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CRYOGENIC FLUID FILM BEARING

TESTER DEVELOPMENT STUDY

by

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ROTORDYNAMICS-SEAL RESEARCH

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1.0 INTRODUCTION

Rolling element bearings have a DN (Diameter in mm X Speed in rpm) limit of approximately 2-2.5 X 10^6 and require lubrication to achieve a reasonable operational life. Fluid film bearings do not have a DN limit and offer life and speed characteristics which are unachievable with rolling element bearing technology. In order to unlock the potential of fluid film bearings for cryogenic turbomachinery applications, research is required to define their characteristics in the cryogenic environment. This research should include materials evaluation, steady-state performance, and rotordynamic coefficients.

The determination of rotordynamic coefficients is important for the development of new bearing technologies, validation of analytical models, and the application of bearing technology to high speed rotating machinery. While fluid film bearings hold great potential for long life, low cost, and reliable operation, there is a general dearth of experimental data for their rotordynamic performance. This is especially true of foil bearings and other bearings operating in cryogens.

This report documents the results of a study to define the conceptual design of a test apparatus and associated facility for the identification of rotordynamic coefficients of fluid film bearings in a cryogenic environment.

1.1 ROTORDYNAMIC COEFFICIENTS

Historically, for rotordynamics, bearing testing has had the objective of determining the coefficients in the following bearing force-displacement model:

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} + \begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix} \begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} \quad (1)$$

where $(X, Y, \dot{X}, \dot{Y}, \ddot{X}, \ddot{Y})$ are the rotor displacements, velocities and accelerations. (F_x, F_y) are the fluid film forces acting on the rotor, and (K,C,M) are the stiffness, damping, and inertia coefficients of the fluid film, respectively.

For bearings with a large length to diameter ratio a more extensive force-displacement model may be required which takes into consideration angular rotation of the shaft. This force/moment-displacement/rotation model is expressed in the following form:

$$-\begin{Bmatrix}F_x\\F_y\\M_x\\M_y\end{Bmatrix}=\begin{bmatrix}K_{xx}&K_{xy}&K_{x\alpha}&K_{x\beta}\\K_{yx}&K_{yy}&K_{y\alpha}&K_{y\beta}\\K_{\alpha x}&K_{\alpha y}&K_{\alpha\alpha}&K_{\alpha\beta}\\K_{\beta\alpha}&K_{\beta y}&K_{\beta\alpha}&K_{\beta\beta}\end{bmatrix}\begin{Bmatrix}X\\Y\\\alpha\\\beta\end{Bmatrix}+\begin{bmatrix}C_{xx}&C_{xy}&C_{x\alpha}&C_{x\beta}\\C_{yx}&C_{yy}&C_{y\alpha}&C_{y\beta}\\C_{\alpha x}&C_{\alpha y}&C_{\alpha\alpha}&C_{\alpha\beta}\\C_{\beta x}&C_{\beta y}&C_{\beta\alpha}&C_{\beta\beta}\end{bmatrix}\begin{Bmatrix}\dot{X}\\\dot{Y}\\\dot{\alpha}\\\dot{\beta}\end{Bmatrix}+\begin{bmatrix}M_{xx}&M_{xy}&M_{x\alpha}&M_{x\beta}\\M_{yx}&M_{yy}&M_{y\alpha}&M_{y\beta}\\M_{\alpha x}&M_{\alpha y}&M_{\alpha\alpha}&M_{\alpha\beta}\\M_{\beta x}&M_{\beta y}&M_{\beta\alpha}&M_{\beta\beta}\end{bmatrix}\begin{Bmatrix}\ddot{X}\\\ddot{Y}\\\ddot{\alpha}\\\ddot{\beta}\end{Bmatrix} \qquad (2)$$

1.2 TRANSFER FUNCTION MEASUREMENT

The aforementioned force-displacement models of Equations 1 and 2 are linear models. The term linear refers to the inherent assumption that the force coefficients K,C, and M are not frequency dependent. It is not good practice to assume the characteristics of data ahead of time. In some cases, such as gas hydrostatic bearings and magnetic bearings, analysis will indicate ahead of time that the linear model may not apply. Therefore, the actual measurement objective for dynamic testing of bearings (and also seals) is the transfer function. Equation 1 can be rewritten in terms of the complex transfer function as follows:

$$-\begin{Bmatrix}F_x\\F_y\end{Bmatrix}=\begin{bmatrix}H_{xx}&H_{xy}\\H_{yx}&H_{yy}\end{bmatrix}\begin{Bmatrix}X\\Y\end{Bmatrix} \qquad (3)$$

where (for the linear model of Equation 1)

$$H_{ij}=K_{ij}-\omega^2 M_{ij}+j\omega C_{ij}$$

Once obtained from test data, the transfer functions will determine whether or not the linear model applies. If the linear model does apply, the K,C,M coefficients can be obtained using linear and polynomial curve fits of the transfer function data. If the linear model does not apply, the transfer function information is used in the rotordynamic analysis in place of stiffness, damping, and mass coefficients. An example set of transfer functions for fluid film bearing testing is shown in Figure 1-1. A similar expression can be developed for the more complicated model of Equation 2.

1.3 TEST APPARATUS FLEXIBILITY

When developing a test apparatus for fundamental research it is important to make the apparatus as flexible as possible. The test results, and not test rig design, must determine what characteristics are important. The evolution of bearing and seal testing for rotordynamic coefficients has demonstrated that limitations in test rig design because of preconceived notions about the relative importance of certain characteristics has rendered useless (or suspect) much of the available data.

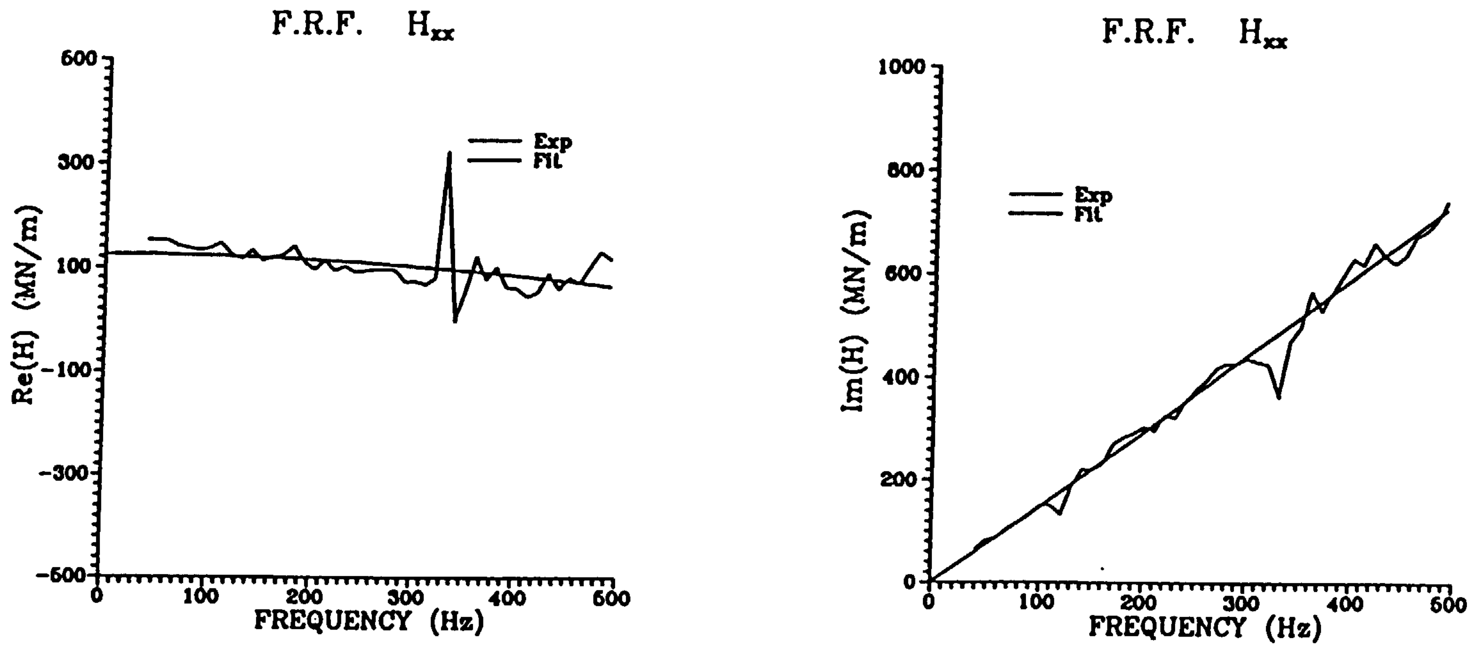

Figure 1-1. Experimental fluid film bearing transfer function from Rouvas (1992)

This point can be illustrated by examining the development of annular seal test rigs. NASA MSFC funded the development of a test apparatus at Texas A&M University for the measurement of rotordynamic coefficients of annular seals in Halon. This test program lasted approximately 6 years and was thought to be successful at the time. However, the test apparatus design was too limited in scope because the conventional knowledge at the time was that hydrodynamic bearing theory and test approaches would suffice. Therefore, no accommodations were made for inlet circumferential velocity control or separation of the rotordynamic coefficients. Unfortunately, both have been shown to be necessary for understanding the rotordynamic coefficients of annular seals. Although the data is useful for a qualitative assessment of the relative performance of different seal geometries, it is not useful for validation of computer codes or design of new components.

Now, we are faced with a similar situation with respect to moment coefficients and tilt. Fortunately, several other researchers have investigated these effects and found that in some cases the influence is substantial. However, it is not possible to extend the findings for a limited sample of test articles to a broad class of yet to be investigated bearing technologies. Therefore, it is imperative to develop a test apparatus with moment coefficients and tilt in mind. The apparatus must either measure all parameters necessary to determine the characteristics or be designed such that all parameters related to tilt and moment coefficients are controlled to the point of being insignificant. With this in mind, the focus of this study will be on test apparatuses which have the test article located between the slave bearings. As the discussion will point out, overhung test apparatuses are subject to excessive tilt and moment effects which can not be eliminated through design.

2.0 SYSTEM CONSIDERATIONS

The test measurement system includes many components both within and external to the apparatus. Due to the difficulty of the task at hand, all components must be designed and/or selected for optimum performance to achieve acceptable experimental uncertainties. It is fair to say that the measurement system is like a chain, and is only as strong as its weakest link. This section briefly discusses the links that form the chain, and some of the primary factors that go into each.

2.1 TESTER CONCEPTUAL DESIGN

Selection of a basic tester concept is probably the most important decision in the conceptual design phase of an apparatus for rotordynamic coefficient testing. To a very large extent this determines, among other things, tester functionality, cost, suitability, ease of use, and overall uncertainty in the experimental result. A variety of alternative tester concepts can be found in the open literature as both basic and applied research of this type has taken place in the U.S. and around the world (see APPENDIX 1). A thorough discussion of each application is beyond the scope of this effort. There are different ways to classify testers. Some of these are:

- test article location
    - overhung
    - centrally mounted
- dynamic excitation source
    - shaker, independently controlled
        - electrodynamic
        - hydraulic
        - magnetic
        - mechanical (i.e., ball vibrolator)
    - synchronous unbalance or eccentricity
    - asynchronous unbalance or eccentricity
    - impulse hammer
    - step input (eg., strap)
- point of application of dynamic excitation
    - test article housing
    - directly on shaft
    - indirectly on shaft through a bearing
        - support bearing
        - auxiliary (non-support) bearing
- load measurement scheme
    - reaction force within test article stator
    - dynamic model matching from response only
        - simple - rigid test article housing

        - complex - flexible test shaft
    - force input measurement (usually involves some model matching, minimal to extensive)
    - film pressure integration
- test article support
    - fixed housing
    - floating housing
- type of test article
    - rolling element bearing
    - fluid film bearing
    - magnetic bearing
    - seal
- test conditions
    - low/high test shaft rotational speed
    - low/high test frequency range
    - cryogenic/ambient/high temperatures or multiple
    - low/high feed flow rates
    - low/high pressures (inlets, deltas, films, etc.)
    - presence of thrust reaction
    - linear/nonlinear conditions
        - deadband clearances
        - large displacements
        - cavitation
        - structural
    - transient/steady state
    - condition matching (do speeds, loads, pressures, temperatures, etc., match or scale with the target application)

This is just a sampling of some of the factors that differentiate the many approaches to rotordynamic coefficient testing. There are numerous interrelated pros and cons for each. Another basic consideration is the overall goal of the test program. The primary goal may be tester accuracy. Or perhaps versatility is paramount such that accuracy is compromised so that test articles can be more easily interchanged. Cost is always a factor, and whether it is realized or not, budget constraints can and do affect the accuracy of the experimental result. More often than not the number one trade off is between accuracy and everything else. Only through a thorough understanding of all facets of rotordynamics and test methods can meaningful accuracy be attained while adhering to the constraints of the test program.

2.2 TRANSDUCER SELECTION, INSTALLATION AND CALIBRATION

Ideally transducer selection should be an integral part of tester conceptual design. For example, standard series Bently Nevada proximeter probes are often used to measure shaft motion. For many tester applications that motion is on the order of several hundred microinches, or less. Due consideration need be given to the fact that such probes are generally considered to have a resolution of roughly 50 microinches. Thus, 200 microinches can only be measured accurately with a 50 microinch device if it is calibrated and applied correctly. Experience has shown that the use of higher gain proximeters can, and does, greatly enhance resolution. But the corresponding reduction in useful transducer measurement range can make it very difficult, if not impossible, to obtain useful readings. Installation of displacement probes should always be done a quad fashion (4 per plane). For static measurements this allows the cancellation of thermal drift, and centrifugal/thermal/pressure growth. For dynamic testing it reduces random measurement errors. Proximeter probe calibration should always be performed at the temperature and pressure to which their subjected. Also, calibrations should be dynamic, not just static, since gain and phase lag in eddy current probes have been measured by the authors to be frequency dependent to a considerable extent.

Accelerometers also are usually a critical part of the test measurement system. Their selection is usually a trade off on size, sensitivity, and range. Proper attention to their installation is also important, especially in high flow fluid bearing testers. Since flow turbulence beyond the test frequency measurement range can cause full or partial saturation of the accelerometer electronics. Like displacement proximeters, accelerometers should be mounted in redundant arrays to reduce random measurement errors. Accelerometer calibrations are generally performed only for amplitude sensitivity. For rotordynamic coefficient measurements, it is imperative that they also be phase calibrated.

For many tester concepts dynamic loads are measured with load cells. Piezoelectric load cells potentially offer greatly enhanced accuracy and range over other types of sensors. Like proximeter probes, load cells generally are calibrated only for load sensitivity. For this application calibrations need to be for sensitivity and phase, both as functions of frequency. Their installation requires tight machining tolerances on the mating parts which they contact. Also, their installation needs to be assessed from the standpoint of incremental loadings brought on by thermal and pressure effects at test conditions. For cryogenic applications, the installation and location of piezoelectric load cells must be done carefully. Dynamic testing of piezoelectric load cells by the authors in both liquid helium and liquid nitrogen showed that the crystal would delaminate or crack under low cycle load conditions. Piezoelectric load cells must be preloaded accurately and maintain a known preload during operation. Therefore, piezoelectric load cells must be insulated from the cryogenic test fluid to avoid changes in preload during operation and possible failure.

2.3 TEST PROCEDURE

The test procedure is generally a direct consequence of the tester concept being used, and the mode of operation of the facility. The effect on experimental uncertainty comes about from the direct influence on what kind of data, and how much, is acquired for a given test article operating condition. Tester design largely determines what kind of data can be acquired, and this is a major factor in establishing potential bias errors which cannot be "corrected" or "calibrated" out.

The "quantity" of data that can be obtained for any single test condition directly influences random errors in the experimental result. Test time made available at any given test condition is strongly influenced by the type of facility (continuous closed loop vs blow down). Cost is also an obvious factor. Repeating a test, or simply running it for a longer length of time, increases confidence and lowers random errors (practically the same thing).

