Ekofisk Revisited - Bearing Optimization for Improved Rotor Stability

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Xdot Engineering and Analysis

Outline

- Background and introduction to the Ekofisk field problem
- Rotor configurations and shaft models
- Rotordynamic characteristics of original (Phase I) rotor
- These bearings are going to be a disaster....
- Let’s try a squeeze film damper
- What bearing might have worked?
- What about the revised (Phase IV) rotor?
- Closing comments

Background and Introduction

- Two specific failures to predict instability in high pressure reinjection compressors pushed the development of tools and techniques
  - Kaybob, 7 months to fix in field
    - Chevron, Alberta, Canada
    - 1971, 9 stage, reinjection, 3175 psi discharge, 11,400 RPM
  - Ekofisk, 10 months to fix in field
    - Phillips Petroleum, North Sea
    - 1974, 8 Stage, 22,000 HP, 9200 psi discharge, 8426 RPM
- Have been several retrospective papers in the past few years
  - Cloud, Pettinato, and Kocur’s 2018 Turbomachinery Symposium paper on Ekofisk is the inspiration for this paper by E.J.
    - The Cloud, et.al. paper focuses more on seal effects

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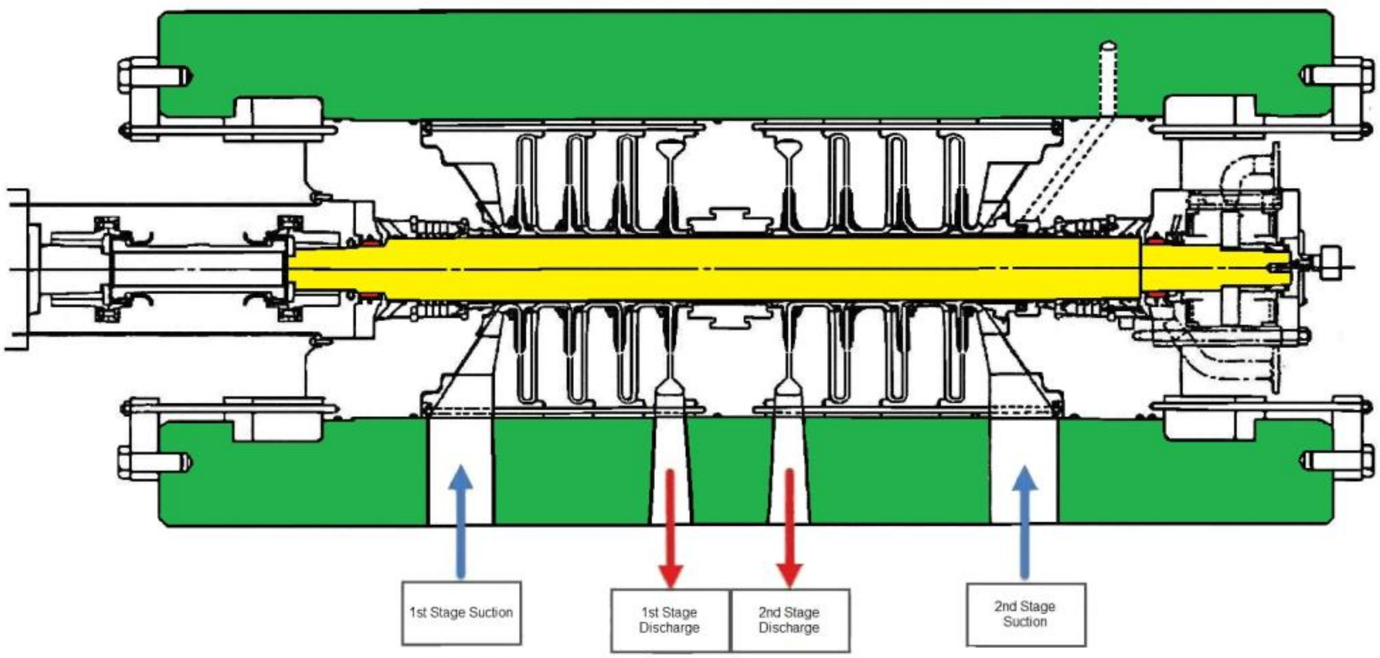
Background and Introduction

- Both failures were encountered for high discharge pressure compressors that were really pushing the state of the art at the time
  - Machines went unstable due to aerodynamic (Alford) cross-coupled stiffness forces
  - Were not bearing induced instability
    - Lund's 1964 work had showed that plain journal bearings are unstable
    - Both compressors had narrow five-pad, load-on-pad tilting pad bearings
- 1970's state of the art transfer matrix codes could compute undamped critical speeds and unbalance response
  - But they could not adequately analyze rotordynamic stability

Background and Introduction

- Three other helpful pieces of background information
  - API specs at the time dictated that compressors should not operate near critical speeds
    - Compressor developers were motived to avoid operating near the second critical speed
    - This requirement influenced bearing choices
  - Bently proximity probes were starting to be installed in critical machinery such as these compressors, so there is some data available
  - Both machines looked fine during the mechanical run test at low pressure/low power

