SURVEYING TILTING PAD JOURNAL BEARING AND GAS LABYRINTH SEAL COEFFICIENTS AND THEIR EFFECT ON ROTOR STABILITY

by

John A. Kocur, Jr.

Machinery Engineer

Plant Engineering Division

ExxonMobil Research & Engineering

Fairfax, Virginia

John C. Nicholas

Owner and President

Rotating Machinery Technology, Inc.

Wellsville, New York

and

Chester C. Lee

Manager, Rotordynamics Design and Test

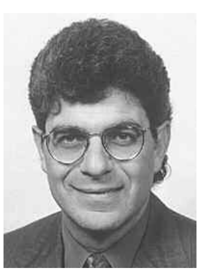
Gas Compressor Engineering Department

Solar Turbines, Inc.

San Diego, California

John A. Kocur, Jr., is a Machinery Engineer in the Plant Engineering Division at ExxonMobil Research & Engineering, in Fairfax, Virginia. He has worked in the turbomachinery field for 20 years. He provides support to the downstream business within ExxonMobil with expertise on vibrations, rotor/aerodynamics, and health monitoring of rotating equipment. Prior to joining EMRE, he held the position of Manager of Product Engineering and Testing at Siemens Demag Delaval Turbomachinery. There Dr. Kocur directed the development, research, engineering, and testing of compressor and steam turbine product lines.

Dr. Kocur received his BSME (1978), MSME (1982), and Ph.D. (1991) from the University of Virginia and an MBA (1981) from Tulane University. He has authored papers on rotor instability and bearing dynamics, lectured on hydrostatic bearings, has been a committee chairman for NASA Lewis, and is a member of ASME. Currently, he holds postions of API 617 Vice-Chair and API 684 Co-Chair.



John C. Nicholas is owner and President of Rotating Machinery Technology, Incorporated, in Wellsville, New York. His company repairs and services turbomachinery, and manufactures bearings and seals. Dr. Nicholas has worked in the turbomachinery industry for 30 years in the rotor and bearing dynamics areas, including five years at Ingersoll-Rand and five years as the Supervisor of the Rotordynamics Group at the Steam Turbine Division of Dresser-Rand.

Dr. Nicholas, a member of ASME, STLE, and the Vibration Institute, has authored over 40 technical papers concerning rotordynamics and tilting pad journal bearing design and application. He received his B.S. degree from the University of Pittsburgh (Mechanical Engineering, 1968) and his Ph.D. degree from the University of Virginia (1977) in rotor and bearing dynamics. Dr. Nicholas holds several patents including one for a spray-bar blocker design for tilting pad journal bearings and another concerning bypass cooling technology for journal and thrust bearings.

Chester C. Lee is the Group Manager of Rotordynamics Design and Test in the Gas Compressor Engineering Department of Solar Turbines Inc., in San Diego, California. He has been with this group since he joined Solar 17 years ago. His major responsibility is to support rotordynamic analysis in design, manufacturing, testing, and field operation.

Dr. Lee received his Ph. D. degree (Mechanical Engineering) from the University of Virginia, specializing in rotating machinery. Before joining Solar, he worked for Mechanical Technology Inc., in Latham, New York, on various rotating equipment.

ABSTRACT

The evaluation of rotor system stability, which has become an essential part of rotordynamic analyses and rotating machinery design, relies heavily on the bearing and seal dynamic coefficient modeling to obtain an accurate prediction of the turbomachinery behavior. Lacking experimental validation, analytical predictions can be widely varied and even divergent as more complex procedures and models are created. To measure the variability of bearing and labyrinth seal coefficient predictions, a survey of 60 turbomachinery users, manufacturers, consultants, and academicians was conducted under the auspices of the American Petroleum Institute (API). Coefficients received from the respondents were incorporated into a common rotordynamic model to determine the impact on the predicted rotor stability. In addition, several of the most popular analytical codes for the prediction of tilt pad journal bearings are compared. Starting with an iso-viscous prediction and proceeding through more complicated thermal and structural deformation solutions, the authors compare the variability and divergent nature of these codes. The measured variability of the data collected clearly illustrates the need for the resolution

of fundamental bearing issues (i.e., synchronously versus nonsynchronously reduced bearing coefficients) and labyrinth seal predictions based on repeatable experimental data.

INTRODUCTION

Centrifugal compressor instability remains a major concern causing reduced unit availability and project commissioning delays, leading to lost revenue for both users and vendors of the equipment. The evaluation of rotor stability has become an essential part of rotordynamic analyses and rotating machinery design. In the latest edition of American Petroleum Institute (API) 617, Seventh Edition (2003), specifications for performing stability analyses of centrifugal compressors were added for the first time. As a continued effort to improve API specifications, a study of the current state of bearing and labyrinth seal dynamic coefficient predictions is undertaken.

Tilting pad bearing analysis started over 40 years ago spawned by work by Lund (1964). API 684 (2005), Section 2.5.4, and Nicholas (2003) provide an excellent historical perspective of the development of tilt pad bearing analysis. While work has advanced to include flexibility of the pad and pivot, nonsynchronous behavior, thermal effects of the fluid, and pad deformation, surprisingly little experimental work has been published on the dynamic behavior of tilting pad bearings. A series of papers was published in the 1990s on the measurement of coefficients concluding with Wygant, et al. (1999). However, bearing operating speeds were not representative of the high speed compressor applications that typically suffer stability problems. In addition, the use of synchronous versus nonsynchronous coefficients in stability analysis was not addressed. Cloud (2006) has investigated this argument from a rotor system standpoint. While the published research provides valuable insight into the appropriate coefficients to use in a stability analysis, it still relies on the assumed accuracy of the bearing prediction methods employed.

Gas labyrinth seal research seems to follow a different path. While significant efforts to understand the dynamic behavior of these seals has been undertaken (Iwatsubo, et al., 1982; Childs and Scharrer, 1986; Kirk, 1990), the validity of these approaches is often called into question by comparison to experimental work, summarized by Childs (1993). Poor matching of the data produced by analytical methods has raised questions regarding continued modeling efforts of labyrinth seals from the single volume approach of Childs and Scharrer (1986), to the three volume approach of Nordmann and Weiser (1990). Current experimental efforts seem to indicate the more complicated analytic models are not providing better results. For the most part, these discussions focus on the tangential coefficients of the labyrinth seal. Radial coefficient predictions suffer such a wide variation to experimental data as to be ignored in their use.

Testing of short labyrinth seals has also proved to be a challenge. Short seals with smaller pressure drops produce small forces that have resulted in high uncertainty levels during testing. While the individual impeller eye seal may not provide the same force magnitude compared to the balance piston (BP), the multiple stages usually result in a total force approaching the balance piston. In addition, the importance of data taken near application conditions was emphasized by Childs and Ramsey (1991), Elrod, et al. (1995), and Wagner and Steff (1996). The later, while extending test conditions, held much of the data proprietary.

The main objective of this paper is to determine the magnitude of variations in tilting pad journal bearing and gas labyrinth seal dynamic coefficient predictions. This is approached in two fashions: first, as a survey sent to industry wide participants using a common data set; and second, by comparing coefficients obtained from several widely used tilting pad bearing analysis codes. The survey is intended to measure the existing variations in the industry when supplied with bearing and seal dimensions from an unstable compressor. The code comparison effort attempts to identify the source of the tilting pad bearing coefficient variations as found in the survey.

At the conclusion of this work, answers to the following questions are sought:

- Is an API Level I type analysis still justified given the current state of analysis technology?
- Has sufficient experimental work been performed to act as the basis for comparison of the existing analysis packages?
- Do the available analysis codes give reasonable agreement when supplied with similar input?
- Why do different tiling pad journal bearing codes provide different bearing coefficients?
- Do the different journal bearing codes converge to identical results after stripping out all of the parameters that cause the coefficient variations?

