

Hydrodynamic Journal Bearings

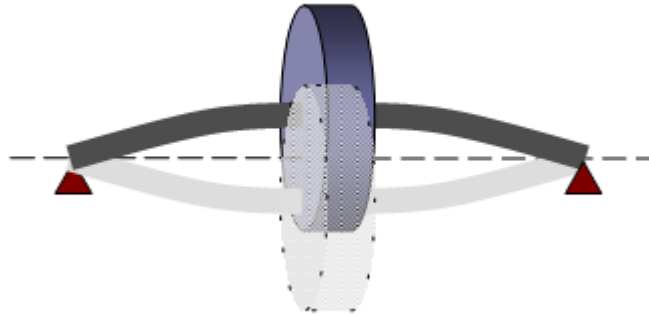
Nicolas Péton



imagination at work

Agenda

- The rotor doesn't vibrate but it is bowed and whirls about its bearing centerline
- Plain journal bearing
- To a stationary external observer the rotor appears to vibrate but it is just the planar projection of the orbit as seen from one side

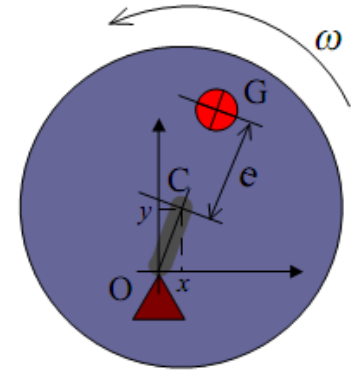


- Let's consider the Jeffcott rotor in the hypothesis that damping is negligible
- The equilibrium in radial direction can be written as

$$-M\omega^2(e + OC) = -K OC$$

- Assuming a perfectly balanced rotor ($e=0$)

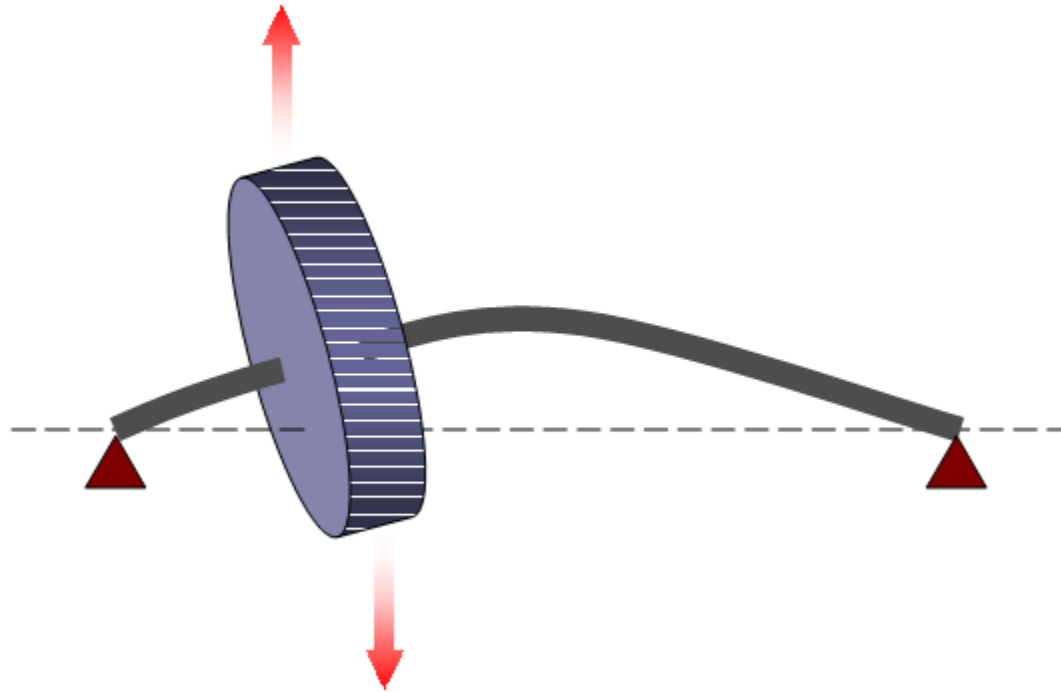
$$(M\omega^2 - K) OC = 0$$



- The solutions are $OC = 0$ or $OC \neq 0$ if $\omega = \sqrt{\frac{K}{M}}$

- In other terms when the unbalance is zero the shaft can have a whirl only if the rotational speed coincides with the undamped natural frequency

- Gyroscopic effect tends to “stiffen” the rotor that whirls increasing the critical speeds
- The effect is related to polar moment of inertia