Ekofisk Compressor



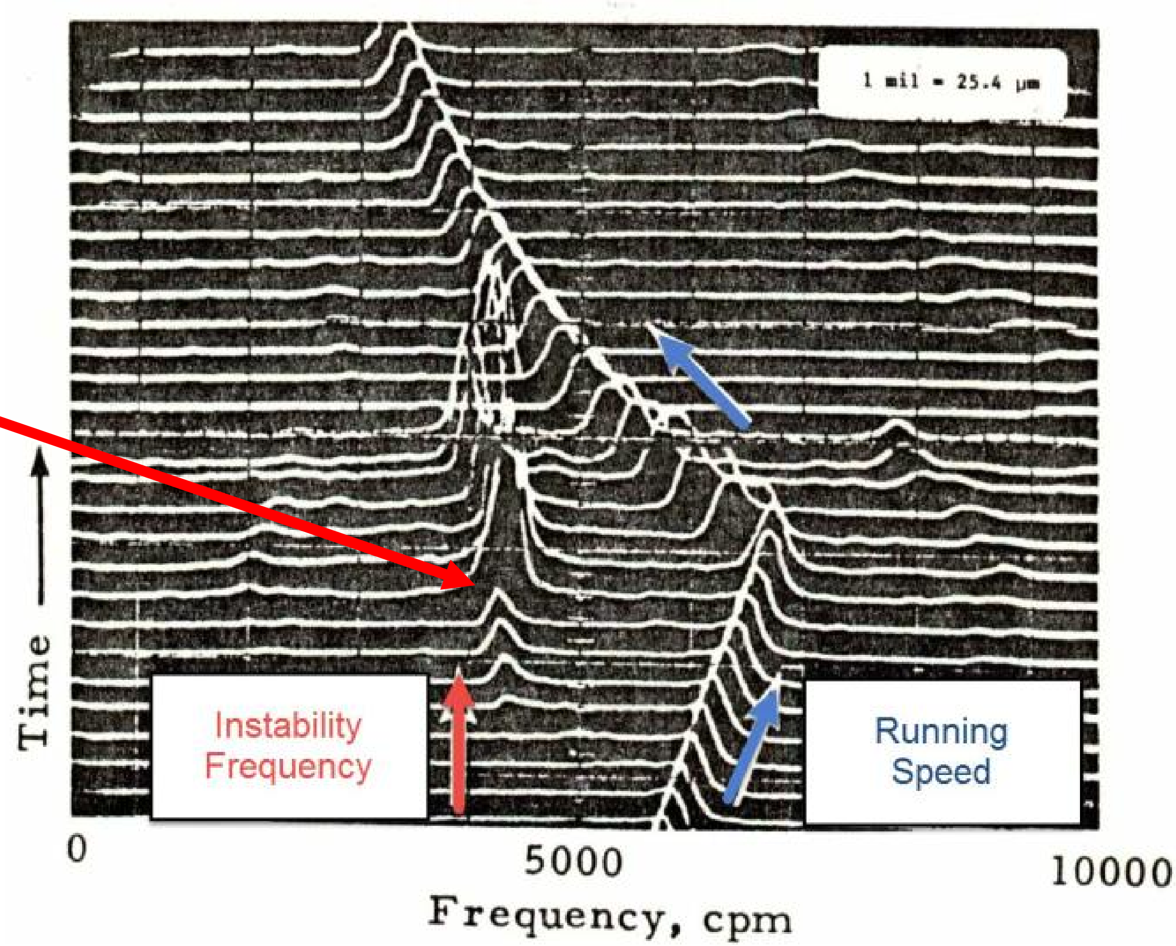
Elliot 25 MBHH

(2^{nd} of two compressors in series)

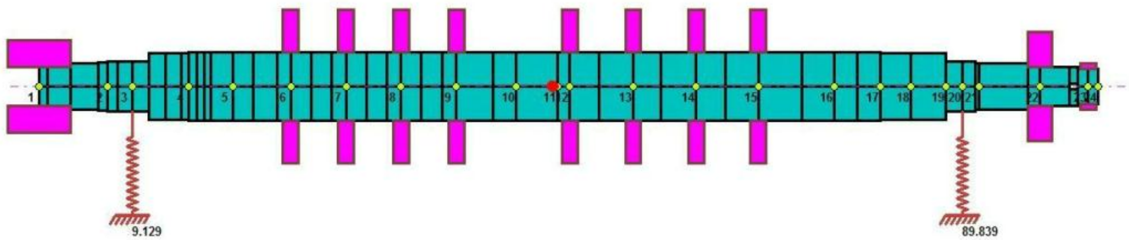
Startup Vibration Waterfall Plot

4400 CPM

(had been calculated to be 3800 cpm, with a <u>rigid</u> bearing critical speed of 4200 cpm)



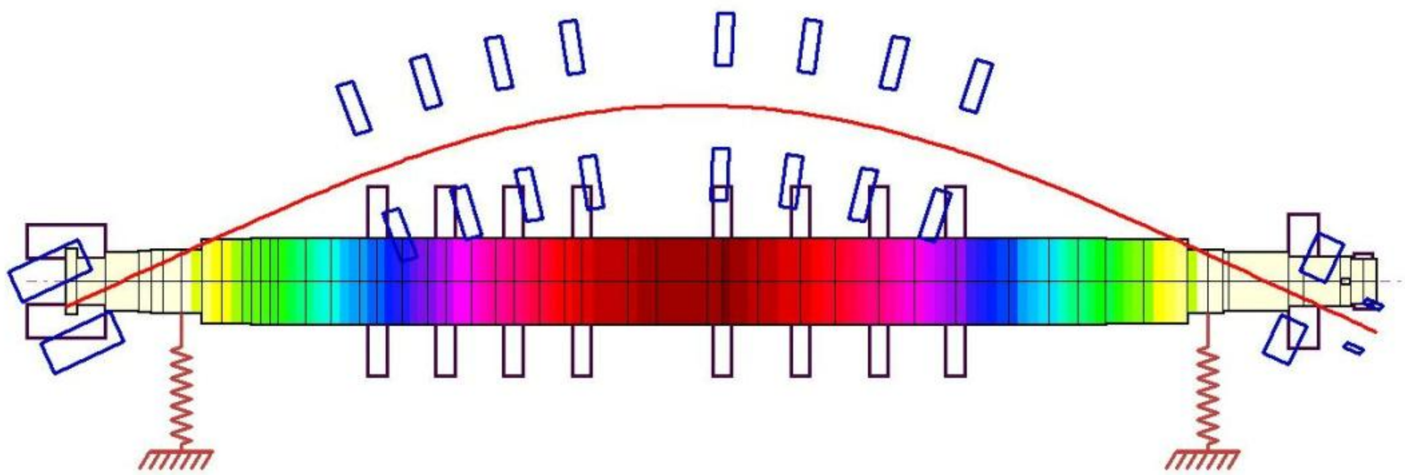
Phase I Rotor Modeling



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

- Dyrobes model based on a rotor model presented by Cloud et., al.
  - Includes disk gyroscopic effects (not included in earlier work?)
  - 80.7 inch bearing span
  - 6.7 inch main diameter
  - L/D ratio 12.0
    - High!

1st Undamped Critical Speed



Phase I Rotor First Critical Speed

Kbrg =1.0e6 Lb/in , N1 = 4,352 RPM

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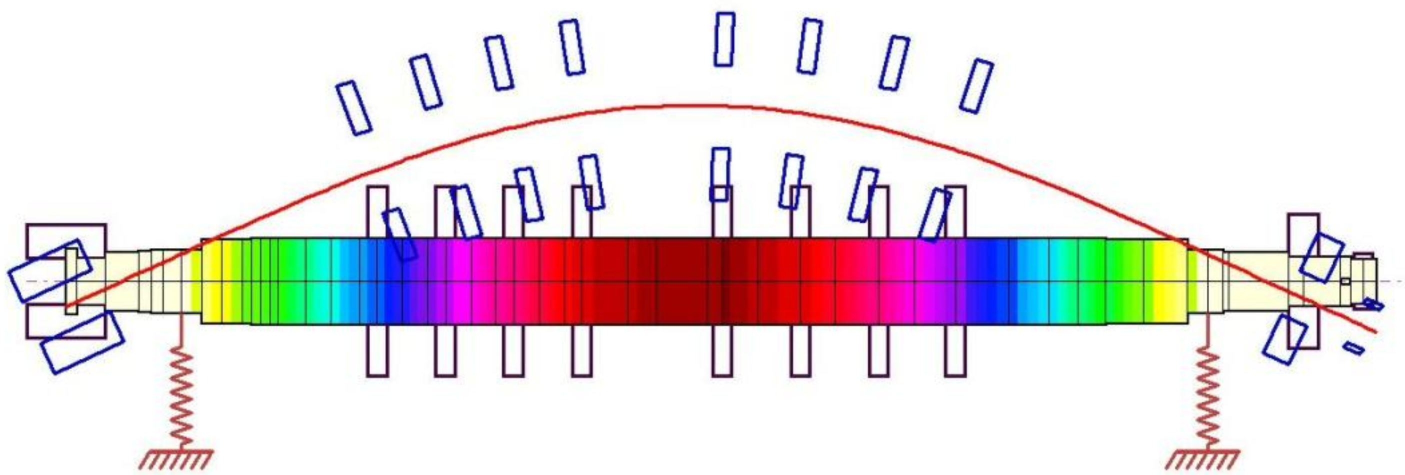
1st Undamped Critical Speed



Phase I Rotor First Critical Speed

Kbrg =1.0e6 Lb/in , N1 = 4,352 RPM

Close Correspondence with measured 4400 cpm instability

(had been calculated as 3800 cpm!)

