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C. Hunter Cloud
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Lloyd E. Barrett



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BRG Machinery Consulting, LLC
703 Highland Avenue
Charlottesville, Virginia 22903 USA
Phone\Fax: (434) 293–9926
www.BRGmachinery.com

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# Influence Of Tilting Pad Journal Bearing Models On Rotor Stability Estimation

**C. Hunter Cloud**
BRG Machinery Consulting
Charlottesville, Virginia USA

**Eric H. Maslen**
James Madison University
Harrisonburg, Virginia USA

**Lloyd E. Barrett**
University of Virginia
Charlottesville, Virginia USA

## ABSTRACT

When predicting the stability of rotors supported by tilting pad journal bearings, it is currently debated whether or not the bearings should be represented with frequency dependent dynamics. Using an experimental apparatus, measurements of pad temperatures, unbalance response and stability are compared with modeling predictions for two tilting pad bearing designs. Predictions based on frequency dependent tilting pad bearing dynamics exhibited significantly better correlation with the stability measurements than those assuming frequency independent dynamics.

## NOMENCLATURE

| | |
|---|---|
| $c_b$ | Bearing assembled clearance |
| $c_p$ | Bearing pad machined clearance |
| FC | Full coefficient |
| $m_p$ | Bearing preload ($m_p = 1 - c_b/c_p$) |
| $q$ | Cross-coupled stiffness |
| SRC | Synchronously reduced coefficient |
| $\omega_d$ | Damped natural frequency |
| $\zeta$ | Critical damping ratio |

## INTRODUCTION

Developed in the early 1900s almost simultaneously by Kingsbury and Michell, tilting pad journal bearings were recognized by Hagg [1] to have superior stability performance. Hagg established that this superior performance stems from the ability of the pads to tilt in response to rotor motion. This tilting means a bearing with $N_{pad}$ pads has $N_{pad}+2$ degrees of freedom, and actually requires $5N_{pad}+4$ stiffness and $5N_{pad}+4$ damping coefficients to define its dynamic properties [2]. Lund [3] showed that these "complete" or "full" coefficients (FC) can be reduced to the typical eight coefficients in order to be compatible with rotordynamic code computational limitations present at the time. The key assumption for this reduction or simplification is that the vibrational frequencies of the pads and shaft are equal to the rotational speed, resulting in "synchronously reduced coefficients" (SRC).

Warner and Soler [4] recognized that, for stability analysis purposes, this synchronously reduced assumption was not appropriate. They showed that the eight reduced coefficients were actually frequency dependent, particularly at higher Sommerfeld numbers. The effects of this frequency dependency on the reduced coefficients and the overall system stability has been extensively investigated [5,6,7,8,9], which all conclude that stability predictions are often strongly influenced by the choice of tilting pad bearing representation (FC or SRC).

Over the last 40 years, however, machines have been designed using SRCs and most of the confidence factors surrounding destabilizing force empirical models and acceptable stability levels (i.e., $\delta \geq 0.1$ for stable operation) were based on the synchronously reduced representation. This experience on the part of manufacturers, users and consultants has led to a general lack of acceptance regarding the theoretical work on tilting pad bearings' frequency dependency. Such practice contrasts sharply with the general acceptance of the importance of considering frequency dependent dynamic characteristics of magnetic bearings and honeycomb/hole pattern gas annular seals. These components can be strongly frequency dependent, which is not always the case for tilting pad bearings. Furthermore, the standing literature on experimental efforts to measure tilt pad bearing dynamics does not provide compelling evidence of the frequency dependence that would necessitate use of FC models [10,11,12].

There are two primary arguments supporting the use of a synchronously reduced representation. The first argument acknowledges the validity of the FC representation, but argues that the effect of the added dynamics begins at frequencies too high to affect stability predictions for the first several modes of the machine. If so, then a component modal reduction of relatively complicated FC based model is appropriate and, arguably, leads to an SRC model. For such an argument to be correct, the stability predictions for these lower modes should be unaffected by replacing the FC model with an SRC model. Hence, this argument can be assessed through computational effort alone and does not require experimentation.

The second argument claims that, because the dominant vibratory frequency within the bearing at the stability threshold is synchronous, it is sufficient to consider the SRC model to at least establish stability thresholds [13]. This argument is primarily heuristic and difficult to justify on theoretical grounds, but it has considerable currency in the industry. Clearly, it is not possible to evaluate this argument using computational means, so we resort here to experiment.

The ongoing debate over which representation more accurately predicts stability was recently highlighted in a survey conducted by the American Petroleum Institute [14]. To address this debate, the central objective of the present investigation was to try to distinguish between tilting pad behavior which is frequency independent (supporting the use of SRCs) and that which is frequency dependent (supporting the use of the FCs) by examining the agreement between predicted and measured damped eigenvalues. Given a typical rotor system, well established theoretical foundations already predict significant differences in damped eigenvalues and stability thresholds between these two representations: evidence which tends to refute the modal truncation argument. Therefore, we develop experimental approaches to work in conjunction with these predictions to attempt to address the second, more heuristic argument. A fair assessment of which predictive model correlates best with the measurements requires two things: (1) the measurements and the models must not be artificially biased toward a false correlation, and (2) the uncertainties in the measurements and the model are not so large as to be inconclusive. These requirements translate into a need for accuracy and precision in both the stability predictions and measurements.

