



Integration Methodology For Oil-Free Shaft Support Systems: Four Steps to Success

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Abstract

Commercial applications for Oil-Free turbomachinery are slowly becoming a reality. Micro-turbine generators, high-speed electric motors, and electrically driven centrifugal blowers are a few examples of products available in today's commercial marketplace. Gas foil bearing technology makes most of these applications possible.

A significant volume of component level research has led to recent acceptance of gas foil bearings in several specialized applications, including those mentioned above. Component tests identifying such characteristics as load carrying capacity, power loss, thermal behavior, rotordynamic coefficients, etc. all help the engineer design foil bearing machines, but the development process can be just as important. As the technology gains momentum and acceptance in a wider array of machinery, the complexity and variety of applications will grow beyond the current class of machines. Following a robust integration methodology will help improve the probability of successful development of future Oil-Free turbomachinery. This paper describes a previously successful four-step integration methodology used in the development of several Oil-Free turbomachines. Proper application of the methods put forward here enable successful design of Oil-Free turbomachinery. In addition when significant design changes or unique machinery are developed, this four-step process must be considered.

Nomenclature

$C_{\alpha\alpha}$	damping (kNs/m, α = matrix index)
D	shaft diameter (m)
$K_{\alpha\alpha}$	stiffness (MN/m, α = matrix index)
L	bearing axial length (m)
S'	modified Sommerfeld number in air
	$S' = \frac{D(D\omega)}{W/DL}$
W_{applied}	applied bearing load (N)
W_{LC}	load capacity at speed (N)
D	journal bearing performance coefficient (N/m ³ /k _{rpm})
ω	shaft speed (krpm)
ω_{lo}	loft-off speed (krpm)
ω_{min}	minimum speed (krpm)

Introduction

In the realm of traditional turbomachinery, there is a wealth of historical, analytical, and experimental experience to bring to bear when designing a new machine. Many "new" turbomachines with conventional bearing technology are really improvements of existing designs involving small, incremental, and evolutionary changes. As a result, often the approach is to use a design that has worked well in the past, or to slightly modify an existing design to enhance its capabilities. As long as the new application does not extend beyond the bounds of prior art, design guidelines and tools can be used with success. With Oil-Free turbomachinery, this is typically not the case. Usually, these machines are unique with little historical knowledge from which to draw. In addition, there is less analytical and experimental data in the literature than with conventional bearings. Thus, when developing new foil bearing turbomachines, it stands to reason that risk minimization would necessitate a well-defined method of analysis combined with experimental validation.

That necessity is at the heart of the integration methodology outlined in this paper. Experience has taught that all development programs negotiate through a similar process, but the most successful plan for it from the outset. The basic methodology hinges on using analysis to conduct preliminary design and layout trade studies followed by experimental verification of the resulting design with iteration possible anywhere along the way. Formally, the process breaks down into four distinct steps: 1) Preliminary feasibility and design layout, 2) Component level bearing testing, 3) System level rotor simulator testing, 4) Final design full-scale demonstration.

While success can never be guaranteed, the four-step process maximizes its probability and helps identify applications that have high risk of failure at the full-scale level. Many proprietary examples exist of attempts to develop new turbomachines without following all four steps, only to result in failure that could have been avoided. This paper outlines the four steps of the development process with the goal of showing how they can be utilized when developing hardware.

Foil Gas Bearings

The four-step integration methodology can be readily applied to any development program, but it is especially

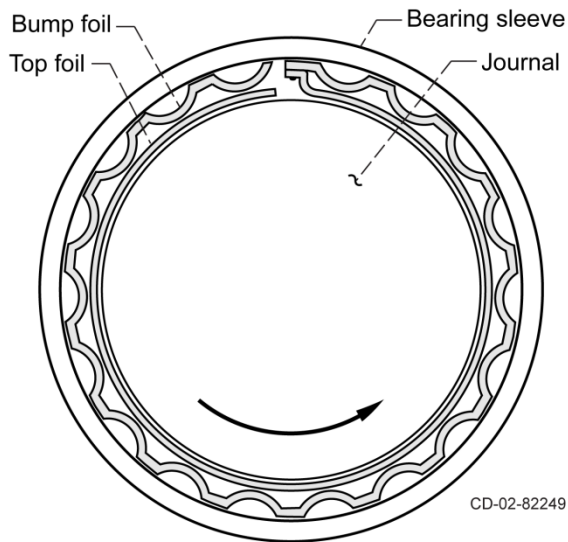


Figure 1.—Cross-section view of a typical Generation I (Ref. 7) bump-type radial foil bearing.

important when a new application is outside the previous design space. This is especially true in the development of Oil-Free turbomachinery. Since many Oil-Free applications utilize gas foil bearings, an introductory discussion on their operation and construction follows.

