Notes 0

INTRODUCTION TO FLUID FILM BEARINGS AND SEALS

A turbomachinery is a rotating structure where the load and/or the driver handle a process fluid from which power is extracted or delivered to. Examples of turbomachines include pumps and compressors, gas and steam turbines, turbo generators and turbo expanders, turbochargers, APU (auxiliary power units), etc.

Most turbomachinery is supported on oil lubricated fluid film bearings, although modern advances and environmental restrictions are pushing towards the implementation of process fluid bearings and even gas bearing applications. Fluid film bearings are used due to their adequate load support, good damping characteristics and absence of wear if properly designed and operated.

Turbomachines also include a number of other mechanical elements which provide stiffness and damping characteristics and affect the dynamics of the rotor-bearing system. Impeller seals, floating ring seals, thrust collars and balance pistons are a few of these elements.

The adequate operation of a turbomachine is defined by its ability to tolerate normal (and even abnormal) vibrations levels without affecting significantly its overall performance (reliability and efficiency).

The rotordynamics of turbomachinery encompasses the structural analysis of rotors (shafts and disks) and the design of fluid film bearings and seals that determine the best dynamic performance given the required operating conditions. This best performance is denoted by well-characterized natural frequencies (and critical speeds) with amplitudes of synchronous dynamic response within required standards and demonstrated absence of subsynchronous vibration instabilities.

A rotordynamic analysis considers the interaction between the elastic and inertia properties of the rotor and the mechanical impedances from the fluid film bearing supports, oil seal rings, seals, etc.

The most commonly recurring problems in rotordynamics are

1. Excessive steady state synchronous vibration levels.
2. Subharmonic rotor instabilities.

Steady state vibration levels may be reduced by:

a) Improving balancing.
b) Modifying rotor-bearing systems: tune system critical speeds out of RPM operating range.
c) Introducing damping to limit peak amplitudes at critical speeds that must be traversed.

Subharmonic rotor instabilities may be avoided by:

a) Raising the natural frequency of rotor system as much as possible.
b) Eliminating the instability mechanism, i.e. change bearing design if oil whip is present.
c) Introducing damping to raise onset speed above the operating speed range.

Rotordynamic instabilities have become more and more common as the speed and horsepower of turbomachinery have increased. These instabilities can sometimes be erratic, seemingly increasing vibration amplitudes for no apparent reason. A common denominator among many stability problems is that they tend to grow with time as the affected component(s) begins to wear or fatigue.

For example, two typical destabilizing forces well documented in the technical literature are due to the aerodynamic effects of labyrinth seals and the hydrodynamic effects of cylindrical bearings and floating oil ring seals in centrifugal compressors. Load, gas molecular weight, and oil pressure and temperature appear to be among the factors bringing severe problems in problematic turbomachinery.

The detailed study of rotordynamics demands accurate knowledge of the specific mechanical elements that support the rotor, i.e. fluid film bearings and seals.

Fluid film bearings

Fluid film bearings are machine elements designed to produce smooth (low friction) motion between solid surfaces in relative motion and to generate a load support for mechanical components. The lubricant or fluid between the surfaces may be a liquid, a gas or even a solid.

Fluid film bearings, when well designed and operated, are able to support static and dynamic loads, and consequently, their effects on the performance of rotating machinery are of great importance.

Our study will concentrate on the analysis of bearings with a full film separating the mechanical surfaces. The word film implies that the fluid thickness (gap or clearance) separating the surfaces is several orders of magnitude smaller than the other dimensions of the bearing, i.e. width and length.

The basic operational principles of fluid film bearings are hydrodynamic, hydrostatic or hybrid (a combination of the former two).

Hydrodynamic fluid film bearings or self-acting bearings

In these bearings, see Figure 1, there is relative motion between two mechanical surfaces with a particular wedge shape. The fluid is dragged into the film and hydrodynamic pressures are generated and able to support an externally applied load.

Hydrostatic fluid film bearings or externally-presurized bearings

In these bearings, see Figure 2, an external source of pressurized fluid forces the lubricant or fluid between the surfaces, thus providing their separation and the ability to support a load without surface contact.

Hydrodynamic or self-acting fluid film bearings

Advantages	Disadvantages
Do not require external source of pressure. Fluid flow is dragged into the convergent gap in the direction of the surface relative motion. Support heavy loads. The load support is a function of the lubricant viscosity, surface speed, surface area, film thickness and geometry of the bearing. Long life (infinite in theory) without wear of surfaces. Provide stiffness and damping coefficients of large magnitude.	Thermal effects affect performance if film thickness is too small or available flow rate is too low. Require of surface relative motion to generate load support. Induce large drag torque (power losses) and potential surface damage at start-up (before lift-off) and touch down. Potential to induce hydrodynamic instability, i.e. loss of effective damping for operation well above critical speed of rotor-bearing system.

