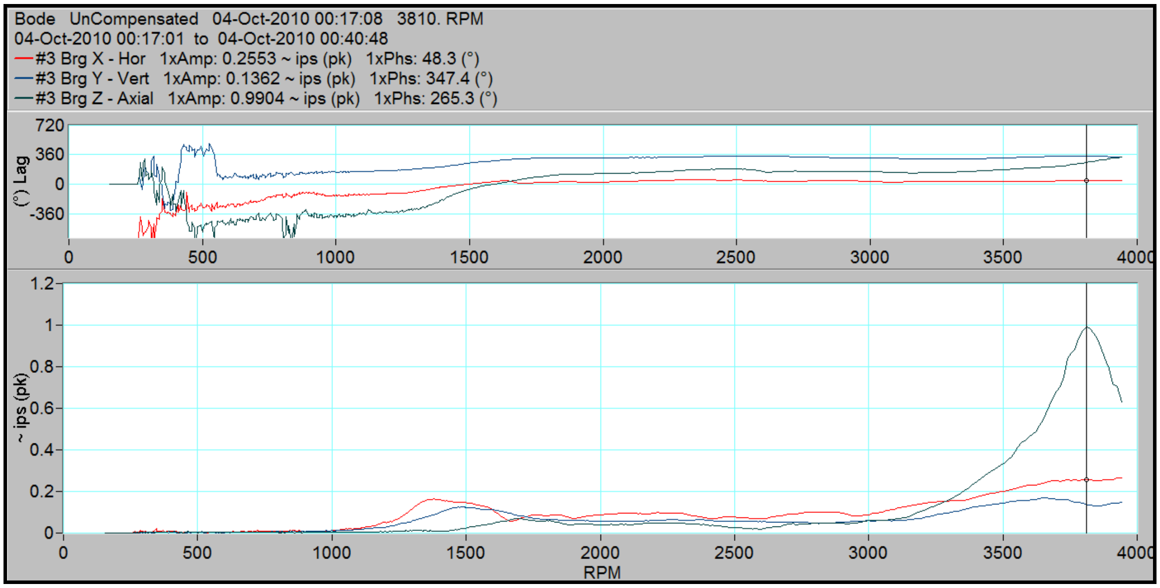

Figure 10. Generator Inboard Bearing Cap – 1X Bode Plot.

Results: During inspection a significant soft-foot condition of 0.027” was found at the generator inboard bearing, right-side frame rail (reference Figure 5). Additional soft-foot issues were found along both frame rails in the plane of the outboard bearing (reference Figure 6). The frame rails were re-shimmed accordingly, and acceptable soft-foot was verified at each hold-down bolt location using a dial indicator. The inboard turbine and generator bearings were also inspected, and found to be in good condition. The alignment was also modified to accommodate thermal growth values that had been measured by us during an optical survey of the unit.

After the unit was re-started in late October, the generator inboard bearing cap vibration levels were 0.35 in/s-pk, representing a dramatic improvement that will certainly increase the long term reliability of both the bearing and the stator.

Conclusions: By performing an ODS analysis prior to beginning any mechanical inspection work we were able to gain a thorough understanding of the problem areas in the generator support frame. A high-quality, well-detailed model showed intricacies within the ODS modes that precisely indicated the location of problems areas without the need to shutdown the unit. These details would have been easily overlooked if a less detailed model were used. Further, by combining the ODS model with a transient speed analysis and a thermal growth study, we were able to direct the customer in the inspection & realignment process, and get the unit back on-line with acceptable vibration levels. Combining ODS and transient speed analyses also allowed us to identify the resonance characteristics of the shaft and structure without resorting to using the usual ‘shakers’ or impact methods associated with traditional modal analysis. Using shakers or impact testing on large turbo-machines is not practical from a physical perspective due to the size and weight of the machines and the forces required to excite the machinery. And, the off-line time required for that type of testing will rarely be tolerated by plant management due to lost generation costs.