The remainder of the paper will first present the test apparatus and develop an analytical model for it, establishing the base stability predictions with FC and SRC bearing models. These results comment on the validity of modal truncation, showing that the stability predictions of the two bearing models are not similar, even for low frequency modes. With this prediction in place, the remainder of the paper examines experimental estimates of stability, showing stronger correspondence to the FC model than to the SRC model. These results comment on the heuristic argument, supporting the proposition that deriving the bearing behavior from synchronous vibration arguments is not valid.

## MODELING OF TEST APPARATUS

The experimental apparatus consisted of a 1.55 m (61 in.) long rotor supported by two tilting pad journal bearings and incorporated three magnetic actuators. Figure 1 displays the overall rig layout. Two of the magnetic actuators, designated Actuators 1 and 2, were located between the journal bearings for application of cross-coupling stiffness. These locations correspond to those of various destabilizing components, such as oil and labyrinth seals, found in industrial between-bearing machines. The third actuator or shaker was located at the outboard end, where previous industrial stability measurement investigations [15, 16] applied external excitation. Five planes of vibration were monitored using eddy current proximity probes. With a midspan diameter of 90 mm (3.5 in.) and bearing span of 1.22 m (48 in.), the shaft was made of AISI 4340 steel.

Each ⌀70 mm (2.75 in.) tilting pad bearing consisted of five, 52.3 mm (2.06 in.) long steel pads, oriented in the load-between-pad configuration. Two sets of bearings were tested, one set with 0.3 preload pads ($m_p = 0.3$), and the other with 0.1 preload pads. With their assembled clearance maintained at approximately 173 $\mu$m (6.8 mils) diametral, the two sets' preloads were varied through pad machined

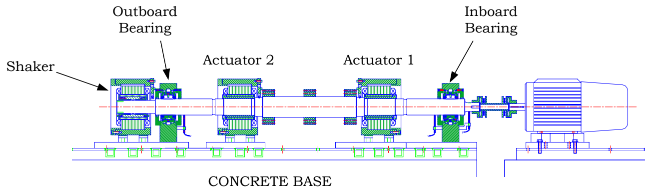



Figure 1: Rig layout

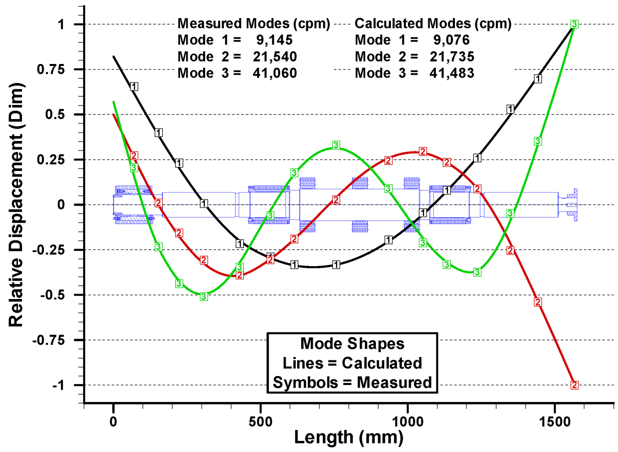



Figure 2: Comparison between calculated and measured free-free rotor modes

clearance differences. Both designs shared common bearing housings and ball pivots. Thus, preload was changed by installing new pads with a different pad machined bore. Complete details of the rig are given in [17].

### Component Modeling

Following the typical modeling process, individual component models were developed for the dynamics of the rotor, tilting pad journal bearings and bearing pedestals. For measurement and control of the actuator dynamics, a force model for the magnetic actuators was also developed.

Model development of the rotor assembly's dynamic properties was a multiphase process involving free-free modal testing before and after mounting of various components. The final reconciled rotor model achieved excellent agreement with the measured modal parameters. The first six free-free natural frequencies agreed within approximately five percent. For the first three free-free modes, Figure 2 compares the measured mode shapes, established from 15 response locations along the rotor, with those predicted by the final reconciled model. Natural frequency predictions for these three modes are within 1% of measured.

It has been well documented by many investigators that the dynamic properties of the bearings' supporting structure can have a very strong influence on the overall rotor system dynamics. To account for any such influence on this test rig, each bearing's pedestal structure was modally tested and identified for modeling purposes. Direct and cross-talk fre-