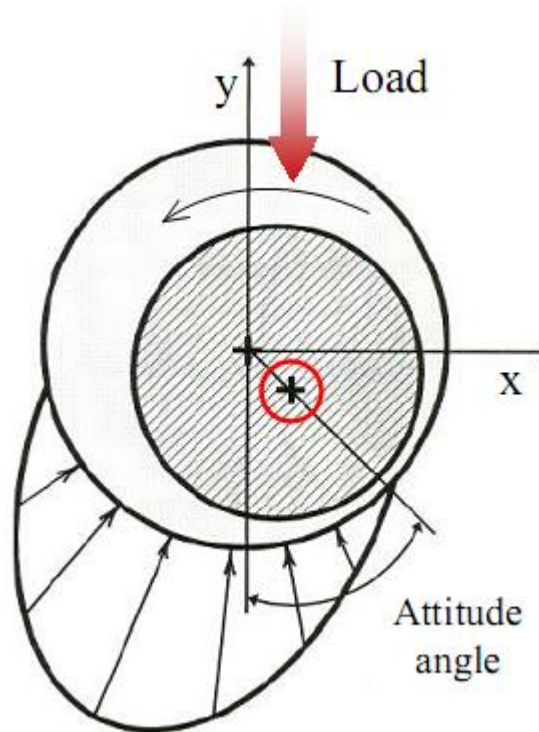


- A couple of radial bearings supports the weight of the rotating shaft by developing a hydrodynamic over pressure in the converging wedge

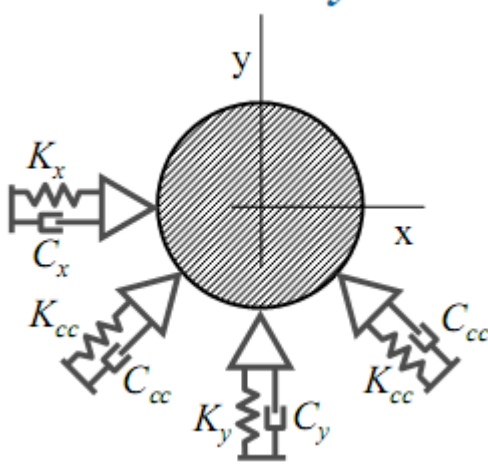


- The pressure profile originates forces that are proportional to centerline displacement and precession velocity

- In the simple case of a plain bearing the shaft, pushed by pressure, climbs up in the bearing forming an attitude angle between the line of centers and the load vector



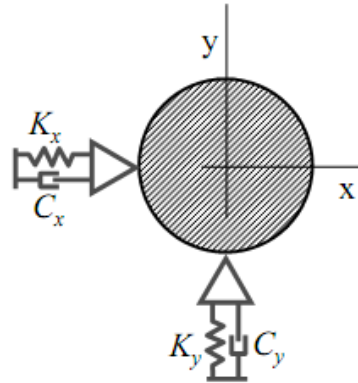
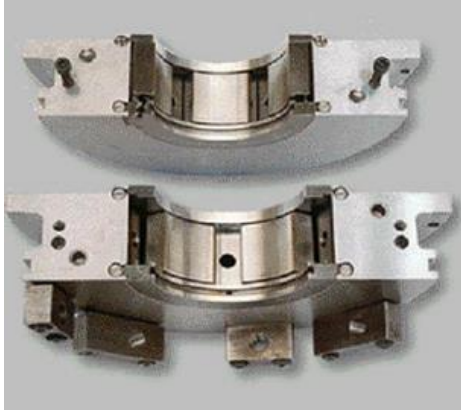
- The load acting on the journal bearing results in a displacement along the load direction and a displacement orthogonal to the load direction (cross coupling)



$$\begin{pmatrix} F_x^{stiffness} \\ F_y^{stiffness} \end{pmatrix} = \begin{pmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{pmatrix} \begin{pmatrix} x \\ y \end{pmatrix} = \begin{pmatrix} K_{xx}x + K_{xy}y \\ K_{yx}x + K_{yy}y \end{pmatrix}$$

$$\begin{pmatrix} F_x^{damping} \\ F_y^{damping} \end{pmatrix} = \begin{pmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{pmatrix} \begin{pmatrix} x \\ y \end{pmatrix} = \begin{pmatrix} c_{xx}x + c_{xy}y \\ c_{yx}x + c_{yy}y \end{pmatrix}$$