2.4 CABLING AND SIGNAL CONDITIONING

Transducer cabling and signal conditioning equipment are often not considered for their effect on experimental uncertainty. Normally, cabling is not a factor as long the usual precautions are taken to prevent tribological effects in the case of capacitive transducers, and good connections are always verified. Signal conditioning equipment like amplifiers and filters should be treated like any high frequency transducer. They should be calibrated for their amplitude and phase characteristics as functions of frequency.

2.5 DATA ACQUISITION

In modern day test labs, data is acquired with computer systems using high speed analog to digital converters. This avoids the use of FM tape recorders which potentially can introduce large errors in rotordynamic coefficients. The data acquisition system is one of the most important "links" in the overall chain of the test system. It must be accurate, it must have adequate precision, and these must be verified by thorough calibration. A computerized data acquisition system is mostly just a fast digital volt meter with multi-channel and self storage capability. But in addition to that it has integral parts such as a high speed clock and bank of analog filters. Calibration of the acquisition system should verify the high frequency characteristics of both gain and phase for the complete system.

Quite often, the calibration of the whole multi-channel instrumentation system, except the transducers themselves, can be done at one time. This is accomplished by passing an appropriate broadband calibration signal simultaneously through all cabling and signal conditioning equipment, and recording the signal with all channels of the acquisition system. This "measurement" provides the means to calibrate the complete system for amplitude and phase across the entire test frequency range.

2.6 DATA PROCESSING AND REDUCTION

In rotordynamic coefficient testing the data processing phase is the final step to obtaining the desired experimental result. That is to essentially calculate the rotordynamic coefficients from a multitude of data channels. In addition to simply producing measured estimates of the rotordynamic coefficients, the calculation method should suppress the effects of noise and minimize uncertainties. It is often overlooked that the calculation method itself has the potential to either magnify or reduce experimental uncertainties, and this applies to both bias and random errors in the rotordynamic coefficients.

To make the most of this opportunity to reduce errors requires a thorough knowledge of digital signal analysis techniques and methods. For example, there are multiple ways to compute the stiffness from a pair of load and deflection measurements. Under ideal circumstances it rarely matters which method is used. When test conditions are less than ideal, such as in the presence of significant instrumentation noise or other types of noise, there are methods which are specifically optimized to reduce, of not eliminate, the effect of various types of noise.

3.0 CANDIDATE DESIGNS

Numerous tester designs for measuring rotordynamic coefficients have been published in the open literature. Nearly all such testers fall into one of two broad categories: overhung or centrally mounted. This refers to the location of the test article (seal, bearing or impeller) with respect to the support bearings (also called slave bearings). A vast majority of the testers in existence are of the centrally mounted variety. This is a natural consequence of the pros and cons that apply to each concept. Without attempting a complete treatise on the subject, the following table suggests some of the more significant pros and cons:

	Tester Concept Type	
	Centrally Mounted	Overhung
PROS	• immune to shaft tilt • slave bearings loaded less than test article	• Easy to change test articles
CONS	• interchanging test articles can be time consuming	• Subject to errors associated with shaft tilt • Slave bearing(s) loaded more than test article

3.1 CONSIDERATIONS ABOUT OVERHUNG TESTERS

An example overhung test apparatus is shown in Figure 3-1. One of the points brought out in the table is that tilt of the test shaft has a detrimental affect. For test articles having an axial length as much or more than the shaft diameter, moments produced by tilt of the shaft can be significant. If measurement of moment coefficients is beyond the scope of the test program, it is important to minimize their potential effect. The only sure way to accomplish this is to preclude the presence of tilt of the test shaft at the test article location. This is best done by positioning the test article midway between two slave bearings. Overhung test articles, on the other hand, are particularly prone to the effect of shaft tilt. This effect, should it occur, can have serious consequences when trying anchor analytical models to test data unknowingly influenced by the presence of fluid film moments produced by tilt.

Another point brought out in the table is the loading to which the slave bearings will be subjected. When testing bearing concepts for advanced turbopump designs, the high operating DN's and large working loads can be beyond the capability of mechanical bearings. This usually suggests the use of fluid film bearings. Testing such a bearing will require rather large flow rates of working fluid at extremely high pressure. This creates a problem for slave bearing design because there may not be enough high pressure flow capacity to supply two fluid film slave bearings <u>and</u> a test bearing. So the natural slave bearing choice would seem to be ball bearings, except that the speed and load conditions are so severe. One thing that can be done is to minimize the loads to which the slave bearings are subjected. The centrally mounted tester concept accomplishes this by sharing the test bearing load bewteen two slave

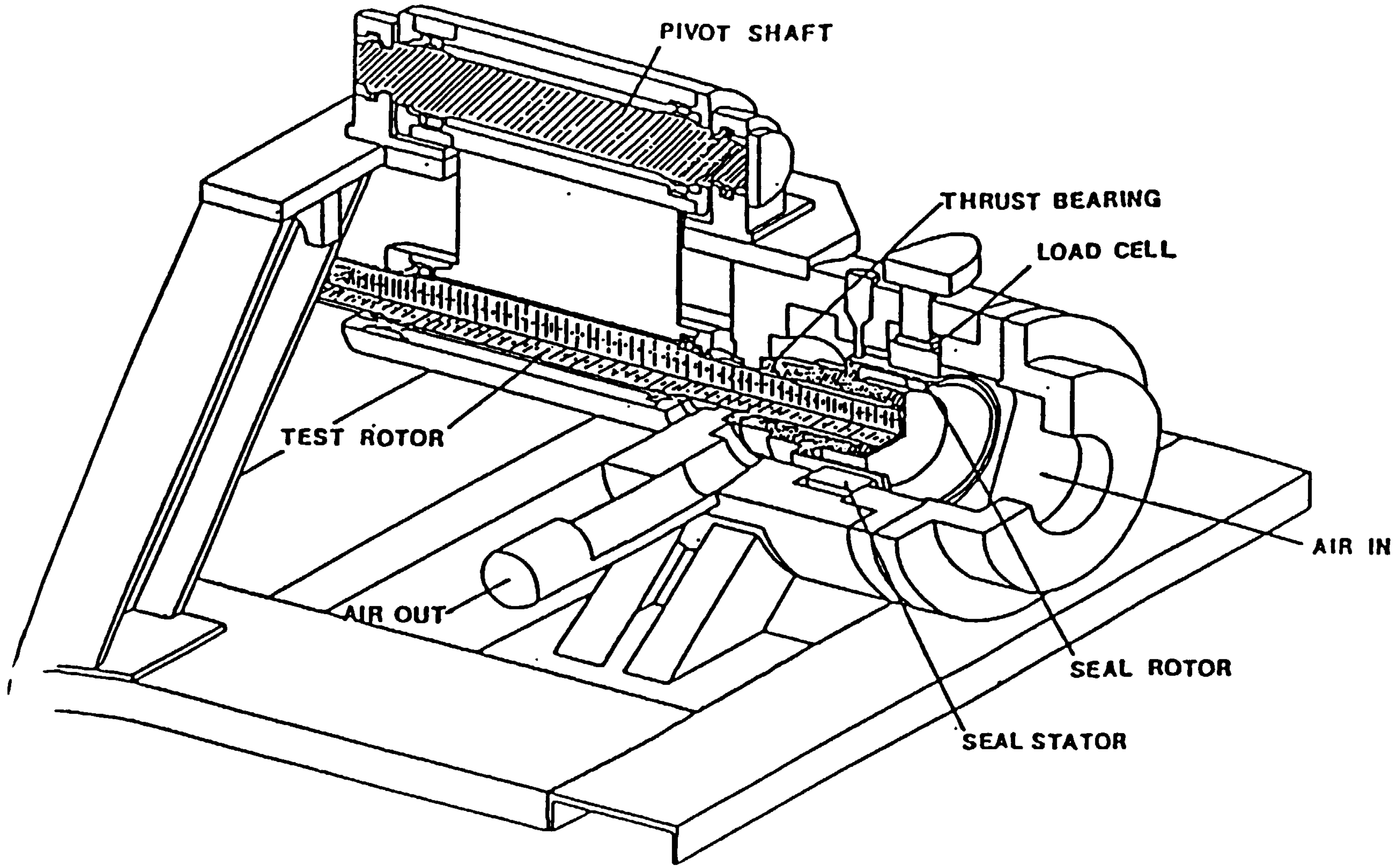

Figure 3-1. Example overhung test rig

bearings. In an overhung tester the near slave bearing actually experiences more load than the test bearing.

The last point brought in the table is the aspect of test article interchangeability. The primary advantage of the overhung concept is the relative ease with which the test article can be removed and replaced. For the centrally mounted concept utilizing rolling element bearings, removing the test article either requires horizontally split bearing housings, or removal of one slave bearing pedestal. If magnetic or fluid film bearings are used with the centrally mounted concept, the shaft can be easily removed for access to the test article.

3.2 SLAVE BEARING CONSIDERATIONS

The type and characteristics of slave bearings has a profound influence on the development of not only the test apparatus, but the entire facility. The overall accuracy and quality of the test results obtained from the test apparatus are heavily dependent on the dynamic characteristics of the rotor-bearing-test article system. If at all possible, the slave bearing rotordynamic characteristics should remain constant for all test conditions. For cryogenic testing, this is a very difficult requirement to meet.

Fluid film bearing rotordynamic characteristics are very clearance and fluid viscosity dependent. It is nearly impossible to design a fluid film bearing which will have the same or similar rotordynamic characteristics for operation at room temperature, LOX temperature, and LH2 temperature. If fluid film slave bearings are preferred, different geometries will have to be designed and fabricated for each temperature and fluid condition. Otherwise, the different dynamic and load characteristics will have to be accounted for by changing the test method and data reduction procedure for each test condition. The facility requirements for fluid film slave bearings are as extensive as for the test article and will severely limit test time in a blow down facility.

Rolling element bearing rotordynamic characteristics are less sensitive to test fluid properties than fluid film bearings. However, they are sensitive to changes in bearing deadband. It may be possible to arrive at 2 rolling element bearing designs which will suffice; one for cryogenic and one for room temperature. Considering the range of test fluids and temperatures that this test apparatus must accommodate, the rolling element bearing is preferred over the fluid film bearing. Also, the facility requirements for rolling element bearings are much less than for fluid film bearings.

Magnetic bearings offer the greatest flexibility for this type of test apparatus. They are relatively insensitive to speed and temperature changes and can provide the same dynamic characteristics for all test conditions. In fact, the magnetic bearing can function as the exciter, eliminating the need for shakers. Magnetic bearings also provide the simplest means for measuring the effects of tilt and moment coefficients. The facility requirements for magnetic bearings are minimal.

3.3 CANDIDATE DESIGNS FOR EVALUATION

The previous discussion has shown that apparatuses with overhung test articles have many deficiencies which limit their usefulness and quality of data. By comparison, apparatuses with centrally mounted test articles are more likely to produce accurate data. Furthermore, fluid film slave bearings may be too sensitive to thermal conditions and fluid properties to function reliably over the entire range of expected tester conditions. Based on these arguments, further discussion and analysis will be limited to apparatuses with centrally mounted test articles utilizing rolling element or magnetic slave bearings. The four candidate configurations are:

1 Floating Housing Rig With Magnetic Bearing Exciter/Support
2 Fixed Housing Rig With Magnetic Bearing Exciter/Support
3 Floating Housing Rig With Shakers
4 Fixed Housing Rig With Shakers

3.3.1 <u>Floating Housing Rig With Magnetic Bearing Exciter/Support.</u> A concept sketch of this rig is shown in Figure 3-2. The stator or housing of the test article is supported only by the fluid film bearing to be tested (the test article). The stator is held parallel to the initial shaft centerline by tilt control wires that have a negligible lateral stiffness. Magnetic bearings are used as both slave bearings and an excitation source. The excitation force is input to the shaft at the slave bearings and the reaction force is measured by measuring the acceleration of the floating housing. This approach eliminates thermal difficulties associated with load cells in a cryogenic tester. A static load can be applied through a cable attached directly to the floating housing. The static load is supplied by a pneumatic cylinder through an isolation spring to the cable. The tilt control wires can also be used to add intentional tilt to the housing to determine the effect of static tilt on the transfer function to be measured.

While this tester concept may eliminate most or all of the thermal constraints of load measurement, it falls short in its ability to control the dynamic motion at the test article. For this concept to be successful, multiple arrays of accelerometers must be placed carefully on the test article and sophisticated calibration and data reduction techniques developed. This design will not be able to generate data for the determination of moment coefficients.

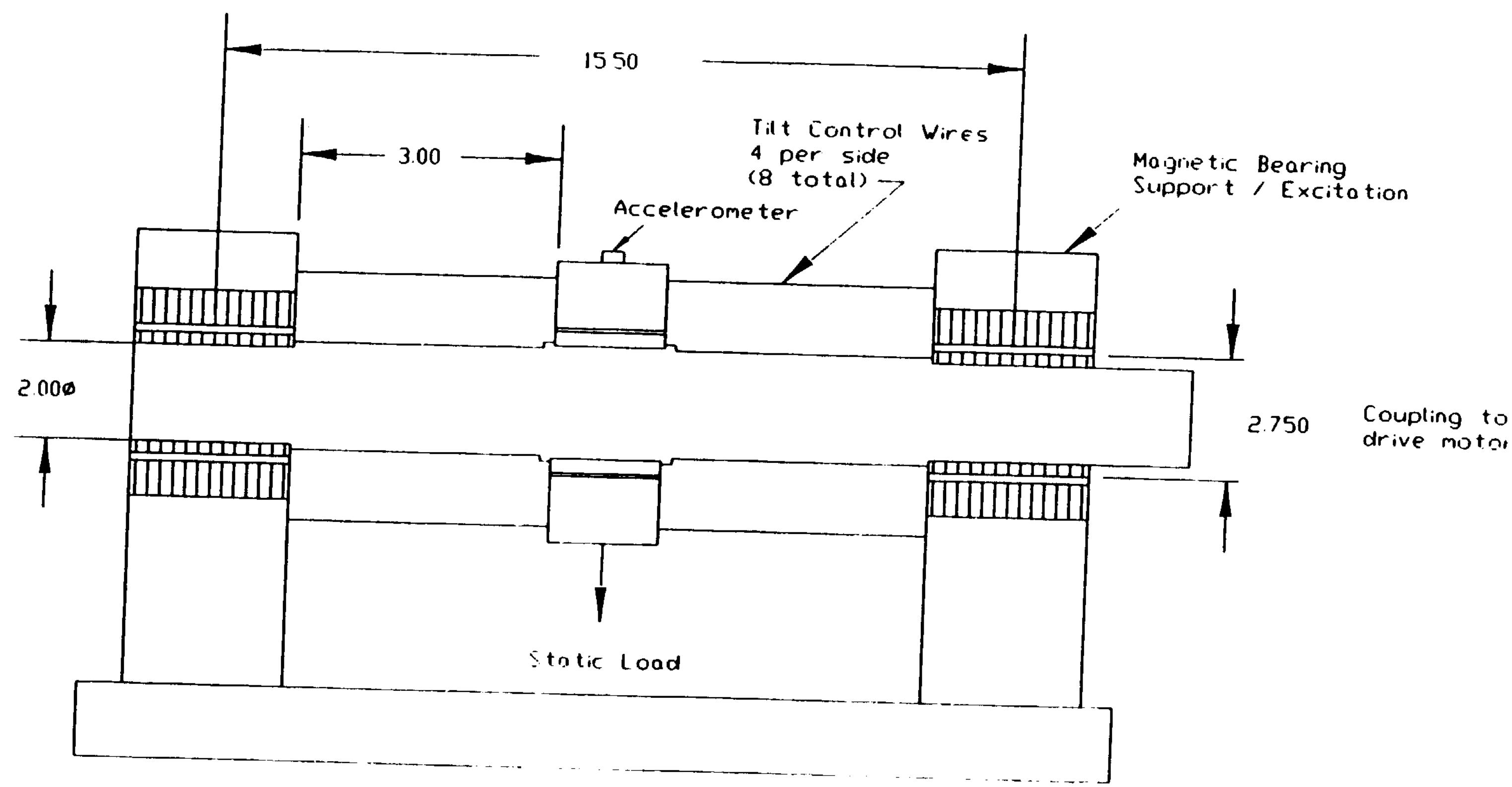

Figure 3-2. Floating housing rig with magnetic bearing exciter/support

3.3.2 Fixed Housing Rig With Magnetic Bearing Exciter/Support. This rig, shown in Figure 3-3, is similar to the floating housing rig (1) except that the test article stator is firmly supported by a pedestal. A thermally resistant material is used to isolate a load cell table in the pedestal used to measure test article load. Accelerometers are used to correct for stator inertia force. The excitation force is applied directly to the shaft at the slave bearings. A static load can be applied using the magnetic bearings or by moving the tester support. Using two independently adjustable, symmetrically mounted magnetic bearings makes it possible to prevent dynamic tilt (a common problem with fixed housing test rigs). This tester works best with a stiff shaft.

