Experimental Results and Implications of Perturbation Testing with ServoFluid™ Control Bearings

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During development and testing of Bently Nevada's new ServoFluid™ Control Bearing (SFCB), additional studies were performed on a modified Clark 1M6 compressor fitted with SFCBs to better understand the behavior of the bearing as well as the subject machinery and its foundation. Significant new discoveries were made and documented.

This article focuses on a portion of those studies: namely, the use of *non-synchronous* perturbation methodologies and their role in identifying the Dynamic Stiffness parameters of the SFCB. [Editor's Note: A comprehensive explanation of the various types of perturbation testing was in the article *Adaptation of Perturbation Techniques to Full-Sized Machinery*, ORBIT, Vol. 21 No. 1, 2000, pp. 41-44.] The majority of benefits associated with this advanced bearing design were determined from data collected with the experimental configuration described in this article. The findings show significant differences in characteristics between the SFCB and conventional bearings, and indicate the superiority of the SFCB for many applications.

[Editor's Note: We recommend first reading the companion article *In Pursuit of Better Bearings* on page 33 of this issue. It provides excellent background information that will aid you in understanding the significance of the experimental results presented in this article.]

ServoFluid Control Bearing

The ServoFluid™ Control Bearing is an externally pressurized, fully lubricated fluid bearing. Prevailing conventional practice is to avoid full lubrication and bearing pressurization in the belief it leads to instability in fluid-film bearings applied to turbomachinery. Research conducted by Bently Rotor Dynamics Research Corporation has shown just the opposite: a pressurized, fully lubricated bearing design actually has numerous practical advantages, including vastly improved stability and the ability to adjust the characteristics of the bearing with the machine running. The key is to use *adequate* pressure (generally in the hundreds of psi), considerably higher than the pressures of 10 to 20 psi used in the fluid delivery systems of typical fluid-film bearings.

The unique design of the SFCB forces fluid to flow primarily along the shaft, rather than around it, forming a predominantly axial support wedge, rather than a predominantly circumferential support wedge as with conventional fluid-film bearings. This is the basis of the SFCB's stability and other unique characteristics, a portion of which are detailed in this article.

Figure 1 shows a basic representation of the new ServoFluid™ Control Bearing. Working fluid is supplied to the bearing using a pressurized fluid delivery system. The pressurized fluid flows through orifices to specially designed bearing ports and pockets. Rotor motion and the bearing's geometric design create a pressure differential between opposing
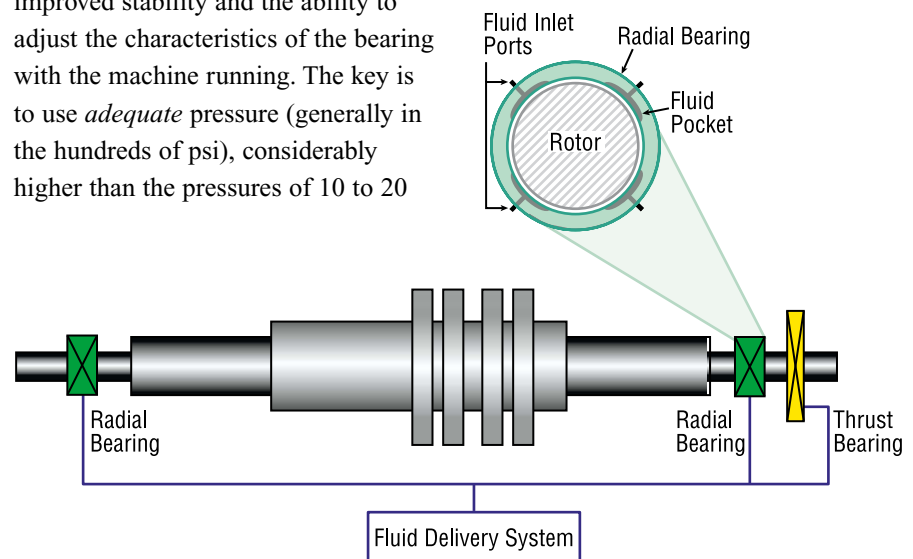

Figure 1. ServoFluid™ Control Bearing.

pockets (180 degrees apart) to provide a restoring force to the rotor during operation, thus centering the rotor and creating a stiff and stable bearing. More information on the operating principles of this bearing are available in the product brochure included in this issue of ORBIT, and in the article *Dynamic Stiffness and the Advantages of Externally Pressurized Fluid-Film Bearings*, which appeared in the First Quarter 2000 ORBIT (Vol. 21 No. 1, 2000, pp. 18-24).