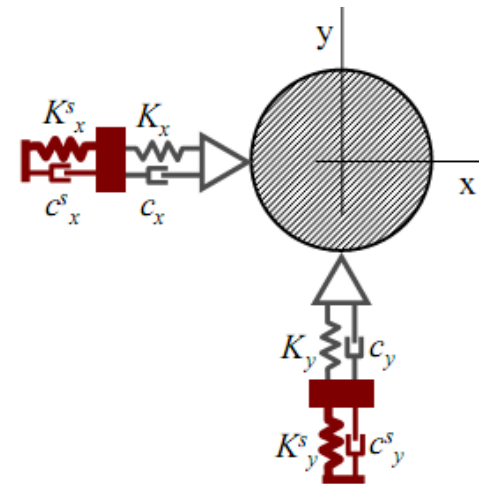
- The pads pivot and rotate in response to the load applied and produces a symmetric reaction force on the journal
- The attitude angle is approximately zero
- Tilting pad bearings achieve very low cross coupling



$$\begin{pmatrix} F_x^{stiffness} \\ F_y^{stiffness} \end{pmatrix} = \begin{pmatrix} K_{xx} & 0 \\ 0 & K_{yy} \end{pmatrix} \begin{pmatrix} x \\ y \end{pmatrix} = \begin{pmatrix} K_{xx}x \\ K_{yy}y \end{pmatrix}$$

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- For most turbo machines the support stiffness is much greater than the bearing stiffness and therefore has a negligible effect on rotor dynamics
- For centrifugal compressors and steam turbine when support stiffness is less than 3.5 times the bearing stiffness, support effect has to be included in the calculation
- In industrial gas turbine the interaction between rotor and casing is many times very strong.

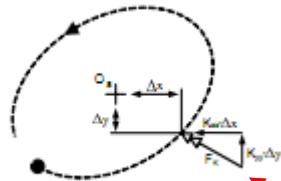


- The principal or direct coefficients are:

$$K_{xx} = \Delta F_x / \Delta x \quad \text{Horizontal Principal Stiffness}$$

$$K_{yy} = \Delta F_y / \Delta y \quad \text{Vertical Principal Stiffness}$$

- They relate the change in force in one direction due to a displacement in the same direction. In other words, these direct stiffness provides a restoring force that pushed the journal back toward its steady state equilibrium position



Force that tries to
push the journal
back to O_s

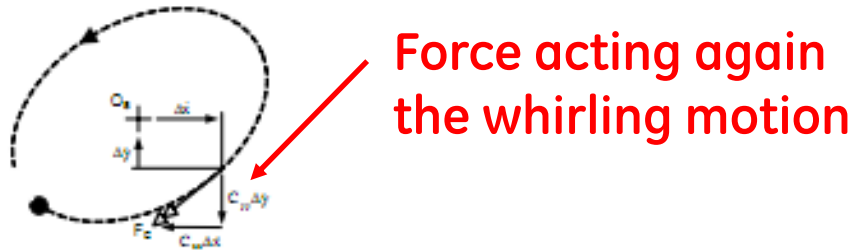
- Large asymmetry for K_{xx} & K_{yy} is the main cause for split critical speeds and noncircular orbit shapes
- Two principal or direct damping coefficients are also present

$$C_{xx} = \Delta F_x / \Delta \dot{x} \quad \text{Horizontal Principal Damping}$$

$$C_{yy} = \Delta F_y / \Delta \dot{y} \quad \text{Vertical Principal Damping}$$

- The damping coefficient relates the change in force due to a small change in velocity. Because of the whirling motion the combination of these two principal damping coefficients produces forces that are tangential to the vibration orbit.

- Furthermore, as shown in the figure below, this direct damping force acts against the whirling motion, helping to retard or slow it

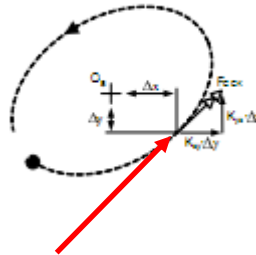


- The off-diagonal stiffness coefficients K_{xy} and K_{yx} are the cross coupled stiffness coefficients.