This concept attempts to address every major issues facing the development of a new cryogenic bearing test apparatus. The magnetic slave bearings provide temperature independent support and excitation, the load cell table is thermally isolated, the magnetic bearings are capable of exciting the shaft in every conceivable manner, and test article access is simplified by removing the shaft. The combination of the load cell table and magnetic bearing excitation at the slave bearings will enable the determination of all force and moment coefficients.

3.3.3 Floating Housing Rig With Shakers. This type of rig is the most common for measuring rotordynamic coefficients for fluid film bearings. A somewhat standard design has evolved over the past several years as researchers have perfected the test technique for journal bearings. A concept sketch of this type of tester is shown in Figure 3-4. The excitation force is applied to the floating stator through a load cell. Stator mounted accelerometers are used to correct for stator inertia. Although a calibrated hammer can be used as the excitation source, hydraulic shakers are more common due to the flexibility they provide in controlling the spectral content of the input signal. As with tester (1), the static load can be applied through a cable attached to the floating housing, and tilt control wires can be used to hold the housing parallel to the initial shaft centerline or provide intentional static tilt.

The major concerns about this type of tester design are tilt/moment control and isolation of the load cell in the cryogenic environment. The load cell is mounted on a stinger between the shaker head and the test article. For extreme environments, design difficulties may occur when trying to keep the load cell and shaker head at a safe temperature while limiting the dynamic and mass characteristics of the stinger and related insulation. This design will not be able to generate data for the determination of moment coefficients.

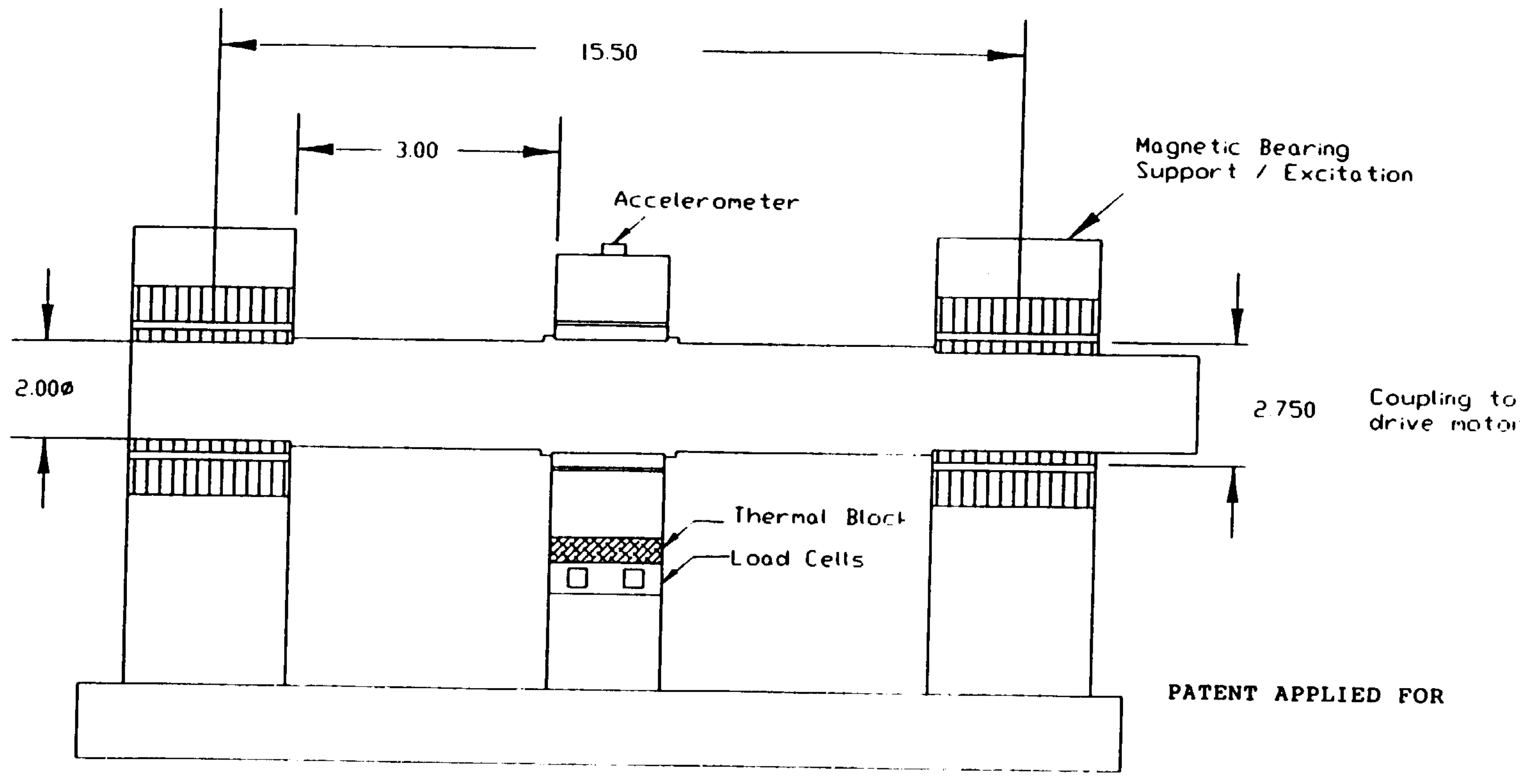

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Figure 3-3. Fixed housing rig with magnetic bearing exciter/support

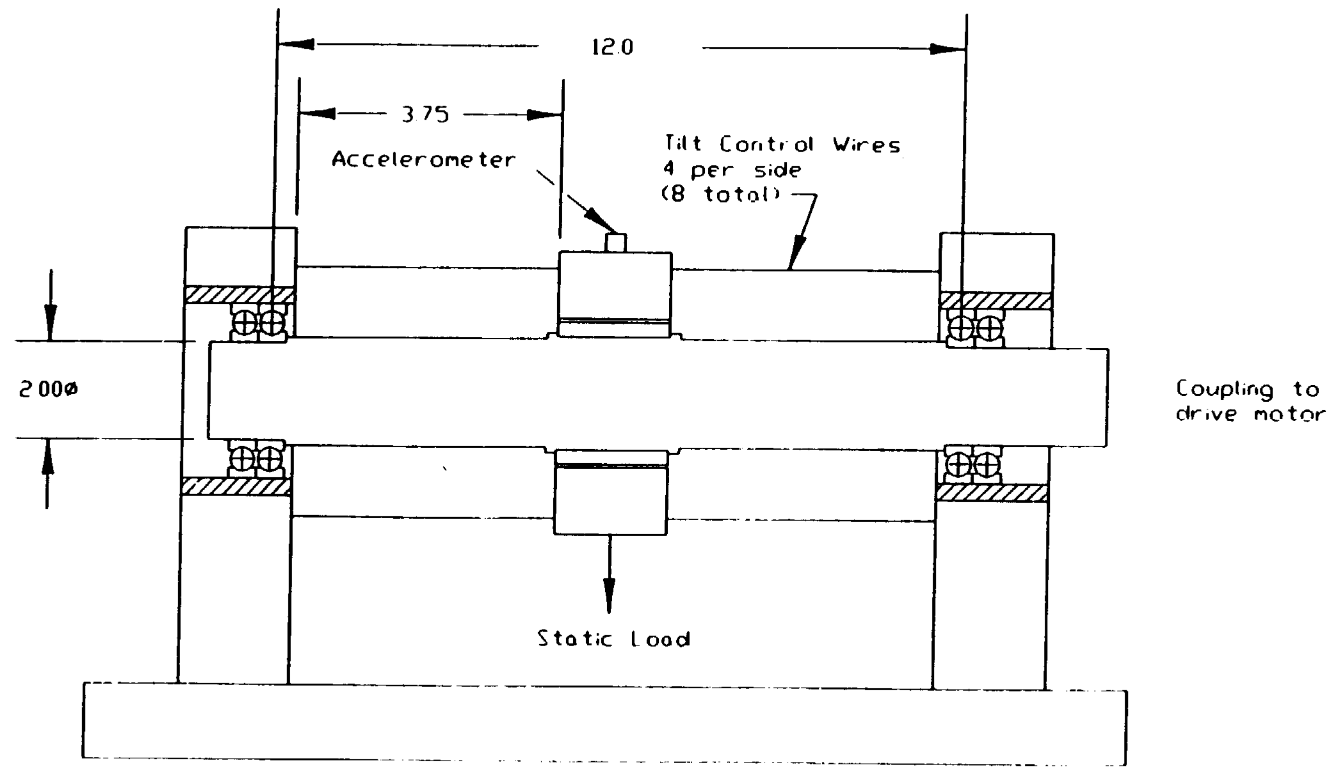

Figure 3-4. Floating housing rig with shakers

3.3.4 Fixed Housing Rig With Shakers. Figure 3-5 is a concept sketch of the floating housing rig of Figure 3-4 modified to utilize a fixed test article. In this type of tester, the excitation force is applied directly to the shaft through a shaft riding ball bearing. Load cells supporting the stator measure the reacted load. Accelerometers are used to correct for stator inertia force, but this correction is much less significant than with a floating housing tester. Again, either a hammer or shaker can be used as the excitation source. A static load can be applied directly to the shaft but is more conveniently applied by moving the tester support. A difficulty with this technique is that it is more difficult to adjust the load. It is also difficult to apply the dynamic load without causing tilt between the rotor and stator, unless two additional shakers are used. This type of tester requires a softer shaft than the other techniques.

This design allows the load cells to be thermally isolated, but it does not allow either tilt or moments to be controlled effectively. In addition, test article change out is more difficult with this arrangement.

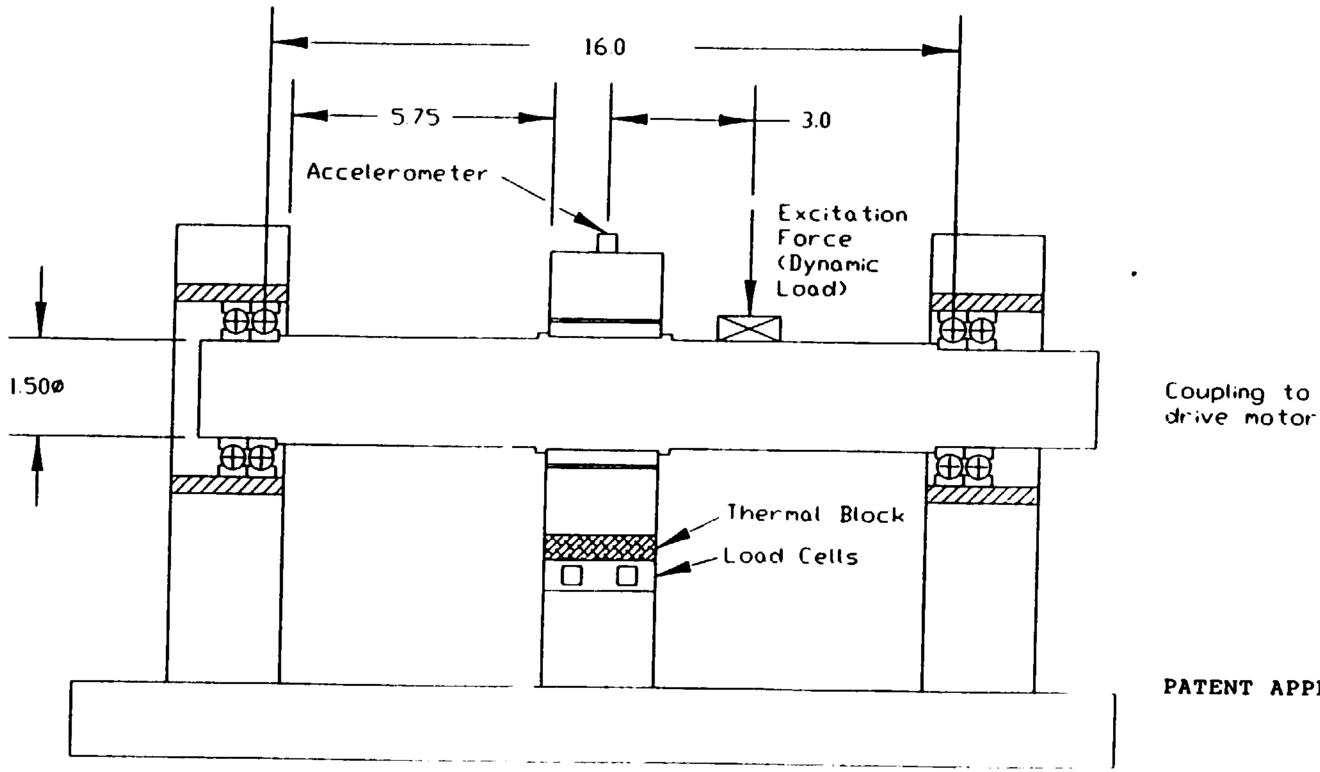

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Figure 3-5. Fixed housing rig with shakers

4.0 EVALUATION OF CANDIDATE DESIGNS

Frequency response and uncertainty analyses were performed for the two magnetic bearing supported testers to determine how they should be applied. The results of a previous analysis of the two shaker excited testers for a NASA funded SBIR (Imlach and Hawkins, 1992) were reviewed also. A finite element model was built for each tester concept. Using these models, linear frequency response analyses were performed using standard rotordynamics analysis techniques modified to account for certain special characteristics of magnetic bearings such as sensor noncollocation.

Tester designs utilizing magnetic bearings were analyzed specifically for this project. Designs utilizing rolling element bearings were originally analyzed by RSR for a NASA funded SBIR. Results from this previous analysis are review and discussed.

4.1 FREQUENCY RESPONSE ANALYSIS

The frequency response analysis was performed for a constant rotor speed of 9,600 rpm and a flat input force spectrum of 100 lb. In the analysis of the two magnetic bearing supported testers, 50 lb is applied at each magnetic bearing. In the floating housing tester of Figure 3-2, 100 lb is applied at the test article casing. In the fixed housing tester of Figure 3-3, 100 lb is applied directly to the shaft 3.0 inches from the test article center.

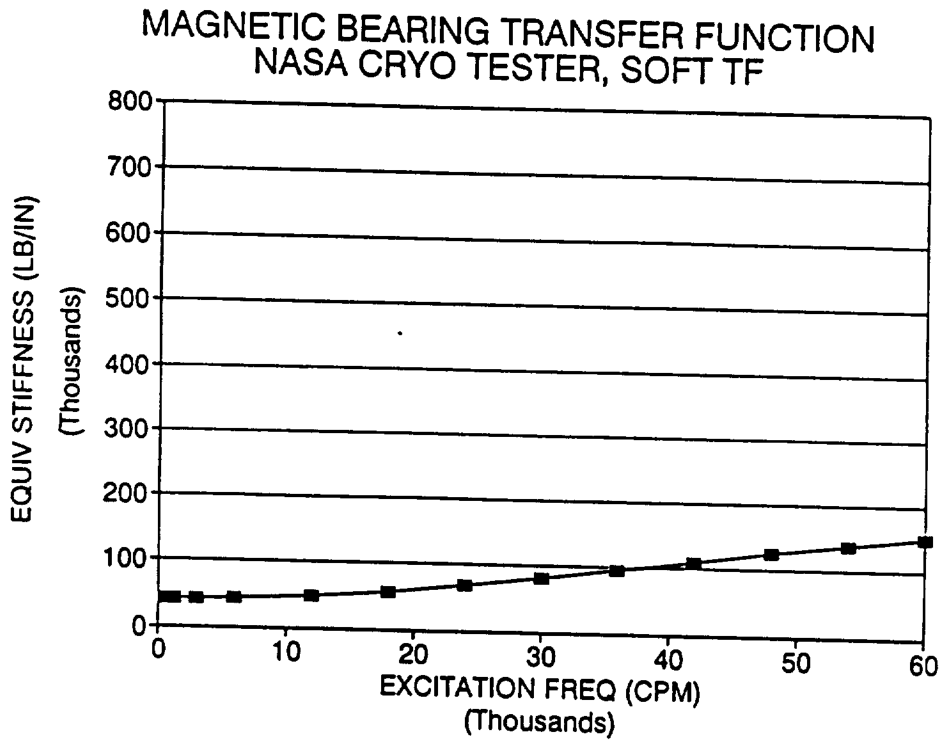
The frequency response analysis was performed for three reasons: 1) to determine the general frequency response characteristics of each test method, 2) to provide the necessary input for an uncertainty analysis, and 3) to determine the required input force levels and measurement resolution requirements. The results provide a basis for comparing and evaluating the relative merits of the test approaches.

