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# A New High-Speed Oil-Free Turbine Engine Rotordynamic Simulator Test Rig

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# **A New High-Speed Oil-Free Turbine Engine Rotordynamic Simulator Test Rig**

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## **Abstract**

A new test rig has been developed for simulating high-speed turbomachinery rotor systems using Oil-Free foil air bearing technology. Foil air bearings have been used in turbomachinery, primarily air cycle machines, for the past four decades to eliminate the need for oil lubrication. The goal of applying this bearing technology to other classes of turbomachinery has prompted the fabrication of this test rig. The facility gives bearing designers the capability to test potential bearing designs with shafts that simulate the rotating components of a target machine without the high cost of building “make-and-break” hardware. The data collected from this rig can be used to make design changes to the shaft and bearings in subsequent design iterations. This paper describes the new test rig and demonstrates its capabilities through the initial run with a simulated shaft system.

## **Introduction**

Foil air bearings are self-acting, hydrodynamic gas bearings with a flexible inner surface. The flexibility derives from the foil structure that acts as the bearing surface. In the bearings of interest to this paper, the foils are formed in the shape of corrugated bumps as seen in figure 1. The compliance of the bearing surface allows changes in the bearing geometry in response to local pressure variations in the fluid film. This flexibility of the bearing results in a thicker fluid film on average than would otherwise be present in a rigid air bearing. The benefits of this arrangement include tolerance of thermal and centrifugal distortions that would destroy conventional rigid bearings, as well as misalignment tolerance, increased damping from frictional rubbing of the foil structure, and the capability to tailor bearing properties to the application (refs. 1 to 4). With successful integration of foil air bearings into various classes of turbomachinery (such as gas turbine engines, turbo generators, auxiliary power units, etc.) safety, reliability, and performance will increase while weight will decrease (ref. 5). Thus, the driving force for the current interest in foil air bearing research is clear.

The Oil-Free Turbomachinery Program at NASA’s Glenn Research Center (GRC) has focused on advancing the state-of-the-art in foil bearing technology through research and collaboration with industry for the past 13 years. The testing capabilities at NASA GRC include high speed and high temperature load capacity and durability testing as well as coating design and development. While NASA GRC’s goal is to develop Oil-Free turbo engines, the role also extends to advocating for and fostering new applications for foil air bearings.

A process has been developed in conjunction with industry partners to bring new applications from concept to prototype that has worked well in the past (ref. 6). The main steps are: 1) Conduct a feasibility study to determine the potential for success and develop a detailed design including bearing and shaft geometry, 2) Build and test candidate bearings on a component level, 3) Build and test a simulated version of the machine of interest, 4) Build and test a working prototype. Each step may involve some iteration to arrive at a working design before moving on to the next step. Step three, simulated engine testing, can be thought of as a proof-of-concept before building a working engine, and is the focus of this work. Until now, GRC only had the capability to do the component level testing of foil air bearings. The test rig presented was conceived and built to extend GRC’s test capabilities into step 3.

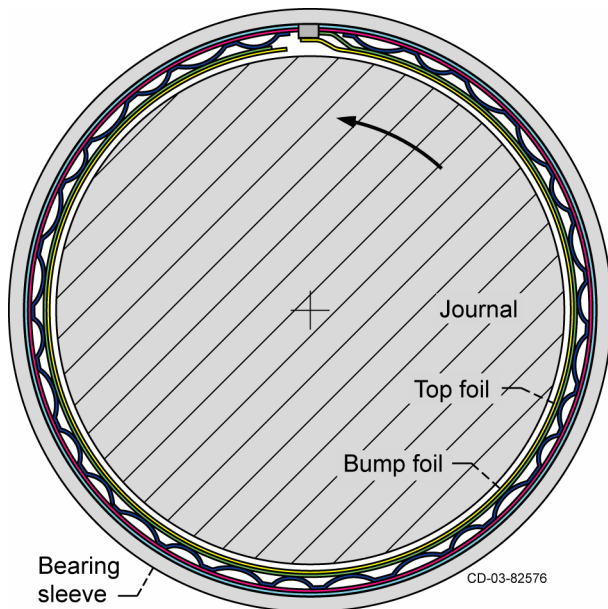


Figure 1.—Foil bearing cross section.

