Ekofisk Revisited - Bearing Optimization for Improved Rotor Stability

Edgar J. Gunter
Prof. Emeritus Mechanical and Aerospace Engineering, Univ. Of Virginia
Former Director Rotor-Bearing Dynamics Laboratory
Fellow ASME

Abstract

In this paper, the stability characteristics of the original Ekofisk rotor are examined and how the stability could have been greatly improved through the use of squeeze film dampers or four pad LBP (Load Between Pad) bearings. The emphasis of the study is directed towards the bearings and how they effect the rotor stability. The previous papers written on Ekofisk have mainly concentrated on the evaluation and improvement of the seals in order to increase rotor stability.

The renewed interest in the Ekofisk project was made possible from the extensive paper recently presented by C. Hunter Cloud, Brian Pettinato and John Kocur on ***Predicting, Understanding and Avoiding the EKOFISK Rotor Instability 40 Years Later*** *(9). This paper presented extensive details on the rotor designs and the bearings used in the various compressor configurations. The original Ekofisk LOP (Load On Pad) 5 pad bearing design had a narrow aspect ratio of L/D of 0.284 This represents a bearing design with an aspect ratio that was less then what was installed in the unstable failed Kaybob compressor (12,21).*

In the early 1970s, several compressor manufacturers were encouraged to switch to a load between pad bearing design (LBP). This concept was rejected for the following reasons. First, with the load on pad bearing design, the vibration probes mounted in the bearing caps would register a lower vibration level than would be with a load between pad bearing design.

The primary reason stated, however, was that the reduced bearing vertical stiffness values would place the second critical speed in the operating speed range. This then would be in direct conflict with the 1969 API Code 612 initiated to protect petrochemical rotors from high vibrations. A secondary consideration for the 5 pad LOP design was the incorrect assumption that bearing asymmetry would improve rotor stability (14,15,18).

In general, it was determined that the 4 pad LBP bearing design is superior to the 5 pad LOP design for stability. Also, squeeze film dampers for multi-stage compressors are only required to compensate for the poor stability characteristics of the 5 pad LOP design.

Resume

Edgar J. Gunter, PhD

Dr. Gunter is a retired Prof. of Mechanical and Aerospace Engineering at the University of Virginia. He received his Mechanical Engineering degree from Duke University and Masters and PhD degrees from the University of Pennsylvania in Engineering Mechanics.

He was employed as a centrifugal compressor design engineer for four years at Clark Brothers, Olean, New York, now a division of Dresser-Rand. Based on his compressor design projects, he was awarded a National Defense Fellowship to pursue the PhD degree in Engineering Mechanics.

During his graduate studies, he received an internship with the SKF Ball Bearing Research Center to study fatigue life of rolling element bearings. In his graduate program, he majored in applied mathematics, vibration and dynamics, fluid mechanics and lubrication theory.

After completing his formal training at the University of Pennsylvania, he assumed the position of Senior Research Scientist at the Franklin Institute Friction and Lubrication Laboratories in charge of the Gas Bearing Division. While at the Franklin Institute, he received a NASA Lewis Research Grant to study rotor - bearing stability. The study was initiated since at that time the Franklin Institute had some of the world's largest digital and analog computers at the Institute. The report on Rotor Bearing Stability was published by NASA as a special CR report and given national distribution. This report formed the basis of his PhD dissertation.

Upon receiving his formal PhD degree, Dr. Gunter was then offered the position of tenured Associate Prof at the University of Virginia. At the University of Virginia, he developed the Rotor Bearing Dynamics Laboratory to assist industry in the development of reliable high-speed rotating equipment.

He has been elected to the following honorary engineering societies of Pi Tau Sigma, Tau Beta Pi and Sigma Xi. He was elected as a fellow of ASME in 1996.

In 2008, Dr. Gunter was awarded the first Jack Frarey Memorial Metal by the Vibration Institute for contributions to the field of rotor dynamics.

Background and Introduction

The Kaybob compressor failure of 1971 and the Ekofisk failure in 1974 represented a new class of failures encountered with the advent of high pressure discharge compressors. These failures were not caused by self-excited rotor instability due to the fluid film bearings, but from the Alford effects created by the cross-coupling forces developed by turbines and compressors under power, as reported in 1965*(3).* Both of the rotors in question were supported by narrow five pad, load on pad (LOP), tilting pad bearings.

Lund*(16)* in 1964 presented a definitive paper on the stiffness and damping characteristics of tilting pad bearings. The use of tilting pad bearings allowed compressor designers to increase rotor operating speeds to higher levels that were not previously possible with the use of plain journal bearings.

Lund at this time, was also able to take the transfer matrix method of Prohl on critical speed analysis and adapt it to the digital computer. He later expanded the undamped transfer matrix theory to include bearing damping in order to compute synchronous rotor unbalance response. However, at the time that Kaybob and Ekofisk were designed, no computer programs existed to calculate rotor 3-dimensional complex damped eigenvalues, including disk gyroscopic moments, to compute modal log decrements, or rotor stability with Alford forces acting on the impellers *(17,8)*.

At that period of compressor development, there was great concern about operating at or near the rotor second critical speed. As such, the API code 612*(1)* presented in 1969, stated that turbines and compressors should not operate in the critical speed region. Nicholas*(20)*, later in 1989, reviewed how the API codes 612 and 617 were later updated to allow operation through the second critical speed. As a result of the desire to avoid the second critical speed in both of these rotor designs, narrow five pad, load on pad (LOP), bearing designs were chosen.
At this same time, the Bently noncontact proximity probes were now being installed in the bearing housings to monitor rotor motion. Under standard test conditions, with ambient pressure, no initial problems were detected with either compressor design.

The renewed interest in the Ekofisk project was made possible from the extensive paper presented in 2018 by C. H. Cloud, B. C. Pettinato and J. A. Kocur*(9)* on ***'Predicting, Understanding and Avoiding the EKOFISK Rotor Instability 40 Years Later "***. The Ekofisk paper presented extensive details on the rotor designs and the bearings used in the various compressor configurations.
An initial finite element computer model of the Phase I rotor was developed based on the rotor model and bearing coefficients presented in reference *(9)*, with the addition of disk gyroscopic effects *(8)*. The original computed first critical speed was at 3,462 rpm. The reported whirl frequency was 4,400 cpm *(9)*. It was speculated that the seals contributed a stiffness of 500,000 Lb/In in order to achieve the observed whirl frequency. The original critical speed calculations by the transfer matrix method appears to be substantially in error. Considerable effort was spent examining seal performance and on seal modifications. In reality, the seals had a very minor effect on the observed instability frequency.
An initial damped eigenvalue analysis was computed for the Phase I rotor with the 5 pad bearing coefficients as presented in *(9)*. The resulting log decrement at 8,500 rpm was computed to be 0.18 with a whirl frequency of 4,352 rpm.

A stability analysis, including aerodynamic cross coupling with the original bearings, indicated that the rotor would be unstable with a cross-coupling value of Q = 20,000 Lb/in. According to the experimental research of Atkins and Perez *(2,13)* on 5 pad, load on pad (LOP) bearings, the actual damping achieved with load on pad bearings may be less than 50% of the predicted analytic values. This would place the original stability limits of the Ekofisk rotor with a Q value of only 10,000 Lb/In.
The original Ekofisk load on pad 5 pad bearing design had a narrow aspect ratio of L/D of 0.284. This represents a bearing design with an aspect ratio that was less then what was installed in the unstable failed Kaybob compressor *(12,21).*

In the early 1970s, several compressor manufacturers were encouraged to switch to a load between pad bearing design (LBP). This concept was rejected for the following reasons. First, with the load on pad bearing design, the vibration probes mounted in the bearing caps would register a lower vibration level then would be with a load between pad bearing design.
The main reason for the rejection of a 4 pad (LBP) design was stated that the reduced bearing stiffness would place the second critical speed in the operating speed range. This then would be contrary to the1969 API Code 612 *(1)*. A secondary reason for the load on pad design was the incorrect assumption that bearing asymmetry would improve rotor stability *(14,15).*

In the ongoing analysis of Ekofisk I, the bearing coefficients for the narrow five pad bearing were re-computed with a moderate preload of m =0.3. A revised rotor model was developed with a bearing support stiffness value of K_f= 2.8e6 Lb/In. It should be noted that the addition of a flexible support under the tilting pad bearings cannot be accurately formulated using the transfer matrix theory.

A stability analysis was conducted with this model showing that at a speed of 8,500 rpm, the maximum value of Q at the threshold of stability was 19,000 Lb/In. The computed whirl frequency was 4,407 cpm which matches the observed experimental whirl frequency exactly! It therefore may be concluded that the seals had no influence on the observed whirl frequency. The investigation into the seal behavior was basically" ***a wild goose chase***".

Originally, a squeeze film damper was attempted to correct the stability problem. It had limited success. An analysis is presented that shows a properly designed a squeeze film damper would allow an aerodynamic excitation Q of ~ 100,000Lb/in. Failure to add the central groove to a squeeze film damper has resulted in the failure of several previous attempts to apply squeeze film dampers. This generates a long bearing damper which creates excessive damping and stiffness. The resulting situation is referred to as damper lockup.

The bearing coefficients for a wider four pad LBP bearing were computed. A wider LBP bearing would allow an aerodynamic loading of Q ~ 90,000 Lb/In. A damped eigenvalue analysis of the system shows that the second critical speed is nonexistent due to the damping. However, such a design in 1972 would have been in violation of the 1969 API 612 Code as it theoretically places the second critical speed in the operating speed range. As can be seen from a damped eigenvalue analysis that this would not be a problem. However, at that time, computer codes did not exist to compute damped eigenvalues for higher order modes *(8,17)*. Thus, the additional rotor redesigns for Ekofisk would not have been necessary if either a properly designed squeeze film damper had been applied or if they were allowed to use a wider four pad LBP bearing.

Ekofisk High Pressure Reinjection Compressor

Ekofisk was a high pressure gas reinjection centrifugal compressor for the Phillips Group Norway for their North Sea Ekofisk reservoir. The maximum design speed was 8426 RPM with a maximum discharge pressure of 9200 psi. The unit was basically an eight stage barrel compressor designed by Elliott in 1972 with four stages separated by a high pressure balance piston seal as shown in Figure 1 *(10)*.

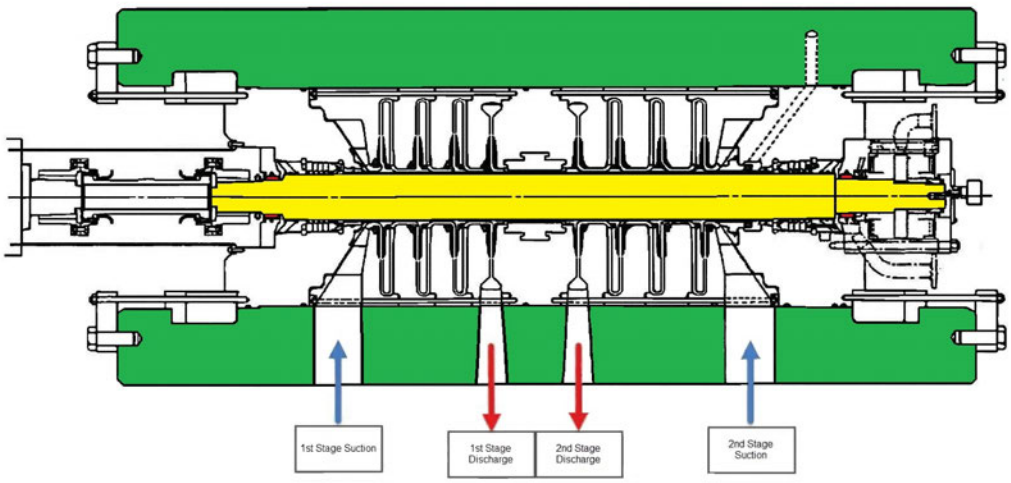

Figure 1 High Pressure 8 Stage Reinjection Compressor
(Geary, Damratowski, and Seyer, 1976)

In June 1974, the gas injection train started under load. Upon reaching minimum governor speed, the compressor tripped on high vibration levels. The plot of the frequency spectrum is shown in Figure 2.

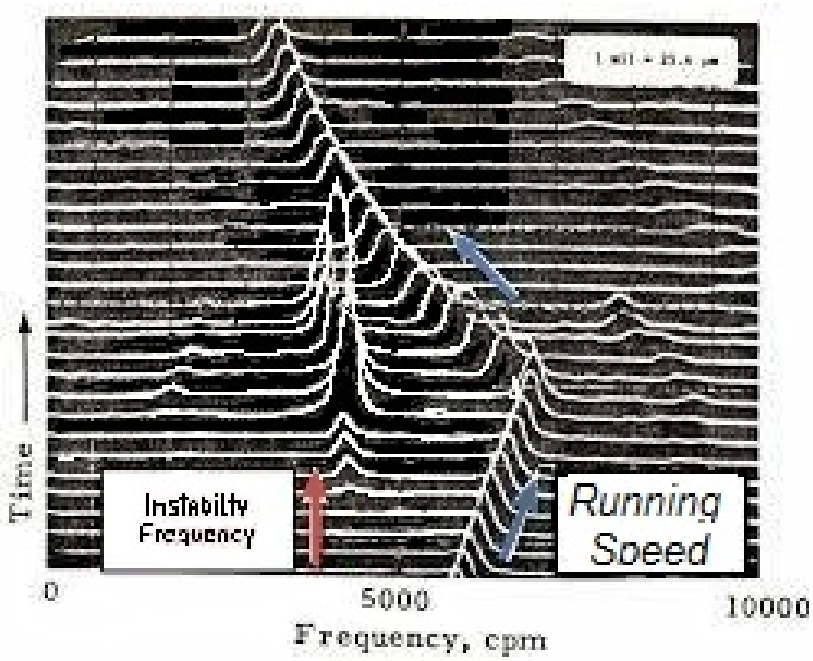

Fig. 2 High Subsynchronous Vibrations on Start Up
(Wachel 1975)

Wachel stated in 1975 that the calculated first critical speed of the rotor was 3800 cpm for a bearing span of 81 inches. The frequency of the nonsynchronous whirl instability was recorded to be 4400 cpm as shown in Fig. 2. It was stated that this recorded whirl frequency was higher than the calculated rigid bearing critical speed of 4200 cpm. It was also claimed that this must have been caused by the floating oil seals to be locked up, thereby effectively reducing the bearing span. A computer simulation of the rotor was performed with the seals assumed to have a stiffness of 500,00 lb/in required in order to match the experimental data.

It will be shown that the original calculations of the shaft critical speeds are ***totally incorrect***! A current analysis of the Ekofisk rotor was performed based on the recent extensive paper on the subject presented at the 47th TUBOMACHINERY Symposium in 2018 by C. Hunter Cloud, Brian C. Pettinato, and John A. Kocur *(9)*. In this paper are shown the original rotor configuration as well as the redesigned rotors, referred to as EKO IV and EKO Modern.