Gas foil bearings (Fig. 1) are gas lubricated, hydrodynamic bearings with a compliant inner sleeve surface. The compliant surface is usually comprised of several layers of sheet metal, called foils. One or more layers of the sheet metal foil are formed into bumps (bump-type foil bearing) or other structural element (overlapping leaf, etc) that behaves as a spring. The compliant layer is covered by one or more sheets of flat metal foil, called the top foil. The top foil acts as the bearing inner surface, against which the hydrodynamic pressure acts to support the shaft weight.

When the rotor spins, pressure is developed in the gas film between the shaft and top foil. As the speed increases, the pressure increases, and eventually equals that of the weight of the rotor, suspending the rotor on a film of air. Once the rotor is suspended, there is nominally no contact between the rotor and the top foil surface.

The foil bearing operating principle just described makes clear some of the advantages of foil bearings over other types of bearings. The advantages of gas foil bearings make them attractive for application to gas turbine engines and other turbomachinery (Ref. 1). Owing to the low viscosity of air relative to other lubricants, the torque (or power loss) in gas foil bearings is low compared to most other bearing types. The gas film does not degrade at temperatures up to the temperature limits of the bearing materials, as liquid lubricants do, resulting in high temperature capability. Once the bearing is lifted off, there is no wear giving foil bearings long life and minimal maintenance. The compliance enables the foil bearing to expand and conform to the pressure generated in the lubricating film. While rigid gas bearings can be vulnerable to

seizure due to thermal and centrifugal distortion and growth, the compliance of gas foil bearings allows them to tolerate shock loads, misalignment and thermal and centrifugal growth (Ref. 2). In addition, the performance characteristics of gas foil bearings can be tuned by varying the different geometry parameters associated with the foil structure, such as bump pitch, foil thickness, etc (Refs. 3 and 4).

As rotor support elements in turbomachinery, foil bearings also have a few limitations in comparison to other bearing types. The low viscosity and compressibility of the gas lubricant results in low load capacity compared to liquid-lubricant hydrodynamic bearings and rolling element bearings of similar size. However, the load capacity does increase with a linear relationship to speed (to a point), resulting in useful load capacities at high speed. Therefore, foil bearings are generally well-suited to high-speed, lightly-loaded turbomachines (Ref. 5). In addition, the damping due to viscous energy dissipation is low compared to oil-lubricated bearings or squeeze film dampers. However, there is evidence that frictional damping in the foils adds stability to foil bearing systems, and typically low cross-coupled stiffness results in less need for high damping (Ref. 6). These unique characteristics of gas foil bearings increase the importance of exercising a well-defined development methodology when designing new machinery.

Four-Step Integration Methodology: Introduction and Definition

The four-step method for integrated design of a new turbomachinery system is outlined in flowchart form in Figure 2. The flowchart is intended to help visualize all the process steps, and the potential for multiple iterations within and across all four steps.

The first step, preliminary feasibility and design layout, is to determine if a proposed application is plausible from the standpoint of bearing performance and system rotordynamics. During this step, trade studies of various rotor configurations are performed to determine bearing loads and sizing, bearing positions (straddle mount, overhung, etc.), calculation of critical speeds, and any other basic information needed to down-select a rotor configuration for further consideration.

In step two, component level testing, the bearings chosen in step one are fabricated and subjected to a battery of tests to verify their capability to perform as needed in the system. Examples of tests include load capacity, power loss, start-stop durability, stiffness and damping, thermal management, etc.

Step three is a rotordynamic simulation intended to ensure that the system designed in steps one and two performs as expected. This is the step that usually gets skipped to the detriment of a development program. Rotordynamic simulation is an important step because it helps to identify system level integration issues that cannot be vetted in steps one and two. It has been the authors' experience that bypassing this step was the cause of many failures experienced in development programs that went from design straight to