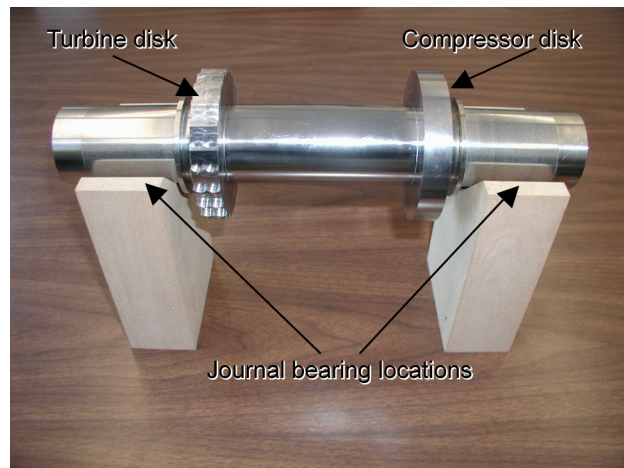


Figure 2.—Initial test rotor.

The rig has a modular design for flexibility in modeling different shaft configurations. The overall length, diameter, and mass of the shaft system can be changed as well as the positions of the components relative to one another to model various engines of interest. The initial rotor (fig. 2) has two foil journal bearings, a double acting thrust bearing immediately inboard from one of the journal bearings, and a turbine disk immediately inboard of the other journal bearing. The test rig is instrumented to collect acceleration, displacement, torque, and temperature data for rotor dynamic analysis of the system. This paper describes the test rig and demonstrates its capabilities through the initial run-up.

## Test Facility Design

The test rig is designed to be modular, meaning the individual shaft components can be moved relative to each other to represent different types of shaft systems. The major components found on a typical engine shaft are journal/thrust bearings, seals, compressor disk/s, and turbine disk/s. A majority of all turbomachinery systems have these components, but the layout can be very different. Some systems have single or few large compressor and turbine disks, others have many smaller disks. Some systems have disks between the bearings (straddle mounted configuration), others have the disks outboard of the bearings (overhung configuration), and any number of other configurations. Therefore, a desirable capability of the test facility is the ability to model many different shaft designs. The initial shaft can be seen in figure 2 and has a single simulated compressor disk and a single simulated turbine disk located inboard of two foil journal bearings. The modularity of the rig allows one to add disks, remove disks, move disks outboard of the bearings, etc. to model different rotor configurations.

The test rig mounts on a table that is 1.5 by 0.90 m (60 by 36 in.). With a containment shield in place, the maximum length of shaft that can be tested is approximately 1.2 m (48 in.). The bearings are held in two separate support structures that can be moved relative to one another to change the bearing span. The minimum span is approximately 200 mm (8 in.), and the maximum span is the same as the maximum length, 1.2 m (48 in.). The centerline of the shaft is 130 mm (5 in.) up from the table. Allowing for adequate space between the table and rotating hardware, the maximum disk that could be installed on a shaft is approximately 240 mm (9.5 in.) in diameter. The foil journal bearing size limitations are set by the bearing support and the bearing carriage. The maximum diameter for the foil journal bearings is 70 mm (2.75 in.) and the maximum length is 50 mm (2 in.).

The disks mentioned earlier are referred to as simulated because they do not function as they do in the engine being modeled. The compressor disk has no aerodynamic blades, and does no work. It is simply on the shaft to simulate the mass and gyroscopic properties of a compressor. Likewise, the simulated turbine disk does not function the same as the turbine in the actual machine, but adds mass and gyroscopic properties. Unlike the simulated compressor though, the simulated turbine is operational, and drives the shaft. It differs from the turbines in engines in that it functions as an impulse turbine with buckets rather than aerodynamic blades like an engine's turbine would have. This difference is for simplicity of design and manufacturing. It is symmetric about its axis, and the aerodynamic forces will be zero or very small in all directions except circumferentially. While this simplified turbine eliminates the ability to model the effects of certain aerodynamic forces (ref. 7) on the shaft, that is not the intention of the rig, and should not affect the measurement of the parameters of interest.

