Review of Atkins and Perez Rotor Stability Tests with Various Five Pad Tilting Pad Bearings

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Abstract

This paper reviews and evaluates some of the important experimental findings of the paper entitled "***Assessing Rotor Stability Using Practical Test Procedures***" by Kenneth E. Atkins and Robert X. Perez presented in 1992 at the Twenty-First Turbo Machinery Symposium in Houston, Texas.

The paper given by Atkins and Perez presents an extensive amount of experimental data on the testing of an industrial five stage compressor mounted in narrow five pad tilting pad bearings with various amounts of bearing preload and clearance. The motivations for the testing were derived from concerns for the compressor stability operating at high speeds and discharge pressures. These apprehensions were based on previous experiences with unstable rotors such as occurred with the Kaybob compressors.

The narrow L/D bearings tested were thought to have superior stability characteristics due to the influence of bearing asymmetry effects. A variety of bearings with different clearances and preloads were tested. An exciter was used to determine the experimental response over a wide range of speeds for the various bearing configurations. From the experimental test results it was determined that the narrow five pad L/D bearing had a very high amplification factor of over 38 for the 5 mil clearance bearing measured at 11,000 rpm. With the open bearing clearance of 8.5 and reduced preload, the amplification factor was reduced to 25. This is still an unacceptable level.

A number of stability analysis were performed on the 5 stage compressor corresponding to various bearing configurations tested. It was found that in general, regardless of the bearing clearance or preload, if any of these configurations were used in a high-speed, high pressure environment such as Kaybob, then violent self-excited whirling would have occurred. The heavy seal and compressor rubs that would have incurred would lead to catastrophic rotor failure.

A three pad bearing is presented and is shown to have superior stability characteristics over the operating speed range. In general, the five shoe load-on-pad bearing design has been overused and is usually poorly suited for use in flexible mutistage compressors operating at high speed and high pressure conditions. Many of the cases that required a squeeze film damper could have been eliminated by the use of a three or four pad bearing design.

Historical Background and Introduction

Starting in the 1960's, the compressor companies started switching from plain journal bearings to the tilting pad bearing design to improve rotor stability. This allowed compressor designs to increase in speed and horsepower from the previous limits of about 7,500 RPM with plain journal bearings to over 12,000 RPM. These new compressor designs with tilting pad bearings performed extremely well under factory tests at very low pressures; however, when installed in high pressure operations such as oil recovery, the results were catastrophic.

In 1975, the results of the Kabob compressor disaster was reported by Smith at the ***Fourth Turbomachinery Symposium*** (7). Under high pressure operation, the compressors would exhibit self-excited whirl instability resulting in rotor destruction. Smith reported that a number of bearing designs were attempted to improve bearing asymmetry and attempts were made to install squeeze film dampers. None of these design changes were successful. Several new rotor designs were attempted. The final rotor design had a shortened and stiffened shaft. This new rotor design was the final solution. The final cost for design and loss of production was estimated to be over $100 million. After this experience, Cooper gave up building compressors for over a 10 year period. A thorough review of the Kabob disaster is presented in reference (2).

The paper given by Atkins and Perez in 1992 at the ***21th Turbomachinery Conference*** presents the results of extensive experimental tests to measure the log decrement of the Citgo five stage compressor mounted in load on pad tilting pad bearings. It was considered that the five pad load on pad ***(LOP)*** bearing design with an L/D aspect ratio of less than 0.5 would promote rotor stability. The tests involved the forced response of the rotor over various speed ranges with various bearing clearances and preload ratios with the load on pad bearing design now commonly used with this class of compressor.

Figure 1 shows a picture of the five stage Citgo compressor to be tested.

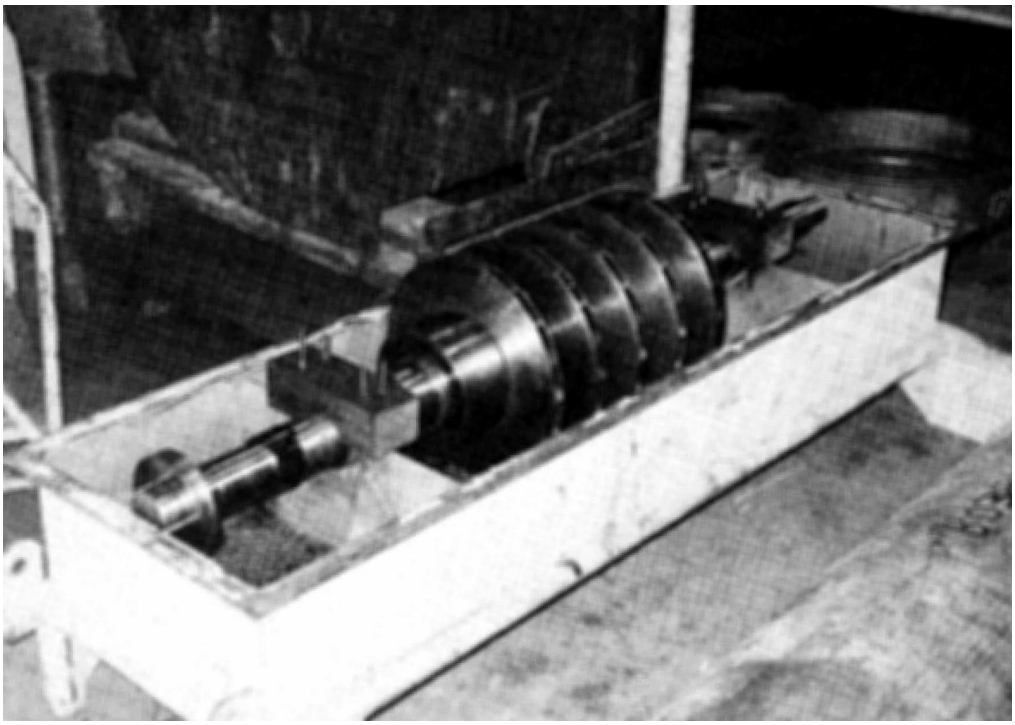

Figure 1 Five Stage Centrifugal Compressor Rotor
2,800 HP @ 12,000 RPM -*(Atkins & Perez -1992)*

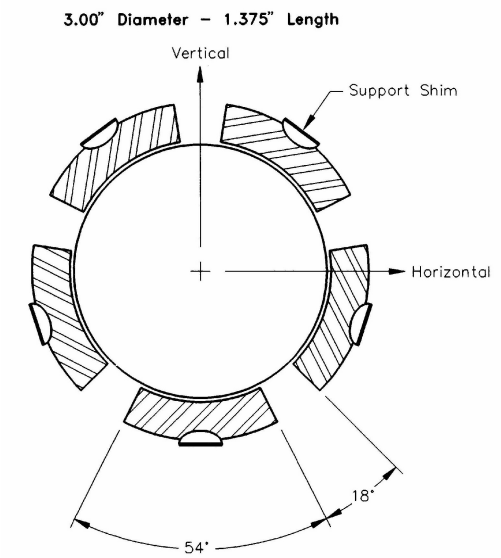

Fig 2 5 Pad Tilt Pad Bearing, LOP
(Atkins & Perez -1992 p.152)

Table 1. Summary of Rotor-Bearing Configurations Tested.

Test Number	*Assembled Clearance (mils)*	*Preload*	*Rotor*	*Shaker Location*
1	7.0	0.30	Main	Inboard
2	7.0	0.30	Main	Outboard
3	5.0	0.50	Main	Inboard
4	8.0	0.20	Main	Inboard
5	6.0	0.45	Spare	Inboard
6	8.5	0.23	Spare	Inboard

Figure 3 Bearing Preload and Clearance Combinations
(Atkins & Perez -1992 p.153)

Figure 2 represents the five shoe load-on-pad bearing configuration as used in the experimental testing for the determination of the rotor log decrement at various speeds. Figure 3 represents the various clearances and preloads used in the testing. The preloads and clearances tested ranged from a very high preload of .5 with a tight bearing clearance of 5 mils to an open bearing clearance of 8 mils with a light preload of .2. No attempts were made to generate the bearing stiffness and damping coefficients for these various bearing combinations selected.

The stiffness and damping characteristics of the five pad bearing are well documented. In 1977, for example, Nicholas, Gunter and Allaire presented a paper on "***Stiffness and Damping Coefficients for the Five Pad Tilting Pad Bearing***"(3) for L/D ratios of .5 and 1.0.

On the Origin and Use of Five Pad Bearings for Centrifugal Compressors

The use of tilting pad bearings starting in the 1960's and 1970's greatly extended the operating speed of multi-stage centrifugal compressors. As a young engineer, my first position was with Clark Bros. which had an excellent training program for young engineers in that they spent one year in the factory working with reciprocating compressors, centrifugals and gas turbines.

As a centrifugal compressor design engineer, in those days, all drawings were done on a drafting board and calculations were by sliderule. At that time there were no computer programs to calculate either the bearing characteristics or rotor dynamics such as critical speeds or stability analysis (damped eigenvalues and log decrements). The only rotor dynamics calculations performed was a simple graphical critical speed iteration method based on the Stodola method which assumed rigid bearings and did not take disk gyroscopic effects into consideration. The design rule at that time was that one should not exceed an operating speed of 1.45 times the predicted rigid bearing critical speed.

When I asked the head German master mechanic why this rule he said: "***son, above those speeds bad things happen!"***

What was particularly intriguing with the new M6 line, which represented a split casing machine was six impellers, was that when the top half of the bearing cartridge was assembled, the master mechanic placed a 10 mil shim between the upper cartridge and the casing.

