Failure of a Test Rig Operating with Pressurized Gas Bearings: a Lesson on Humility

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Abstract

Process fluid lubrication in rotating machinery offers compactness and efficiency as it dispenses with complicated oil lubricant supply systems. Prior work in a dedicated test rig demonstrated the performance of water lubricated bearings operating at high speed and high load conditions applicable to a cryogenic turbo pump. The test rig was revamped to operate with gas bearings, externally pressurized and flexure pivot-tilting type, in a program aiming to measure the performance of gas thrust bearings for micro turbomachinery.

Maiden tests began with the bearings supplied with air at 7.9 bar, then 6.5 bar, and at 5.1 bar, while the shaft speeds reached 25 krpm (surface speed=50 m/s). The data recorded showed a very lightly damped system with a critical speed at ~ 6 krpm, and susceptible to excite sub synchronous whirl motions (SSV) when the rotor turned with a speed above twice the first critical speed.

Ignoring the initial warnings, the rotor was brought to a high speed of 28 krpm and a low air supply pressure equaling 5.1 bar. Suddenly, the shaft experienced large amplitude SSVs, contacted the bearings, and produced a catastrophic failure: a broken coupling, a twisted rotor, sheared covers, and welded pads into one of the bearing casings.

Post-mortem analysis demonstrates the failure is a whirl instability of the first rigid body rotor-bearing mode, and a rotor dynamics model, which includes the quill shaft and diaphragm coupling, produces predictions in agreement with the last vibration data set acquired prior to the humiliating incident.

The sobering experience demonstrates the need for following proper operating procedures while also paying attention to early evidence that could have prevented the costly mishap.

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BS and MS (2015) in Mechanical Engineering at Texas A&M University. Rotating machinery engineer at Exxon-Mobil Baton Rouge Refinery.

Objective

Evaluate the performance of gas lubricated hybrid radial and <u>thrust</u> bearings for high speed rotating machinery.

Externally pressurized bearings allow **rub-free** operation during rotor **start up** & **shut down**.

Major issues with gas bearings:

Little damping & Instability (whirl & hammer)

Brief Literature Review

½ frequency whirl is well known in (lightly loaded) hydrodynamic bearings. The whirl is usually benign until locking to a natural frequency to produce an instability (large amplitude motion).

Osborne and San Andrés (2006) **ASME J. Eng. Gas Turbines Power, 128**

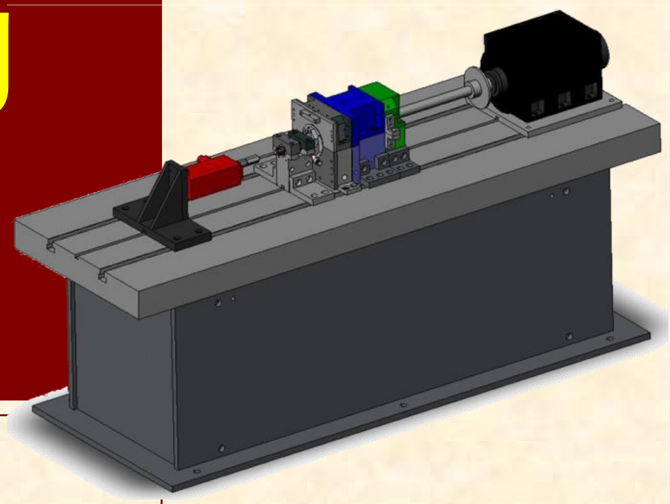
On a rigid rotor on hydrostatic gas bearings: As supply pressure into bearings increases, the system critical speed increases and the damping ratio decreases → the threshold speed of instability increases.

Int. Conf. on Noise and Vibration Engineering

Waumans et al. (2006, 2011) **Int. Conf. on Sustainable Construction & Design**

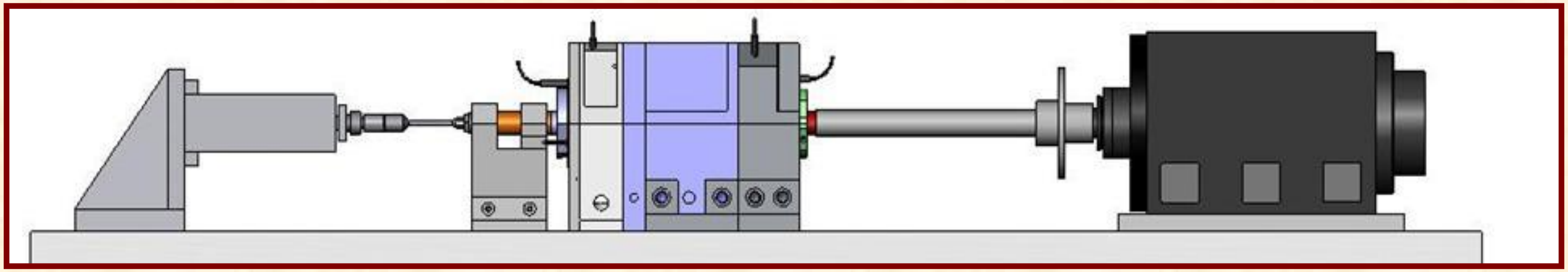
Micro-turbine rotor on gas bearings: Sub synchronous whirl is more apparent at low supply pressure and high rotor speed. Methods to improve the stability of gas bearings → decrease film clearance and modify bearing geometry.

A thrust bearing test rig



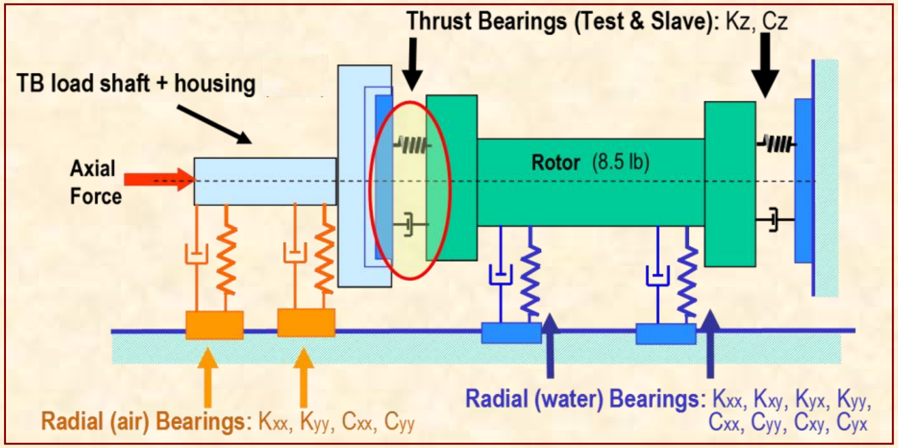
Funded by **USET (AF) program** (2005-2008)

Program Objectives



Hydraulic shaker load shaft test rotor coupling drive motor

- **Validate hybrid thrust bearings predictive tools for application to cryogenic turbo pumps.**
- **Measurements of forced performance of water lubricated hybrid thrust bearings for operation with high speed (25 krpm) and high supply pressure (70 bar).**
- **Measurements show remarkable correlation with predictions from XLHYDROTHRUST® model.**



Test Rig Description

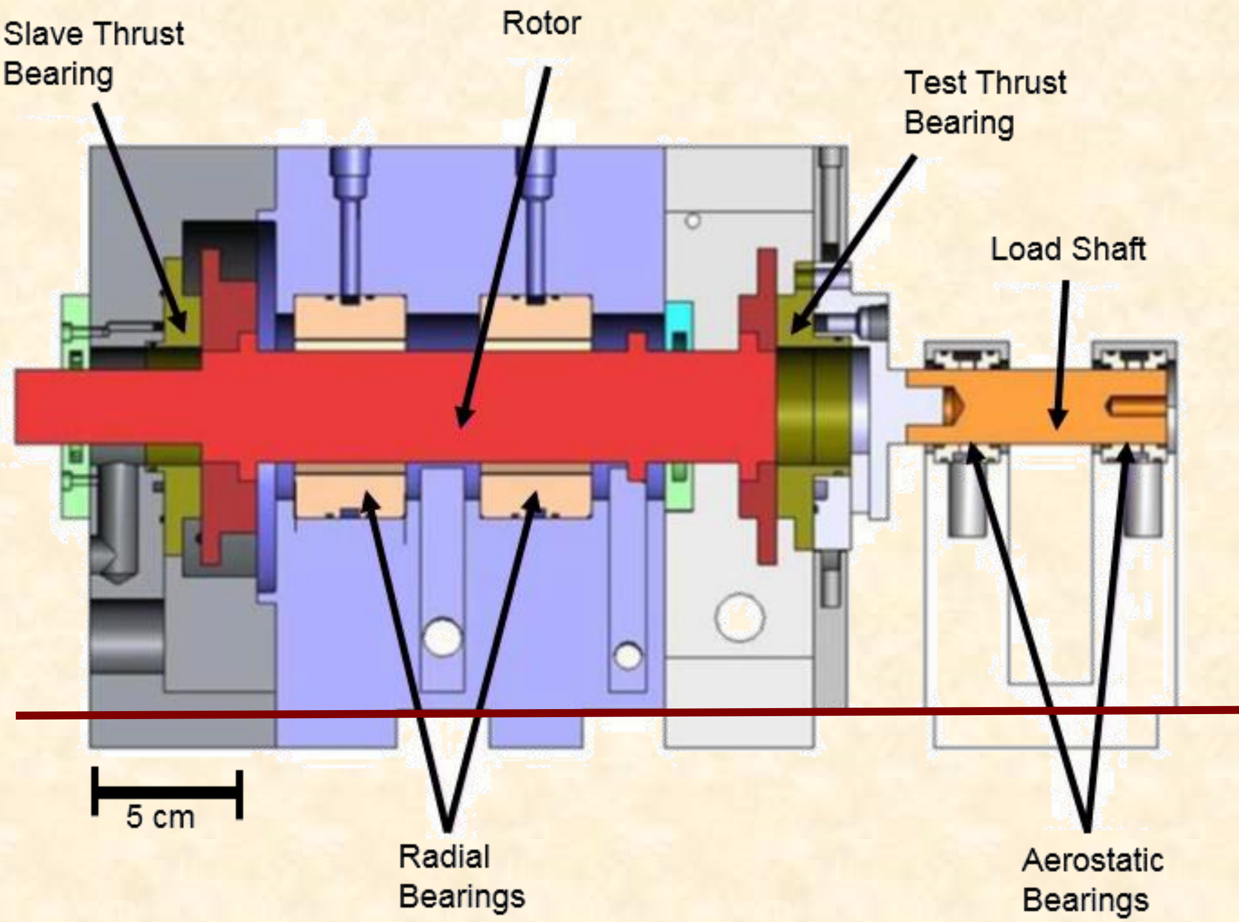
Water lubricated bearings

(a) **Motor drives rotor through coupling.**

(b) **Two radial bearings support rotor.**

(c) Load shaft **applies load to** **test thrust bearing** **and pushes on thrust collar in the shaft.**