Using the modern finite element based software program, ***DyRoBeS***, the first critical speed of the original rotor design was computed to be 4,352 RPM, and the rigid rotor critical speed was computed to be 4,792 RPM. Using the 5 pad bearing data provided in the recent Ekofisk paper, the whirl instability of 4,353 rpm initially calculated without the aid of the oil seals. Details of the critical speeds and stability analyses for the various rotor designs are presented in later sections.

It is important to note that the extensive analysis conducted on the various floating seal designs was of importance, but they did not have a significant impact on stabilizing the compressor. The extensive studies of the seals were conducted because of the erroneous calculations of the rotor critical speeds. The key element leading to the highly unstable Ekofisk compressor was similar to that which was encountered with the Kaybob rotor. The problem in both systems was the use of narrow five pad bearings with load on pad design. This presented a situation in which these rotors became highly susceptible to self excited whirling due to the Alford effect.

It will be shown that the instabilities encountered with both Kaybob and Ekofisk rotor systems and their subsequent rotor redesigns were due to the strict adherence to the API Codes 612 & 617 of 1969. In these codes, it was stated the requirement to avoid operation at or near the second critical speed regardless of the sensitivity of the second critical speed or the computed value of the 2nd mode log decrement.

It is now apparent that these earlier API codes, which were enacted to ensure reliable rotor designs, were themselves responsible for millions of dollars of losses in the petrochemical industry and required unnecessary rotor redesigns with stiffer shafts and reduced bearing spans. Both the Kaybob and Ekofisk rotors could have easily been stabilized with the use of larger four or three pad bearings with load between pads. These bearing changes would not have been allowed at that time because it would have placed the second critical speeds of these rotors in the operating speed range.

An important paper on this subject was presented by John C. Nicholas in 1989 on '***Operating Turbomachinery on or Near the Second Critical Speed in Accordance with API Specifications"*** at the Eighteen Turbomachinery Symposium *(20)*. In this paper, Nicholas explains the operation of rotors near the second critical speed and the corresponding changes in the API codes 612 and 617 on allowing rotors to operate on or near the second critical speed. It is apparent that the highly deficient, narrow five pad, load on pad bearing designs were forced on the compressor manufacturers in order to meet the unrealistic API codes of 1969.

EKOFISK Rotor Configurations

In the review paper on the Ekofisk instability problem *(9)*, three rotors Are reviewed as shown below in Figure 3.

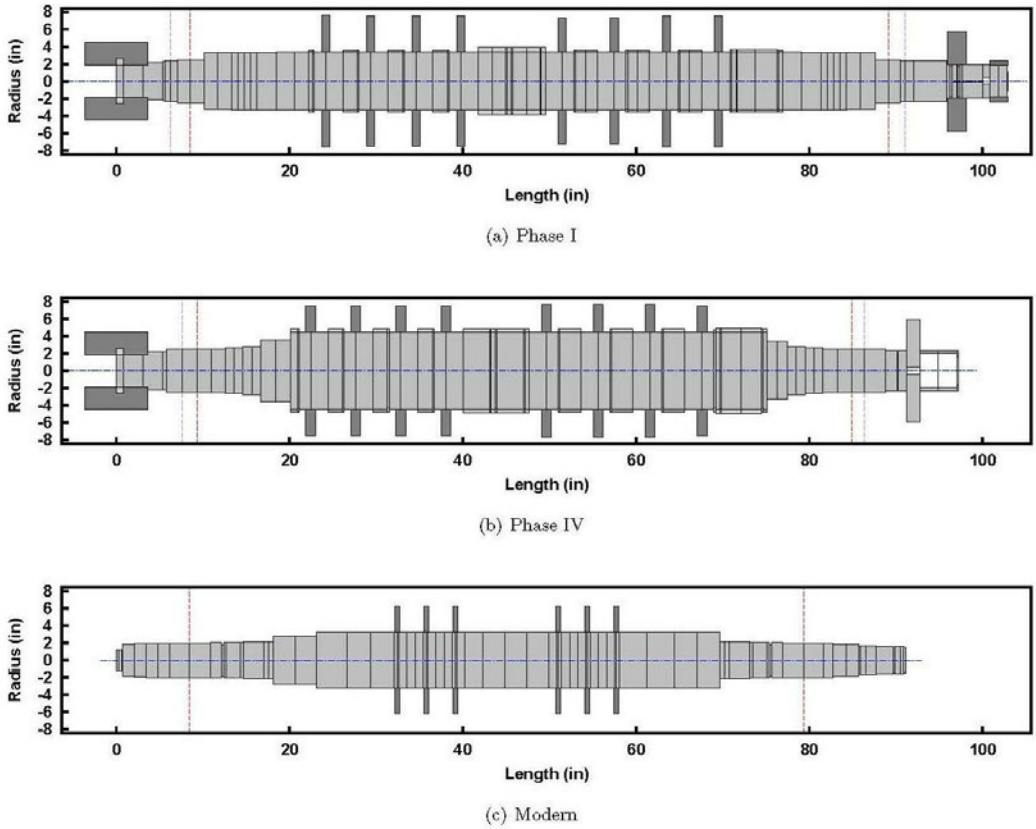

Figure 3 EKOFISK Rotor Configurations
(Cloud, Pettinato & Kocur, 2018)

The first rotor configuration shown, Phase I, represents the original 8stage rotor design. The second rotor shown, labeled as Phase IV, was also an 8 stage design and was a significant improvement in rotor stability. The final rotor shown, labeled as Modern, was designed to operate at a higher speed with only six stages.
Table 1 represents the various rotor properties as shown in Fig. 3 *(9)*.

	Phase I	Phase IV	Modern
Bearing Span (L)	80.7 in (2049 mm)	75.6 in (1920 mm)	71.0 in (1803 mm)
Main Diameter (D)	6.7 in (171 mm)	9.0 in (229 mm)	6.4 in (163 mm)
Shaft L/D	12.0	8.4	11.1
OSR (rpm)	6741–8847	6741–8847	9636–12647
CSR (dim)	2.35	1.5	2.59

Table 1 Rotor Properties *(7)*

In the Phase IV rotor design, it is seen that the bearing span has been reduced from 80.7 in. to 75.6. In addition, the effective shaft diameter D, has been significantly increased to 9 inches. This reduces the L/D ratio of the original design from 12 to 8.4. This design was an apparent effort to raise the second critical speed to well above the operating speed range with the new load between the bearing design referred to as LBP.

The final design referred to as Modern, has a reduced bearing span of 71 inches and a slightly smaller shaft diameter. It was designed with only six stages and to operate at a higher speed.

EKOFISK Original Phase I Rotor Critical Speed Analysis

Fig. 4 represents the computer model of the Phase I rotor generated by the Dyrobes finite element software.

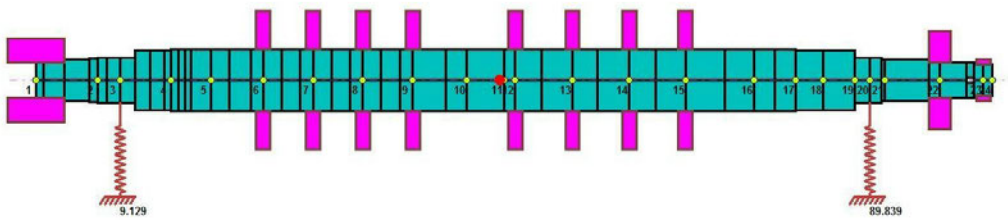

Figure 4 EKOFISK Original Phase I Rotor Configuration
Shaft Modal Stiffness, Ks= 270,000 Lb/In
Nrigid brgs = 4,792 RPM

Phase I Rotor Model Summary	
1	Shaft
23	Elements
51	SubElements
3	Materials
0	Unbalances
11	Rigid Disks (4 dof)
0	Flexible Disks (6 dof)
2	Linear Bearings
0	NonLinear Bearings
0	Flexible Supports
0	Axial Loads
0	Static Loads
0	Time Forcing Functions
0	S.S. Harmonic Excitation
24	Stations
96	**Degrees of Freedom**

Table 2 Summary of Phase I Rotor Computer Model

Table 2 shows the features of the Phase I rotor computer model. The Phase I model consists of 24 stations with 96 degrees of freedom and 11 disks with disk gyroscopic moments included. The disc gyroscopic effects do not appear to have been included in the earlier analyses.

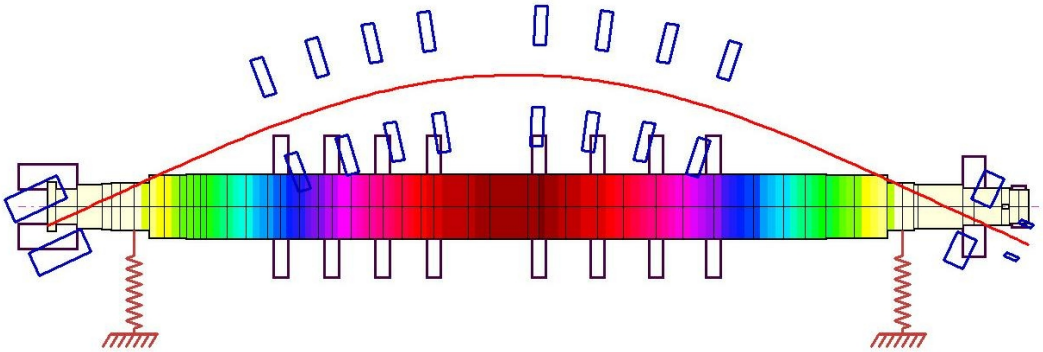

Figure 5 Phase I Rotor First Critical Speed
Kbrg =1.0e6 Lb/in , N1 = 4,352 RPM

The reported 1st critical speed was stated as 3,800 cpm and the rigid bearing critical speed was listed as 4,200 cpm. The original computed rotor critical speed values are considered to be in considerable error. The whirl frequency was recorded as 4,400 cpm, which corresponds closely with the first mode as computed by the Dyrobes finite element based software. More details on the stability calculations will be presented in the following sections. It is apparent from the original errors in the computations of the rotor first mode, that the conclusion of 500,000 Lb/in of seal stiffness was required to match the experimental data. This assumption is completely in error as it masked the real problem .

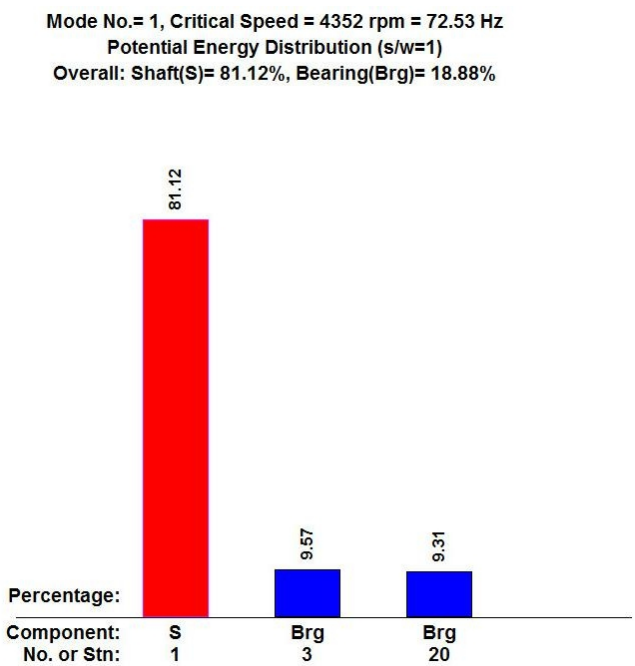

Fig. 6 1st Mode Rotor Strain Energy Distribution

Figure 6 represents the strain energy distribution for the first mode. With the assumed average bearing stiffness of 1e6 lb/in, over 81% of the system strain energy is in shaft bending. Each bearing has less than 10 % strain energy. Hence, the bearings can not be very effective in damping out both synchronous and not synchronous vibrations. To be effective, each bearing must have at least 20% strain energy for that mode.

EKOFISK Phase I Rotor Second Critical Speed Analysis

It is very important to review the second critical speed calculations for the Ekofisk Phase 1 original rotor design. It will be seen that the narrow five pad LOP bearing designs were forced on both Kaybob and Ekofisk in order to meet the stringent API Code 612 of 1969. Below is shown the predicted rotor second critical speed for a nominal bearing stiffness of Kb = 1.0e6.

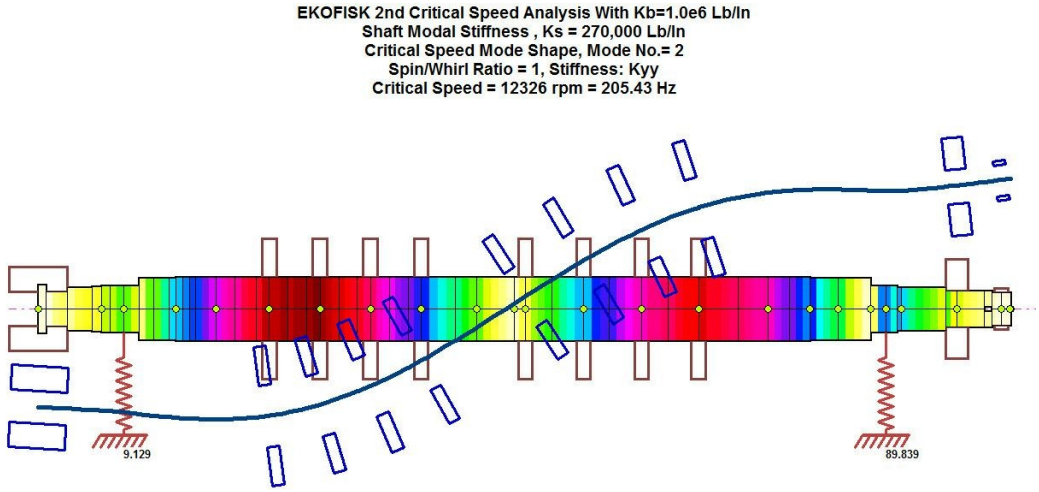

Figure 7 Phase 1 Rotor Predicted Second Critical Speed with Kb=1.0 Lb/In
Nc2 = 12,326 RPM

The predicted rotor 2nd critical speed is at 12,326 RPM and is well above the design operating the 8,400 RPM. The corresponding strain energy for the shaft and bearings for the 2nd critical speed is shown in Figure 8 below.