4.1.1 Magnetic Bearing Supported/Excited Testers

The analyses for testers (1) and (2) were performed for the following conditions:

Test Article Stiffness:	200,000 lb/in	and	1,000,000 lb/in
Test Article Damping:	100 lb-s/in		
Test Bearing Journal Diameter:	2.0 in	and	6.0 in
Magnetic Bearing Stiffness:	low	and	high

The two magnetic bearing transfer functions are shown in Figures 4-1 and 4-2 in terms of equivalent stiffness and damping coefficients. The larger test article diameter causes two significant differences in the analysis presented here: 1) higher



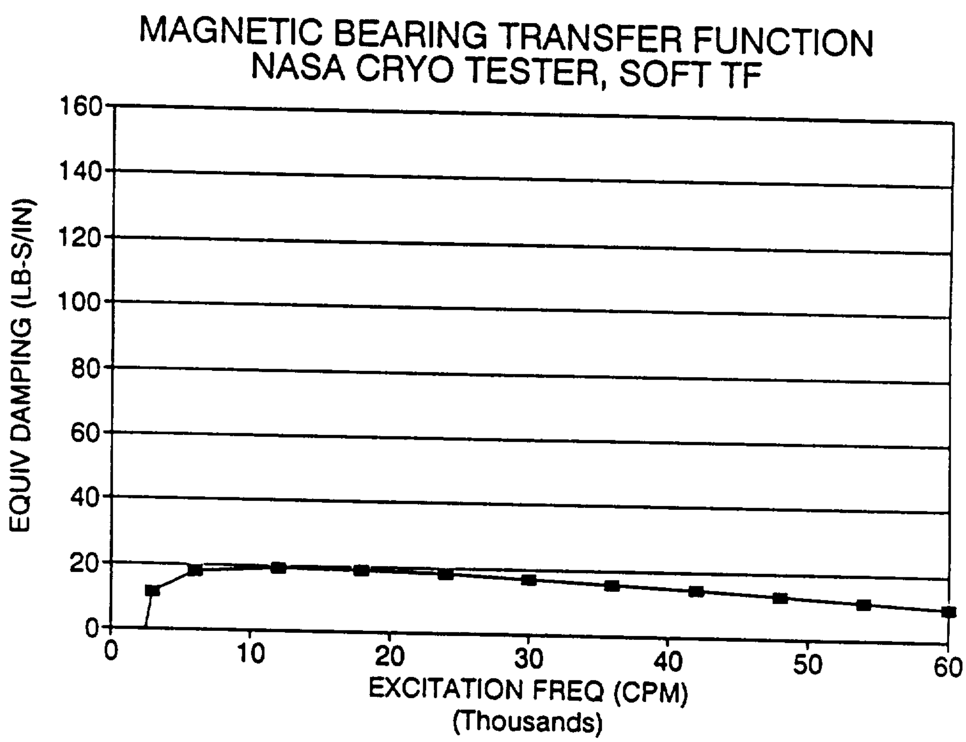
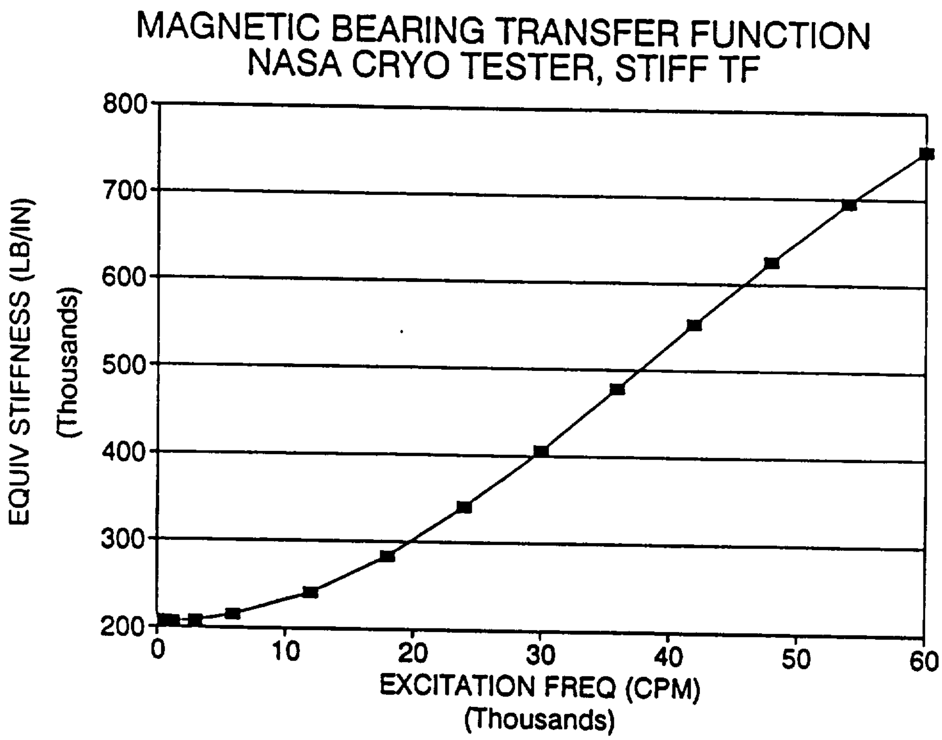


Figure 4-1. Stiff magnetic bearing transfer function used in analysis



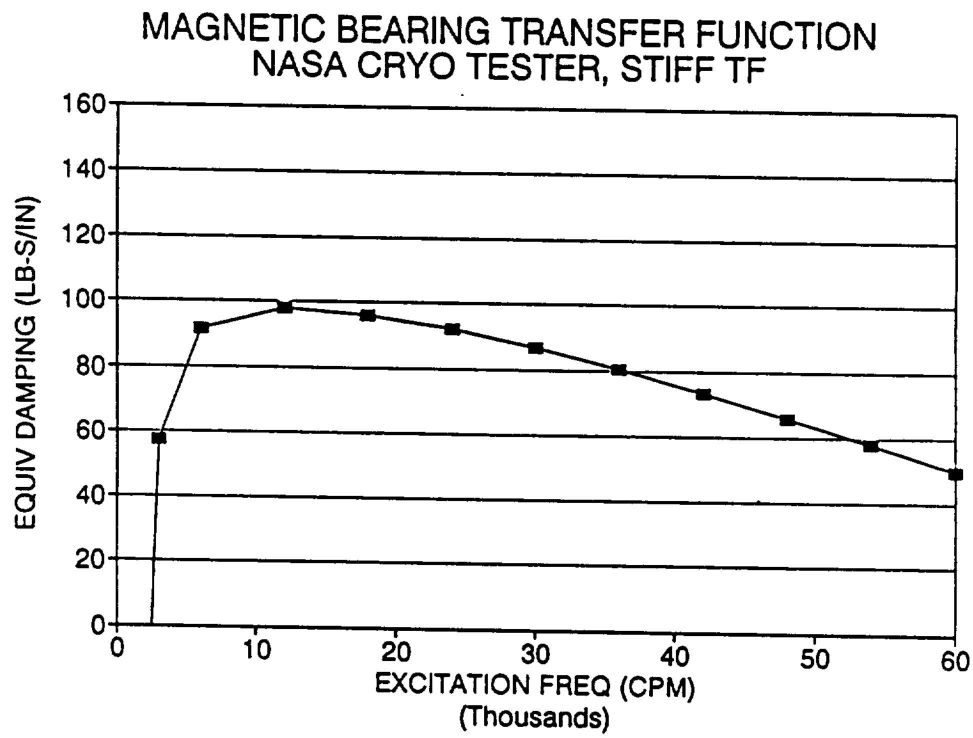


Figure 4-2. Soft magnetic bearing transfer function used in analysis

test article stator mass - 168 lb compared to 25 lb, an effect that tends to drive the rotor natural frequencies lower, and 2) a stiffer rotor which tends to increase the rotor natural frequencies.

4.1.1.1 Floating Housing (1). Frequency response analysis results with the floating housing are shown in Figures 4-3-4-12. Figure 4-3 shows the relative rotor/stator response at the shaft center for a 2 inch test article journal diameter, 200,000 lb/in test article stiffness and the stiff magnetic bearing transfer function. The two response peaks are due to the first and second mode of the tester. The response is near zero at low excitation frequency because the input is through the slave bearings. In order to get relative motion across the test article, the excitation frequency has to be high enough to excite one of the modes of the shaft/test article. The response at high frequency becomes small due to the wide separation in the second and third natural frequencies of the system. The response magnitude is above the minimum threshold of 0.00005 inches from about 100 Hz to 900 Hz. This range could be extended somewhat with an increase in input force. Figures 4-4 and 4-5 show test article acceleration and test bearing load as a function of excitation frequency for this case. The response levels are within acceptable limits from about 100 Hz to 900 Hz.

Figures 4-6-4-8 show relative rotor/stator displacement for three additional cases - Figure 4-6 is for a soft magnetic bearing transfer function, Figure 4-7 is a stiff magnetic bearing transfer function and 1,000,000 lb/in test article stiffness, and Figure 4-8 is for a soft magnetic bearing transfer function and 1,000,000 lb/in test article stiffness.

The characteristics illustrated in Figures 4-3-4-8 show the major problem with floating housing testers. The response at the test article is completely dependent on test article natural frequencies which may change dramatically depending on the type of bearing tested and the fluid medium used for the test. A flat response spectrum is desired in order to produce the lowest uncertainties and to reduce any amplitude nonlinearity effects. A much flatter response spectrum can be produced by spectrally shaping the input force spectrum. This can easily be done with either magnetic bearing or hydraulic shaker excitation. However, in either case, the response at null frequencies, such as the low frequency region of Figure 4-3, can not be increased significantly with a practical force input magnitude. Large changes in system dynamic characteristics may require a flexible and extensive data reduction software package which may need to be customized for each test series. Situations may also occur where the response at the test article is below measurement capability over the frequency range of interest (typically 0-1kHz).

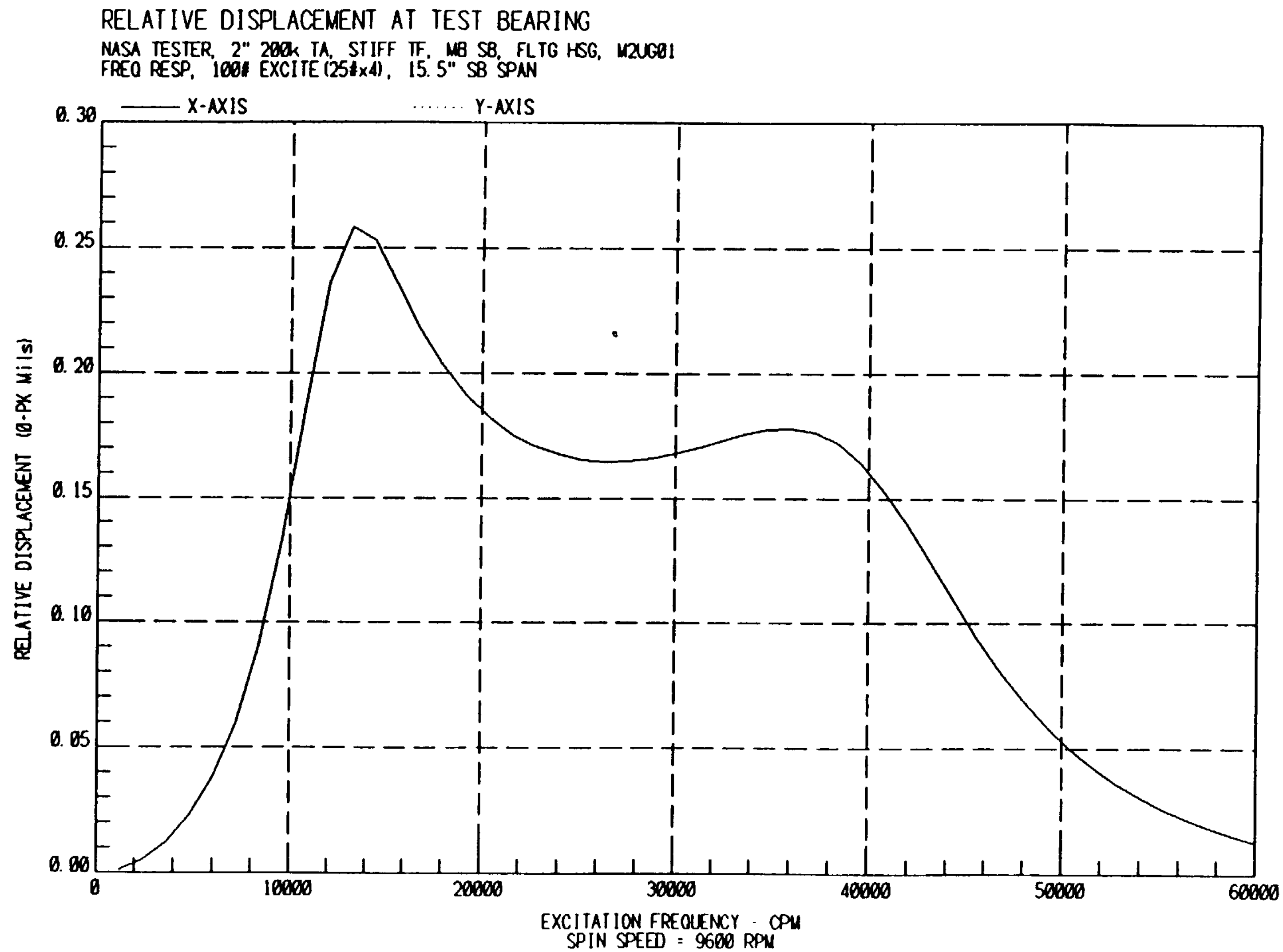


Figure 4-3. Relative test article response for floating housing/magnetic bearing rig, 2" journal, 200,000 lb/in test article, stiff magnetic bearing