When I inquired why this was done, since it was not specified in any of the drawings, he again replied "***this is for the bearing crush!".*** He then formed his fingers together to present an ellipse to me. What he had done by intuition or maybe from experience in Germany was that the 10 mil shim had in reality turned the plain journal bearings into two lobe elliptic bearings. These machines were able to run successfully, increasing the speed limit of 7,500 to over 10,000 RPM without incurring problems. *(Note: All major compressor designs originally came from engineers trained in Germany and Switzerland such as Bill Trumpler,Chief Engineer at Clark Bros. and trained under Stodola)*

However, a number of years later when these machines were disassembled for cleaning and maintenance, the shims were unnoticed and were not replaced. Upon reassembly of the units, most experienced self-excited oil whirl instability. Needless to say, this caused great consternation and hand wringing back at the head office.

The tilting pad bearing design has been known for many years. They were not initially considered for use in centrifugal compressors as they were considered to be too expensive as well as too difficult to manufacture and maintain. However, when the five pad tilt pad bearings were installed in the M line, the self excited oil film whirl instability occurrences were eliminated and speeds of 10,000 RPM and higher were achieved.

At the time this was occurring in the industry, there were no reliable instruments to monitor the rotor shaft orbits, frequency spectrum, or to understand the mechanism of self-excited whirling. The best that could be done under testing was to place the Davie reed vibrometer on the casing for basic frequency measurement and to use the IRD shaft rider stick to get a crude idea of shaft amplitudes. Placing a wooden shaft rider stick on a rotor spinning up to 14,000 RPM was a task that most people did not relish doing.

Up until the late 1960's, compressors were not instrumented to determine shaft behavior. For example, at the Franklin Institute at the Gas Bearing Laboratory, which I directed for a number of years, we could observe the high-speed motion of the gas bearing rotors using two sets of Wayne Kerr capacitance probes fed into an oscilloscope to view and measure the orbits. These probes could measure linear motion up to 10 mils, but could not be used in an oil environment. These probes were extremely expensive and costly to replace, and so were not suited for use in industry.

All this changed with the development of the non-contacting proximity probe by Donald Bently, which completely changed our understanding of rotor dynamics, balancing and particularly of self excited whirl instability effects.

I first met Don Bently in 1967 when he visited my lab at the University of Virginia. He brought with him one of his new single mass rotor kits that he had developed with his X-Y proximity probes to measure the orbits. What was unique about the installation was that he had both X and Y probes to monitor the shaft motion at various shaft locations. In addition he had a third probe called a keyphaser, which monitored a notch on the shaft. This was fed into the Z axis of the oscilloscope and generated a blip on the orbit of the shaft. As one would pass through the critical speed, one could see the amplitude or orbit grow with the timing mark moving opposite the direction of rotation. This was the beginning of the measurement of rotor phase angles. Thus by observing the timing mark and knowing whether the rotor was above or below the critical speed, one could estimate an ideal location to apply the first trial weight for balancing. The current least squares error balancing programs make use of amplitude and phase measurements for single and multiplane balancing calculations.

At that time *(1967),* we did not have the FFT analyzer to evaluate the spectrum. We are indebted to the Russians, the Cold War and the Navy for the development of the FFT. The Office of Naval Research (ONR) spent considerable funds in developing the FFT with both Hewlett-Packard and other companies. The object of this initial intense research was directed towards quiet operation of our nuclear attack submarines.

In 1969, I was called in by Adm. Rickover's office to visit with the engineering group at Westinghouse to evaluate the vibrations in their vertical nuclear water pumps. Westinghouse had installed a vertical pump with plain bearings that was in subsynchronous whirl that they claimed could be detected 200 miles away.

Over the next 10 years, I visited with Don Bently between semesters and worked with him for a week or two in his lab looking at his various rotor models and new equipment that his group had developed. In the beginning Don started in Carson City at the airport with seven personnel. The last time I had met with Don was at the ISCOMA conference in Cleveland in 2005.

One of the first persons to use his probes was Charlie Jackson of Monsanto. Malcolm Leader, who is one our foremost users of Dyrobes, went to work with Charlie in the early days. Don was very impressed with what Charlie was doing with his probes. Don would exclaim gleefully that Charlie was using two of his probes to monitor the axial motion of a thrust bearing. This pleased him to no end.

Charlie was also one of the first to install radial probes on the Monsanto compressors. The easiest place to install these probes appeared to be in the vertical direction in the bearing housing. No attempts were ever made to monitor the rotor center span motion as this presented complications. Charlie was also very active in the API code specification for compressors. He encouraged other petrochemical companies to install probes for monitoring axial and lateral rotor response. This soon became standard procedure with the petrochemical industry and became required by API.

I was very grateful to Mr. Bently, not only for the equipment he provided for my rotor dynamics lab, but also with each purchase of his proximity probes, he included a copy of my thesis on stability of rotor bearing systems(4). NASA had supported my research and published my thesis which they sold for the magnificent price of one dollar. Don bought up most of their publications. In the thesis, a great deal of emphasis was placed on the importance of bearing or support asymmetry in promoting rotor stability as observed by Bert Newkirk on his earlier compressor work at GE.

This early concept of bearing asymmetry promoting rotor stability has led to a number of problems with the selection of bearings and the misconception of how bearing asymmetry improves rotor stability. What was not realized at the time was the importance of the ratio of the bearing vertical stiffness to the shaft stiffness. If the ratio of bearing to shaft stiffness exceeds 0.5, then bearing asymmetry has very little effect in improving rotor stability.

Thus the five pad bearing was preferred because it was thought to provide more bearing asymmetry than a four pad bearing. Also by having load on pad, this made the manufacturers quite happy on testing as there was minimal amplitude of motion at the probes when the rotor was passing through both first and second critical speeds.

The four pad bearing at this time was discouraged because of the higher amplitudes recorded at the bearing; however, when Westinghouse developed their line of gas turbines such as the W501, they had the know-how of an advanced bearing lab from by the leading researchers that initially started the ASLE, the American Society of Lubrication Engineers. Their turbine bearing of choice was the four pad bearing with load between pads. It is of interest to note that the generator bearings are still using 110 year old 120 partial deg arc bearings which are unstable! The generator division did not wish to go to the extra expense of installing tilt pad bearings.

We shall see later that the five pad bearing with load on pad usually is poorly matched to a compressor. The reason for this is that the vertical bearing stiffness exceeds that of the fundamental shaft modal stiffness. When this occurs, high amplification factors are encountered and susceptibility towards rotor instability is encountered at high speeds and pressure ratios. Many of these units that had five pad bearings could only be corrected by applying squeeze film dampers. It was shown in a later paper by Gunter and Weaver (15) that with a properly designed four pad bearing, a squeeze film damper would not be required.

Rotor Experimental Setup

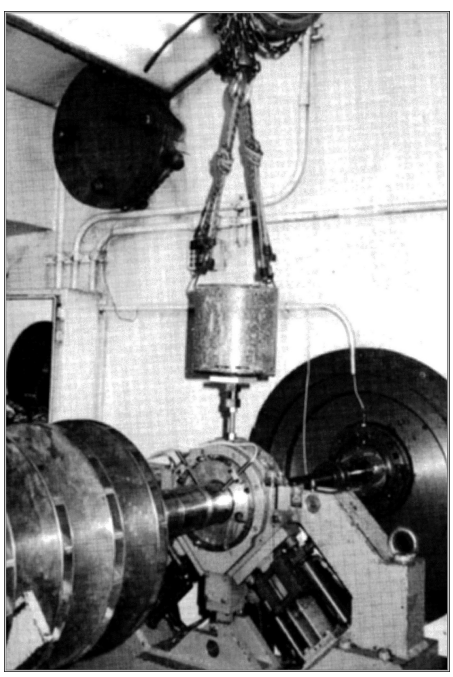

Fig.4 Shaker Mounted on Inboard Bearing Pedestal

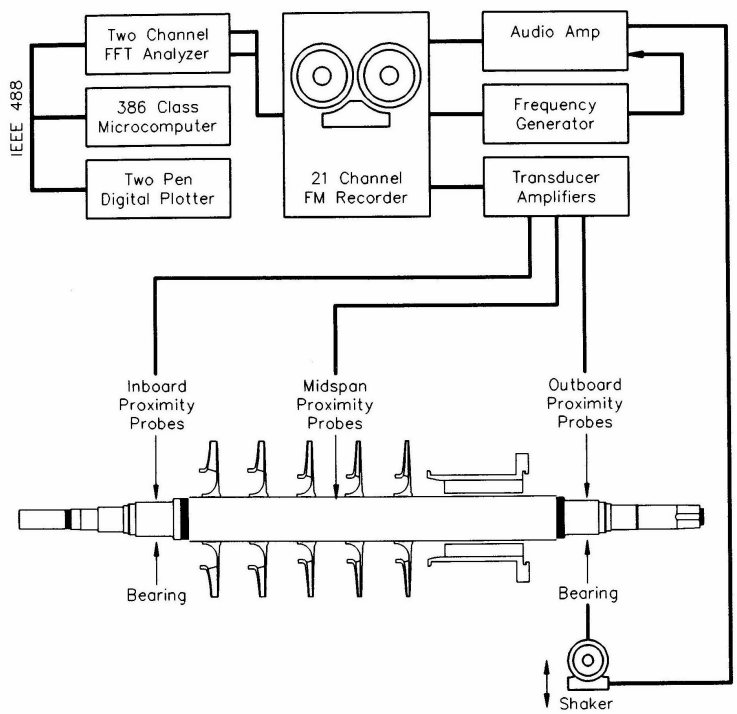

Figure 5 Instrumentation Schematic
(Atkins & Perez)

Figure 4 shows the external shaker mounted on the inboard bearing pedestal. Figure 5 shows the instrumentation schematic used to analyze the data and to plot the amplitude and phase plots corresponding to the various speeds of operation.

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We shall later see that the five pad bearing design with load on pad is usually a poor choice to use with a multistage compressor that must operate at speeds of over 12,000 RPM. The reason for this, as previously mentioned, is that the vertical bearing stiffness exceeds that of the fundamental shaft modal stiffness. When this occurs, high amplification factors are encountered and susceptibility towards rotor instability. Many of these units that had five pad bearings could only be corrected by applying squeeze film dampers. It was shown in a later paper by Gunter and Weaver *(15)* that with a properly designed four pad bearing, a squeeze film damper would not have been required.