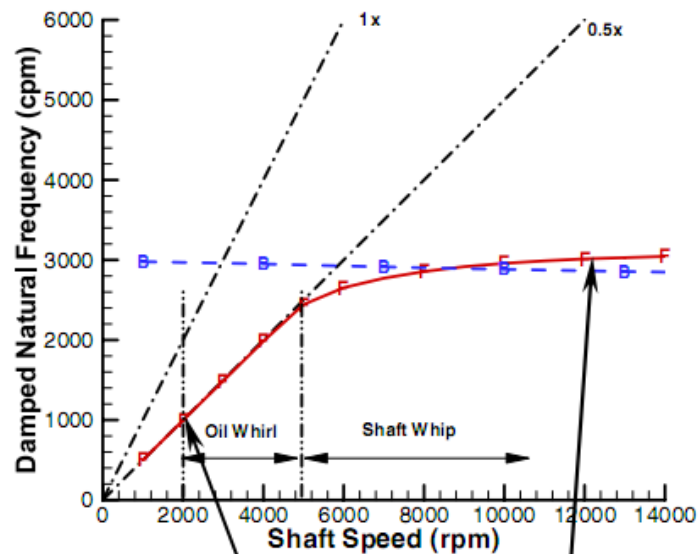
$$K_{xy} = \Delta F_x / \Delta y$$

$$K_{yx} = \Delta F_y / \Delta x$$

- As an example the coefficient K_{yx} relates a vertical force due to a horizontal displacement
- Rotor systems are unique in that the symmetry usually doesn't exist
- $K_{xy} \neq K_{yx}$ and usually $K_{xy} > 0$, $K_{yx} < 0$
- Instead of opposing the rotor's whirling motion like the direct damping the cross coupled stiffnesses combine to create a force pointing in the whirl direction, promoting the shaft vibration



- When the direct damping force is unable to dissipate the energy injected by the cross coupled stiffness force, the natural frequency (typically the lowest one with forward whirling direction) will become unstable causing the shaft to whirl at this frequency

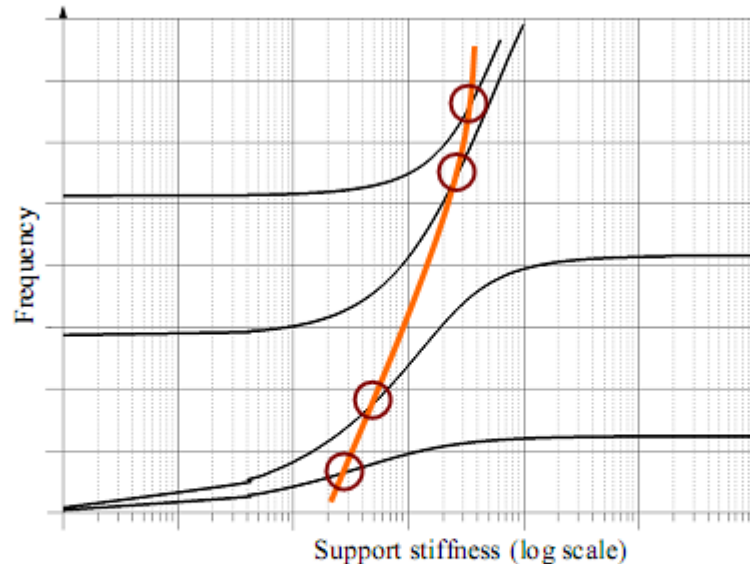


- In real life most unstable machines exhibit shaft whip directly without exhibiting oil whirl behavior
- Oil whirl is one exception where the bearing's dynamic properties dominate the rotordynamics behavior of the system
- Unlike oil whirl, the rotor's mode shape in whip undergoes noticeable bending and its flexibility plays a significant role in the system's overall dynamics.

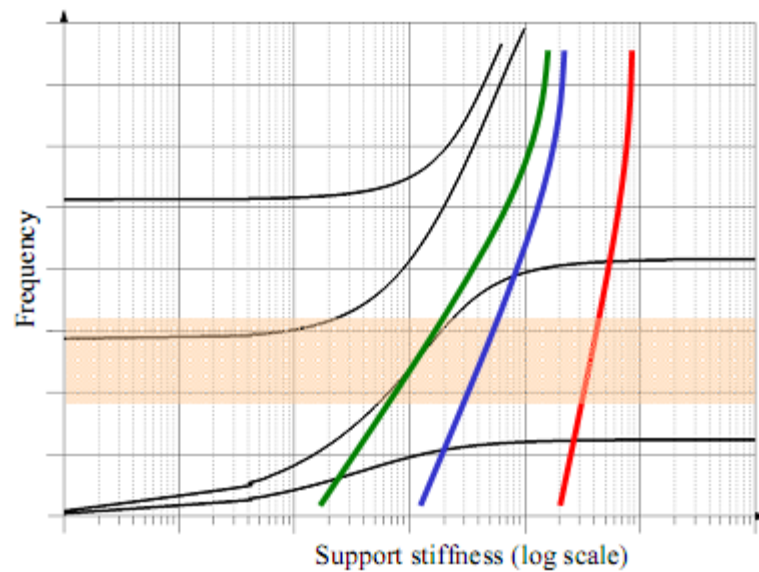
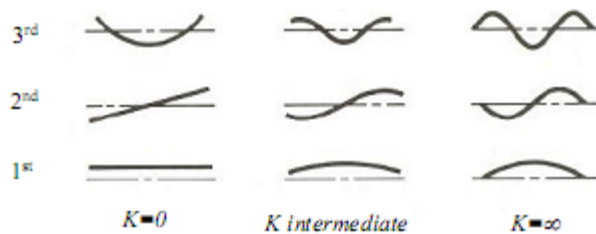
- The key to understand is, does this instability occur at a speed that is below twice the first critical ? If it does it takes less energy to lock onto a frequency that is exactly 50% of synchronous frequency “Oil Whirl”
- Once above twice the first critical it is easier for the rotor to respond at the critical frequency as opposed to a 50% component
- If there is an exact 50% component we would look first for a rub (for example) but if it is an old machine with plain cylindrical sleeve bearing it may be bearing induced instability