Equipment and Procedures

The photo and diagrams shown in Figures 2, 3a, and 3b illustrate the general layout of the Clark 1M6 compressor and the installed transducer
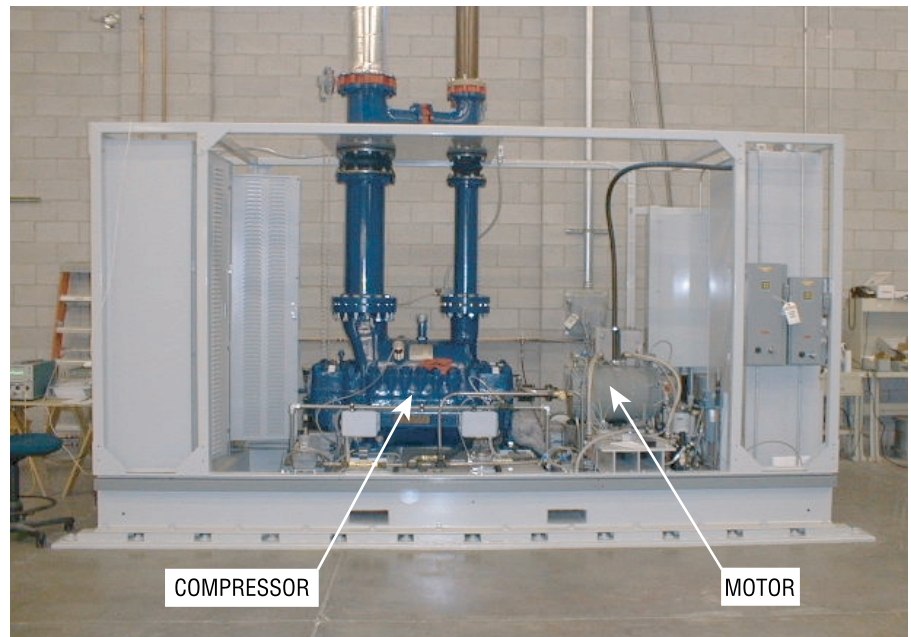

COMPRESSOR   MOTOR

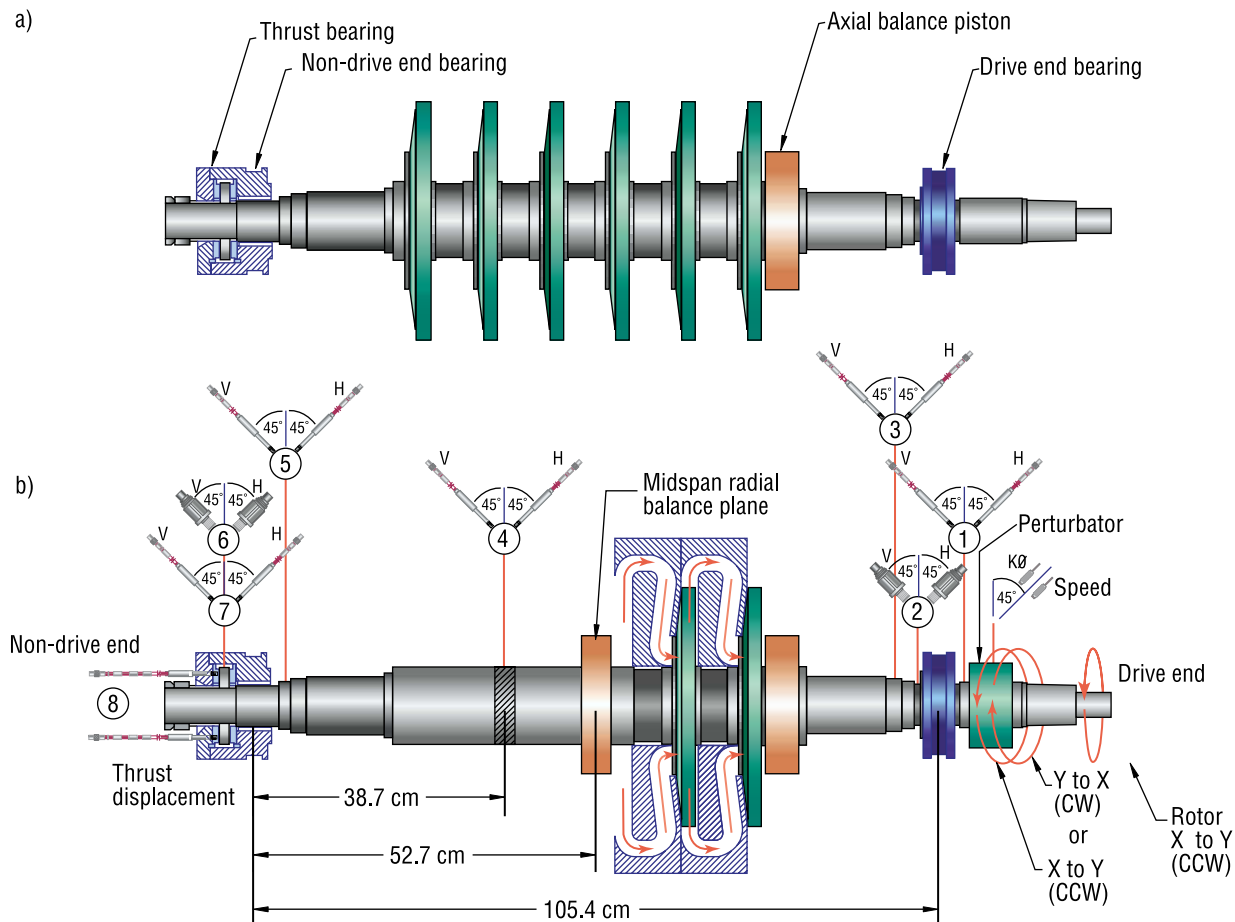**Figure 2. Clark 1M6 compressor with electric motor drive located on right.**