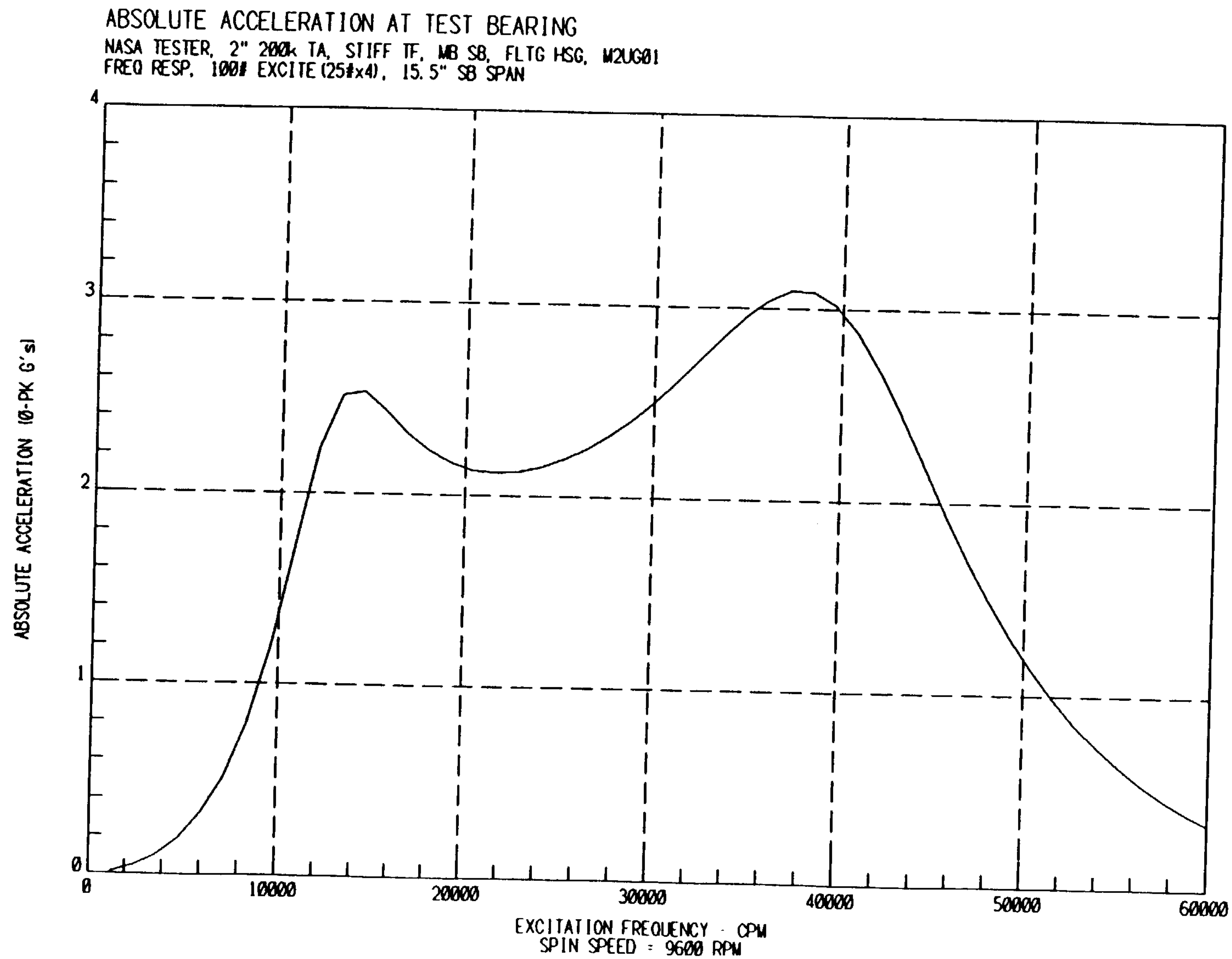


Figure 4-4. Test article acceleration for floating housing/magnetic bearing rig, 2" journal, 200,000 lb/in test article, stiff magnetic bearing

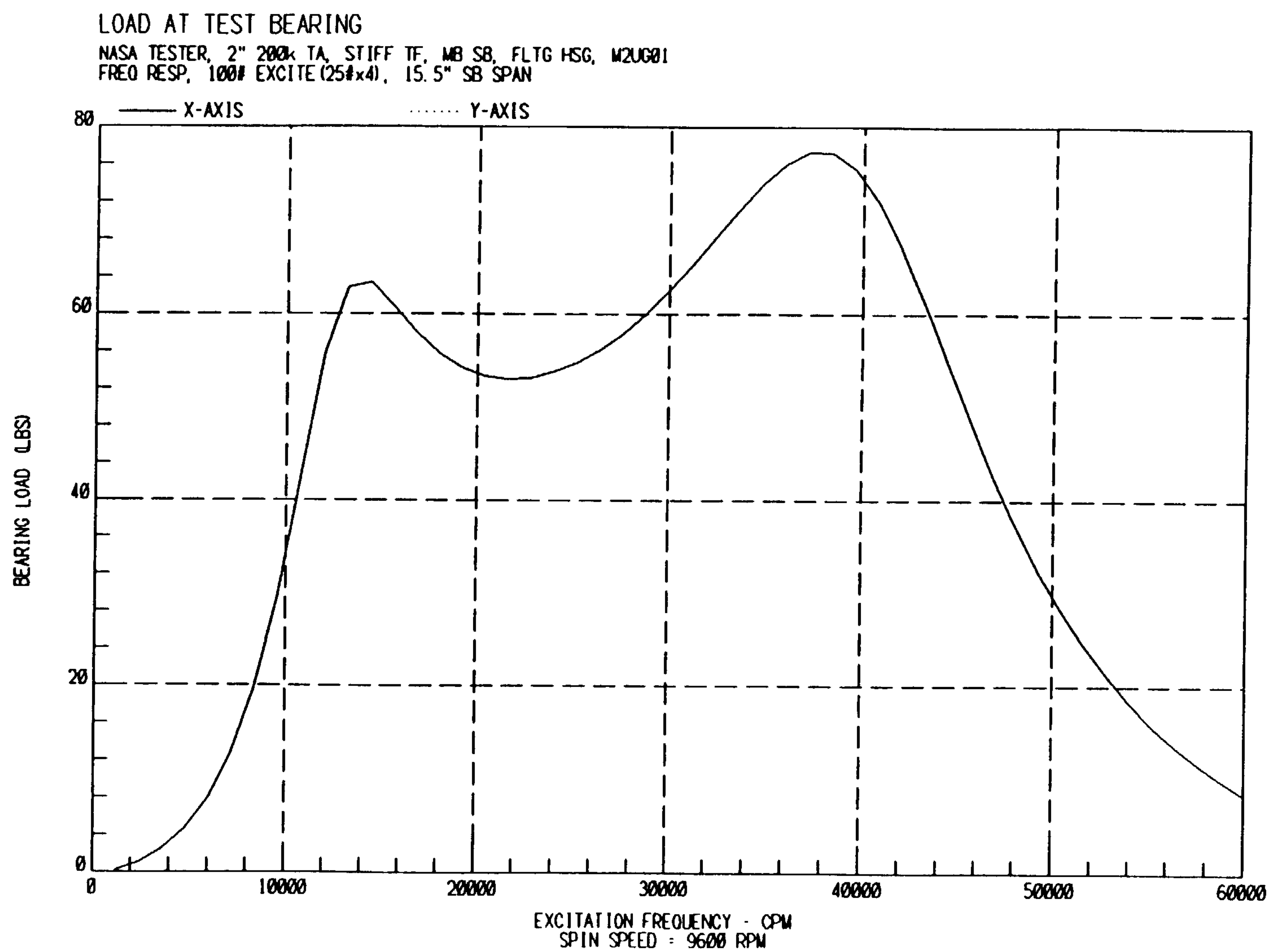


Figure 4-5. Test article load for floating housing/magnetic bearing rig, 2" journal, 200,000 lb/in test article, stiff magnetic bearing

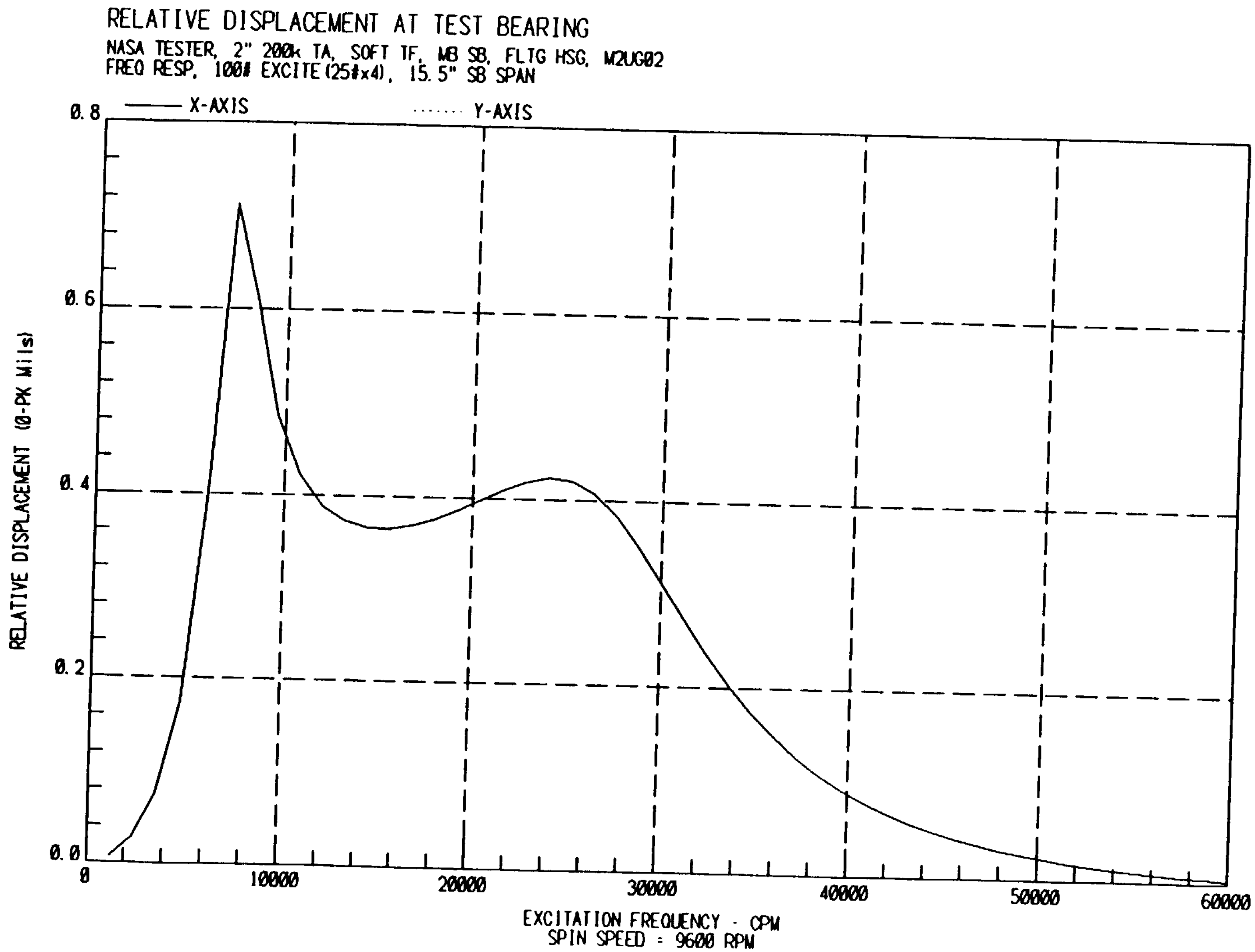


Figure 4-6. Relative test article response for floating housing/magnetic bearing rig, 2" journal, 200,000 lb/in test article, soft magnetic bearing

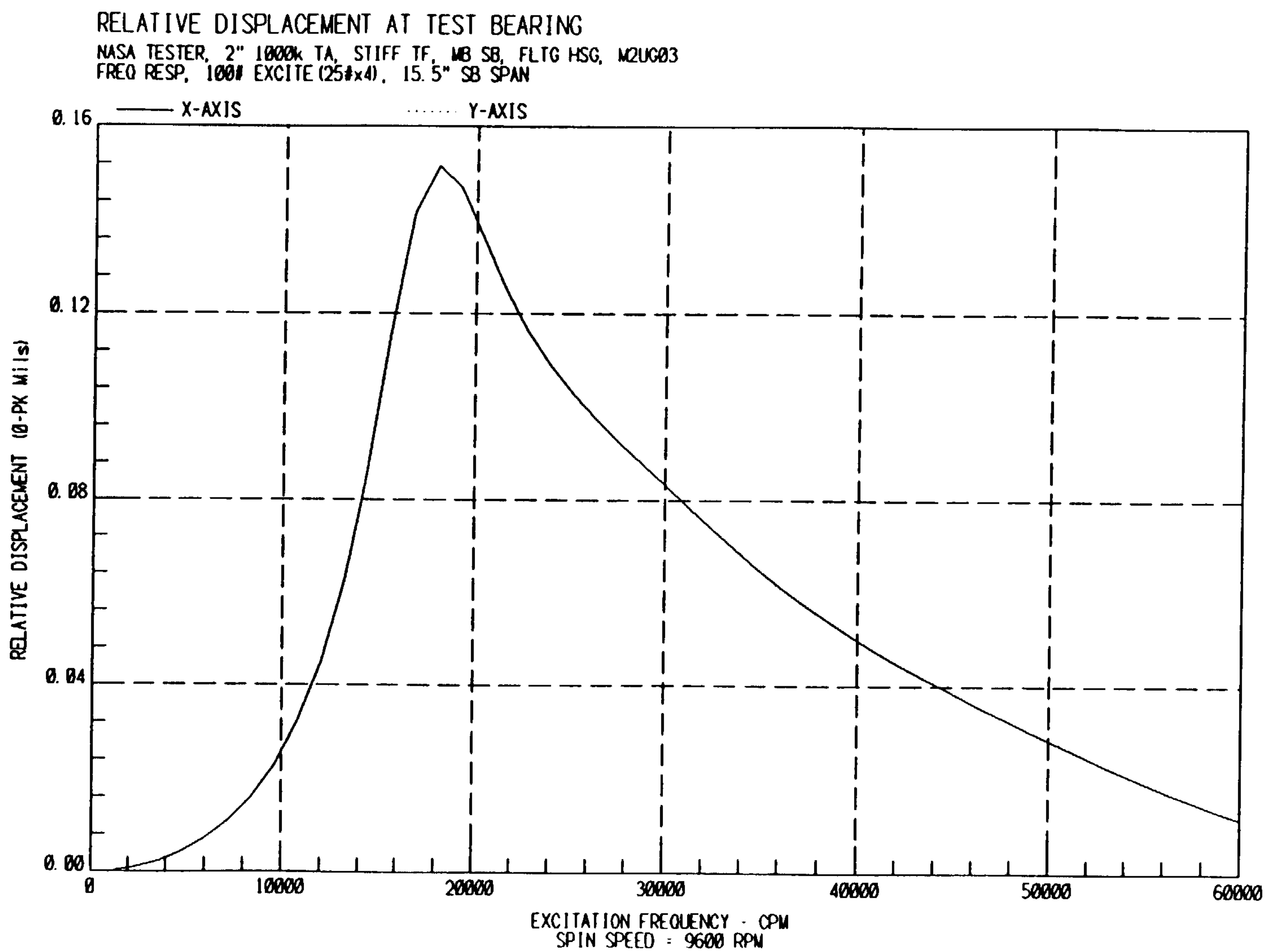


Figure 4-7. Relative test article response for floating housing/magnetic bearing rig, 2" journal, 1,000,000 lb/in test article, stiff magnetic bearing