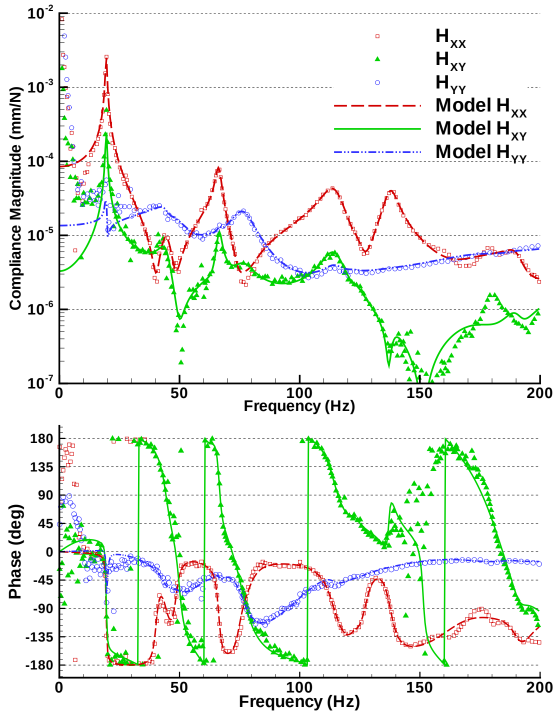



Figure 3: Inboard pedestal model versus measured compliance

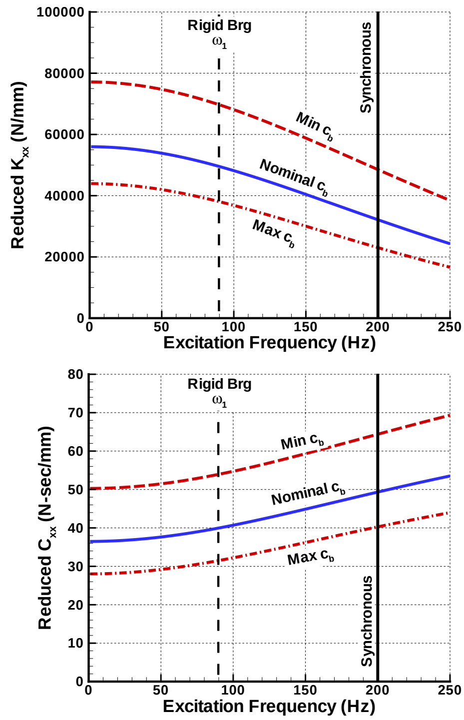



Figure 4: Inboard bearing 0.3 preload design reduced stiffness and damping coefficient bounds (12,000 rpm)

quency response functions (FRFs) were measured on both pedestals. Transfer function models for the pedestal system's dynamics were then obtained by applying a commercial multiple input, multiple output curve fitting algorithm on the measured FRFs. Figure 3 presents an example of the very good agreement between the identified model and measurements that were obtained for the pedestal system.

To accommodate the potential for unequal current bias distribution as well as the multitude of possible operating conditions, magnetic circuit models were identified for use in controlling and measuring the actuators. The magnetic circuit model is one-dimensional and treats the actuator as a magnetic network analogous to an electrical network [18]. In the same spirit of reconciling the rotor and pedestal models using test data, the circuit model for each actuator was identified using known forces along with measured currents and positions. The resulting actuator models were accurate to within 7% of the applied forces. In terms of predictive uncertainty, the circuit models achieved $\pm 22.2$ N (5 lbf) with 95% confidence.

Finally, the tilting pad bearings' dynamics were calculated using the two-dimensional, finite element algorithm developed by He [19]. This algorithm has demonstrated very good accuracy when compared with measurements and is used extensively by industry for bearing design and rotordynamic purposes.

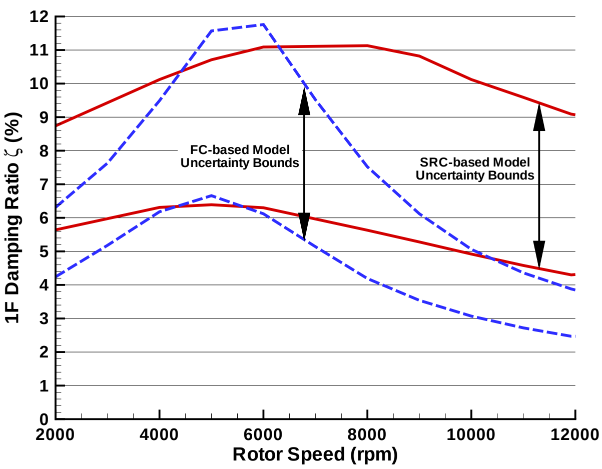
From the stability perspective, the character of the reduced coefficients for subsynchronous frequencies is a primary concern. Figure 4 focuses on the predicted subsynchronous characteristics for the inboard bearing at 12,000 rpm. The three bearing conditions are given which define the estimated uncertainty bounds on the stiffness and damping. The minimum $c_b$, maximum $m_p$, minimum oil temperature condition creates the upper bound on stiffness and damping properties. Lower bounds are established from the maximum $c_b$, minimum $m_p$, maximum oil temperature condition. Depending on the particular excitation frequency, the upper stiffness bound can range from 75% to 112% greater than the lower bound. Likewise, the upper damping bound ranges from 60% to 81% greater than the lower damping bound.

The first rotor mode's predicted undamped natural frequency on rigid bearings is indicated in Figure 4 to highlight its location relative to synchronous frequency. At the nominal bearing condition, the synchronously reduced principle stiffness level is nearly 32,000 N/mm. However, subsynchronous frequencies experience higher stiffness. For instance, the rigid bearing first mode is controlled by 50,000 N/mm, an approximately 55% increase from the synchronously reduced level. Reduced principle damping levels, on the other hand, decrease at subsynchronous frequencies. The model predicts that the rigid bearing first mode experiences nearly 20% less damping than synchronous vibration. While not shown here, the 0.1 preload bearing design's reduced coefficients exhibited even greater frequency dependency.

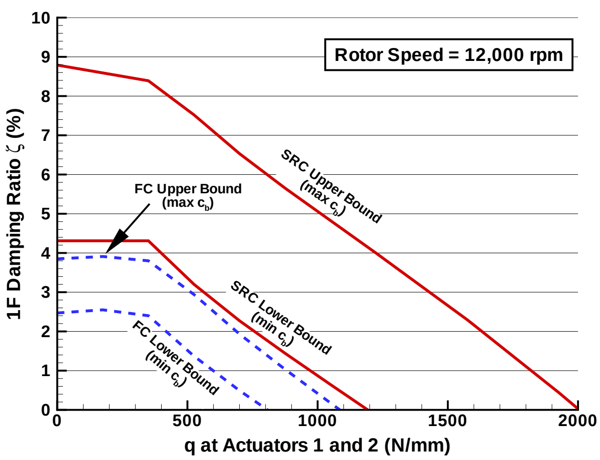
## System Modeling

The various component models were assembled into a system model using an algorithm developed by Vázquez and Barrett [20]. Commonly used by industry to predict stability characteristics of turbomachinery, this algorithm employs a frequency domain search eigensolver.