<i>Bearing Type</i>	<i>Sketch</i>	<i>Direction of operation</i>	<i>Cross Coupling</i>	<i>Stiffness and damping</i>	<i>Comments</i>
Cylindrical				Moderate	
Cylindrical with dammed grooves				Moderate	
Lemon				Moderate	
Three lobes				Good	
Offset halves				Excellent	
Tilting pad				Good	• Compressors, Steam and Gas Turbines • High cost

- The critical speed map collects the values of natural frequencies of the rotor as a function of support stiffness



Given the bearing stiffness the map permits a rough estimation of actual critical speeds



Operating envelope

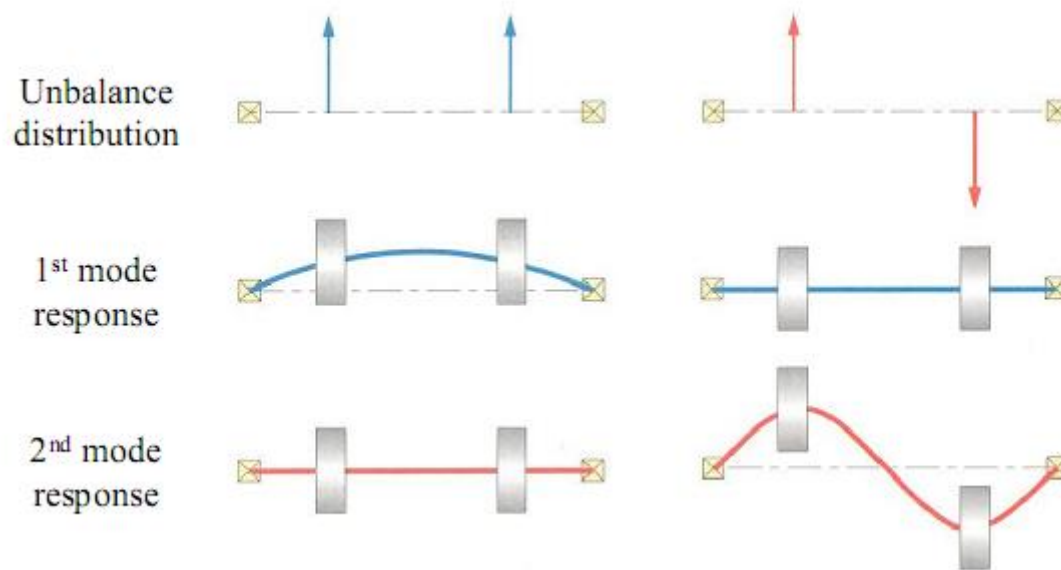
- ☐ Very high bearing stiffness
- ☐ Shaft bending dominated modes
- ☐ Low damping

- ☐ High bearing stiffness
- ☐ Rotor operating between the 1st and the 2nd critical speeds

- ☐ Low bearing stiffness
- ☐ Bearing dominated modes
- ☐ High damping
- ☐ Critical speed within operating envelope

- Critical speed map is useful to roughly position critical speeds but real critical speeds are affected by damping both in terms of frequency and peak amplitude
- The anisotropy of real systems makes the orbits elliptical
- Changing the speed the stiffness and damping characteristics of bearings change as well the shape of the ellipse
- In order to identify the orbit pattern and the maximum vibration amplitude two axes are needed

- The degree of excitation of any mode depends on how well the unbalance distribution fits the particular mode shape.