Figure 3. Schematic of the Clark 1M6 Compressor: a) Standard configuration; b) Modified configuration – transducer locations 1, 3, 4, 5, and 7 are for XY radial proximity probes observing shaft relative motion. Location 8 is for axial proximity probes. Locations 2 and 6 are for XY velocity transducers observing casing motion.
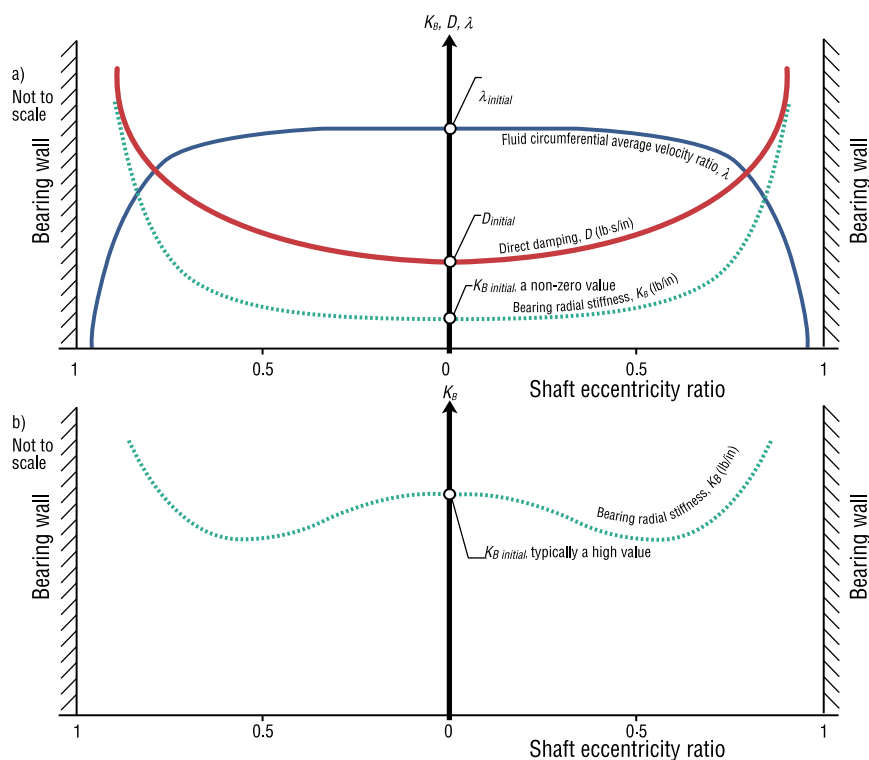
**Figure 4. Radial bearing characteristics as a function of eccentricity for:
a) Conventional hydrodynamic sleeve-type designs; b) ServoFluid™ Control Bearing**

suite. The compressor, driven by a 130 hp (97 kW) electric motor, was retrofitted with two radial ServoFluid™ Control Bearings and one axial (thrust) ServoFluid™ Control Bearing. Operating oil pressures in these SFCBs were 1000 psi (6900 kPa) for the supply and 700 psi (4800 kPa) for the bearing "ports."

To enable better investigation of the SFCB's characteristics, modifications were made to the Clark 1M6 compressor as well. Figure 3a shows the standard compressor configuration; Figure 3b shows the modified compressor. As shown, the number of compression stages was reduced from six to two; the machine was modified to accommodate five sets of XY proximity probes, two sets of XY velocity transducers, and two axial probes observing the thrust bearing movement; a midspan radial balance plane was added; and the compressor was fitted with a special

"perturbation wheel" attached to the rotor drive end. This perturbation wheel can rotate on the shaft *independent* of the main rotor speed and direction, and can accommodate weight placements much like a balance ring, allowing an external force input to the mechanical (rotor-bearing-case) system.

To determine the Dynamic Stiffness characteristics of the SFCB, perturbation testing was used to quantify the system's behavior when subjected to a known applied force.

[Editor's Note: For more information on Dynamic Stiffness, consult the tutorial article on page 44, or the article *Dynamic Stiffness and the Advantages of Externally Pressurized Fluid-Film Bearings*, which appeared in the First Quarter 2000 ORBIT (Vol. 21 No. 1, 2000, pp. 18-24).]

Testing of the compressor with the perturbation wheel was conducted with the compressor stopped and at 7000 rpm.

At these compressor rotation speeds, the perturbation wheel itself was used both with and without weights at perturbator rotational speeds between zero and 10,000 rpm to investigate the effects of nonsynchronous perturbation. The perturbator mass used was a 0.5 oz (14.2 g) weight, (r = 2.5 in, or 6.35 cm) located at 346° relative to the compressor's Y probes.