(d) **Rotor displaces and** **slave thrust bearing** **reacts load.**



Test Rig Description

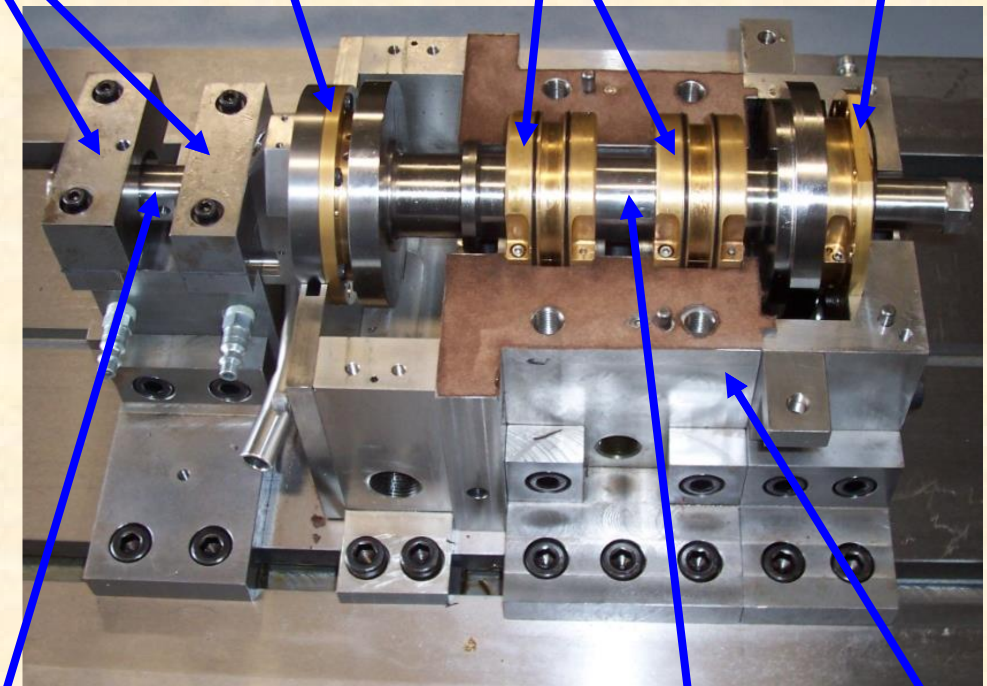
Water lubricated bearings

Aerostatic Bearings

Test Thrust Bearing

Journal Bearings

Slave Thrust Bearing



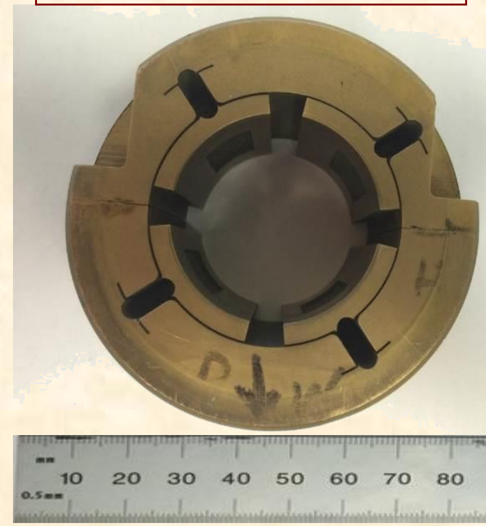
0 5 10 cm

Load Shaft

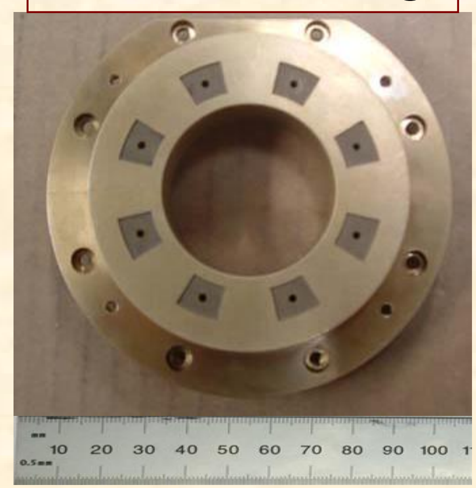
Rotor

Housing

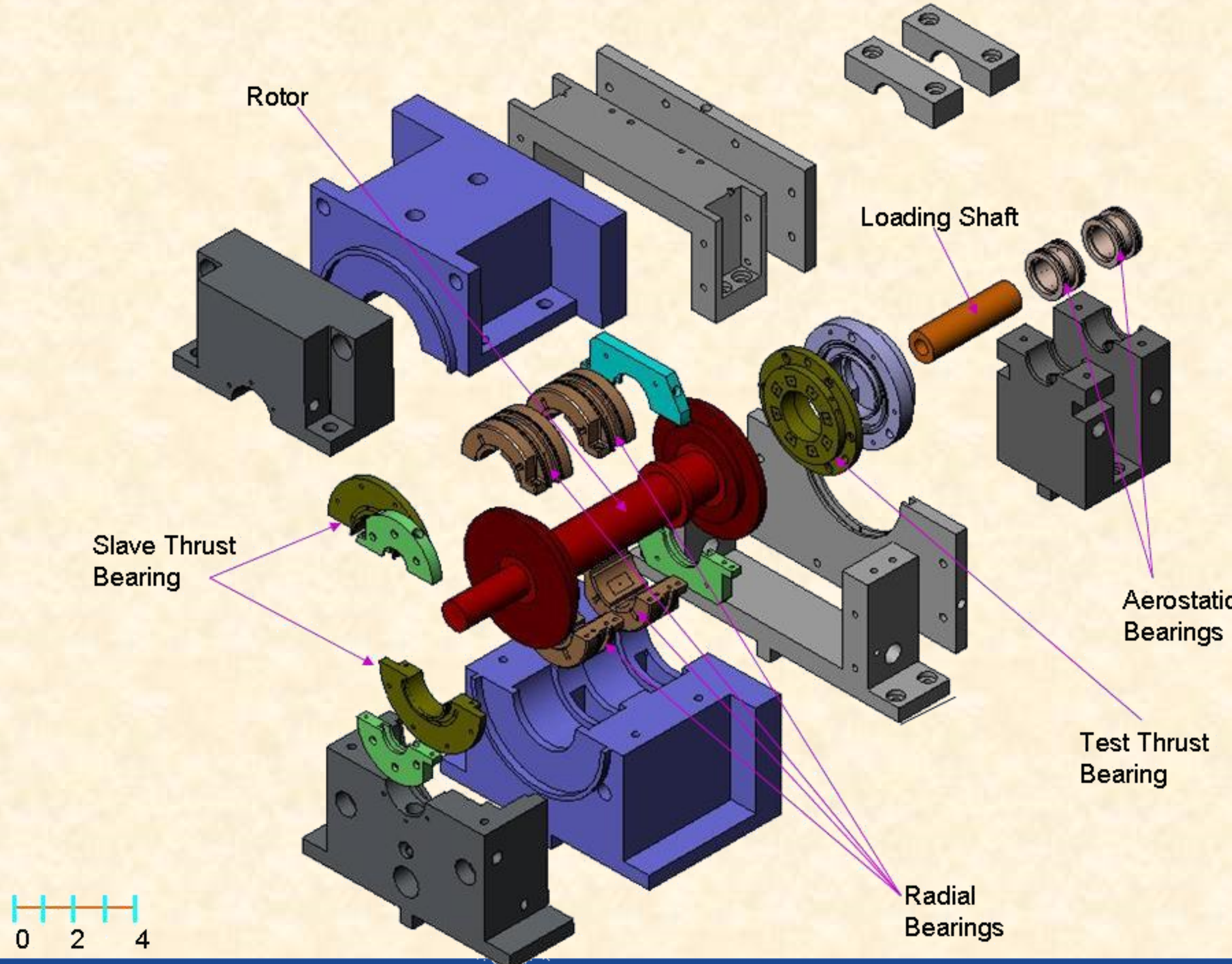
Radial Bearing



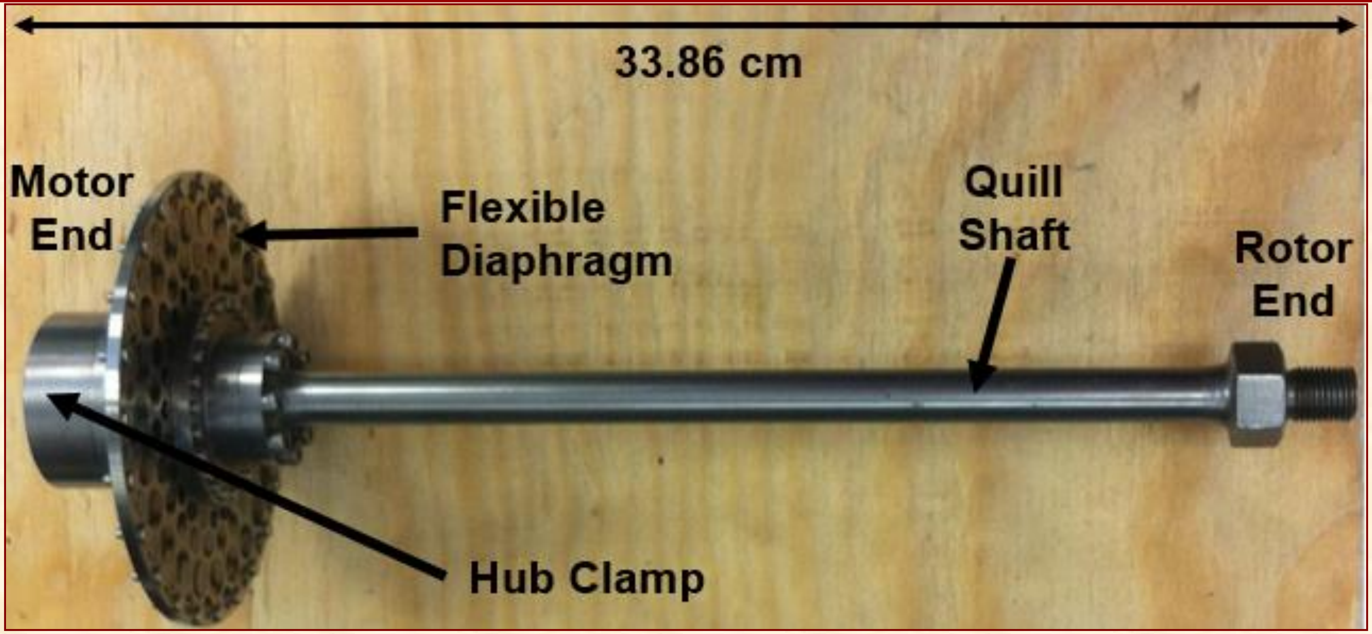
Thrust Bearing



Exploded View of Thrust Bearing Test Rig



Coupling Description



a) Quill shaft threads into rotor.

b) Hub clamp attaches to motor shaft.

c) Diaphragm isolates motor from axial loads.

d) Diaphragm and quill shaft are flexible to allow for misalignment.

Modified thrust bearing test rig

Redesign and manufacture radial bearings for operation with <u>air.</u>

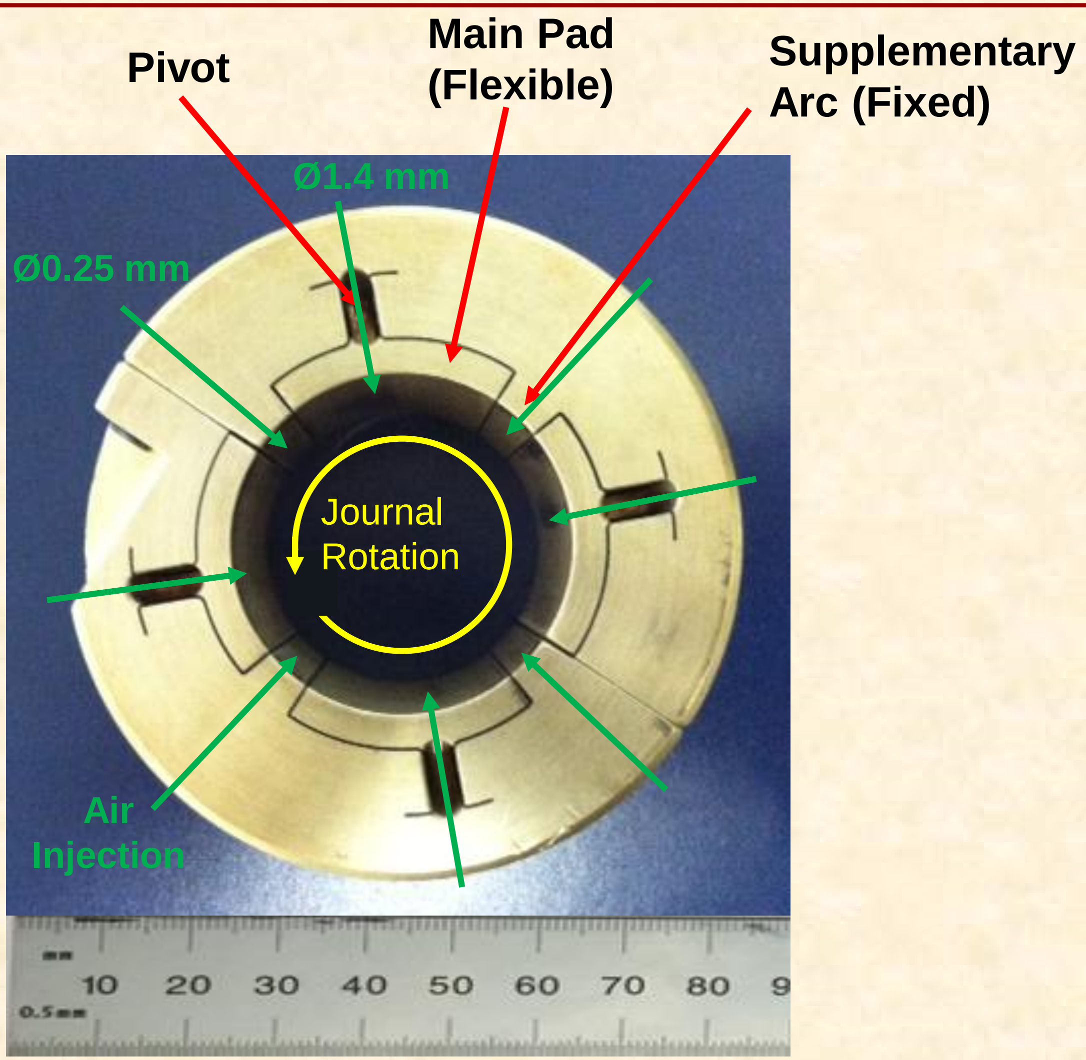