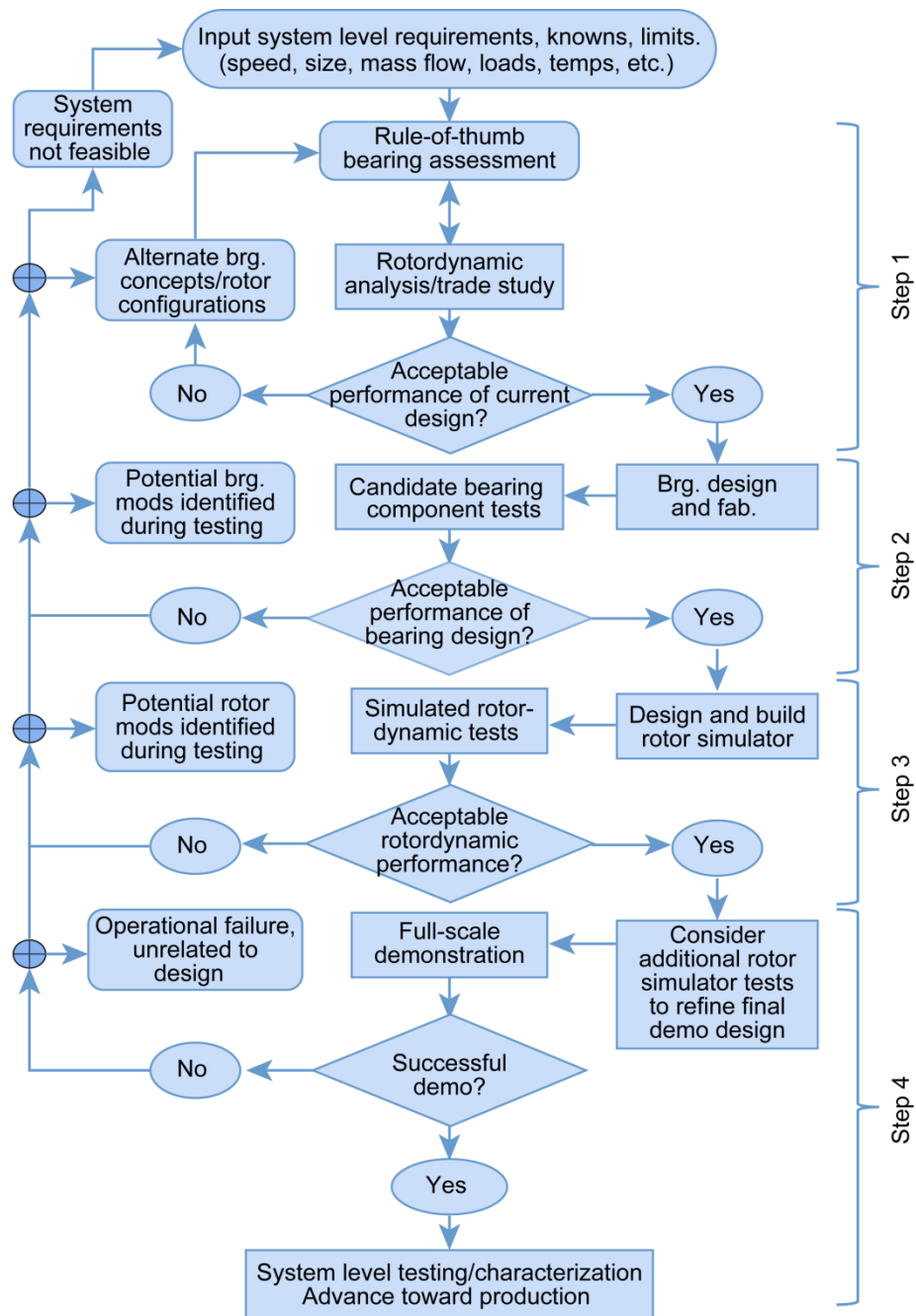


Figure 2.—Flowchart of the four-step integration methodology.

full-scale demonstration. The authors have witnessed program cancellations due to failures of demo hardware that could have been avoided by operating a simulated rotor system. It seems as if there is generally less tolerance of failures when real hardware is involved compare to simulated test hardware. Step three is intended to identify and correct any of these potential unknown issues in a low-cost, simplified test rig before developing in a full-scale demonstration unit.

Step three is implemented by building a rotor system that matches the layout of the actual machine, but with simulated aerodynamic components. Building a rotor with non-functional turbine and compressor components affords an opportunity to test the rotor system without the complications of the thermodynamic process. The benefits of simulated rotor components include easy and low-cost production and testing, if failures occur they are usually non-critical and less costly to

repair, changes can be made faster and at less cost, and more. Additionally, the rotordynamic behavior of the simulated system can be checked against the predicted behavior and changes can be made to optimize the system before any final designs are made. If warranted, the simulated rotor system can be intentionally pushed to failure to identify failure modes and true limits of operation.