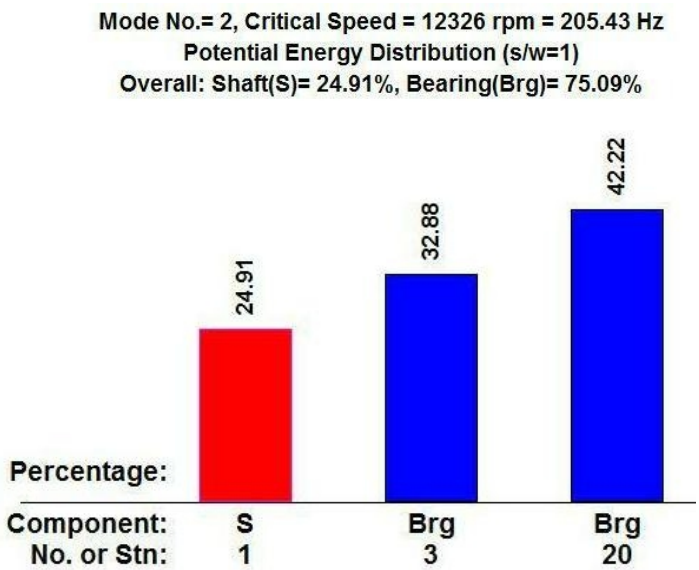

Figure 8 Rotor I 2nd Mode Strain Energy

It is seen that the bearing second mode strain energy is 75%. This is in sharp contrast to the less than 20% bearing strain energy as seen in the first mode with Kb = 1.0e6 Lb/In. Thus, the second mode, with the shaft moving out of phase, will be well damped.

EKOFISK Phase I Rotor Critical Speed Map

Figure 9 represents the critical speed map for the Ekofisk Phase I original rotor design.

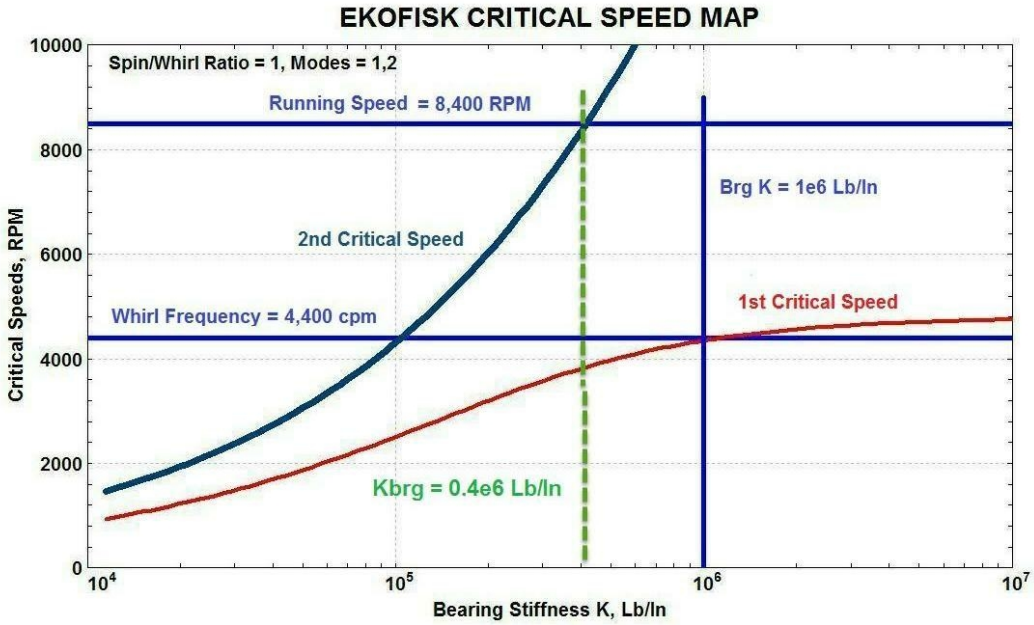

Figure 9 Phase I Rotor Critical Speed Map

In the Phase I rotor design, with the bearing stiffness vertical stiffness at approximately 1e6 lb/in, the predicted first critical speed corresponds closely with the observed frequency of 4,400 CPM. The second critical speed is predicted to be over 12,000 RPM, which is well above running speed.

If a lower vertical bearing stiffness, such as 0.4e6 lb/in, is selected in order to provide better bearing damping, then it is seen that the second critical speed would be placed near the operating speed of 8,400 rpm. To do this would be in violation of the 1969 API code 612.

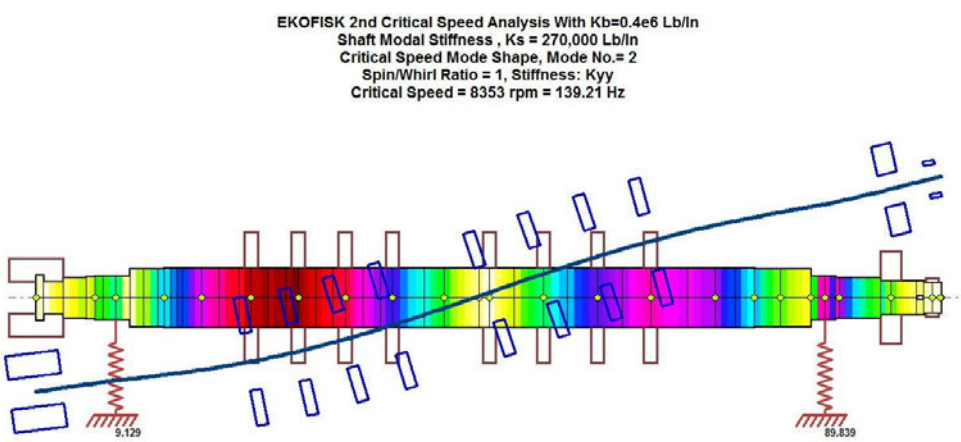

Figure 10 Phase I Rotor 2nd Critical Speed With Kb = 0.4e6 Lb/In
Nc2 = 8,325 RPM

Figure 10 shows the mode shape with an assumed bearing stiffness of Kb = 0.4e6 Lb/In. This mode shape is essentially a rigid body mode with over 90% of the system strain energy in the bearings. Thus, this mode is well damped and can not be excited.

Tilting Pad Bearing Designs

Table 3 represents the bearing design features for the various rotors.

	Phase I	Phase IV	Modern
Loading Direction	LOP	LBP	LBP
L/D	0.284	0.296	0.5
$c_b^{\varnothing}/D$ ($^\circ/_{\circ\circ}$)	0.8–1.2	1.0–1.4	1.7–2.2
m_p (dim)	0.29–0.53	0.07–0.38	0.19–0.40
Pivot Offset (%)	54	50	50

Table 3 Bearing Design Parameters *(7)*

ε (dim)	0.39
K_{xx} (lbf/in)	6.6e5
K_{yy}	9.5e5
C_{xx} (lbf-s/in)	589
C_{yy}	654
Min. Film Thickness (in)	0.0012
Probe Temperature (°F)	212
Power Loss (hp)	7.71

Table 4 Phase I Bearing Coefficients

Table 4 represents the original tilting pad bearing coefficients for the Phase I rotor *(7)*. In this bearing design, the load is on pad, referred to as LOP design. In Phase IV design, a narrow L/D bearing, is employed with the load between pads (LBP). This generates a softer vertical bearing stiffness. In the modern design, a wider L/D bearing design is employed with the load between pads.

An analysis of the original Phase I tilting pad bearing was computed using the DyRoBeS software bearing module. The tilt pad bearing input features are shown below in Table 5.

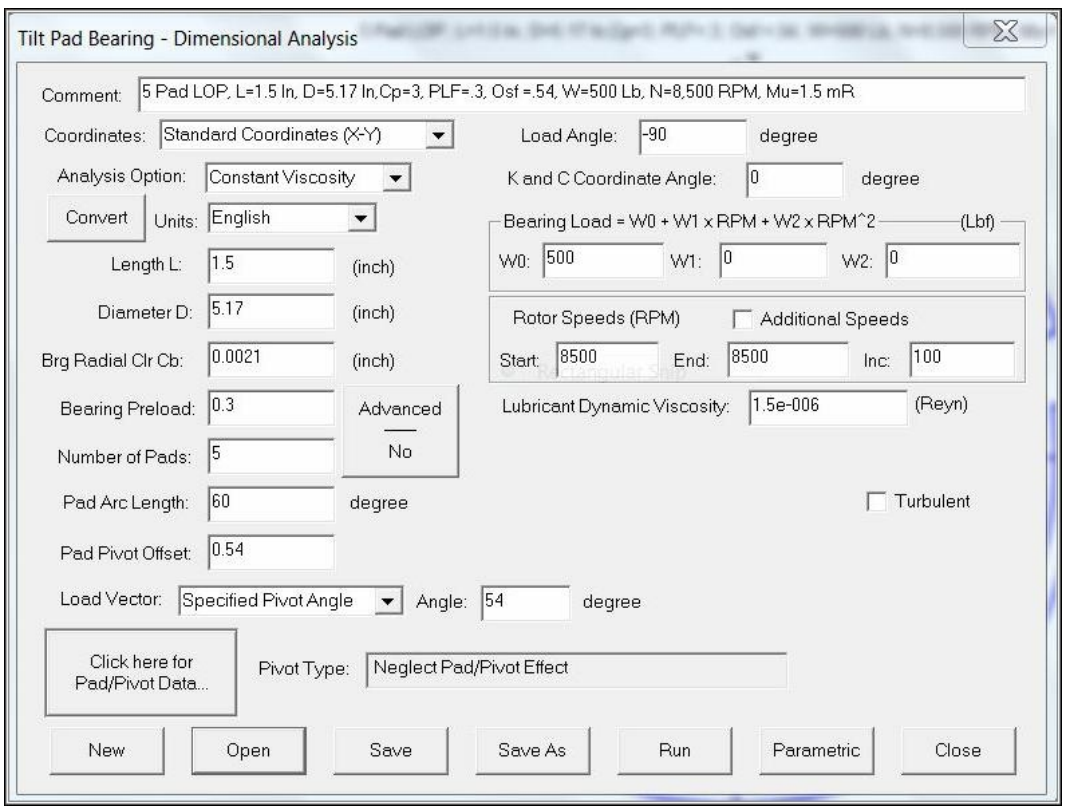

Table 5 Input Parameters For Phase I LOP Bearing at 8,500 RPM

A wide range of clearances and preloads was specified in the original design. In this calculation, the preload was set at 0.3.

Phase I Bearing to Shaft Stiffness Ratio *K*

An important parameter to evaluate, in order to ensure the stability of a compressor, is the ratio of the bearing stiffness to the shaft modal stiffness. This is referred to as the dimensionless parameter ***K*** = 2Kb/Ks, where Ks is the shaft modal stiffness.
The optimum value has been shown to be that ***K*** = 1. The first step is to evaluate the vertical stiffness of the narrow five pad bearings at running speed. From Table 4, it has been shown in *(7)* that Kyy = 9.5e5 Lb/In.

In Table 3, it is shown that the preload for the Phase I tilting pad bearing could vary from m_p = 0.29 - 0.53. The DyRoBeS bearing module was used to compute the tilting pad bearing characteristics at the running speed of 8,500 RPM with m_p = 0.3, which is on the low end of the specified range of preload values.
Figure 11 represents the five pad bearing model with the load on pad (LOP) design.

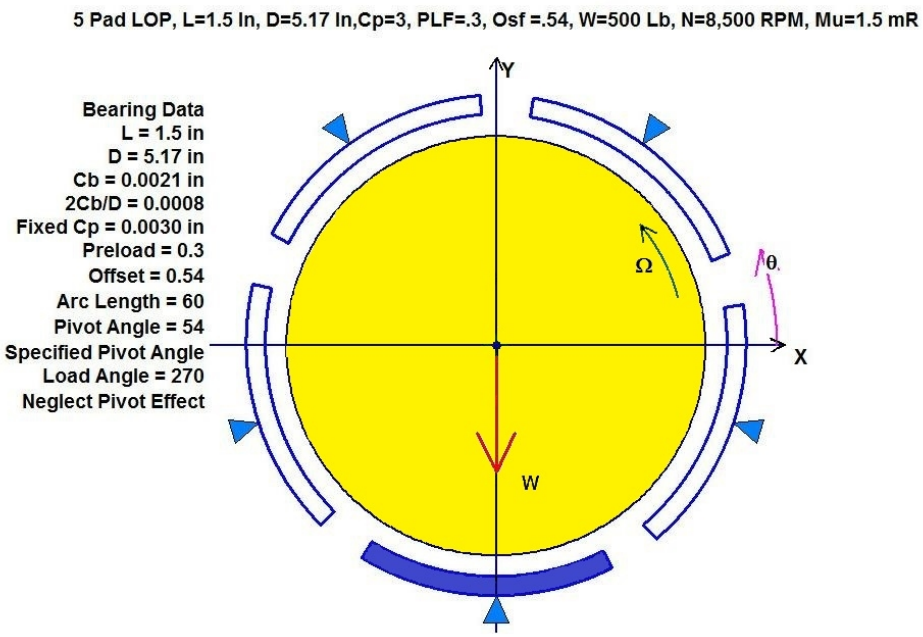

Figure 11 5 Pad Bearing Design with LOP and Preload m_p = 0.3

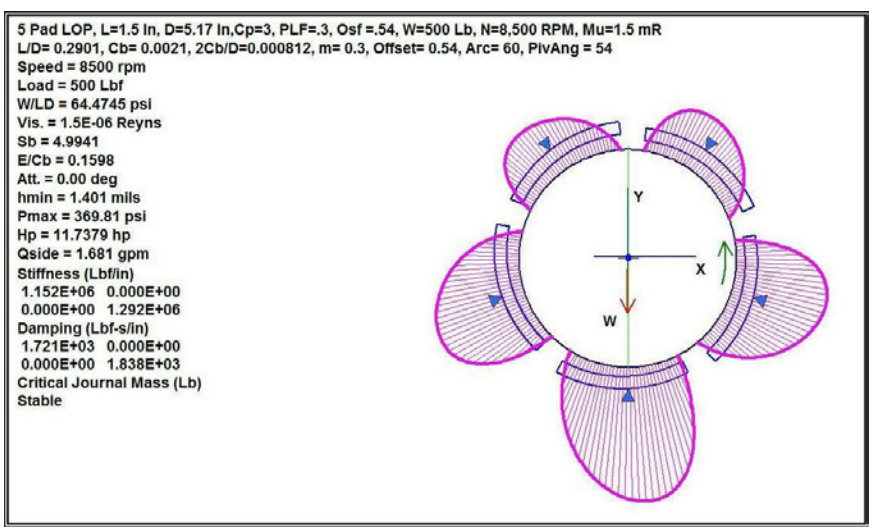

Figure 12 5 LOP Bearing Stiffness & Damping Coefficients at 8,500 RPM

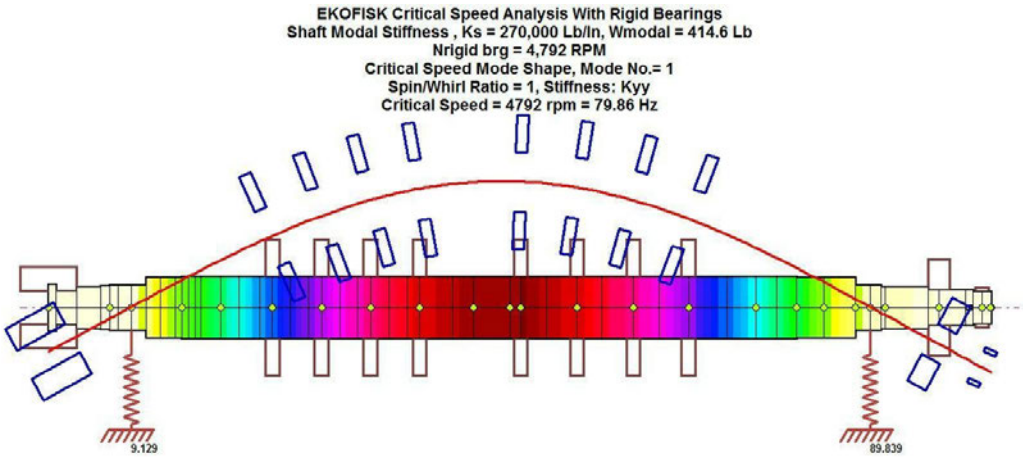

Figure 13 Phase I Rotor Rigid Rotor Critical Speed - 4,749 RPM
Shaft Stiffness Ks = 270,000 Lb/In, Wmodal = 414 Lb

Figure 13 represents the Ekofisk Phase I rotor rigid bearing critical speed computation. Note that the computed rigid bearing critical speed is 4,749 RPM. This is in sharp contrast to the original reported value of the rigid bearing critical speed as 4,200 RPM. The difference in the two values may be due to the rotor critical speed programs used. The original calculations were performed by the transfer matrix based formulation which requires an iteration solution and specified convergence criteria. These programs have often encountered great difficulty in convergence.