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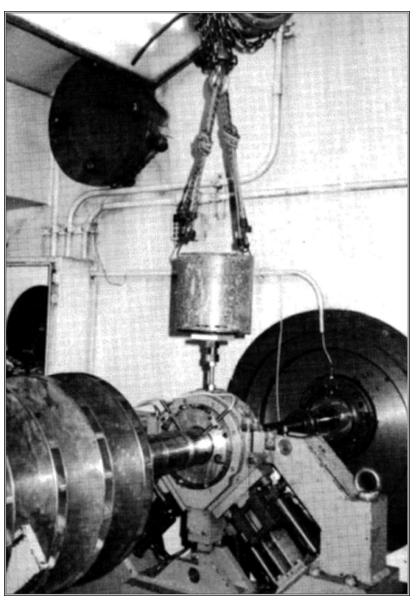

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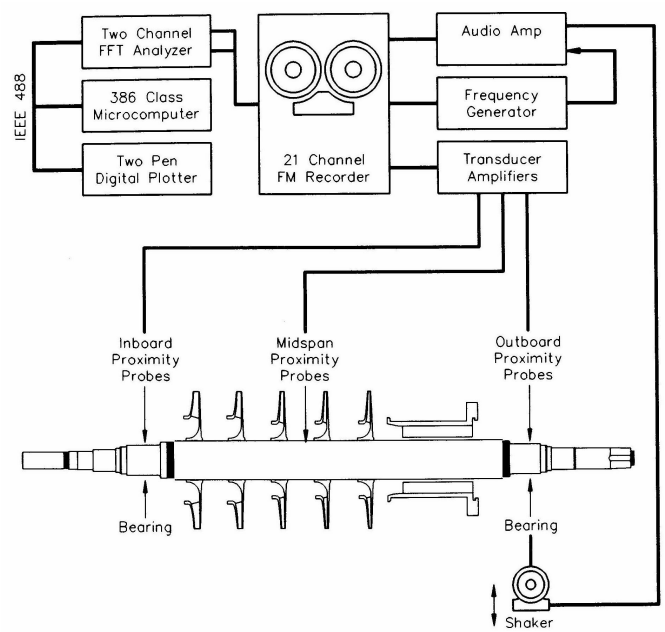

Figure 5 Instrumentation Schematic
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Figure 4 shows the external shaker mounted on the inboard bearing pedestal. Figure 6 shows the instrumentation schematic used to analyze the data and to plot of the amplitude and phase plots corresponding to various speeds of operation.

Experimental Response Plots with Several Preload and Clearance Ratios

Figures 6 and 7 represent some of the experimental data generated by Atkins and Perez on the rotor excitation of the five stage compressor with the five pad bearing with various bearing clearances and preloads. Figure 7 represents a large clearance bearing of 8.5 mils with a low preload ***(PLF)*** of 0.23. With this open bearing clearance and low preload, the recorded transfer amplification factor is a maximum of 2 at 11,000 RPM. The recorded rotor first resonance frequency excited at 11,000 RPM is approximately 3,800 CPM.

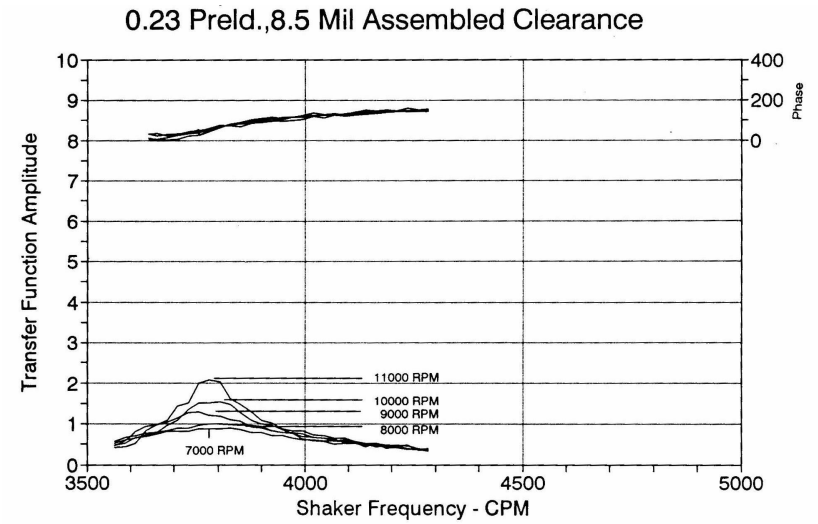

Figure 6 Response Plot with 8.5 Mil Cd & 0.23 Preload
(Atkins & Perez - 1992)

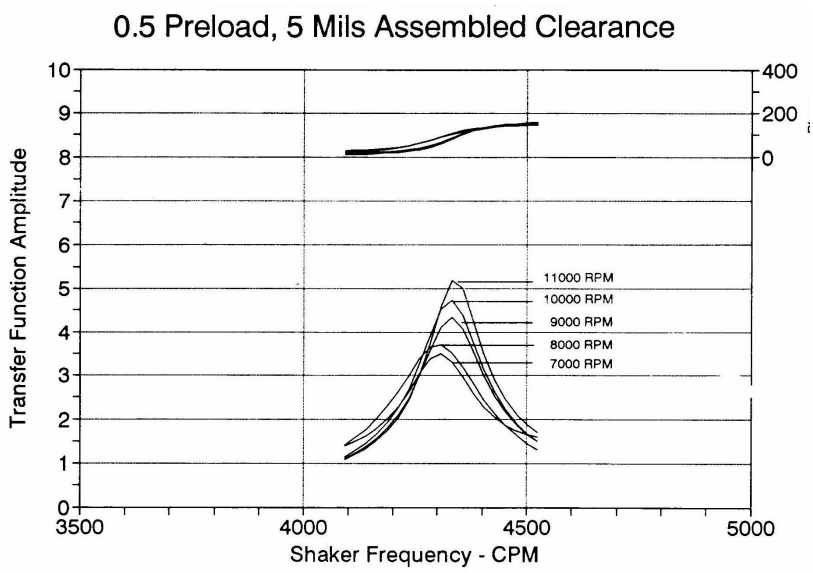

Figure 7 Response Plot with 5.0 Mil Cd and 0.5 Preload
(Atkins & Perez - 1992)

Figure 7 represents the response plot with the tightened diametral bearing clearance of 5 mils and the increase bearing preload of 0.5. With the running speed at 11,000 RPM, it is seen that the first resonance frequency is increased to 4350 CPM and an increased amplification factor of 5 is observed. Thus the tight bearing clearance and large preload has caused a high amplification.

Critical Speed Analysis

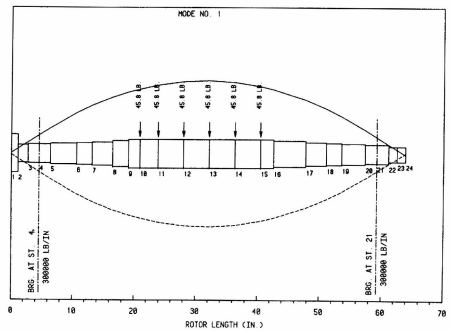

Fig. 8 Critical Speed Model

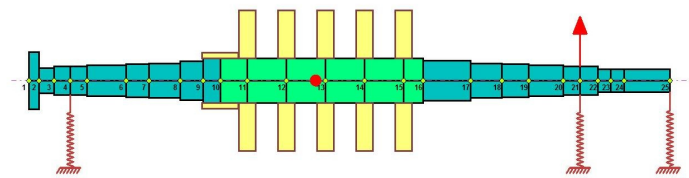

Fig. 9 Modified 5 Stage Compressor Model Including External Exciter

Figure 8 represents the original critical speed model for the five stage compressor. The authors did not show the results of their original critical speed model. When this model was analyzed, the resulting critical speed was only 3,700 RPM. This is well below the values observed in the testing. Figure 7, for example, shows that with a tight bearing clearance of 5 mils and a high preload of 0.5, then the first resonance frequency can exceed 4,300 CPM.

A modified model was developed as shown in Figure 9. It is apparent that the impellers have caused a stiffening of the shaft to raise the critical speed. It is important to generate the critical speed model so that the shaft stiffness value may be properly determined. The shaft modal stiffness value is determined by assuming rigid bearings. A shaft stiffness value of approximately 260,000 Lb/In was computed.

The optimum vertical bearing stiffness should be to $2K_{yy}/K_{shaft} = 1$. If this value is greatly exceeded then high amplification factors will be experienced and the rotor will be sensitive to Alford type of cross coupling forces at high speeds and high operational pressure conditions. Most five pad bearing designs violate this fundamental principle and hence are ill-suited for use in high speed, high pressure multistage compressors.