Since the tilting pad journal bearings have the largest uncertainty of the all component models, it is important to understand how this uncertainty translates into uncertainty in the system's stability predictions. For the first forward mode (1F), Figure 5 presents modal damping uncertainty bounds for the two tilting pad journal bearing representations. Base stability, in Figure 5(a), refers to the stability characteristics established by the system encompassing the rotor, bearings and pedestals alone, without any dynamics from the magnetic actuators. Given the significant overlap that exists between the two bearing representations for base stability, it will be difficult to distinguish which representation most accurately correlates with the damping measurements except at high speeds. However, when actuator cross-coupling is varied in Figure 5(b), no overlap occurs for the stability sensitivity bounds, making it a very good identifier. Unfortunately, the damped natural frequency will likely not serve as a good metric because of the large overlap in predictions.




(a) Base stability




(b) Stability sensitivity

Figure 5: Predicted stability uncertainty bounds ($m_p = 0.3$)

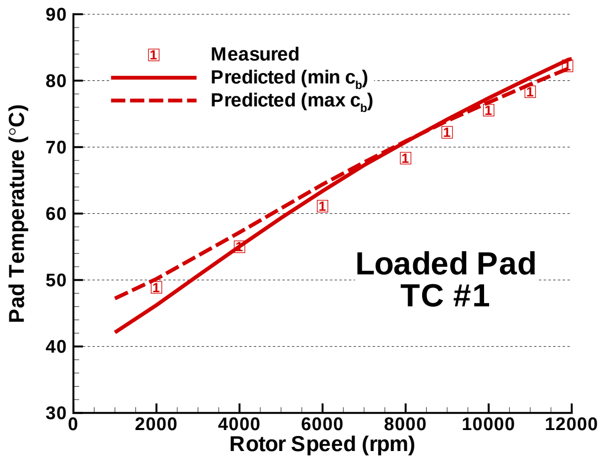



Figure 6: Measured versus predicted pad temperatures ($m_p = 0.3$)

## BEARING PERFORMANCE

Unlike the component models for the rotor assembly, bearing pedestals and magnetic actuators, no component level test data was available to validate the predicted dynamics for the tilting pad bearings. Therefore, the accuracy or bias of the bearing dynamic models remained in question. In an attempt to address this issue, pad temperatures and unbalance response measurements from the test rig were examined with respect to the modeling predictions.

Pad temperatures were recorded for eight speeds in the speed range. At each of these speeds, temperatures were recorded after they had been stable (varying less than $\pm 0.5°$C) for more than 30 minutes, indicating thermal equilibrium had been reached. Figure 6 compares the predicted and measured pad temperatures for one of the two loaded (bottom) pads. The predicted temperatures from the bearing algorithm correlate extremely well with the measurements especially at high speeds. Similar accuracy was achieved for the other three thermocouples within each bearing. While not shown, predictions for the 0.1 preload bearings were approximately 6°C higher than measured. Such good correlations with respect to both bearings' thermal performance provide greater confidence in their predicted dynamics.

Measuring the unbalance response performance provides another platform for assessing the accuracy of the assembled system's dynamic model, of which the bearings' dynamics are a crucial component. This was accomplished by applying 364.4 g-mm (0.506 oz-in) of unbalance on the outboard disc that is adjacent to Actuator 2. After compensating for the residual response, the measured response due to this unbalance alone was compared to the predictions. Figure 7 presents this comparison for the two orthogonal displacement probes at one of the bearings.

Looking at the predictions, the general character of the response is modeled very well. For the $x$ probe, the minimum clearance condition predicts a well-amplified peak within 0.5% of the peak speed measured, and its amplitude is only

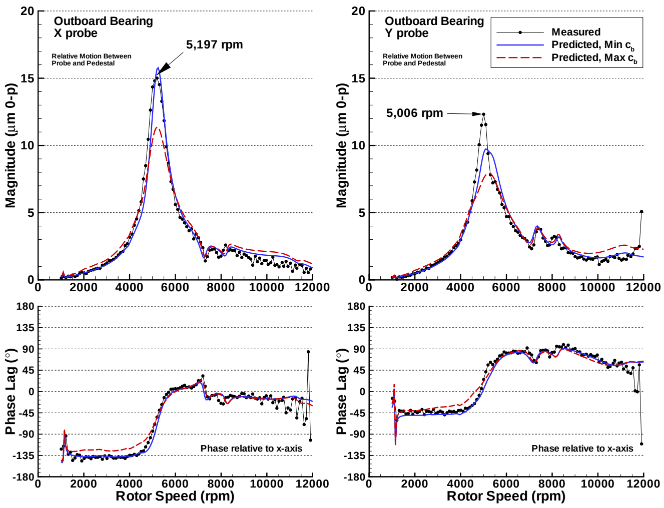



Figure 7: Measured and predicted unbalance response ($m_p = 0.3$)

5% higher than measured. Unfortunately, the $y$ probe's peak response characteristics are less accurately predicted. Here, the minimum clearance condition predicts the peak to occur 1.7% away from the measured peak. The peak amplitude is noticeably underpredicted by 25% using the minimum clearance condition model. Around 8,000 rpm, both peaks associated with the two pedestal structure natural frequencies are predicted relatively well. The second structural mode's peak response speed is predicted to be slightly higher (200 cpm) than measured. As a note, the erratic phase measurements at speeds above 11,000 rpm are due to the very small response amplitudes present, making phase measurement difficult.

