Introduction to rotordynamics and lubricated elements

Dr. Luis San Andres
Mast-Childs Chair Professor
Turbomachinery Laboratory
Texas A&M University
Lsanandres@tamu.edu
Turbomachinery

A turbomachinery is a rotating structure where the load or the driver handles a process fluid from which power is extracted or delivered to.

Fluid film bearings (typically oil lubricated) support rotating machinery, providing stiffness and damping for vibration control and stability. In a pump, neck ring seals and inter stage seals and balance pistons also react with dynamic forces. Pump impellers also act to impose static and dynamic hydraulic forces.

Acceptable rotordynamic operation of turbomachinery:
Ability to tolerate normal (even abnormal transient) vibrations levels without affecting TM overall performance (reliability and efficiency)
Rotordynamics of turbomachinery (TM)

Goals
Conduct structural analysis of rotors (shafts and disks) and design of fluid film bearings and seals to render the best dynamic forced performance at the machine design operating conditions.

Best performance denotes well-characterized natural frequencies (and critical speeds) with amplitudes of synchronous motion within required standards and demonstrated absence of subsynchronous vibration instabilities.
Rotordynamics of turbomachinery (TM)

Best performance

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.

Rotordynamic problems are more frequent in high performance TM since it concentrates more power & operates at ever increasing high speeds. Stability limits are usually determined by load condition, i.e. changes to the operating point.
PV/CV turbochargers

RBS
Fully Floating Bearing

RBS
Semi Floating Bearing

RBS
Ball Bearing
Centrifugal compressor
SSME turbopumps
Most common problems in rotordynamics

Excessive steady state synchronous vibration levels & Sub harmonic rotor instabilities

**Steady state vibration levels** may be reduced by:
- Improving balancing
- Modifying rotor-bearing systems: tune system critical speeds out of RPM operating range
- Introducing damping to limit peak amplitudes at critical speeds that must be traversed

**Sub harmonic rotor instabilities** may be avoided by:
- Raising the natural frequency of rotor system as much as possible
- Eliminating the instability mechanism, i.e. change bearing design if oil whip is present
- Introducing damping to raise onset speed above the operating speed range.

[Ehrich and Childs 1]
Fluid Film Bearings

Fluid film bearings produce low friction between solid surfaces in relative motion and generate a load support for mechanical components.

The lubricant or fluid between the surfaces may be a liquid, a gas or even a solid (coating).

Fluid film bearings, if well designed, support static and dynamic loads, affecting the dynamic performance of rotating machinery.

Basic operational principles are hydrodynamic, hydrostatic or hybrid (a combination of the former two).
Bearings enable smooth (low friction) motion between solid surfaces in relative motion and, if well designed, support static and dynamic loads. **Bearings** affect the dynamic performance of machinery (reliability and availability).

**Generalized Striebeck Curve**

- Boundary lubrication
- Mixed lubrication
- (Elasto) hydrodynamic lubrication

**Friction coefficient**

**Surface velocity x viscosity**

**Specific pressure**

**Full film lubrication**
Hydrodynamic Bearings

**Hydrodynamic pressure** generated by relative motion between two mechanical surfaces with a particular “wedge like” shape.

### Advantages

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.

Schematic view of hydrodynamic (self-acting) fluid film bearing.
Disadvantages

Thermal effects affect performance if film thickness is too small or available flow rate is too low.

Require of surfaces’ 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.
Examples of hydrodynamic bearings

- Plain Journal
- Four Axial Groove
- Elliptical
- Offset Half
- Four-Lobe
- Pressure Dam
- Partial Arc Journal Bearing
- Floating Ring Journal
- Pressure Dam Journal Bearing
- Tilting pad bearings
- Typical cylindrical journal bearings
Hydrostatic Bearings

External source of pressurized fluid forces lubricant to flow between two surfaces, thus enabling their separation and the ability to support a load without contact.

**Advantages**

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 or 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.
## Hydrostatic Bearings

### Disadvantages
- Require ancillary equipment. Larger installation and maintenance costs.
- Need of fluid filtration equipment. Loss of performance with fluid contamination.
- Penalty in power consumption: pumping losses.
- **Limited LOAD CAPACITY ~ f(P_{supply})**
- Potential to induce **hydrodynamic instability** in hybrid mode operation.
- Potential to show **pneumatic hammer instability** with compressible fluids, i.e. loss of damping at low and high frequencies of operation due to compliance and time lag of trapped fluid volumes.