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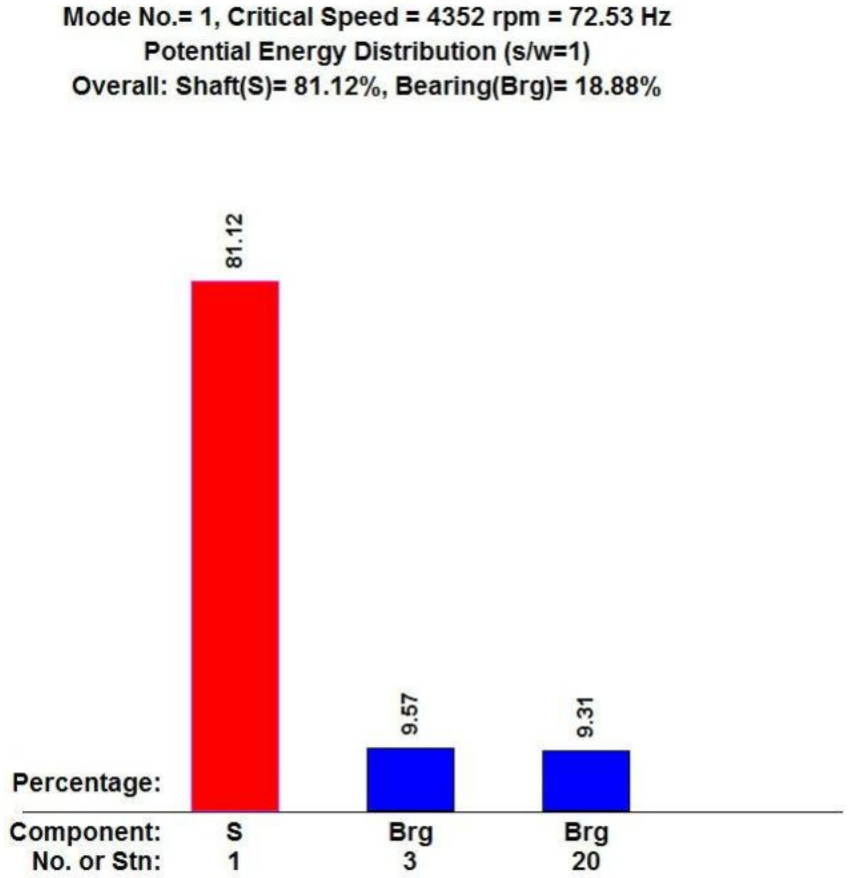
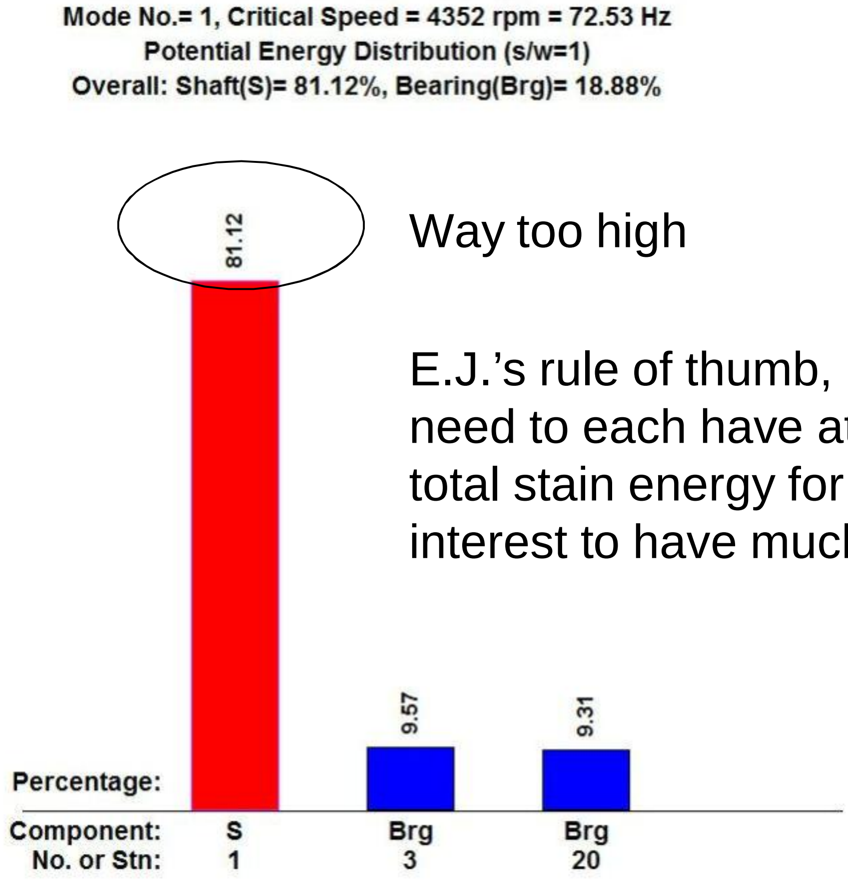
1st Undamped Critical Shaft Strain Energy



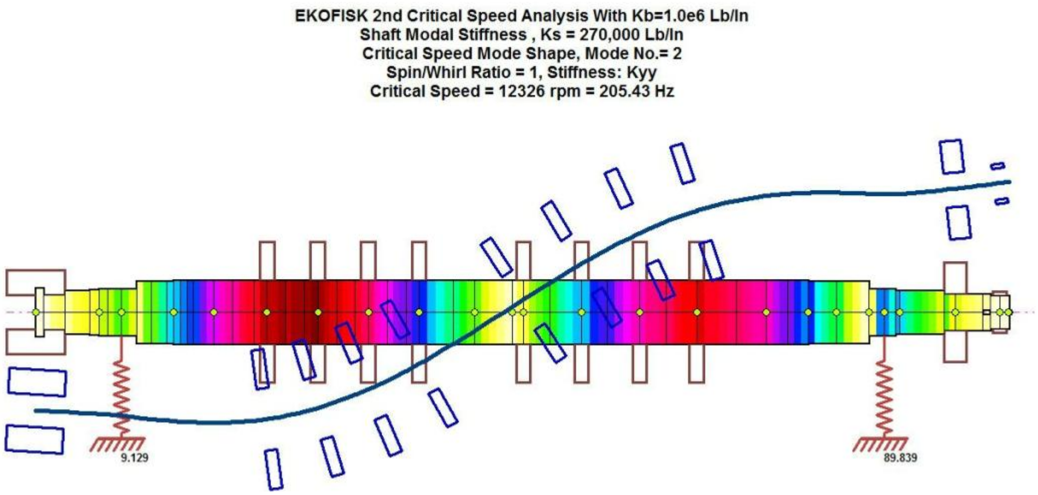
1st Undamped Critical Shaft Strain Energy



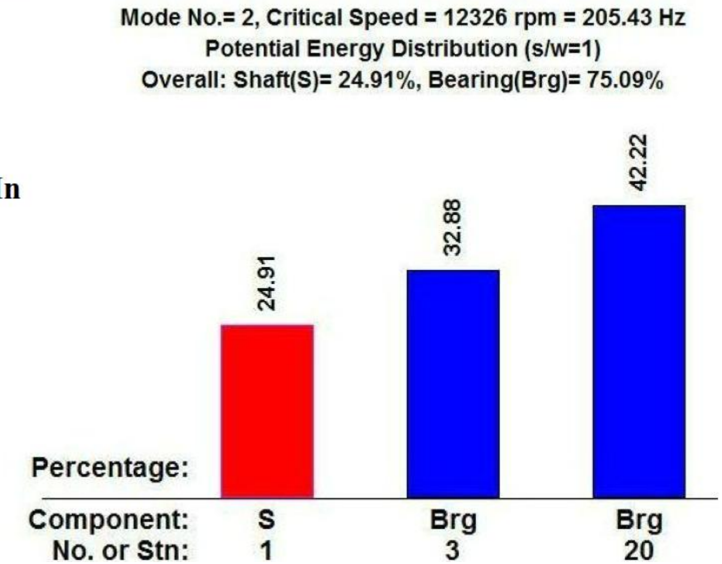
Way too high

E.J.'s rule of thumb, bearings need to each have at least 20% of total stain energy for mode of interest to have much effect