The main advantage of designing the rig with simulated aero components is the ability to run the rig on pressurized shop air instead of combustion air. This means the rig will run cool, and does not need to be exhausted to the outside. The disadvantage is the inability to completely model the thermal environment of an engine, and the aerodynamic forces already mentioned, both of which can have an effect on the rotordynamics of the system. However, valuable information about rotordynamics can be obtained by running this relatively simple, low temperature version of an engine. For example, if a particular bearing design performs poorly in this facility, it will most likely perform even worse in a more adverse (high temperature) environment. Also, analytical rotordynamic models can be verified using the rig and then extended to model the actual hardware. This can help direct design iterations in the direction of bearing improvements.

The two main structures (fig. 3) each house one of the two foil journal bearings. The bearings mount in a carriage, which in turn is mounted in a pair of high precision duplex ball bearings. Therefore, the foil bearings are free to rotate within the bearing support structures. The free rotation of the foil bearings has two purposes, to allow torque measurement and alignment of the two journal bearings. The torque measurement is discussed later.

There is an alignment system with two laser heads that mount to the rotating bearing carriages. Each laser head has a laser and a sensor. The laser in one head points at the sensor in the other head, and vice versa. The heads rotate with the bearing carriages, and after three distinct positions (3, 12, and 9 o'clock) are measured can calculate the misalignment of the bearing carriage centers, and indicate how to adjust the position of each support structure to improve the alignment. Each support bolts down to a bracket on the test rig tabletop with two slotted bolts. The brackets have positioning screws so that the supports can be moved small amounts linearly and angularly with respect to one another. The purpose of this movement is to allow horizontal alignment of the bearing centers using the laser alignment system. Vertical alignment is set using shims under the bearing supports. In this manner, the foil journal bearing alignment can be set to within tolerable limits. The resolution of the system is  $\pm 2.54 \mu\text{m}$  (0.0001 in.) and 0.0001 m/m (in./in.) for linear and angular misalignment, respectively. However, in practice the alignment can usually be set to within 25.4  $\mu\text{m}$  (0.001 in.) and 0.001 m/m (in./in.), respectively because finer adjustments are not generally possible using the relatively crude jacking screws and shims.

Axial forces on the rotor, such as thrust from the turbine, are constrained by a set of double-acting air foil thrust bearings (fig. 3). Double-acting means that two thrust bearings oppose each other, one on either side of a thrust runner (in this case the simulated compressor disk acts as a thrust runner). In this configuration, thrust loads in either axial direction are controlled.

The maximum capability of the rig in terms of speed is dependant on the geometry and material capabilities of the shaft. The smaller the shaft, the faster it can safely spin. The baseline shaft is 50 mm (2 in.) in diameter, with two 95 mm (3.75 in.) diameter disks, and two 50 mm (2 in.) diameter foil journal bearings. It is approximately 330 mm (13 in.) long, and weighs 31.4 N (7 lb). The design speed of this shaft is 62,000 rpm. In rolling element bearing terms, this equals to 3.1 million DN (Diameter in mm  $\times$  speed in rpm) for a rolling element bearing of the same size.