Case History #2 - High-Pressure Air Blowers: A pair of motor-driven, skid mounted air blowers (Figure 11) at a power plant had proven troublesome since plant startup during the early 1990s. Blower #1 in particular was always very sensitive to minor changes in alignment, balance condition, and soft-foot; efforts to shim, precision level, and anchor the frame had provided only marginal improvement. And, although the units are only rated at 150 HP, they supply plant boiler air and are considered critical equipment. The loss of blower air results in a boiler and turbine-generator shutdown and the loss of 140 MW electrical generation revenues for the plant.

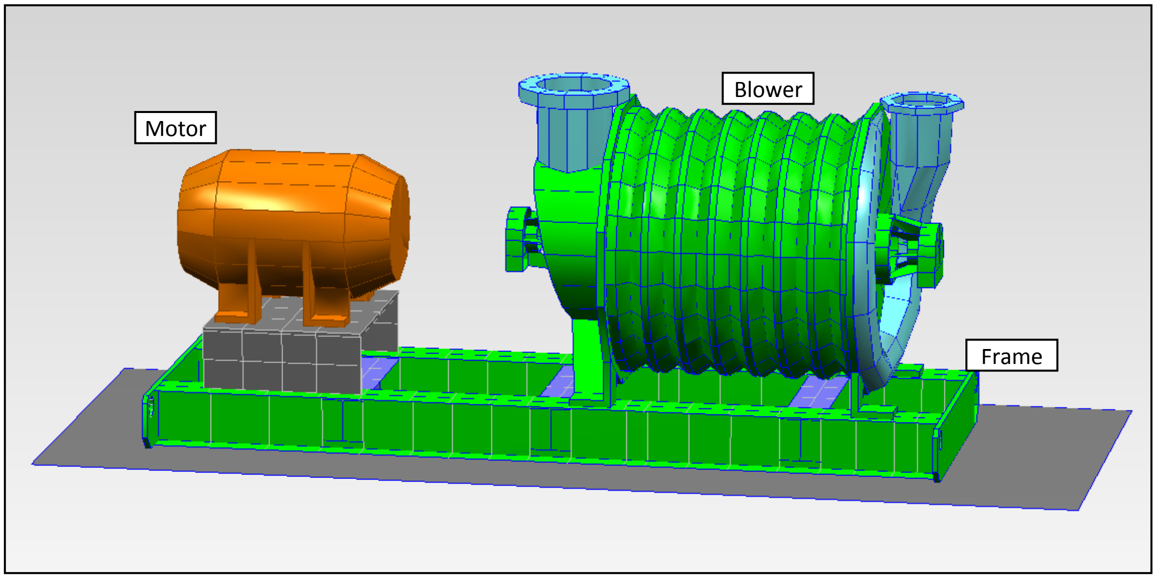

Figure 11. High-Pressure Air Blower Structural Model.

Blower #1 was rebuilt in 2008 but vibration levels at operating speed (3,590 rpm) exceeded 1 inch/second, and the unit was placed in emergency standby until the problem could be resolved. Transient speed vibration analysis revealed several structural resonances, with one occurring directly at operating speed. It was then decided to perform a modal analysis of the unit to determine how the frame could be best modified to alter the resonance and reduce the unit's sensitivity.

After constructing the model, impact testing was performed on the structure to identify any resonances present and their mode shape. A force hammer and tri-axial accelerometer were used to capture single input, multiple output (SIMO) data to generate the necessary FRFs at the selected DOFs.

We were mainly interested in determining the mode shape of the resonance that coincided with operating speed, as that was the most troublesome vibration frequency. However, due to background noise from the adjacent Blower #2, the FRF data at 3,590 cpm proved too noisy to be useful, despite averaging of multiple FRFs. We did note that the data outside of the operating speed range showed more typical modal activity.

"Non-Operating ODS": Since the two units operate at the same speed, it stood to reason that the background vibration from Blower #2 could be used as an excitation source for Blower #1. Since there was sufficient vibration present on Blower #1 when the unit was off, the background vibration would serve us well as a "shaker" tuned at precisely the frequency needed to provide the excitation for what the author has termed a "Non-Operating ODS". A roving tri-axial and a stationary reference accelerometer were then used to provide the cross-spectral data from 117 DOFs on the frame, motor, and blower.