API SURVEY

A survey was conducted under the auspices of the American Petroleum Institute with the intent to determine the conformity level of bearing and labyrinth seal dynamic behavior predictions. Direct comparison of the coefficients and impact on rotor stability are the two methods used to determine the variability of the predicted coefficients. The working hypothesis of the survey can be stated as follows: *"To date, there remain significant differences across the industry in the prediction of dynamic coefficients for fluid film tilt pad bearings and labyrinth seals."*

The survey focuses on the prediction of tilt pad bearings and impeller eye and balance piston labyrinth seals as related to centrifugal compressor applications. Consequently, the goals of the survey are threefold:

- First, to improve the recently introduced specifications in API 617 (2003) and 684 (2005) regarding rotordynamic stability. This can be realized by examining the necessity and/or appropriateness of the steps within the stability methodology.
- Second, highlight and communicate the disparity of the predictions within the industry. Hopefully this can be used as supporting evidence for the continued need of research funding from industry and government agencies.
- Finally, determine the need for experimental data to act as the gold standard.

While some groups have started measuring tilt pad bearing coefficients, there remains a lack of verifiable data that can be used universally to determine the accuracy of analytical methods for either bearings or labyrinth seals.

The survey was proposed and supported by API. Information relevant to the prediction of the tilt pad bearing and labyrinth seals was sent to 60 respondents directly from API. The respondents included industrial users and vendors of rotating equipment (both at a component and turbomachinery level), consultants, and educators. While the list of respondents was known (and developed) by the authors of this paper, the responses were kept anonymous by API. From the 60 requests for information sent, 16 respondents supplied bearing coefficients, 12 of which also sent seal coefficients, and several responded that they did not have the ability to calculate the requested information. Approximately one-third of the group replied in some fashion.

SURVEY INFORMATION

As preferred by the API Subcommittee on Mechanical Equipment (SOME), an actual compressor application was sought as the basis for the survey. The compressor used in the survey represents a multistage nitrogen compressor intentionally designed by the vendor to be unstable for stability testing. Extensive testing by the vendor determined the instability onset speed for several operating conditions. One such condition was selected for the survey representing the highest ΔP across the compressor obtained during the testing program.

To minimize efforts requested from the respondents and to eliminate modeling differences between respondents, a common rotor model and damped eigenvalue solution algorithm (Vázquez and Barrett, 1998; Vázquez, et al., 2001), was proposed for the study. By eliminating these differences, the variations in the predicted damped eigenvalues can be attributed to the supplied bearing and seal coefficients. Respondents were asked to supply bearing coefficients, the first and last impeller eye seal coefficients, and balance piston coefficients for a single operating point representing the onset of instability. Each was instructed to treat the information supplied as they would a new design compressor or one with a known stability problem. The appropriate level of analysis to use (including thermal analysis, pivot stiffness, etc.) was left to the respondent to determine.

As noted, a single rotor model was employed throughout the study. Figures 1 and 2 display the mass (top) and stiffness (bottom) and 3D rotordynamic model of the survey compressor. The compressor is a five-stage tie-bolt design compressor with nitrogen as the working fluid. General information regarding the compressor and operating point in question can be found in Table 1. The operating point selected for the survey consists of an operating speed of 21,662 rpm with the compressor producing 209 bara at a pressure ratio of 2.53 in N_2.

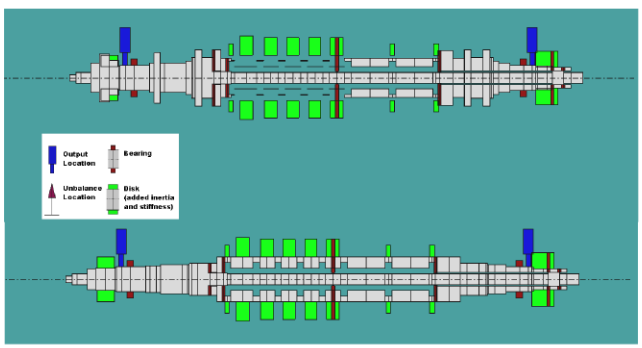


Figure 1. Mass and Stiffness Model of the Survey Compressor.

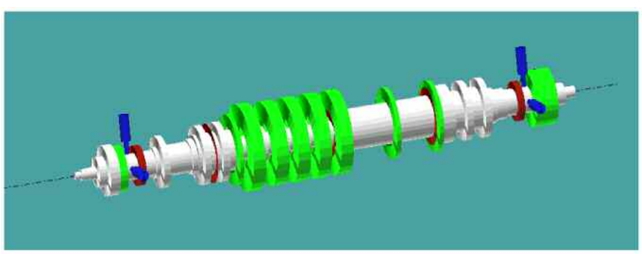


Figure 2. 3D Model of the Survey Compressor.

Table 1. General Information, API Survey Compressor.

Project	API Survey	Unit	Compressor	Number of Impellers	5
Service	Nitrogen	Speed, rpm	21,662	Molecular Weight	28.01
P_1, bara	82.81	P_2, bara	209.3	Temp In - Temp Out	22.2 - 174.4 °C
Rotor Weight, kg	68	Bearing Span, mm	875.2		

The tilt pad bearings consisted of five pads with the gravity load located between the pad pivots. The steel backed babbitted pad arc length was 60 degrees. Bearing clearance ratio was on the order of 1.5 mils per inch. Other details of the bearing can be found in Table 2. Survey information also included oil type, oil inlet temperature and flow, oil viscosity at referenced temperatures, and pivot stiffness.

Table 2. Bearing Information, API Survey Compressor.

Type:	Tilting Pad	No. of Pads:	5	LBP	Pad Angle, deg:	60

	Total Load (N)	Unit Loading (N/mm²)	Length (mm)	Dia. (mm)	L/D	Preload	Diametrical Clearance (μm)		Clearance Ratio C_b / Diameter (0/00)
							C_p	C_b	
Drive-end	347	0.212	28.6	57.2	0.5	0.222	114	89	1.56
Thrust-end	320	0.196	28.6	57.2	0.5	0.222	114	89	1.56

Bearing Operating Conditions

Preload Range			Oil Type	Normal Oil Temperature	Pivot Location (Offset)
-	0.222	-	ISO VG 32	60 °C	Centered

Notes: LOP = Load On Pad.
C_b = Assembled Clearance.
C_p = Machined Clearance.

Labyrinth seals were described as tooth-on-rotor for all impeller eyes and the balance piston. Geometry dimensions included the tooth spacing, tooth tip width, tooth height, operating clearance, and diameter as shown in Figure 3. The impeller eye seals consisted of four tooth seals at a diameter of 133.35 mm (5.25 inches). The balance piston incorporated 11 teeth at a diameter of 127 mm (5.0 inches). The gas preswirl was also given due to the complexity of information needed to accurately compute this parameter. Preswirl was set at 70 percent for all seals. Only coefficients for the first and fifth (last) impeller eye seals were requested (also to minimize efforts). A simple linear variation was used to derive the eye seal coefficients for the middle three impellers.

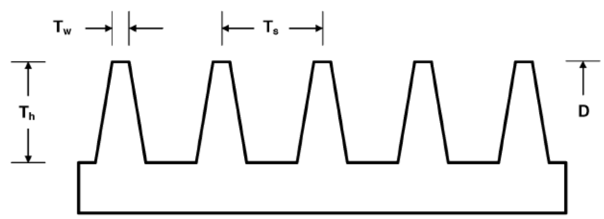


Figure 3. Dimensions Supplied in Survey for the Impeller Eye and Balance Piston Labyrinth Seals.