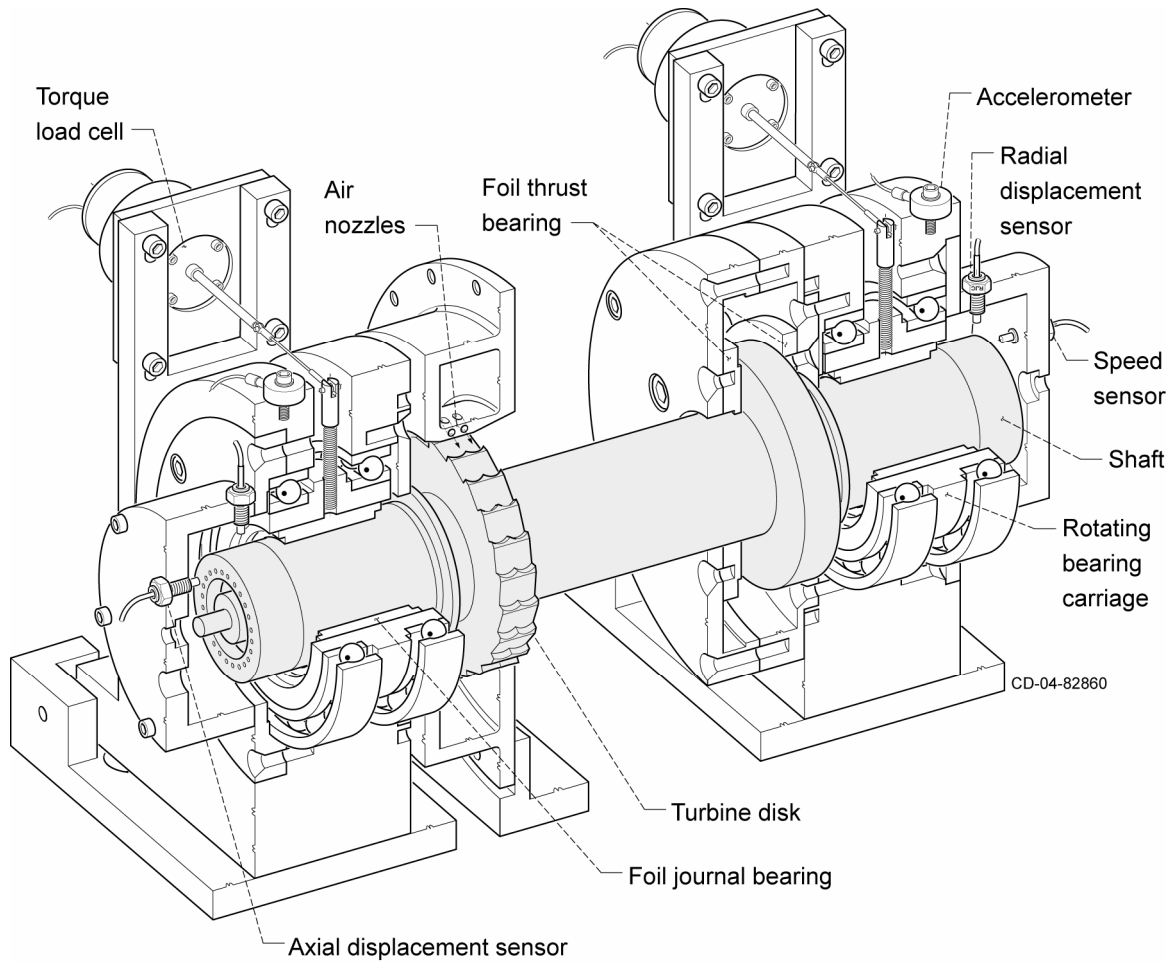


Figure 3.—Section view of the test rig.

The speed is limited by the maximum allowable stresses in the shaft which is determined by the geometry and material strength. In the event of a catastrophic failure of the shaft, a containment shield surrounds the test article, the thickness of which is chosen to contain a tri-burst type failure of the original 50 mm (2 in.) shaft at 78,500 rpm. While a detailed study of any shaft to be tested should be done to determine its maximum safe speed, an estimation of the containment capability of this shield can be made using the rotational energy of the proposed shaft design with the following relation

$$\text{rpm}_{\text{Max}} \approx \left(\frac{1}{\pi}\right) \sqrt{\frac{1.60e^8}{I_p}} \quad (1)$$

where  $I_p$  is the polar moment of inertia of the entire shaft (in units of  $\text{Nm}\cdot\text{s}^2$ ). This relation is based upon the maximum energy the shield can absorb for piercing failure assuming the shaft breaks into three equal sized pieces. This is the most severe type of failure, and represents the worst case scenario.

As stated earlier, virtually any shaft design that fits within the constraints of the test facility can be tested. The length is limited to 1.2 m (48 in.), the diameter to 70 mm (2.75 in.), and the maximum outside diameter (OD) of any disks must be 240 mm (9.5 in.) or less. The impulse type turbine, which is a simple design consisting of buckets machined into the OD of the turbine disk, drives the shaft. Air is delivered through a plenum manifold to several jets aimed tangentially into the buckets. The air exits the manifold