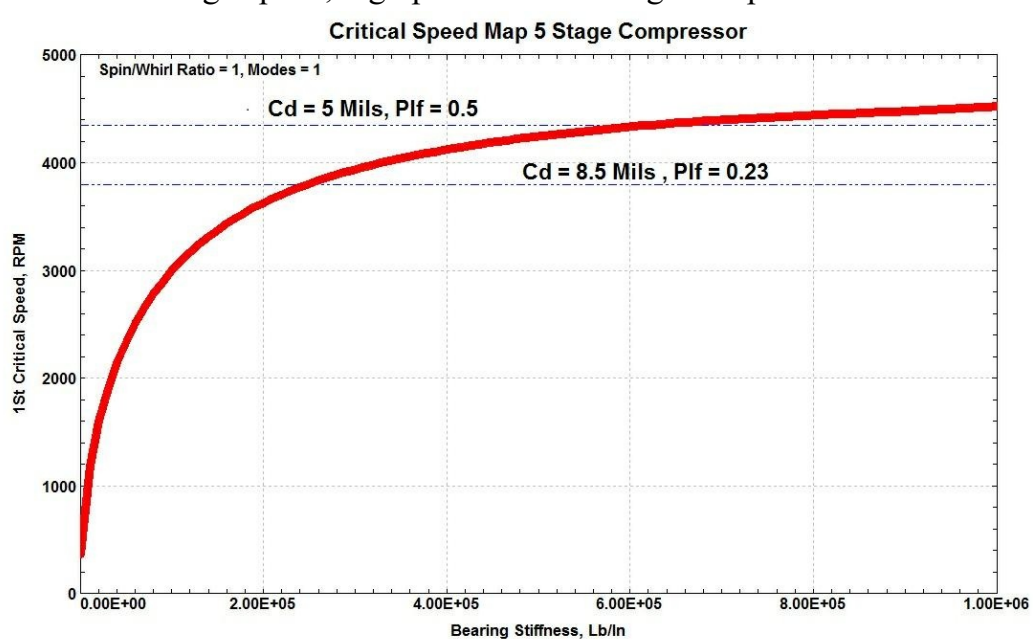

Figure 10 Critical Speed Map Showing Bearing Clearance and Preload Effects

Figure 10 represents the critical speed map for the reinforced rotor. Superimposed upon critical speed map are two lines. The lower line represents critical speed corresponding to the five pad bearing with the large bearing clearance of 8.5 mils and light preload of 0.23. The second line represents the critical speed with the tight bearing clearance of five mils and a high preload of 0.5.

The closing up of the bearing clearance from 8.5 to 5 mils increases the vertical bearing stiffness and raises the first critical speed from 3,800 CPM to over 4,300 CPM. Note that once the effective bearing stiffness approaches 600,000 Lb/In, the critical speed plot becomes asymptotic. This is an indication that there is very little amplitude of motion at the bearings. Once the rotor operates on this section of the critical speed map, then the bearing damping will be very ineffective in suppressing self excited whirling at high speeds such as caused by the Alford effect.

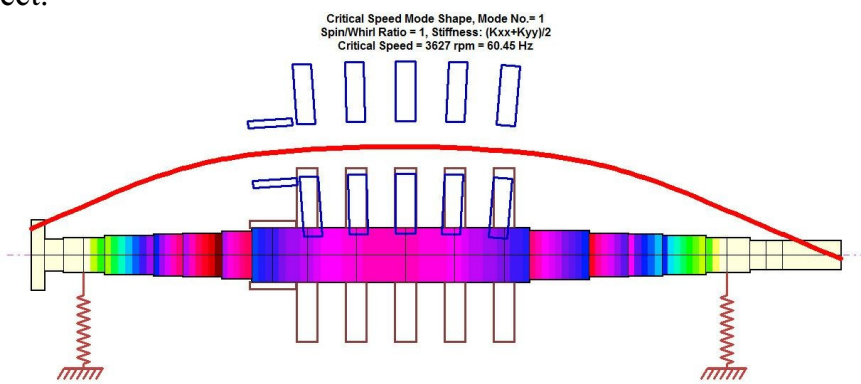

Figure 11 1st Critical Speed With Kb = 200,000 Lb/In
N1 = 3,627 RPM

Figure 11 represents the rotor first critical speed with an assumed bearing stiffness of Kb = 200,000 Lb/In. This corresponds closely to the tested bearing with the large clearance of 8.5 mils and light bearing preload of .23. The larger bearing clearance and lighter preload reduces the critical speed and causes more amplitude to be observed at the bearings.

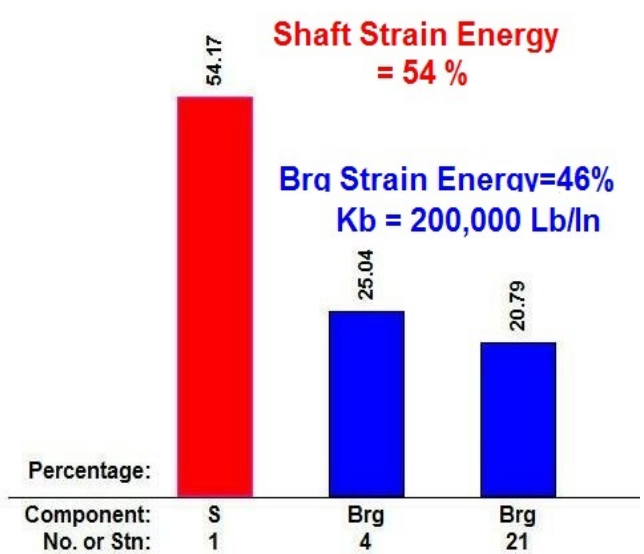

Fig. 12 Shaft and Bearing Strain Energy
For Kb =200,000 Lb/In

Figure 12 represents the energy distribution in the shaft and bearings for the assumed bearing stiffness values of Kb = 200,000 Lb/In. The energy distribution in this case is closely balanced with 54% of the energy in the shaft and approximately 46% of the energy distributed in the bearings. This is considered an acceptable distribution.

The calculated critical speed in Figure 11 of 3,627 RPM with an assumed bearing stiffness of 200,000 Lb/In corresponds closely with the experimental data as observed at 11,000 RPM in Figure 6 with the large bearing clearance of 8.5 mils and the preload of 0.23.

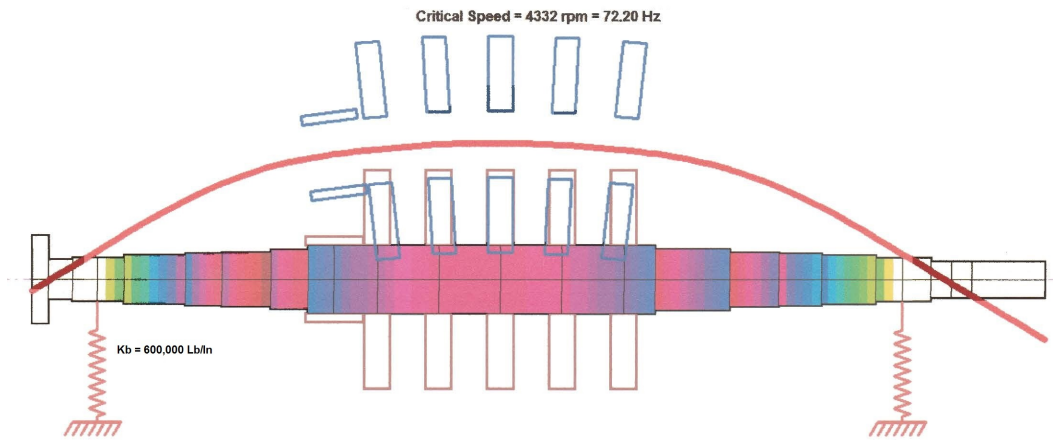

Figure 13 1st Critical Speed With Kb = 600,000 Lb/In
N1 = 4,332 RPM

Figure 13 represents the mode shape of the five stage compressor with the assumed nominal bearing stiffness of 600,000 Lb/In. The resulting critical speed is approximately 4,332 RPM. This corresponds closely with the value observed at 11,000 RPM in the experimental plot on Figure 7 with the tight bearing clearance of 5 mils and 0.5 preload.

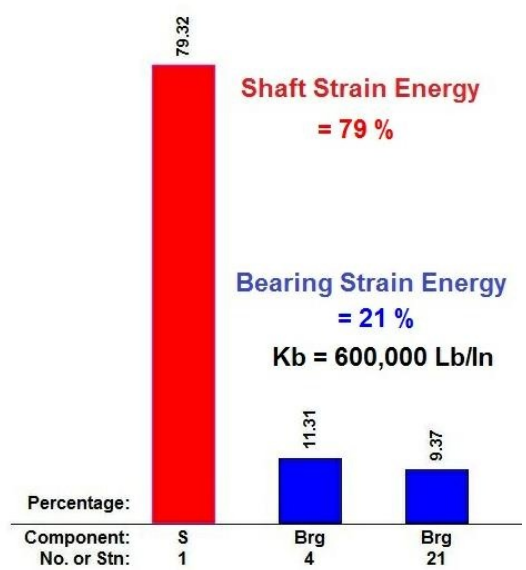

Fig. 14 Shaft and Bearing Strain Energy
For Kb =600,000 Lb/In

Figure 14 shows the distribution of the strain energy of the shaft and bearings with the relative bearing stiffness increased to 600,000 Lb/In. In this case we have seen that strain energy in the shaft increases to over 79% while the strain energy in the bearings is reduced to 21%. This is not an acceptable ratio.

Summary of Experimental Log Decrements And Amplification Factors for Various Five Pad Bearings

Table 2. Summary of Test Results.

Rotor	*Clearance/ Preload (mils)*	*Speed (rpm)*	*Peak Freq. (Hz)*	*Damping %*	*Q*	δ_m
Main	5.0/0.50	7000	71.7	1.73	28.9	0.109
		8000	71.7	2.14	23.4	0.135
		9000	72.2	1.58	31.6	0.099
		10000	72.1	1.53	32.7	0.096
		11000	72.3	1.30	38.5	0.082
Main	7.0/0.30	7000	67.6	3.71	13.5	0.233
		8000	67.5	3.09	16.2	0.194
		9000	67.3	3.16	15.8	0.199
		10000	67.7	2.80	17.9	0.176
		11000	67.3	2.51	19.9	0.158
Main	8.0/0.20	7000	63.7	6.13	8.16	0.385
		8000	64.9	4.27	11.71	0.268
		9000	66.0	4.46	11.21	0.280
		10000	65.0	2.37	21.10	0.149
		11000	65.3	2.78	17.99	0.175
Spare	6.0/0.45	7000	67.9	2.33	21.5	0.146
		8000	68.1	2.54	19.7	0.160
		9000	68.5	2.13	23.5	0.134
		10000	68.7	1.84	27.2	0.116
		11000	68.9	1.64	30.5	0.103
Spare	8.5/0.23	7000	63.4	5.43	9.21	0.341
		8000	63.5	4.23	11.82	0.266
		9000	62.6	3.51	14.25	0.221
		10000	63.0	2.50	20.00	0.157
		11000	63.0	1.95	25.64	0.123

Figure 15 Experimental Test Results of Log Dec of Five Pad Bearings with Various Clearances and Preloads
(Atkins & Perez p. 155)

Figure 15 represents the results of the extensive testing of the five pad bearings with the various combinations of bearing clearance and preload. In this paper, we shall concentrate on examining the results for the two bearing extremes in which we have a tight bearing clearance of 5 mils and high preload of .5. The other extreme to be examined will be the case with the large clearance bearing of 8.5 and preload of 0.23.