The last step is full-scale demonstration. This step involves building a highly instrumented, production-like machine and running it through a series of qualification tests. By the time a development program gets to the fourth step most, if not all, of the design changes should be complete, leaving only to verify that the machine performance meets the design criteria. Small changes may still be needed to realize the desired operation of the new machine in the final step, but most of the integration issues should have been identified and corrected. In the following sections, a simple example of each of the first three steps is given.

Four-Step Integration Method: Example Application—Simulated Turbocompressor

An existing test rig that simulates a small turbocompressor is used as the surrogate turbomachine in this example in order to model realistic hardware. The first three steps are illustrated with sample calculations to demonstrate how one might utilize the four-step method to develop a piece of hardware. The fourth step, demonstration of the final design, is self-evident and only applies to a production machine, and therefore is not included here. It is hoped by navigating through the first three steps in this example readers will understand how this method can be used to methodically progress through a design process to minimize the risk involved in developing new hardware. Also, the example provides an opportunity to demonstrate an array of design tools for gas foil bearings that have emerged from research in the past decade and how one can utilize these tools in practice.

The first objective of step one is to size the bearings. Many factors play a role in dictating the size of the shafting, and therefore the bearing size in turbomachines. Often times, the desired function of the machine dictates much of the design. For example, the process fluid, flow rate, pressures, temperatures, power, etc. are all considered in designing the aerodynamic components. Once the aero components are roughly sized for the desired performance, a rough estimate of the mass of the machine is known. Additionally, the aero components generally dictate the rotational speed and the power transmitted through the shaft. With that information, one can begin to characterize the design space. For this discussion, it is assumed the aero components are designed a priori to do a specific function, and are not part of the four-step integration effort. If, however, that is not the case for a particular application, the aero component design can be incorporated into the four-step process and can evolve iteratively along with the rotor and bearing designs. For instance, if the speed required by the compressor is too low, one

could use a smaller wheel at higher speed. Once the weight and torque constraints are known, the bearing size can be selected.

For gas foil bearings, the correct size is based on design criteria related to load capacity and power loss. First, it is necessary to ensure the load capacity of the bearings is greater than the respective load on each of the bearings. This can be accomplished utilizing the foil bearing load capacity rule-of-thumb (ROT) developed by DellaCorte and Valco (Ref. 7). The ROT states that a foil bearing can support a load proportional to its projected area multiplied by its surface velocity. Specifically, Equation (1) gives the empirically derived relationship.

$$W_{LC} = D \times (L \times D) \times (D \times \omega) = D \times LD^2\omega \quad (1)$$

where D , the performance coefficient, depends upon the bearing design and other parameters such as process fluid, temperature, ambient pressure, etc.

For our simple example, assume a system study predicted a rotor weight of about 67.2 N, with a desired speed range of 10,000 to 30,000 rpm. The load capacity ROT is used to pick a bearing size as follows:

For simplicity, the rotor is assumed to be symmetric with half the weight supported by each of two journal foil air bearings. The best available foil bearing technology will be used, meaning the performance coefficient is around 2.71×10^5 N/m³/krpm (1.0 lb/in.³/krpm) (Ref. 7). At this stage of the design, it is reasonable to choose an L/D ratio for the bearings of 1. The load capacity ROT predicts that the minimum bearing size needed to support the weight of the rotor is about 23-mm diameter by 23-mm long (0.90- by 0.90-in.) using 10,000 rpm ($\omega = 10$ krpm) as the minimum speed.

However, in addition to providing enough load capacity, the bearings should operate in a low power loss state as defined by the performance map first explained in DellaCorte, et.al. (Ref. 8), and later expanded in Radil and DellaCorte (Ref. 9). A sample performance map is shown in Figure 3.

The performance map plots power loss versus a modified Sommerfeld number (speed/load parameter), or versus load and speed in three dimensions, and makes clear two distinct operating regimes. As shown in Figure 3, there is a point in the operating space of a foil bearing where the power loss is at a minimum. To the right of the minimum power loss (higher Sommerfeld number = higher speed and/or lower load), the bearing operates in a lightly loaded, thick film regime. To the left of the minimum (lower speed and/or higher load), the bearing is operating in a highly loaded, thin film regime. It is desired to be to the right of the minimum power loss, in the thick film regime because this region of the operating envelope is stable, meaning that small changes in running conditions correct themselves in time. Operating near the minimum power loss, or to the left of it is thermally unstable because small perturbations tend to grow until failure is experienced. A more detailed discussion is given in (Ref. 9).