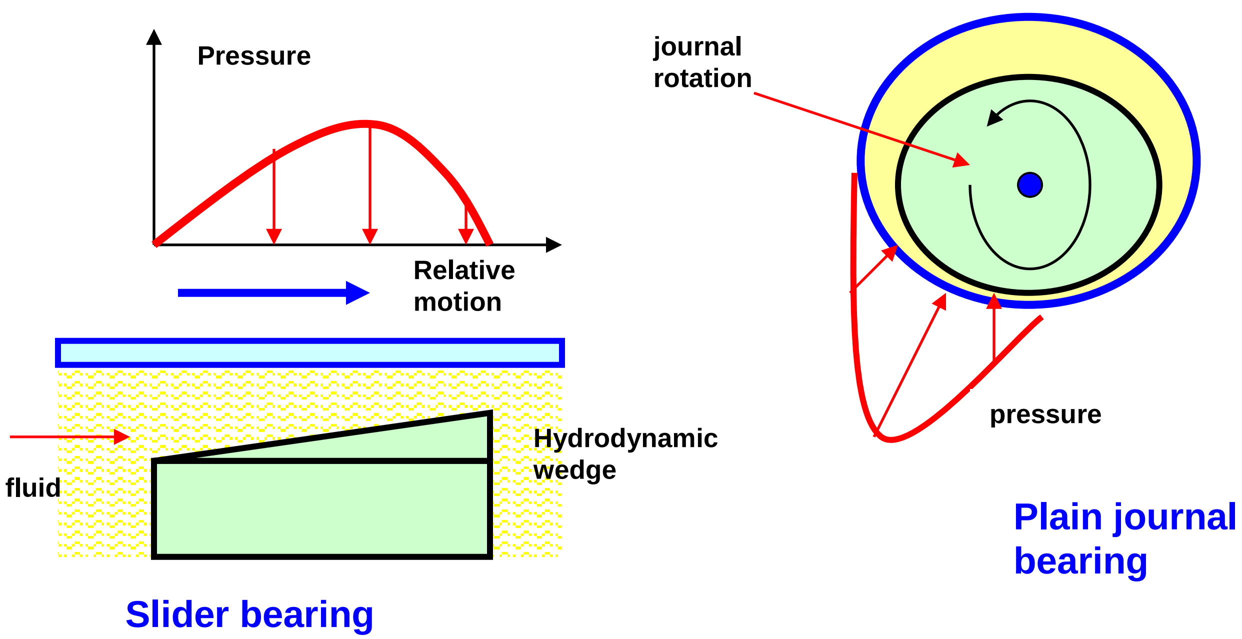


Figure 1. Examples of hydrodynamic (self-acting) fluid film bearings

Hydrostatic or externally pressurized fluid film bearings

Advantages	Disadvantages
Support very large loads. The load support is a function of the pressure drop across the bearing and the area of fluid pressure action. Load does not depend on film thickness nor lubricant viscosity. Long life (infinite in theory) without wear of surfaces Provide stiffness and damping coefficients of very large magnitude. Excellent for exact positioning and control.	Require ancillary equipment. Larger installation and maintenance costs. Need of fluid filtration equipment. Loss of performance with fluid contamination. High power consumption because of pumping losses. Potential to induce hydrodynamic instability in hybrid mode operation. Potential to show pneumatic hammer instability for highly compressible fluids, i.e. loss of damping at low and high frequencies of operation due to compliance and time lag of trapped fluid volumes.