Another factor may be the inclusion of impeller gyroscopic effects. Also, when disk gyroscopic moments are included, the transfer matrix method may converge to the backward mode. Therefore, these apparent earlier inaccuracies in the reported critical speed values had led to the extensive investigation into the seals as the cause of the higher observed frequencies.

In addition to computing modal shaft stiffness of 270,000 Lb/In, the rotor modal weight of 414 lb or modal mass of 1.07 was computed. This allows one to generate a Jeffcott model for the approximate computations of optimum bearing characteristics and aerodynamic stability limits for a multi mass compressor.

With the knowledge of the shaft modal stiffness, modal mass and the vertical bearing stiffnesses, an approximation of the rotor amplification factor and the amount of aerodynamic cross coupling Q that the rotor can tolerate can be estimated. After the EKOFISK stability problems in 1974, research at the University of Virginia Rotor Dynamics Laboratory was conducted to determine the optimum bearing stiffness and damping for multistage compressors in order to maximize rotor stability.

The research work with motivated by a paper by Prof. Henry Black paper presented in 1976 on "***The Stabilizing Capacity of Bearings for Flexible Rotors With Hysteresis***." Prof. Black had also earlier been a visiting scholar at the University of Virginia Rotor Dynamics Laboratory.
In 1978, a paper was presented by Barrett, Gunter and Allaire on ***"Optimum Bearing and Support Damping for Unbalance Response and Stability of Rotating Machinery"***. This paper was an extension of the earlier paper by Prof. Henry Black of Harriet Watt University, Edinburgh.

At this particular time, in the evolution of rotor dynamics, damped eigenvalue analysis was computed by the transfer matrix method of Lund. Proper convergence of the eigenvalues was a problem with the transfer matrix theory of rotor stability, and also had the added problem of missed modes. This paper, presented on optimum bearing and support damping, could be applied to a multi-mass compressor by first computing the rotor modal mass and modal stiffness from the computation of the rotor first critical speed assuming rigid bearings. This then reduces the multi- mass rotor to a Jeffcott rotor.

Optimum Stiffness Ratio *K* and Amplification Factor *A*

An estimate of the rotor amplification factor ***A*** and aerodynamic capacity Q may be estimated by the use of the shaft modal mass and bearing to shaft stiffness ratio ***K*** .
The original vertical bearing stiffness, Kyy was computed as 9.5e5 Lb/In as shown in Table .
The ***Kyy*** ratio is given by 2 Kyy/Ks = 2x 9.5e5/270,000 = 7.03
The Kyy bearing stiffness generated by Dyrobes is 1.29e6 as shown in Figure 12. Thus the ***K*** is as follows:

$$\boldsymbol{K_{yy}} = 2\text{x } 1.29\text{e}6/270{,}000 = 9.56$$

As shown in Ref (4), the optimum value of ***Kyy*** = 1!
The corresponding amplification factor with optimal bearing damping is given by:

$$\boldsymbol{A}_{\text{opt}} = 2(1 + \boldsymbol{K}) \qquad (1)$$

Therefore we see that even with optimum bearing damping, the amplitude is predicted to be between 14 to 19 ! This is clearly excessive and indicates a highly sensitive rotor to self excited forces. As a check of these approximate values of amplification factor, the DyRoBeS damped eigenvalue analysis was performed on the Phase I rotor with the two bearing coefficients calculated.

Figure 14 below shows the computed damped 1st forward mode for the Ekofisk Phase I rotor at 8,500 RPM with the original narrow five pad LOP bearing design with the characteristics as shown in Table 4. The amplification factor is computed as pi/log dec = 17 !

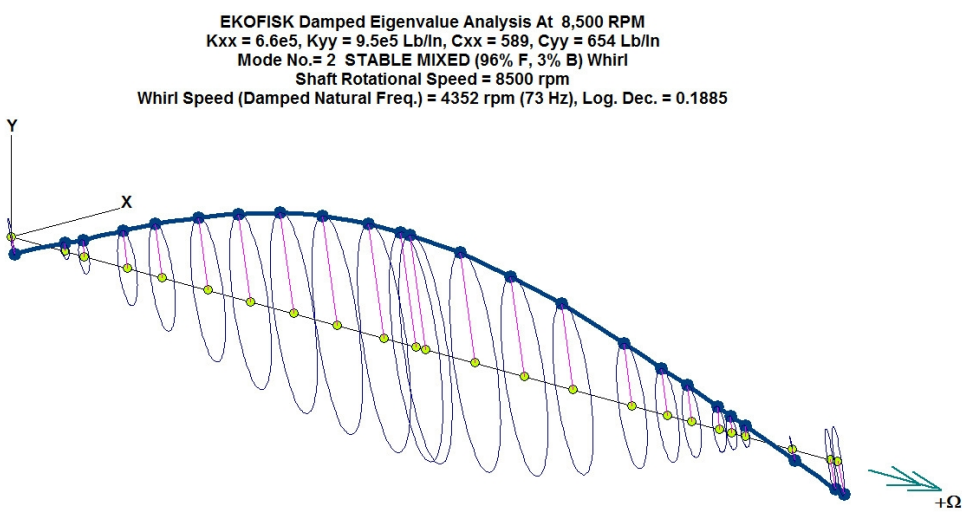

Figure 14 EKOFISK Phase I Rotor Damped 1st Forward Mode at 8,500 RPM
Nf1 = 4,353 CPM, Log Dec = 0.1833, A = 17

Influence of Support Stiffness on Effective Tilting Pad Bearing Damping

In Figure 12 , the tilting pad bearing coefficients, with a preload m of 0.3, as computed by Dyrobes, are given as:

Kxx = 1.15e6 Lb/In , Kyy = 1.29e6 Lb/In

Cxx = 1,731 Lb-Sec/In, Cyy = 1836 Lb-Sec/In

Figure 15 represents the computed 1st forward damped eigenvalue (mode) using these coefficients at the operating speed of 8,500 RPM.

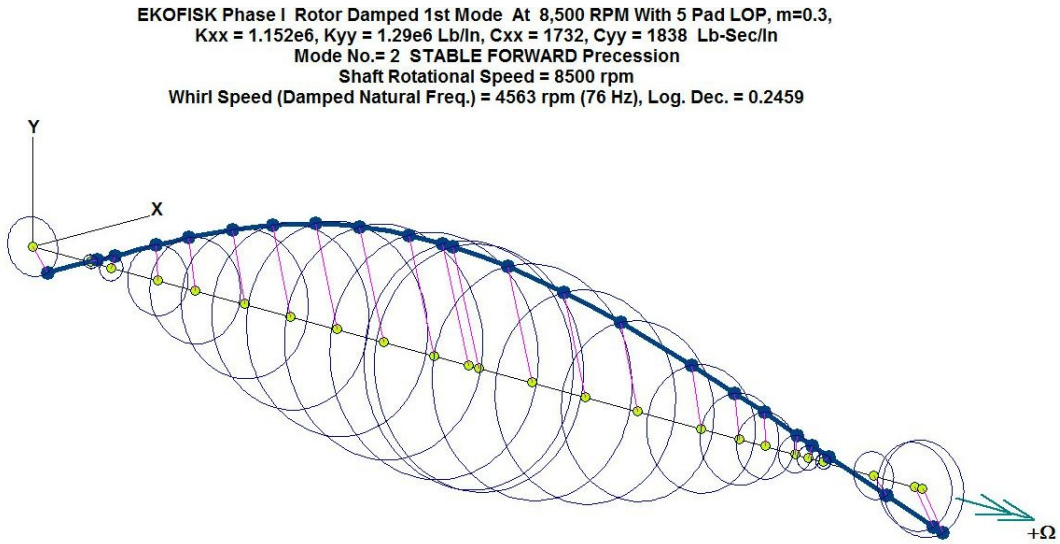

Figure 16 EKOFISK Phase I Damped 1st Forward Mode at 8,500 RPM, m=0.3
Kxx = 1.15e6, Kyy = 1.29e6 Lb/In, Cxx= 1732, Cyy = 1838 Lb-Sec/In
Nf1 = 4,563 CPM, Log Dec = 0.246

The computed first forward whirl mode using the bearing coefficients considering a 0.3 bearing preload, resulted in very high bearing stiffness and damping coefficients. The resulting whirl frequency of 4,563 cpm is higher than the actual recorded whirl frequency of 4,400 cpm. The problem is that in an actual rotor system, it is not possible to have the high damping coefficients predicted in the tilt pad bearing programs. The reason for this is that foundation or support flexibility significantly reduces the effective bearing damping.

In the early 1970s, for example, Charlie Jackson, chief engineer of Monsanto, reported on some earlier experimental testing of a multistage compressor with five pad LOP design bearings. The compressor was deliberately surged to experimentally measure the damped first natural frequency. From the measured experimental data obtained from the testing, when compared with the predicted damped first whirl mode, indicated that the apparent tilt pad bearing damping coefficients were less than half of the predicted values from the computer code.

Further studies were then directed to study the influence of foundation flexibility on the effective bearing stiffness and damping characteristics. It was found that significant reduction in bearing damping occurs with even very high values of support stiffness.

The reduction in bearing stiffness was indicated in the paper presented by Atkins and Perez at the 21st Turbomachinery Symposium in 1992. Extensive testing was performed on a multistage compressor with narrow five pad LOP designed tilt pad bearings. A follow-up study on the experimental test results of Atkins and Perez was initiated in 2017*(12)* using the Dyrobes software to compute the theoretical bearing coefficients. The test results were compared with the theoretical bearing stiffness and damping coefficients. It was determined that less than 50% of predicted tilt pad bearing damping was present in the tests.

From a practical standpoint, the tilting pad bearing damping should be derated at least by 50% to produce realistic computations. The second more sophisticated procedure is to add a flexible support under the tilt pad bearings. It should be noted that it is difficult to produce accurate calculations with support flexibility included when using transfer matrix theory formulation.

Figure 17 represents the Ekofisk 1st whirl mode computed assuming a 50% reduction of the bearing damping values as originally calculated with the preload m = 0.3.

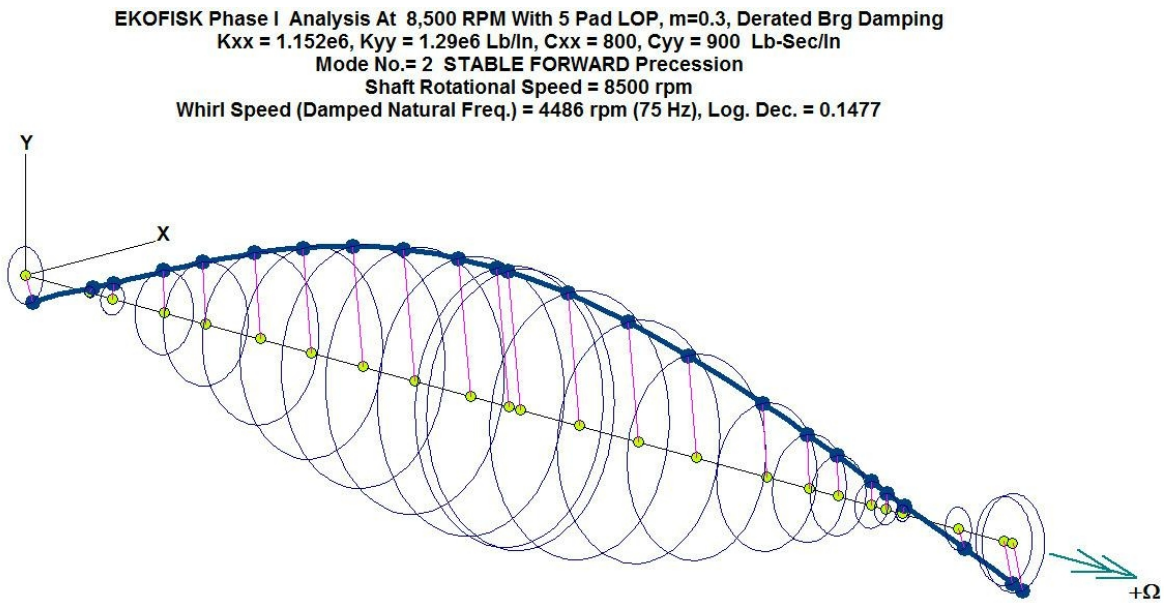

Figure 17 1st Forward Mode at 8,500 RPM With Derated Damping
Kxx = 1.15e6, Kyy = 1.29e6 Lb/In, Cxx= 800, Cyy = 900 Lb-Sec/In
Nf1 = 4,486 CPM, Log Dec = 0.148

By reducing the bearing coefficients by 50%, it is seen that a frequency of N_{f1} = 4,486 cpm is obtained. This value is now more in line with the observed experimental frequency of 4,400 cpm. Note that no seal stiffening effects are required to obtain this frequency.
The amplification factor is A = pi/log dec = 3.14/0.148 = 21.2.

This value is unacceptable and is an indicator of a highly unstable rotor. In the process of trying to avoid the second critical speed by designing a narrow LOP five pad bearing, a highly unstable rotor system has been created which will be extremely sensitive self excited forces.
A more precise method will be presented in the next session in which flexible supports are added to the system under the tilting pad bearings. This then, does not require a degradation of the bearing damping to produce realistic eigenvalues.

EKOFISK Phase I Rotor With Flexible Supports

Figure 18 represents the original Ekofisk Phase I rotor design with flexible supports incorporated with the tilting pad bearings.