2nd Undamped Critical Speed

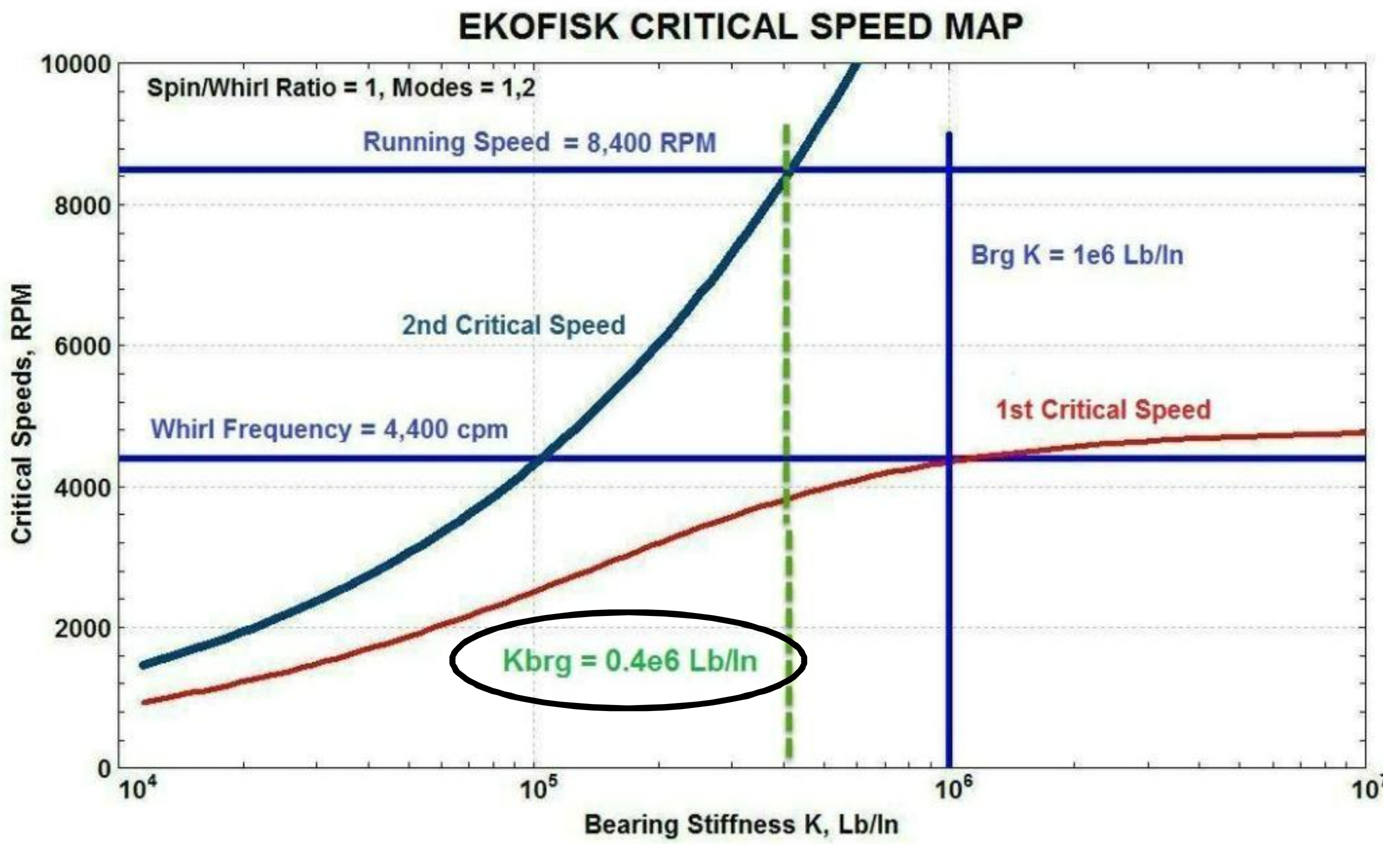


Phase 1 Rotor Predicted Second Critical Speed with Kb=1.0 Lb/In
Nc2 = 12,326 RPM

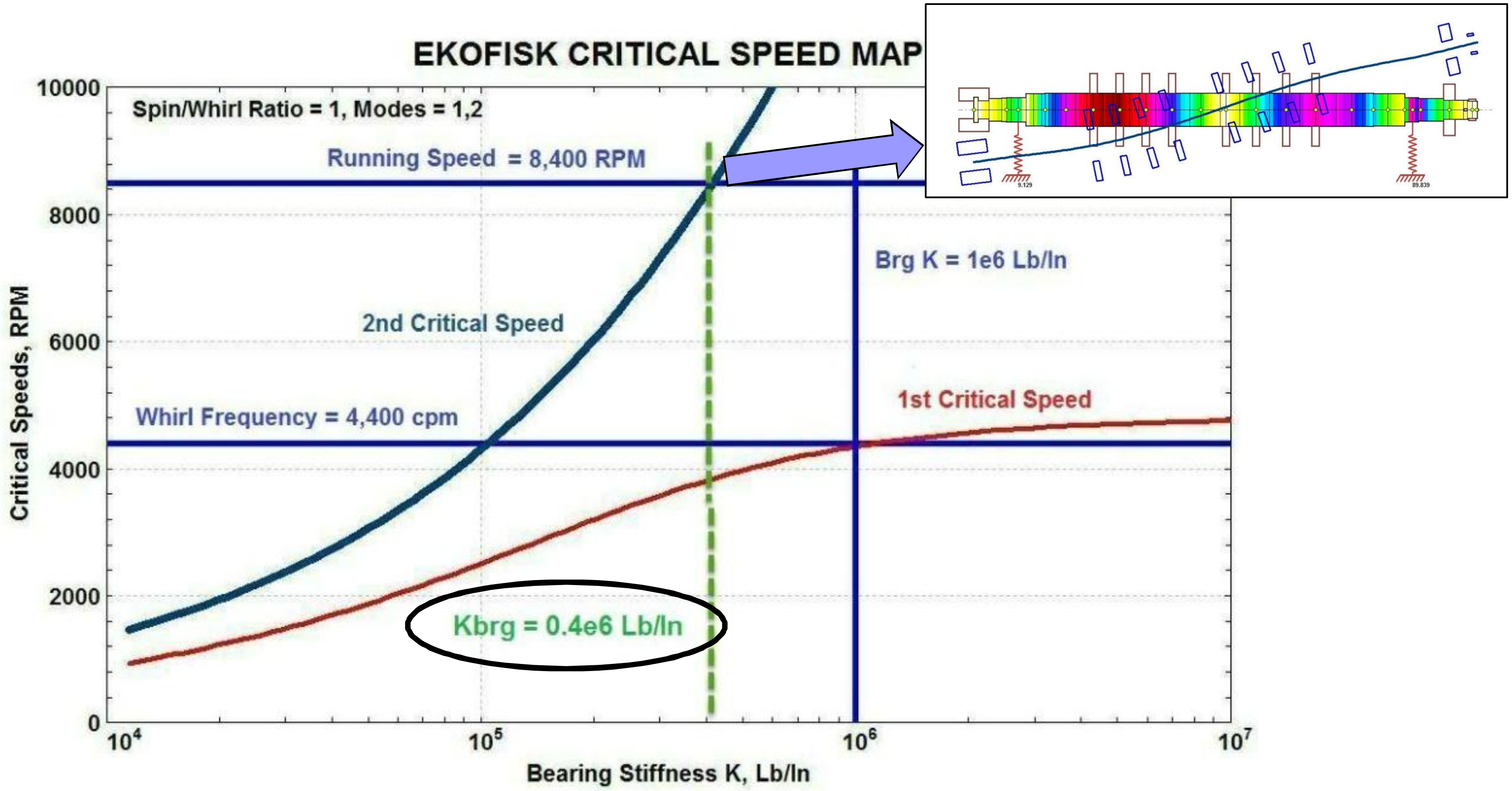


Plenty of bearing strain energy, bearing damping will be very effective

What if the Bearing were Softer?



What if the Bearing were Softer?



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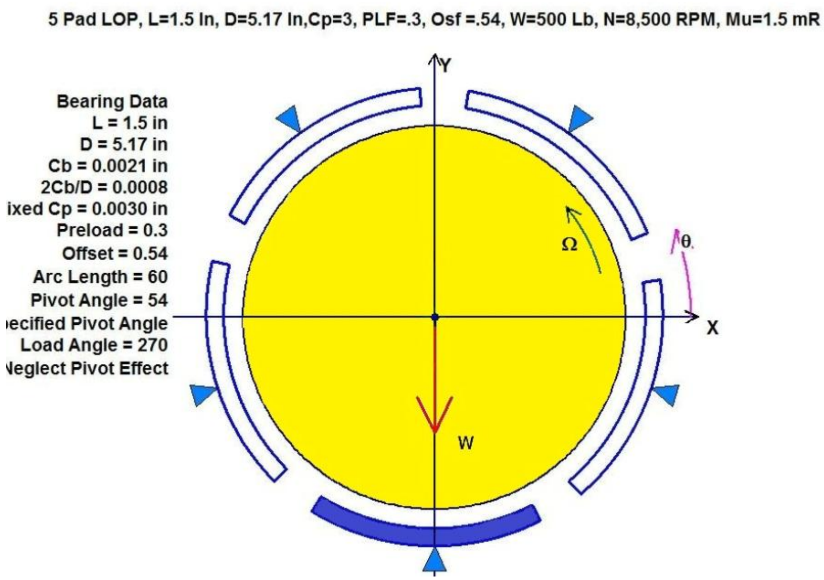
Bearing Analysis Using Dyrobes

- Phase I bearings are narrow (L/D = 0.284), 5 pad, load on pivot tilting pad bearings
  - 0.54 offset
  - 0.29 – 0.53 preload (nominal preload = 0.3 used for this paper)
  - Predicted Kyy stiffness: 1.292e6 lbf/in



Some Observations

- For a simple flexible rotor, it can be shown that there is an optimum shaft to bearing stiffness ratio

  Kyy_ratio = $\frac{2Kyy}{Ks}$,   Kyy_ratio_optimal = 1

- Using the shaft modal stiffness of 2.7e4 lbf/in

  Kyy_ratio_original = $\frac{2Kyy}{Ks} = \frac{2(9.5e5)}{2.7e4}$ = 7.03

  Kyy_ratio_Dyrobes= $\frac{2Kyy}{Ks} = \frac{2(1.29e6)}{2.7e4}$ = 9.56

Some Observations

- For a flexible rotor with optimal stiffness, there is also an optimal damping
- Assuming optimal damping and stiffness, an estimate of the amplification factor can be made

A_optKC = 2 (1 + K_ratio)