Gas conditions for each impeller eye seal were supplied in terms of suction and discharge pressure and temperature for the stage and for the compressor overall. Recognizing that some respondents chose methods similar to the modified Alford's method employed in the API 617, Seventh Edition (2003) Level I stability analysis, impeller diameter and discharge tip width and diffuser minimum width were also supplied in the survey. To further eliminate unintended variations, the gas properties of nitrogen were supplied at key operating points within the compressor. Figure 4 displays a typical set of data for nitrogen.

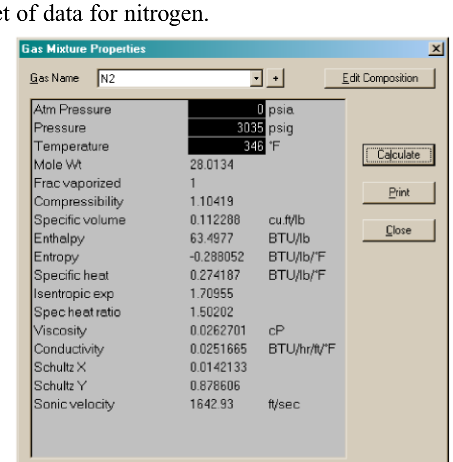


Figure 4. Typical Nitrogen Gas Properties Supplied in Survey.

Finally, an approximate location of the first critical speed was supplied at 6700 cpm. This was intended to be used by those respondents supplying bearing and seal coefficients for the whirl frequency rather than synchronous frequency. Currently, API states that synchronous reduction should be used in the Level I analysis for the bearing coefficients. It should be noted however that nearly half of the participants in that effort to develop the API stability specifications did use nonsynchronous reduction. No such statement is made for the calculation of labyrinth seal coefficients.

SURVEY COEFFICIENT RESULTS

Sixteen sets of bearing data were received from the respondents. The data were normalized using the minimum stiffness supplied for the principle stiffness in the X-direction. The same respondent's principle damping in the X-direction was used to normalize the damping coefficients. Coefficients are presented on a graph plotting K_{xx} versus K_{yy} and C_{xx} versus C_{yy}. This display was selected to show variations in the coefficients due to loading. However, the survey compressor's bearing was very lightly loaded producing almost identical coefficients in the horizontal and vertical directions. Figure 5 displays the variations in stiffness from the survey. Considering the effects of reduction, almost an order of magnitude of difference exists in the supplied coefficients. Using the information supplied by respondent #8, frequency reduced coefficients are assumed to occupy the higher range of the normalized coefficients. Figure 6 plots the variations in damping. As with the stiffness, the variation in damping from the smallest to largest is also nearly an order of magnitude. While using nonsynchronous reduction increases the bearing stiffness, damping is predicted to decrease with the reduction (Figure 6).

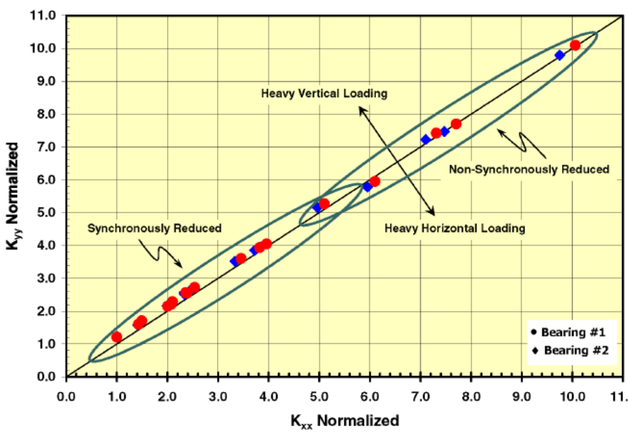


Figure 5. Normalized Principle Stiffness Coefficients.

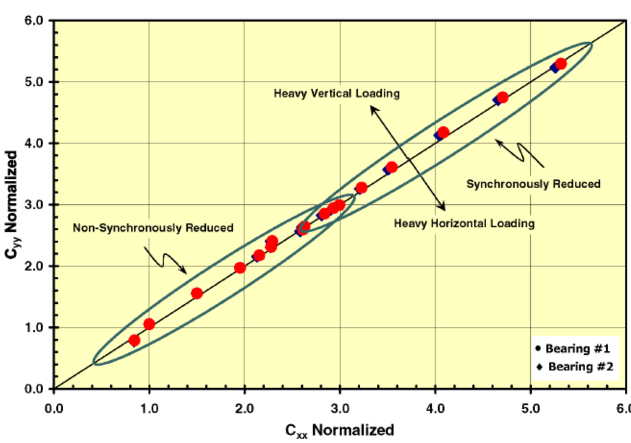


Figure 6. Normalized Principle Damping Coefficients.

It should be noted that the approximate location of the first critical speed was erroneously supplied at 6700 cpm rather than the intended 9700 cpm. Unfortunately, this error was identified after the information was received from the respondents. While this information has no effect on the respondents' using synchronous values for the bearing and labyrinth seal coefficients, it will add some unintended variation to those supplying frequency dependent coefficients.

The frequency impact and reduction schemes are well known (Parsell, et al., 1983, and Barrett, et al., 1988). Examining the effect of this parameter on tilt pad bearings, a set of coefficients was predicted using a solution scheme developed by Branagan (1988). Comparing the frequency reduced coefficients using 6700 cpm, 9700 cpm, and 21,662 rpm, the change in the principle stiffness and damping coefficients are shown in Table 3. As noted, reducing the full coefficient matrix to the lower frequency, increases the stiffness and reduces the damping an additional 10 percent from the synchronously reduced coefficients when compared to the reduction to the intended frequency of 9700 cpm.

Table 3. Predicted Impact of Frequency on Principle Bearing Coefficients.

Frequency	Normalized Principle Stiffness	Normalized Principle Damping
21,662 rpm	1.0	1.0
9700 cpm	1.26	0.68
6700 cpm	1.35	0.59

Figure 7 illustrates the expected change in respondent #8's bearing coefficients if a frequency of 9700 cpm is used in the reduction process instead of 6700 cpm. The difference between synchronous and nonsynchronous reduction is decreased by ≈25 percent assuming the same variation as shown in Table 3. The impact on rotor stability is shown later.

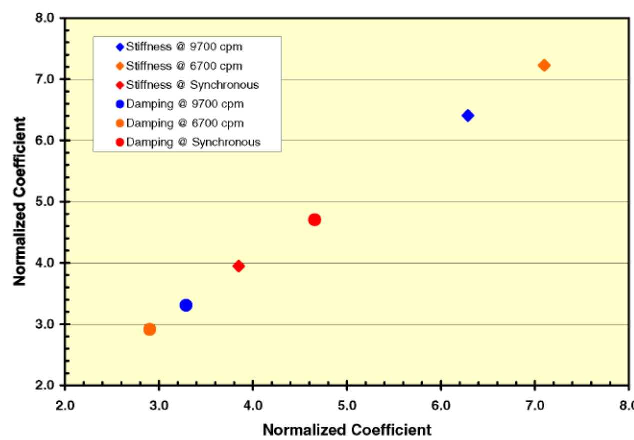


Figure 7. Approximate Impact of Frequency on Bearing Coefficients from Respondent #8.

The coefficient variations for the labyrinth seals are presented on graphs of the tangential force components, K_{xy} versus C_{xx}. All respondents supplied symmetric coefficients for the seals. As previously mentioned, coefficients for impeller #1 and #5 eye seals and the balance piston were requested. The variations for these seals are plotted in Figures 8, 9, and 10. For the most part, the relative variations in K_{xy} are matched by the variations in C_{xx}. One exception is noted for a respondent supplying Alford type forces for the labyrinth seals.