Major objective was to evaluate the dynamic forced performance of hybrid thrust bearings and thrust foil bearings (gas lubricated).

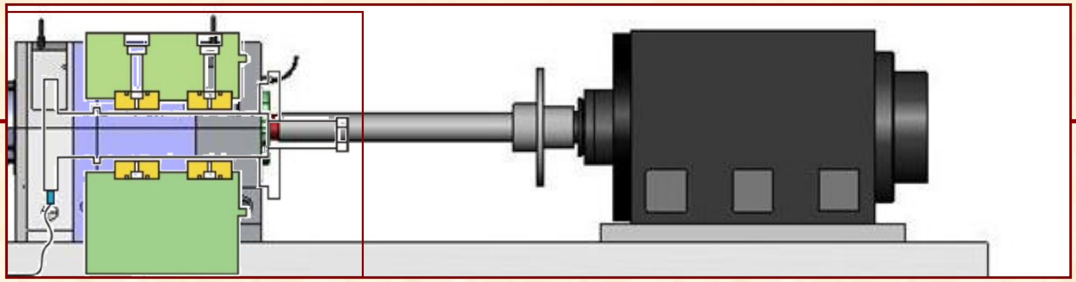
Gas Bearings

Flexure, Pivot Tilting Pad Hydrostatic Gas Bearing

330 Bearing Bronze

Inner Diameter	3.81 cm
Outer Diameter	7.62 cm
Length	3.81 cm
# Pads	4
Arc Length	72°
Pivot Offset	0.6
Radial Clearance	**76 μm**
Orifice Diameter	**1.4 mm**
Arc Length	18°
Radial Clearance	**76 μm**
Orifice Diameter	**0.25 mm**





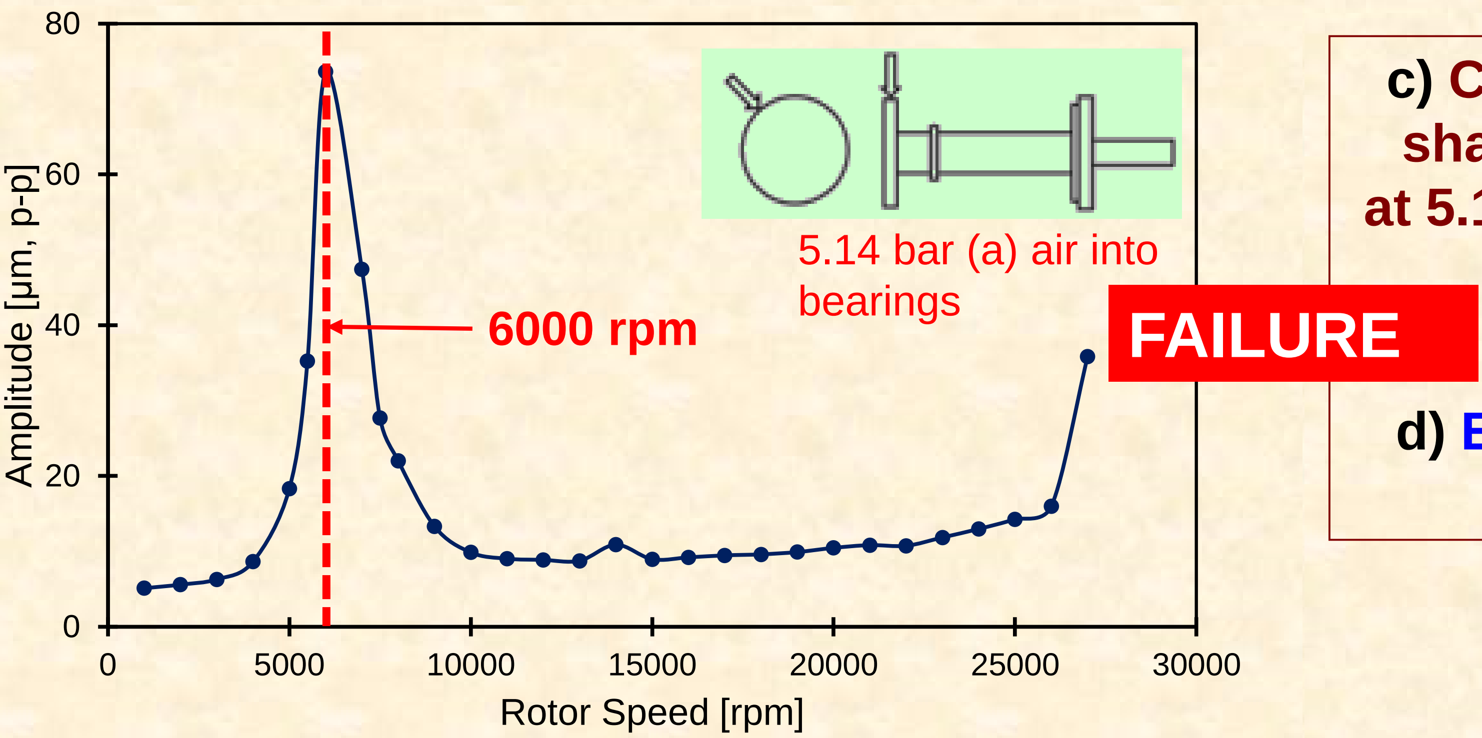
Gas lubricated thrust bearing test rig

Modifications took several months to complete.
After rotor-bearing-coupling system assembly and alignment, verification of operation without pneumatic hammer, test system was ready to operate with shaft rotational speed (thrust bearings NOT active)………

….. and the following happened on day one of test rig operation.