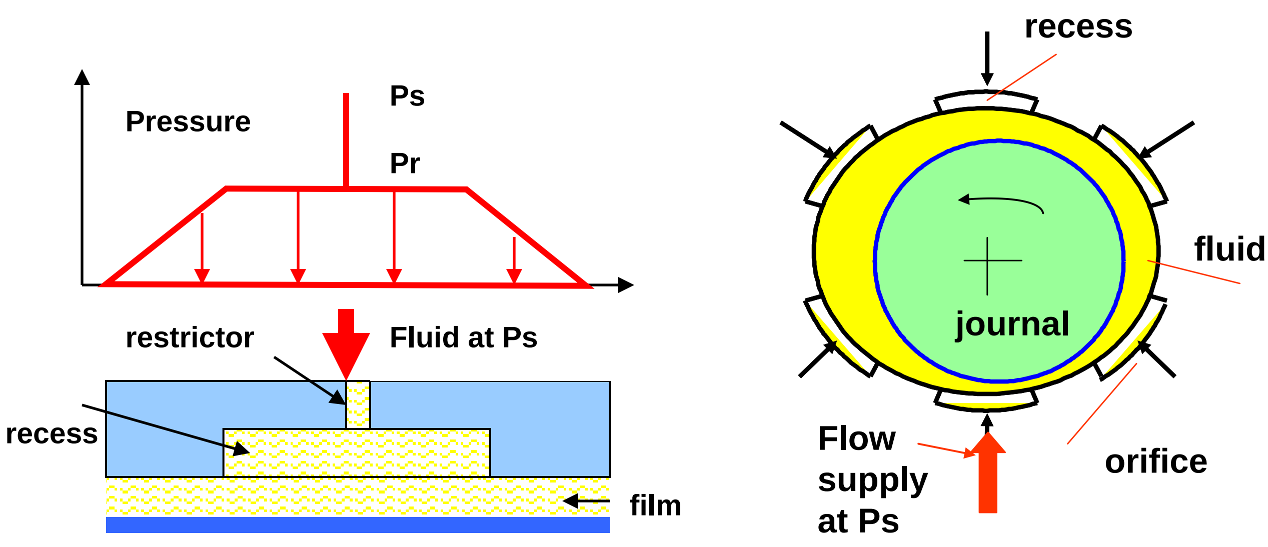


Figure 2. Examples of hydrostatic (externally pressurized) fluid film bearings

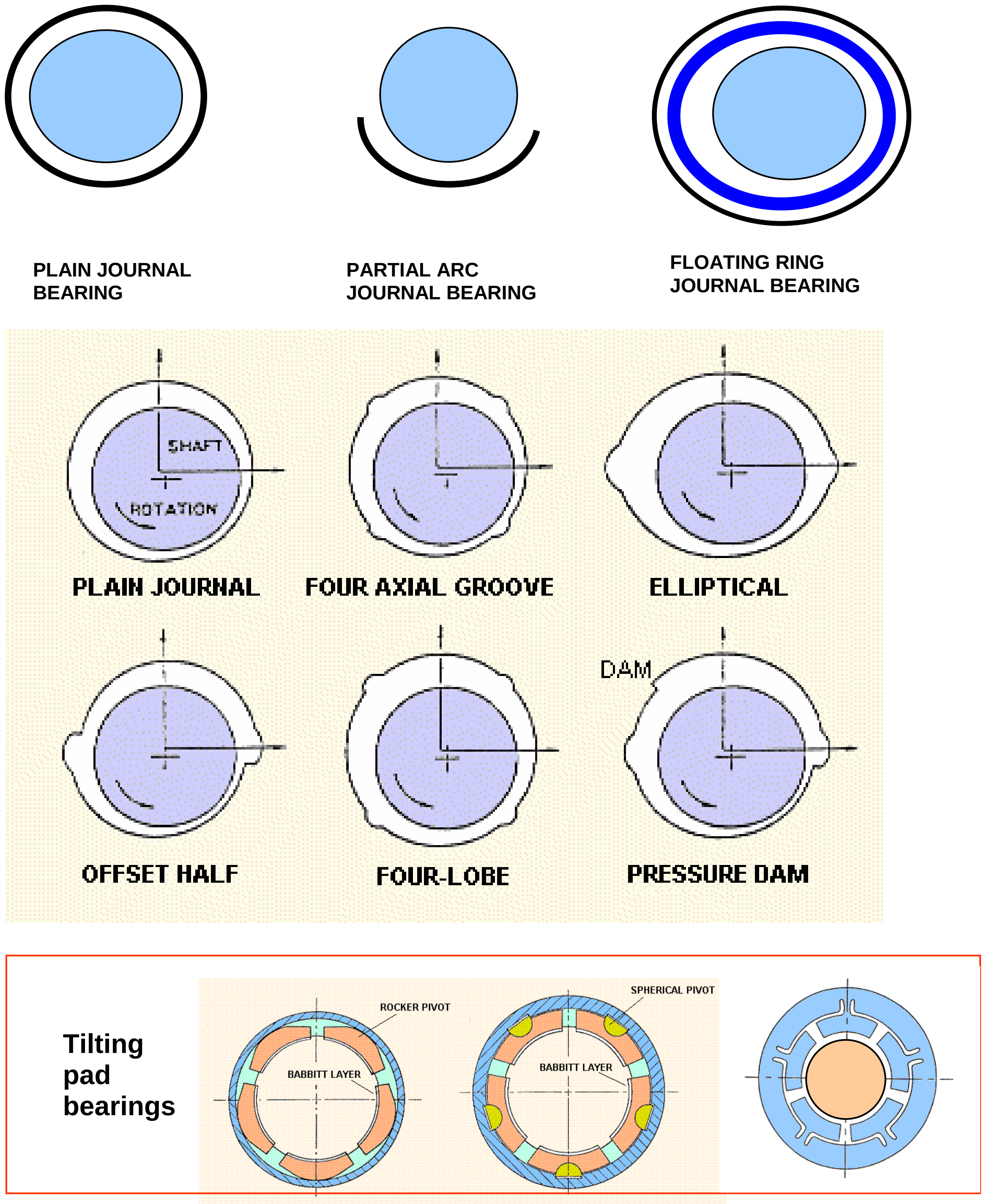


Figure 3. Typical configurations of cylindrical journal bearings

Squeeze film dampers

Oscillatory or periodic motions can also generate hydrodynamic pressures in the thin film separating two surfaces. This squeeze film action mechanism works effectively only for compressive loads, i.e. those forcing the approach of one surface to the other. **Squeeze film dampers** are routinely implemented to reduce vibration amplitudes and isolate structural components in gas jet engines, high performance compressors, and occasionally in water pumps.

A squeeze film damper consists of an inner non rotating journal and a stationary outer bearing, both of nearly identical diameters. Figure 4 shows an idealized schematic of this type of fluid film bearing. A journal is mounted on the external race of a rolling element bearing and prevented from spinning with loose pins or a squirrel cage that provides a centering elastic mechanism. The annular thin film, typically less than 0.250 mm, between the journal and housing is filled with a lubricant provided as a splash from the rolling bearing elements lubrication system or by a dedicated pressurized delivery. In operation, as the journal moves due to dynamic forces acting on the system, the fluid is displaced to accommodate these motions. As a result, hydrodynamic squeeze film pressures exert reaction forces on the journal and provide for a mechanism to attenuate transmitted forces and to reduce the rotor amplitude of motion.

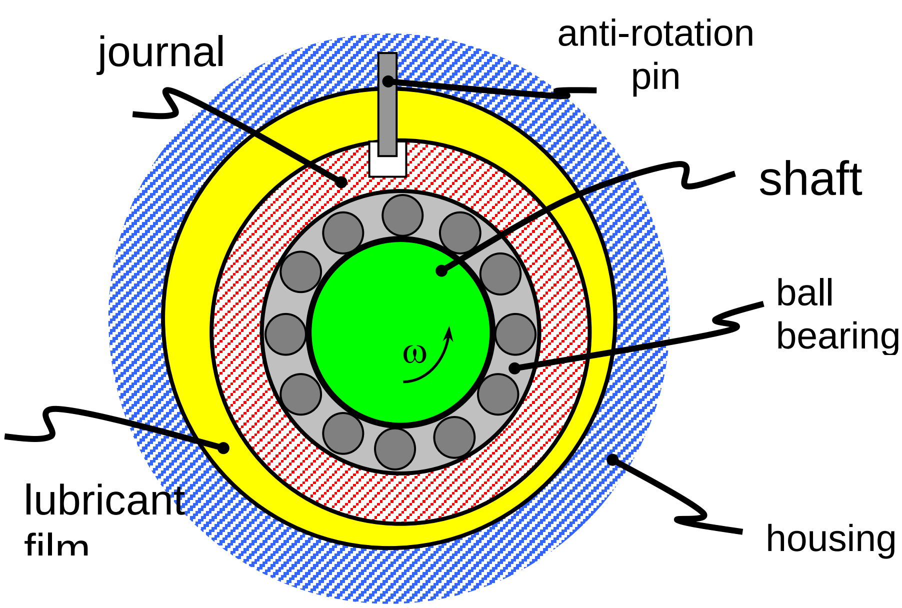


Figure 4. Typical squeeze film damper configuration

Radial seals (annular, labyrinth or honeycomb) separate regions of high pressure and low pressure in rotating machinery and their function is to minimize the leakage and improve the overall efficiency of a rotating machine extracting or delivering power to a fluid. Typical applications include neck ring seals on impeller eyes and interstage seals as well as balance pistons in pump and compressor applications. See Figure 5 for a depiction of these mechanical elements in a typical pump.

Seals have larger clearances than load carrying fluid film bearings. Yet their impact on the rotordynamics of turbomachinery is of importance since seals are located at rotor locations where large vibrations (rotor elastic deflections) occur, as shown in Figure 6.