The bulk of the respondents' impeller eye seal coefficients falls in the normalized range from 5 to 20. One respondent was less (at

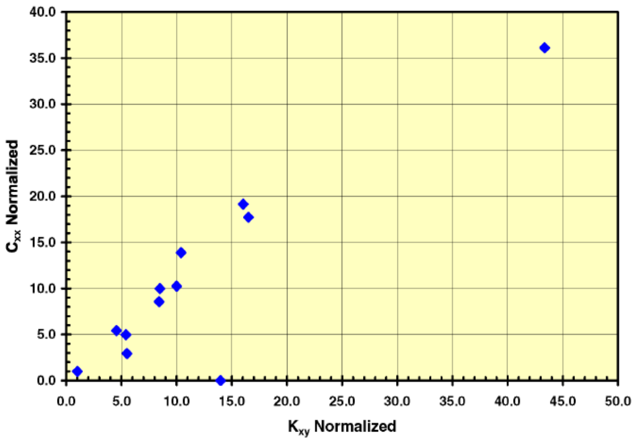


Figure 8. Normalized Seal Coefficients for Impeller #1 Eye Labyrinth.

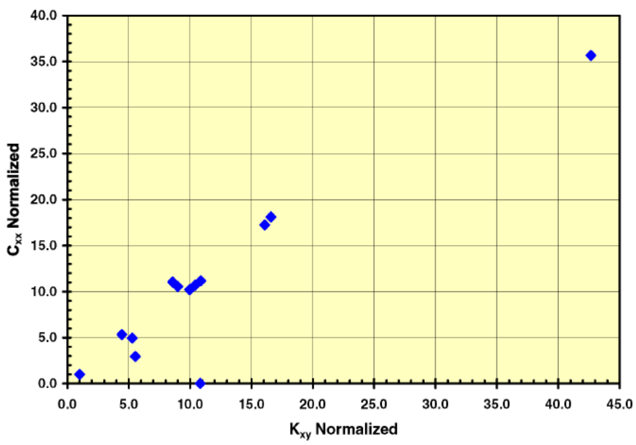


Figure 9. Normalized Seal Coefficients for Impeller #5 Eye Labyrinth.

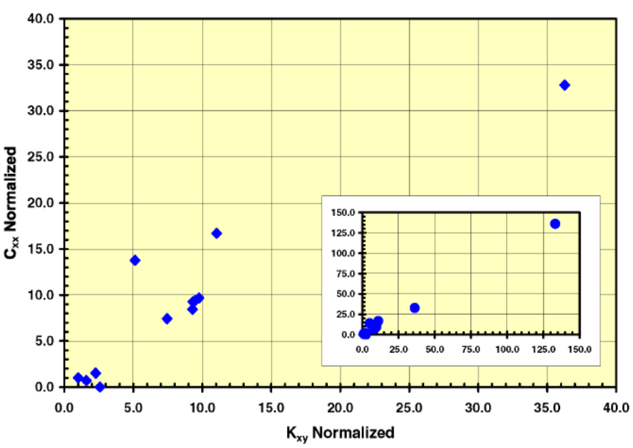


Figure 10. Normalized Seal Coefficients for Balance Piston Labyrinth (Expanded Range Shown in Inset).

1.0) and one higher (at 42.) For the balance piston, the majority fell in the normalized range from 1 to 12 (an order of magnitude) with one at 37 and another at 137 (greater than two orders of magnitude.) (Confirmation of the values supplied by several respondents was requested but not received at the time of this paper.) Since the variations in cross-coupled stiffness and principle damping were nearly equal for all coefficients, it comes as no surprise that the destabilizing force, defined by Equation (1), also shows a similar range in normalized values, Figure 11.

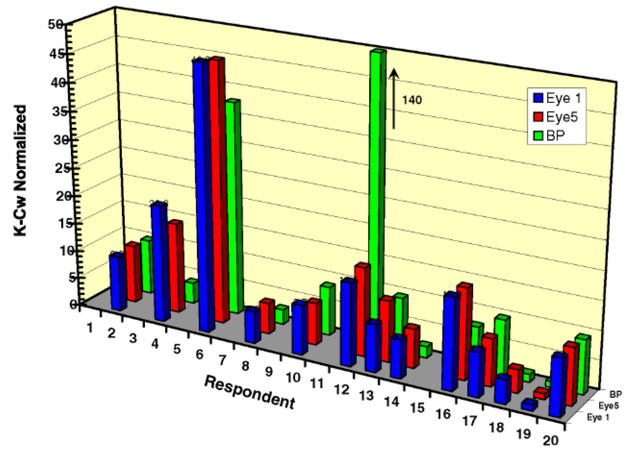


Figure 11. Normalized Destabilizing Force for the Balance Piston Labyrinth.

Using a solution algorithm developed by Kirk (1990), the impact of the whirl frequency ratio (WFR) on the BP coefficients was studied. For the conditions specified, balance piston coefficients were calculated for 0.31 (6700 cpm), 0.45 (9700 cpm), and 1.0 (synchronous) whirl frequency ratios. Both cross-coupled stiffness and direct damping were smaller for the lower whirl frequency ratio. If the destabilizing force is defined as:

$$Q_a = k - C\omega_{wf} \qquad \textbf{(1)}$$

Then calculating the Q_a using the tangential coefficients derived from the three whirl frequency ratios produces the results shown in Table 4. As noted, Q_a is only 10 percent greater for the higher subsynchronous WFR. The impact on stability of using 6700 cpm rather than 9700 cpm would be a secondary effect. Also notice the size of the destabilizing force if a WFR of 1.0 is used to predict the seal coefficients. This represents a 2× increase over the coefficients derived from a WFR of 0.45.

Table 4. Impact of WFR on the Destabilizing Force of the Balance Piston.

WFR Used in Calculating Coefficients	Q_a (lbf/in)
0.31	5538
0.45	5915
1.00	13653

STABILITY RESULTS

Coefficients supplied for the bearings and seals were incorporated into a common model of the compressor rotor. In addition, a single complex eigenvalue solution algorithm was used for each model. The bearings were represented by the principle terms only. Cross-coupling terms were consistently three orders of magnitude smaller than the principle terms and could thus be safely ignored. Labyrinth seal behavior was modeled using only the tangential terms supplied by the respondents due to the variations in the radial terms. Figure 12 plots the variation in the radial terms of impeller #1 eye seal. The principle stiffness varies by two orders of absolute magnitude and the cross-coupled damping by nearly three orders of magnitude, both ranging from positive to negative values. In addition, the relative size of each was large enough to significantly change the predicted frequency of the first forward damped eigenvalue.

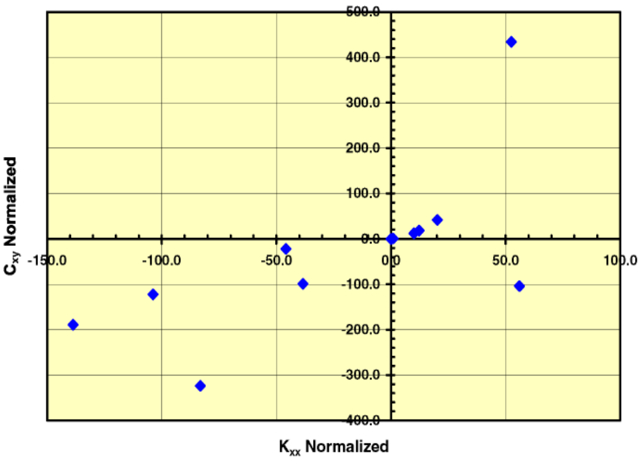


Figure 12. Variations in Radial Coefficients of Impeller #1 Eye Seal.