- Thus, even if we had the optimal damping, the amplification factor at the first critical is expected to be in the range of <u>14 to 19</u>
  - This suggests the rotor will be quite sensitive to aerodynamic instability drivers

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But Wait, It Gets Worse!

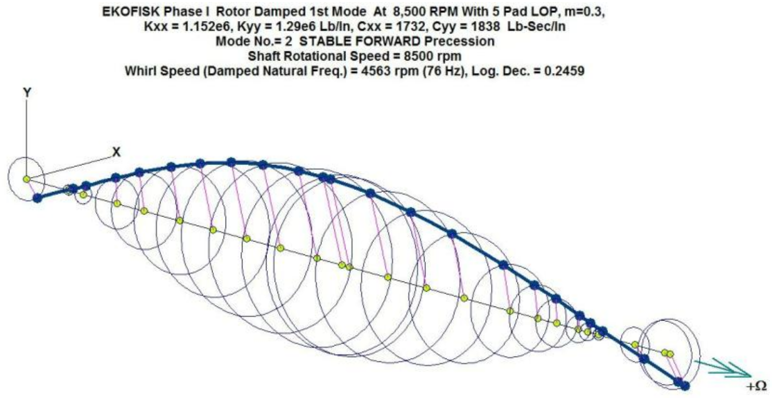
- Dyrobes predicts the tilting pad bearing direct dynamic coefficients to be

  Kxx = 1.15e6 lbf/in   Cxx = 1,731 lbf-s/in

  Kyy = 1.29e6 lbf/in   Cyy = 1,837 lbf-s.in

- However, we know that support and pivot stiffness effects degrade the effective damping. E.J.'s preliminary rule of thumb is derate by 50% for initial calculation in this compressor
  - He suggests the more precise approach is to include a decent estimate of the actual bearing support stiffness
  - How good does his rule of thumb do?

But Wait, It Gets Worse!

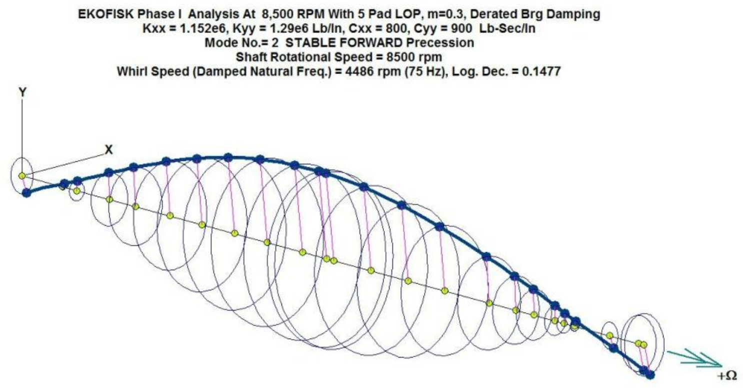


First mode at 8500 rpm

<u>No adjustment</u>

N = 4563 cpm

Log. Dec. = 0.25



<u>50% Derate</u>

N = 4486 cpm

Log. Dec. = 0.15

Measured Instability 4400 cpm

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Rotor Stability

- An approximate rotor stability limit for a symmetric, flexible rotor is given by (I think this assumes optimum damping)