- Undamped mode shapes are used to place unbalances for rotor response

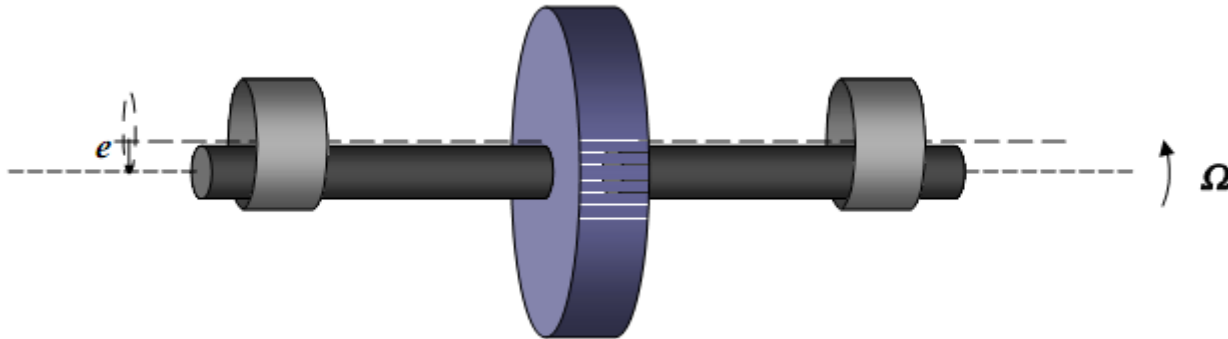
- A real rotor is always characterised by a certain unbalance and therefore is always subjected to a forced vibration
- For a real rotor free vibration is therefore something that superimposes to the unavoidable forced vibration
- Free vibration is excited by an impulse applied to the rotor. The vibration can occur with one or several system natural frequency
- These impulses may be due to the following causes electrical short circuit, surge, internal rubs, slug of liquid
- The free vibrations decay as the initially input energy dissipates at a rate that depends on the amount of damping



- Instability is characterised by whirling of rotor at frequencies other than shaft speed
- Instability always involves the excitation of a natural frequency
- The word instability implies that the motion can increase without limit and this sometimes occurs with destructive consequences
- The phenomena is usually associated with the variation of some pressures around the circumference of a rotor component and results in forces related to rotor displacement and velocity

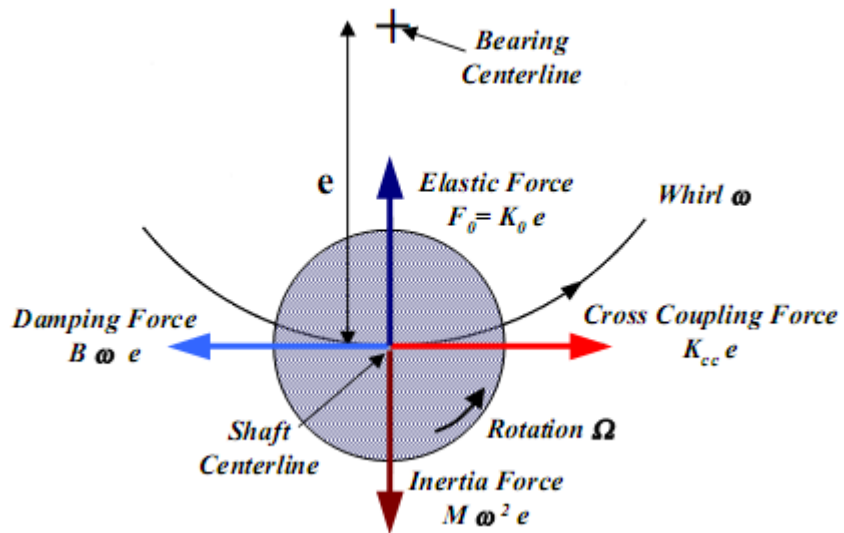
- Machine parts where these phenomena may take place are: journal bearings, gas labyrinth seals, blade tip clearance (Alford's force)
- Oil seals (no more in use) generated forms of instability with the same mechanism of journal bearings
- A different form of self excited vibration is caused by dry friction (backward whirl driven by rubbing friction between rotor and stator)

- A perfectly balanced isotropic stiff rotor of mass M without gyroscopic effect
- Plain journal bearing
- Shaft is rotating around its axis with a constant speed Ω



WHAT ARE THE CONDITIONS FOR A PRECESSION WHIRL?