The predicted compressor stability considering a rotor/bearings only model with the respondents' coefficients is displayed in Figure 13. On the plot, the coefficients supplied by respondent #8 are highlighted for the two reduction schemes. A third point (in red) is plotted for an anticipated reduction to 9700 cpm. The impact on the base stability level of the rotor is secondary even with the 25 percent reduction in coefficient difference noted earlier in the paper. As is expected given the impact on the bearing coefficients, synchronous reduction of the bearing coefficients produces a lower first natural frequency (due to the softer stiffness values) and a higher logarithmic decrement (log dec) (due to the higher predicted damping.) The general trend can be seen on the plot as higher log dec values are produced at lower natural frequencies. This illustrates both the effect of reduction schemes and the increase in relative shaft-to-bearing stiffness making the bearing damping more effective in stabilizing the first mode.

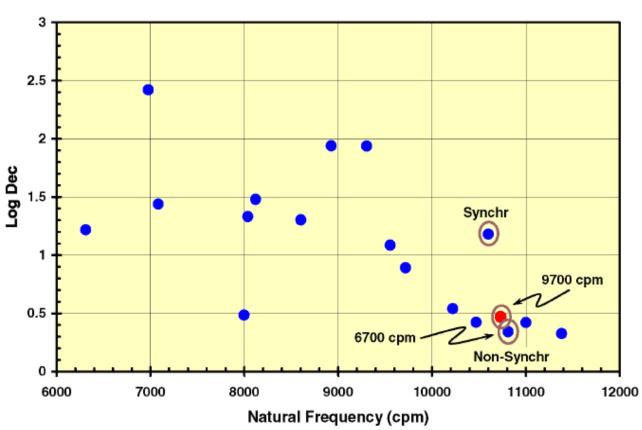


Figure 13. Predicted Compressor Stability with Rotor/Bearings Only.

Adding the labyrinth seals to the stability model impacts the predicted log dec as shown in Figure 14. As before, the two reduction schemes supplied by respondent #8 are highlighted with their seal coefficients. A third point is added that includes both the bearing and seals coefficients altered representing a 9700 cpm location of the first damped forward mode. The change in log dec remains secondary to the overall variations in predicted rotor stability. In addition, a second respondent's seal coefficients were changed by the amount indicated in Table 4. The 10 percent increase in destabilizing force was added to respondent #13's coefficients and the change indicated by the arrow in Figure 14.

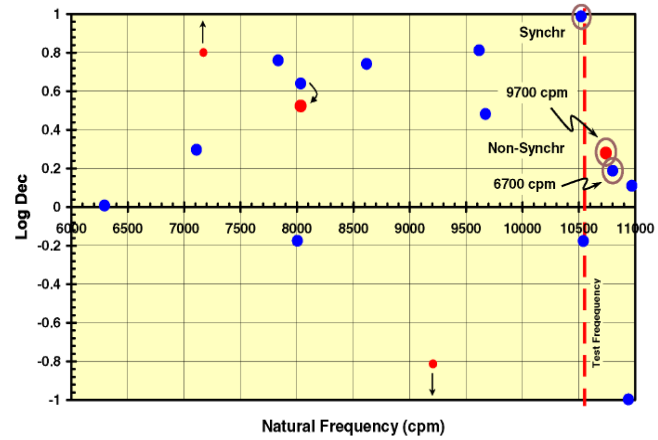


Figure 14. Predicted Compressor Stability with Rotor/Bearings/ Labyrinth Seals.

Two calculated points are not presented in the plotted range in Figure 14, one at approximately a log dec of 1.8 and the other at a log dec exceeding −2.0. These are omitted to more clearly define the influence of respondent #8's coefficients and due to the uncertainty of the validity of these points.

An engineer analyzing this compressor at the onset of the design with the supplied coefficients would conclude a stable compressor design with a WFR less than tested in the majority of the cases. The subsynchronous frequency of the first mode as witnessed on the test stand was 10,700 cpm. This is plotted in Figure 14 against the predicted results. While the change is slight from 9700 cpm to 10,700 cpm, the change for most results would be in the wrong direction, to the left and up in Figure 14. Finally, the impact of the labyrinth seals on the rotor stability (log dec) is shown in Figure 15. The distribution seems skewed to the higher side ranging between a −0.2 to −1.5 change in log dec.

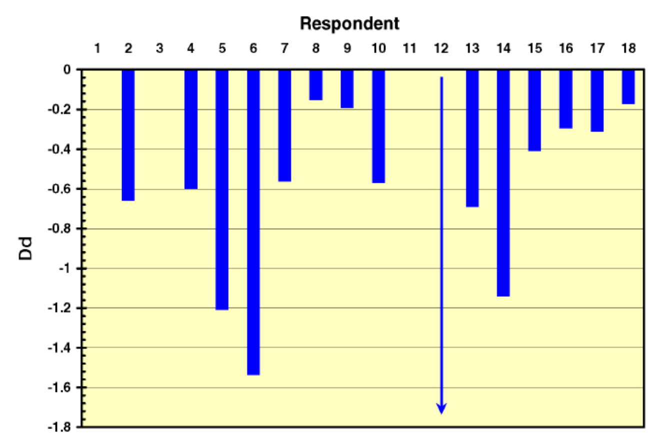


Figure 15. Labyrinth Seal Impact on Predicted Log Dec.

TILTING PAD JOURNAL BEARING STIFFNESS AND DAMPING VARIANCE

From the previous sections, it is clear that there is a relatively wide variation in the stiffness and damping (K and C) properties provided by various computer codes for this stability study. In an attempt to investigate the source of this K and C variation, five different tilting pad journal bearing codes are utilized. References for the five codes are Nicholas, et al. (1979), Branagan (1988), San Andres (1995), He (2003), and Chen and Gunter (2005).

The analysts were given the instructions to run the various codes for the tilting pad bearing from Table 5 as if they were about to perform an API stability analysis. Additional instructions included utilizing whatever thermal solution that was available within the code and to use a pad pivot stiffness of K_{piv} = 8.0 × 10^5 lbf/in. Finally, synchronous coefficients were requested.

Table 5. Stability Results Using Variable Viscosity Derived Coefficients from Four Bearing Codes at 21,662 RPM.

Code #	Sync or Non-Sync	K_{piv} (lbf/in)	T_{max} (°F)	K_{xx} (lbf/in)	C_{xx} (lbf-s/in)	N_d (cpm)	Log Dec
1	Sync	800,000	173	255,000	179	9,628	0.595
2	Sync	Infinite	176	293,000	283	10,550	0.796
2	Non-Sync	Infinite	176	542,700	176	10,810	0.128
3	Sync	800,000	189	188,100	141	8,736	0.672
4	Sync	800,000	221	189,000	128	8,694	0.552

The results are illustrated in Figures 16, 17, 18, and 19 for K_{xx}, K_{yy}, C_{xx}, and C_{yy}, respectively. Note that only codes #1 through #4 are shown as code #5 did not have a variable viscosity capability (or the operator did not know how to use the capability). Additionally, code #2 had to be run with infinite pivot stiffness. Clearly, some K and C variation is evident.

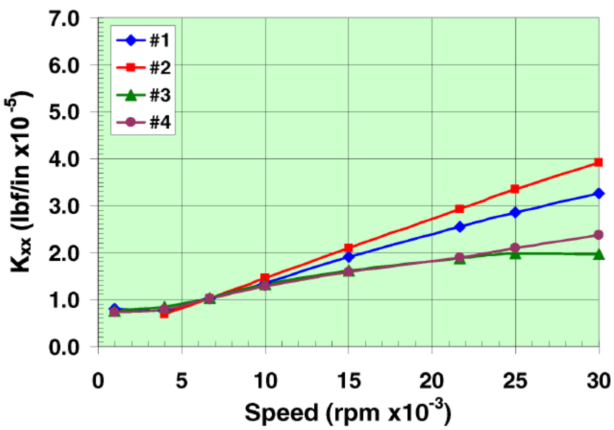


Figure 16. K_{xx} - Variable Viscosity, K_{piv} = 8.0×10^5 lbf/in.