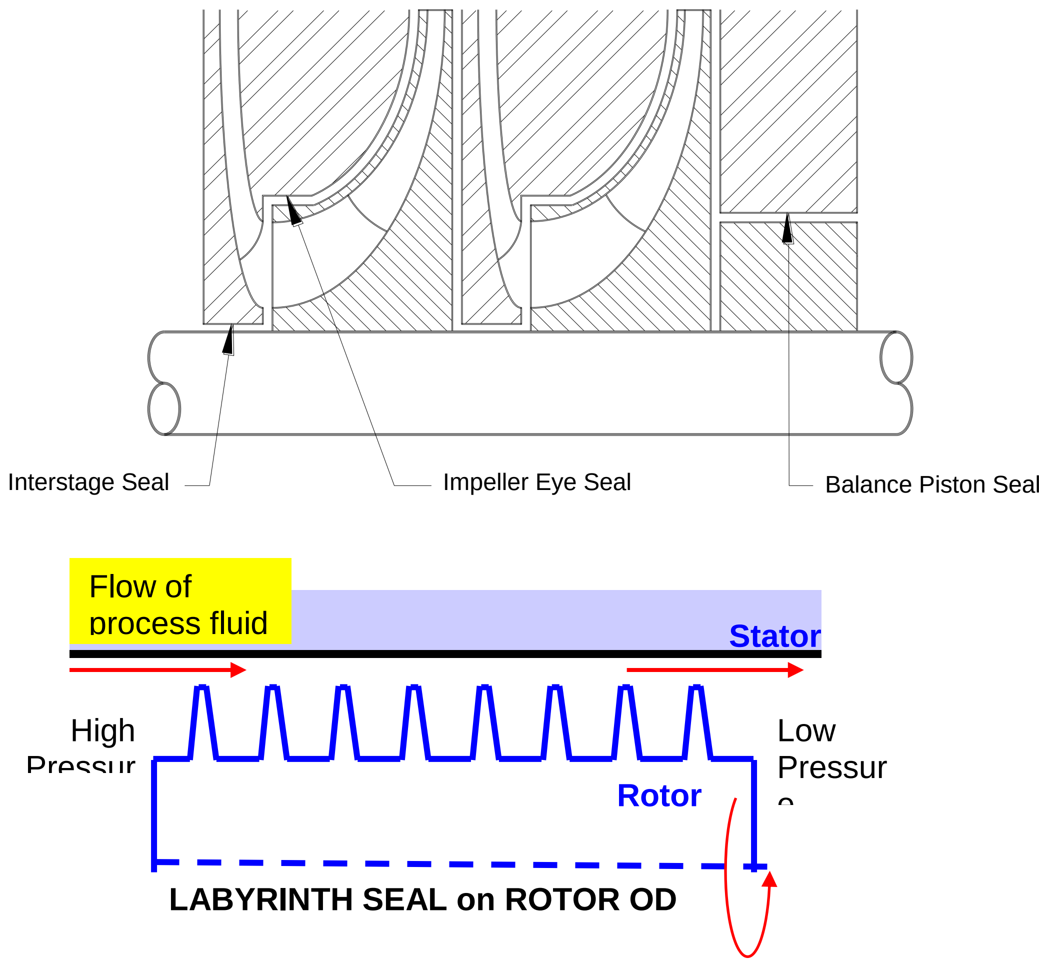


Figure 5. Seals in a Multistage Centrifugal Pump or Compressor

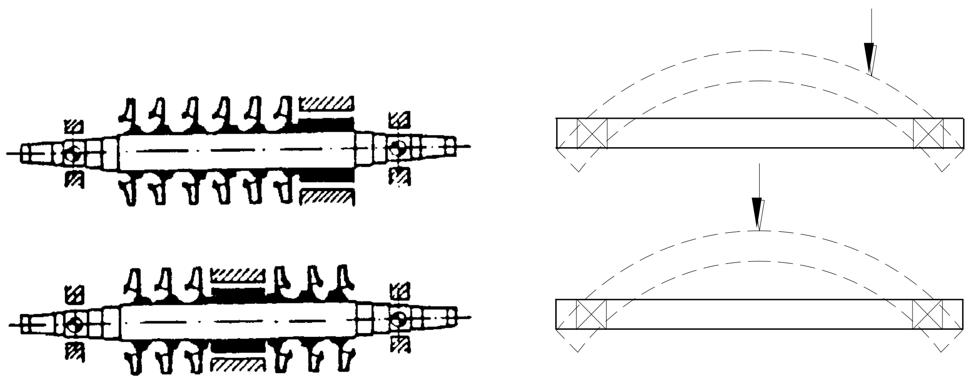
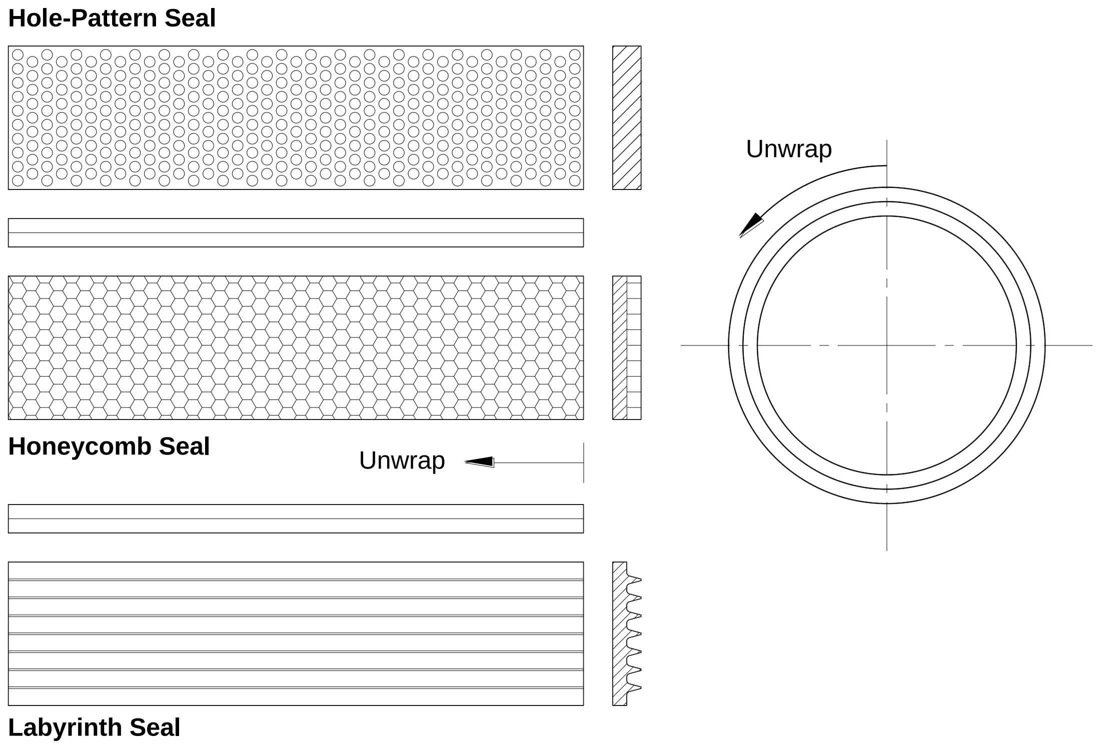
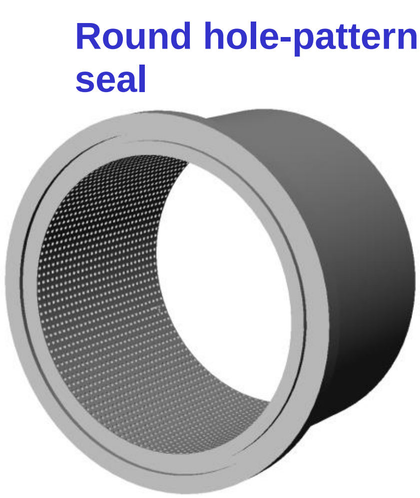