As mentioned previously, no attempt attempts were made to compute either the rotor critical speeds for the test rotor, the bearing coefficients for the various preload cases, or the damped eigenvalues for this system.

In the next section, the bearing stiffness and damping characteristics for the two extreme bearing cases will be computed corresponding to the operating speed of 11,000 RPM. With these computed bearing coefficients, the theoretical log decrements may be computed and compared to the measured experimental values.

Computed Tilt Pad Bearing Characteristics at 11,000 RPM

5 PAD LOP 3 In x 1.375 In, 0.5 Plf, Cb = 5 Mils Dia N=11,000 RPM, Mu=1.0E-6
L/D= 0.4583, Cb= 0.0025, 2Cb/D=0.00167, m= 0.5, Offset= 0.5, Arc= 54, PivAng = -18
Speed = 11000 rpm
Load = 264 Lbf
W/LD = 64 psi
Vis. = 1E-06 Reyns
Sb = 1.0313
E/Cb = 0.4234
Att. = 0.00 deg
hmin = 1.212 mils
Pmax = 311.845 psi
Hp = 1.95204 hp
Qside = 0.810 gpm
Stiffness (Lbf/in)
1.578E+05 0.000E+00
0.000E+00 3.787E+05
Damping (Lbf-s/in)
1.474E+02 0.000E+00
0.000E+00 2.506E+02
Critical Journal Mass (Lb)
Stable

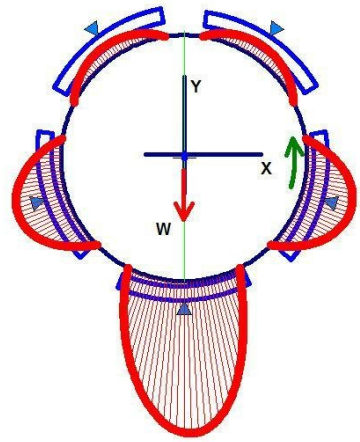

Figure 16 5 Pad Bearing Characteristics at 11,000 RPM
L/D = 0.458 , Cd = 5 Mils , PLF = 0.5

Figure 16 represents the bearing test case as shown in Figure 15 (***Table 2 Atkins & Perez***) with the 5 mil bearing clearance. The data shown represents the bearing characteristics computed for the rotor speed of 11,000 RPM. The vertical bearing stiffness is computed to be 378,700Lb/In. As previously noted, the fundamental shaft stiffness was computed to be 260,000 Lb/In.

This gives the fundamental bearing to shaft stiffness ratio of 2Kb/Ks = 2.91. This value is almost 3 times the optimal bearing to shaft stiffness ratio of 1. High rotor amplification factors would be expected with this bearing design. Thus bearing asymmetry would have little influence in improving rotor stability. This is shown in the response plots of Figure 7.

5 PAD LOP 3 In x 1.375 In, 0.23 Plf, Cb = 8.5 Mils Dia
L/D= 0.4583, Cb= 0.00425, 2Cb/D=0.00283, m= 0.23, Offset= 0.5, Arc= 54, PivAng = -18
Speed = 11000 rpm
Load = 264 Lbf
W/LD = 64 psi
Vis. = 1.5E-06 Reyns
Sb = 0.53525
E/Cb = 0.5855
Att. = 0.00 deg
hmin = 1.493 mils
Pmax = 296.207 psi
Hp = 1.93373 hp
Qside = 0.536 gpm
Stiffness (Lbf/in)
3.668E+04 0.000E+00
0.000E+00 2.771E+05
Damping (Lbf-s/in)
5.538E+01 0.000E+00
0.000E+00 1.841E+02
Critical Journal Mass (Lb)
Stable

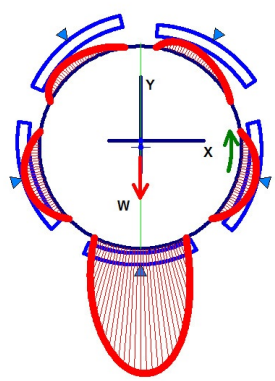

Figure 17 5 Pad Bearing Characteristics at 11,000 RPM
L/D = 0.458 , Cd = 8.5 Mils , PLF = 0.23

Figure 17 represents the bearing characteristics with the large bearing clearance of 8.5 mils and the reduced preload of 0.23. In this case the vertical bearing stiffness has been reduced to 277,000 Lb/In. Thus the bearing to shaft stiffness ratio is computed to be 2.13. This represents an improvement over the tight 5 mil bearing case which had a stiffness ratio of 3. Figure 6 shows a significant improvement in amplitude over the results as shown in Fig. 7.

Comparison of Theoretical and Experimental Damped Eigenvalues

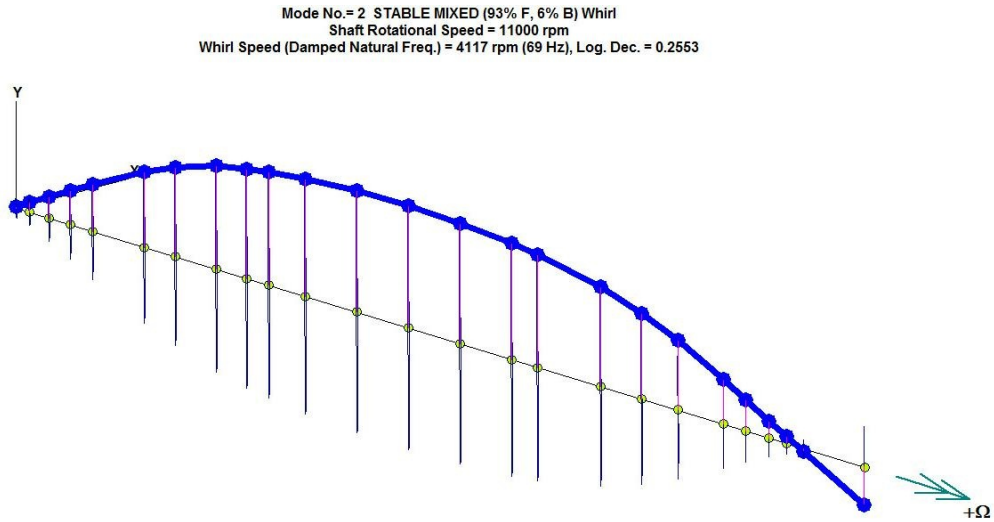

Figure 18 Computed Log Dec for Five Pad Bearing with 5 Mil Cd and 0.5 Preload at 12,000 RPM - LD = 0.255

Figure 18 represents the computed log dec for the five pad bearing with the 5 mil bearing clearance and 0.5 preload as shown in Figure 16. It is seen that the computed log dec based on the calculated bearing characteristics at 11,000 RPM produces a log decrement of 0.255. This value is over 3 times as high as the measured value of 0.082 as shown in Figure 15 taken from ***Table 2 Summary of Test Results*** of Atkins and Perez.

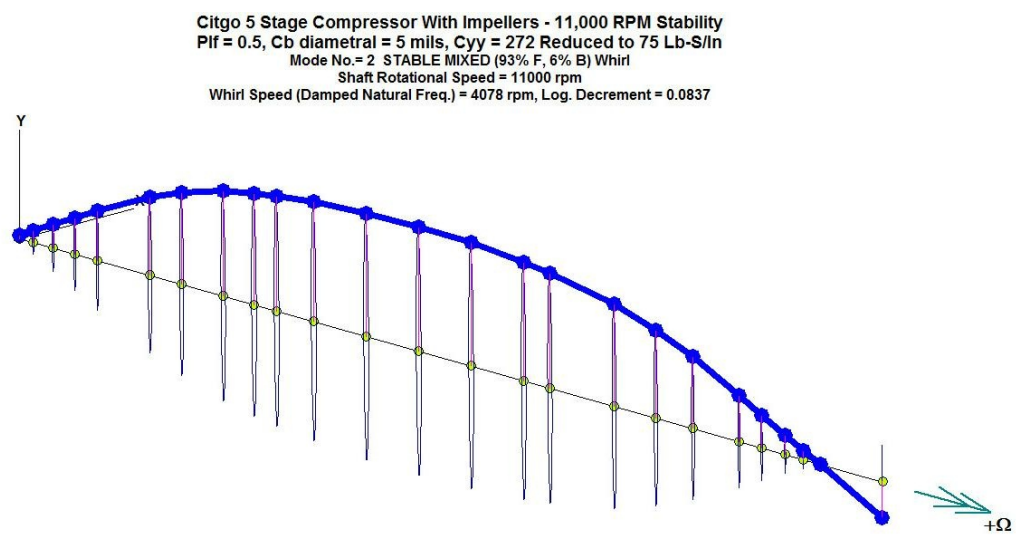

Figure 19 Computed Log Dec for Five Pad Bearing with 5 Mil Cd Using Reduced Cyy = 75 Lb/In at 12,000 RPM - LD = 0.0837

In order to match the low log decrement of 0.082 as determined experimentally, as shown in Figure 15, the damped eigenvalues were recomputed using a reduced vertical bearing damping of 75 Lb/In compared to the original value of 250 Lb/In. This value represents a ***70% reduction*** in the computed vertical damping of the five pad bearing with high preload!