While not presented here, the unbalance response characteristics of the 0.1 preload bearing design were predicted with very similar accuracy. Predicted rotor critical speed locations for both bearing designs are within 3.3 percent of those measured. Such accuracy gives confidence in the fidelity of the combined stiffness characteristics of the bearing and pedestal models.

Neither bearing design's system model, however, was able to reliably predict the amplitudes in the rotor's critical speed region. The accuracy of the amplitude predictions in the rotor critical speed region is of primary interest for stability purposes. Given a particular unbalance distribution, the amplitudes in this region are strongly governed by the damping ratios, or stability levels, of the two forward and backward sister modes which are interacting. Both designs' minimum clearance condition model achieved the best amplitude correlations in this region, yet, amplitudes were still consistently underpredicted. Underprediction of this region's peak amplitudes suggests the models attribute too much damping, or too high a level of stability, to these modes. Measured response also indicates the degree of anisotropy in the system is slightly less than predicted.

## STABILITY TESTING

For the two different preload bearing designs, stability tests were conducted for correlation with the predicted damped eigenvalues of the first forward and backward modes. These measurements were conducted using the methodology described in [21], which includes the use of blocking excitation and the time domain technique, multiple output backward autoregression (MOBAR), for damping estimation. A type of tuned-sinusoidal method, blocking testing effectively tries to isolate a mode by exciting at its natural frequency in its predominant direction. Once the mode is isolated and its signal-to-noise ratio (SNR) is maximized, the blocking excitation is suddenly turned off. Modal damping is then estimated from the resulting free decay transient using the MOBAR technique.

Using the actuator control system's sinusoid generator, the shaker actuator was used to excite the rotor at the desired blocking frequency and direction. The amplitude of the shaker blocking excitation was limited to keep unfiltered vibration at the journal bearings less than 38 $\mu$m (1.5 mils) p-p for linearity concerns. Upon turning off the shaker's nonsynchronous blocking excitation, the transient decay was recorded for use with the MOBAR technique. Five blocking tests were recorded for each direction of excitation. All ten radial probe channels were used in the estimation process.

### Base Stability

Base stability testing was conducted to examine the stability performance as a function of speed with no actuator cross-coupled stiffness present. Measurements were conducted at eight different speeds from 2,000 rpm to maximum speed. For the 0.3 preload bearing design, Figures 8 and 9 present the results of these measurements along with model prediction bounds for the first forward and backward modes, respectively.

For each measurement, the $\zeta$ and $\omega_d$ estimates are averages of estimates obtained from the five blocking tests. Vertical bars on each measurement indicate the range of estimates obtained, serving as an indicator of the estimate's uncertainty. Typical uncertainties were ±0.2% for damping ratios and ±20 cpm (±0.3 Hz) for damped natural frequencies.

Throughout the speed range, the forward mode's damping measurements correlate the closest with FC-based model's lower bound. An approximately 1% separation is maintained between the measurements and this bound for all measured speeds above 2,000 rpm. The SRC-based model's lower bound overpredicts the damping by a factor of 2.75 at maximum speed. For the measured damping levels from 6,000 rpm to maximum speed, the FC-based model's lower bound predicts a nearly identical relative decline of 3.7 percentage points, whereas the SRC-based model's lower bound gives only a two percentage point decrease.

Figure 8 illustrates the strong overlap between the predicted ranges of the first forward mode $\omega_d$ for the two models. This overlap makes it difficult to quantitatively con-

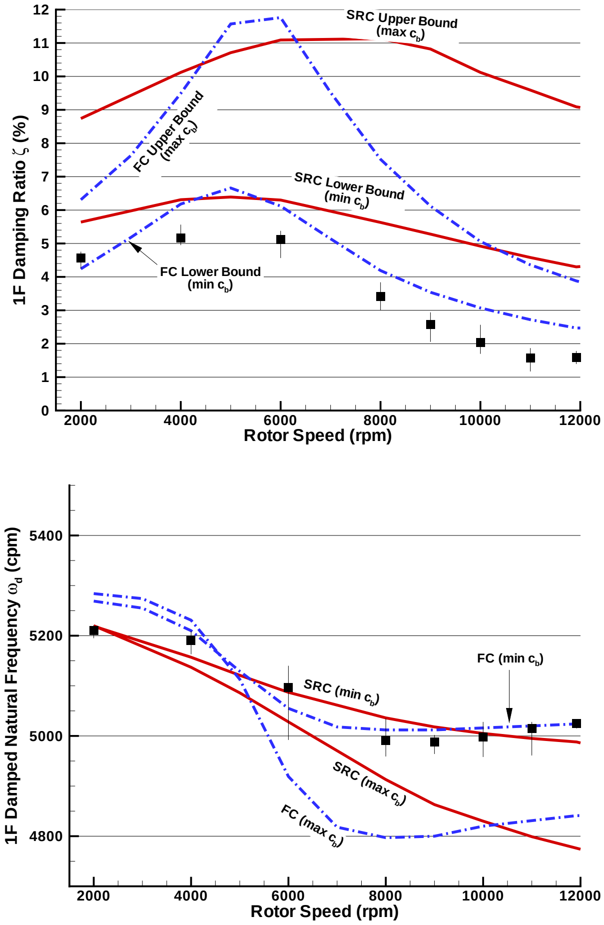



Figure 8: First forward mode base stability measurements versus predictions ($m_p = 0.3$)