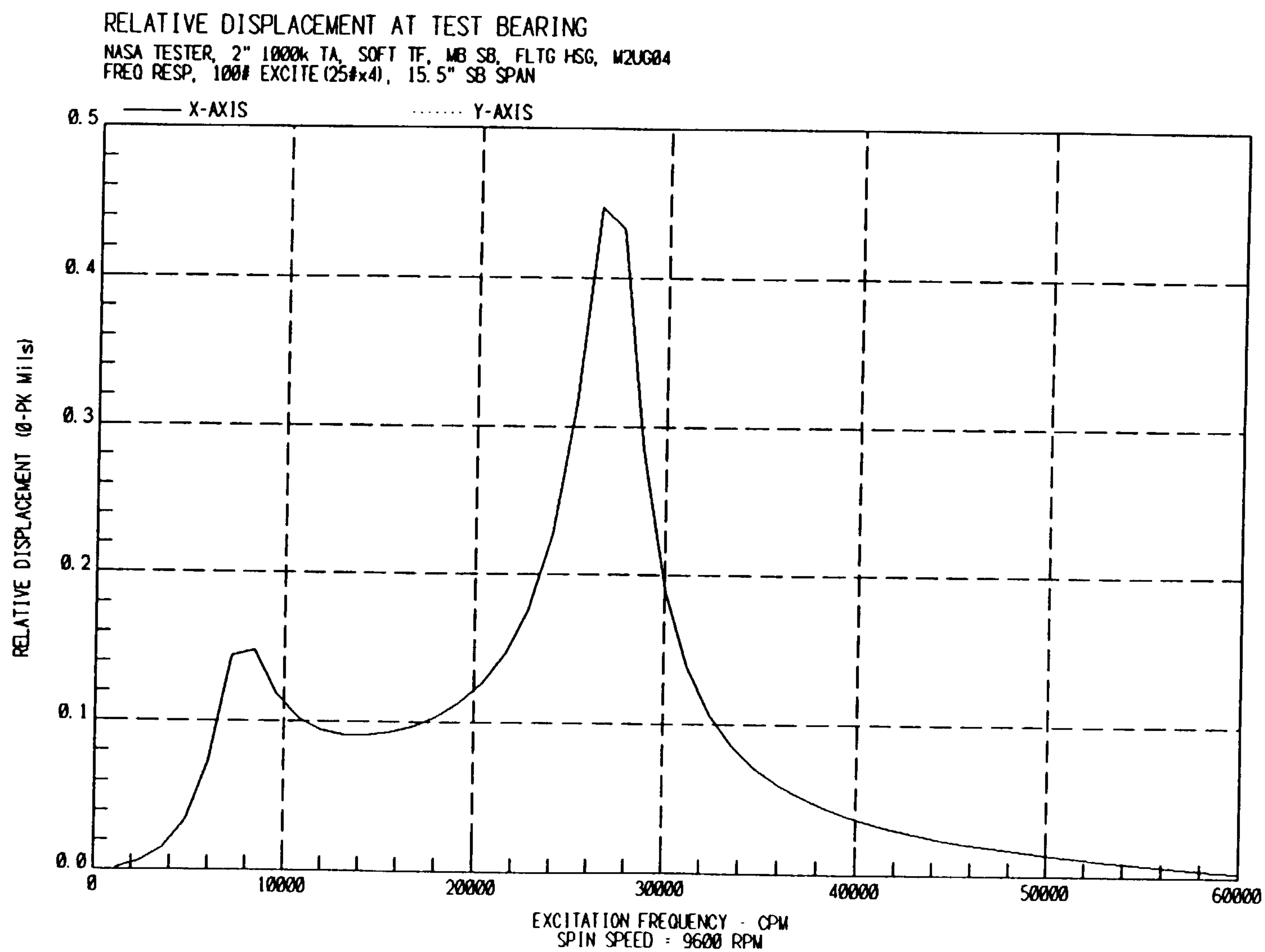


Figure 4-8. Relative test article response for floating housing/magnetic bearing rig, 2" journal, 1,000,000 lb/in test article, soft magnetic bearing

Figures 4-9-4-12 show relative rotor/stator displacement for four additional cases that were analyzed with a 6 inch test article journal diameter. Figures 4-9-4-12 correspond directly to Figures 4-3 and 4-6-4-8 for the 2 inch test article. By comparison of the figures, the higher test article mass reduced the natural frequency of vibration for the responding modes of the tester. As a consequence, three vibration modes are in the frequency range of interest with the larger bearings. Of course, the larger bearings will actually be tested at much higher stiffness values than the smaller 2 inch bearing, partially negating this effect.

4.1.1.2 <u>Fixed Housing (2).</u> Frequency response analysis results with the fixed housing are shown in Figures 4-13-4-22. Figure 4-13 shows the relative rotor/stator response at the shaft center for a 2 inch test article journal diameter, 200,000 lb/in test article stiffness and the stiff magnetic bearing transfer function. The one response peak is due to the second mode of the tester as the first mode is well damped by the analyzed test article damping. Bearings having less net damping (significant cross-coupled stiffness or less direct damping) would have a first mode response. A significant difference between this configuration and the floating housing configuration is that there are no null points in the frequency range of interest. The relative response levels are similar for the similar force input. Figures 4-14 and 4-15 show test article acceleration and test bearing load as a function of excitation frequency for this case. Note that the acceleration is several orders of magnitude below the levels predicted for the floating housing tester. This is because of the very high natural frequency (2-4kHz) of the test article support relative to the frequency range of interest (0-1kHz), and indicates that the acceleration measurement will be much less important with the fixed housing tester. The response levels are such that spectral shaping of the input could be used to produce an acceptably flat response over the entire frequency range of interest.

Figures 4-16-4-18 show relative rotor/stator displacement for three additional cases; Figure 4-16 is for a soft magnetic bearing transfer function, Figure 4-13 is a stiff magnetic bearing transfer function and 1,000,000 lb/in test article stiffness, and Figure 4-18 is for a soft magnetic bearing transfer function and 1,000,000 lb/in test article stiffness. The differences in response among these three cases and Figure 4-13 are due to the differences in the locations and damping levels of the rotor natural frequencies due to the different support stiffness values. In each case flat, measurable response spectrums should be possible with reasonable input force and spectral shaping.

Figures 4-19-4-22 show relative rotor/stator displacement for four additional cases that were analyzed with a 6 inch test article journal diameter. The larger test article diameter results in the same two differences stated before: 1) higher test article stator mass - 168 lb compared to 25 lb, an effect which is not as important with the fixed housing tester, and 2) a stiffer rotor which tends to increase the rotor

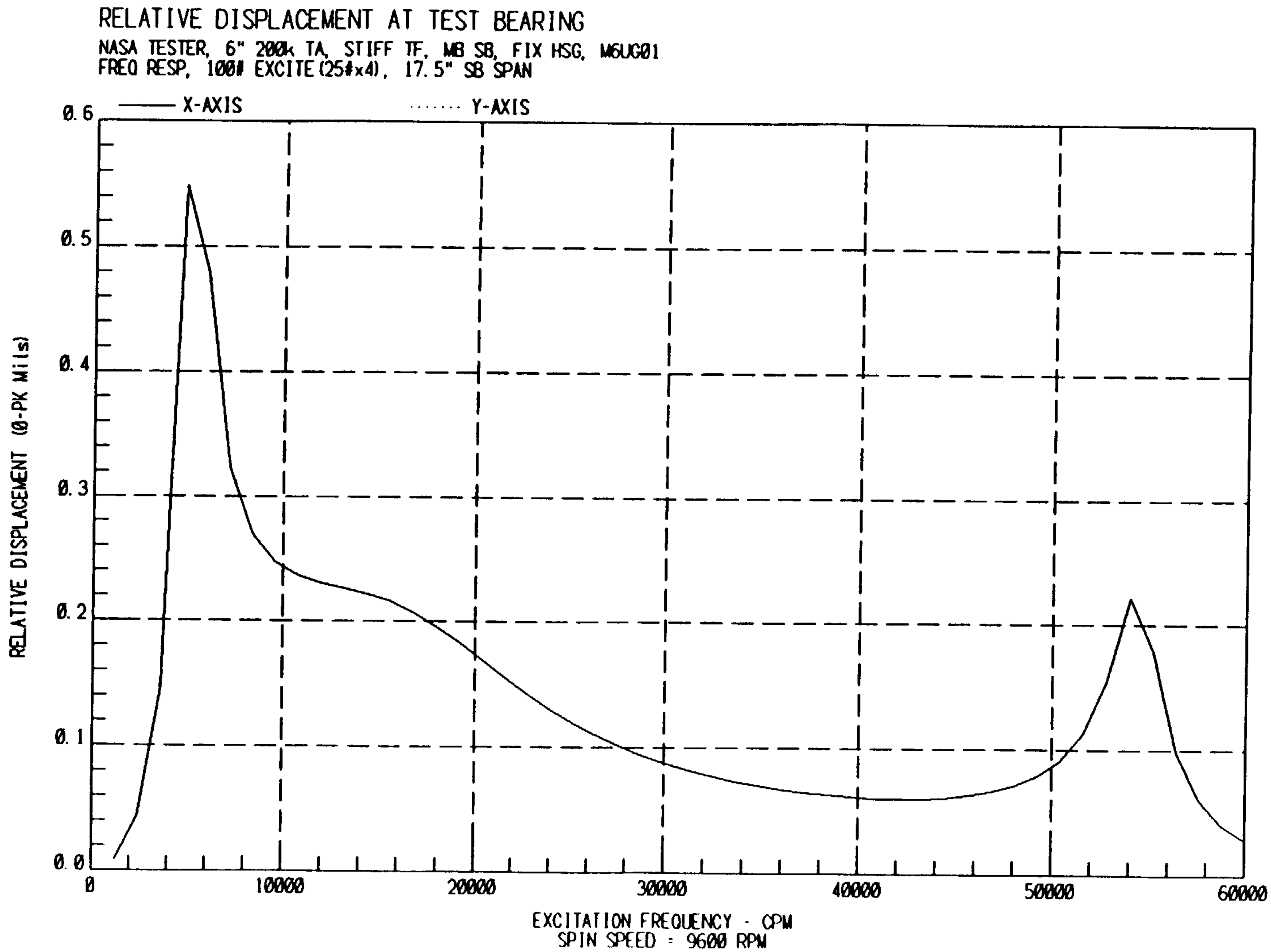


Figure 4-9. Relative test article response for floating housing/magnetic bearing rig, 6" journal, 200,000 lb/in test article, stiff magnetic bearing

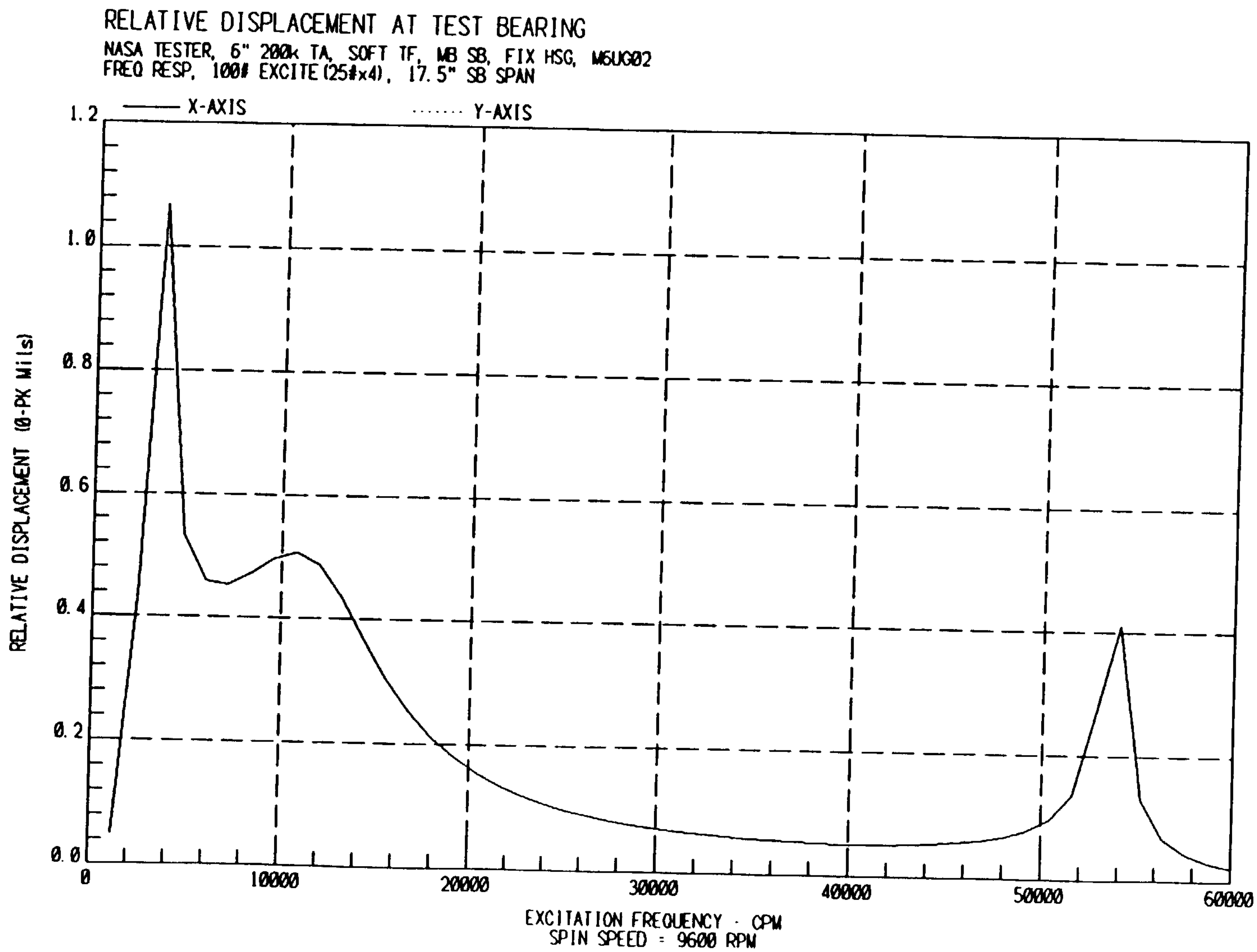


Figure 4-10. Relative test article response for floating housing/magnetic bearing rig, 6" journal, 200,000 lb/in test article, soft magnetic bearing

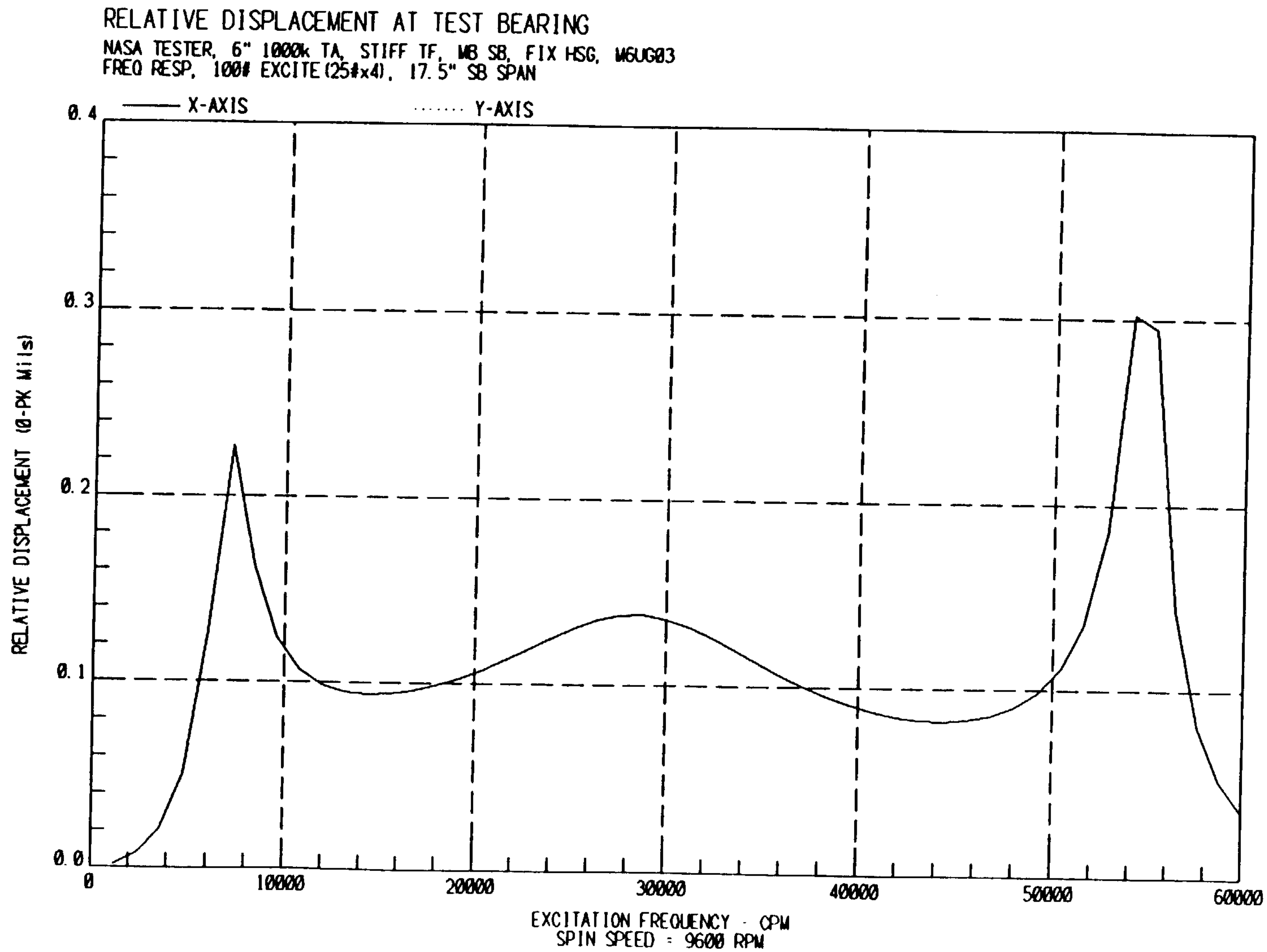


Figure 4-11. Relative test article response for floating housing/magnetic bearing rig, 6" journal, 1,000,000 lb/in test article, stiff magnetic bearing

