New Pressurized Bearing Kills Frequency Shifts in Rotating Machinery

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This article explores the use of pressurized bearings, such as the ServoFluid™ Control Bearing (SFCB), to reduce natural frequency shifts in rotating machinery. The Dynamic Stiffness of conventional journal bearings is highly nonlinear and highly dependent upon journal position within the bearing clearance. Forces other than those considered in the machine design, such as misalignment, can cause the journal position to deviate from the design location. This could result in bearing stiffness modification causing changes in the system's natural frequencies. Often, these frequencies are placed close to operating speed to optimize machinery dynamic properties. In these systems, even slight variation in the natural frequency can cause them to become unacceptably close to operating or other sensitive frequencies.

Introduction

As discussed in "The Death of Whirl" (First Quarter 2001 *ORBIT*, Vol. 22 No. 1, pp.10-13), the application of the ServoFluid™ Control Bearing provides the capability to eliminate oil whirl in the bearings of rotating machinery. This article focuses on another strength of the SFCB: the insensitivity of the bearing stiffness to shaft position in the journal clearance, commonly referred to as eccentricity. One of the problems encountered using conventional non-pressurized fluid-film bearings is the *dependence* on eccentricity for their stiffness. The conventional bearing has a nominal centered stiffness that increases rapidly as the journal approaches the bearing wall. During normal machine operation, a static radial load is applied to the bearing by the main process flow, or in the case of horizontal machines, by gravity. The static radial load sets the nominal operating position for the journal within the bearing clearance, establishing a nominal value for the bearing stiffness and damping properties. These values are used in the design of the machine to place the system's natural frequencies into regions where they don't conflict with operating or other sensitive frequencies.

Finite element model cross section of turbine generator

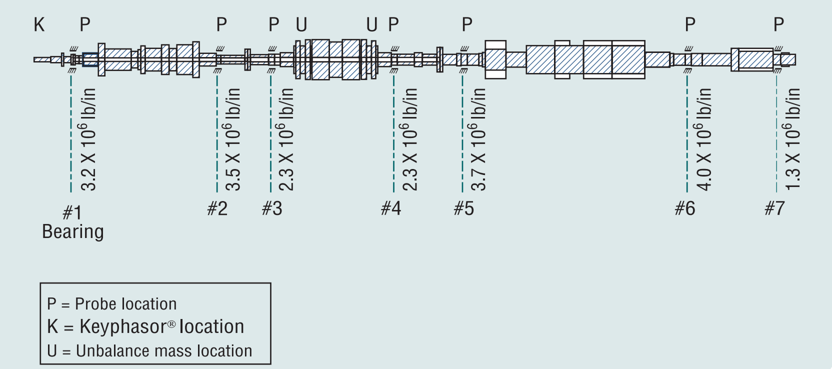

Figure 1.

The Problem

Let's focus on a large steam turbine generator (STG) system used to produce electrical power. For these heavy, horizontal machines, gravity provides the static radial force that sets the design eccentricity point for the bearing. If gravity were the only force acting on the rotor, there would be no problem when using the conventional non-pressurized bearings. However, other forces such as alignment can affect journal position in the bearing. Consequently, machines that have natural frequencies close to the operating or other sensitive frequencies can exhibit unusually high vibrations when the alignment is poor. This produces bearing stiffness values that deviate from the nominal design values.

A 300 MW steam turbine generator is shown in Figure 1. The vibration response at the low pressure (LP) turbine under normal operating conditions is shown in Figures 2 and 3. Although the initial excursion into the resonance region is observable in the data, the actual natural frequency cannot be accurately determined because it is above the maximum allowable speed of the machine. However, a computer model can be used to find the actual second natural frequency. By adjusting the bearing stiffness and residual unbalance in the computer program to produce the observable rotor response below operating speed, a model of the machine is created that can be used to estimate the response above its normal operating speed range.