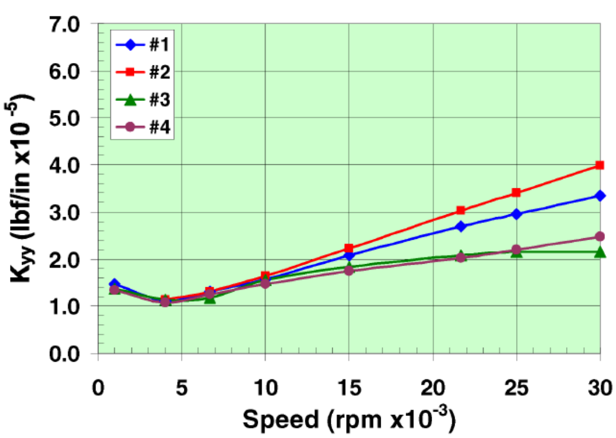


Figure 17. K_{yy} - Variable Viscosity, K_{piv} = 8.0×10^5 lbf/in.

At 21,662 rpm, the maximum pad metal temperature (T_{max}) predictions from the four codes varied from 173°F (#1) to 221°F (#4). Ignoring #2 as it was run with infinite pivot stiffness, the lowest predicted temperature resulted in the highest predicted K and C values (#1). Likewise, the highest predicted temperature resulted in the lowest predicted K and C values (#4).

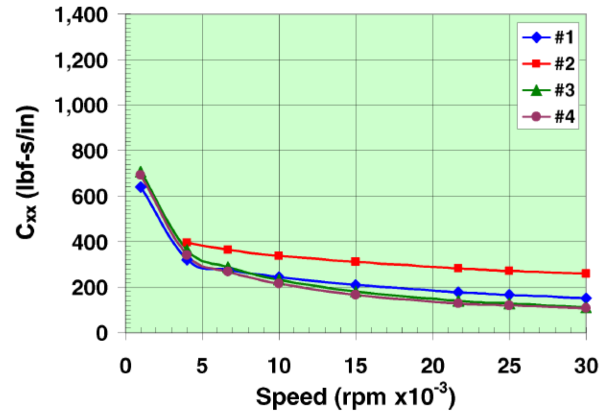


Figure 18. C_{xx} - Variable Viscosity, K_{piv} = 8.0×10^5 lbf/in.

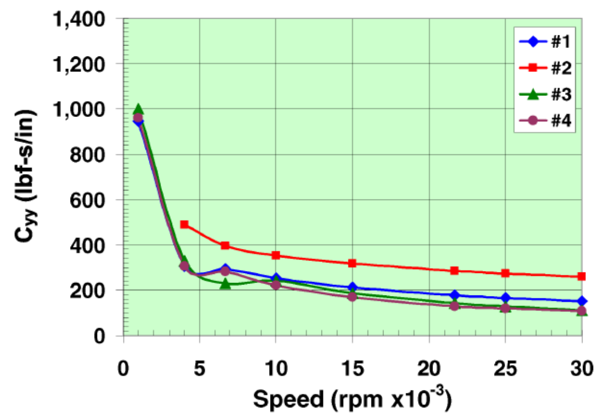


Figure 19. C_{yy} - Variable Viscosity, K_{piv} = 8.0×10^5 lbf/in.

Stability results at 21,662 rpm using the variable viscosity K and C coefficients from the four codes discussed above are summarized in Table 5. Code #2 is listed twice for synchronous and nonsynchronous coefficients. All other results are for the synchronous coefficients from Figures 16, 17, 18, and 19. For synchronous coefficients, the variation in log dec is between 0.552 and 0.796. Code #2's nonsynchronous coefficients produce a log dec value of 0.128.

Eliminating this variation in temperature predictions and thus in viscosity variation, the codes were rerun assuming a constant viscosity. The iso-viscous results are shown in Figures 20, 21, 22, and 23. While the damping variation is negligible (Figures 22 and 23), the stiffness values still show some variation. Closer examination of Figures 20 and 21 reveal that codes #1 and #5 predict the same values while codes #3 and #4 predict the same values. Codes #1 and #5 use the pad assembly solution technique (Lund, 1964) while codes #3 and #4 utilize the full pad solution technique.

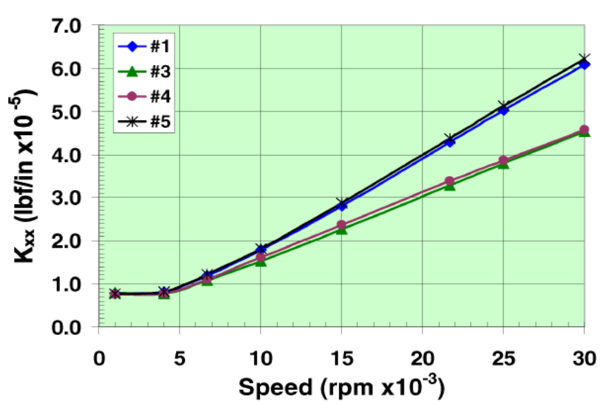


Figure 20. K_{xx} - Iso-viscous, K_{piv} = 8.0×10^5 lbf/in.

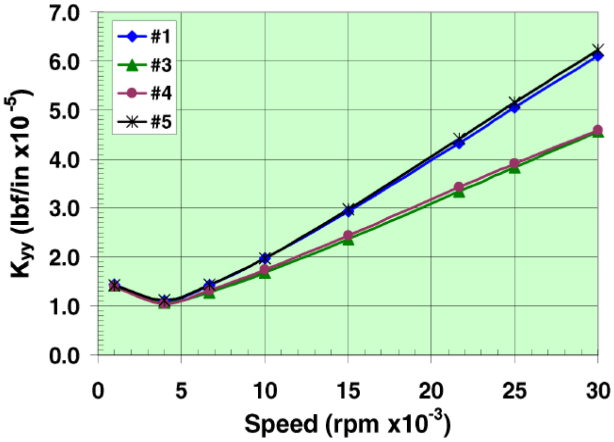


Figure 21. K_{yy} - Iso-viscous, K_{piv} = 8.0×10^5 lbf/in.

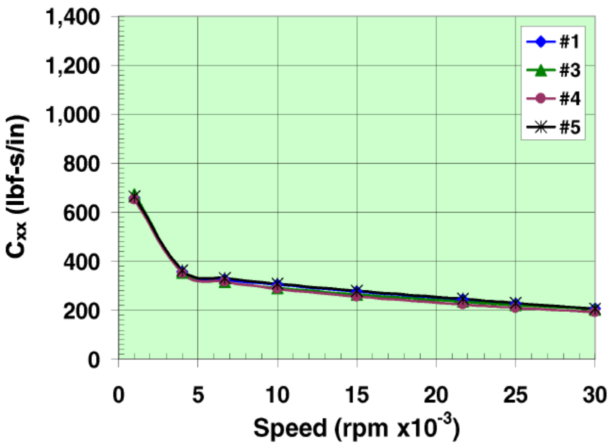


Figure 22. C_{xx} - Iso-viscous, K_{piv} = 8.0×10^5 lbf/in.