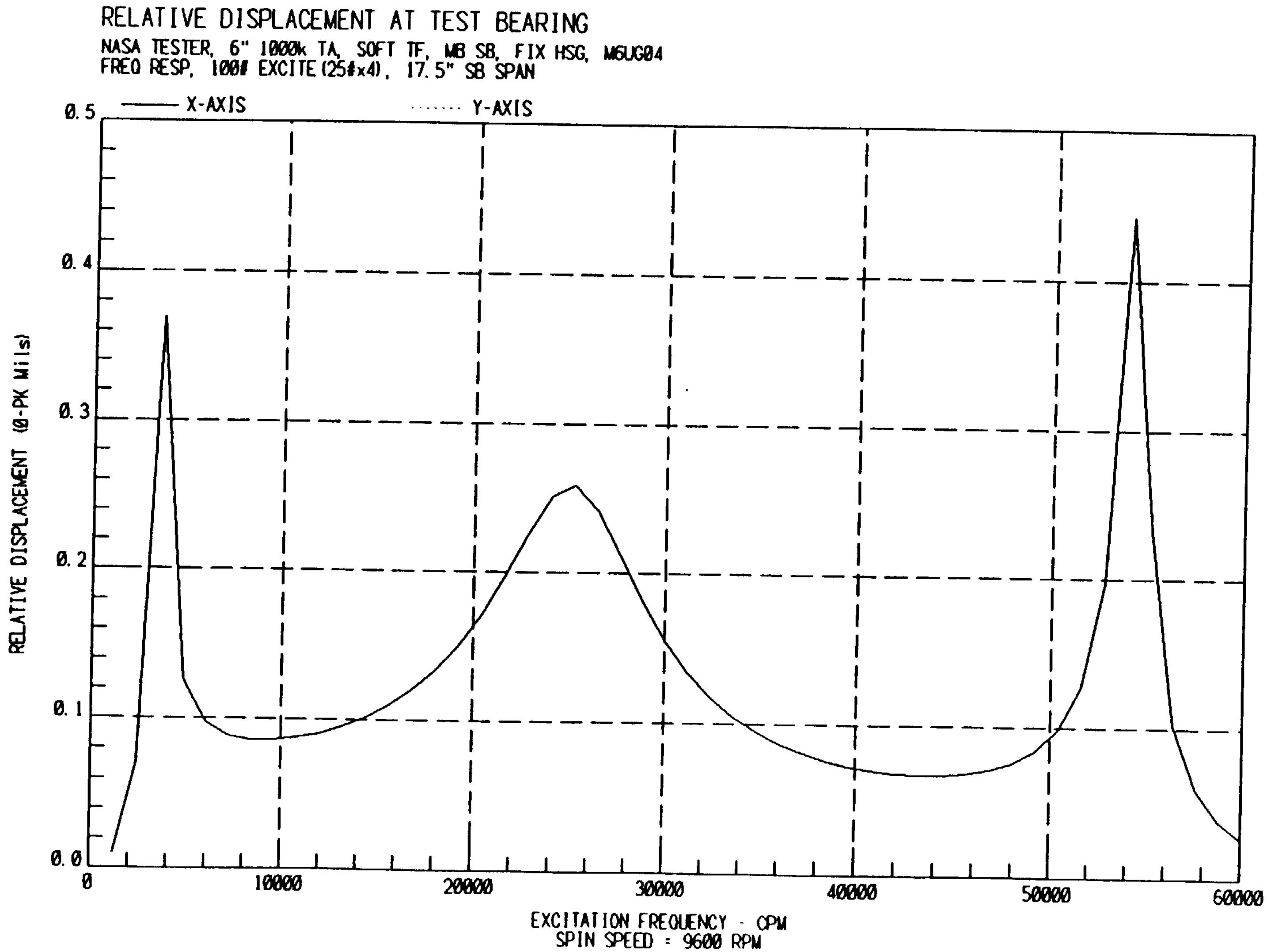


Figure 4-12. Relative test article response for floating housing/magnetic bearing rig, 6" journal, 1,000,000 lb/in test article, soft magnetic bearing

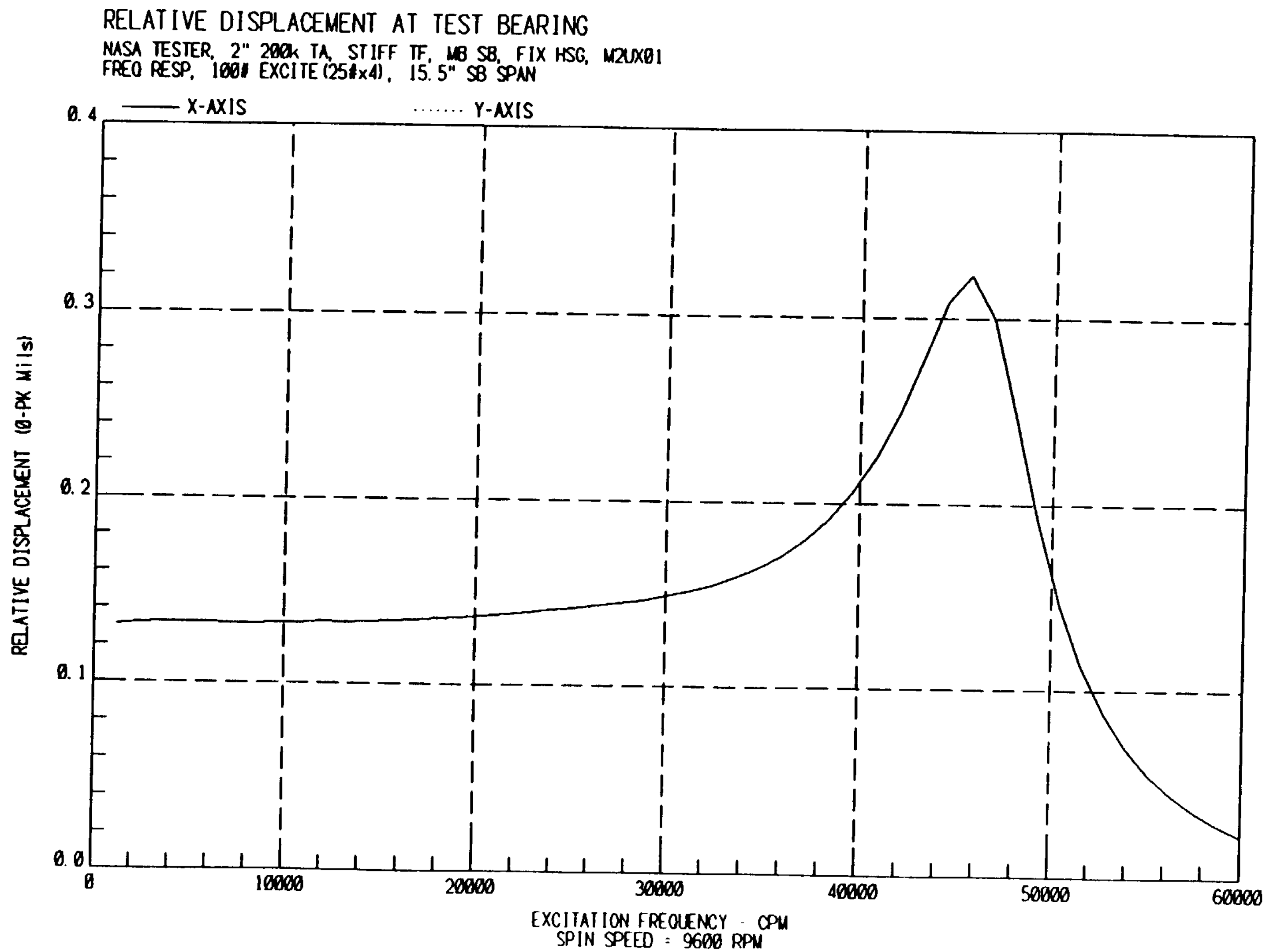


Figure 4-13. Relative test article response for fixed housing/magnetic bearing rig, 2" journal, 200,000 lb/in test article, stiff magnetic bearing

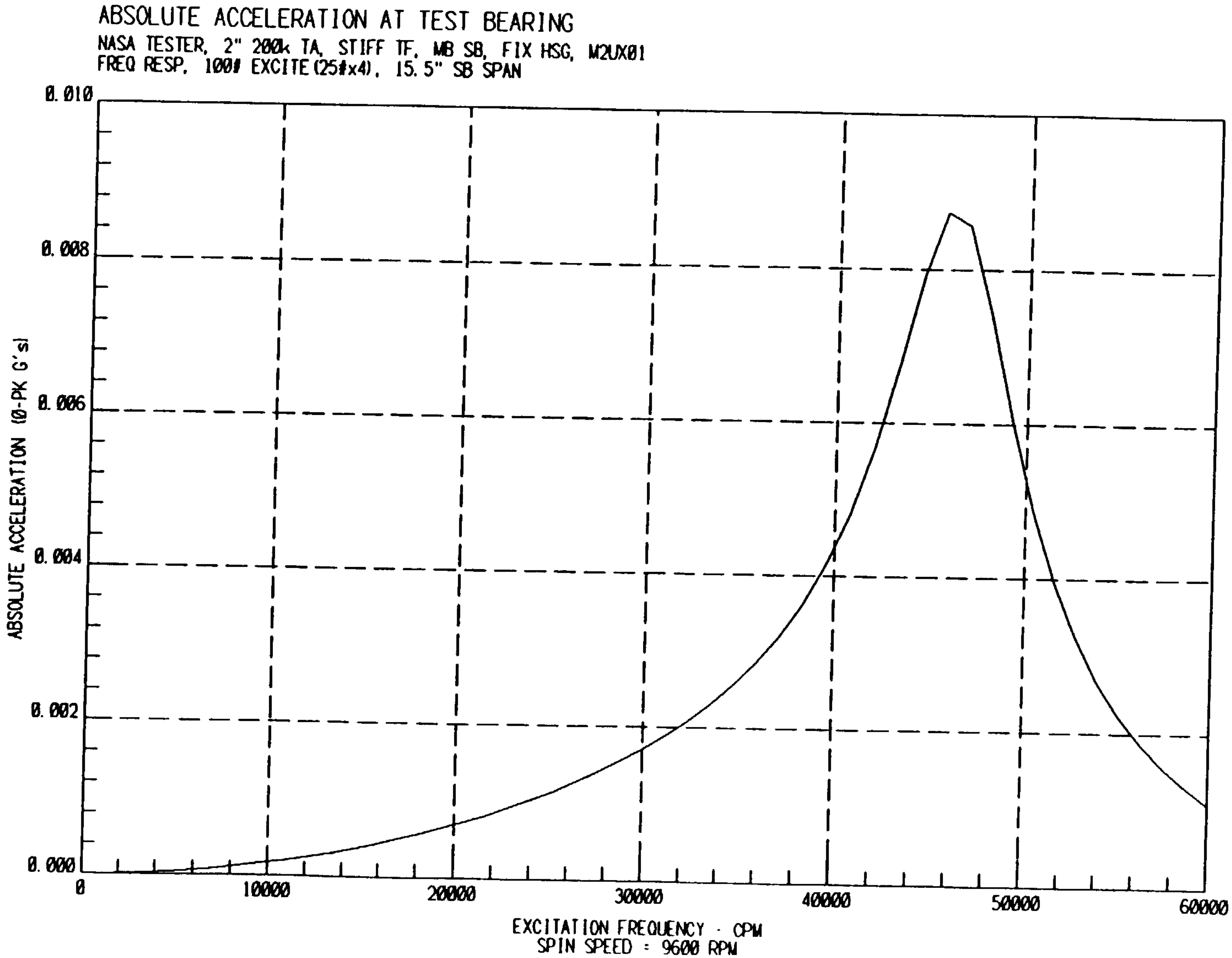


Figure 4-14. Test article acceleration for fixed housing/magnetic bearing rig, 2" journal, 200,000 lb/in test article, stiff magnetic bearing

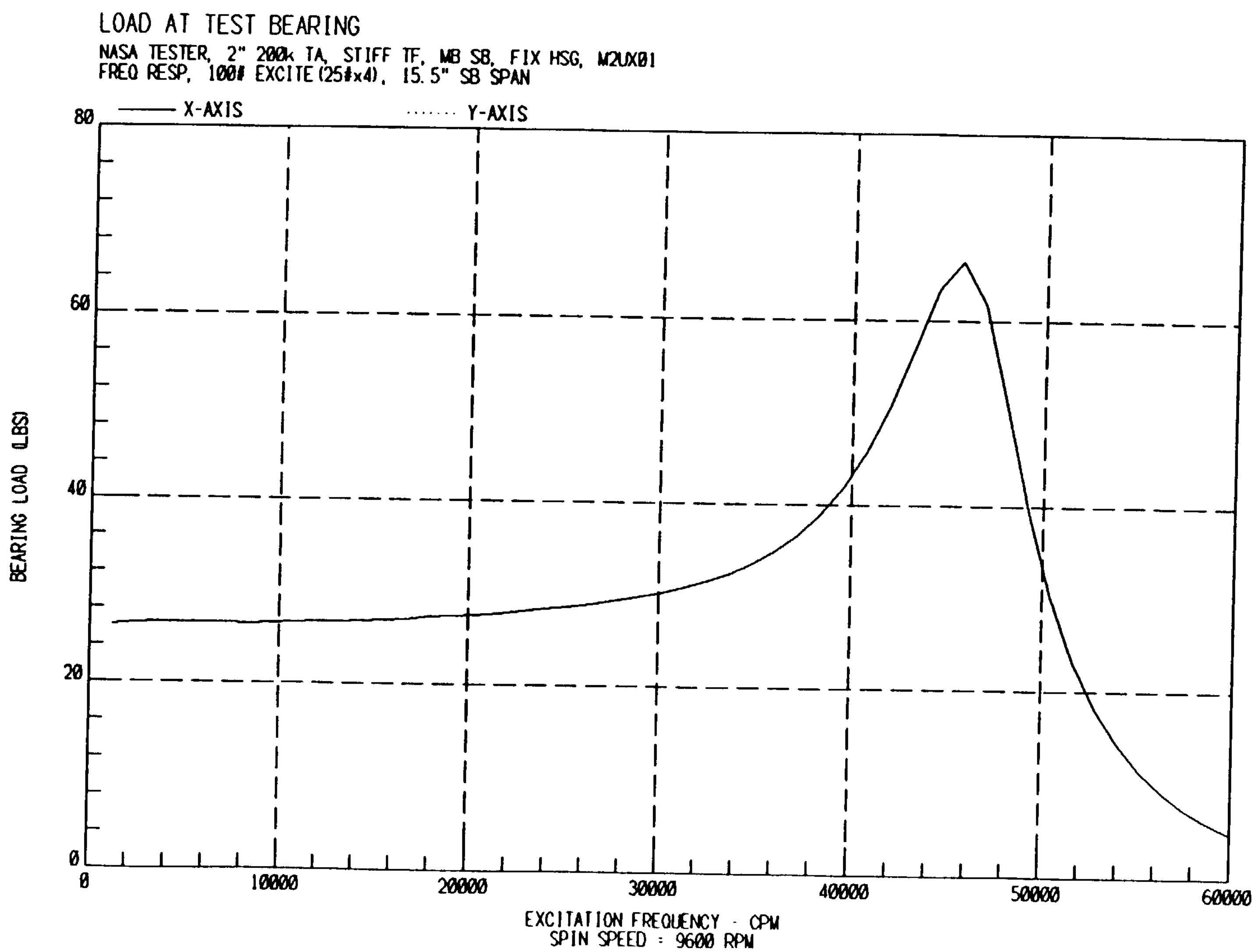


Figure 4-15. Test article load for fixed housing/magnetic bearing rig, 2" journal, 200,000 lb/in test article, stiff magnetic bearing