Maiden Operation

No active thrust bearings



c) Contact between shaft and bearings at 5.14 bar(a) and 28 krpm.

d) Emergency stop initiated.

a) <u>Operating conditions</u>

- a) Air supply pressure to bearings = 7.89, 6.52, 5.14 bar(a)
- b) Rotor speed = 0 – 25 krpm (50 m/s rotor surface speed)
- c) Supply pressure = 5.14 bar(a), rotor speed > 25 krpm

b) Natural frequency ~ 6 krpm (100 Hz), damping ratio ~ 6%

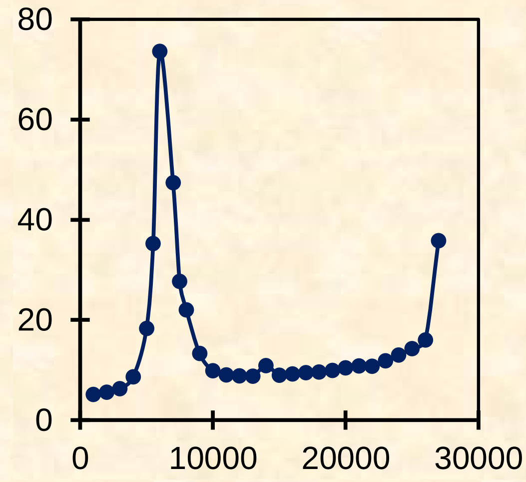
Troubleshooting of rotor response

a) Large 1X amplitude rotor motion recorded while crossing lowest critical speed (6 krpm).

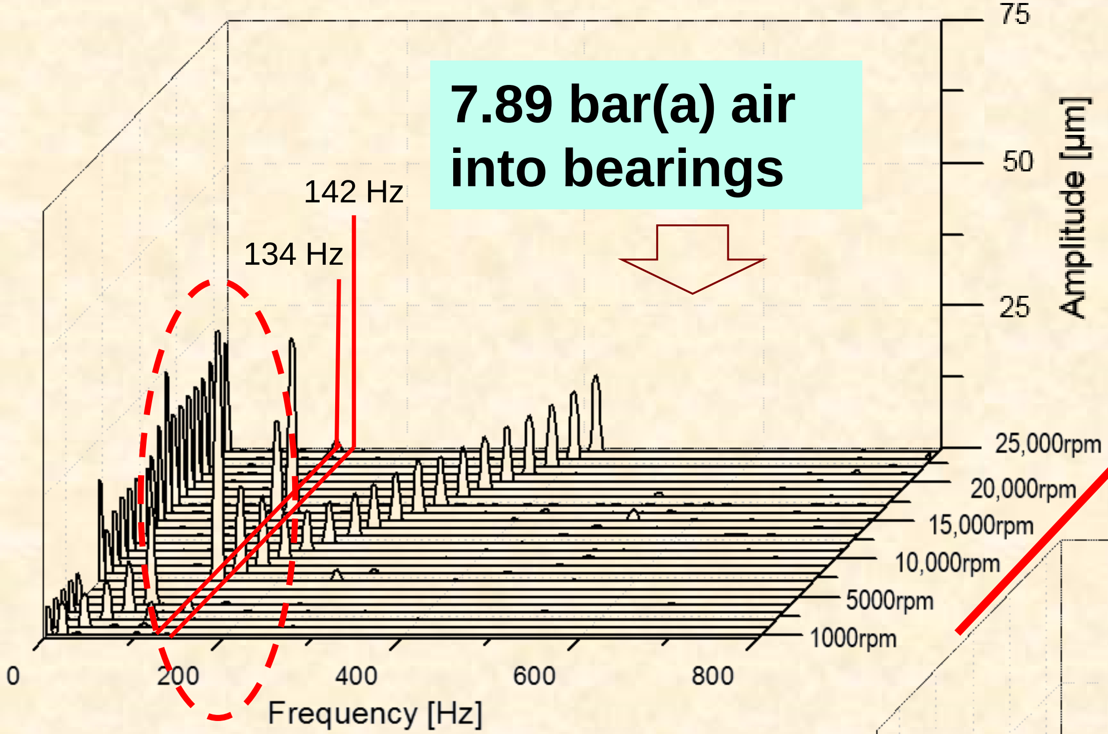
b) Above 12 krpm, operator taps rotor to excite it. Rotor motion appears at first natural frequency (~ 100 Hz) and decays → stable.

c) At high speed (~23 krpm), decay time increases to ~7 s while rotor vibrates at first natural frequency.

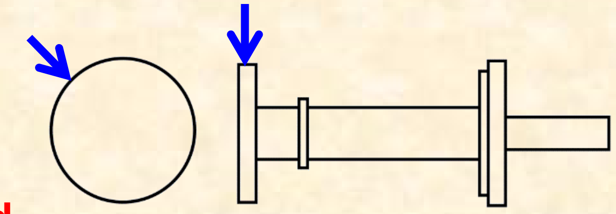
Findings: Rotor-bearing system has little damping (~3.6%) at its first natural frequency. When operating above 12 krpm, rotor is prone to show subsynchronous whirl motions at its first natural frequency.



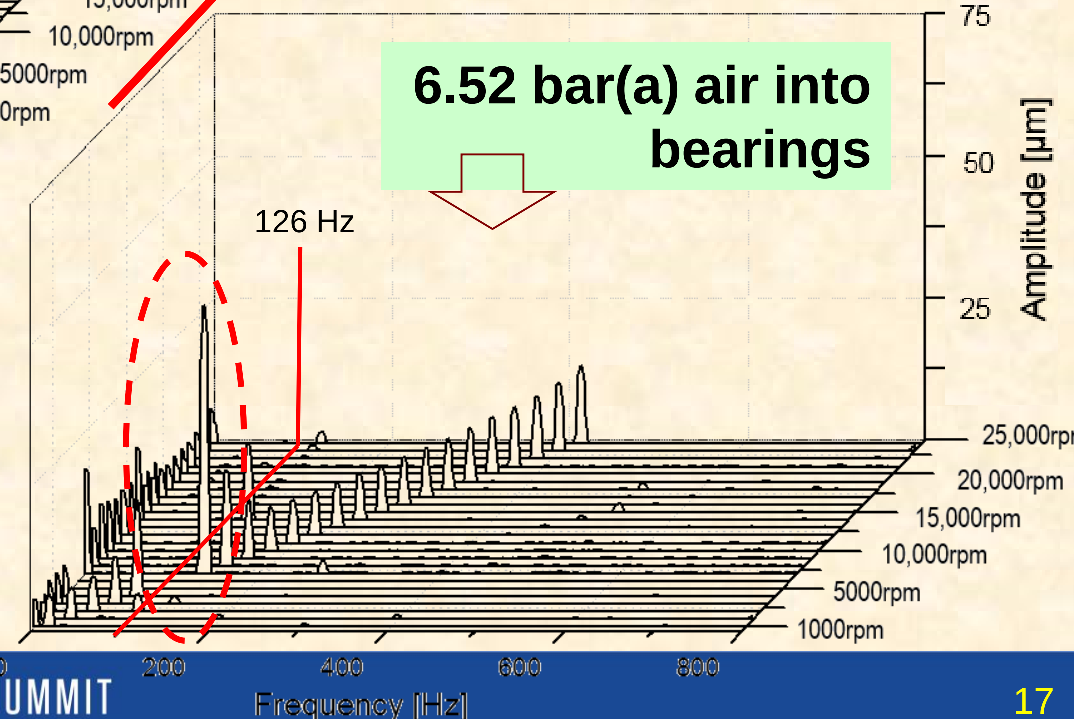
Maiden Operation — Decrease in supply pressure

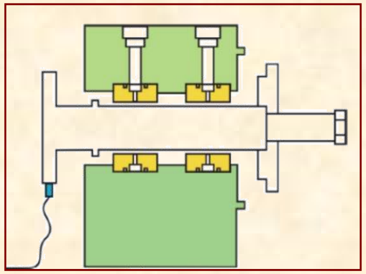


Rotor displacement recorded at collar on rotor free end:



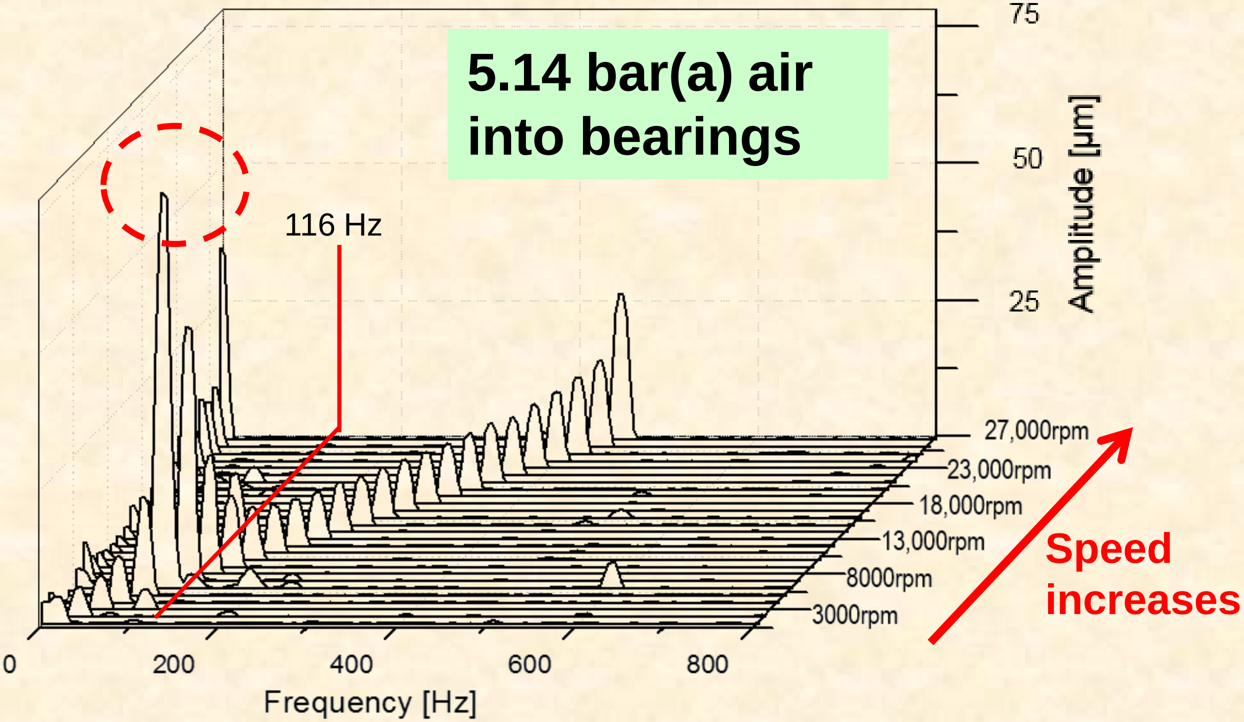
Speed increases



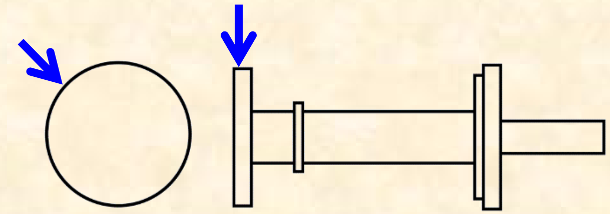


Maiden Operation

Decrease in supply pressure



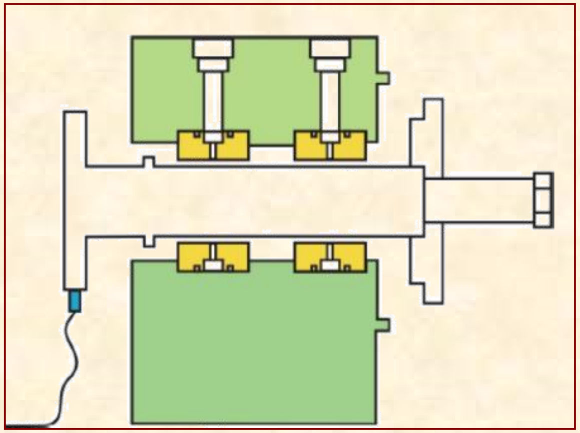
Findings: Subsynchronous rotor motion appear. As supply pressure increases, amplitude and frequency of subsynchronous vibration **(SSV)** increases.



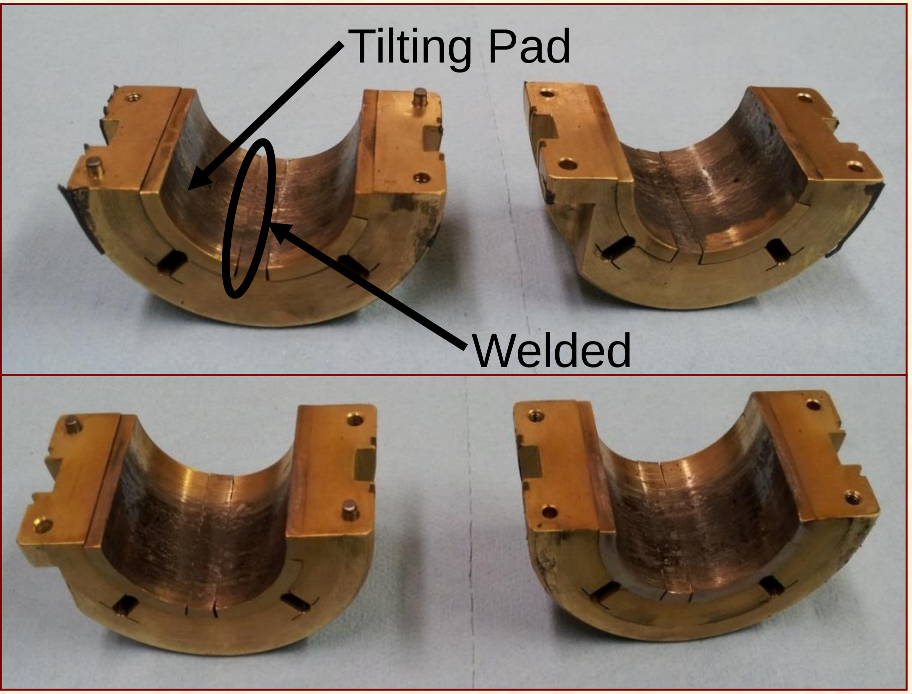
Whirl frequency ratio (WFR)=0.50

Air Supply Pressure	Rotor Speed	Frequency of SSV	Amplitude of SSV
5.14 bar	233 Hz (14 krpm)	**116 ± 2 Hz**	11 µm
6.52 bar	250 Hz (15 krpm)	**126 ± 2 Hz**	13 µm
7.89 bar	267 Hz (16 krpm)	**134 ± 2 Hz**	18 µm
7.89 bar	283 Hz (17 krpm)	**142 ± 2 Hz**	31 µm

Post-Mortem

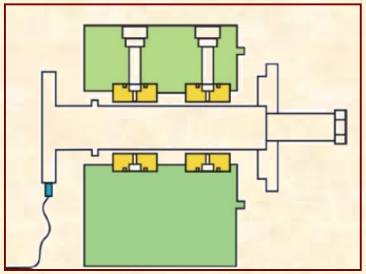


Damaged Components

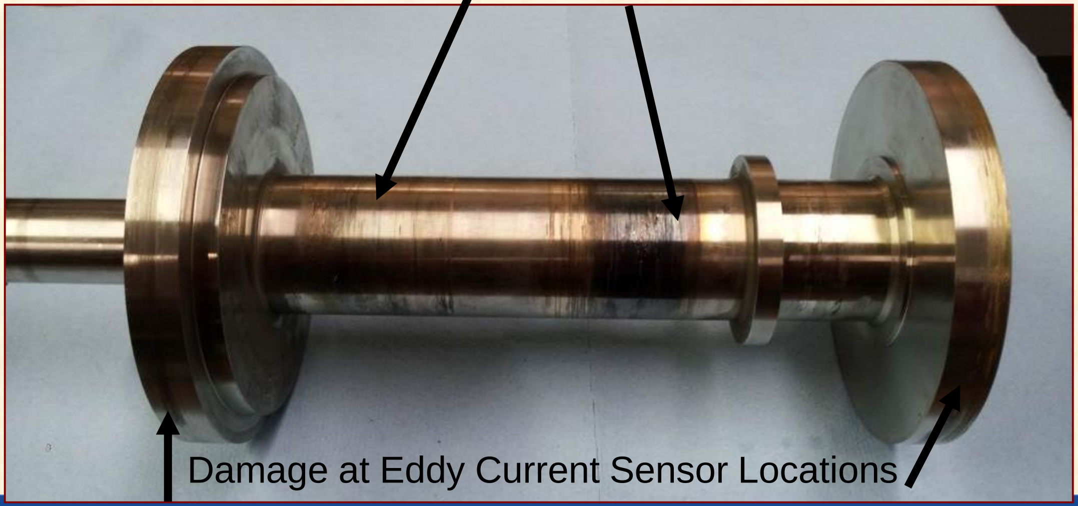


Rotor contacted and rubbed on radial bearings producing so much heat (and temperature) that welded the pads to the bearing.

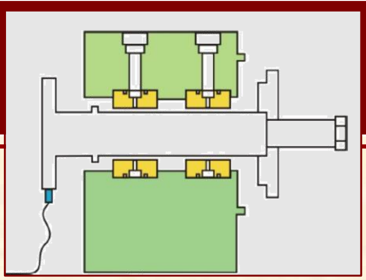
Rotor severely worn & twisted.