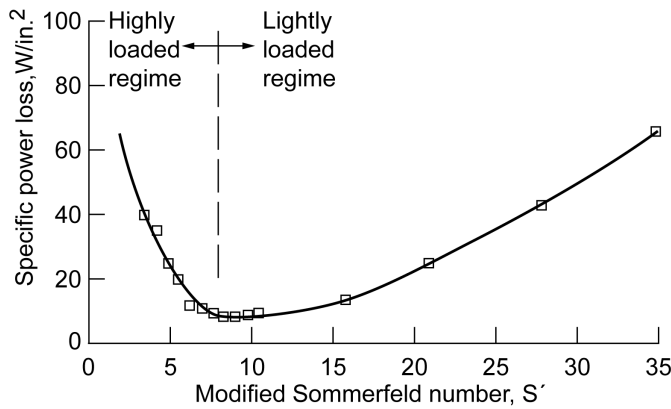


Figure 3.—Sample performance map for a gas foil journal bearing.

Without having a bearing to test, which is likely to be the case in step one, it is not possible to generate a performance map, and therefore, it is not possible to know where the minimum power loss occurs. In the absence of the power loss map, the load capacity ROT can be used to estimate the minimum power loss speed for a given load. A loose correlation has been experimentally observed that the power loss minimum occurs at roughly 2 to 6 times the lift-off speed for a given load. The load capacity ROT can be used in reverse to estimate a lift-off speed for a known load because the lift-off speed is the speed at which the applied load is the load capacity. Rearranging Equation (1) gives:

$$\omega_{LO} = \frac{W_{\text{applied}}}{DLD^2} \quad (2)$$

Rearranging to solve for the bearing size, and making the substitutions: $D = L$ (from $L/D = 1$), and $D = 2.71 \times 10^5$, gives:

$$D^3 = \frac{W_{\text{applied}}}{2.71 \times 10^5 \omega_{LO}} \quad (3)$$

In order to satisfy the power loss requirement of 2 to 6 times the lift-off speed as a minimum operating speed, use:

$$\omega_{LO} = \left(\frac{1}{2} \text{ to } \frac{1}{6} \right) \omega_{\text{min}} \quad (4)$$

Then, for the above example with $W_{\text{applied}} = 33.6$ N per bearing (7.55 lb), $\omega_{LO} = 1/2$ to $1/6$ (10 krpm) = 5.0 to 1.67 krpm, the approximate desired minimum bearing size ranges from 29 mm (~1.2 in.) to 42 mm (~1.7 in.) diameter. A designer might use his experience to determine a bearing size based on this range. Many times, a trade-off between competing design criteria will feed into this decision. For example, one may choose to pick a bearing at the large end of the range to ensure operation to the right of the power loss minimum. Alternately,

there might be a maximum size limitation for the given application, or a minimum rotor diameter to transmit the necessary torque. The usefulness of the ROT is to provide a reasonable range for a starting point. The example test rig utilizes 50.8-mm diameter by 40.6-mm long (2.0- by 1.6-in.) bearings. This bearing size is well above the minimum for load capacity, and also above the minimum acceptable range of diameter based on power loss. The L/D ratio < 1 was a matter of availability of bearings. A 50.8- by 40.6-mm bearing was available as a standard size from the bearing supplier. Using the ROT, the 50.8- by 40.6-mm bearing yields a lift-off speed of about 1200 rpm. Thus, the bearings may be oversized for the application. Regardless, the bearing design chosen in this manner then becomes the preliminary input to the feasibility study. If the choice does not satisfy the design criteria as the process progresses, it can be modified in a future iteration.

Once a tentative bearing size is chosen, dynamic coefficients can be predicted using one of the techniques outlined in various papers (Refs. 10 and 11), or one's own in-house analysis techniques. Then, a rotordynamic analysis is done to assess the feasibility of the proposed design and generate a rotor layout. The rotordynamic layout trade study with the bearing characteristics for the tentative bearing size, and an idea of what the rotor will look like, is used to predict the system critical speeds, stability threshold, unbalance response, etc. to ensure the design criteria are met. If not, the layout or the bearing size might be modified and another iteration performed until the rotor behaves as desired.

Using the test rig as an example, the 50.8- by 40.6-mm bearings are analyzed using the technique developed by San Andres and Kim (Ref. 10) to determine dynamic coefficients. The resulting bearing coefficients are given in Table 1.