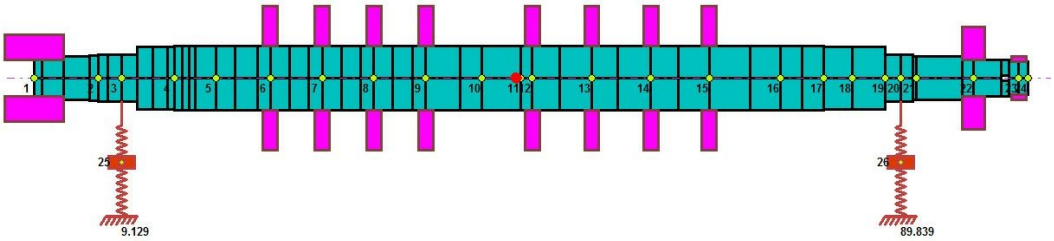

Figure 18 EKOFISK Phase I Rotor With Flexible Bearing Supports

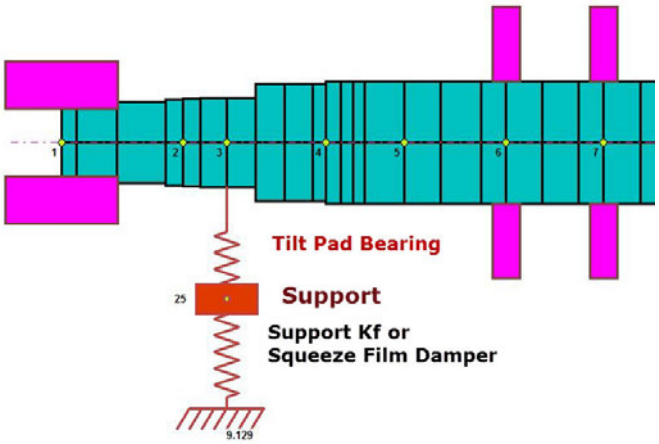

Figure 19 Expanded View of Bearing and Support

Figure 19 represents an expanded view of the tilting pad bearing and the supporting structure. The supporting structure can represent a simple stiffness, referred to as K_f, or it could be used to incorporate a linear or nonlinear squeeze film damper with the tilting pad bearing. In order to have two bearings connected in series, a supporting mass must be included. The Eigen solver will not compute if station 25 is a massless station.

Figure 20 below, represents the input file to specify the support mass. A nominal mass of 25 pounds was selected to represent the structure supporting the tilt pad bearing.

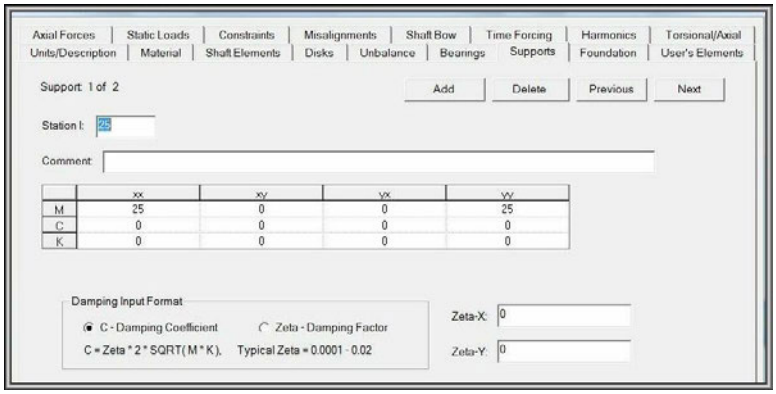

Figure 20 Support Structure Input Table

Stability Capacity of EKOFISK Phase I Rotor

An estimate of the Ekofisk Phase I rotor stability limits may be estimated from the rotor modal properties, the shaft modal stiffness and the vertical bearing and support stiffness values*(5).*

The estimated stability capacity, Q is given by:

$$Q = \frac{M_{modal}}{2(1+K)} \times \omega_{cr}^2 \left[\frac{1+2K}{2(1+K)}\right]^{0.5}$$

Where:

M_{modal} = 1.07, w_{cr} = 497 rad/sec (*Critical Speed on rigid bearings*)

$1/K_{eff} = 1/K_{yy} + 1/K_f$

K_{yy} = 1.29e6 Lb/In , K_f = 2.8e6 Lb/In *(Support Stiffness)*

K_{eff} = 883,000 Lb/In

$K = 2\,K_{eff}/K_s$ = 2x 883,000/ 270,000 = 6.54 , K_s = *Modal shaft stiffness*

***Q* = 17,500 Lb/In**

The approximate capacity of the Phase I rotor, using modal properties, indicates that the aerodynamic cross coupling capacity is less than 20,000 Lb/In. This value is considerably below the desirable aerodynamic capacity of 100,000 to 200,000Lb/In. It is apparent that the problem with the Phase I rotor is that the rotor is extremely sensitive to self excited whirling due to the high bearing to shaft stiffness ratio. The problem with the phase I rotor design is not due to the seals, but it is due to the narrow five pad load on pad LOP bearing design which was employed in order to avoid the rotor second critical speed.

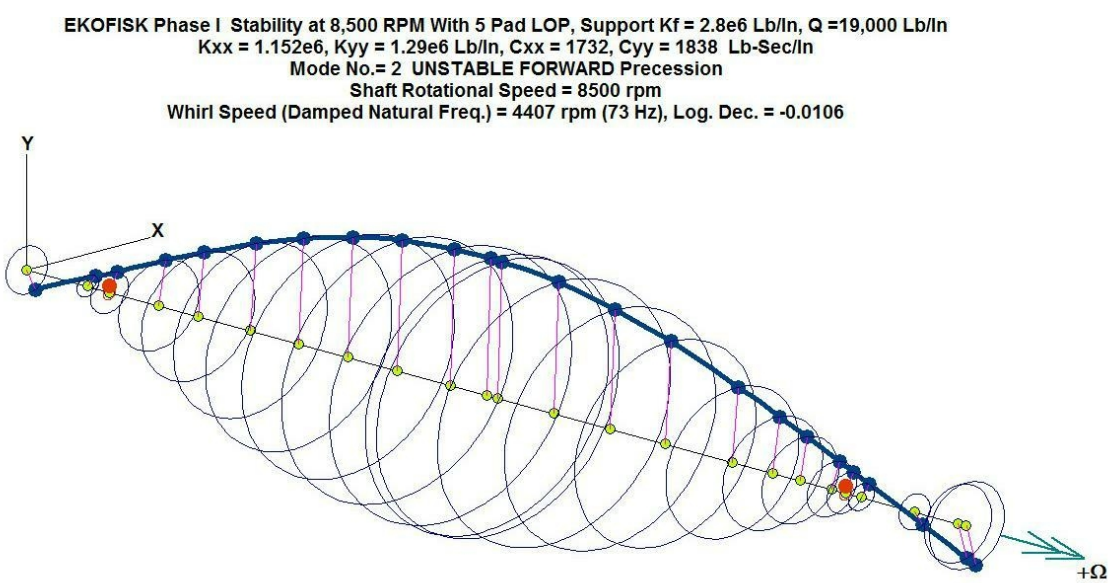

Figure 21 EKOFISK Phase Rotor I Stability Limit With An Aerodynamic Cross-Coupling of Q =19,000 Lb/In

EKOFISK Phase I Rotor Time -Transient Analysis

A time transient analysis will be generated for the Ekofisk Phase I rotor.

```
** System Parameters ***************

   1  Shafts
  23  Elements
  51  SubElements
   3  Materials
   1  Unbalances
      1  Mass Unbalances (mew^2)
  11  Rigid Disks (4 dof)
   0  Flexible Disks (6 dof)

   5  Linear Bearings
   0  NonLinear Bearings
   2  Flexible Supports
   0  Axial Loads
   0  Static Loads
   0  Time Forcing Functions
   0  S.S. Harmonic Excitation

   0  Natural Boundary Conditions
   4  Geometric Boundary Conditions
   4  Constraints

  26  Stations
 100  Degrees of Freedom
```

Table 6 Rotor Specifications

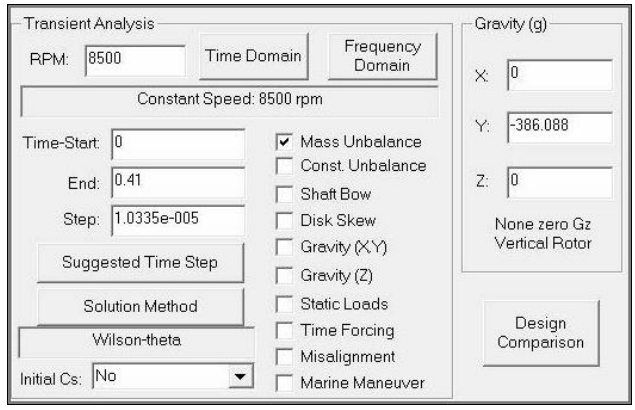

Table 7 Transient Analysis Specifications

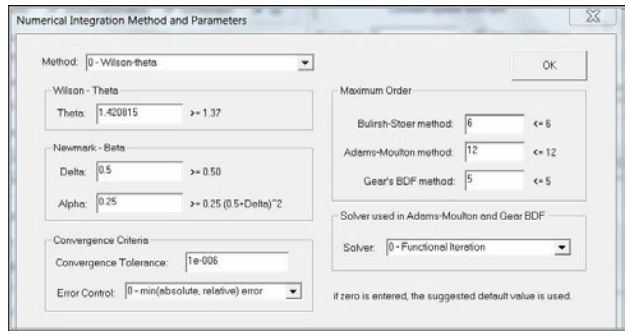

Table 8 Wilson-Theta Integration

Table 6 represents the specifications for the Phase I rotor with the flexible supports. The rotor has 24 major mass stations which represent 96 degrees of freedom. The two additional supports under the bearings represent four more degrees of freedom for a total of 100 degrees of freedom.

Table 7 represents the input options for the time transient analysis. In this model, the rotor will be operating at a speed of 8,500 RPM or 141.7 rev/sec. The time for one cycle of motion is 7.059e-3 sec/cycle. The total time is show as 0.41 sec. Hence the total number of cycles of motion is 58. The time step increment is 1.033e-5 sec. Thus to complete one cycle of motion requires 683 time steps. Therefore, to complete the transient motion, requires approximately 4 million solutions of the equations of motion. Time transit analysis is not possible with a matrix transfer formulation.

It should be noted that at the time period that both KAYBOB and EKOFISK were designed, the finite element theory of rotor dynamics was not fully formulated, and also the computers did not exist to perform the numerical calculations required. Hence, at that time, it was not able to perform linear or nonlinear dynamical analysis of the actual rotor motion with fluid film bearings including rotor unbalance, aerodynamic excitation and squeeze film dampers. This was simply not possible to do at that time period. It was not even possible to accurately calculate the damped second critical speed amplification factor with bearing damping.

EKOFISK Phase I Rotor Transient Motion at 8500 RPM With Q=50,000 Lb/In

Figure 22 represents the time transient motion of the Phase I rotor supported in the original narrow five pad load on pad bearings. An aerodynamic cross coupling of Q was assumed to be 50,000 Lb/In. As shown by the stability analysis, this value is greatly in excess of the predicted stability limit of Q =19,000 Lb/In.

A large value of rotor unbalance of 10 oz-in was added to the rotor center. There is no evidence of synchronous motion in the rotor response as it is completely dominated by the self excited whirl behavior of the rotor.

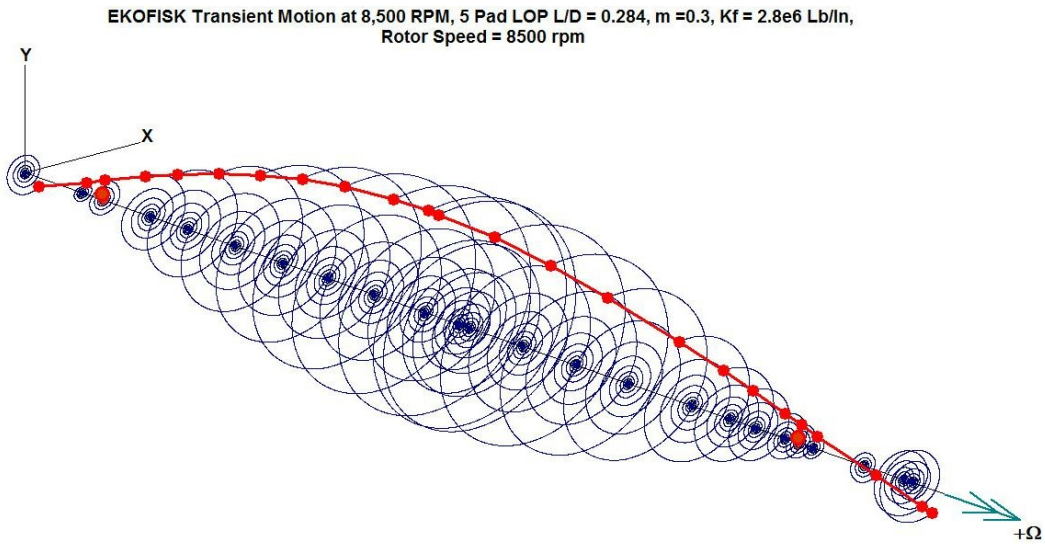

Figure 22 Time Transient Motion of Phase I Rotor With Q =50,000 Lb/In

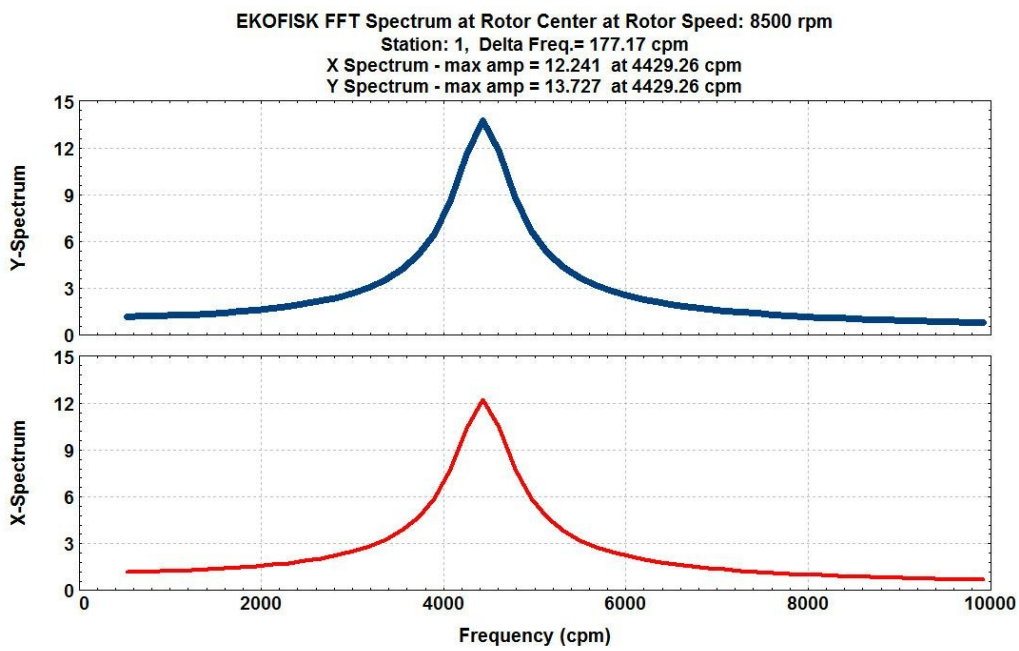

Figure 23 FFT of Unstable Rotor Spectrum Showing Whirl at 4,429 CPM

Figure 23 shows the vibration spectrum to be at 4,429 CPM. This whirl frequency matches the unstable whirl motion observed in the original rotor testing. There is no seal influence.