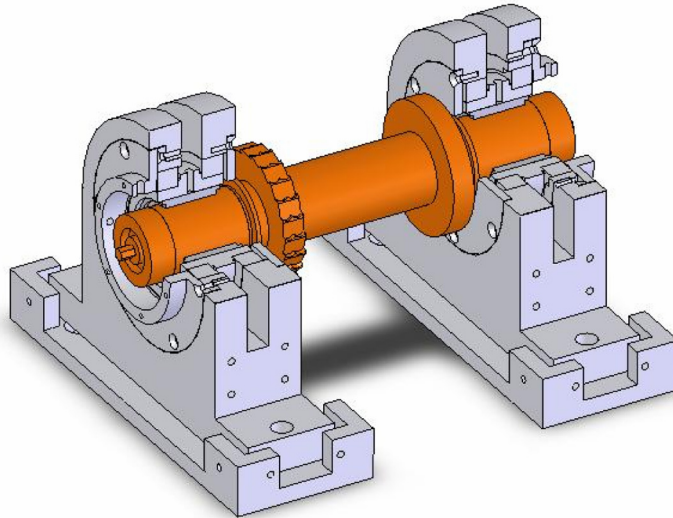


Figure 4.—Current shaft configuration with inboard masses.

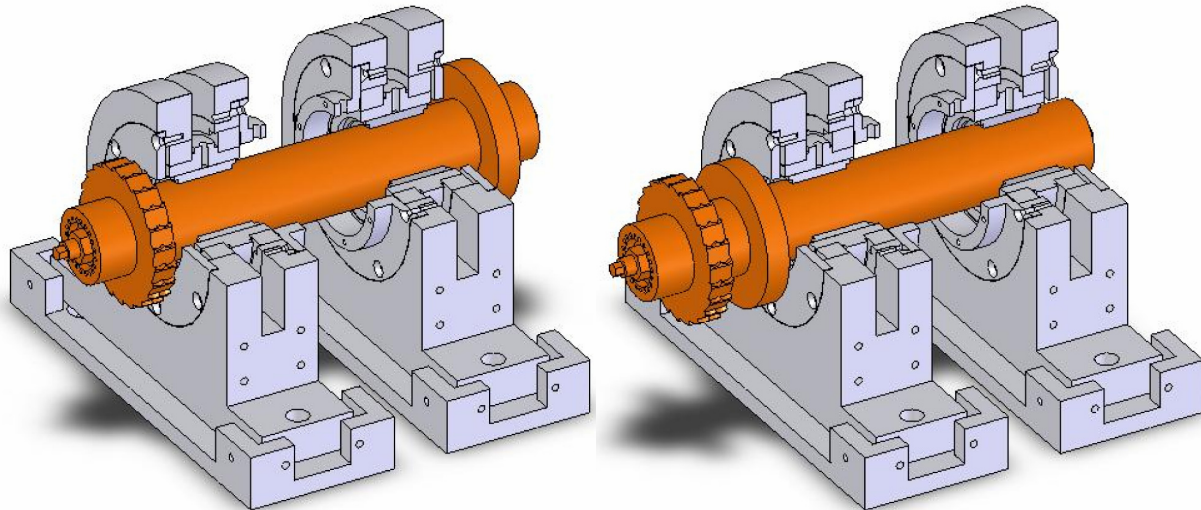


Figure 5.—Potential configuration with outboard masses. Figure 6.—Potential configuration with over-hung masses.

at high velocity and impinges on the buckets, forcing the shaft to rotate. The turbine disk and nozzle manifold can be positioned anywhere along the length of a test shaft as long as sufficient room exists between the nozzle structure and the bearing support structure. Figures 4 to 6 show the current configuration as well as two additional configurations that illustrate the potential flexibility of the facility.

### Test Facility Capabilities

The data acquisition capabilities of the test facility are typical of those of most rotor dynamic test rigs. There are four inductance type displacement transducers, two at each end of the shaft. One probe is mounted vertically and one horizontally in the same plane to measure the shaft motion at each bearing location (fig. 7). These sensors are used to plot orbit trajectories, and vibration amplitude on Bode plots. There is one other inductance type displacement probe to measure the shaft position along its axis of rotation. This information is mainly used to monitor the health of the thrust bearings and to identify any axial vibrations that may exist.

