A NEW METHOD OF ACCURATELY IDENTIFYING THE LOCATION AND MOVEMENT OF FORWARD AND BACKWARD BENDING MODES USING MAGNETICALLY SUSPENDED ROTOR SYSTEMS

by

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ABSTRACT

Rotordynamic calculations done on most turbomachinery often discount or even completely ignore the “backward” modes. In fact, many rotating equipment engineers have never observed a backward mode, leading some to doubt their very existence. The author first observed such a mode in 1990, and has since routinely dealt with them, largely due to the ever increasing use of magnetic bearings in cryogenic turboexpanders. This paper will provide a brief background on these modes, discuss their importance in magnetic bearing systems, and present a novel method of measuring the frequencies of both forward and backward bending modes using a traditional Campbell diagram obtained by simply generating a cascade plot of the transfer functions during the run up or run down of the machine.

INTRODUCTION

In traditional beam type rotating machinery, the gyroscopic effects are usually relatively small. This can be better understood if a typical six stage centrifugal compressor rotor is considered. The six impellers are generally about the same diameter and spread out along the shaft relatively evenly, such that the diameter of the rotor is relatively small compared to its length. This means that the polar moment of inertia will be smaller than the transverse moment of inertia. It is the ratio of these two inertias that determines the gyroscopic effects that will be present when the rotor is running in the machine.

In a typical cryogenic turboexpander, the rotor is typically a “double overhung” design in which both impellers are located outboard of the bearings, and the impeller diameters are relatively large compared to the shaft, generally resulting in higher gyroscopic action than a typical beam type machine.

Despite the much larger gyroscopics, a typical turboexpander with oil lubricated bearings will not generally show any response to the high frequency forward and backward bending modes, since there is no means of exciting them. However, when the rotor is suspended on magnetic bearings these modes now have a means of excitation, and thus the modes are observable if the methods described in this paper are used to look for them.

DESCRIPTION OF TURBOEXPANDER

A cryogenic turboexpander is very similar in cross section to a turbocharger. There is a radial inflow expander impeller on one end of the shaft and a centrifugal compressor on the other end. In between these two impellers there is a bearing system, which supports the rotor on an oil film or else the rotor is magnetically suspended. This paper will deal mainly with rotors that are suspended in magnetic bearings since they are used in gathering the data discussed later in this paper. The general concepts regarding forward and backward bending modes, however, apply equally to rotors using any bearing system.

A typical rotor model layout is shown in Figure 1 to help the reader visualize the general rotor systems under consideration.

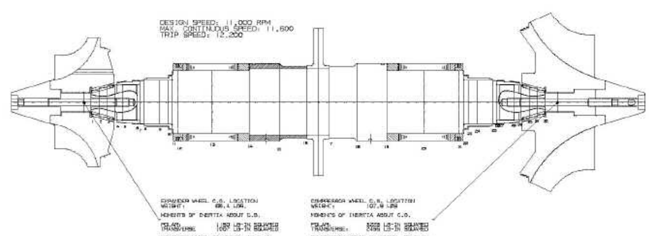

Figure 1. Typical Turboexpander Rotor Model.

DESCRIPTION OF MAGNETIC BEARINGS

Magnetic bearings are very common today in many cryogenic turboexpanders. Figure 2 shows a pictorial of how the rotor is supported in a housing containing magnetic bearings. Figure 3 shows the general method in which the radial shaft position is determined by the radial sensors and the relative locations of the magnetic and auxiliary bearings. Figure 4 shows that the sensors determine the position of the shaft within the bearing, then directs this information to a proportional, integral, derivative (PID) control algorithm that then varies the current to the individual bearings several thousand times per second to maintain the shaft in the proper location within the bearing.

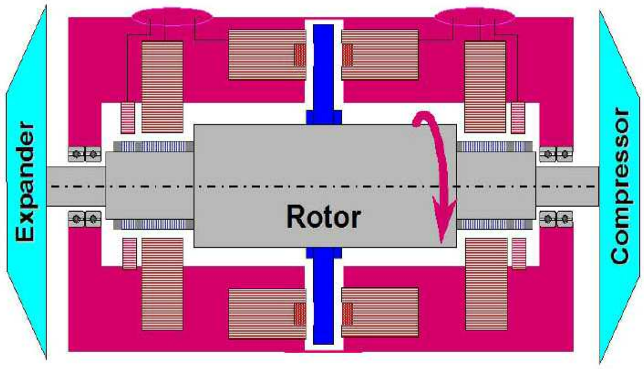

Figure 2. Turboexpander with Bearings.