Squeeze Film Design For Phase I Rotor

In order to design a nonlinear squeeze film damper, it is first necessary to evaluate the optimum values of damping corresponding to various support stiffness values.
Table 9 shows the maximum aerodynamic cross-coupling values Q that are possible for various values of support damping for three different support stiffness values.

The optimum value of support stiffness should be approximately one-half the modal shaft stiffness, which is 270,000 Lb/In for the Phase I rotor.
With the support stiffness at 150,000 Lb/In, the maximum Q value attainable is slightly greater than 100,000 Lb/In stability. The required damping may vary from 600 to 900 Lb-s/In. Below 600 Lb-s/In the damping is insufficient and above 900 Lb-s/In leads to damper lockup.

Table 9 Maximum Q values For Various Support Stiffness and Damping

Kf support, Lb/In	***Cf support damping, Lb-S/In***	Maximum **Q** ***Lb/In***
150,000	600 - 900	***100,000***
300,000	1,000 - 1,300	75,000
500,000	1,200 - 1,800	60,000

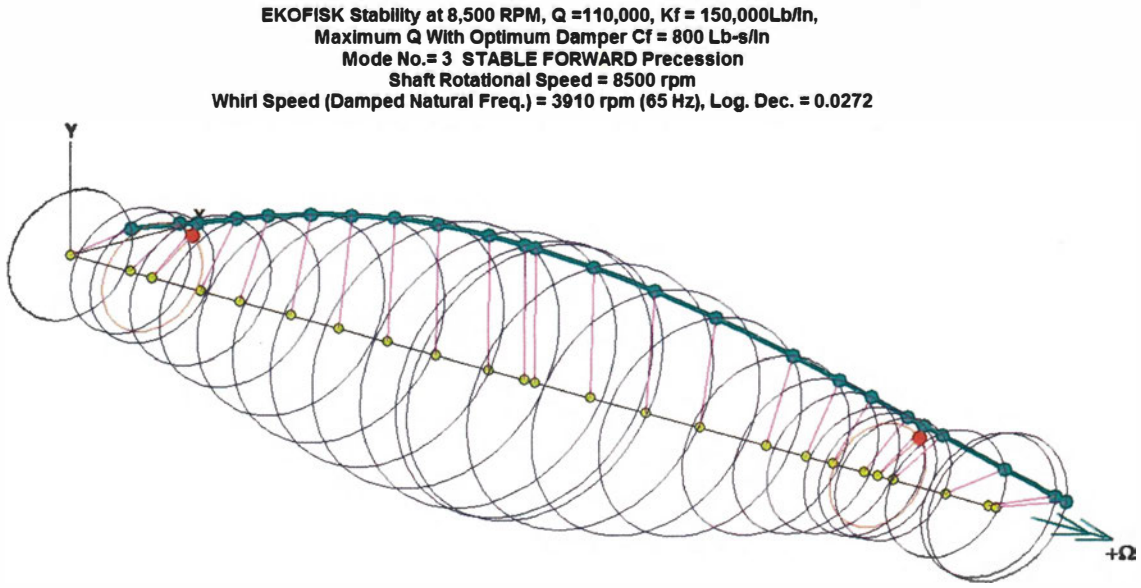

Figure 24 Phase I Stability With *Kf* = 150,00 Lb/In & *Cf* = 800 Lb-s/In
Q max = 110,000 Lb/In

Figure 24 shows the mode shape of the first forward whirl mode at 8,500 RPM with an aerodynamic loading of 110,000 Lb/In with optimum support damping of 800 Lb-s/In.
This level of aerodynamic stability capacity of over 100,000 Lb/In is in sharp contrast to the original stability limits of less than 20,000 Lb/In.

Squeeze Film Damping Properties

To create a squeeze film damper, the tilting pad bearing cartridge is supported in O-rings with a clearance for the oil film. In this damper design, a center groove is placed in the damper to generate what is referred to as a short film damper bearing. If no center groove is added, then the damper is a long film bearings damper. Table 10 below represents the bearing stiffness and damping coefficients for various bearing configurations with or without considering oil cavitation.

Bearing	Film	Motion	Stiffness	Damping
Short Bearing	π film	Circular Synchronous Precession	$\frac{2\mu R L^3 \varepsilon \omega}{C^3(1-\varepsilon^2)^2}$	$\frac{\mu R L^3 \pi}{2C^3(1-\varepsilon^2)^{3/2}}$
	2π film		0	$\frac{\mu R L^3 \pi}{C^3(1-\varepsilon^2)^{3/2}}$
	π film	Pure Radial Squeeze Motion	0	$\frac{\mu R L^3 \pi (2\varepsilon^2+1)}{2C^3(1-\varepsilon^2)^{5/2}}$
	2π film		0	$\frac{\mu R L^3 \pi (2\varepsilon^2+1)}{C^3(1-\varepsilon^2)^{5/2}}$
Long Bearing	π film	Circular Synchronous Precession	$\frac{24\mu R^3 L \varepsilon \omega}{C^3(2+\varepsilon^2)(1-\varepsilon^2)}$	$\frac{12\mu R^3 L \pi}{C^3(2+\varepsilon^2)(1-\varepsilon^2)^{1/2}}$
	2π film		0	$\frac{24\mu R^3 L \pi}{C^3(2+\varepsilon^2)(1-\varepsilon^2)^{1/2}}$

Where

R = damper radius
L = damper axial length
C = radial clearance
ω = whirl speed
μ = oil viscosity
ε = eccentricity ratio

Table 10 Squeeze Film Damping Properties
(Barrett, & Gunter 1975)

The bearing stiffness and damping coefficients for both the short and long bearing damper assumptions were generated assuming synchronous circular precession about the journal center. The bearing stiffness and damping coefficients were computed assuming a cavitated pi oil film and also without oil film cavitation.

If the film cavitates, then radial forces are generated. These radial forces increase greatly with eccentricities exceeding 30% of the bearing clearance. There is however, one compressor company that supplies squeeze film dampers with its five pad tilting pad bearings. They use the assumption in their analysis that the squeeze film dampers do not cavitate. This assumption provides double the damping of the cavitated damper, and the problem of damper lockup is avoided. Such an assumption of the uncavitated squeeze film damper is unrealistic as the supply pressure to the damper can never be high enough to prevent cavitation. A proper damper design, therefore, should assume a cavitated film.

One very interesting exception to this is the last case of a two pi film long damper. This equation is actually the assumption used to design an air squeeze film damper. An air squeeze film damper was designed for aircraft ball bearing supported ACM unit which had experienced repeated failures.

These units are basically double overhung rotors in which the ball bearing cartridge is supported in 0-rings. Two air squeeze film dampers were successfully installed with the bearing cartridge. No additional air supply is required with an enclosed air squeeze film damper. This design concept would also be suitable for small high-speed turbochargers with ceramic bearings.

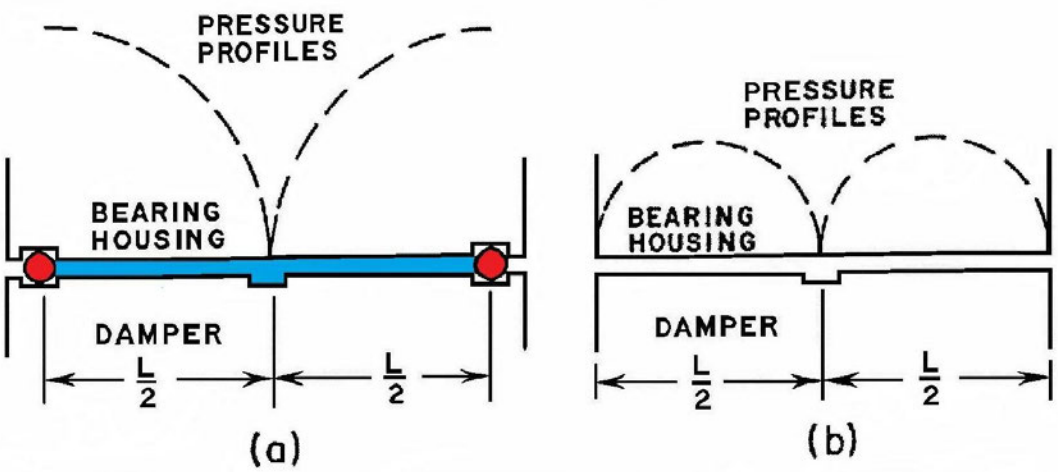

Figure 25 Squeeze Film Dampers With Center Oil Supply Groove
(a) Closed End Damper (b) Open Ended Damper

Figure 25 shows a typical squeeze film damper with the central groove installed. The O-rings seal the damper and provides support. Care must be applied in the design of the O ring grooves. If a standard O-ring groove is used, then the weight of the rotor will collapse the damper. A narrow groove must be used that supports the O-ring so that it will support the rotor weight with out bottoming out in the cartridge. If this occurs, then the damper will not operate successfully.

A second consideration is that the upper damper groove must be slightly deeper than the lower groove. This is so that excessive loading is not applied to the damper when assembled.

Sizing Squeeze Damper Dimensions

From Table 9, an estimate can be made of the required squeeze film damping properties required in order to carry an aerodynamic excitation load of Q = 100,00 Lb/In.

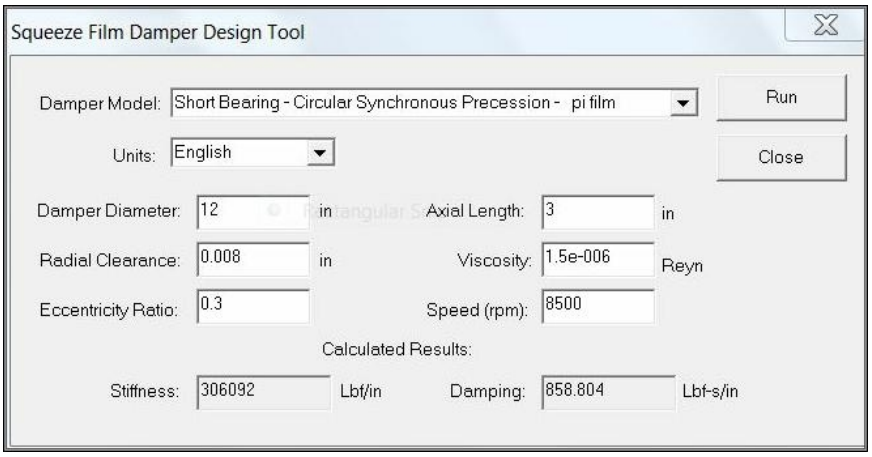

Figure 26 Synchronous Squeeze Film Damping Coefficients
N= 8,500 RPM, D =12 In, L = 3 In, Cr =0.008, Ecc =0.3

Figure 26 shows the synchronous stiffness and damping coefficients for the above conditions. The resulting damper stiffness and damping are given as:

K_{damper} ~ 300,000 Lb/In , C_{damper} ~ 860 Lb-S/In

The total effective stiffness of the damper system is the combination of the stiffness of the supporting O-rings plus the added stiffness of the damper oil film. The total system damper stiffness increases rapidly when the orbital damper motion exceeds 30% of the damper clearance. Thus, large values of rotor unbalance can cause damper "*lockup*" at some speeds resulting in subsynchronous whirl instability.

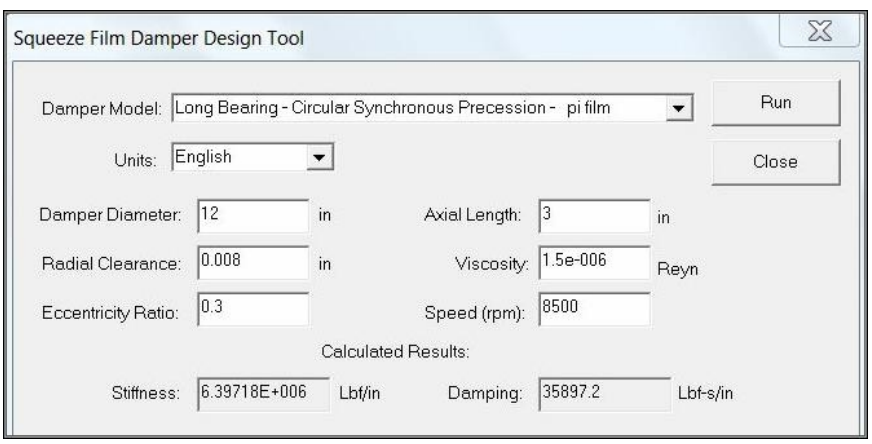

Figure 27 Long Bearing Squeeze Film Damping Coefficients
N= 8,500 RPM, D =12 In, L = 3 In, Cr =0.008, Ecc =0.3

Figure 27 represents the squeeze film damper characteristics generated assuming a long bearing damper. This occurs with the damper when the center groove is not added. As a result, the damper stiffness is over 6e6 Lb/In and the damping is approximately 36,000 Lb-S/in. Thus, the damper is locked up and acts more as a rigid support. This is precisely what happened with the KAYBOB rotor in that the center groove was not added to the damper. The squeeze film damper could have been made operational by either adding a central groove or increasing the bearing damper clearance. Instead, the damper clearance was reduced!

EKOFISK I Response With Nonlinear Squeeze Film Damper

The Ekofisk I rotor model as shown in Figure 18 has the linear bearing supports replaced with nonlinear squeeze film dampers. In addition to the nonlinear dampers, at the rotor center station 10 is added aerodynamic cross coupling Q of 100,000 Lb/In and rotor unbalance of Ub of 5 Oz-In.