- If the phenomenon takes place its precession frequency is the undamped natural frequency.
- The bearing physics is such to introduce a cross coupling force normal to the radial deflection e in the direction of the whirling motion
- The cross coupling balances the damping forces
- The precession is forward; except for dry friction whirl virtually all rotor dynamics instabilities are forward



Radial force balance

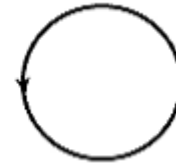
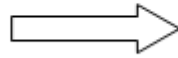
$$-M\omega^2 e = -K_0 e \implies \omega = \sqrt{\frac{K_0}{M}}$$

Tangential equilibrium

$$0 = -B\omega e + K_{cc} e$$

- When the cross coupling balances the damping forces the precession is stable

$$B\omega e = K_{cc}e$$



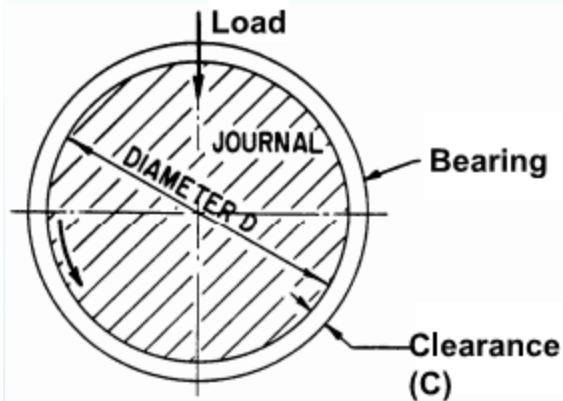
- If the coupling exceeds the damping force the rotor will absorb energy increasing the precession amplitude (in theory) without bound

$$B\omega e < K_{cc}e$$

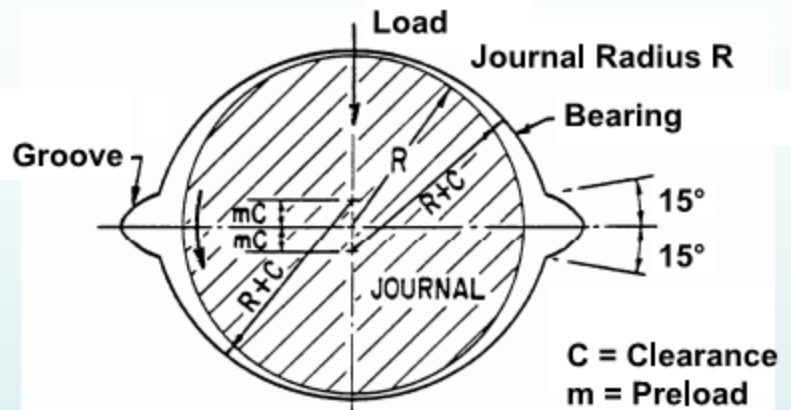


- Typically the rotor motion will increase until system nonlinearities restrain the rotor to some sort of limit

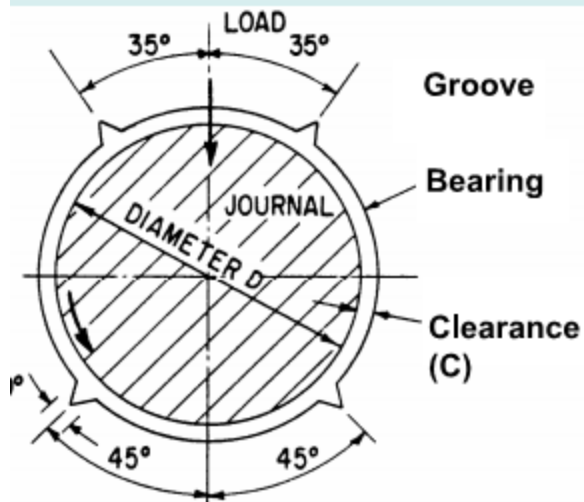
- Due to the structure of the bearing dynamic coefficients oil whip occurs at a rotational speed that is ~2 times the first natural frequency.