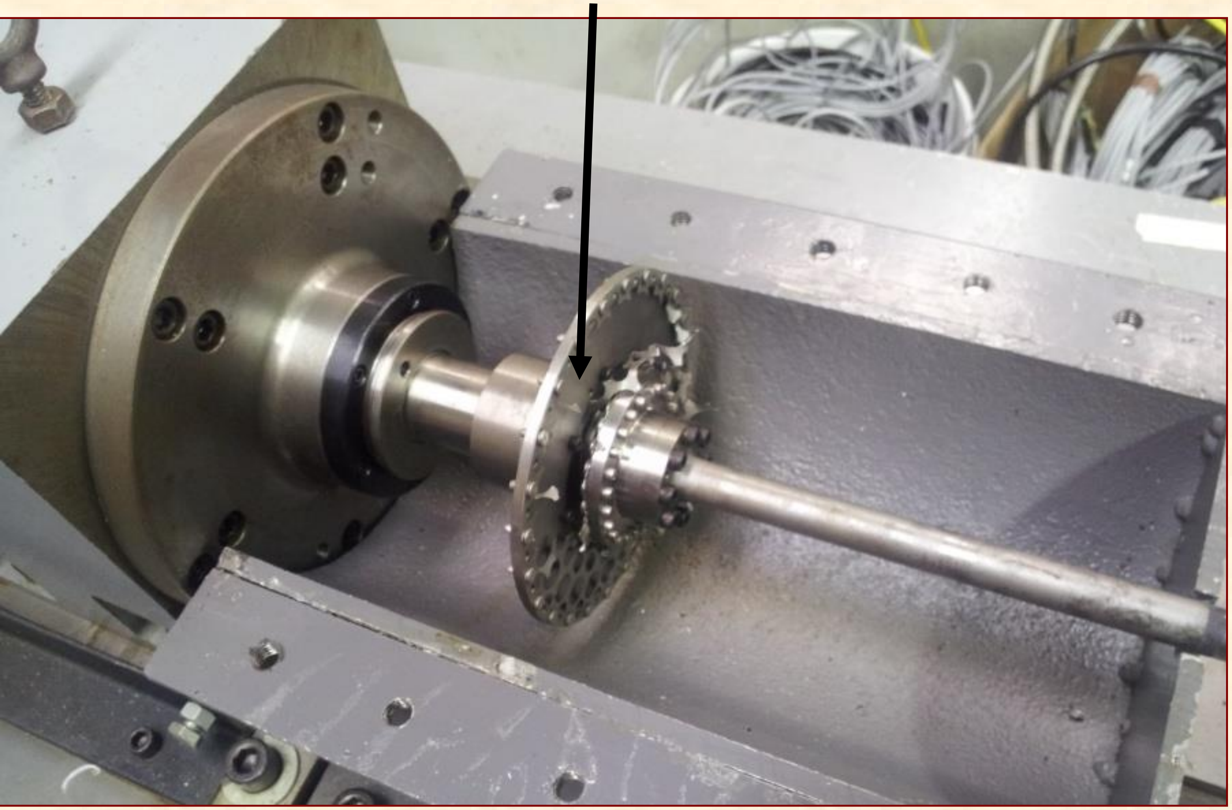
Damage at Bearing Support Locations



Damaged Components



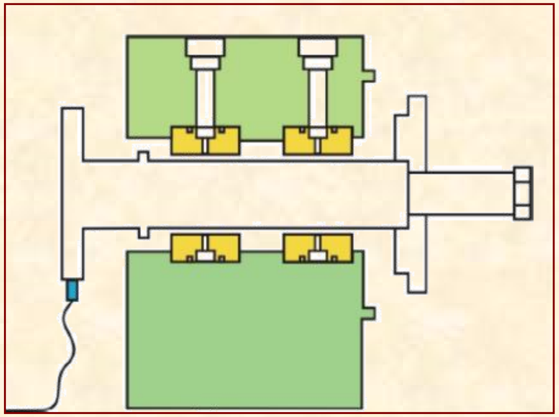
Ruptured Diaphragm



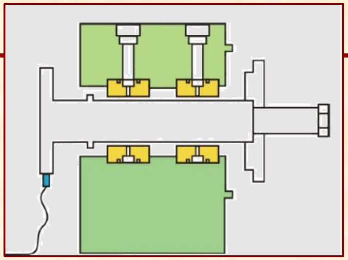
Emergency (sudden) stop placed large torque that ruptured coupling diaphragm while rotor kept spinning.

Sensors damaged, seals damaged, pins and connecting bolts sheared off.

What Caused the Failure?

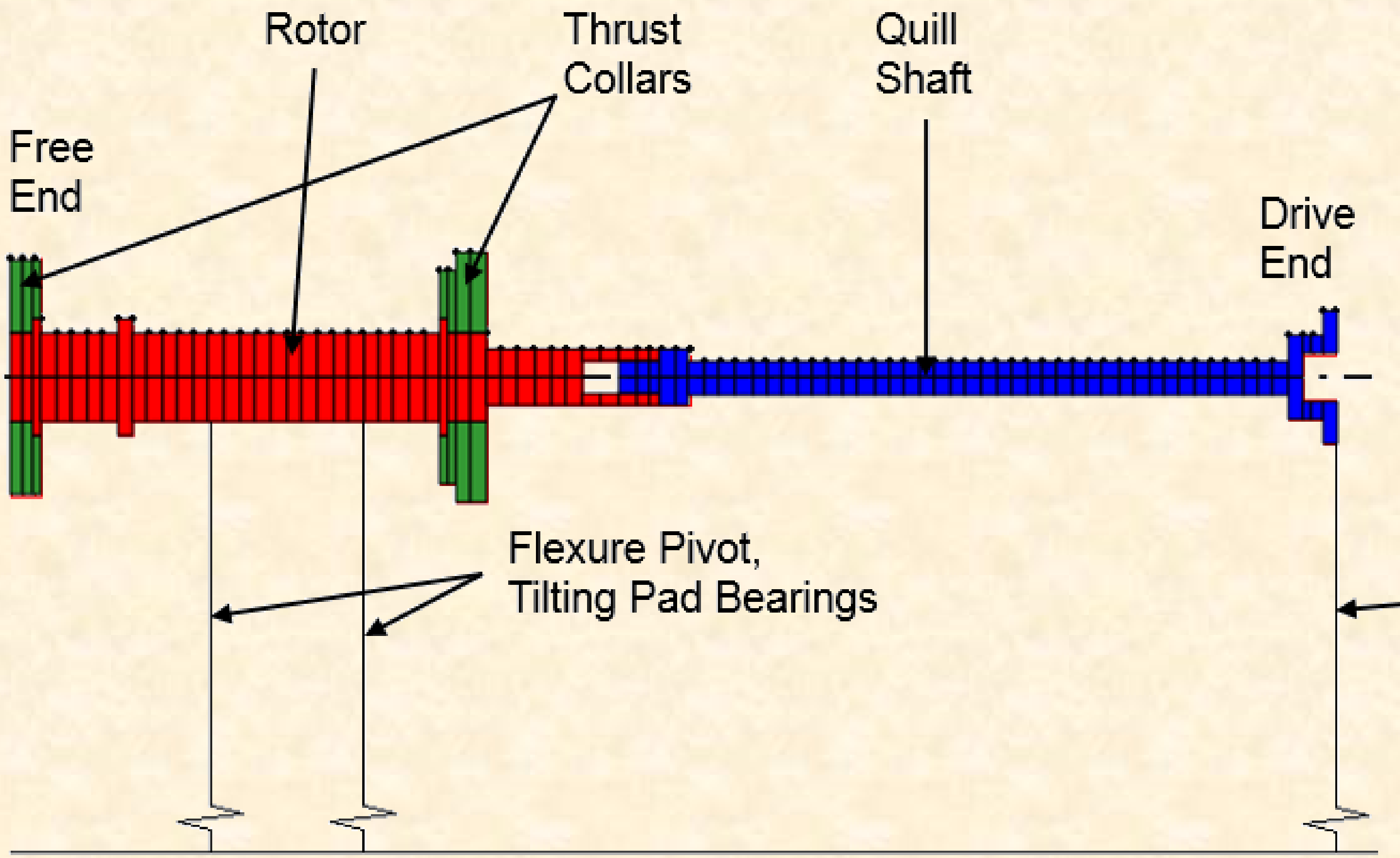


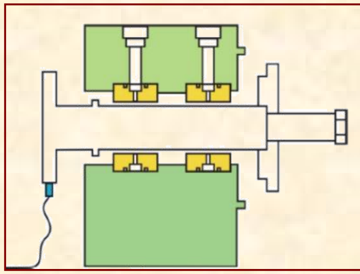
What do we need to know to explain/predict the failure?



a) Structural analysis of rotor and coupling.
b) Force coefficients of gas bearings.
c) Accurate model of rotor-bearing system.
d) System natural frequencies and damping ratios.
e) System imbalance response.

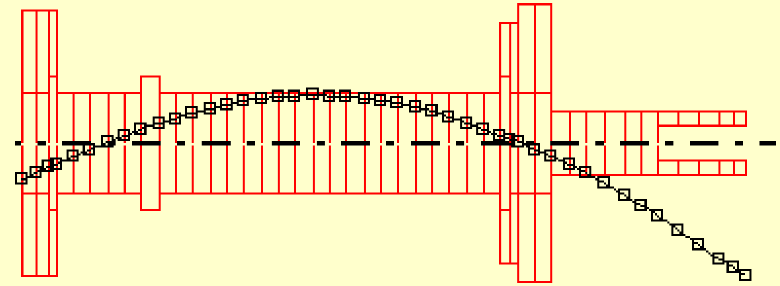
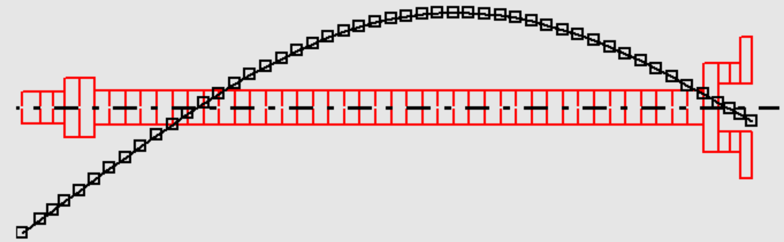
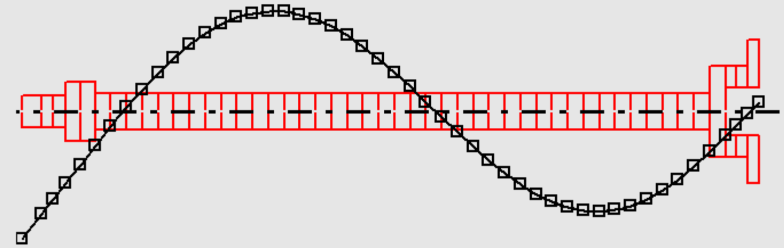
Rotor-Coupling-Bearing System Model





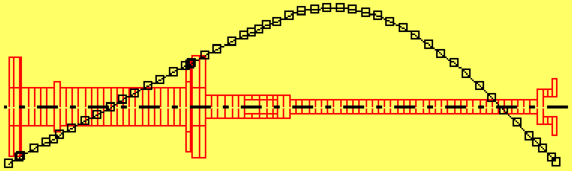
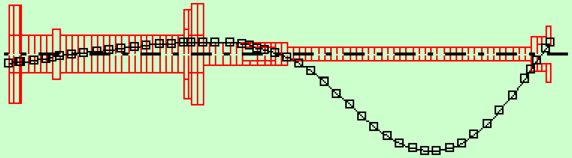
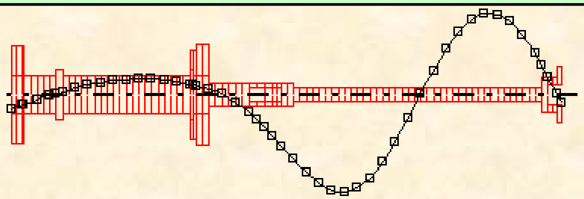
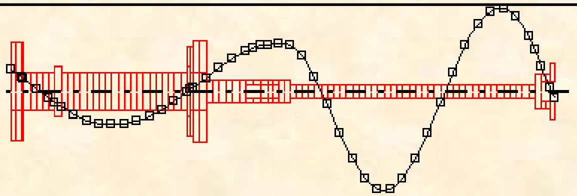
Question: Is coupling dynamics "decoupled" from rotor dynamics?