![Schematic view of hydrostatic/hydrodynamic journal bearing](image)
Normal surface motions can also generate hydrodynamic pressures in the thin film separating two surfaces.

The **squeeze film action** works effectively only for compressive loads, i.e. those forcing the approach of one surface to the other.

**Squeeze film dampers** are routinely used to reduce vibration amplitudes and isolate structural components in gas jet engines, high performance compressors, and occasionally in water pumps.
Annular Pressure Seals

Seals (annular smooth, labyrinth or honeycomb) separate regions of high pressure and low pressure and their principal function is to minimize the leakage (secondary flow); thus improving the overall efficiency of a TM extracting or delivering power to a fluid. Seals have larger clearances than load carrying bearings.

Seals in a Multistage Centrifugal Pump or Compressor
Annular Pressure Seals

Due to their relative position within a rotor-bearing system, seals modify sensibly the system dynamic behavior. Seals typically "see" large amplitude rotor motions. This is particularly important on back-to-back compressors and long-flexible multiple stage pumps.
Steam turbine

- 50Hz 660MW - HP turbine

Note: 1, 2, 3, 4, 5, 6, 7, 8, 133, 134, 138: Seal location
Steam IP turbine

Note) 11, 12, 13, 14, 15, 16, 17, 18, 136 : Seal location
Intentionally roughened stator surfaces (macro texturing) reduce the impact of undesirable cross-coupled dynamic forces and improve seal stability.

Annular seals acting as Lomakin bearings could be support elements (damping bearings) for cryogenic turbopumps as well in process fluid pumps & high pressure compressors.
Rotordynamic Analysis

**Model structure**: (shaft and disks) and find free-free mode natural frequencies

**Model bearings and seals**: predict mechanical impedances (stiffness, damping and inertia force coefficients)

**Eigenvalue analysis**: predict damped natural frequencies and damping ratios for various modes (rigid and elastic) of vibration as the rotor speed increases (typically 2 x operating point)

**Synchronous response analysis**: predict peak amplitudes 1X motion, safe passage through critical speeds and estimate bearing loads

**To certify reliable performance** of rotor-bearing system satisfying established engineering criteria (**API 610 qualification**) and to emit recommendations to improve the system performance (response and stability)
Rotordynamic Analysis

Equations of motion:

\[
([M] + [N])_R \{\ddot{u}\} - \Omega [G]_R \{\dot{u}\} + [K]_R \{u\} = \{F(u, \dot{u}, t)\}
\]

- **Rotor inertia**
- **Rotor gyroscopics, \( fn \) (rotor speed)**
- **Rotor elastic properties**
- **Forces: external and from bearings & seals**

**DOFs at a node:** 2 translations \((X, Y)\) and two rotations \((\delta X, \delta Y)\)
Linear EOMs for rotor-bearing-seals system

\[ [M] \{\ddot{u}\} + \left[ [C] - \Omega [G]_R \right] \{\ddot{u}\} + [K] \{u\} = \{F_{ext}\} \]

- **Mass matrix**
- **Damping & Gyroscopic matrices**
- **Rotor + bearing + seal stiffness matrix**
- **External forces (Imbalance, shocks, etc)**

\[
[M] = ([M] + [N])_R \cup \sum [M]_S; \\
[K] = [K]_R \cup \sum [K]_B \cup \sum [K]_S; \\
[C] = \cup \sum [C]_B \cup \sum [C]_S;
\]

**Eigenvalues:**

\[
[K] + s [C] - \Omega s [G]_R + s^2 [M] \{v\} = \{0\}
\]

\(s = \lambda + i \omega; \quad \lambda < 0 \text{ for stability}\)

**Synchronous response**

\[
[K] + i \Omega [C] - i \Omega^2 [G]_R - \Omega^2 [M] \{w\} = \{mxue^{i\phi}\}
\]
Component-Mode Synthesis (CMS)

- Timoshenko-beam, FE-formulation
- Calculates real modes
- Reduces model dimensionality by using a limited number of modes

Rotor structure model
Rotor structural FE model

Compressor  thrust disk  shaft  turbine

Typical TC rotor hardware

• Beam Finite-Element Formulation

Typical FE rotor structure model
Validate rotor model

2 rotor model

Validate rotor model with measurements of free-free modes (room Temp)

Rotor: 6Y gram
SFRB: Y gram

Static weight load distribution
Compressor Side: Z
Turbine Side: 5Z

Rotor model with measurements of free-free modes (room Temp)
Free-free natural frequency & shapes