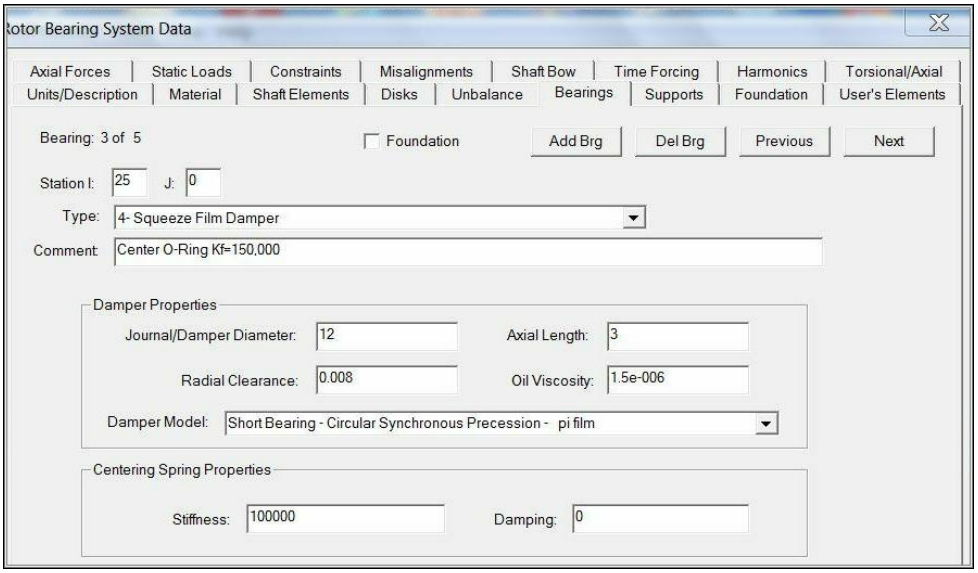

Figure 28 Specification of Nonlinear Squeeze Film Dampers

Figure 28 shows the specifications for the nonlinear squeeze film dampers. A short bearing damper has been chosen similar to that shown in Figure 25 (a). The outer diameter of the bearing cartridge D is specified as 12 in. and the axial length is 3 in. A damper radial clearance of 8 mils is assumed. The effectiveness stiffness of the rubber O-rings are assumed to be 100,000 Lb/In.

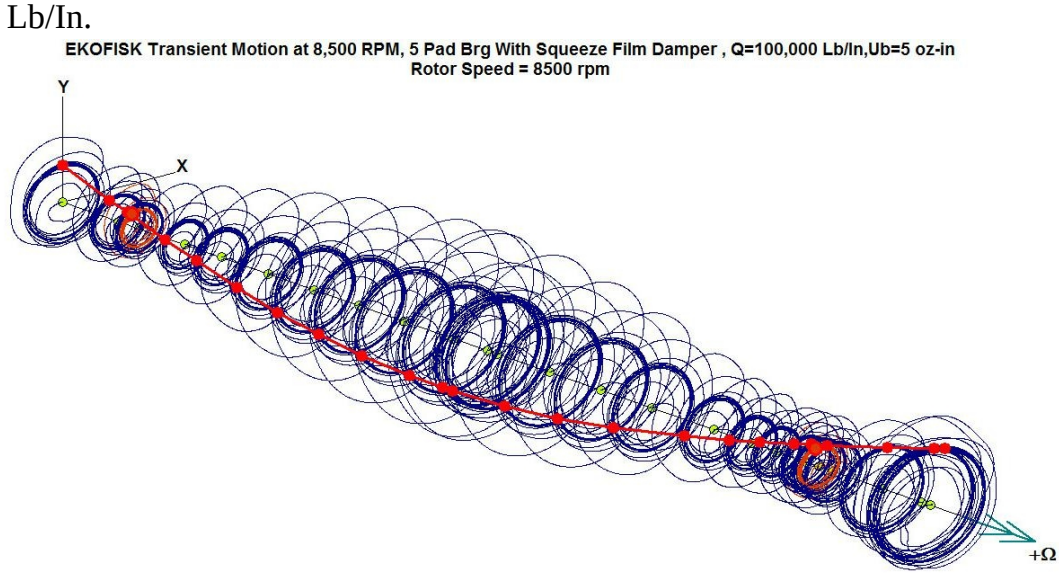

Figure 29 Transient Motion With Nonlinear Squeeze Film Dampers
Kf =100,000 Lb/In, Cr = 8 mils, Q = 100,000 Lb/In - Stable System

The rotor system is stable with an excitation of Q =100,000 Lb/In with the squeeze film dampers with a radial clearance of 8 mils.

EKOFISK I Response With Nonlinear Squeeze Film Damper Kf =150,000 Lb/In

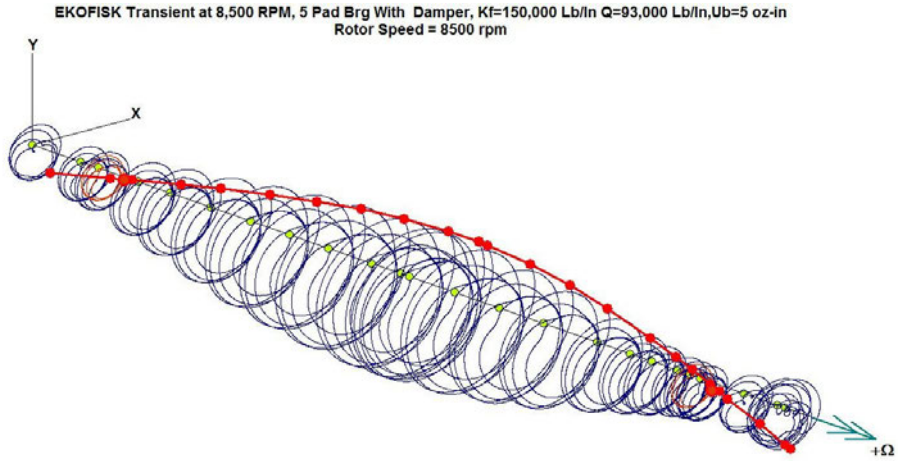

Figure 30 Rotor Response With Kf = 150,000Lb/In & Q =93,000 Lb/In

The effective stiffness of the damper is a combination of the stiffness of the supporting O-rings plus the addition of the stiffness generated by the oil film. The ideal effective total support stiffness should be approximately one half the shaft modal stiffness, which would be approximately 135,000 Lb/In. As the total effective stiffness exceeds the optimal value, then the amount of aerodynamic cross coupling that the rotor is capable of withstanding diminishes.

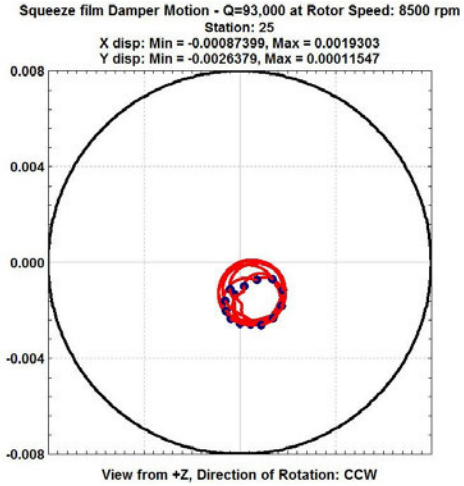

Figure 31 Damper Motion

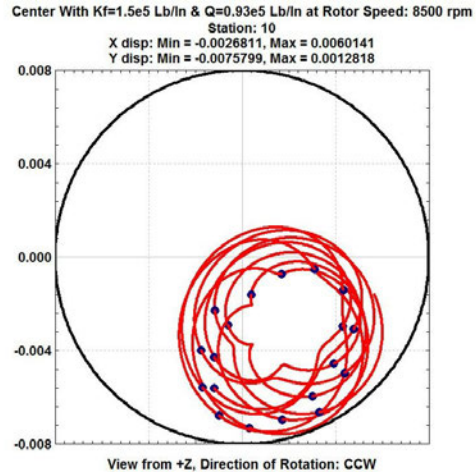

Figure 32 Center Motion

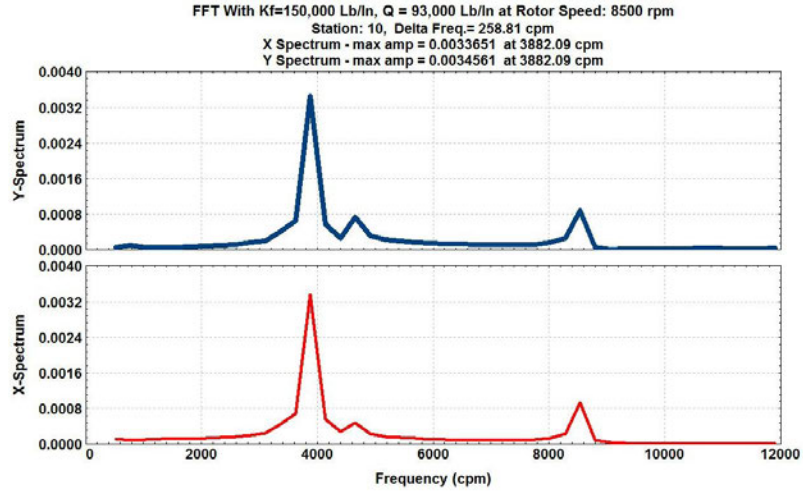

Figure 33 FFT of Center Span Motion With Kf = 1.5e5 Lb/In

EKOFISK I Rotor Stability With Four Pad LBP Bearings

The original Ekofisk I rotor design had a very narrow five pad LOP bearing design. This produced a very high vertical bearing stiffness in relationship to the fundamental shaft stiffness. The original concept was to avoid the second critical speed by placing it above the operating speed range. As such the stability capacity of the original design was Q < 20,000 Lb/In. Table 9 shows that if the bearing support stiffness is reduced to as low as 150,000 Lb/In, then an aerodynamic value of Q ~ 100,000 Lb/In is possible.

In order to reduce the effective vertical bearing stiffness, a four pad LBP bearing design has been selected. In addition, as shown in Figure 34 below, the bearing length has been increased to L = 3 in. A light preload of m =0.1 has been chosen to reduce top pad loading.

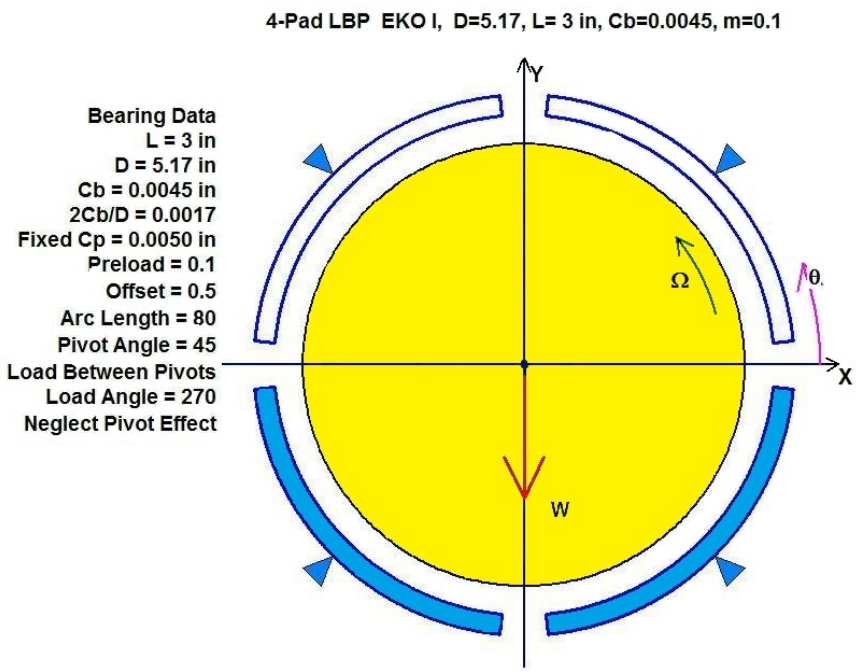

Figure 34 Specification of 4 Pad LBP Bearing With L = 3.0 In

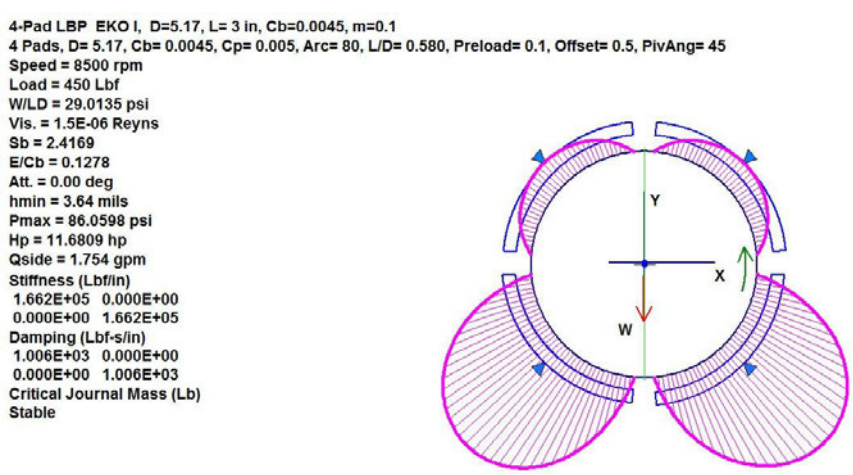

Figure 35 Four Pad Bearing Stiffness and Damping Coefficients

Figure 35 shows that the four pad LBP bearing design has highly desirable stiffness and damping coefficients. The stiffness values are Kxx = Kyy = 166,000 Lb/In and the damping coefficients are Cxx = Cyy = 1006 Lb-s/In. These coefficients will greatly improve stability.

Figure 36 represents the stability characteristics for the original rotor with enlarged four pad LBP bearings. The stability characteristics for the rotor have been increased from Q =19,000 to 88,000 Lb/In. Thus, the use of four pad LBP bearings will not require the installation of complicated and sensitive squeeze film dampers to enhance the system stability.