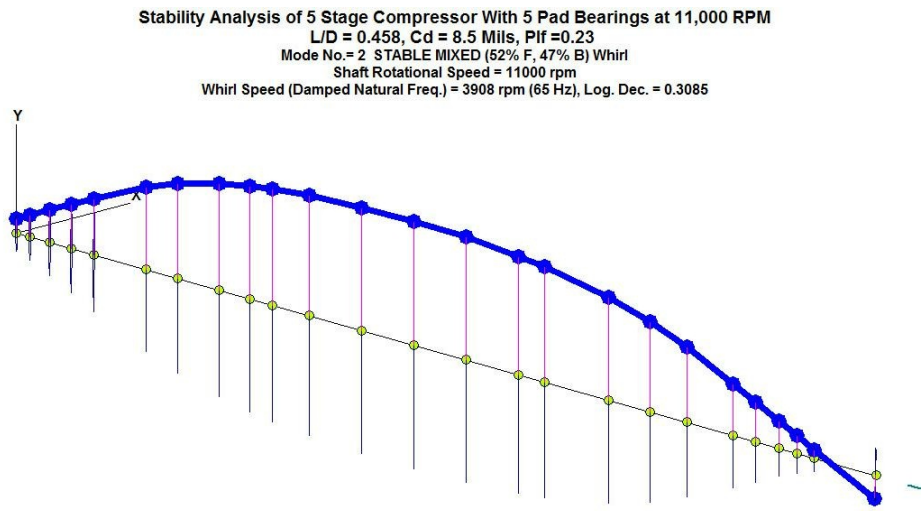

Figure 20 Computed Log Dec for Five Pad Bearing with 8 Mil Cd and 0.23 Preload at 12,000 RPM - LD = 0.308

Figure 20 represents the 1st damped forward mode for the case of the large 8 mil clearance bearing with the low preload of 0.23. Using the bearing characteristics as shown in Figure 17, the computed log decrement is 0.308. The measured log dec as shown in Figure 15 at 11,000 RPM is given as 0.123. Thus the computed log dec is two and half times higher than the actual measured test data with the large clearance bearing.

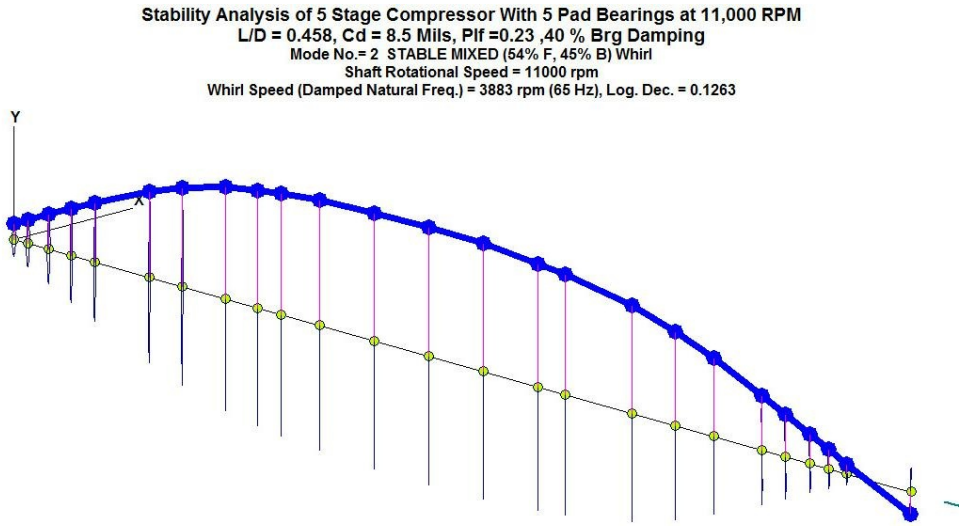

Figure 21 Computed Log Dec for Five Pad Bearing with 8 Mil Cd and 0.23 Preload at 12,000 RPM, 40% Cyy, LD = 0.126

Figure 21 represents the recomputed first forward damped mode with the large 8 mil clearance bearing and low preload of the 0.23. In order to be able to match the experimental data, the vertical bearing stiffness Cyy was reduced by 60%. The calculated log dec as shown in Figure 21 is 0.126 as compared to the measured value of 0.123.

Stability with Reduced Five Pad Bearing Damping Coefficients

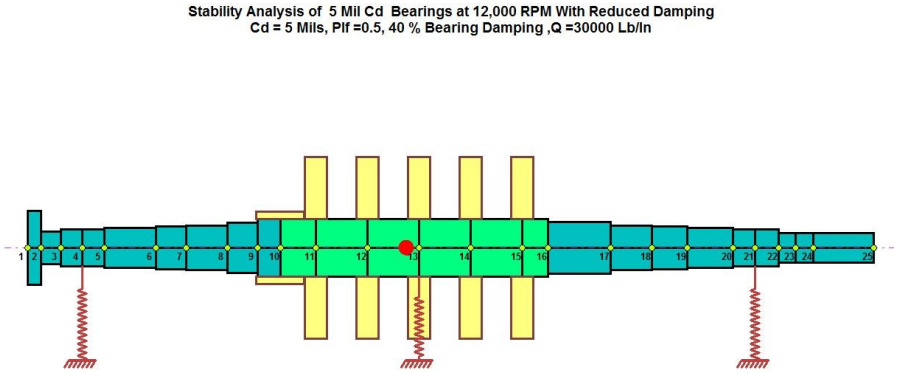

Figure 22 Five Stage Compressor Model Showing Aerodynamic Cross Coupling Acting at Rotor Center

Figure 22 represents the five stage compressor with aerodynamic cross coupling assumed acting at the center. The concept of aerodynamic cross coupling was first presented by Joel Alford of GE Gas Turbines in 1964. He demonstrated that turbines and compressors will develop cross coupling components when operating at high speed and high pressures. The aerodynamic cross coupling is represented by a bearing in which Kyx is equal to -Kxy.
For the five stage compressor, if we assume a cross coupling value of 6,000 Lb/In acting at each stage, then this would generate a total of 30,000 Lb/In which is sufficient to cause the rotor to become unstable as shown in Figure 23. These characteristics are similar to what was observed in the Kaybob compressors.

When rotors are tested in the factory under normal operating pressures, the aerodynamic cross coupling is not observed. This is only observed when operating under high compressor discharge pressures such as in petrochemical operations. Rather than place the aerodynamic cross coupling at all five stages, a modal amount is placed at the rotor center since it is the first mode frequency that is excited by aerodynamic cross coupling.

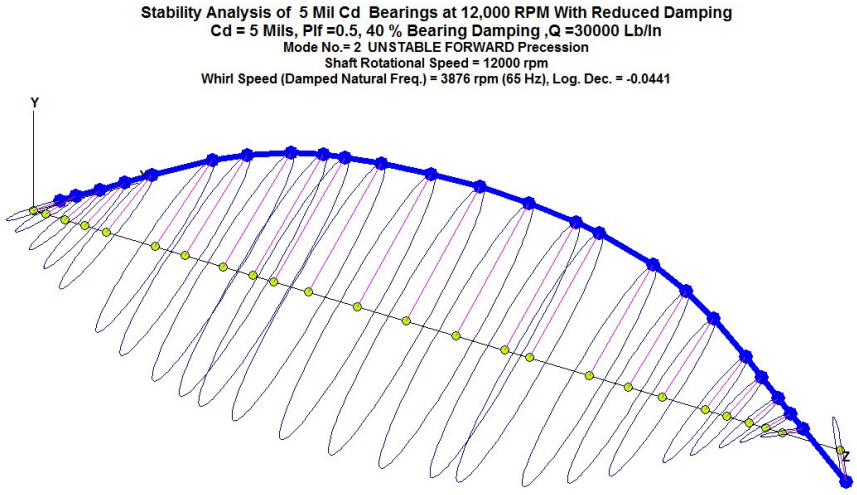

Figure 23 Unstable Compressor First Mode at 12,000 RPM with Aerodynamic Q = 30,000 Lb/In

Determination of Shaft Stiffness from Field Measurements

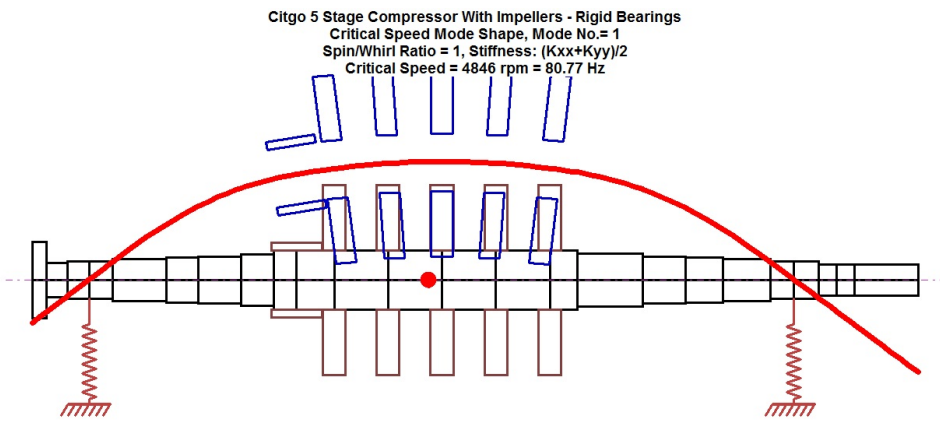

Figure 24 Natural Frequency on Rigid Supports

If a compressor in the field is to be retrofitted with a new tilting pad bearing, then it is important to be able to determine if that bearing is suitable for use with that particular compressor. With a replacement tilting pad bearing, the manufacturer will normally supply the user with the computed stiffness and damping characteristics.