Results of SFCB Perturbation Testing

In a conventional hydrodynamic radial sleeve bearing, it is known that the following characteristics change non-linearly as a function of rotor eccentricity position (e):

- The bearing's fluid-film stiffness, K_B, (with units of lb/in)

- Damping, D, (with units of lb·sec/in)

- Lambda (λ), the fluid circumferential average velocity ratio, (with dimensionless units)

- Attitude angle, Ψ, (with units of angular degrees)

The elements of K_B, D, and λ as functions of eccentricity position are shown in Figure 4a. In short, a conventional sleeve bearing exhibits decreased λ, increased radial stiffness, and increased damping as rotor eccentricity position increases. Since high stiffness and damping, and low values for λ are generally desirable in a fluid-film bearing, it is readily apparent that this is achieved in a conventional bearing at the expense of large rotor eccentricity ratios (values closer to 1).

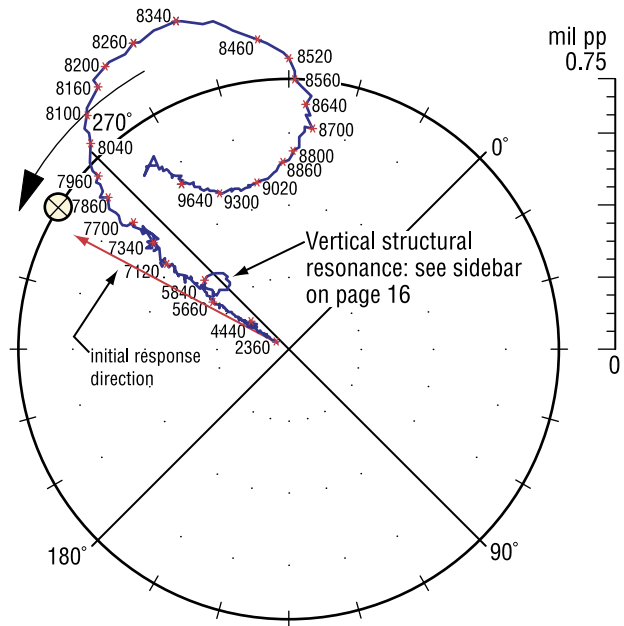The ServoFluid™ Control Bearing's characteristics are quite different, as shown next.

- **Lambda (λ)**

    Again referring to Figure 4a, it is seen that λ starts high with low
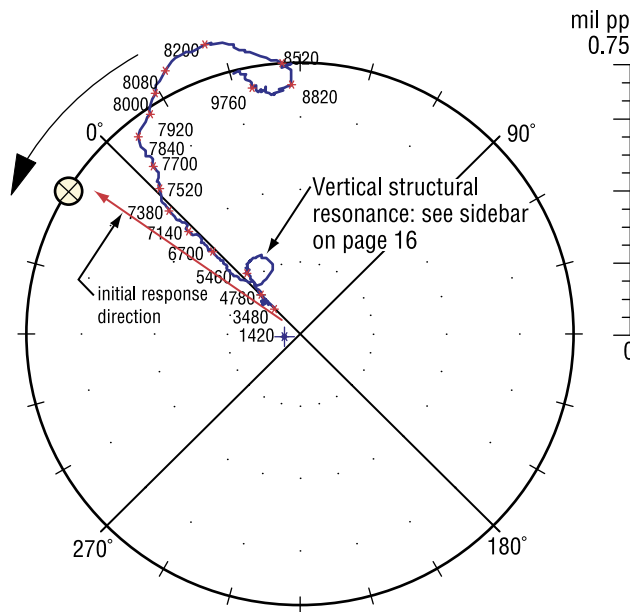
0.75 mil pp FULL SCALE

CCW ROTATION

Figure 5a. Polar plot response, of compressor at probe location 1, to perturbator speeds between 0 and 10,000 rpm with compressor at 0 rpm.
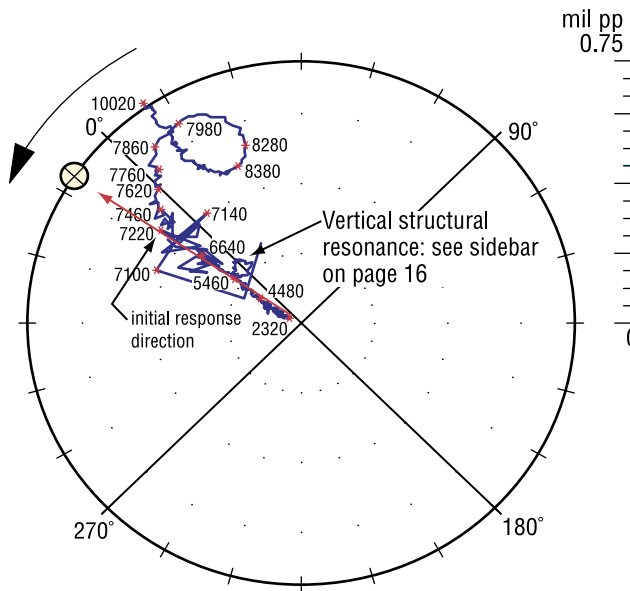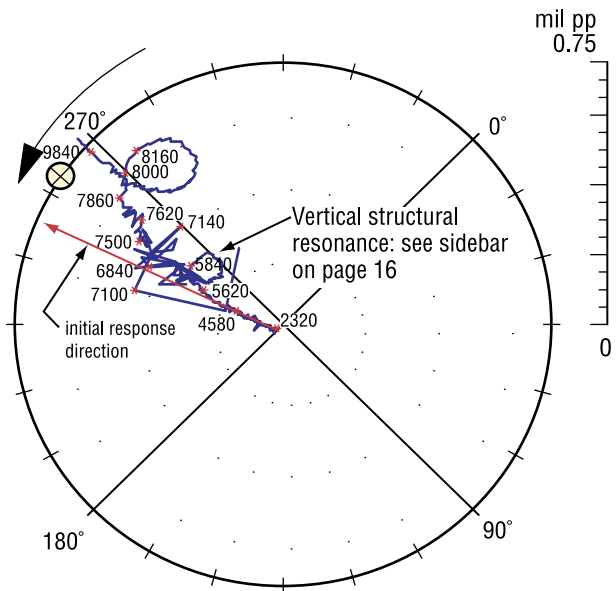
0.75 mil pp FULL SCALE