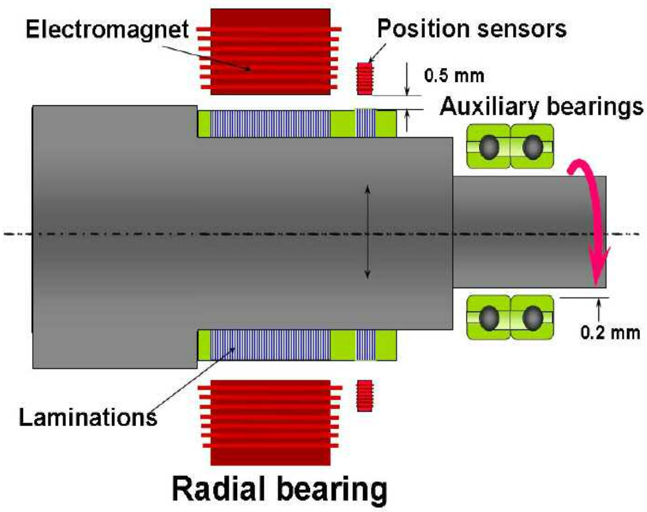

Figure 3. Radial Magnetic Bearing Configuration.

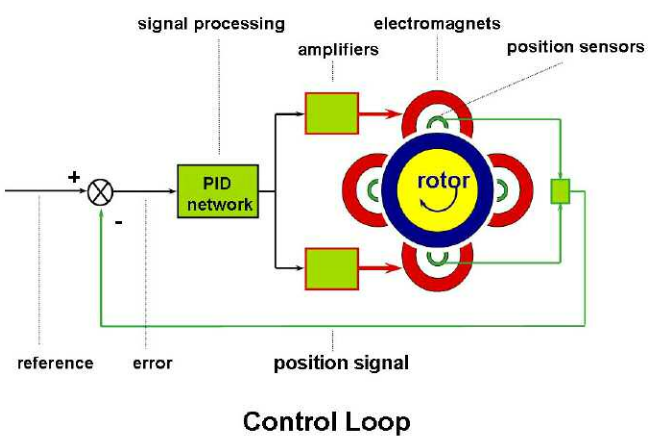

Figure 4. PID Control Loop.

Figure 5 shows a typical skidded package prior to installation in a major Gulf Coast refinery in the USA. This particular machine has been running successfully for approximately 10 years without being taken apart for any reason.

While magnetic bearings are considered “new technology” by many rotating equipment engineers, while writing this paper the author has just recently returned from the site of the first two magnetic bearing turboexpanders in North America. They were originally built in 1990, and are still successfully running today in a major ethylene plant in the USA. More information on this project in particular and magnetic bearings in general can be found in papers by Jumonville, et al. (1991), and Schmied (1991).

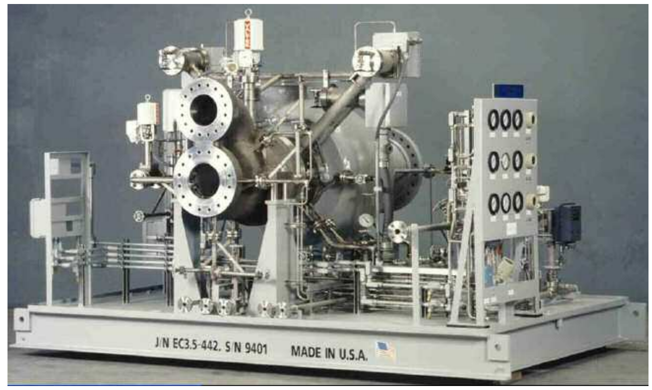

Figure 5. Typical Magnetic Bearing Turboexpander.

OIL VERSUS MAGNETIC BEARINGS

The area of concern for most beam type turbomachinery operating on oil lubricated bearings usually extends no higher than the first or second bending mode, since there is no reason to believe that the rotor will be excited at higher frequencies.