It is therefore important to have an understanding of what the shaft stiffness value is to determine if the bearings are suitably matched with respect to the proper bearing to shaft stiffness ratio.

1. **Natural Lateral Frequency Measurement**

There are a number of ways to determine an approximate value of the shaft stiffness. One method is to mount the compressor on stationary V blocks and to perform a rap test to determine the first undamped natural frequency on fixed supports.

The shaft stiffness is computed from the following equation:

$$\omega_{cr} = \sqrt{\frac{K_{shaft}}{M_{Modal}}} \; ; \mathrm{K} = \mathrm{M}_{Modal} \left(\omega_{cr}\right)^2 \tag{1}$$

The modal mass for a uniform beam is one-half the total beam weight. A good approximation of the modal weight for a 5 stage compressor is to assume 75% of the weight.

$$K_{Shaft} = \left(\frac{0.75 \times 512 Lb}{386 Lb - Sec / In}\right) \times \left(507.4 rad / \sec\right)^2 = 256{,}155 Lb / In \tag{2}$$

The computed stiffness ratio of 2Kb/Ks for the 5 pad bearing with the tight bearing clearance of 5 mils and 0.5 preload is computed as 3.0. This is far in excess of the desired ratio of 1. This tight bearing clearance design would thus be immediately rejected for use in this compressor as a replacement bearing!

By opening up the bearing clearance to 8.5 mils and reducing the preload to 0.23, it is seen that the vertical bearing stiffness Kyy is reduced to 277,100 Lb/In. This produces a reduced bearing to shaft stiffness ratio of 2.2. This value is still not within the acceptable range.

2. Static Deflection Measurement

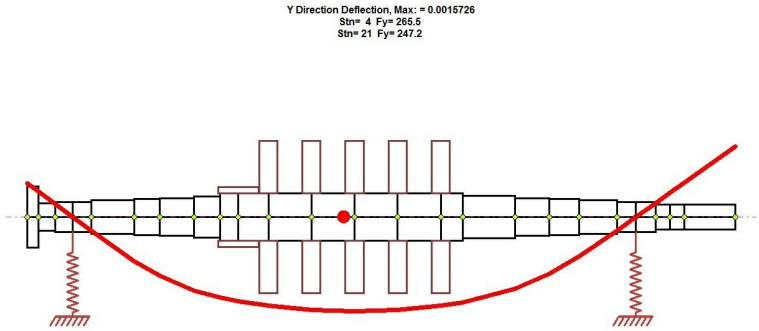

Figure 25 Shaft Static Deflection, 1.57 Mils

$$\omega_{cr} = \sqrt{\frac{g}{\delta}} = \sqrt{\frac{386}{0.00157}} = 495.8 rad/\sec; \quad K = M_{Modal} \times (495.8)^2 = 244{,}274\, Lb/In \tag{3}$$

A second method to determine the approximate shaft stiffness is to measure the static deflection of the shaft while resting on V blocks. The theory is based on simple spring-mass vibration theory. In this case it is seen that the measured static deflection is 1.57 mils. From the simple spring mass theory, an approximate shaft stiffness as shown above is computed to be 244,274 Lb/In.

3. Static Load Deflection

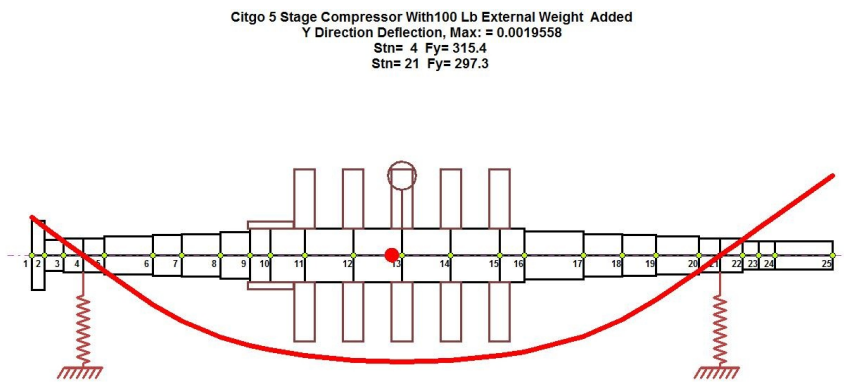

Figure 26 Rotor With 100 Lb External Weight Applied

$$F = K\delta = K(\delta_1 - \delta_0) = K(0.00196 - 0.001572) \tag{4}$$

$$K = \frac{100LB}{0.000388} = 257{,}732 Lb/In \tag{5}$$

It can be difficult to measure the gravitational sag of the shaft from the bearing centerlines. A third method to determine the relative shaft stiffness is to apply a known weight at the rotor center and measure the resulting static deflection. This proves to be a highly accurate method of quickly determining the shaft spring rate. In the above equations it is seen that the applied weight causes a 0.388 mil deflection. The computed shaft spring rate as show above is:

$$K_{shaft} = 257{,}732 \text{ Lb/In} \tag{6}$$

This value agrees well with method 1 based on the natural lateral frequency given as:

$$K_{shaft} = 256{,}155 \text{ Lb/In} \tag{7}$$

Stability Plots of Atkins and Perez

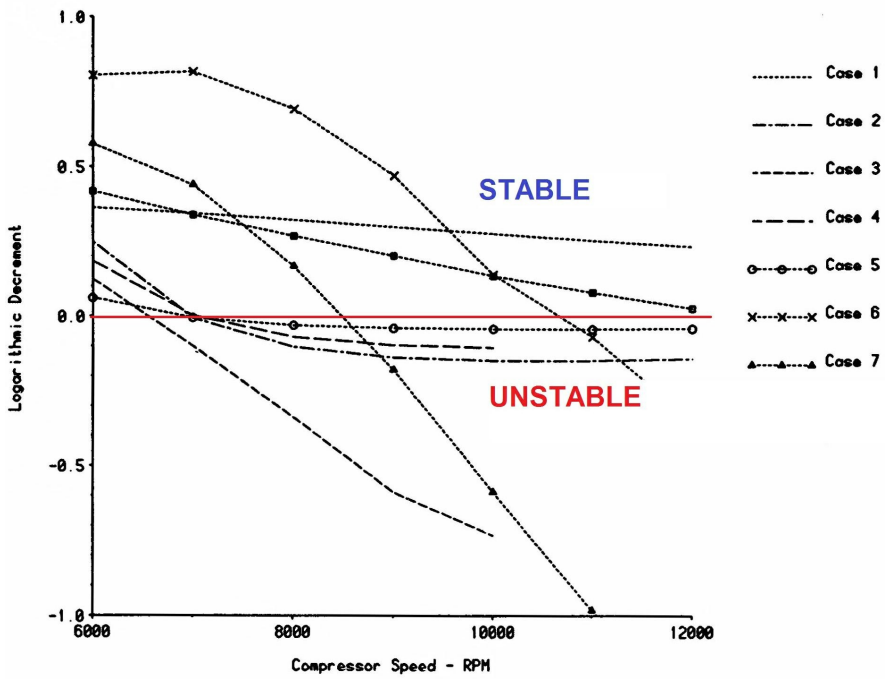

Figure 27 Stability Plot of Atkins & Perez for 5 Pad Bearing, L/D = 4.58 ***(Fig. 22, p 158)***

Figure 27 is a representation of the stability map as shown by Atkins and Perez, Figure 22 of their paper. The figure shows seven case studies but the stability map has eight cases plotted. It is apparent that by the time the authors reached this stage of their paper, it appears they had run out of time, space and energy to describe what their stability plots represented. There is no indication of what these bearing cases are that are shown in the plot or what assumptions were used in the stability analysis.

Of the 8 cases shown, it is seen that 6 cases are unstable at 7,000 RPM and below. Two cases appear to be stable throughout the operating range. It is apparent from the stability analysis of the system, that the tight bearing clearance case of 5 mils will have the lowest stability margin. The higher clearance bearing of 8.5 mils has an improved stability range, but it still would not represent a satisfactory design. The authors do point out a number of important stability considerations for proper design and operation, namely:

The critical speed ratio should be less than two
The amplification factor should be less than 5
The stiffness ratio should be less than 2

The authors never attempted to evaluate the five pad bearings in light of these criteria. For example, it is seen that the speed ratio of operating speed to critical speed with this design would be three. The amplification factors for the tight bearing clearance are over 38 for the 5 mil clearance bearing and over 25 for the 8.5 mil clearance bearing at 11,000 RPM. These values are in excess of the desired guidelines. The authors have shown by their stability map this bearing would be totally unsatisfactory for operation in high speed, high pressure conditions.

Compressor Stability with a Three Pad Bearing Design

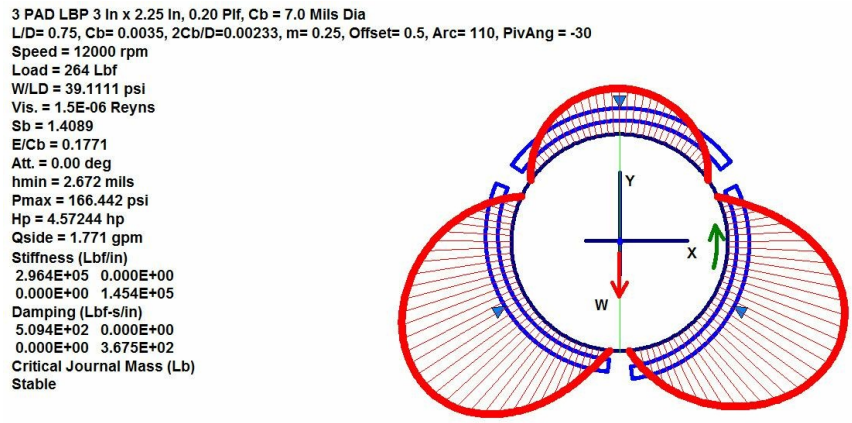

Figure 28 3 Pad Bearing Design With Load Between Pads
N =12,000 RPM, Cd = 7Mils, Plf = 0.20

Figure 28 represents the bearing characteristics for a three pad bearing at 12,000 RPM. The bearing design is load between pads and the bearing length has been increased to 2.25 inches. The computed vertical bearing stiffness of Kyy = 145,000 Lb/In is ideally suited to the shaft which has a stiffness of approximately 260,000 Lb/In. By having the load between pads with the three pad bearing design, a larger clearance and length of pads, we have an almost ideal bearing to shaft stiffness ratio for the five stage compressor operating at speeds of 12,000 RPM and higher.