CCW ROTATION

Figure 5b. Polar plot response, of compressor at probe location 1, to perturbator speeds between 0 and 10,000 rpm with compressor at 7,000 rpm.

eccentricity ratios and decreases with increasing eccentricity ratios in a conventional hydrodynamic sleeve bearing. High values of λ are one measure of a bearing's propensity to become unstable and exhibit classic fluid whirl or whip. It is notable that Figure 4b does not show λ. This is because its value is negligible in the SFCB. Experimental confirmation of the small value of λ in the SFCB can be noted by examining the phase response in Figures 5a and 5b.

Figure 5a is a polar plot showing the 1X amplitude and phase response with the compressor at zero rotational speed and the perturbator wheel running from 0 to 10,000 rpm. Figure 5b shows a similar plot, but this time the well-balanced compressor is running at a constant rotational speed of 7000 rpm while the perturbator is run from 0 to 10,000 rpm.

It is well known that the presence of Quadrature Dynamic Stiffness (QDS) will cause a rotating shaft to move not only in the direction of the applied force (for example, an imbalance force), but also in a direction at right angles to the force. QDS only exists for a rotating shaft. Thus, while we expect the initial response in Figure 5a to reflect no QDS (the shaft is not rotating) and the response to be in line with the unbalance force, we do not necessarily expect this to be the case for the shaft revolving at 7000 rpm in Figure 5b. However, since we observe virtually no change in the phase of the initial response between Figures 5a and 5b, we conclude that QDS is

very small, and λ is likewise very small. The virtual absence of λ in the ServoFluid™ Control Bearing confirms that it is extremely stable.

- **Attitude Angle (ψ)**
  As explained in Don Bently's article *In Pursuit of Better Bearings* on page 33, attitude angle is an excellent indicator of stability. The SFCB has a very small attitude angle, indicative of a very stable bearing. This can be observed in Figure 5b with the compressor running at 7000 rpm. Notice that the polar plot's initial

response points almost directly at the heavy spot (the direction of imbalance force). Since the excitation force and response are in line with one another, we conclude that the ServoFluid™ Control Bearing has virtually no appreciable attitude angle and is extremely stable.

- **Eccentricity Ratio (e)**
  The desirability of machines that can run with small rotor eccentricity ratios is also discussed in Mr. Bently's article, as mentioned above. Experimentally, this was verified for the SFCB as shown
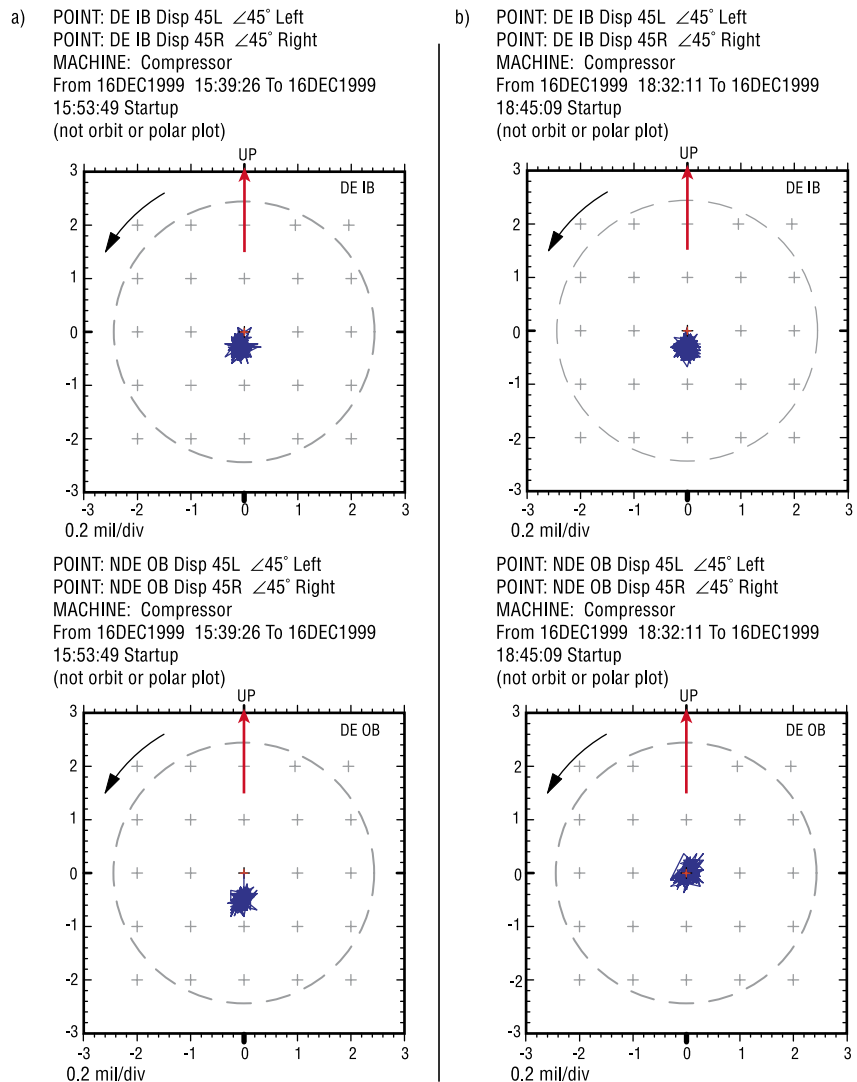
a)
POINT: DE IB Disp 45L $\angle 45°$ Left
POINT: DE IB Disp 45R $\angle 45°$ Right
MACHINE: Compressor
From 16DEC1999 15:39:26 To 16DEC1999
15:53:49 Startup
(not orbit or polar plot)