TABLE 1.—STIFFNESS AND DAMPING COEFFICIENTS FOR 50.8- BY 40.6-mm FOIL BEARING.

[Evaluated at frequency = synchronous speed, static load = 33.6 N (in Y direction)]

Speed (rpm)	Kxx (MN/m)	Kxy	Kyx	Kyy	Cxx (kNs/m)	Cxy	Cyx	Cyy
5000	2.5	-1.6	-2.5	5.9	2.6	-2.5	0.63	5.9
10000	2.8	-1.1	-2.0	4.8	1.8	-1.2	0.62	2.6
15000	3.0	-0.78	-1.6	4.4	1.4	-0.75	0.56	1.6
20000	3.3	-0.54	-1.3	4.2	1.1	-0.55	0.51	1.2
25000	3.6	-0.38	-1.0	4.2	1.0	-0.45	0.45	0.91
30000	3.8	-0.26	-0.80	4.2	0.86	-0.38	0.39	0.76

Using the bearing data from an analysis code, one can conduct a rotordynamic trade study and system layout. The process can involve iteration back and forth between bearing design and rotor layout in the search for a feasible design. Since the test rig already exists, this step is used to illustrate what one iteration might look like in a design program. The rotor geometry is modeled in a rotordynamic analysis package

using the bearing data from Table 1. A sketch of the test rig is shown in Figure 4. The primary result of step one of the integration method is a bearing design and rotor layout that can be carried forward into the later steps for testing. As such, the rotordynamic analysis of the system is compared to the desired operating envelope to determine if the system will behave acceptably.

As an example, the test rig model in Figure 5 was analyzed, and its unbalance response is shown in Figure 6. As seen in Figure 6, there are predicted resonances at around 5,000 rpm

and 12,000 rpm. Since the desired speed range is 10,000 rpm to 30,000 rpm, another iteration of bearing design and/or rotor layout could be performed to try to move the 12,000 rpm resonance out of the operating range. In this case, the test rig is already built, but this illustrates how the iterative nature of the four-step process can be beneficial. In this case, it could have been used to improve the frequency response of the test rig.

In a similar manner, the rotordynamic model can be used to predict stability, and the design can evolve on paper until both the unbalance response and the stability are acceptable.

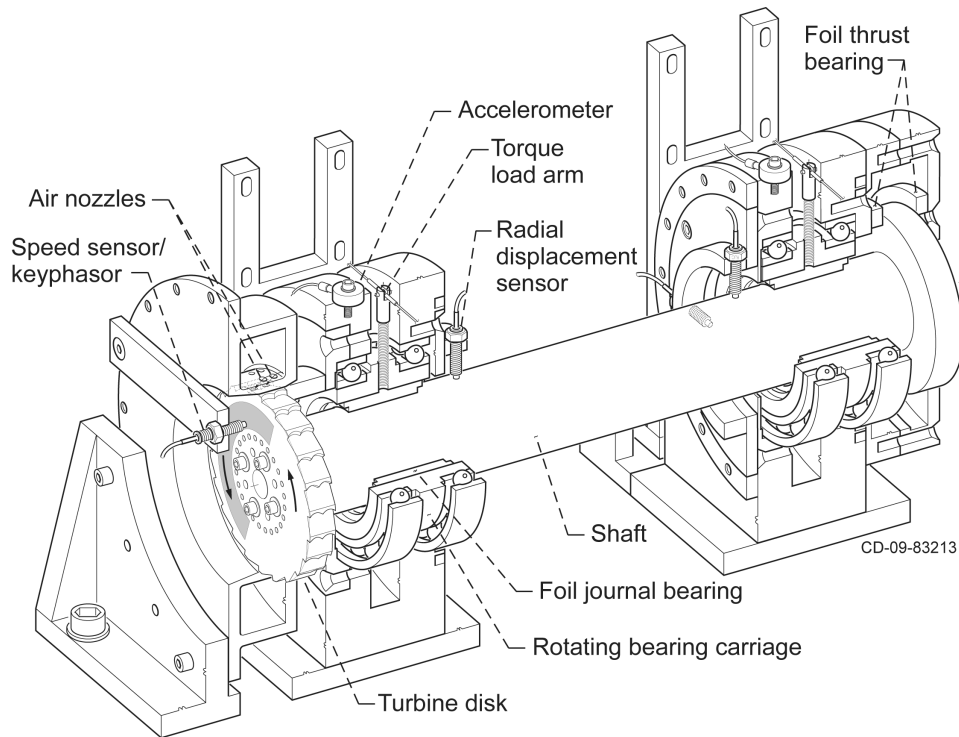


Figure 4.—Sketch of the test rig used for example calculations.