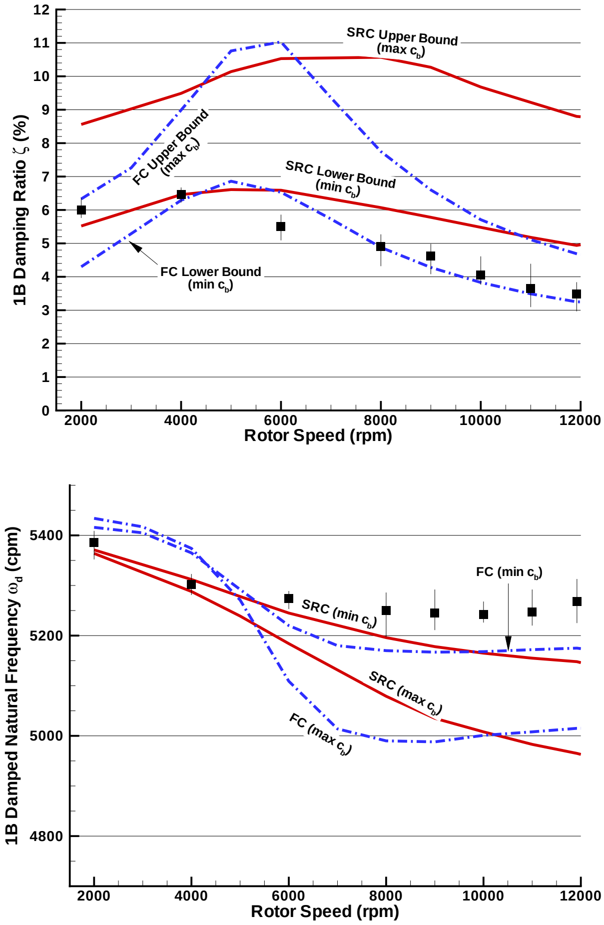



Figure 9: First backward mode base stability measurements versus predictions ($m_p = 0.3$)

clude which bearing representation best correlates with the measurements. Qualitatively, the forward mode's damped natural frequency measurements tend to achieve a better correlation with the FC-based model. The SRC-based model predicts that the mode's frequency will decline almost linearly with speed. However, the FC-based model predicts the majority of the frequency change to occur over a relatively small speed range, from approximately 4,000 rpm to 7,000 rpm. The forward mode's measurements tend to follow this abrupt frequency change behavior as well.

Damping estimates for the backward mode exhibit somewhat better correlation to the model, as shown in Figure 9. In the vicinity of the 5,000 rpm critical speed region, its damping appears somewhat smaller than the lower bounds' predictions, although it is difficult to make a definite quantitative conclusion. The backward mode's damping measurements fall within both models' bounds at 2,000 rpm and 4,000 rpm. For speeds of 8,000 rpm and above, however, the backward mode's measured $\zeta$ agrees extremely well with the lower bound of the FC-based model. Unfortunately, these good correlations are not consistently achieved for the backward mode's $\omega_d$.

For both bearing designs, the FC-based model consistently provided the most accurate prediction of both the first forward and first backward modes' damping ratios. The FC-based model did overpredict the forward mode's damping in both bearing designs. At maximum speed, the FC-based model overpredicts the first forward mode's base damping by approximately one percentage point for both bearing designs. On the other hand, the SRC-based model overpredicts forward mode stability by over twice as much (2.7 percentage points) for the 0.3 preload bearing, and nearly five times as much (4.7 percentage points) for the 0.1 preload bearing design.

## Stability Threshold

A mode's stability threshold, where $\zeta = 0$, provides another datum for correlation with the two different bearing representations. Testing focused on the stability threshold characteristics of the first forward mode as a function of speed and cross-coupled stiffness at Actuator 2. With no shaker excitation and the rotor operating at the desired speed, increasing amounts of cross-coupled stiffness, $q$, were applied using Actuator 2 until the rotor exhibited unstable vibrations. Here, "unstable" is defined in the classical sense, where the mode's vibration increases with time. This testing methodology was performed at the same eight speeds where base stability testing was conducted. An additional test was also performed at 1,000 rpm.

Figure 10 presents the unstable vibrations exhibited during one of the stability threshold tests. Slow, exponential growth is observed, an expected behavior when a mode's damping ratio is just slightly negative. At approximately 3.2 seconds, the cross-coupling was turned off to avoid a damaging level of vibration. The rotor vibration returns to a stable level when the actuator's cross-coupling is removed.