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

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*

$$\boxed{\mathbf{Q = 17{,}500\ Lb/In}}$$

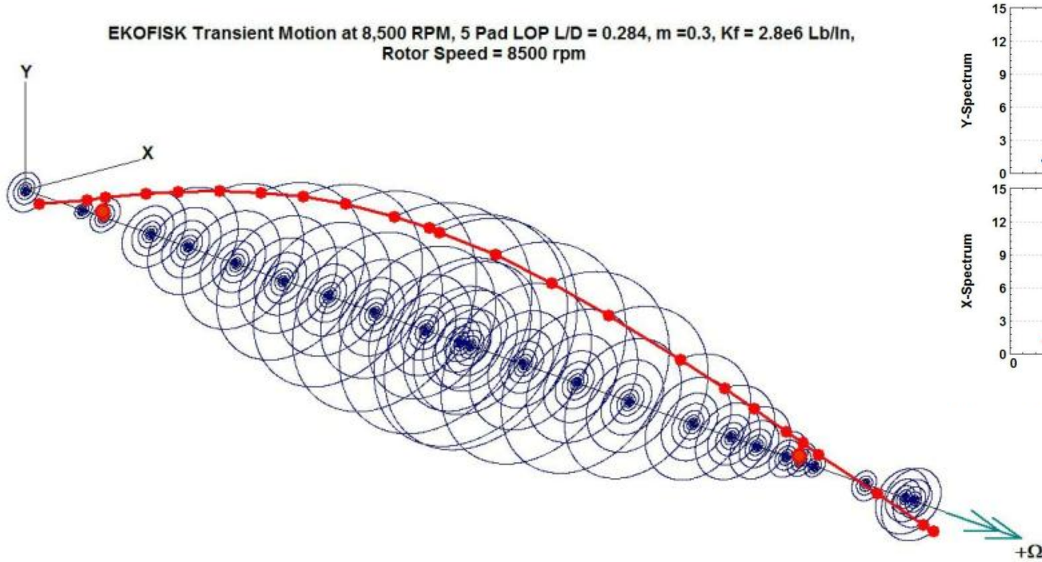
- This is well below the desired 100,000 to 200,000 lbf/in
  - Cloud et al. give the API Level 1 cross coupling as 207,000 lbf/in

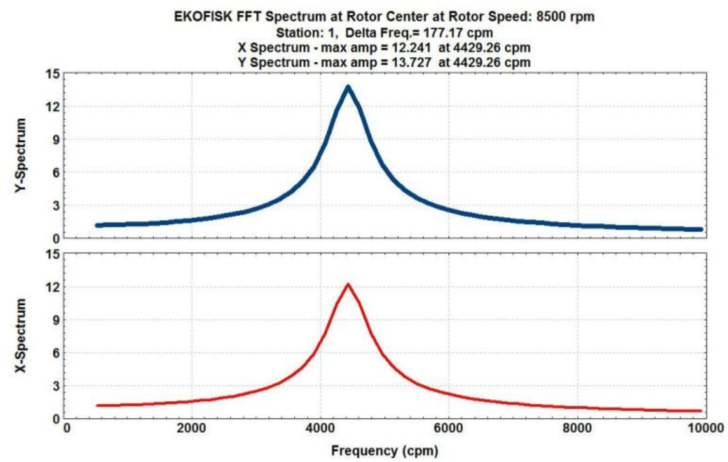
Rotor Stability

- This compressor will almost certainly be unstable!
  - Even without consideration of oil ring seal effects
- Fundamental issue is that the narrow five pad LOP bearings are too stiff
  - They were probably selected to push the second critical speed up to well above the operating speed range
  - Causes the rotor to be very sensitive to self-excited whirling forces
  - This characteristic is also sometimes seen in more modern compressors if tilting pad bearings with large offset are used
    - A machine with stiff bearings relative to the shaft stiffness is almost always undesirable unless the machine is running below the first critical speed

Time Transient Response

- Modern rotordynamic tools such as Dyrobes can perform a time transient response with the full bearing calculation performed at every time step
  - These tools were not available in the 1970's
  - Will show the instability





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

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Solution 1 – Squeeze Film Damper

- Paper has considerable detail
- Can use optimal support stiffness (half shaft modal stiffness) for an initial estimate of target damper stiffness
  - Too high damper stiffness reduces the amount of acceptable aerodynamic cross-coupling
- Can run parametric study to determine the range of damping that works and the maximum amount of aerodynamic cross-coupling that can be applied and the system remain stable
  - Not enough damping -> the amount of cross-coupled stiffness that can be applied drops
  - Too much damping -> damper “lock-up” and is not effective

Solution 1 – Squeeze Film Damper

Maximum Q values For Various Support Stiffness and Damping

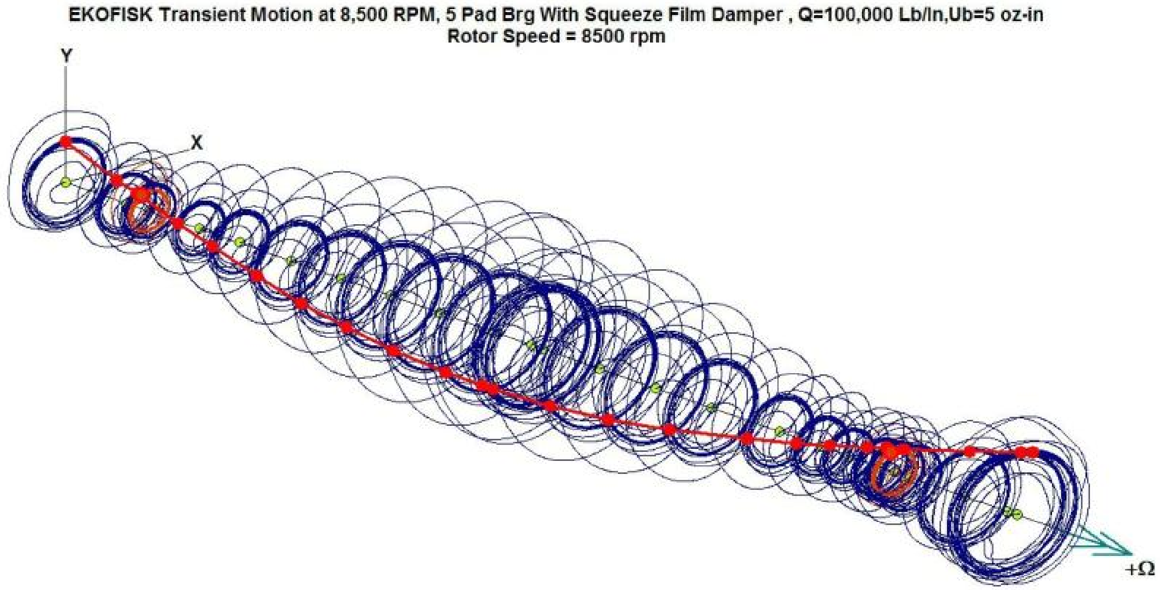
K*f support, Lb/In*	C*f support damping, Lb-S/In*	Maximum **Q** ***Lb/In***
150,000	600 - 900	***100,000***
300,000	1,000 - 1,300	75,000
500,000	1,200 - 1,800	60,000

All are much higher than the original limit of less than 20,000 lbf

Solution 1 – Squeeze Film Damper

- Sizing and Dimensions
  - Dyrobes has a calculation tool that can be used
- Central circumferential groove
  - Be very careful about central grooves, some equations assume the damper has a central circumferential groove. If it is modeled, but not present in the hardware, there is a high risk of lockup
    - E.J. believes this was a problem with one of the Kaybob fixes
- O-Rings
  - O-ring end seal stiffnesses must be included in parallel with the damper coefficients if o-rings are used to seal the ends of the damper
  - Be aware of the potential for shaft weight to crush the o-ring
    - Centered dampers are preferred