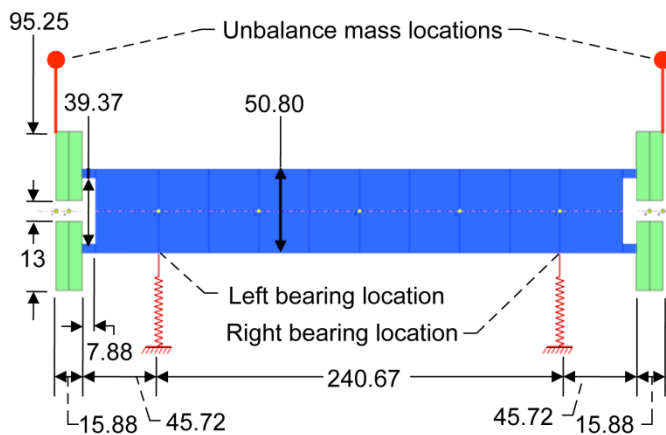


Figure 5.—Example rotordynamic model used to assess system feasibility in step 1 of the four-step integration method (dimensions in mm).

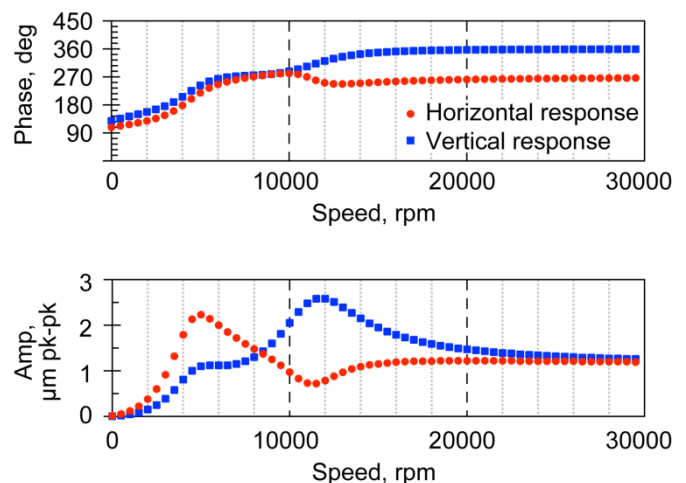


Figure 6.—Unbalance response prediction at a bearing location.

In step two of the integration methodology, the bearing design chosen in step one is tested to verify the desired performance is achieved. As seen in the flowchart, the first action of step two is to fabricate or procure the bearings designed in step one. Next, the bearings are tested for acceptable performance. The testing performed in step two can depend upon the application, but may include measurements of load capacity, power loss, dynamic coefficients, thermal behavior, wear, etc. As an example, a power loss plot for the bearings used in the test rig is shown in Figure 7. The test was done for a fixed load, which means the only variable in the modified Sommerfeld number is speed. Thus, plotting with speed on the abscissa is qualitatively the same, and it makes it easier to see that for speeds between 10,000 and 30,000 rpm, the test rig will indeed be operating to the right of the power loss minimum, i.e., the lightly loaded regime.

Any number of other component level tests can be done to determine that the bearings will behave as designed. If they do not, sometimes modifications are identified during testing, and bearings can be altered and retested. Otherwise, the designer can back up and re-iterate through step one to change the bearing or rotor design to correct for shortcomings uncovered in step two.

In step three, rotordynamic simulation testing serves to verify that a rotor/bearing system with the same bearings and mass properties as the desired turbomachine will behave as designed. A rotor with matching mass, geometry, and gyroscopic properties is built and tested using the bearing design chosen in step one and verified in step two. Here, the predicted behavior from the feasibility study can be verified, and the rotor system can be tested for overall desirable dynamic response. Other integration issues such as build techniques and procedures can be tested out as well as the fitment of various components. Lingering questions about any design or build-up issues can be answered in step three, such as shaft interference fits, bearing mounting (alignment), material pairs, etc. Once the design successfully completes step three there should be little doubt that the full-scale demo will be successful. Again, like the other steps, step three affords an opportunity to identify and make changes to the bearings or rotor and re-test, or major design flaws can be worked out by stepping back into the previous steps and re-iterating.