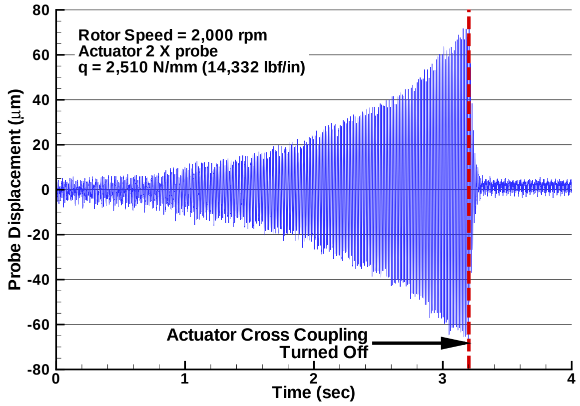



Figure 10: Example waveform showing unstable rotor motion

Figure 11 presents the measurements along with the bounds predicted using the FC and SRC tilting pad bearing representations. Because of the difficulty in measuring where the cross-coupling produces $\zeta = 0$ for the first forward mode, it is necessary to try to bound the threshold cross-coupling between a stable and an unstable $q$ level. Therefore, Figure 11 shows two measured $q$ values at each speed, one for the highest tested $q$ amount where the rotor was stable, and one for the lowest tested $q$ amount where the rotor was unstable. For clarity purposes, only the uncertainties for the unstable $q$ amounts are shown.

In Figure 11, the threshold $q$ measurements consistently fall significantly under both models' predicted lower bounds. The exceptions occur at 1,000 rpm and 2,000 rpm which agree with the FC-based model's lower bound. Throughout the speed range, the forward mode's threshold $q$ measurements correlate the closest with the FC-based model's lower bound. At maximum speed, the SRC-based model's lower bound overpredicts the amount of threshold cross-coupling by 325 percent, whereas the FC-based model's lower bound overpredicts by 225 percent.

The two model's predicted ranges of the first forward mode $\omega_d$ overlap substantially throughout the experimental range, particularly at speeds greater than 5,000 rpm. This overlapping makes it impossible to quantitatively conclude which bearing representation best correlates with the measurements. As observed in the base stability results, the forward mode's threshold $\omega_d$ measurements achieve a better qualitative correlation with the FC-based model. The SRC-based model predicts that the mode's frequency will decline almost linearly with speed. However, like the measurements, the FC-based model predicts the majority of the frequency change to occur over a relatively small speed range. Furthermore, the increase of the mode's frequency for speeds above 8,000 rpm is another characteristic predicted by the FC-based model, although not to the same degree as that measured.

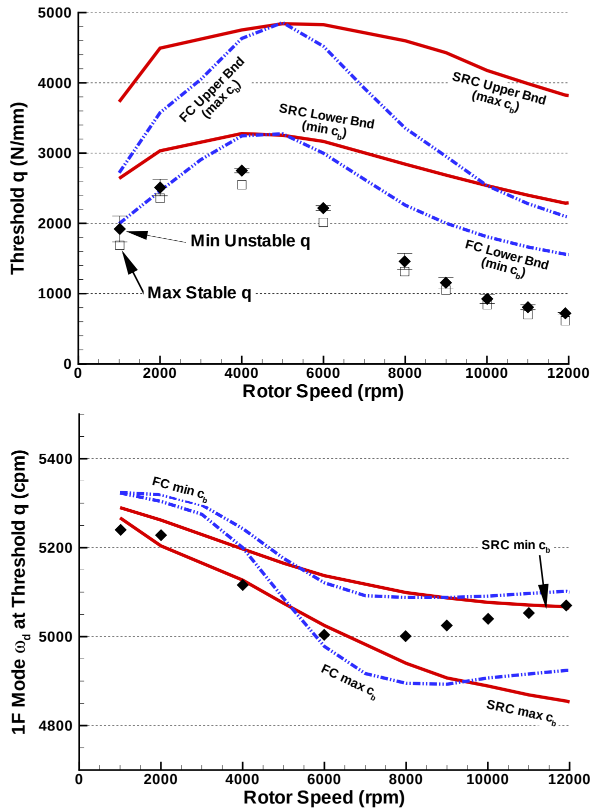



Figure 11: First forward mode stability threshold measurements versus predictions ($m_p = 0.3$)

## Stability Sensitivity

Our purpose for this type of testing is to examine the apparatus's stability as a function of cross-coupled stiffness, a relationship commonly of interest to industry. With the speed held constant, the first forward and backward modes' stability sensitivities were measured between the base and threshold results just described. Damped eigenvalues of the two modes were measured through blocking with the shaker, while simultaneously applying cross-coupling with Actuator 2.

Figures 12 and 13 present the first forward and backward modes' stability sensitivity results for the two speeds tested. Absent from these plots is the upper bound for the SRC-based model, since it is well above the damping levels of interest. For the 9,000 rpm predictions in Figure 12, some minor overlapping exists between the two models' $\zeta$ bounds. This is not much of a factor since the measured damping values fall below both models' lower bounds. Greater separation between the two models' $\zeta$ bounds occurs at maximum speed (Figure 13), especially for the first forward mode.