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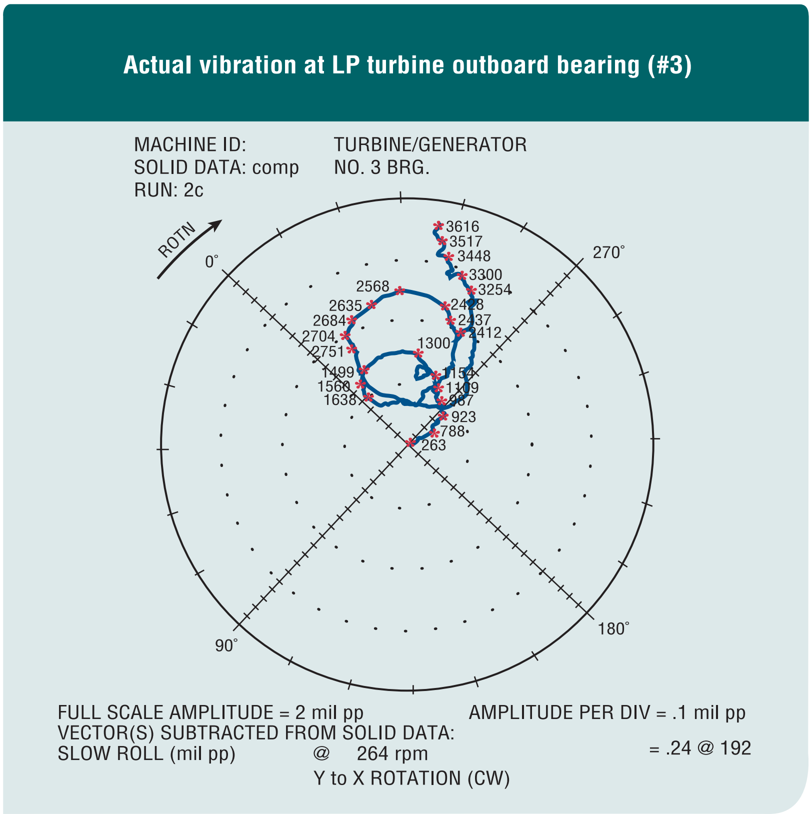

Figure 2.

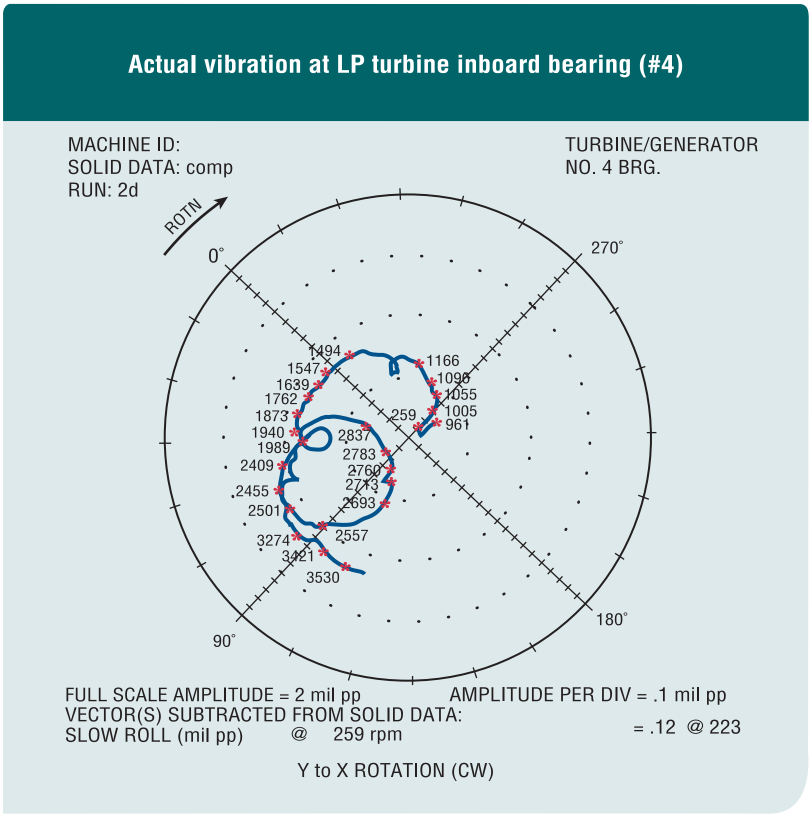

Figure 3.