The use of the three pad bearing has been seriously neglected by the compressor industry. It is impossible for a five pad bearing design with load on pad to satisfy the desired stiffness ratio required for high stability.

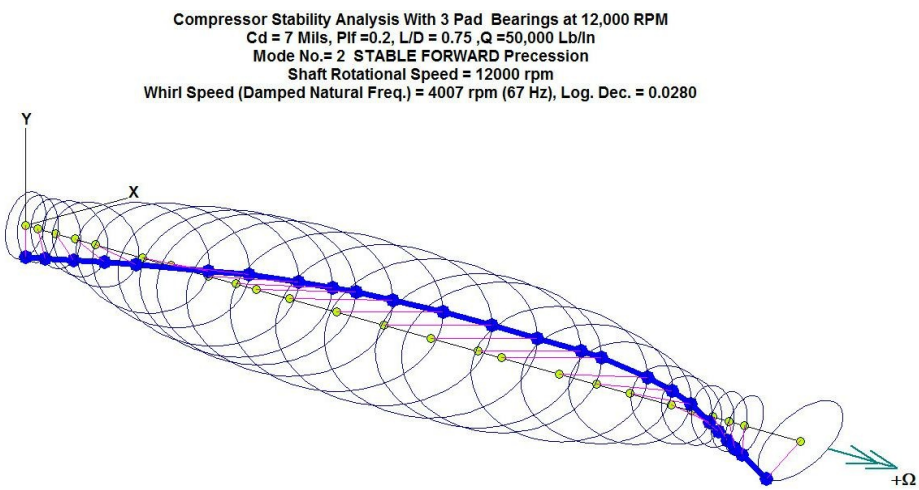

Figure 29 Rotor Stability with a Three Pad Bearing Design
N = 12,000 RPM, Q = 50,000 Lb/In

Figure 29 represents the stability calculation of the five stage compressor with the three pad bearing design. In this analysis the bearing coefficients as shown in Figure 28 were used for the stability analysis. Assuming an Alford cross coupling effect of 50,000 Lb/In, it is seen that the compressor is highly stable. This bearing design would eliminate the need for a squeeze film damper to promote stability.

Discussion and Conclusions

The paper presented by Atkins and Perez at the ***Twenty First Turbomachinery Symposium*** represents a monumental work concerning the experimental measurements of the bearing characteristics of a narrow five pad bearing for application in a 5 stage compressor.

The five pad bearing with load on pad has a very narrow L/D ratio of 0.458. This is a very common bearing that had been used in the petrochemical industry for many multistage compressors operating at high speed and high pressure ratios. The reason for the extensive testing was because of the concern of the stability characteristics of this bearing.

The design of the five stage Citgo compressor is in many ways reminiscent of the Kaybob compressor of Alberta presented at the Texas A&M ***Fourth Turbomachinery Symposium***. Before the Kabob stability problems were corrected, they had incurred losses of over $100 million in today's currency.

In the paper, the authors present a computer model for the rotor and show the bearing design. However, no details of the rotor critical speeds, bearing coefficients, or damped eigenvalues (stability calculations) are presented by the authors.

The paper generated a number of surprises when comparing the experimental data with the computed rotor critical speeds and the bearing coefficients. For example, using the rotor model as shown in Figure 8, an initial critical speed of 3,700 RPM was computed. The experimental data, such as shown in Figure 7, shows that with the tight 5 mil bearing clearance, the first mode could be over 4,300 CPM. The apparent discrepancy between the original computed critical speed of 3,700 RPM and the measured value of 4,300 CPM at 11,000 RPM is attributed to the stiffening effects of the impellers.

The authors externally excited the compressor over a range of speeds with the shaker system as shown in Figure 4 and 5. What is seen in all cases is that as the rotor speed is increased, the amplification factor of the excited resonance frequency is higher at the higher speeds. The worst-case observed was at 11,000 RPM with the tight 5 mil bearing clearance. The amplification factor increased from 28.9 at 7,000 RPM to 38.5 at 11,000 RPM.

The best case obtained (which is still too excessive!) was with the large clearance bearing of 8.5 mils and a low preload of .23. This bearing recorded an amplification favor of 9.21 at 7,000 RPM but rapidly increased to 25.64 at 11,000 RPM. These values are deemed excessive and are not suitable for use in multistage compressors operating at high speeds with high discharge pressures. Under these conditions the compressor cross-coupling coefficients caused by Alford effects become significant.

The bearing coefficients at 11,000 RPM for the tight bearing clearance case of five mils and the larger clearance of 8.5 mils were computed. Using these bearing characteristics, the rotor amplification factors were predicted. In order to match the amplification factors as observed in the experimental data, the vertical damping for the tight bearing clearance had to be reduced by 70% and for the large clearance bearing of 8.5 mils, the damping had to be reduced by 60%.

A large reduction in the computed damping was necessary in order to match the experimental data. This was startling but not totally unexpected. A number of years ago, Charlie Jackson of Monsanto and Malcolm leader deliberately surged one of their compressors with a five pad bearing configuration with the load-on-pad design. They measured the log decrement from the transient response curve. They computed that in order to match the experimental data with the computer predictions, that the computed bearing damping had to be reduced by over 50% in order to match the experimental data.

At the time this brought up great concerns of the accuracy of the tilt pad bearing program. At that time I contacted Dick Elwell of the GE Bearing Lab to review our bearing calculations. He computed the bearing coefficients using the GE code and reported that he had obtained similar bearing values to the ones that we had generated with our finite element tilting pad bearing program.

This result then launched us into the investigation of the influence of the flexibility effects of pivot and foundation on tilting pad bearing stiffness and damping coefficients. For example, with added support flexibility, the stiffness reduces similar to springs in series; however, the damping significantly reduces due to support, pivot and pad flexibility and speed. The effects of this can be quite dramatic. For example, with the Westinghouse W501 gas turbines, a flexible support is used to support the four pad bearing at the hot turbine end. The resulting flexibility of the support reduces the bearing damping by over 90%. This great reduction in damping at the gas turbine location is what leads to such unusual dynamic characteristics observed in the field.

With the large Westinghouse nuclear water pumps with three pad bearings supporting the motor, it was observed that pivot and support flexibility caused over an 80% reduction of the effective motor bearing damping.

The tilt pad bearing design with load on pad has been used for a number of years because of its seemingly superior stability characteristics. Given that the most important characteristic for a tilt pad bearing is the matching of the pad vertical stiffness to the shaft stiffness, if the bearing stiffness to shaft ratio of 2Kb/Ks exceeds 1, then bearing asymmetry plays an insignificant role in improving rotor stability.

As five pad bearings became more popular in industry for replacement of plain journal bearings, the load on pad design was favored because of the low vibration amplitudes of motion recorded at the bearings during factory testing and certification. Also the narrow L/D pad design had lower horsepower losses. On testing, the multistage compressors mounted in the five pad bearings with low L/D tested adequately under ambient test conditions, however once placed in the field at high speeds over 12,000 rpm and under high discharge pressure conditions, the Alford effect would cause these rotors to become unstable resulting in large amplitude whirl motion which, in turn, resulted in seal and rotor failures. Squeeze film dampers were then necessary to correct problems these whirl problems at great expense.

The whole purpose of the squeeze film damper is to provide a soft support so that the damping may become effective. For example, for the five stage compressor analyzed, the shaft stiffness is approximately 250,000 Lb/In, therefore a damper spring rate of 125,000Lb/In or less is desirable. A squeeze film damper is incorporated with a 5 pad bearing design in order to compensate for the poor characteristics of the five pad bearing.

The incorporation of a squeeze film damper with a five pad bearing to promote stability is essentially an attempt to make a ***silk purse out of the sows ear!*** as the saying goes. Many of the current units that have squeeze film dampers could have been corrected by using a properly designed four or three pad bearing design with load between pads.

It is apparent that if the five stage Citgo compressor with the L/D ratio of 0.458 had been put into operation similar to Kaybob at 14,000 RPM with high discharge pressure, then disaster would have occurred! In general, the five pad bearing, regardless of L/D ratio , clearance or preload, is usually poorly suited for use in multistage compressors operating at speeds over 12,000 RPM. Very rarely does the five pad bearing with load-on-pad have the proper bearing stiffness to shaft ratio desired. The five pad bearing has simply been over-used in industry and has few of the qualities required to promote adequate rotor stability.

Most of the designs that required a squeeze film damper support could have been eliminated by the use of a properly designed three or four pad bearings with a longer L/D ratio. When inquired of senior management why they did not incorporate the three or four pad bearing design with their compressors instead of the five pad bearing, the response was they preferred the lower amplitudes recorded with the five pad bearings on the factory certification tests. End of discussion on the subject!

In sum, a five pad bearing design with load-on-pad should never be incorporated into a high-speed compressor unless the stiffness to shaft ratio has been properly computed. The correct bearing stiffness to shaft ratio is the most important parameter of a tilting pad bearing for high-speed centrifugal compressor operating under high discharge pressures. Bearing asymmetry has little to no influence in promoting rotor stability if a proper bearing to shaft stiffness ratio is not first obtained. This never seems to be the case with the use of five pad bearings. The three or four pad bearings have superior stability characteristics over all five pad bearings.

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