Figure 6. Straight-through and back-to-back bompressor configurations and their fundamental elastic mode shapes

Extensive testing has shown that seals with macroscopic roughness; i.e. **textured stator surfaces**, offer major improvements in reducing leakage as well as cross-coupled stiffness coefficients. Figure 7 depicts two textured seals and a conventional labyrinth seal (teeth on stator). A textured surface like a round-hole pattern or a honeycomb increases the friction thus reducing leakage, and aids to retard the development of the circumferential flow velocity -the physical condition generating the cross-coupled stiffness coefficients. Since the late 1990s, compressor and pump manufacturers, as well as end users, implement or use textured seals with noticeable improvements in pump or compressor efficiency and ensured rotordynamic stability.





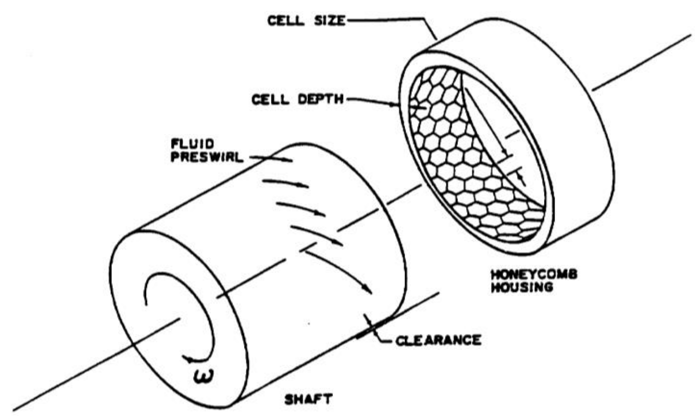


Figure 7. Hole-pattern, honeycomb and labyrinth seal configurations

Example of a rotordynamic analysis

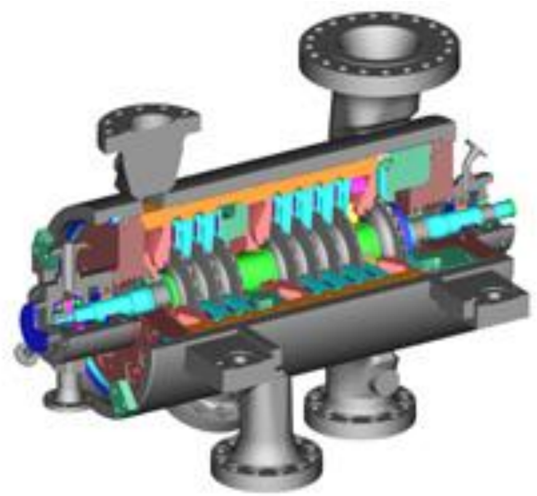


Figure 8. Cut-away view of a centrifugal compressor

The following brief description shows the major elements of a rotordynamic analysis performed on a seven-stage compressor handling a light hydrocarbon mixture.

The objectives of a rotordynamic analysis are:

a) To model the rotor (shaft and disks) and to determine its free-free natural frequencies.
b) To model the fluid film bearing and seal elements and to calculate the mechanical impedances (stiffness, damping and inertia force coefficients) connecting the rotor to its casing.
c) To perform an **eigenvalue analysis**, i.e. to predict the natural frequencies and damping ratios for the different modes (rigid and elastic) of vibration of the rotor as the rotor speed increases to magnitudes well above the design operating conditions. Positive damping ratios evidence the absence of rotordynamic instability.
d) To perform a **synchronous response analysis** to calibrated imbalances in order to predict the maximum amplitudes of vibration, the safe passage through critical speeds and to estimate the loads transmitted through the bearing supports.

   The results of studies (c) and (d) must satisfy stringent conditions as requested by API norms (API 610, for example).

Note that the typical rotordynamic analysis is linear, i.e. it relies on the representation of the bearings and seals as linear mechanical elements. That is, the second order differential equations describing the motion about an equilibrium position are linear. Of course a nonlinear analysis could also be performed but its efficiency and (improved) accuracy are, to this date, questionable. Furthermore, a linear analysis is mandatory to determine the operability of the turbomachine.