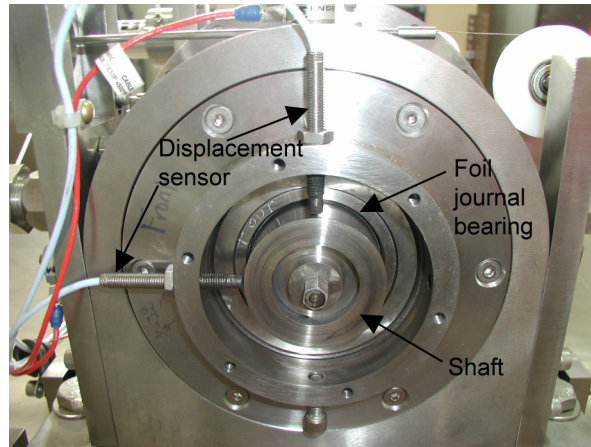


Figure 7.—End view showing displacement sensors.

In addition to displacement transducers, there are two piezoelectric accelerometers at each bearing location to measure structural vibration in the vertical and horizontal directions. Contrary to most rolling element bearing machines, Foil Air journal bearings are soft compared to the structure supporting them. In general, little force transmits to the structure. Therefore, accelerometers will not be as useful as the displacement sensors for vibration information. They are included in the data acquisition system to detect any structural vibrations, as well as being useful for potential future modifications whereby a shock could be applied to the structure to observe the shaft response to external stimuli.

As mentioned earlier, the foil bearings are mounted in rotating fixtures to collect torque data. To facilitate this measurement, there is a lever arm attached to each bearing fixture that extends upward through the support structure. The lever arms are attached at their free end to a load cell through a cable such that the force necessary to restrain the bearing from rotation can be measured, figure 3. Unfortunately running torque measurements of the air foil bearings cannot be made because the friction in the ball bearings holding the fixture is greater than the torque in the foil bearings. However, the system is still useful to warn of an imminent foil bearing failure because the torque in the foil bearing overcomes the ball bearing friction as failure approaches. Thus, absolute torque values cannot be measured, but failure can generally be anticipated using the torque measurement system. Future enhancements may include reduced friction ball bearings or hydrostatic bearings.

Speed measurements are made by using a fiber optic based displacement transducer that detects a dark spot on the end of the shaft. For every shaft rotation, the dark spot passes by the sensor and sends a low voltage signal to the data acquisition system, which is otherwise a high voltage signal. In this manner, the speed signal takes the form of a square wave with frequency equal to the run speed. The data acquisition system can count that frequency with high accuracy to determine the running speed.

The data acquisition system is highly customized, and was developed using a commercially available graphical programming language. It has the standard analysis capabilities of other rotor dynamic packages including Bode plots, waterfall plots, orbit trajectories, Fast Fourier Transforms (FFT), as well as data logging and playback.

## Sample Output

The following is a discussion of some sample output from the test facility to illustrate its capabilities. This output was obtained using the initial rotor configuration in figure 2. Figure 8 shows a plot of horizontal and vertical displacement at the turbine end of the rotor. The displacement is filtered to show only the synchronous frequency component of vibration, which is useful for identifying critical speeds as well as trim balancing the rotor system. The plot shows a vibration trace during an acceleration from about 5,000 to 56,000 rpm.

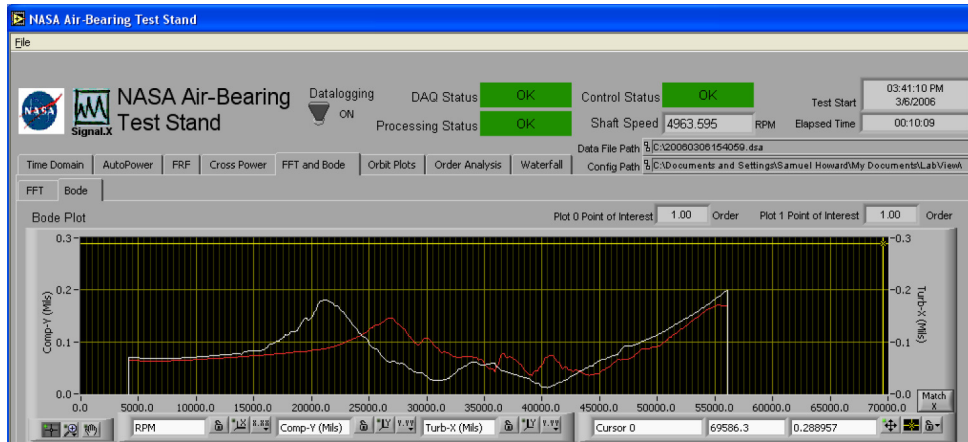


Figure 8.—Bode plot of horizontal and vertical synchronous vibration at the turbine end.