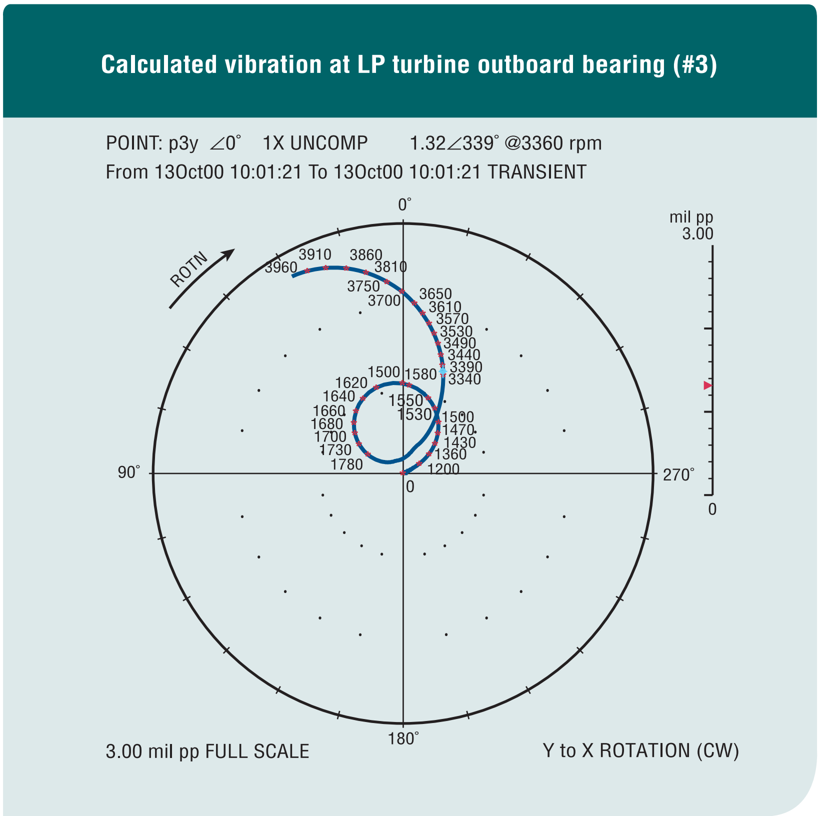

Figure 4.