Rotor and Coupling Natural Frequencies

	Measurement	Prediction	Mode Shape
Rotor	**1,760 ± 8 Hz**	**1,817 Hz**	
Coupling	**496 ± 8**	**491 Hz**	
	1,504 ± 8	**1,535 Hz**	

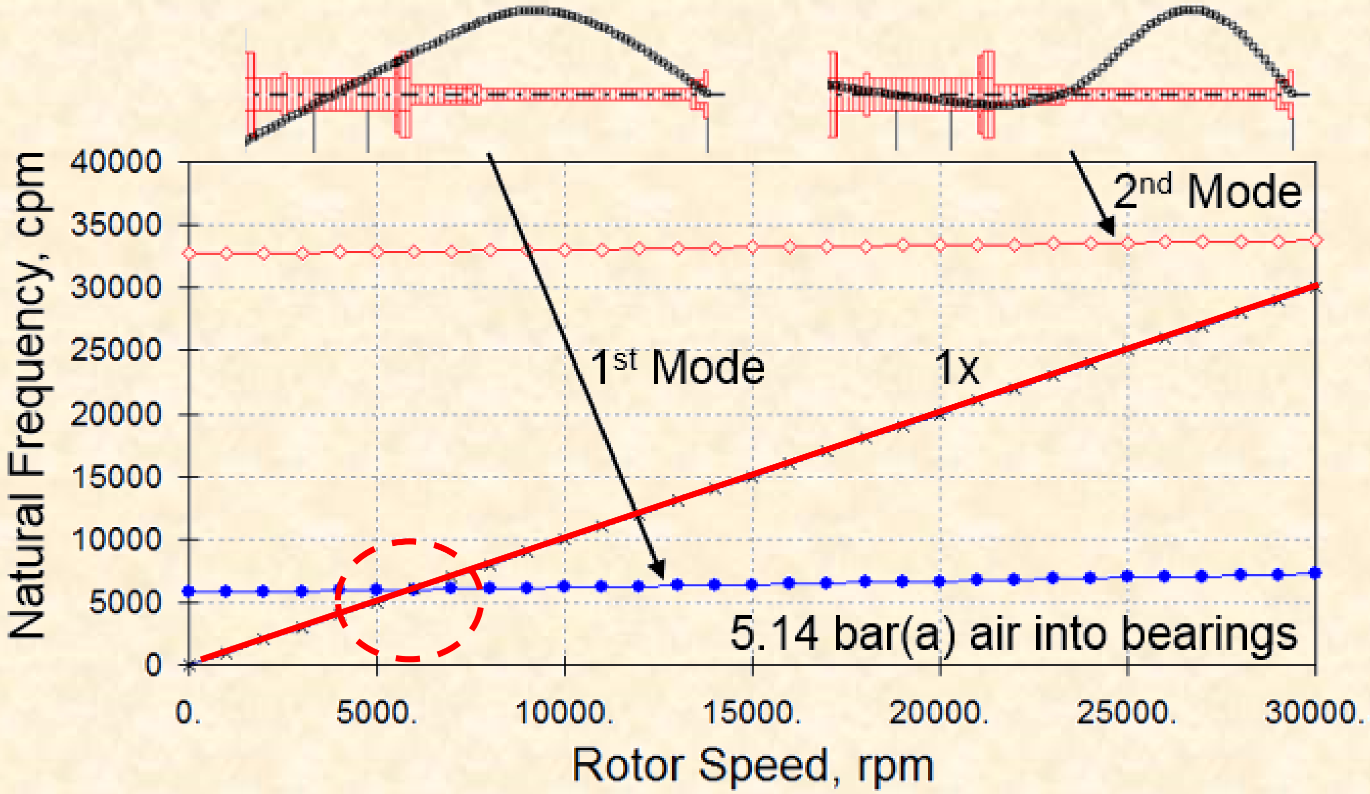
Finding: Structural models for both rotor and coupling predict well their <u>free-free mode</u> natural frequencies.

Free-free modes of rotor & coupling

Measurement [Hz]	Prediction [Hz]	**Free-Free Mode Shape**
104 ± 8	101	
552 ± 8	560	
1,272 ± 8	1,332	
1,944 ± 8	2,029	

Findings: **Modes show quill shaft is <u>too</u> flexible. For operation below 30 krpm, both quill shaft and rotor operate as a single unit.**

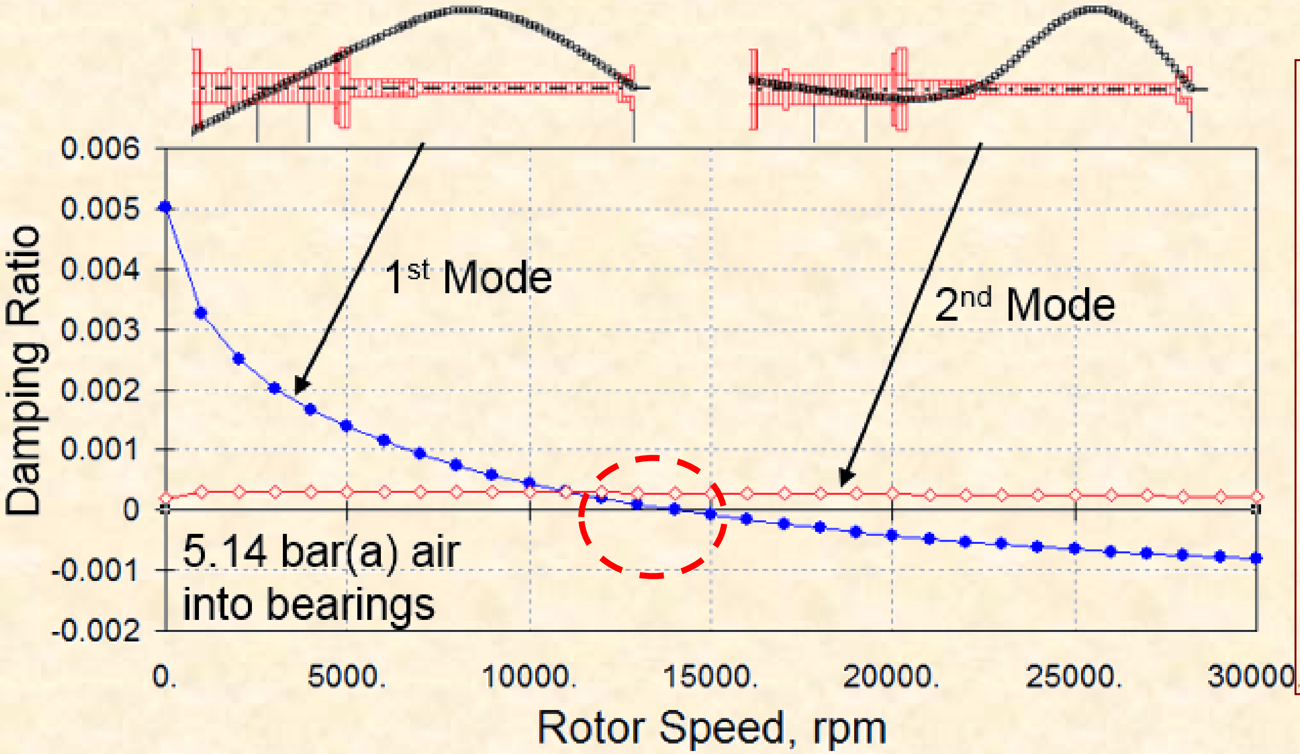
System natural frequencies & 1st critical speed



Predicted critical speed agrees with measured one at 6 krpm [100 Hz].

Findings: Gas bearing stiffness does not affect natural frequencies and critical speed (6 krpm [100 Hz]) as flexibility of quill shaft determines its location. Bearing clearance is too large for adequate stiffness.

System Damping Ratio



Predicted instability at 14 krpm with whirl frequency ratio (WFR)= 0.42.

As supply pressure into bearings increases, threshold speed of instability also increases.