<table>
<thead>
<tr>
<th>Mode</th>
<th>Measured (Freq)</th>
<th>Predicted (Freq)</th>
<th>% Diff</th>
</tr>
</thead>
<tbody>
<tr>
<td>First</td>
<td>1.799 KHz</td>
<td>1.823 KHz</td>
<td>1.3%</td>
</tr>
<tr>
<td>Second</td>
<td>4.938 KHz</td>
<td>4.559 KHz</td>
<td>7.7%</td>
</tr>
</tbody>
</table>

Measured and predicted free-free natural frequencies and mode shapes agree: rotor model validation
Bearings and seals

Support rotor with low friction. These elements react with forces that depend on the rotor motion.
Bearing dynamic forces

DOF lateral displacements (X,Y)

Measure of stability: Whirl frequency ratio

\[ \text{WFR} = \frac{K_{xy}}{C_{xx} \omega} \]

Typically:

- No fluid inertia or moment coefficients accounted for
- Force coefficients independent of excitation frequency for incompressible lubricants. Functions of speed & load
Seal forces

Liquid seal:

\[
\begin{bmatrix}
F_x \\
F_y
\end{bmatrix} = -\begin{bmatrix}
K_{XX} & K_{XY} \\
K_{YX} & K_{YY}
\end{bmatrix}_S \begin{bmatrix}
X \\
Y
\end{bmatrix} - \begin{bmatrix}
C_{XX} & C_{XY} \\
C_{YX} & C_{YY}
\end{bmatrix}_S \begin{bmatrix}
\dot{X} \\
\dot{Y}
\end{bmatrix} - \begin{bmatrix}
M_{XX} & M_{XY} \\
M_{YX} & M_{YY}
\end{bmatrix}_S \begin{bmatrix}
\ddot{X} \\
\ddot{Y}
\end{bmatrix}
\]

Stiffness coefficients

Damping coefficients

Inertia coefficients

Gas seal:

\[
\begin{bmatrix}
F_x \\
F_y
\end{bmatrix} = -\begin{bmatrix}
K_{XX}(\omega) & K_{XY}(\omega) \\
K_{YX}(\omega) & K_{YY}(\omega)
\end{bmatrix}_S \begin{bmatrix}
X \\
Y
\end{bmatrix} - \begin{bmatrix}
C_{XX}(\omega) & C_{XY}(\omega) \\
C_{YX}(\omega) & C_{YY}(\omega)
\end{bmatrix}_S \begin{bmatrix}
\dot{X} \\
\dot{Y}
\end{bmatrix}
\]

Typically: frequency dependent force coefficients
Concept of stability and cross-coupled forces

Forces driving and retarding rotor whirl motion

Cross-coupled force = $K_{rt} \Delta e$

Damping force = $-C_{tt} \omega \Delta e$

Rotor spin, $\Omega$

Whirl orbit, $\omega$

Measure of stability: $WFR = \frac{K_{rt}}{C_{tt} \omega}$
Example: Compressor

**OBJECTIVE:** perform complete rotordynamic analysis of compressor

<table>
<thead>
<tr>
<th>Compressor C-2100</th>
<th>Physical units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of impellers</td>
<td>7</td>
</tr>
<tr>
<td>Shaft length</td>
<td>85.6 “ (2.17 m)</td>
</tr>
<tr>
<td>Rotor weight includes thrust collar</td>
<td>1,024 lb (4,550 N)</td>
</tr>
<tr>
<td>Center of mass from coupling side</td>
<td>43.65 “ – station 34</td>
</tr>
<tr>
<td>Mass moment of inertia (transversal)</td>
<td>302,815 lbm-in^2</td>
</tr>
<tr>
<td>Mass moment of inertia (polar)</td>
<td>16,749 lbm-in^2</td>
</tr>
<tr>
<td>Static load on bearing (coupling side)</td>
<td>469 lb (2,085 N)</td>
</tr>
<tr>
<td>Static load on bearing (free end)</td>
<td>554 lb (2,465 N)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Stage</th>
<th>5,700 RPM</th>
<th>9,850 RPM</th>
<th>9,850 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Pressure (bar)</td>
<td>Temperature (K)</td>
<td>Pressure (bar)</td>
</tr>
<tr>
<td>0</td>
<td>20.00</td>
<td>311.0</td>
<td>21.00</td>
</tr>
<tr>
<td>7</td>
<td>27.00</td>
<td>338.0</td>
<td>33.00</td>
</tr>
</tbody>
</table>