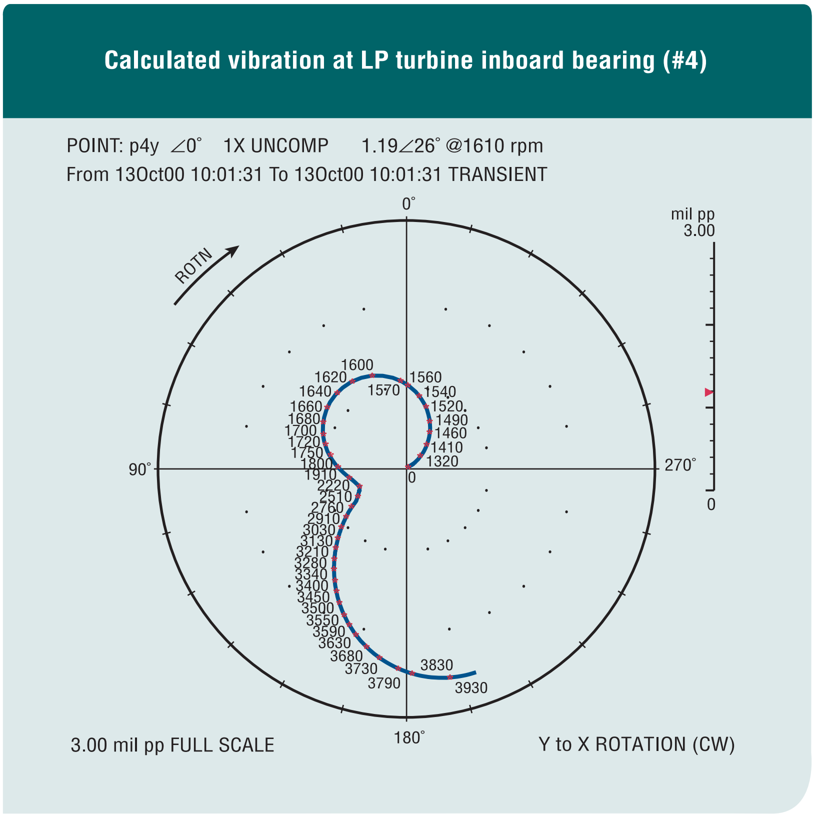

Figure 5.

Calculated responses from the computer program are shown in Figures 4 and 5. One can see that they agree very well with the actual data in Figures 2 and 3, except for the region of 2700 rpm. The response at 2700 rpm is caused by the second resonance of the generator. It is not observed in the calculated data because it is not excited by the unbalance distribution used to explore the natural frequencies of the LP turbine.

The unbalance in the LP turbine, as determined by the computer program, was 40 in•lbs at 295 degrees in the balance plane away from the generator and 15 in•lbs at 125 degrees in the balance plane adjacent to the generator. Bearing stiffness values for the LP turbine were 2.3 x 10^6 lb/in, while those for the generator were 3.7 x 10^6 lb/in for the bearing adjacent to the LP turbine and 4.0 x 10^6 lb/in for the bearing away from it. Using these bearing stiffness values, and extending the rotational speed range of the calculations above the normal operating speed, produces the response shown in Figure 6. This data indicates that the second natural frequency is at 4110 rpm, 15% higher than the operating speed of 3600 rpm, with a mode shape as shown in Figure 7. This margin between the natural frequency and operating speed is within commonly accepted limits.

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Nonetheless, in the assembly and operation of the machine, factors such as misalignment, bearing tolerance, and wear can affect the bearing stiffness values and reduce the natural frequency to a level unacceptably close to operating speed. For example, assume bearings that are designed to operate at a 0.6 eccentricity ratio (a common design objective). When the machine is misaligned, it produces an unloading in the LP turbine bearing next to the generator. The stiffness would decrease for the LP turbine bearing and increase for the adjacent generator bearing.

The question becomes how much does each stiffness change? This can be calculated using the design eccentricity of 0.6 and the formula for stiffness versus eccentricity for journal bearings, $\frac{K_0 e}{\left(1-e^2\right)^{3/2}}$, where K_0 is the centered stiffness of the bearing and e is the eccentricity of the journal in the bearing. A three to four times decrease in stiffness in the LP turbine bearing could be expected as the misalignment forces the journal to lower eccentricities. Given that the generator bearing is larger, there would be a slightly smaller increase in its stiffness as it is forced to carry the load that is normally supported by the LP turbine bearing. Therefore, the unloaded stiffness of the LP turbine bearing would decrease to 6.0 x 10^5 lb/in and the stiffness of the generator bearing would increase to 1.0 x 10^7 lb/in, and this would change the natural frequency.

Again the question becomes, how much? Replacing the design values in the computer program with the misaligned values produced the response shown in Figure 8. This data indicates that the second natural frequency would decrease to 3720 rpm, with the mode shape shown in Figure 9. *This second natural frequency is only 3% above operating speed and is unacceptably close.*