Figure 9.—FFT plots showing frequency and order components respectively at the compressor end of the rotor.

Figure 9 is a Fast Fourier Transform (FFT) plot of the vibration data. This plot shows the vibration amplitude at all frequencies and is useful for determining possible sources of excitation that cause vibration. For example, high sub-synchronous vibration levels, at frequencies near one half the run speed, are often associated with a type of self-excited instability known as whirl. Synchronous, or one times the run speed vibrations are usually the result of unbalance. Two times the run speed, or second harmonic

vibrations can be caused by bearing misalignment or a bent shaft, and so on (ref. 8). Thus, diagnosing vibration problems can be made easier by looking at the frequency components of the vibration signal. The data acquisition software can display the FFT data in the frequency domain, where the original vibration signal is broken down into its frequency components, or in the order domain, where the original vibration signal is decomposed into order components. An order is a multiple of the running speed. So, a synchronous vibration component is the first order component, a second harmonic component is the second order component, and so forth. The distinction between order domain and frequency domain is that a fixed order vibration (synchronous for example) will always appear at the same order in the order domain even if the running speed changes. Conversely, a synchronous vibration will move in the frequency domain as the speed changes. Likewise, a fixed frequency component will appear at a fixed frequency in the frequency domain, but will move in the order domain as the speed changes. The order domain is convenient if one is tracking a vibration that is fixed to the run speed, and the frequency domain is convenient for tracking a fixed frequency vibration. The plot in figure 8, for example is constructed by tracking the magnitude of the first order (synchronous) vibration component in the order domain as the speed is increased from 5,000 to 56,000 rpm.

A useful visualization tool for monitoring rotating machinery health is the orbit plot. Figure 10 shows two orbit trajectories for the sample rotor of figure 2. The rotor trajectory is a convenient way to visualize the actual motion of the rotor in a “real time” format. There exists a wealth of knowledge that can help the turbomachinery operator interpret the health of a running machine by observing the orbit trajectories (ref. 9). For example, the orbit plot offers a way to discern subsynchronous vibration quickly while running a machine. As mentioned above, subsynchronous vibrations are often associated with self-excited instability, that can be the source of failure. A subsynchronous vibration will appear as an internal loop on the primary orbit trajectory. For example, the compressor (left) orbit in figure 11 shows two internal loops due to two subsynchronous frequencies. An operator can quickly see this and take the appropriate corrective action. Orbit trajectories can also be useful in a research capacity for matching analytical models to the “real world” and observation of different stimuli effects on rotor behavior.