POINT: NDE OB Disp 45L $\angle 45°$ Left
POINT: NDE OB Disp 45R $\angle 45°$ Right
MACHINE: Compressor
From 16DEC1999 15:39:26 To 16DEC1999
15:53:49 Startup
(not orbit or polar plot)

b)
POINT: DE IB Disp 45L $\angle 45°$ Left
POINT: DE IB Disp 45R $\angle 45°$ Right
MACHINE: Compressor
From 16DEC1999 18:32:11 To 16DEC1999
18:45:09 Startup
(not orbit or polar plot)

POINT: NDE OB Disp 45L $\angle 45°$ Left
POINT: NDE OB Disp 45R $\angle 45°$ Right
MACHINE: Compressor
From 16DEC1999 18:32:11 To 16DEC1999
18:45:09 Startup
(not orbit or polar plot)

Figure 6. Shaft average centerline plots with the compressor at:
**a) 0 rpm shaft rotational speed   b) 7000 rpm shaft rotational speed**

> "*Research conducted by Bently Rotor Dynamics Research Corporation has shown just the opposite: a pressurized, fully lubricated bearing design actually has numerous practical advantages, including vastly improved stability and the ability to adjust the characteristics of the bearing with the machine running.*"

by the shaft centerline plots of Figure 6a. Notice that the rotor in the SFCB system starts at zero rotational speed with a very small (approximately 0.2) eccentricity ratio. The rotor effectively "self-centers" in the bearing clearance due to the external pressurization used in the SFCB, even at zero rotational speed. As speed is increased to 7000 rpm, Figure 6b shows that the shaft centerline at both DE and NDE bearings moves very little relative to its position at zero rotational speed in Figure 6a. Consequently, the eccentricity ratio (and attitude angle) remains virtually unchanged.

- **Radial Fluid Stiffness (K_B)**

  Figure 4b shows the radial fluid stiffness as a function of eccentricity for the SFCB. When compared to the characteristics of a conventional bearing in Figure 4a, some items are noteworthy:

  – The initial value of radial fluid stiffness, $K_{B(initial)}$, is much higher in the SFCB than in a conventional bearing.
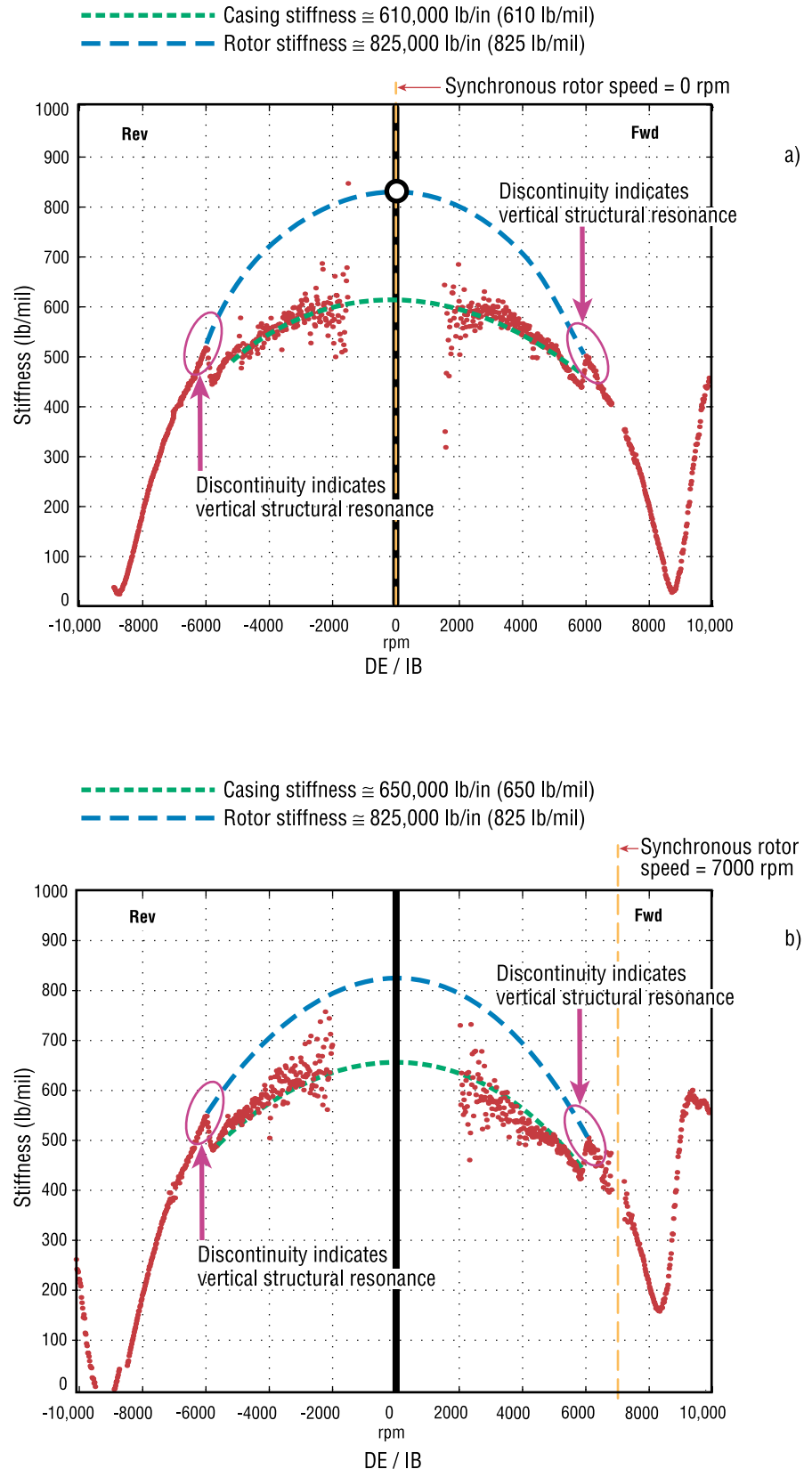
Figure 7. **Observed Direct Dynamic Stiffness of drive end radial ServoFluid™ Control Bearing with compressor at: a) 0 rpm; and b) 7000 rpm.**
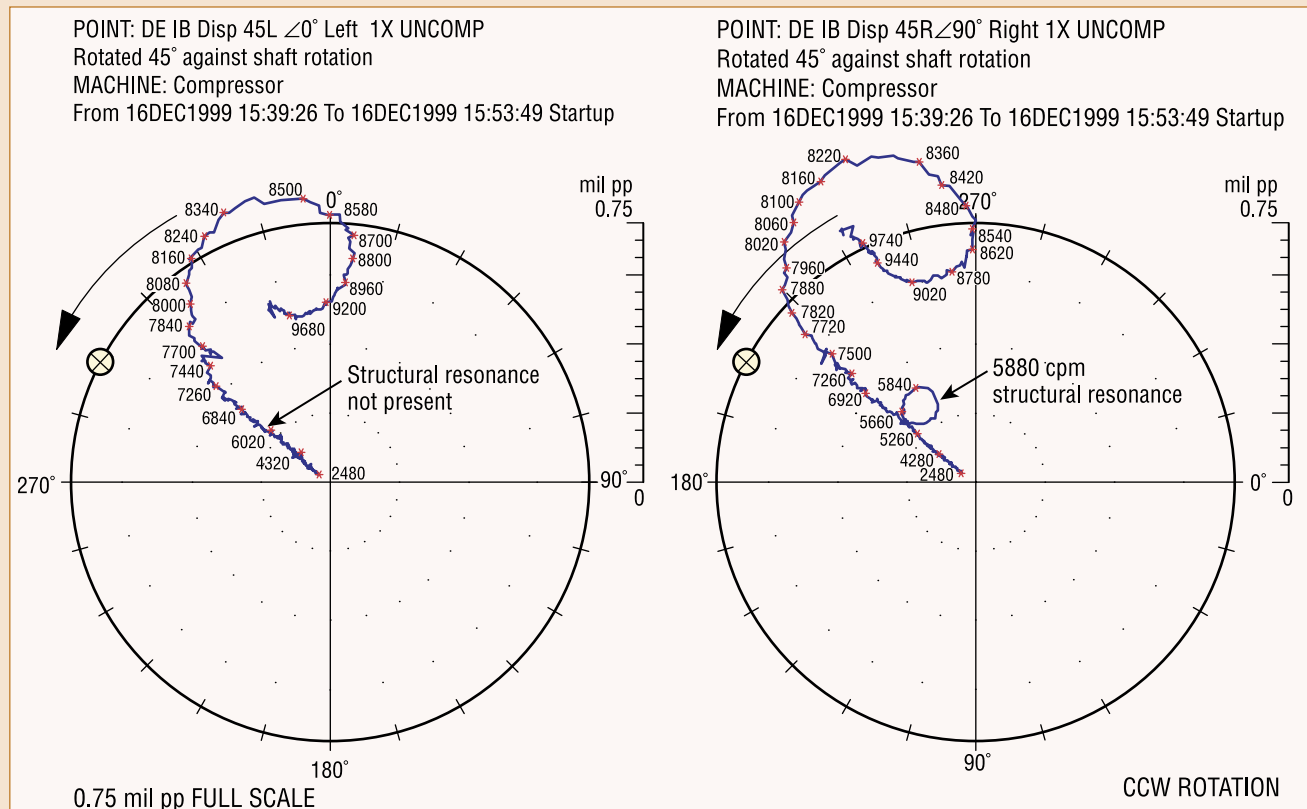
Using the Power of ADRE® for Windows to Analyze a Casing Resonance