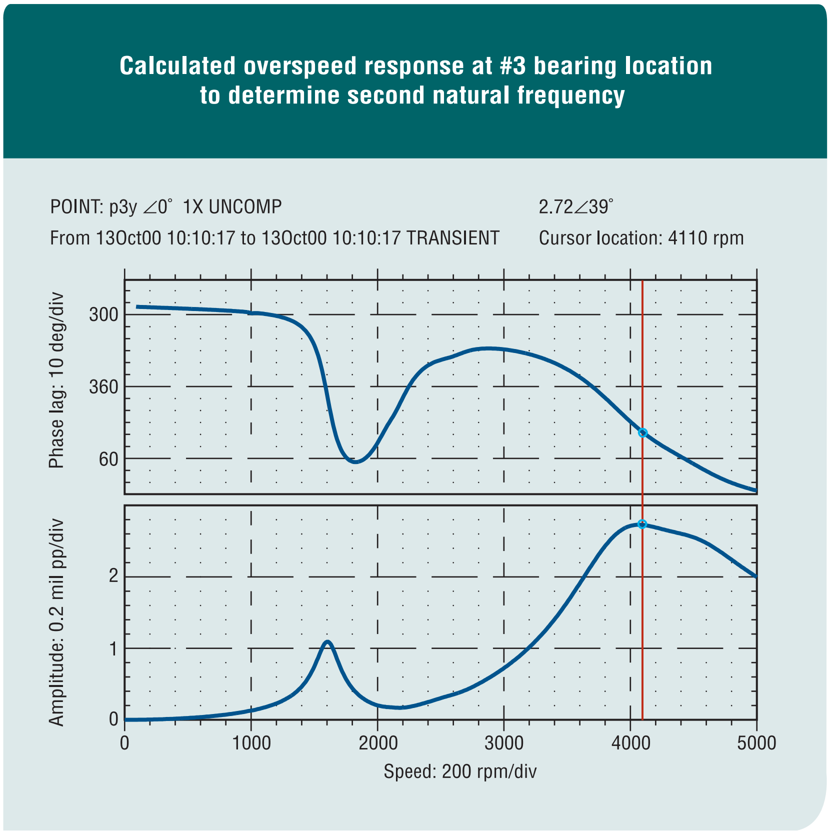

Figure 6.

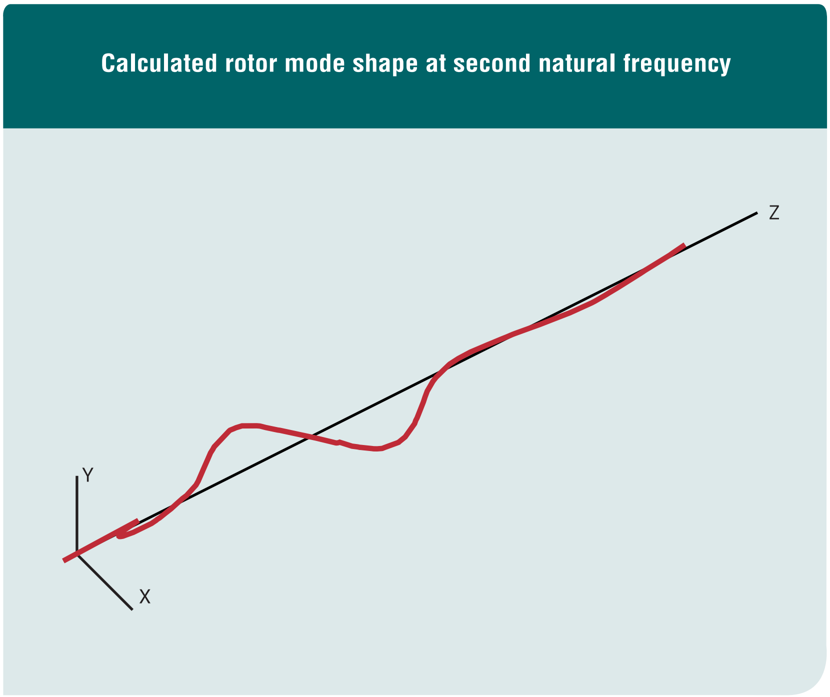

Figure 7.