Solution 1 – Squeeze Film Damper



Transient Motion With Nonlinear Squeeze Film Dampers

Kf =100,000 Lb/In, Cr = 8 mils, Q = 100,000 Lb/In - Stable System

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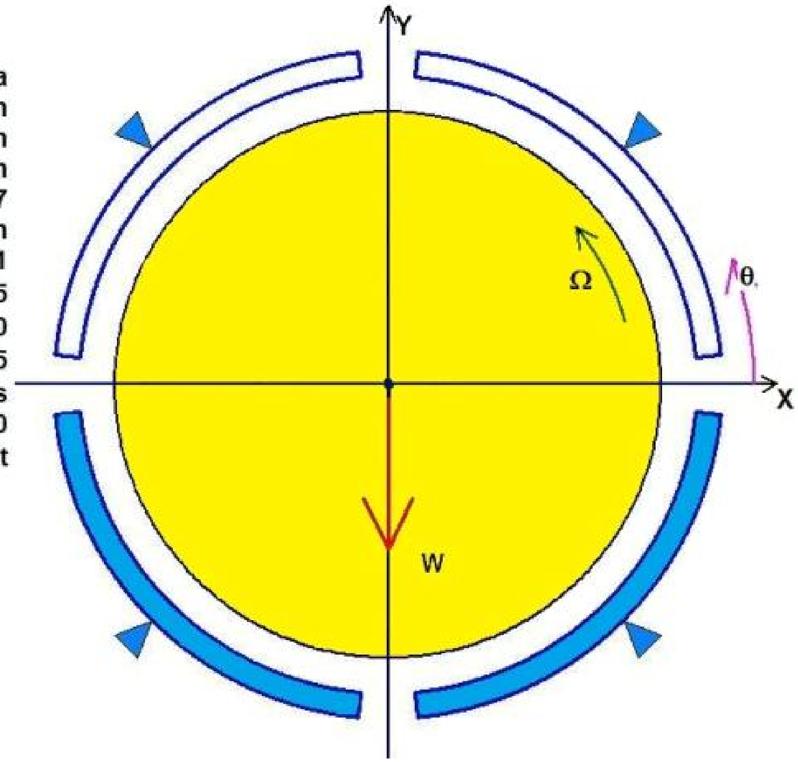
Solution 2 – Softer Bearing, 4 Pad LBP

Longer bearing (was 1.5 inch)

Bearing Data
L = 3 in
D = 5.17 in
Cb = 0.0045 in
2Cb/D = 0.0017
Fixed Cp = 0.0050 in
Preload = 0.1
Offset = 0.5
Arc Length = 80
Pivot Angle = 45
Load Between Pivots
Load Angle = 270
Neglect Pivot Effect

4-Pad LBP EKO I, D=5.17, L= 3 in, Cb=0.0045, m=0.1



Stiffness (Lbf/in)

1.662E+05	0.000E+00
0.000E+00	1.662E+05

Damping (Lbf-s/in)

1.006E+03	0.000E+00
0.000E+00	1.006E+03

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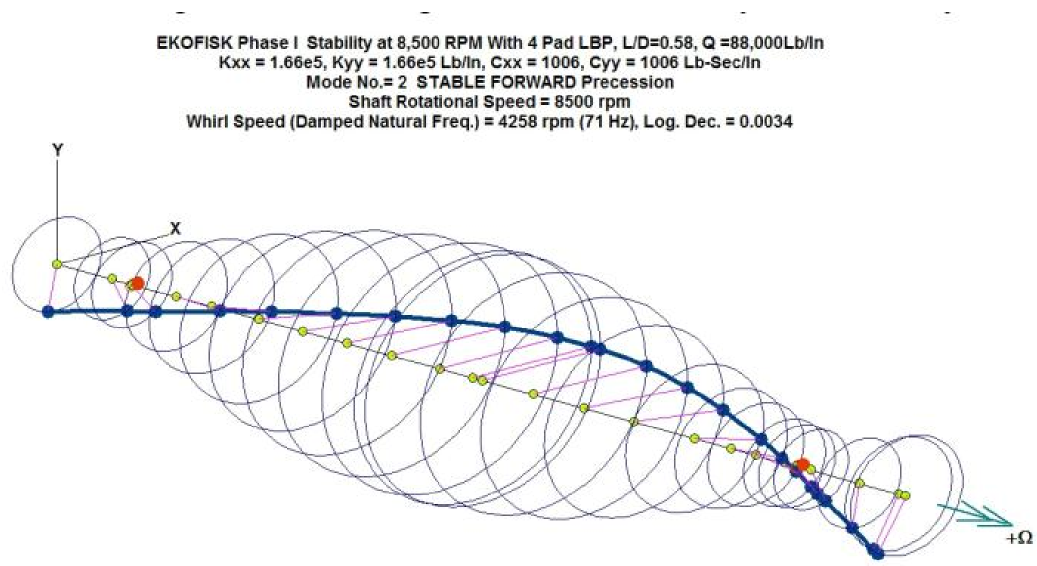
Solution 2 – Softer Bearing, 4 Pad LBP



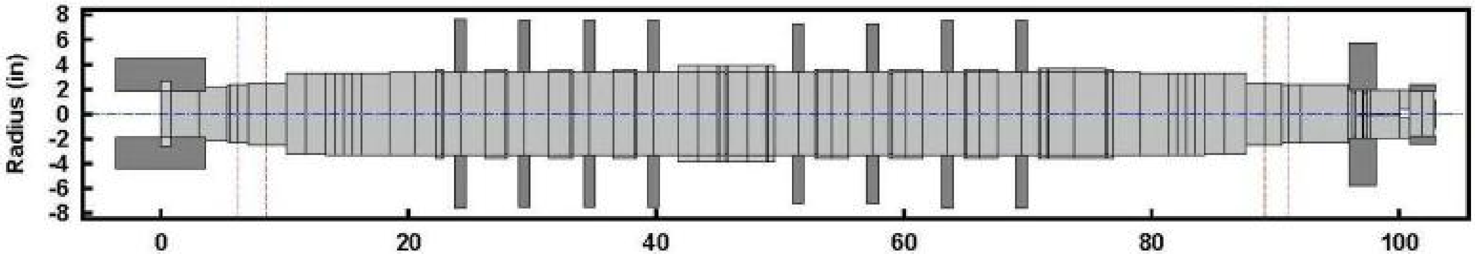
EKO I Rotor Stability With Four Pad LBP Bearings

Kxx = Kyy = 166,000 Lb/In $\boxed{\textit{Qmax = 88,000 Lb/In}}$

Solution 2 – Softer Bearing, 4 Pad LBP

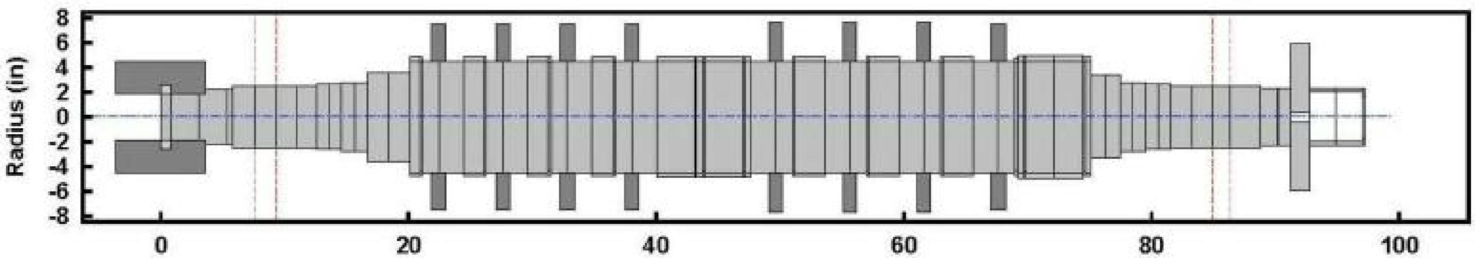
- But wait ... what about the second critical speed?
- The softer bearing has enough damping that the second mode is an overdamped, rigid body conical mode
  - Does not get excited
- Next critical speed is well above operating speed
- The early 1970's tools would have had a hard time verifying that the second critical speed was not a concern