It is important to stress that the tasks (objective) described above need of extensive experimental and field support verification. Analysis without adequate measurements is usually not very useful in rotordynamics.

The example intends to show the complexity of a typical analysis. Figure 9 depicts the structural model with the rotor partitioned into 36 stations (each with inertia and inertia properties). The circles denote added inertias such as those from the impellers and thrust collars. The spring-like connections to ground denote the bearing and seal elements supporting the rotor.

Tables 1 to 3 show the physical properties of the rotor, the compressor operating conditions (current and desired), and a brief description of the bearings and seals in place.

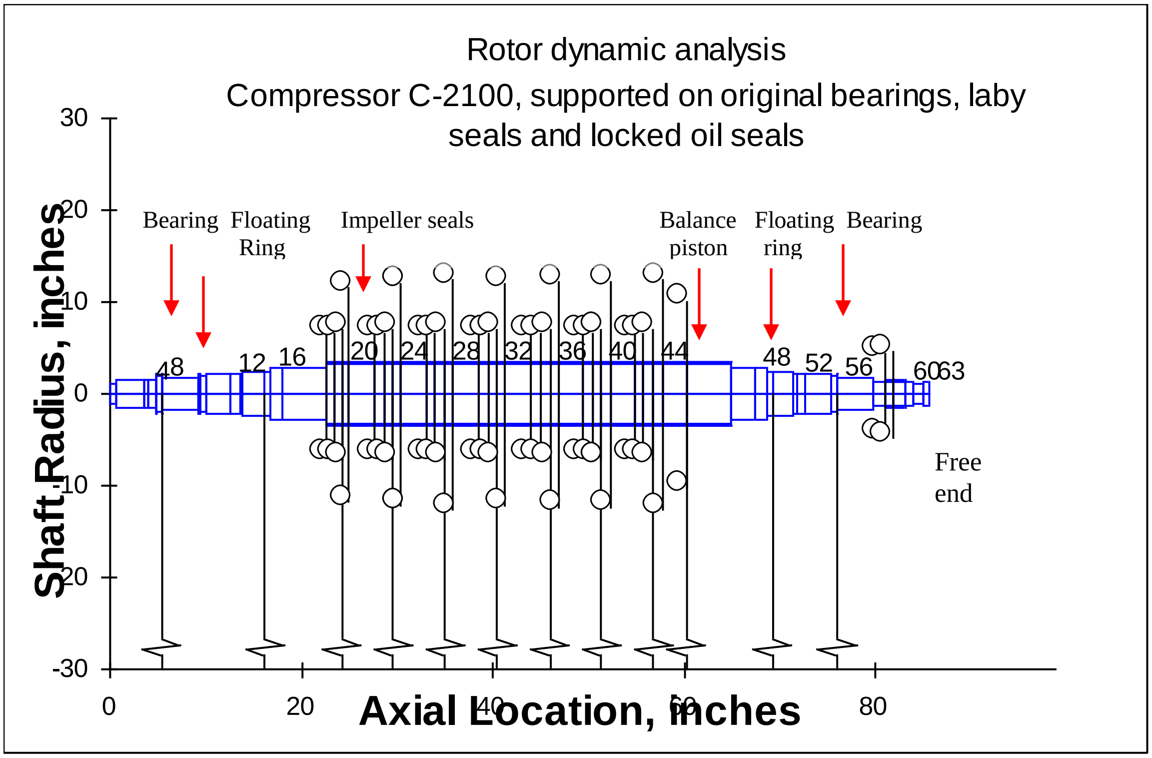


Figure 9. Structural rotordynamic model of a multiple-stage centrifugal compressor

Table 1. Geometry of rotor for compressor

Compresor C-2100	Physical units
Number of impellers	7
Shaft length	85.6 “ (2.17 m)
Rotor weight incluydes thrust collar	1,024 lb (4,550 N)
Center of mass from coupling side	43.65 “ – station 34
Mass moment of inertia (transversal0	302,815 lbm-in^2
Mass moment of inertia (polar)	16,749 lbm-in^2
Static load on bearing (coupling side)	469 lb (2,085 N)
Static load on bearing (free end)	554 lb (2,465 N)

Table 2. Operating conditions (actual and desired) for compresor

Hydrocarbon mixture (molecular weight 8.72)

	5,700 RPM		9,850 RPM	
Stage	Pressure (bar)	Temperature (K)	Pressure (bar)	Temperature (K)
0	20.00	311.0	21.00	311
7	27.00	338.0	33.00	360

Table 3 shows the location of the mechanical impedances to ground. The rotor is supported on two multiple lobe cylindrical bearings operating with ISO VG 32 oil. Furthermore, pressurized floating oil seal rings isolate the process gas from the environment. There are also seven eye impeller seals and six interstage seals of the labyrinth type. At the free end of the compressor there is a long balance piston.

Note to reader: f you have a genuine interest in the example, ask the lecturer for a more detailed description of the bearings and seals, i.e. dimensions and operating conditions.

Table 3. Bearing and seal locations in compressor

Station	Mechanical element	Description
8	Hydrodynamic bearing	Three lobe bearing (coupling end)
56	Hydrodynamic bearing	Three lobe bearing (free end)
15	Floating ring seal	Pressurized, lubricant
50	Floating ring seal	Pressurized, lubricant
46	Balance piston	Process Gas, 27 teeth
20, 24, 28 32, 36, 40	Impeller seals– neck ring (eye) and interstage	Labyrinth type, process gas 4 teeth
44	Eye Impulsor # 7 seal	Labyrinth type, process gas

The structural analysis predicts the free-free mode natural frequencies of the rotor, as given in Table 4. q free-free mode is an elastic natural mode of the rotor without any connection to ground, i.e., without bearings and seals. The good correlation with the field measurement is encouraging. The field test usually consists of hanging the rotor from long cables; then raping the shaft with a heavy object; and, recording the natural frequency (and mode shape) of motion.