When magnetic bearings are used, the potential excitations from the control system itself extend to much higher frequencies, thus calculation and actual measurement of these higher bending modes is done to ensure the control loop is stable. In cryogenic turboexpanders, the rotor is usually operating below the first bending mode (both forward and backward), so there is no excitation of these higher modes to unbalance.

The reason that the higher modes are evaluated and measured is that the control system itself can excite them if the tuning of the control system is not optimized for these modes. The larger the bandwidth of the controller, the higher the number of modes that can be observed and need to be stabilized by the control system.

The difference between the two means of suspending the rotor can best be understood by an example. Suppose there are two identical rotors, one supported by oil bearings and the other supported by magnetic bearings. The oil bearing machine will have the rotor sitting in the bottom of the bearing at 0 rpm, and there is no possibility of exciting any of the modes under these conditions. The rotor still has many bending modes that can be calculated as natural frequencies of the system, but they have no means of excitation, thus the rotor just “sits there.”

The rotor suspended by magnetic bearings does not sit in the bottom of the bearing at 0 rpm, but rather is supported in the center of the bearing by the force of the magnetic field. Sensors observe the shaft position and feed the information to the control system thousands of times per second to adjust the current levels in the bearings and keep the rotor centered in the bearing, hence the term “active magnetic bearing.”

If the PID control loop for the active magnetic bearing is not tuned properly, the rotor could experience a resonance of perhaps the fifth or sixth bending mode even at 0 rpm! Because of this, accurate determination of these higher order bending modes is very important.

BACKWARD AND FORWARD BENDING MODES

When most rotating equipment engineers refer to the first bending mode, they are generally referring to the first *forward* bending mode, even though the first backward bending mode actually occurs at a lower frequency and is therefore usually closer to the running speed of the machine! In fact, many times the backward modes are not even calculated. This is probably because the backward modes are so rarely observed in typical oil lubricated turbomachinery that they are not deemed worthy of calculation.

The concept of backward modes of vibration in rotors is not new. *Mechanical Vibrations* by Hartog (1985) is based on a series of lectures given in the period 1926 to 1932, and the latest revision in 1985 is simply a reprint of the 1956 edition. In this excellent reference, Hartog gives detailed calculations relating to both forward and backward whirl, and yet he states "The author of this book looked around and asked his friends for fifteen years about it without results and was just about ready to conclude that the reverse critical speed was imaginary, when a case actually occurred" (Hartog, 1985). He also cites work by Aurel Stodola in his book "Steam and Gas Turbines" (Stodola, 1927) first published in English in 1927, in which he observed a "rough critical" that Stodola termed a "reverse whirl" or a "retrograde precession."

More recent books on rotordynamics such as the excellent one by Vance (1988) also give detailed calculations for backward and forward whirl frequencies. Vance makes the following comment in his book: "The actual occurrence of backward whirl in turbomachinery has been doubted in the past, but modern instrumentation confirms that it does occur. The author has observed it in his laboratory" (Vance, 1988).

As can be seen from these two references, for most oil lubricated turbomachinery the backward modes are rarely observed in practice other than laboratory test rotors, hence the lack of attention paid to them in traditional rotordynamics analysis.

By contrast, the use of turbomachinery with magnetic bearings makes the observation of the backward modes commonplace, especially in machines where the gyroscopic effects are large, such as some cryogenic turboexpanders.

It is important to note that backward and forward modes exist equally in both oil and magnetic bearing machines, the difference is that one can easily observe them in rotors suspended by active magnetic bearings, whereas one would need to build in a means of high frequency excitation to an oil bearing machine in order to observe these modes in oil bearing machines.

One last point that needs to be understood about both forward and backward modes is that the frequency of each mode changes with the speed of the rotor. In general, the backward and forward modes are the same frequency at 0 rpm, but as the rotor speed is increased, the forward mode rises in frequency while the backward mode drops in frequency. The amount that these modes change frequency is strongly related to the gyroscopic effects discussed earlier, and they change in a complex way for each bending mode, which can be seen later in the plots of actual measured data.

Obviously, the rotordynamic calculations for the higher order modes is inherently a nonsynchronous calculation, since there is no need to determine the response to unbalance for these modes, and in fact the modes would shift to different frequencies if the rotor attempted to run at these high speeds.