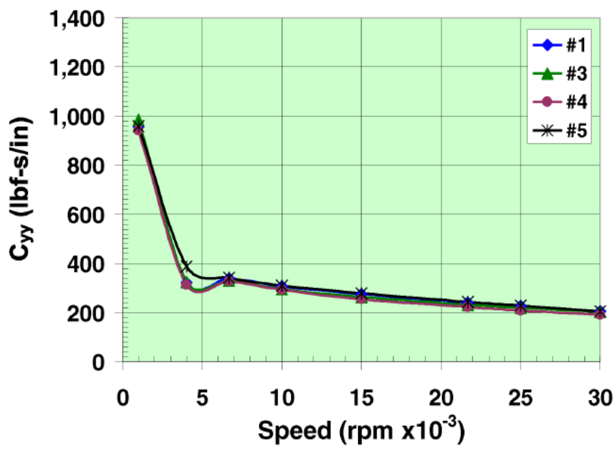


Figure 23. C_{yy} - Iso-viscous, K_{piv} = 8.0×10^5 lbf/in.

Eliminating the pad pivot stiffness as a possible source of variation, the codes were rerun assuming infinite pivot stiffness and constant viscosity. The iso-viscous, infinite pivot stiffness results are shown in Figures 24, 25, 26, and 27. Now, all five codes predict essentially the same results for speeds above 4,000 rpm. Below 4,000 rpm, four out of the five codes predict essentially identical results.

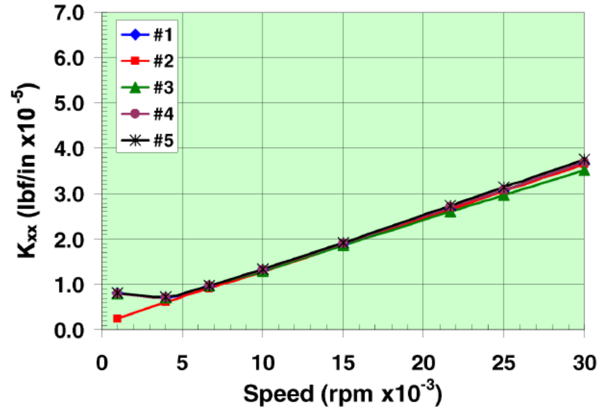


Figure 24. K_{xx} - Iso-viscous, K_{piv} = Infinite.

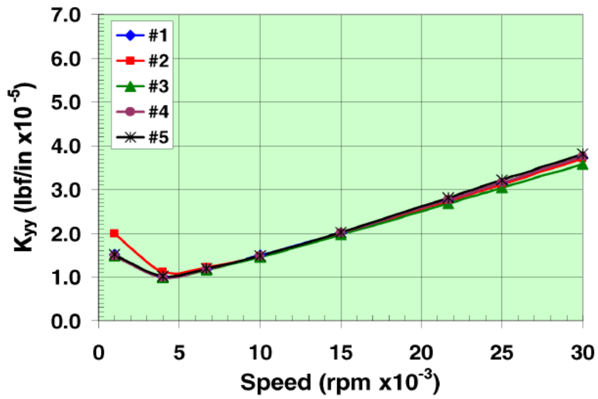


Figure 25. K_{yy} - Iso-viscous, K_{piv} = Infinite.

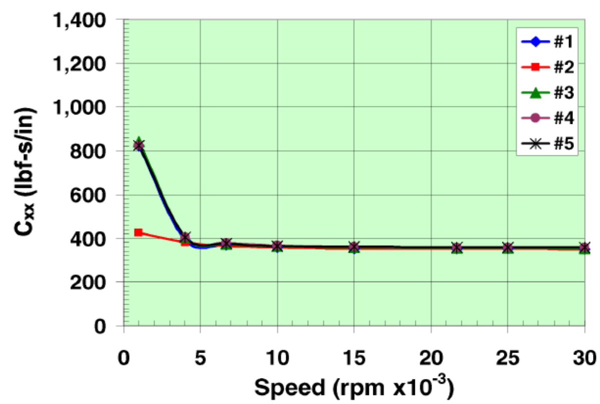


Figure 26. C_{xx} - Iso-viscous, K_{piv} = Infinite.

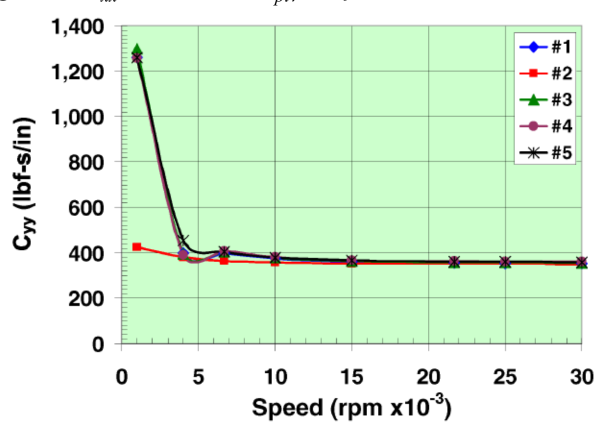


Figure 27. C_{yy} - Iso-viscous, K_{piv} = Infinite.

CONCLUSIONS

This work presents surveyed bearing and seal coefficients from various industry sources including academia, manufacturers, users, and consultants. The variation in those coefficients and their impact on an example compressor rotor stability prediction was highlighted. In an effort to explain the origin of the bearing coefficient differences, five widely used bearing codes were compared. Various input options affecting the thermal solution and pivot stiffness were investigated for their effect on the predicted stiffness and damping of the tilting pad bearing.

The industry survey supplied sufficient (but maybe not exhaustive) data to confirm the working hypothesis that significant differences exist in the prediction of dynamic coefficients for tilting pad journal bearings and gas labyrinth seals. The differences arise from several sources, some due to the solution algorithm and some traced to the user. Whatever the source, sufficient evidence exists to validate the following conclusions:

- A gold standard of experimental data is needed for both tilting pad journal bearings and gas labyrinth seal dynamic coefficients. The data should be used to validate analytical prediction methods. Preferably, the experimental data should be obtained for identical components from several sources. Due to variations in testing procedures, test equipment, etc., multiple sources should be encouraged and funded to obtain the data. Emphasis should be placed on obtaining the component information. While system behavior may provide insight into overall component behavior, i.e., nonsynchronous versus synchronous reduction (Cloud, 2006), it does not provide enough information to validate component coefficient predictions.

- The Level I analysis in the API specifications is still needed. The original intent of that analysis was to provide a screening tool to identify rotors requiring only a simplified analysis and *to provide a common methodology for stability calculations.* The wide variations in bearing and seal coefficients and continued debate on synchronous versus nonsynchronous reduction of tilting pad bearing coefficients, still necessitate a common methodology to permit valid comparisons across the industry.

- For the survey compressor, nonsynchronous reduction of bearing coefficients appears to represent the rotor support situation more accurately. Synchronous coefficients tended to underpredict the frequency and overpredict the stability level of the first forward mode. However, strong caution is advised in drawing widespread conclusions without addressing the significant variations in bearing and seal coefficients shown in the study. Further research, both on a component and system level, is needed in this area.

The predicted stability of the survey compressor was greatly affected by the variations in the bearing and seals coefficients. Frequency predictions for the first forward mode ranged from 6000 to 11,300 cpm. Log dec magnitudes from +1.0 to −1.0 even after ignoring the extremes. While the authors understand that not all compressors will show this sensitivity, the survey compressor shares traits with rotors typically showing instability problems, namely, lightweight rotors operating at high speeds and pressures.

- Analytical predictions of labyrinth seals, both short and long, are still incomplete and, in some cases, insufficient. As noted in the study, the radial force coefficients of the seals were ignored due to the extreme variations (approaching three orders of magnitude) in the coefficients. The study also showed that for the balance piston seal, the destabilizing force, Q_a, produced from synchronously derived coefficients to be 2× greater than that produced from coefficients derived from a WFR of 0.45.