Solution 3 – Shorter, Larger Diameter Rotor



(a) Phase I

L/D = 12



(b) Phase IV

L/D = 8.4

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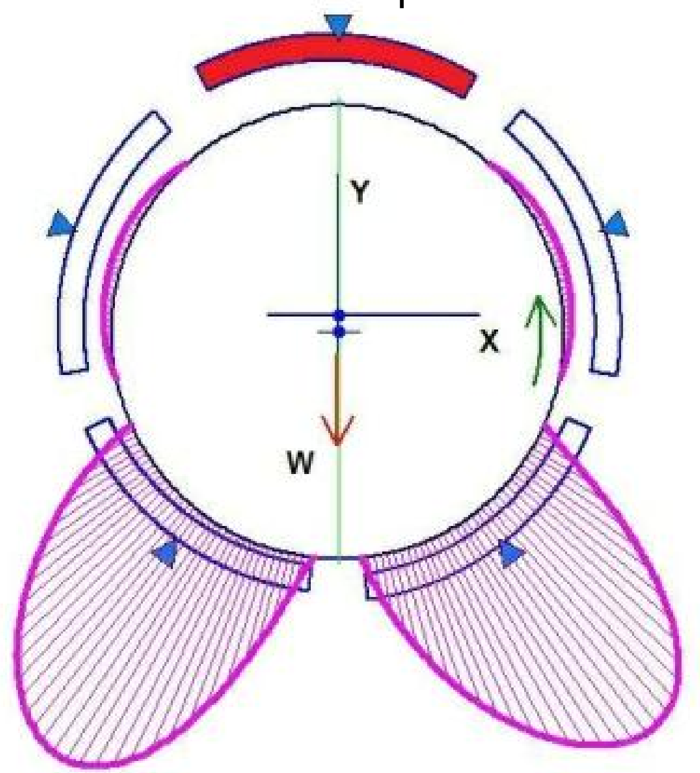
Solution 3 – Clock Bearing to be LBP

Also went to center (50%) pivot and increased clearance which reduces preload

Unloaded top pad – flutter risk



Stiffness (Lbf/in)

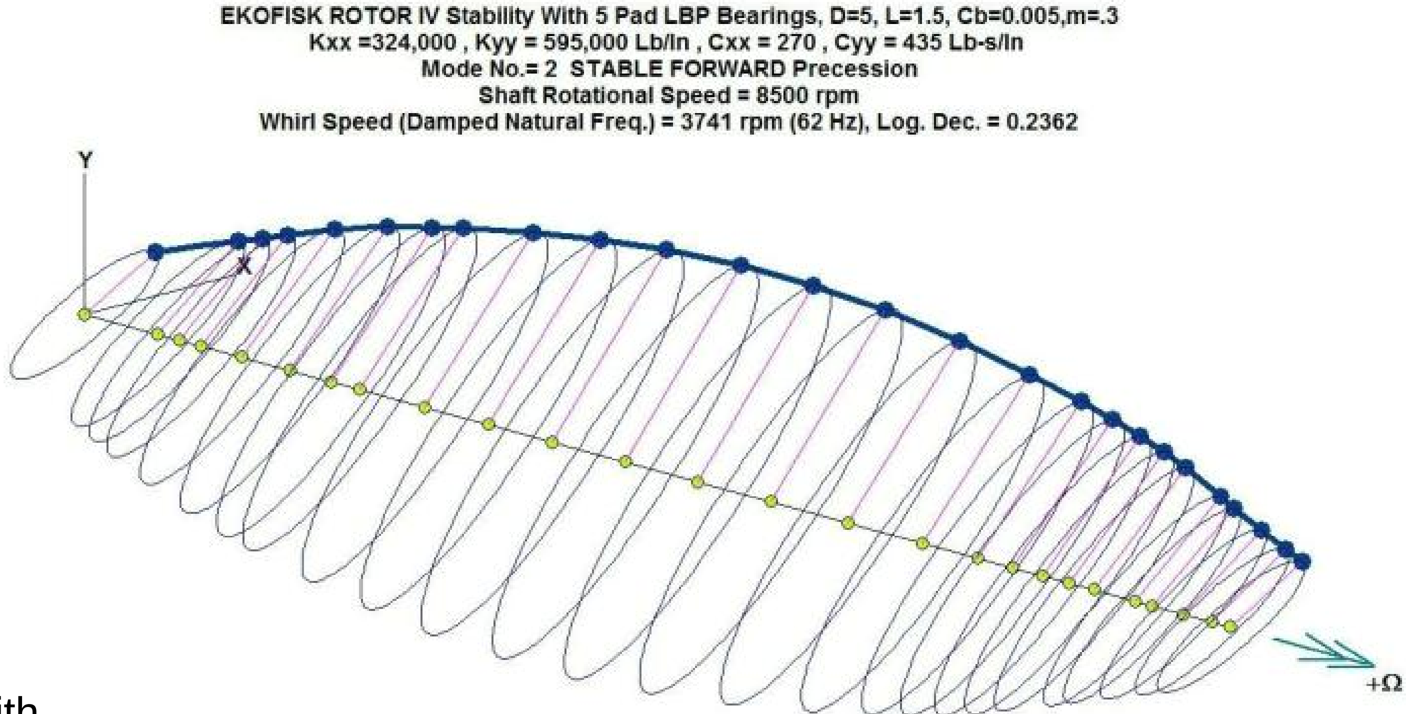
3.239E+05	0.000E+00
0.000E+00	5.952E+05

Damping (Lbf-s/in)

2.702E+02	0.000E+00
0.000E+00	4.350E+02

Arguably, this is actually a 4 pad LBP bearing, since the top pad is not doing much

Solution 3 – Clock Bearing to be LBP



Stable with
100,000 lbf/in
cross-coupling

Discussion and Conclusions

- Design practices and codes of the time drove the Ekofisk design towards a long (flexible) shaft and narrow, very stiff bearings
- This is generally not a desirable combination
  - Modal strain energy is all in the shaft
  - Bearing damping is not vey effective in reducing amplification factors or providing rotordynamic stability
- The Dyrobes re-analysis does a very good job matching the observed instability frequency without considering oil seal effects
  - Original analysis did not match very well

Discussion and Conclusions

- Solutions
  - Major rotor redesign
    - Implemented in 1974
    - Shorter, larger diameter rotor (stiffer)
    - Switched to load between pivot bearings with more clearance (softer)
  - Squeeze film damper
    - Carefully designed damper probably would have worked with original rotor
  - 4 pad load between pads bearing
    - Carefully designed 4 pad bearings probably would have worked with original rotor

Thank-You for Listening!