A "NEW" TYPE OF CAMPBELL DIAGRAM

The forward and backward bending modes of the rotor system are observable in magnetically suspended rotor systems because an external excitation signal can be injected into the bearing control circuit and the response of the rotor can be observed. When the response, or "output," is divided by the injected signal, or "input," the ratio of these two is called a transfer function. Because we are interested in the response within a given frequency range, a swept sine excitation signal is used.

Transfer functions are commonly used for tuning magnetic bearing systems, but typically they are done at a single speed. Because the forward and backward modes move with machine speed, and the magnitude of the response for these modes is usually very low, it can be very difficult to identify the forward and backward modes using a single transfer function at a single speed.

To facilitate this procedure, a method was developed by the author in which a cascade plot (often called a waterfall plot) of the transfer functions taken over a wide speed range is generated that essentially generates a Campbell diagram directly from the data taken during either a machine startup or rundown. This allows both the forward and backward modes to be identified at all speeds within the operating speed range. It has also been valuable in troubleshooting, as will be discussed later.

EXAMPLES OF MEASURED CAMPBELL DIAGRAMS

Figure 6 shows a typical Campbell diagram generated using the methods described above. The swept sine wave used for excitation of this machine was from 0 to 2000 Hz, as shown on the X axis. The line of peaks at "A" represents the 1× vibration. Well above this in frequency is the first backward and forward bending mode, shown at "B." The second backward and forward bending modes are shown at "C," the third at "D," and the fourth at "E." As can readily be seen, the backward modes all drop in frequency as the speed increases (higher speed lines occur as one moves up the Y axis), while the forward modes all rise in frequency as the speed increases. This behavior is what would normally be expected.

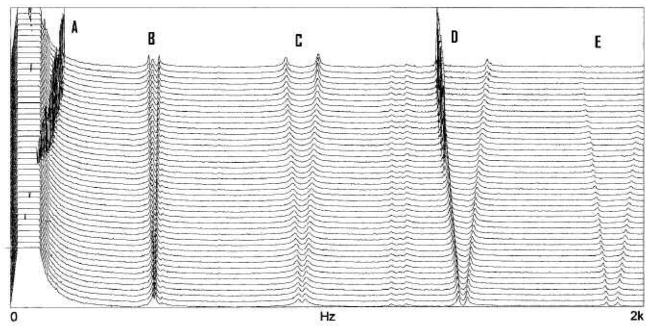

Figure 6. Campbell Diagram with 90 degree Axis.

Figure 7 shows the same plot as Figure 6, except that one axis has been tilted for a different view. It now becomes more obvious that the third backward bending mode has substantially more response beginning about 65 percent of full speed. This is due to inadequate damping in this region, and was easily fixed by changing the tuning parameters of the magnetic bearing control system.

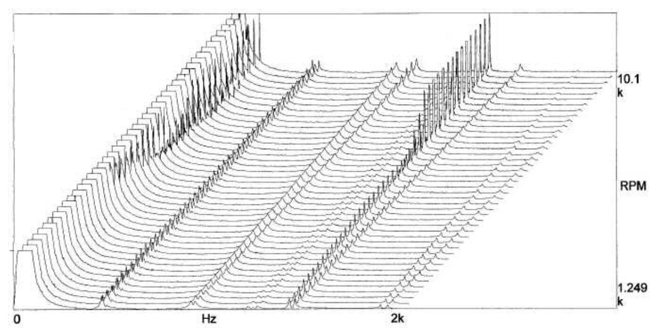

Figure 7. Campbell Diagram with Tilted Axis.

The author has found that for most plots, the data are most easily viewed when the axis angles are 90 degrees apart, as in Figure 6. However, as can be seen in Figure 7, there are instances when a tilted axis can be very useful.

Figure 8 contains a plot of the expander transfer functions on the top plot and the compressor transfer functions on the bottom plot. Recall that both the expander and compressor impellers are on a single shaft, so these plots are simply the two ends of the same rotor. This graphic shows many things, including that the response to a fixed excitation is not always the same for the forward and backward modes. At "A," the third backward mode responds much more than the forward mode to the same excitation. The 1× line for this machine is not visible, since it is buried in the "noise" to the left of the chart. However, the 7× harmonic is shown at "B." Notice that this system is well damped in this region, and thus the harmonic line crossing the forward and backward modes have very little effect, as shown at "C." At location "D," a stator resonance can be seen. This was not investigated to find the exact part that was resonating, but it was most likely a flanged stub pipe resonance or something similar. Note that this frequency does not change as the machine speed changes.