Table 4. Free-free mode natural frequency of rotor (no thrust collar)

	calculated	Field measurement
Fundamental frequency	14,431 (RPM) (240 Hz)	14,400 (RPM)
2nd frequency	27,081 ‘’	Not recorded
3rd frequency	40,927 ‘’	‘’

The rotordynamic analysis predicts the eigenvalues (**damped natural frequency map,** see Figure 10, and **damping ratios,** see Figure 11) of the rotor operating on its bearings and seals for speeds to 20,000 rpm, twice the design value. The predictions show a lightly damped critical speed at 4,000 rpm. Most importantly, the analysis reveals a *rotordynamic instability* at 8,163 rpm. This speed is known as the *threshold speed* of instability. The instability is due to the loss of effective damping (needed to dissipate mechanical energy of vibrations) and excitation of the system (lowest) natural frequency with dangerously high amplitude vibrations. The field measurements evidence of the subsynchronous vibration at a lower speed, i.e. 7.850 rpm!

Table 5. Threshold speed of instability: predicted and measured

	Threshold speed	Whirl frequency	Whirl ratio	Mode
Predicted	8,163 rpm	4,000 rpm	0.49	Elastic
Field data	7,850 rpm	3,532 rpm	0.44	

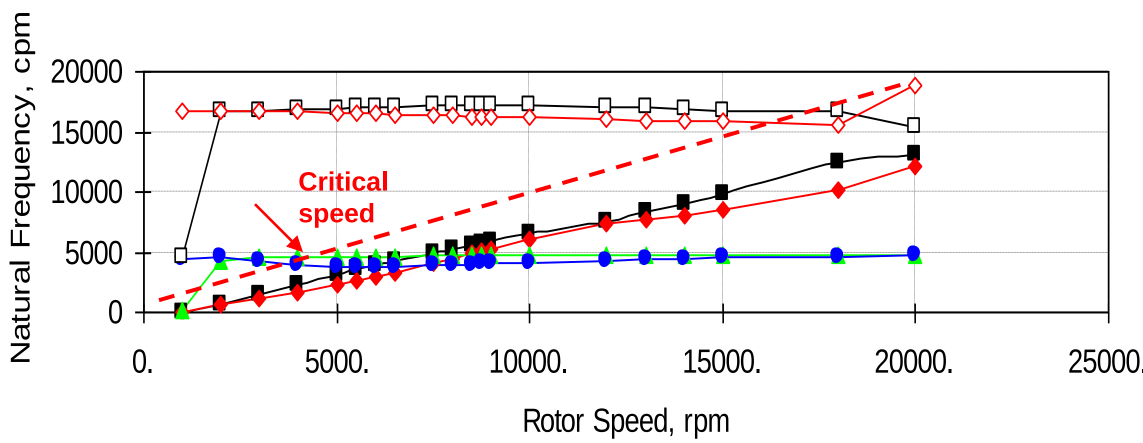


Figure 10. Natural frequency versus rotor speed for multiple-stage compressor

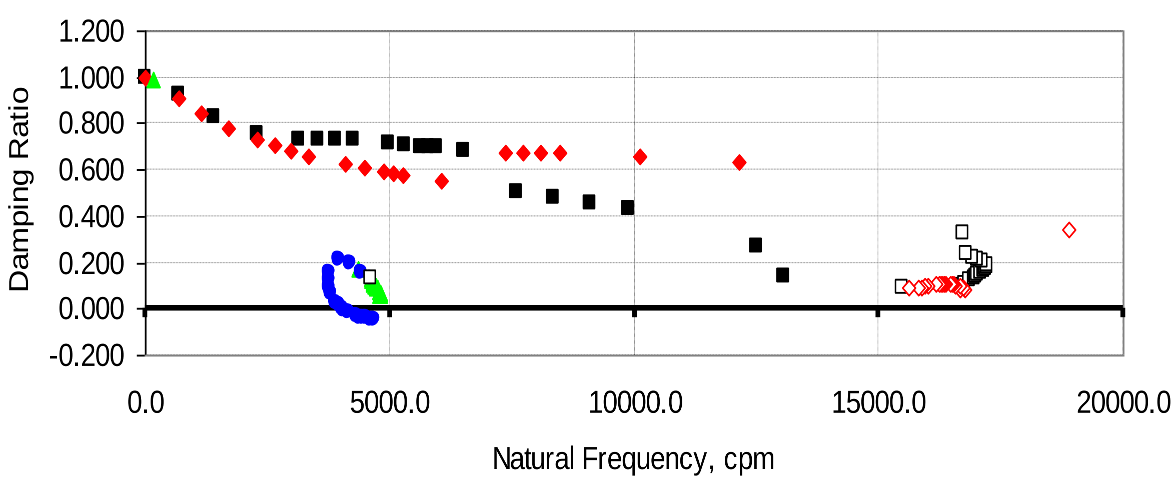


Figure 11. Damping ratio versus natural frequency for multiple-stage compressor

Figure 12 shows the amplitude of synchronous rotor motion versus rotor speed for a specified mass imbalance distribution. The predictions are shown for the rotor at one of the bearing locations. Figure 13 depicts the mode shape of vibration at a rotor speed of 8,750 rpm. Note that the predicted results are not valid for rotor speeds above the threshold speed of instability since the rotor (would) vibrate with a subsynchronous whirl frequency component of much larger amplitude than the one synchronous with rotor speed.