Again referring back to the test rig, an example of the types of tests one can perform in step three is a comparison of the unbalance response to the predicted response in step one. Figure 8 shows the measured unbalance response curves from the same bearing location on the test rig. In this example case, one can see that a reasonable prediction was made during step one. This also illustrates that foil bearing design tools are currently becoming quite capable for design of turbomachinery.

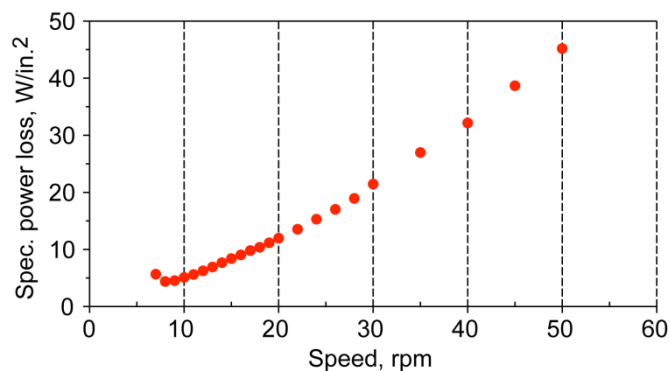


Figure 7.—Power loss versus speed for 50.8 mm diameter foil bearing at constant load.

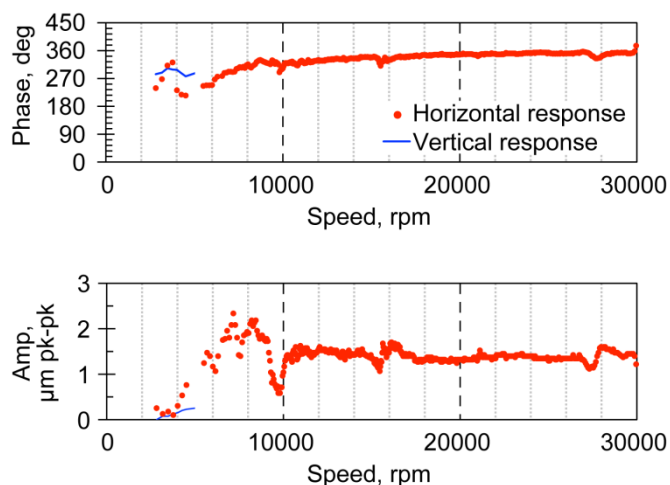


Figure 8.—Measured unbalance response plots from the test rig for comparison to Figure 6.

Step three is followed by fabrication of a full-scale demonstration unit. As with the previous steps, this can be an iterative process to fine tune the design, or revert back to a previous step if major issues surface. Once successful, the demonstration unit can lead into a production turbomachine.

Summary Remarks

As Oil-Free turbomachinery applications gain momentum in the marketplace, more new machines will be developed to take advantage of the unique capabilities of gas foil bearings. A four-step integration methodology is outlined in this paper that, if followed during development, will increase the probability of successful design of new turbomachines. A structured approach to development such as this is especially important for applications that are a significant departure from machines that currently exist. Many times, development efforts will skip one or more of the steps outlined here and

rush to prototype demonstration only to fail prematurely. Following all the steps provides opportunities to gauge the performance of the design and make changes along the way to maximize the chance of success without expensive hardware failures.

Several examples are given to illustrate the types of testing and analysis that can be done to feed into the design and to measure the chances of successful development. While the examples and descriptions given are based on gas foil bearing development, the process can be applied to any development program to enhance the potential for success.

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14. ABSTRACT Commercial applications for Oil-Free turbomachinery are slowly becoming a reality. Micro-turbine generators, high-speed electric motors, and electrically driven centrifugal blowers are a few examples of products available in today's commercial marketplace. Gas foil bearing technology makes most of these applications possible. A significant volume of component level research has led to recent acceptance of gas foil bearings in several specialized applications, including those mentioned above. Component tests identifying such characteristics as load carrying capacity, power loss, thermal behavior, rotordynamic coefficients, etc. all help the engineer design foil bearing machines, but the development process can be just as important. As the technology gains momentum and acceptance in a wider array of machinery, the complexity and variety of applications will grow beyond the current class of machines. Following a robust integration methodology will help improve the probability of successful development of future Oil-Free turbomachinery. This paper describes a previously successful four-step integration methodology used in the development of several Oil-Free turbomachines. Proper application of the methods put forward here enable successful design of Oil-Free turbomachinery. In addition when significant design changes or unique machinery are developed, this four-step process must be considered.					
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