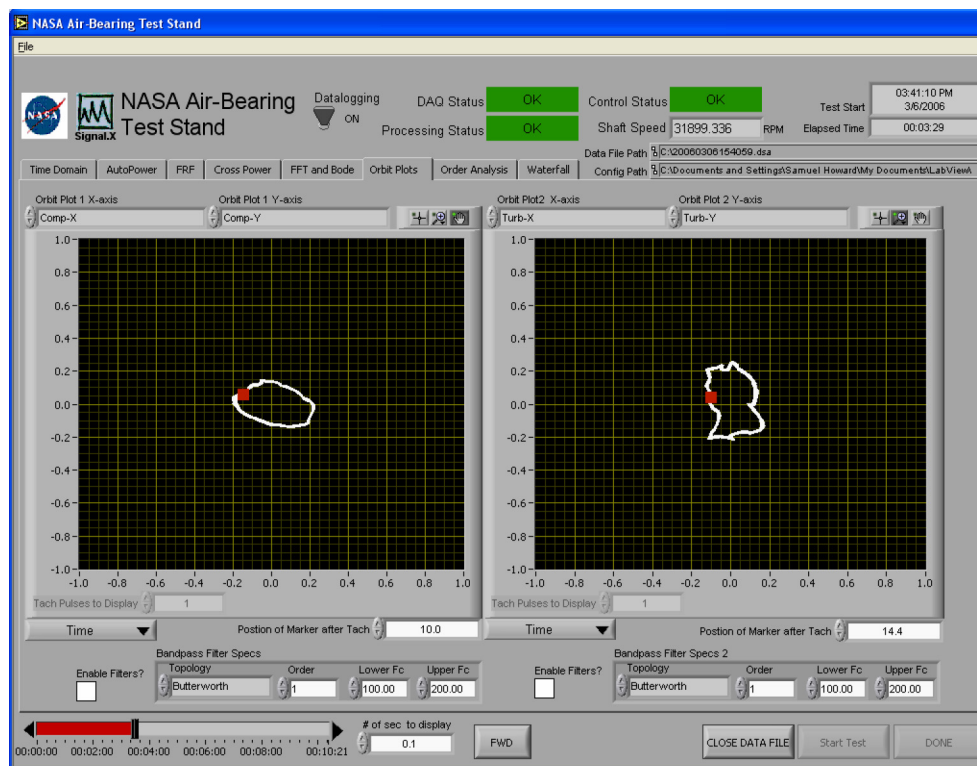


Figure 10.—Orbit trajectories at both ends of the rotor.

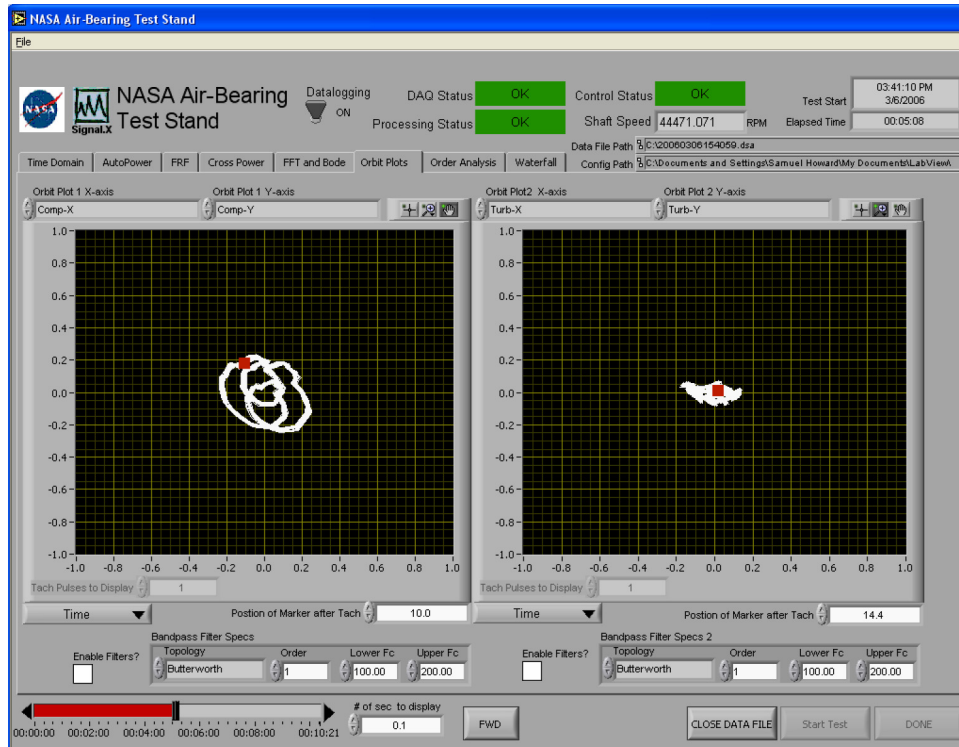


Figure 11.—Orbit trajectories showing subsynchronous vibration components.

## Discussion and Conclusion

A new foil bearing test rig has been developed to conduct research on the rotordynamic behavior of Oil-Free turbomachinery. The facility offers the capability to simulate the operation of a wide range of turbomachines of different configurations. A data acquisition system and analysis software package has been developed to aid in interpretation of the rotor system characteristics. It is anticipated the new facility will give foil bearing and turbomachinery designers a useful tool for the development of advanced Oil-Free turbomachinery technology.

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