During the course of perturbation testing on the ServoFluid™ Control Bearing, polar plots revealed a casing resonance as noted by the small interior loop at 5880 cpm in Figure 5a on page 13. Notice that this loop is present in the plots from both X and Y probes.

Using a standard feature in ADRE® for Windows software known as "virtual probe rotation," data can be manipulated to show the response that would have been measured if the probes were rotated 45 degrees against shaft rotation, thus orienting the Y probe truly vertical and the X probe truly horizontal instead of the actual probe locations of 45 degrees left and right of vertical.

The figure below shows the polar plot response when the data used in Figure 5a was processed using this "virtual probe rotation" feature. Notice that the interior loop is no longer present for the Y (vertical) probe, but is present for the X (horizontal) probe. This indicates that there is a casing resonance purely in the horizontal direction, where the machine moves side-to-side but not up and down. A resonance that is only present in a single direction can often be indicative of machine support problems.

This ability to perform virtual probe rotation is highly useful, as permanently installed transducers are rarely mounted in line with the stiffness axes of the machine. At times, as in this example, the stiffness axes can be truly vertical and horizontal. Other times, they may not be. Regardless, ADRE® for Windows software allows rotation of probes to any axis(es) desired. The ability to conduct such "what if" investigations through ADRE® for Windows' special transducer rotation algorithm allows the user to see what the response would be for a "virtual" transducer mounted in the desired (rather than actual) angular orientation. This feature of ADRE® for Windows is used frequently in the field by Bently Nevada's Machinery Diagnostics Engineers. ↻

POINT: DE IB Disp 45L ∠0° Left 1X UNCOMP
Rotated 45° against shaft rotation
MACHINE: Compressor
From 16DEC1999 15:39:26 To 16DEC1999 15:53:49 Startup

Structural resonance not present

0.75 mil pp FULL SCALE

POINT: DE IB Disp 45R∠90° Right 1X UNCOMP
Rotated 45° against shaft rotation
MACHINE: Compressor
From 16DEC1999 15:39:26 To 16DEC1999 15:53:49 Startup

5880 cpm structural resonance

CCW ROTATION

– The radial fluid stiffness, K_B, increases much faster with increasing shaft eccentricity. This means that as the rotor moves away from the center of the bearing in the SFCB, the stiffness (and therefore the restoring forces) are much larger and increase much faster. This tends to keep the rotor "self-centered" and also means that the SFCB has significantly better stiffness and load-carrying capacity than conventional bearings, particularly at low eccentricities. Figure 7a shows the Direct Dynamic Stiffness (DDS) for the drive end radial SFCB with the compressor at zero rotative speed. Because the perturbator can be rotated either with or counter to the direction of compressor shaft rotation, both forward and reverse values for DDS are shown, illustrating the symmetry. For illustrative purposes, the system's DDS is shown as two distinct

components: the bearing's stiffness, $K_{D(\,Bearing\,)}$, and the casing stiffness, $K_{D(\,Casing\,)}$. Figure 7b shows a similar plot with the compressor running at a constant speed of 7,000 rpm.

The following observations from these figures are noteworthy:

– The bearing provides very high stiffness at zero rotative speed.

– There is virtually no difference in stiffness characteristics between the compressor when stopped and when running at 7,000 rpm. This indicates that the system is extremely stable and that there is no fluid inertia effect or fluid circumferential average velocity ratio (λ) that can cause the system to go unstable.
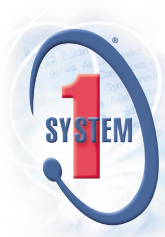
Summary

Experimental testing on a modified Clark 1M6 compressor resulted in clear understanding of the differences between behavior of a conventional

hydrodynamic sleeve bearing and Bently Nevada's new ServoFluid™ Control Bearing. Direct Dynamic Stiffness, λ (lambda), eccentricity ratio, independently adjustable damping, and attitude angle were all observed experimentally using perturbation techniques. The new bearing's design was found to have numerous advantages, including excellent stability from zero through operating speed. It was shown that conventional hydrodynamic bearings rely on their bearing characteristics to be generated at higher eccentricities (journal closer to the bearing wall), while the ServoFluid™ Control Bearing's characteristics are generated by the differential fluid flow across the journal surface – at much lower eccentricities.

More information on the ServoFluid™ Control Bearing is available in the brochure attached to this issue of ORBIT, from your nearest Bently Nevada sales or service professional, or at our website, www.bently.com. ↺