**Compressor operating conditions (actual and desired)**

Hydrocarbon mixture (molecular weight 8.72)
**Structural model and supports**

Free-free mode natural frequencies

<table>
<thead>
<tr>
<th>Station</th>
<th>Mechanical element</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>Hydrodynamic bearing</td>
<td>Three lobe bearing (coupling end)</td>
</tr>
<tr>
<td>56</td>
<td>Hydrodynamic bearing</td>
<td>Three lobe bearing (free end)</td>
</tr>
<tr>
<td>15</td>
<td>Floating ring seal</td>
<td>Pressurized, lubricant</td>
</tr>
<tr>
<td>50</td>
<td>Floating ring seal</td>
<td>Pressurized, lubricant</td>
</tr>
<tr>
<td>46</td>
<td>Balance piston</td>
<td>Process Gas, 27 teeth</td>
</tr>
<tr>
<td>20, 24, 28, 32, 36, 40</td>
<td>Impeller seals—neck ring (eye) and inter stage</td>
<td>Labyrinth type, process gas 4 teeth</td>
</tr>
<tr>
<td>44</td>
<td>Eye Impeller # 7 seal</td>
<td>Labyrinth type, process gas</td>
</tr>
</tbody>
</table>

**Fundamental frequency**
- C-2100
  - Calculated: 14,431 RPM (240 Hz)
  - Measurement: 14,400 RPM

**2nd frequency**
- 27,081 RPM
- Unknown

**3rd frequency**
- 40,927 RPM
- ""
Natural frequencies and damping ratio

Rotordynamic Damped Natural Frequency Map

<table>
<thead>
<tr>
<th>Threshold speed</th>
<th>Whirl frequency</th>
<th>Whirl ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Predicted</td>
<td>8,163 rpm</td>
<td>4,000 rpm</td>
</tr>
<tr>
<td>Field data</td>
<td>7,850 rpm</td>
<td>3,532 rpm</td>
</tr>
</tbody>
</table>

Rotordynamic Root Locus Plot

Field vibration spectrum showing rotordynamic instability
Rotordynamics → applications

21st century turbomachinery
Ultra-performance (reinjection) compressors: > 15,000 psi (1,000 bar)

Combined cycle turbines (gas/steam): efficiency > 60%

Aircraft: Larger high-bypass geared turbofans (GR>5)

Electric distributed propulsion systems
- GTs → batteries → electric fans for thrust
- Larger efficiency & lower noise. Braking regenerative power

Unmanned Aerial Vehicles (Drones): war at a distance & no casualties, surveillance, parcel mail delivery

Reusable rocket engines: LH₂ and LOₓ with fluid film bearings
Subsea pumping & compression

High pressures & extreme temperatures

Subsea Engineering or SURF –

**Subsea**

**Umbilicals**

**Risers**)

**Flowlines**

*Wet compression systems must be reliable (5 y operation)*

**Meso-micro turbomachinery:**
portable packs (5 kW), 1 million rpm

**Oil-free gas turbines and generators:**
(mid size to 0.5 MW): foil gas bearings, damper seals.

Rotordynamics, 3D printing, materials

coatings: solid lubes gas lubrication & rotordynamics
Largest power to weight ratio
Compact & low # of parts
Reliability and efficiency
Low maintenance
Extreme temperature and pressure – multiple phases
Environmental safe (low emissions)
Lower lifecycle cost ($ kW)

Microturbomachinery needs & hurdles

High speed

Materials
Ceramic rotors and components
Bearings & Sealing
Rotordynamics & (Oil-free)
Automated agile processes
Additive manufacturing: $ & #
Low-NOx combustors for liquid & gas fuels. Scaling to low Reynolds #
Coatings: for low friction and wear

Processes & Cycles
Low-NOx combustors for liquid & gas fuels. Scaling to low Reynolds #

Manufacturing
Automated agile processes
Additive manufacturing: $ & #

Fuels
Best if free (bio-fuels)
ACMs, APUs, blowers, compressors.....

Successful with rigid rotors and limited in damping. Must enable operation above rotor flexural modes.
Superchargers & micro-power gen

Ready technology

Hybrid vehicles: 50 miles/gal & 0 NOx $\rightarrow$ fuel cells. Issues are high temperature, materials and NL rotordynamics

2014 KIST (Lee, Kim & Kim)
Questions (?)