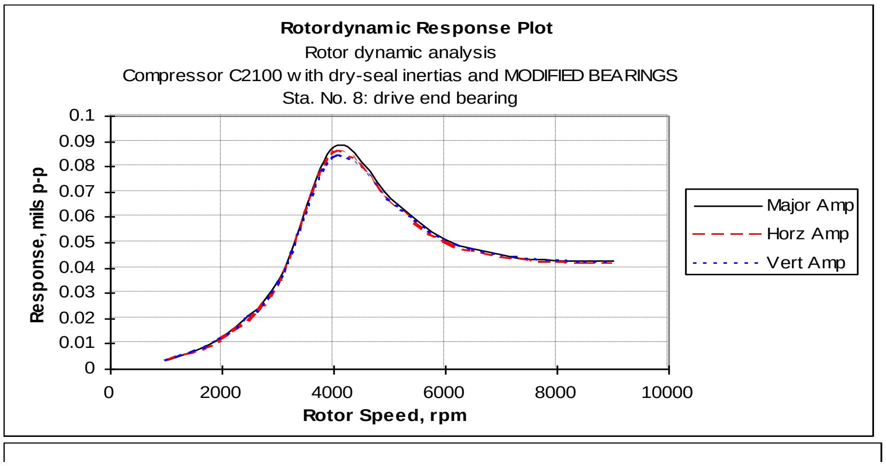


Figure 12. Predicted rotor imbalance response at drive end bearing

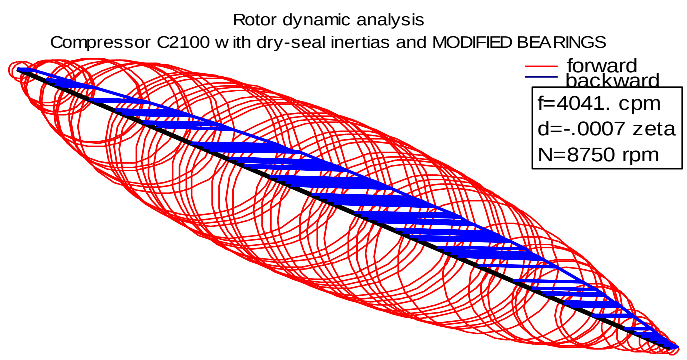


Figure 13. Deflected rotor mode shape at 8,750 rpm

The mode shape shows that at the bearing locations the rotor motion is quite small, while the vibration amplitude at the seal locations (rotor midspan) is much larger. Hence, the instability is certainly associated with a poor design of the support multiple-lobe bearings and the unfortunate lockup of the oil seal rings.

Finally, Figure 14 presents the field recorded vibration spectra. The rotor speed is 7,860 rpm and the dangerously high amplitude subsynchronous vibration develops at 3,532 rpm. The rotordynamic predictions are overly conservative!

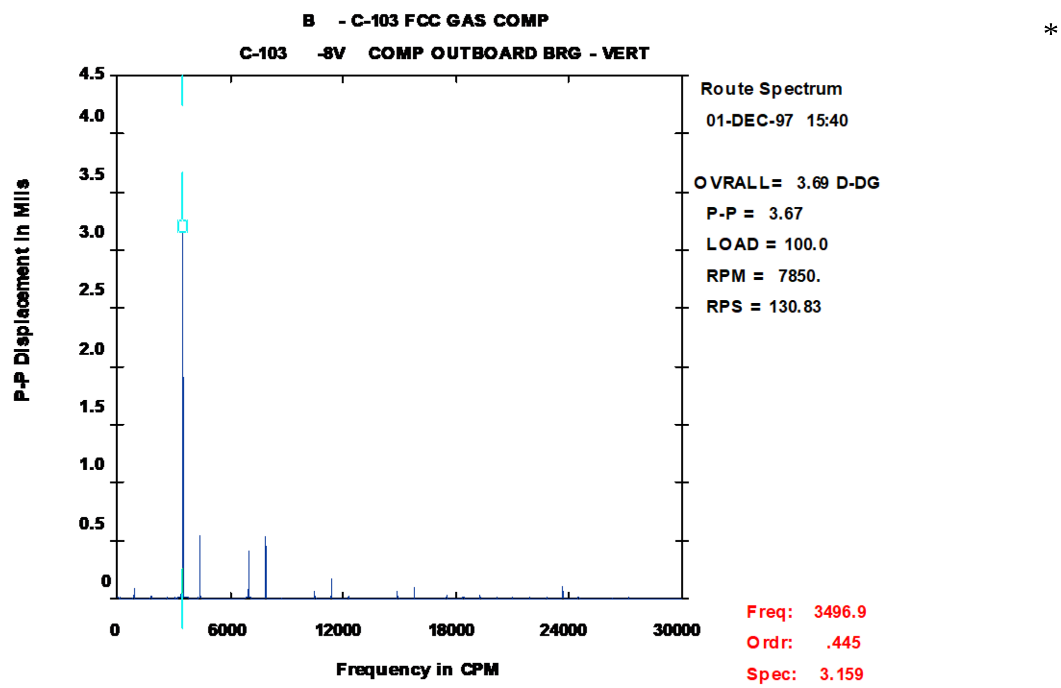


Figure 14. Field vibration spectrum showing rotordynamic instability for compressor example.

The example evidences the importance of fluid film bearings and seals on the dynamics of rotating machinery. Note that the example referred brought an unexpected stop in the operation of the unit with an enormous cost to the owner, several hundred thousand of dollars per day over an undisclosed amount of time. Fortunately, current monitoring techniques enabled the engineers to prevent a catastrophic failure with a potentially enormous financial impact and even human lives cost.

Closure

The following lectures detail the fundamentals of fluid film lubrication and rotordynamics that will enable the interested reader to begin analyzing fluid film bearings and seals for applications in rotating machinery. The course begins with a detailed analysis of the fundamentals of lubrication theory and its applications to oil-lubricated bearings and ring seals. Next, seals and squeeze film dampers are thoroughly covered. The importance of fluid inertia and flow turbulence on modern (currently used) bearing and seal applications is also covered.

Performance Objectives for the Modern Lubrication Course

1. **To learn about the physical concepts and mathematical models for the analysis and design of fluid film bearings and seals.**

2. **To acquire knowledge based on the detailed review of the literature on fluid film lubrication and rotordynamics.**

3. **To identify the mechanical effects of importance on the static and dynamic forced performance of fluid film bearings.**

4. **To learn about the effects of fluid film bearings on the rotordynamics of turbomachinery.**

5. **To identify the future trends in applications of bearing and seal technologies and the needs for further research.**

6. **To provide the basics of efficient computational skills for the prediction of the static and dynamic forced performance of fluid film bearings.**