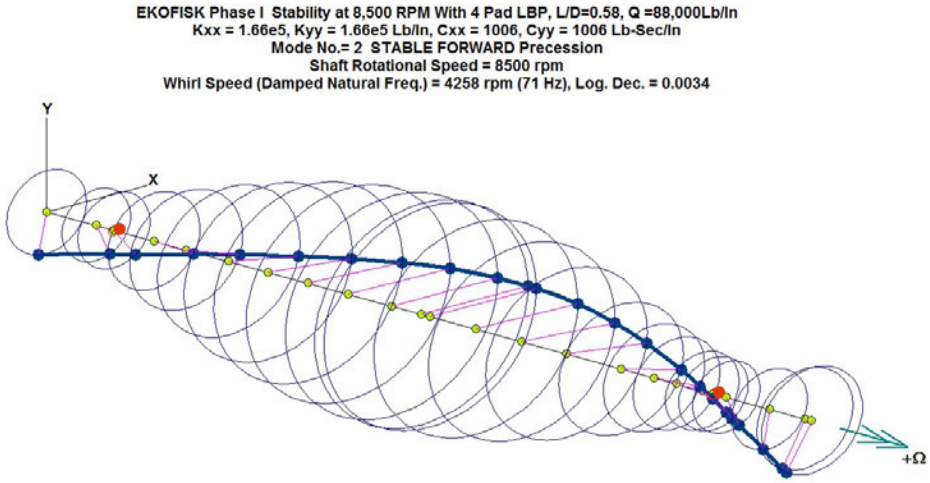

Figure 36 EKO I Rotor Stability With Four Pad LBP Bearings
Kxx = Kyy = 166,000 Lb/In, Qmax = 88,000 Lb/In

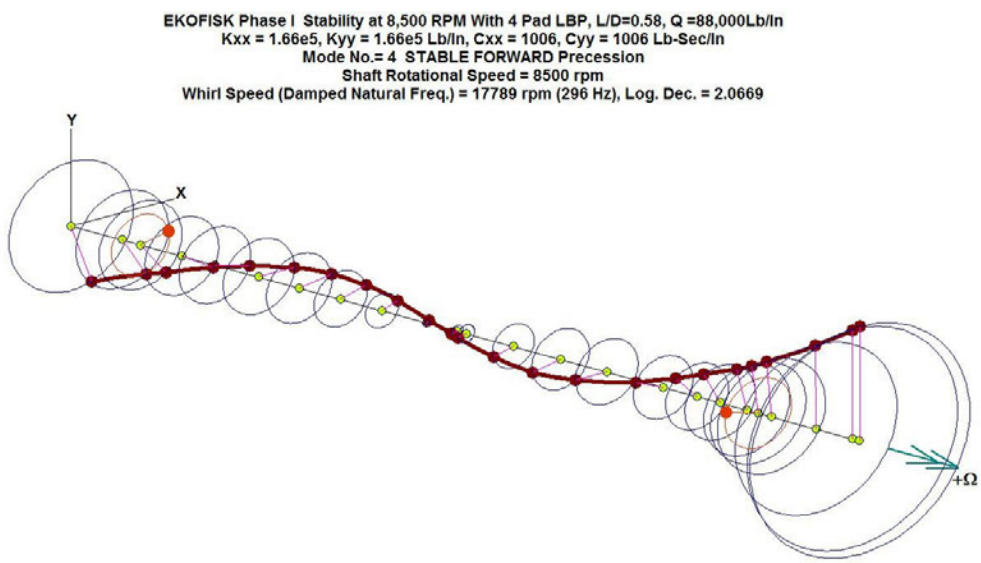

Figure 37 Forward Bending Critical Speed at 17,789 RPM

The original reason for the use of the narrow five pad LOP bearing design was the concept that the elevated bearing stiffnesses would avoid the operation near the second critical speed region. Figure 37, as shown above, is actually the rotor bending third critical speed. The second critical speed is basically a rigid body conical mode, which is over damped and is nonexistent. Hence with the four pad bearing as shown there is no second critical speed to be excited. However, at the time of the original design of EKOFISK I, finite element damped eigenvalue computer codes were not available to compute the log decrement of higher order modes.

EKOFISK Phase IV Rotor Stability With 5 Pad LBP Bearings

Figure 3 shows a cross-section of the Phase IV rotor. The Phase IV rotor has the bearing span reduced from the Phase I value of 80.7 to 75.6 in. In addition to shortening the bearing span, the shaft main diameter has been increased from 6.7 in to 9.0 in.
A computer model of the Phase IV rotor is shown below in Figure 38.

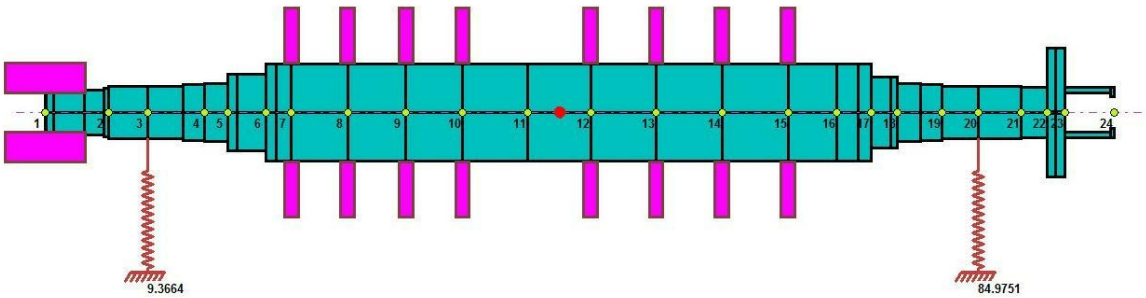

Figure 38 Phase IV Rotor With Reduced Bearing Span and 9 in Main Diameter

Figure 39 represents the critical speed of the Phase IV rotor with rigid bearings. The modal stiffness Ks of the rotor may now be computed. The value of Ks is ~ 860,000 Lb/In. This is in comparison to a value of 270,000 Lb/In computed for the original Phase I rotor system. Thus the Phase IV rotor has over 3 times the stiffness of the original rotor design.

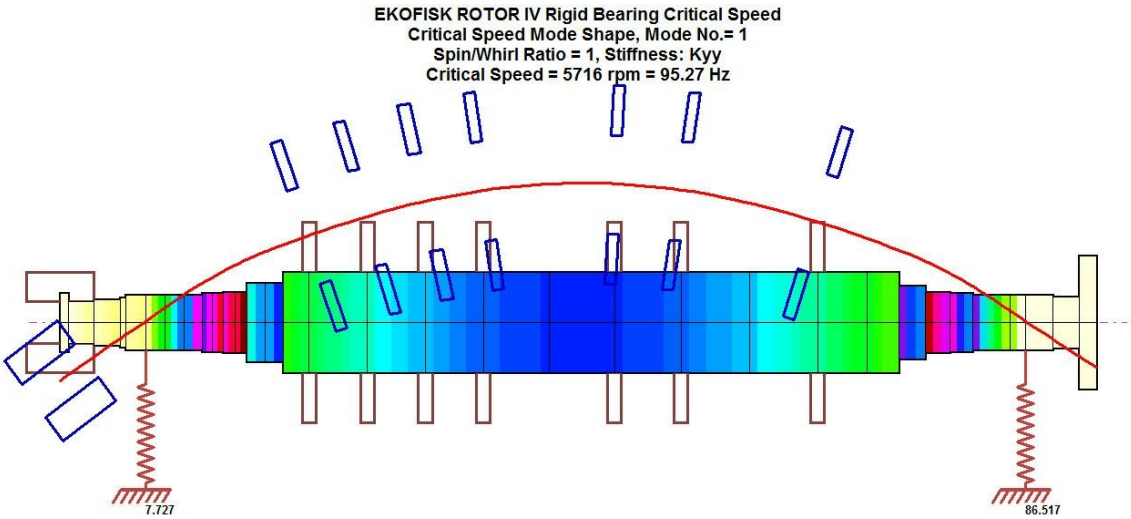

Figure 39 Phase IV Rotor Critical Speed on Rigid Bearings
Ncr = 5,716 RPM, Mmodal = 2.39 , Kshaft = 860,000 Lb/In

The next step in the evaluation of the Phase IV rotor is to evaluate the stiffness and damping coefficients of the Phase IV bearings. The data for the bearings is given in Table 3 on pg. 11. The bearings designed for the Phase IV rotor system are 5 pad bearings with the load between pads(LBP). The pivot location for this design has been changed from 0.54 to 0.50, which is a center pivot location. It will be shown that this is not a particularly desirable feature to have in a 5 pad LBP bearing design because of the problem of having an unloaded top shoe.

Figure 40 represents the bearing pressure profile and bearing coefficients for the phase IV rotor. With the load between pads (LBP) design, the vertical stiffness is greatly reduced. However, with the preload of 0.3, it is seen that the top pad is unloaded. With the center pivot of 0.5, the top pad is unstable.

When the film thickness hp, at the pad pivot location, becomes larger than the pad clearance Cp, then the pad may go into flutter or may lock up, with the leading edge contacting the shaft. This is referred to as shoe *"spragging"*. This condition may be corrected by removing 10% of the pad babbitt from the shoe trailing edge. This, in effect, creates a pad with a 55% pivot location.

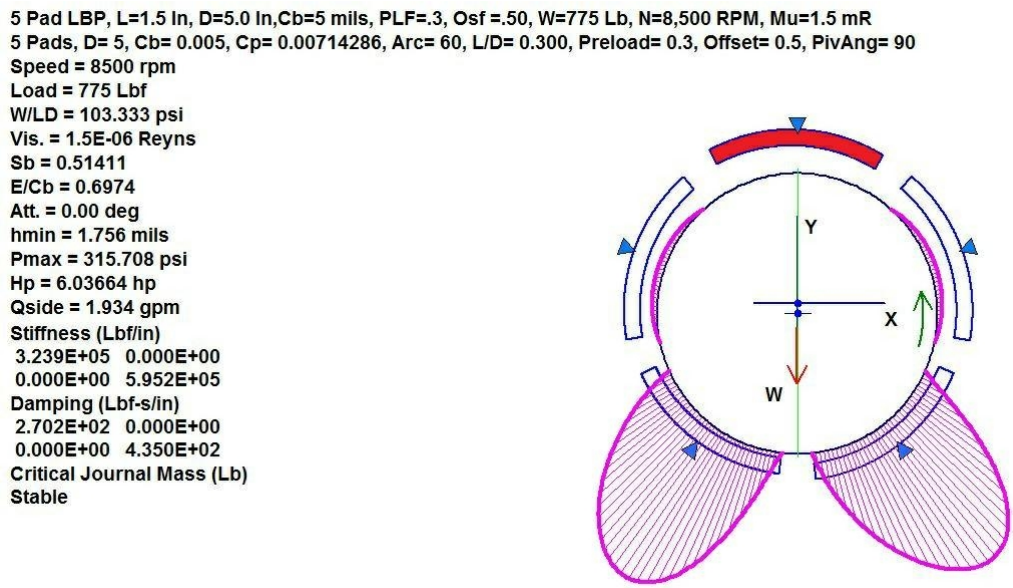

Figure 40 5 Pad LBP Bearing Coefficients For Phase IV Rotor

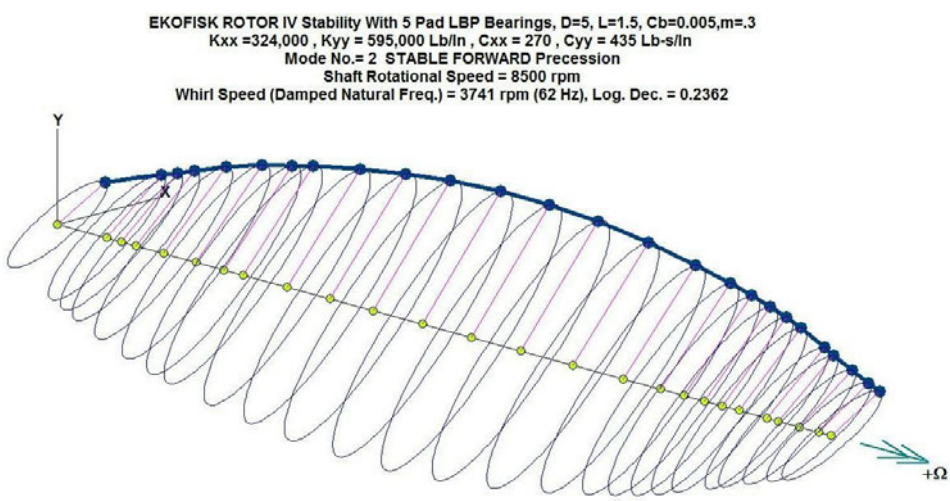

Figure 41 Stable Phase IV Rotor With 5 Pad LBP Bearings at 8,500 RPM
Q = 100,000 Lb/In

The Phase IV rotor with the 5 pad LBP design is highly stable at 8,500 rpm with an aerodynamic excitation force of Q =100,000 Lb/In. The later bearing designs for the Ekofisk rotors were changed to load between pad (LBP) designs as shown in Table 3, pg11. The reason for this is to reduce the vertical bearing stiffness. The 5 pad load on pad (LOP) bearing design is seldom justified. Although the (LBP) bearing design is preferred to load on pad, this bearing is acting as an inefficient 4 pad bearing with m values over 0.3 since the top pad is unloaded !

Discussion and Conclusions

The Ekofisk offshore high pressure barrel compressor was unique in its design as it was the highest pressure compressor application at that time. The Ekofisk compressor used narrow five pad load on pad (LOP) bearings very similar to the design used in the failed Kaybob compressor application. At the time of the design of both the Kaybob and Ekofisk compressors, the technology did not exist to calculate the damped natural frequencies ***(complex eigenvalues)*** of rotors including bearing damping and aerodynamic cross-coupling forces.

Aerodynamic destabilizing cross-coupling forces, as occurring in gas turbines, was first reported by J. Alford of GE in 1965*(3)*. These effects were relatively unfamiliar to compressor designers in the 1970s when Kaybob and Ekofisk were designed. In an effort to protect turbomachinery from high vibrations, the API Code 612 of 1969 did not allow turbines or compressors to operate near the second critical speed.

As such, narrow five pad LOP bearings were chosen for both of these compressors as their high stiffness values placed the second critical speeds above the operating speed range. Shop testing of both of these compressor systems under ambient air conditions indicated no problems. Under high pressure applications, both of these units became unstable and exhibited self excited whirling of the first critical speed.

The extensive research on the Ekofisk seals, in particular, was caused by the initial calculations of the system critical speed, which was substantially lower than the whirl frequency observed on the offshore platform. As a result of the discrepancy between the computed critical speed and the measured whirl frequency, a great deal of effort was expended in the investigation of the seals as responsible for the elevated whirl frequency.

The ability to review the various Ekofisk designs was due to the extensive paper on the history of Ekofisk presented in 2018 by Cloud, Pettinato, and Kocur*(9).* Computer models of the various Ekofisk designs were generated using current finite element methods including disk gyroscopics and bearing support effects*(8)*. The computation of the instability whirl frequency of the original Ekofisk Phase I rotor design matched the measured experimental frequency of 4,400 CPM. Hence it was concluded that the seals did not have a significant effect on the original instability observed on the Ekofisk rotor.

An analysis of the narrow 5 pad LOP bearings for both Kaybob and Ekofisk found out that this design is totally unsuited for use in multi-stage compressors. The reason for this is that the vertical bearing stiffness values generated by the narrow 5 pad LOP designs are excessive in relationship to the fundamental shaft stiffness. The high vertical bearing stiffness values makes the rotors very sensitive to aerodynamic excitation forces. As a result, shorter, stiffer rotors were redesigned to compensate for the excessive vertical bearing stiffness values.

The other option for the 5 pad LOP bearing designs are to install squeeze film dampers. The correct design of these squeeze film dampers have a very narrow window as to the proper values of damper stiffness and damping. O-ring supported bearing cartridges are normally designed with a center groove in the damper. Failure to incorporate the central groove results in a long bearing damper which has excessive stiffness and damping properties. This then leads to the situation referred to as "***damper lockup***". This was the situation with Kaybob. No center groove was installed. The dampers would have been operational if the center grooves had been added, or if the damper clearances had been increased. Instead, the damper clearances were reduced!

It is shown that good stability with the original Ekofisk Phase I rotor could have been achieved by the installation of 4 pad LBP bearings. The use of 4 pad LBP bearing designs is highly recommended over the 5 pad LOP and LBP bearing designs regardless of bearing length. A properly designed multi-stage compressor should never require the addition of squeeze film dampers.

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