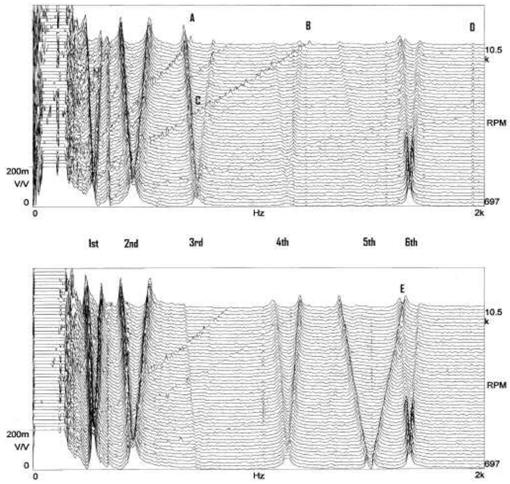

Figure 8. Expander on Top, Compressor on Bottom.

The lower half of Figure 8 at location "E" shows that the forward and backward modes can cross without any problems. It is also interesting to note that the fifth bending mode shown on this plot is strongly influenced by gyroscopic effects, hence the large change in natural frequency for a given speed change (a "wide vee"), compared to other modes, such as the adjacent sixth mode (a "narrow vee").

In Figure 9, there are three different plots. The lower plot shows the effect of 500 ft-lbs (680 N-m) of torque on the nut holding the impeller onto the tapered shaft. This is below the design value, and it can be seen at "A" that the forward bending mode begins to drop off in frequency at higher speeds. In the middle plot, the torque is increased to 550 ft-lbs (748 N-m) and an improvement is seen at "B." Finally, in the upper plot, the forward mode is nearly normal in shape at "C" with 575 ft-lbs (782 N-m) of torque on the nut. This shows the influence of wheel to shaft attachment integrity on both the forward and backward bending mode frequencies.

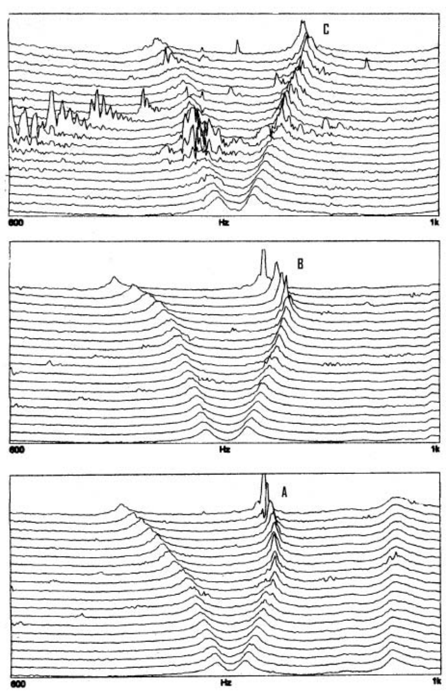

Figure 9. Shaft Attachment Effects on Natural Frequencies.

Figure 10 shows an interesting plot. As the speed on this rotor was increased, the first forward bending mode remains completely normal, but the first backward bending mode drops off rapidly. Note that this did not have any effect on normal operation of the machine, but the shaft to impeller fit was modified to eliminate the effect just to be on the safe side.

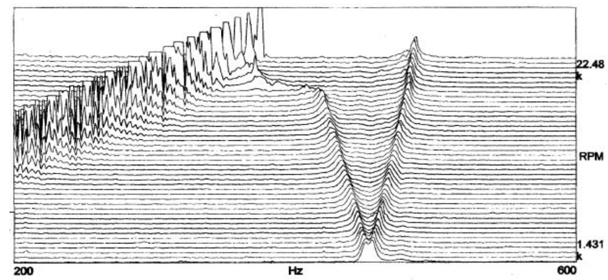

Figure 10. Rapid Backward Mode Shift Without a Change in Forward Mode.