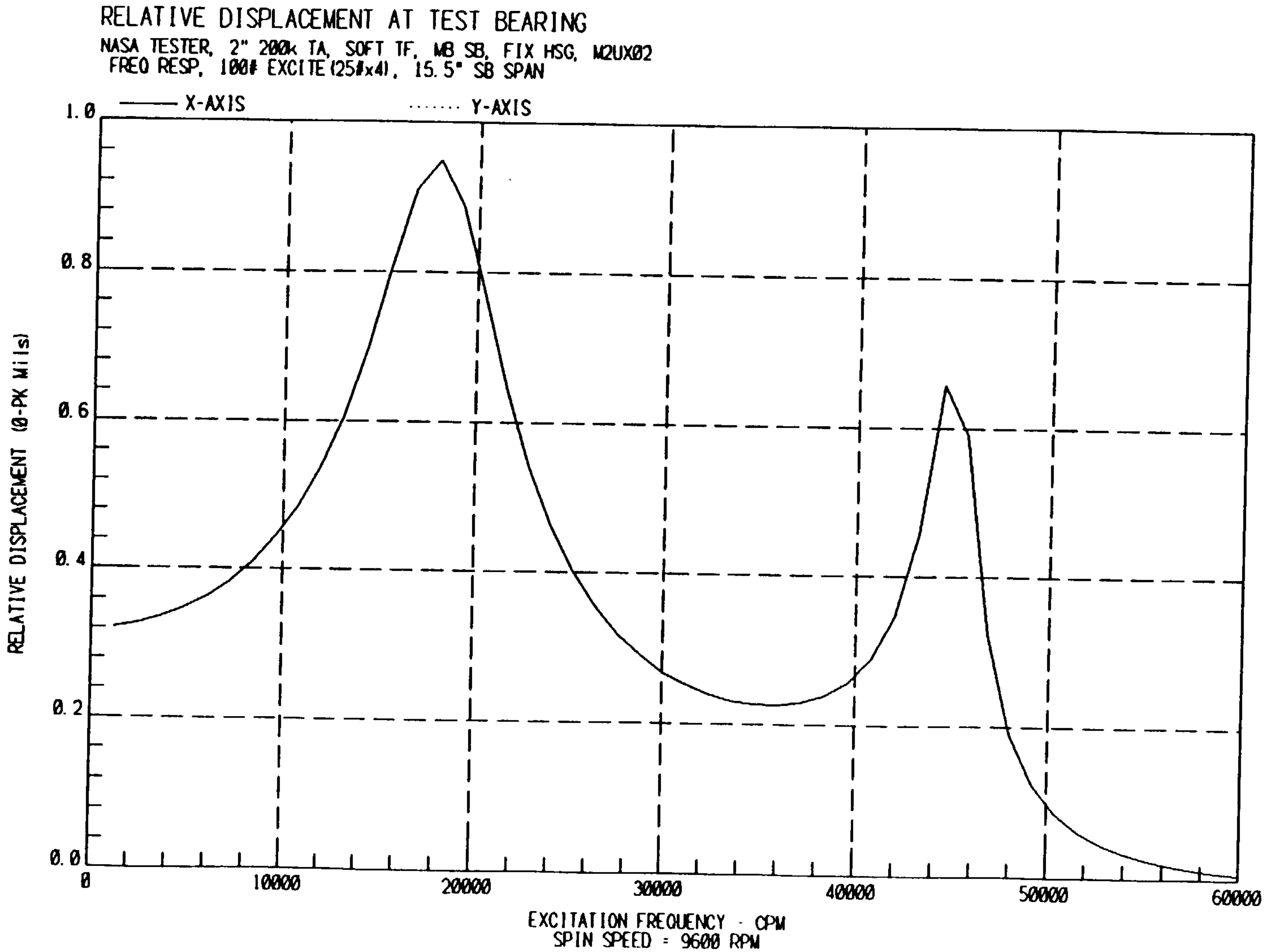


Figure 4-16. Relative test article response for fixed housing/magnetic bearing rig, 2" journal, 200,000 lb/in test article, soft magnetic bearing

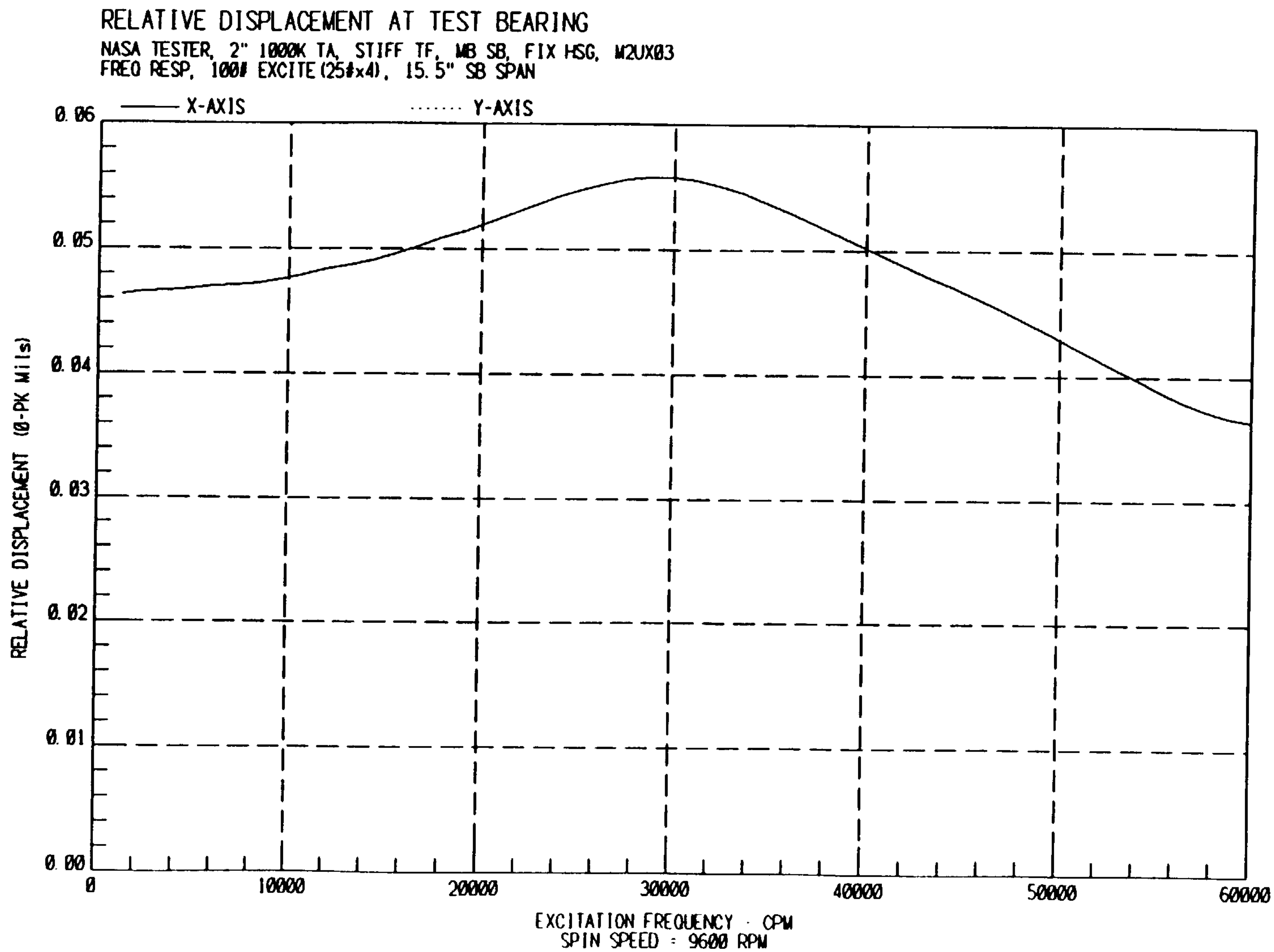


Figure 4-17. Relative test article response for fixed housing/magnetic bearing rig, 2" journal, 1,000,000 lb/in test article, stiff magnetic bearing

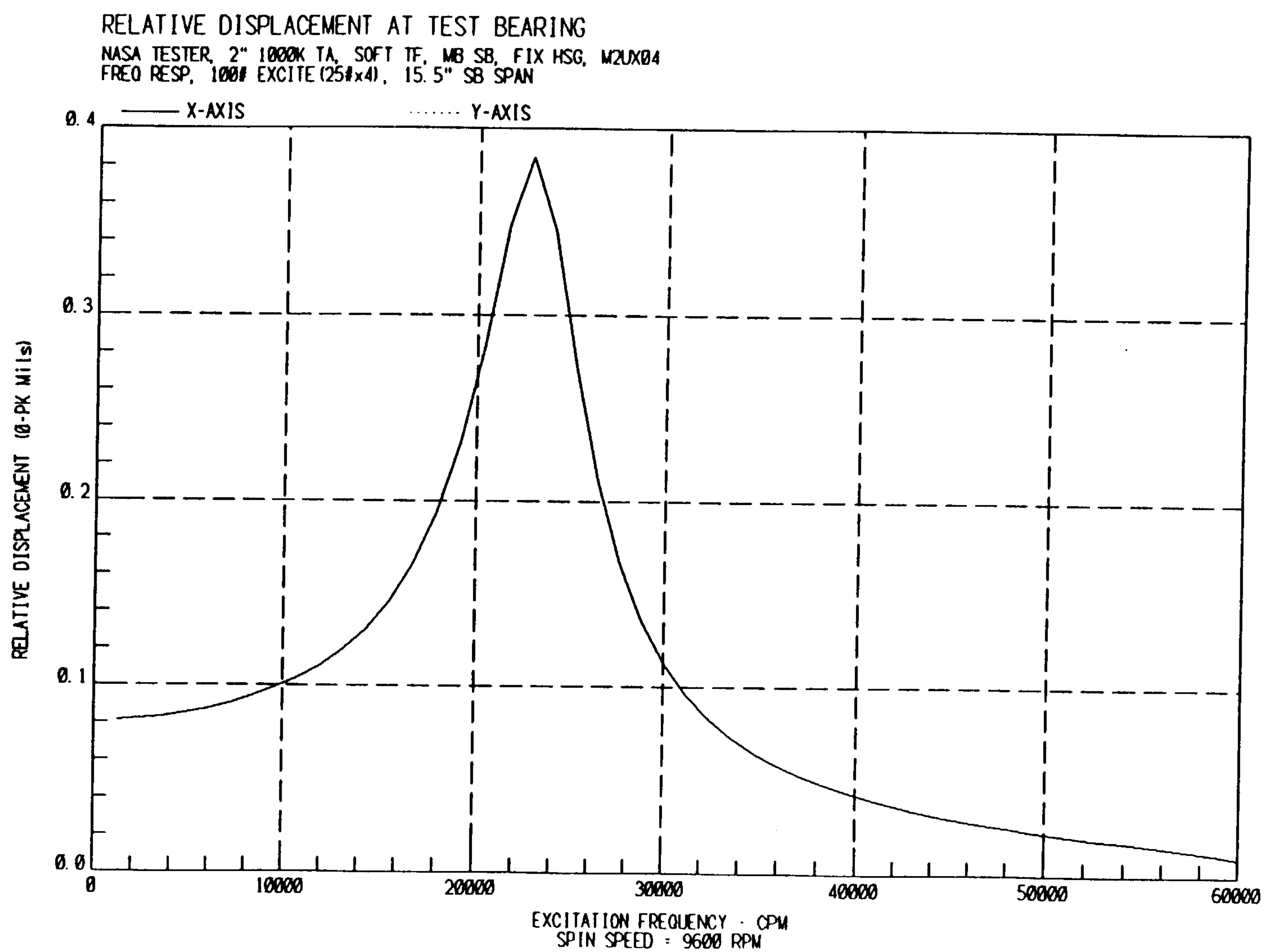


Figure 4-18. Relative test article response for fixed housing/magnetic bearing rig, 2" journal, 1,000,000 lb/in test article, soft magnetic bearing

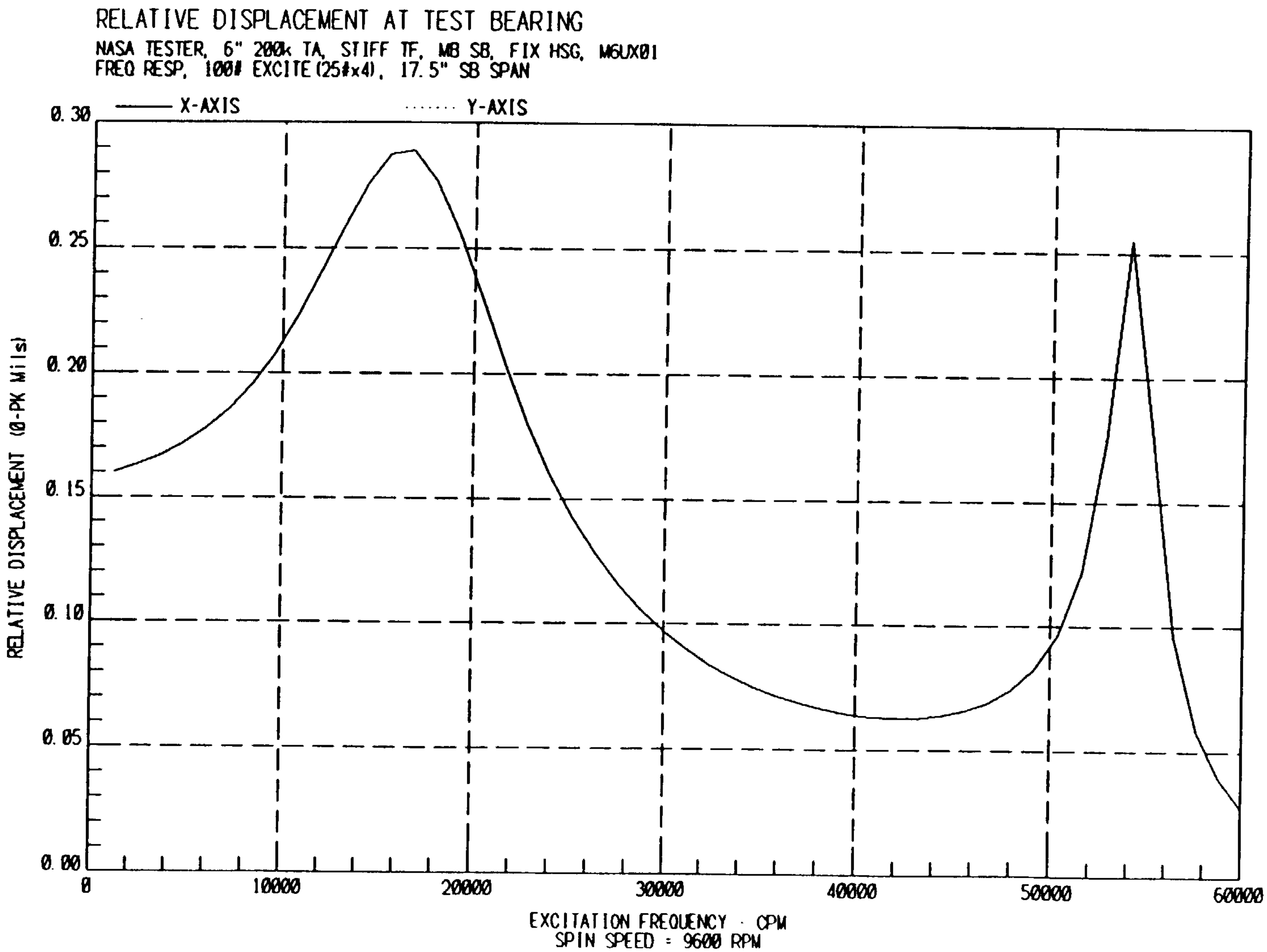


Figure 4-19. Relative test article response for fixed housing/magnetic bearing rig, 6" journal, 200,000 lb/in test article, stiff magnetic bearing