Calculated second natural frequency at #3 bearing location with misalignment

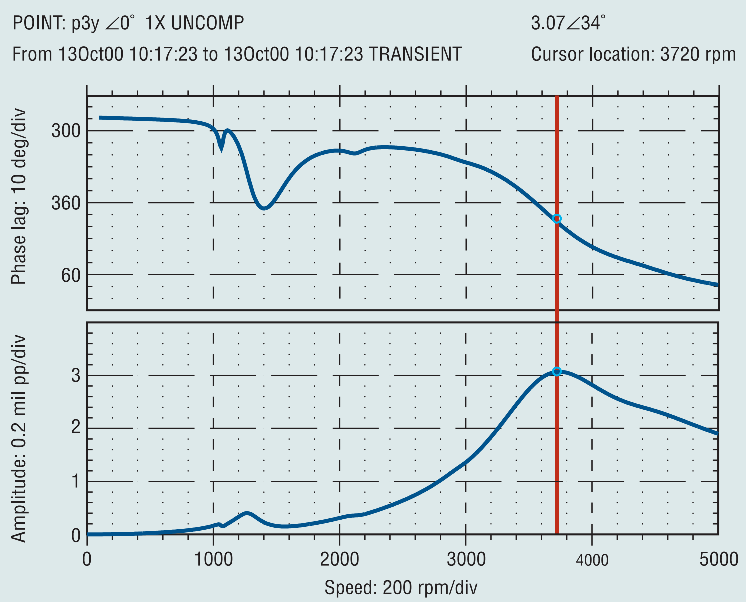

Figure 8.

Calculated rotor mode shape at second frequency with misalignment

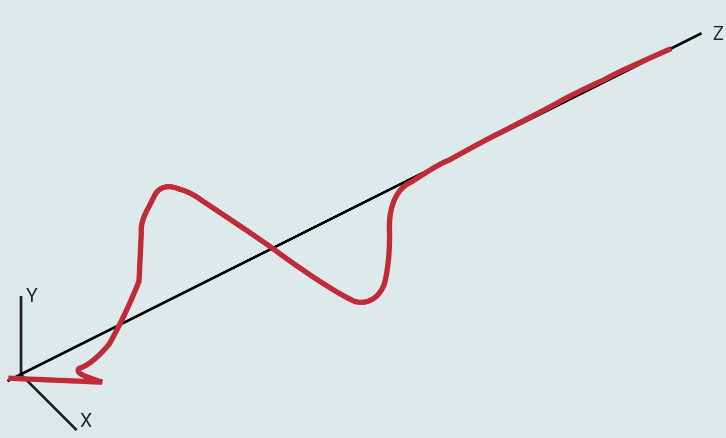

Figure 9.

The Solution

How do we prevent this occurrence? The obvious answer is to never misalign the machine. Unfortunately, machines are never perfectly aligned, and even if they started out as such, the alignment will change with time and operating conditions. A less obvious answer is to use a bearing that doesn't significantly change its stiffness when the eccentricity changes: the Bently ServoFluid™ Control Bearing. This bearing is designed to operate in the center of the journal clearance and exhibits only a slight decrease in stiffness as eccentricity increases, which can be compensated for by increasing the fluid supply pressure. Design bearing stiffness is sustained over varying static radial forces, and unlike conventional journal bearings, the second natural frequency at 4110 rpm is maintained and does not encroach upon the operating speed of 3600 rpm.

"... use of the ServoFluid™ Control Bearing yields a machine that is significantly more tolerant to changes in alignment."

Conclusion

The use of pressurized bearing technology, such as the Bently ServoFluid™ Control Bearing, can effectively eliminate frequency shifts and the problems they cause. This is because bearing stiffness is held relatively constant, regardless of changes in shaft eccentricity position.

While a perfectly aligned machine is a laudable goal, it can rarely be achieved in practice and will change with time and operating conditions. These changes in alignment condition result in changes in eccentricity and – in a conventional fluid-film bearing – changes in stiffness that significantly alter the natural frequency of the rotor dynamic system. Pressurized bearing technology does not suffer from these phenomena. Thus, use of the ServoFluid™ Control Bearing yields a machine that is significantly more tolerant to changes in alignment.