Figure 11 shows an instance at "A" where the 2× harmonic intersects the second backward bending mode and shows a slight increase in response at this intersection, indicating that the mode is not as well damped as an earlier plot in which the harmonic had almost no effect. Points "B" and "C" show where the third and fourth forward and backward bending modes begin to drift down in frequency at a relatively low speed, despite having no indication of any problems at the first and second modes. This is believed to be due to the mode shape differences between the modes, where the impeller to shaft is more likely to flex when excited at the two higher modes. This effect is in stark contrast to the previous figure, in which the frequency drops very rapidly at a certain speed.

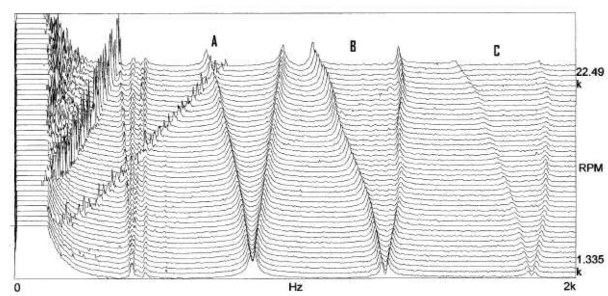

Figure 11. Slow Shift in Two Modes While Other Two Remain Unchanged.

Figure 12 is rather difficult to read, but it is included to show just how high this method will allow the bending modes to be viewed. This is a very high tip speed machine, and the data were taken during a rundown from 22,500 rpm. Marked on the plot are the sixth, seventh, eighth, and ninth measured forward and backward bending modes, obtained with a swept sine frequency range of 0 to 5000 Hz. This means that with a machine operating at 375 Hz, bending modes can accurately be observed, both forward and backward, at frequencies as high as 5000 Hz, or more than 1300 percent above running speed!

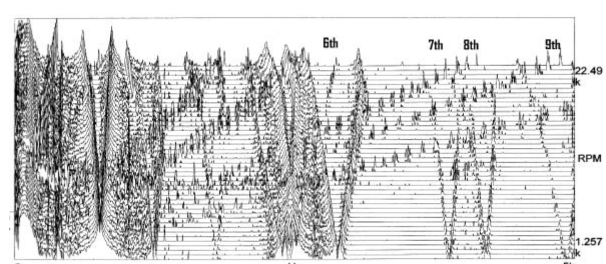

Figure 12. Accurate Measurement Up to Ninth Bending Mode.

The final plot is shown in Figure 13. This figure shows an unusual situation in which the fifth bending mode splits into *two* different bending modes, each having a forward and backward mode. This bifurcation is due to the flexibility of the impeller itself, though in other cases it can be due to the flexibility of the shaft to impeller attachment. When the impeller motion is in phase with the shaft motion, the frequency is lowered from the nominal "rigid attachment" value. When the impeller motion is out of phase with the shaft motion, the frequency is raised to a value slightly above the nominal "rigid attachment" value. This topic alone could be a separate paper, perhaps in future years, but it will not be addressed further at this time.

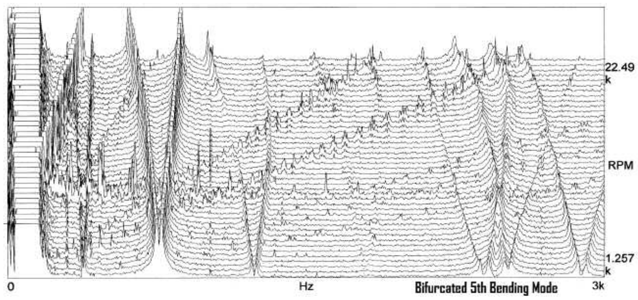

Figure 13. Fifth Bending Mode Split into Two Modes.

CONCLUSION

The methods presented here for obtaining accurately measured data regarding both the shape and magnitude of the Campbell diagrams should help shed new light on the subject of forward and backward modes for high speed turbomachinery. Few turbomachinery engineers have knowledge of or first hand experience with backward bending modes, and it is hoped that this paper will whet their appetite for obtaining more knowledge of this interesting subject. As machines are constantly being pushed to operate at higher speeds, and as magnetic bearings continue to find their way into ever larger and faster turbomachinery, it seems clear that a better understanding of these modes (and how to accurately model them) will become increasingly more important.

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