Plain Cylindrical Bearing

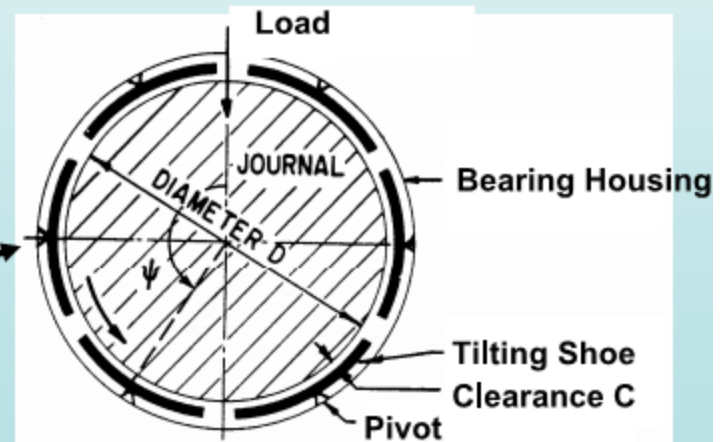


Elliptical Bearing



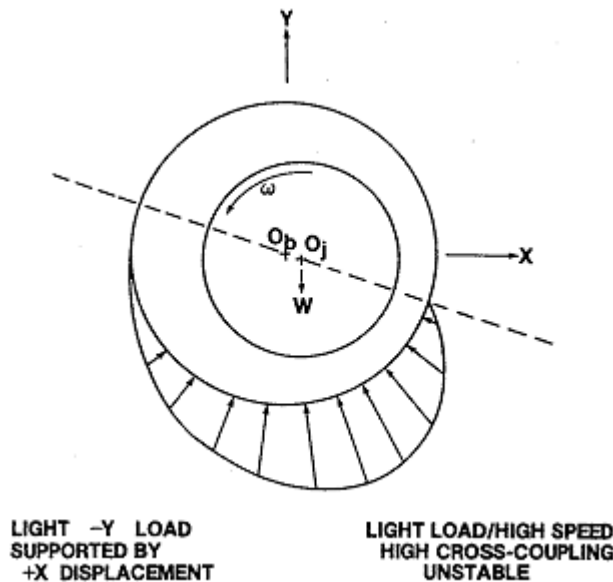
4-AXIAL GROOVE BEARING

Most Stable Bearing



Tilting Pad Bearing
Load Between Pads

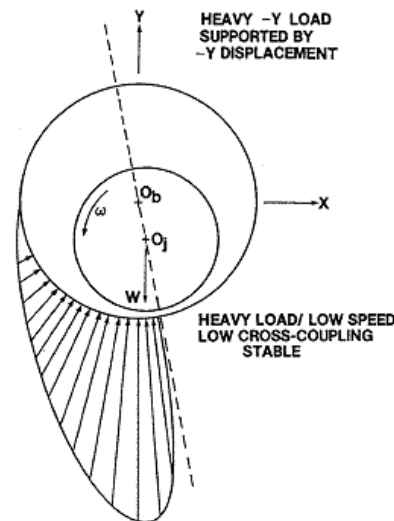
- Fixed lobe or sleeve bearings are becoming unstable at relatively high speed and/or light loads.
- The problem is that those bearings support a vertically downward load or force with a displacement that is not directly downward but at some angle with rotation from bottom dead center.



- A light $-Y$ direction load is supported by a $+X$ displacement. This occurs since the load is so light, the resulting pressure profile becomes very small with very little change from the maximum film to the minimum film location
- The summation of all vertical components of the hydrodynamic forces times the area must equal and opposite to the external load. Likewise the sum of all horizontal forces must be zero. This can only occur for attitude angles that approach 90°

- Since a downward load is supported by a horizontal displacement any downward force perturbation will result in a horizontal displacement which will result in a horizontal force and a vertical displacement, etc...
- Thus the bearing produces unstable cross coupling forces that actually drive the rotor and cause it to vibrate at frequency that is 50% or less of running speed

- Heavily loaded stable journal bearing is shown below
- The attitude angles is $\sim 0^\circ$ and eccentricity ratio is large
- In this manner a heavy $-Y$ direction load is supported by a $-Y$ displacement. This occurs since the load is so heavy, the resulting pressure profile becomes very large with very large gradient from the maximum film to the minimum film locations.



- $\sum F_x=0$ & $\sum F_y=Load$, this can only occur for attitude angles that approaches 0° . Since a downward load is now supported by a vertical displacement cross coupling forces are at minimum and the bearing is stable
- Usually bearing are designed not to go unstable until the rotor speed exceeds twice the rotor's first critical speed. Thus an exact $0.5X$ is a rare occurrence and bearing induced instabilities usually show up as shaft whip at frequencies less than 50% of synchronous speed

For reference:

[1] Robert C. Eisenmann, Sr., P.E & Robert C. Erisenmann, Jr « Machinery Malfunction Diagnosis and Correction », PTR Prentice Hall, Englewood Cliffs, New Jersey 07632, 1997

Take home message

- Still to be written

Thank You !



imagination at work