The Non-Operating ODS data in Figure 12 indicated the frame was subject to a significant lateral (horizontal) resonance at 3,590 cpm. This lateral resonance and mode shape clearly explained the vibration sensitivity of Blower #1 to any changes in alignment, frame position, and balance quality. With the resonance being coincident with operating speed, any changes in these items would directly affect the unit's 1X vibration response in a non-linear manner.

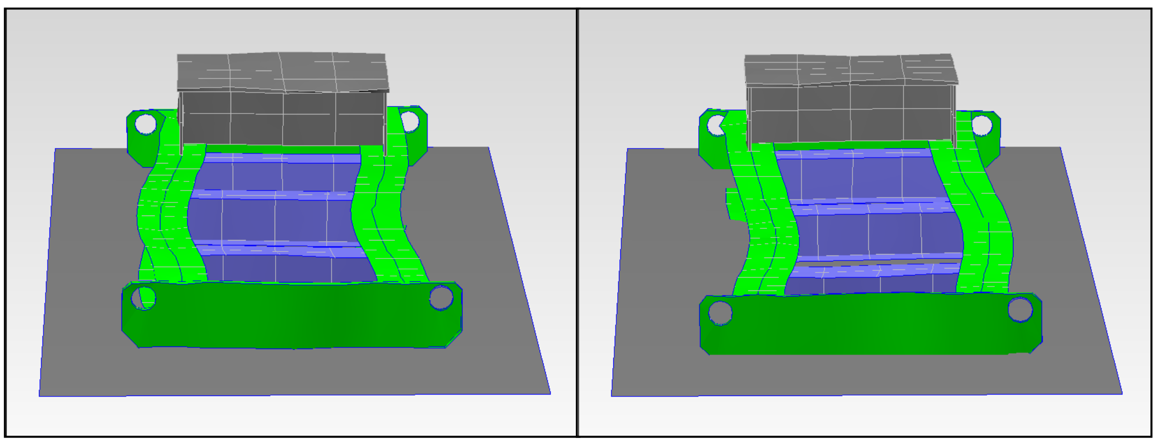

Figure 12. Blower Frame – 3,590 cpm Lateral Mode Shape.

Results: To correct this situation, 1" thick baseplates were designed and installed beneath the motor and blower. The blower baseplate was 30.5" wide x 88" long and added 740 pounds to the structure. It effectively eliminated the resonance and the plate's added mass decreased the background vibration present on the blower, which will decrease the false brinnelling of the rolling element bearings occasionally noted in the past.

Overall Conclusions: This paper highlighted two interesting cases that were difficult to solve using 'typical' vibration analysis techniques. Modal & ODS analysis provided excellent insight into the machinery dynamics and helped solve some perplexing reliability problems. Additional examples may be found at http://www.mbesi.com/ModalGallery.htm. Some important points highlighted above worth remembering include:

- Constructing an accurate model with sufficient complexity at the start of a project lets us accurately depict global and local mode shapes on our first pass through the system. It also minimizes the re-work time and effort that is often encountered when ‘coarse’ models are used ‘just to get started’.
- ODS analysis shows deflection shapes at operating speed and can be used to manage 1X vibration.
- When ODS and transient speed analysis are combined on large turbo-machines, most resonances can be accurately identified without resorting to the use of large, inconvenient ‘shakers’ or impact methods associated with traditional modal analysis.
- Resonances near running speed, or ones excited during transient speed operation, can reduce reliability due to excessive vibration. ODS and transient speed analysis can be used to help determine corrective actions.
- The author introduced the concept of a Non-Operating ODS, whereby effective ODS analyses can be performed using background vibration from adjacent machinery as an excitation source.

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[2] K Blakley, D Kientzy, M Richardson, “Using Finite Element Data to Set Up Modal Tests”, Sound and Vibration Magazine, June 1989, page 20.