Findings: **Damping ratio is very low because most motion is at quill shaft (no damping). At natural frequency, predicted damping ratio (ζ = 0.001) is lower than estimated damping ratio (ζ = 0.04).**

Nat freq. & damping ratio vs supply pressure

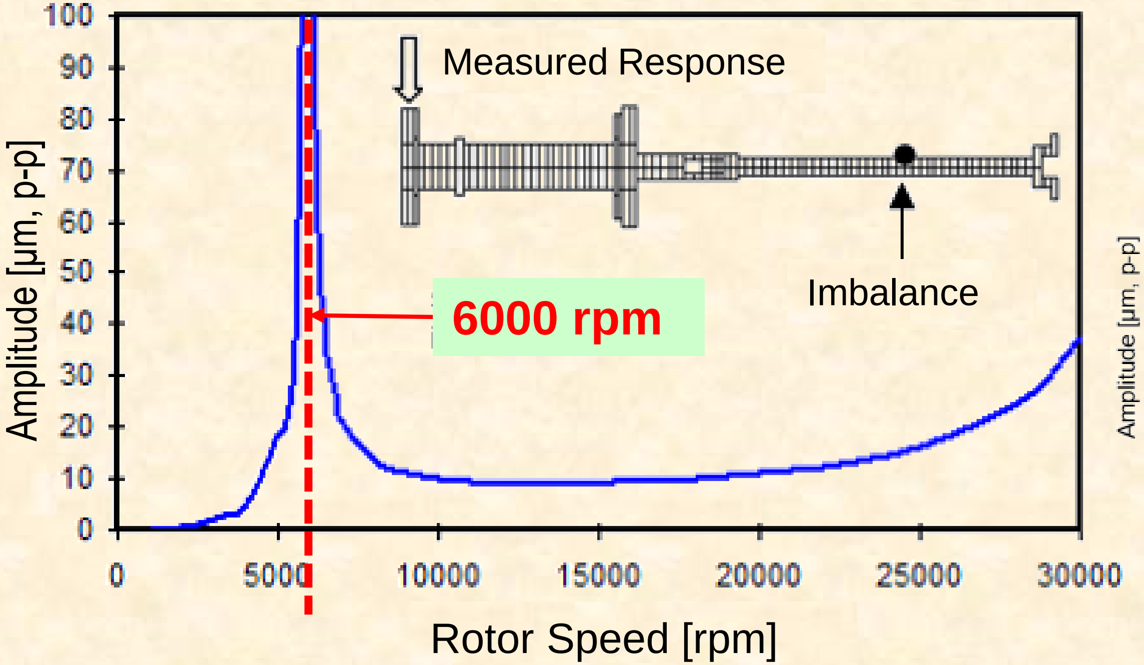
Supply pressure into bearings	5.14 bar	6.52 bar	7.89 bar
1^{st} Natural Frequency	**99 Hz**	**100 Hz**	**101 Hz**
Damping Ratio, ζ	**0.0011**	**0.0014**	**0.0019**
Threshold Speed of Instability	**14 krpm**	**16 krpm**	**21 krpm**
Whirl Frequency Ratio	**0.42**	**0.38**	**0.29**
2^{nd} Natural Frequency	563 Hz	563 Hz	563 Hz
Damping Ratio, ζ	0.0002	0.0001	0.0001

Predictions: Quill shaft dominates rotordynamics: first and second natural frequencies remain constant as supply pressure increases.

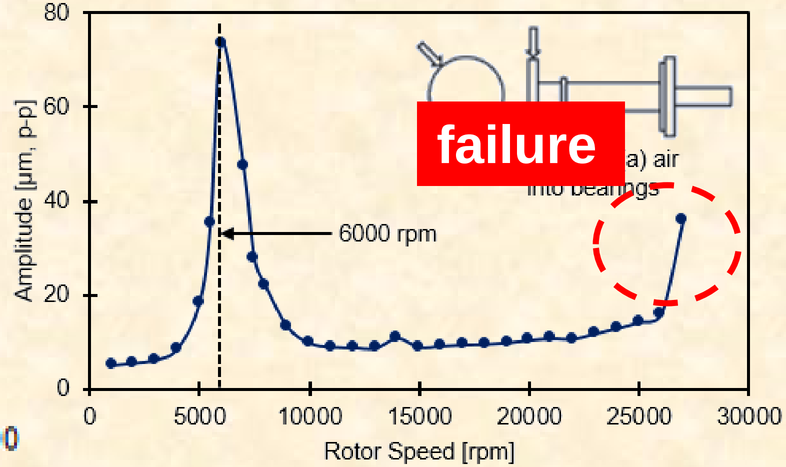
Damping ratio is very low and threshold speed of instability increases with supply pressure.

Response from Imbalance

Prediction vs data



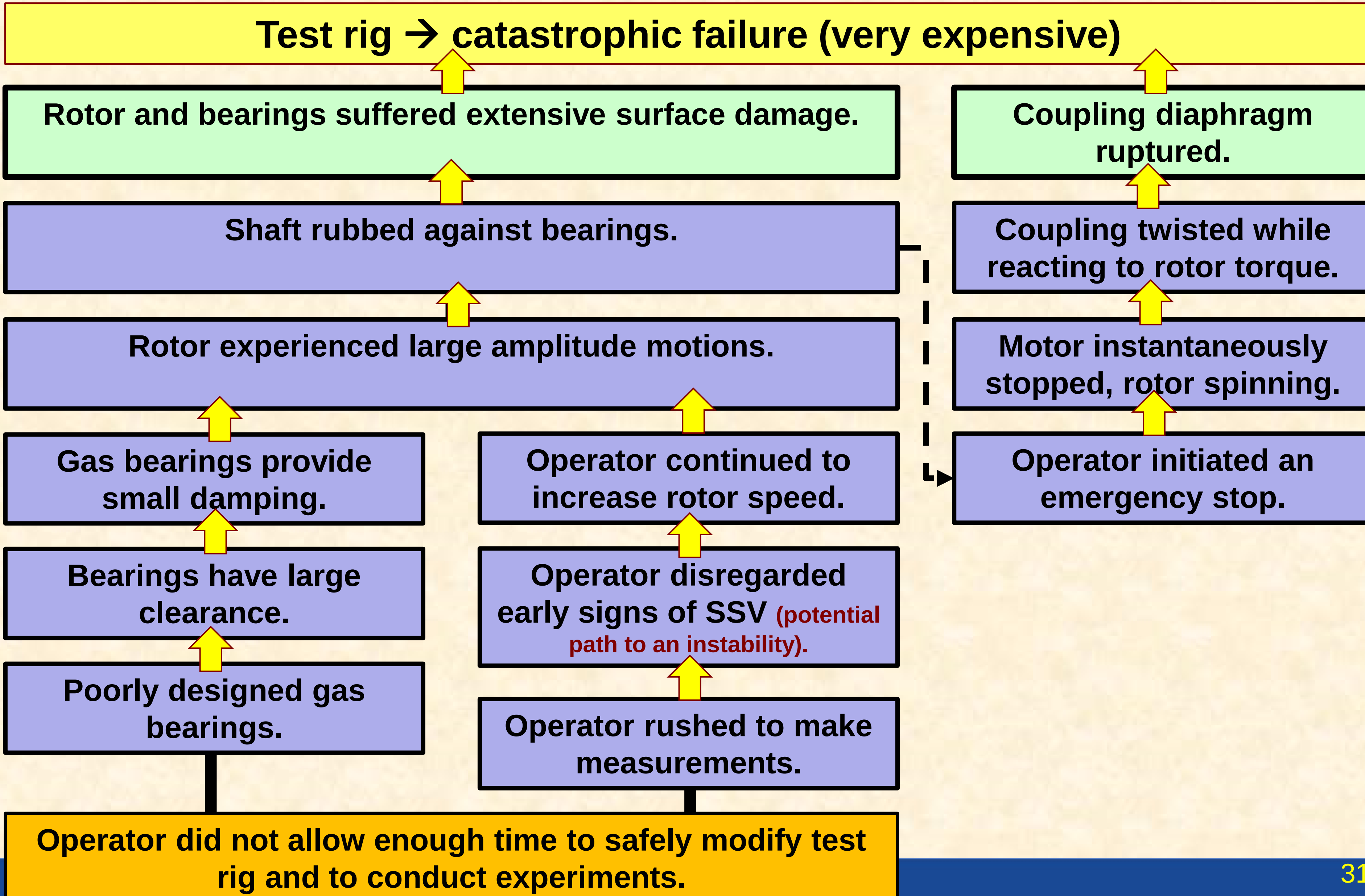
5.14 bar(a) air into bearings



Measured

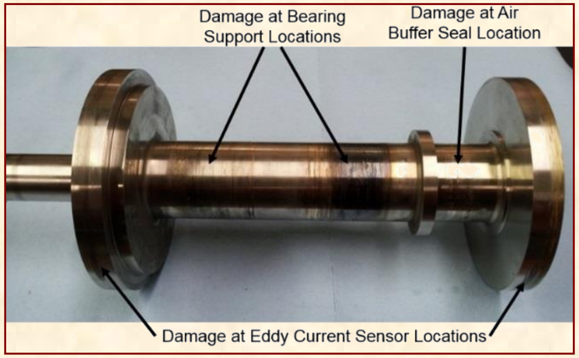
Findings: **Predicted 1X response is similar to measured response. Operation > 25 krpm produces an increasing amplitude of motion as system approaches its second natural frequency (34 krpm [563 Hz]).**

Anatomy of the Failure



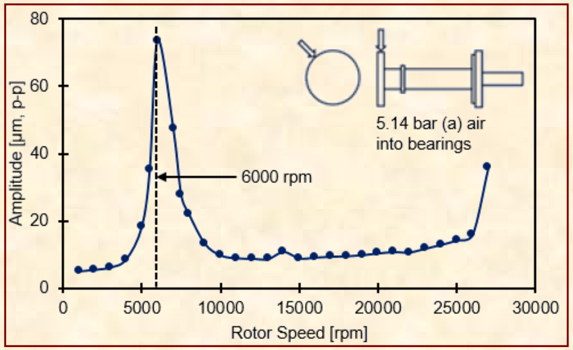
Conclusion & lesson learned

Modified test rig to operate with <u>air lubricated</u> hybrid journal bearings and thrust bearings.



- **(a) During maiden operation: sudden contact & rubbing between the shaft and its bearings led to catastrophic failure of the test rig.**

- **(b) Large amplitude rotor motions are a result of hydrodynamic instability from gas bearings plus too+ flexibility of quill shaft.**



Incident could have been avoided had operators not ignored early signs of an instability (very little damping from air bearings).

Acknowledgments

Thanks to TAMU Turbomachinery Laboratory & Turbomachinery Research Consortium.

Questions:

Learn more at http://rotorlab.tamu.edu