- The impact of the subsynchronous frequency used in solving for either the bearing or seal coefficients was also studied using two specific solution algorithms. A change in WFR from 0.33 to 0.45 decreases by 25 percent the difference between synchronously and subsynchronously reduced bearing coefficients and increases the destabilizing force by 10 percent for the balance piston seal. Both were shown to produce only secondary effects.

From the computer code survey, code-to-code variations in the tilting pad journal bearing stiffness and damping coefficients were found to be due to several sources:

- Synchronous versus nonsynchronous coefficients
  - This is the major source of the K and C variation.
- The temperature solution technique and the resulting pad metal temperature prediction and, thus, the viscosity variation
- The methodology of the inclusion of pivot stiffness
  - The pad assembly codes handle the inclusion of pivot stiffness differently compared to the full solution codes.

Stripping out the above three variations, all five codes produce essentially identical synchronous coefficients for the infinite pivot stiffness, iso-viscous case.

With regard to the industry survey, variations in the bearing coefficients stem from user assumptions affecting pivot stiffness and the frequency reduction of the coefficients. These variations appear to be related more to the user controlled analysis options rather than the analytical methods employed.

The authors hope the presented work can be used to stimulate funded research in the areas of tilting pad and labyrinth seal coefficient predictions/measurements and the impact of frequency dependency on both. As compressor development continues to expand in size and power with an accompanied increase in the cost of unexpected downtime or project delays, accurate prediction of the compressor dynamic behavior is essential.

NOMENCLATURE

C_s	= Seal diametral clearance, mm (mils)
C, C_{xx}	= Principle damping, N-s/mm (lbf-s/in)
D	= Labyrinth seal diameter, mm (in)
H_c	= Minimum width of the impeller or discharge volute, mm (in)
K, K_{xx}	= Principle damping, N-s/mm (lbf/in)
k, K_{xy}	= Cross-coupled stiffness, N/mm (lbf/in)
Q_a	= Destabilizing force, N/mm (lbf/in)
T_w	= Labyrinth seal tooth tip width, mm (in)
T_h	= Labyrinth seal tooth height, mm (in)
T_s	= Labyrinth seal tooth spacing, mm (in)
WFR	= Whirl frequency ratio = w_{wf}/operating speed
w_{wf}	= Whirl frequency of first natural frequency, rad/sec

REFERENCES

American Petroleum Institute Standard 617, 2003, “Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services,” Seventh Edition, American Petroleum Institute, Washington, D.C.

American Petroleum Institute RP 684, 2005, “API Standard Paragraphs Rotordynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsionals, and Rotor Balancing,” Second Edition, American Petroleum Institute, Washington, D.C.

Barrett, L. E., Allaire, P. E. and Wilson, B. W., 1988, “The Eigenvalue Dependence of Reduced Tilting Pad Bearing Stiffness and Damping Coefficients,” *ASLE Transactions, 31,* pp. 411 - 419.

Branagan, L. A., 1988, “Thermal Analysis of Fixed and Tilting Pad Journal Bearings Including Cross-Film Viscosity Variations and Deformation,” Ph.D. Dissertation, University of Virginia, Charlottesville, Virginia.

Camatti, M., Vannini, G., Fulton, J., and Hopenwasser, F., 2003, "Instability of a High Pressure Compressor Equipped with Honeycomb Seals," *Proceedings of the Thirty-Second Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 39-48.

Chen, W. J. and Gunter, E. J., 2005, "Introduction to Dynamics of Rotor-Bearing Systems," Victoria, British Columbia, Canada: Trafford Publishing.

Childs, D. W., 1993, *Turbomachinery Rotordynamics: Phenomena, Modeling, and Analysis,* New York, New York: John Wiley & Sons, Inc.

Childs, D. W. and Scharrer, J. K., 1986, "An Iwatsubo-Based Solution for Labyrinth Seals: Comparison to Experimental Results," ASME Journal of Engineering for Gas Turbines and Power, *108,* (2), pp. 325-331.

Childs, D. W. and Ramsey, C., 1991, "Seal Rotordynamic-Coefficient Test Results for a Model SSME ATD-HPFTP Turbine Interstage Seal With and Without a Swirl Brake," ASME Journal of Tribology, *113,* pp. 113-203.

Cloud, C. H., 2006, "Stability of a Rotor Supported on Tilting Pad Journal Bearings," Ph.D. Dissertation, University of Virginia, Charlottesville, Virginia.

Elrod, D. A., Pelletti, J. M., and Childs, D. W., 1995, "Theory Versus Experiment for the Rotordynamic Coefficients of an Interlocking Labyrinth Gas Seal," ASME Paper 95-GT-432, Presented at the International Gas Turbine and Aeroengine Congress and Exposition, Houston, Texas.

He, M., 2003, "Thermoelastohydrodynamic Analysis of Fluid Film Journal Bearings," Ph.D. Dissertation, University of Virginia, Charlottesville, Virginia.

Iwatsubo, T., Matooka, N., and Kawai, R., 1982, "Spring and Damping Coefficients of the Labyrinth Seal," NASA CP-2250, pp. 205-222.

Kirk, R. G., 1990, "A Method for Calculating Labyrinth Seal Inlet Swirl Velocity," *ASME Journal of Vibration and Acoustics, 112,* (3), pp. 380-383.

Kocur, J. A., Jr. and Hayles, G. C., Jr., 2004, "Low Frequency Instability in a Process Compressor," *Proceedings of the Thirty-Third Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 25-32.

Lund, J. W., 1964, "Spring and Damping Coefficients for the Tilting-Pad Journal Bearing," ASLE Transactions, 7, pp. 342-352.

Nicholas, J. C., 2003, "Lund's Tilting Pad Journal Bearing Pad Assembly Method," ASME Journal of Vibrations and Acoustics, *125,* (4), pp. 448-454.

Nicholas, J. C., Gunter, E. J., and Allaire, P. E., 1979, "Stiffness and Damping Coefficients for the Five Pad Tilting Pad Bearing," ASLE Transactions, *22,* (2), pp. 112-124.

Nordmann, R. and Weiser, H., 1990, "Evaluation of Rotordynamic Coefficients of Look-Through Labyrinths by Means of a Three Volume Bulk Model," *Rotordynamic Instability Problems in High-Performance Turbomachinery,* NASA CP-3122, pp. 141-157.

Parsell, J. K., Allaire, P. E., and Barrett, L. E., 1983, "Frequency Effects in Tilting-Pad Journal Bearing Dynamic Coefficients," *ASLE Transactions, 26,* pp. 222-227.

San Andres, L. A., 1995, "Bulk-Flow Analysis of Flexure and Tilting Pad Fluid Film Bearings," TRC-B&C-3-95, Turbomachinery Laboratory, Texas A&M University, College Station, Texas.

Vázquez, J. A. and Barrett, L. E., 1998, "Representing Flexible Supports by Polynomial Transfer Functions," ASME Paper 98-GT-27.

Vázquez, J. A., Barrett, L. E., and Flack, R. D., 2001, "A Flexible Rotor on Flexible Bearing Supports: Stability and Unbalance Response," Journal of Vibration and Acoustics, ASME Transactions, *123,* (2), pp. 137-144.

Wagner, N. G. and Steff, K., 1996, "Dynamic Labyrinth Coefficients from a High-Pressure Full-Scale Test Rig Using Magnetic Bearings," *Rotordynamic Instability Problems in High-Performance Turbomachinery,* NASA CP-3344, pp. 95-111.

Wygant, K. D., Barrett, L. E., and Flack, R. D., 1999, "Influence of Pad Pivot Friction on Tilting-Pad Journal Bearing Measurements—Part II: Dynamic Coefficients," *STLE Tribology Transactions, 42,* (1), pp. 250-256.

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