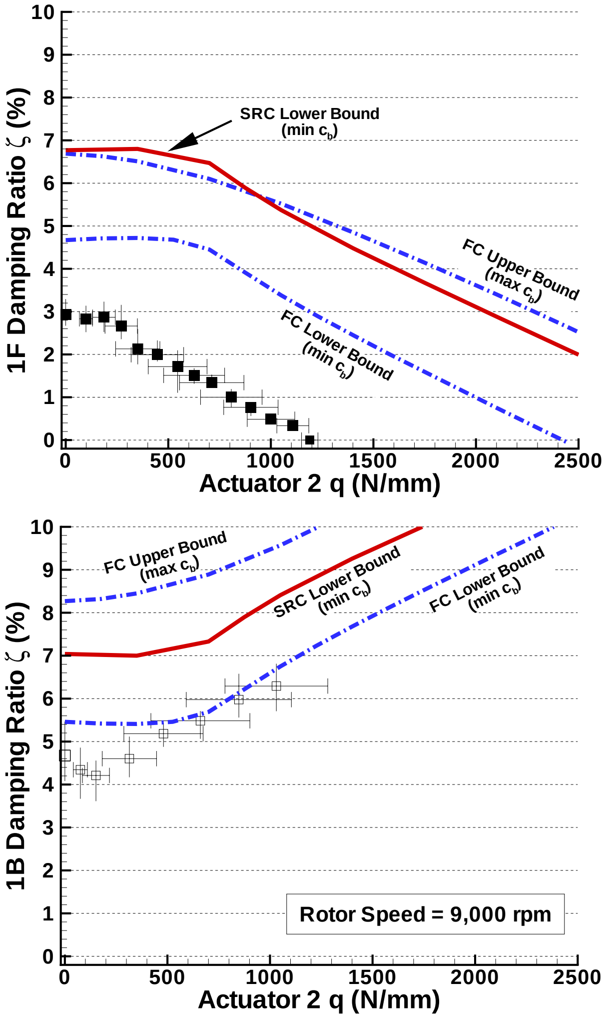



Figure 12: First forward and backward mode stability sensitivity measurements versus predictions at 9,000 rpm ($m_p$=0.1)

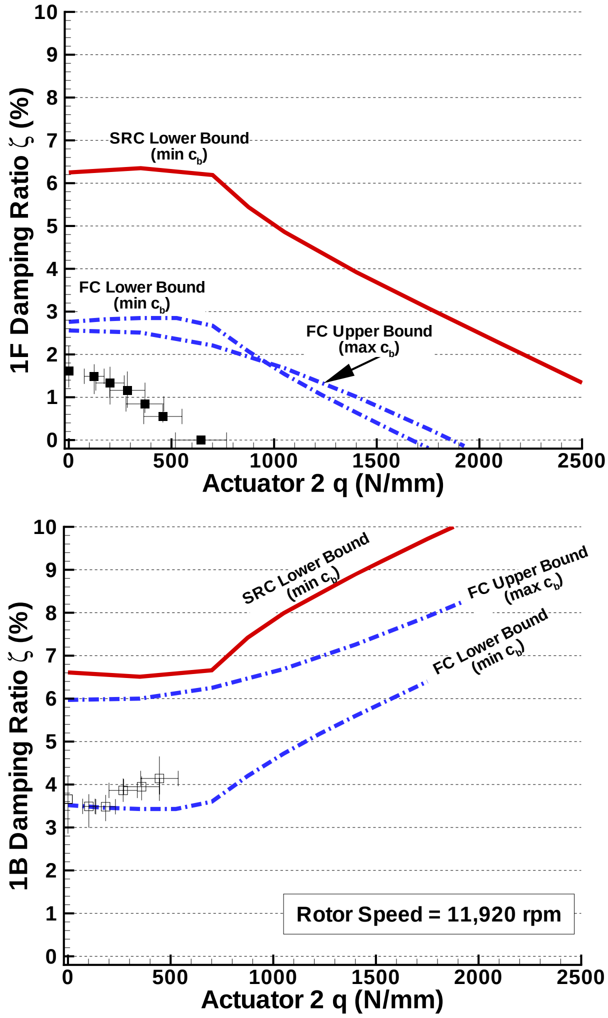



Figure 13: First forward and backward mode stability sensitivity measurements versus predictions at 11,920 rpm ($m_p$=0.1)

As with base stability, the FC-based model provides the most accurate prediction for both modes while still overpredicting the first forward mode's damping. In comparison, at maximum speed, the SRC-based model's lower bound predicts more than twice as much damping for the forward mode than the highest FC bound.

Qualitatively, neither bearing representation completely captures the $\zeta$ sensitivity trends. Both models forecast a large plateau region where the modes' damping is insensitive to additional cross-coupling. However, the $\zeta$ measurements show a very small plateau region creating an almost linear dependence on cross-coupling. This indicates the models are overpredicting the degree of anisotropy in the bearing system, a characteristic also observed in the unbalance response measurements.

## CONCLUSIONS

This investigation has provided a comprehensive set of correlations (two bearing designs, two modes, multiple speeds and cross-coupling levels) between measured damped eigenvalues and those predicted by the two tilting pad bearing representations. The differences between predicted lower mode damping as well as between predicted stability thresholds for the FC and SRC models were substantial, indicating at least for this machine configuration that a modal truncation argument for use of SRC models is not valid.

Both representations overpredicted all of the stability measures that were examined. However, the FC-based model consistently achieved superior quantitative and qualitative agreement with the base, threshold and sensitivity stability measurements. While it is not possible to argue that this result validates the FC model, it does tend to refute the heuristic argument that SRC is *better* at predicting stability thresholds because of dominance of synchronous vibration at the onset of instability.

It remains unclear exactly why the measured stability levels were consistently lower than either prediction. Because the rotor model was extensively validated using free–free mode shapes and frequencies and because the pedestal was derived directly from modal testing, we have relatively high confidence in these models. In contrast, the bearing models cannot be corroborated by direct component level testing, leading to a lower level of confidence. From this we conclude that these results highlight the need for continued improvements in fluid film bearing modeling.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