Hydrodynamic Bearing Basics

Hydrodynamic journal bearings operate by forming a thin film of oil that completely separates the shaft from the bearing surface at operating speeds. The clearance between the shaft and the bearing surface permits the formation of the thin film which the shaft rides on and as such allows the shaft center to move to one side, as shown in Figure 1.

The oil film thickness is a function mainly of radial load, shaft surface speed, and oil viscosity. The thickness of the oil film, among other characteristics, determines how the bearing will respond to rotating unbalance in the system at different speeds. This is important because the shaft center shown in Figure 1 is the theoretical equilibrium point at which the formed film pressure distribution equates to the applied radial load on the bearing. In reality, rotating assemblies are not perfectly balanced and thus residual imbalance exists which causes the shaft center to orbit around the equilibrium point. The orbit frequency, amplitude, and shape are a function of the bearing geometry, operating conditions and fluid-film characteristics, all of which determine the dynamic behavior of the rotor/bearing system.

Dynamic Coefficients

To predict bearing rotordynamic behavior, fans (as an example) are modelled using the Jeffcott rotor model, shown in Figure 2, which assumes an unbalanced, center hung mass on an elastic shaft and two bearings modelled as a spring-mass-damper system. Likewise, each hydrodynamic bearing can be modelled as a two-dimensional spring-
mass-damper system as shown in Figure 3. When analyzing bearing applications, hydrodynamic bearing software will generally produce 8 dynamic coefficients, as shown in Figure 4.

The $K$ values represent the spring stiffness of the fluid film, which generates a force as a result of the shaft center position movement away from the equilibrium point. The $C$ values represent the damping in the fluid film, which generates a force as a function of the shaft center’s velocity movement away from the equilibrium point. The subscripts of the stiffness or damping (i.e. $xx$, $xy$, $yx$, $yy$) represent displacement or velocity in a direction producing a stiffness or damping force in the same or orthogonal direction.

For both stiffness and damping, there are direct and cross-coupled coefficients (and respective forces). Direct coefficients have the same directional subscripts (i.e. $K_{xx}$, $C_{yy}$, etc.) and the resultant force produced is in the same direction to the motion. For example, if the shaft center moves purely in the $y$ direction, the resultant force generated by $K_{yy}$ will be in the $y$ direction. The effects of the direct forces are shown in Figure 5. Cross-coupled coefficients have different directional subscripts (i.e. $C_{yx}$, $K_{xy}$, etc.). The first subscript denotes displacement/velocity, the second subscript denotes the force direction, and the resultant force is orthogonal to the direction of motion. For example, if the shaft center moves purely in the $x$ direction, the resultant force generated by $K_{xy}$ would be in the $y$ direction. Direct stiffness, such as $K_{xx}$ or $K_{yy}$, produces a force opposite to the displacement, which can be viewed as pushing the shaft toward the equilibrium point, reducing the orbit size (and amplitude of vibration). Direct damping, such as $C_{xx}$ or $C_{yy}$, produces behavior similar to $K_{xx}$ or $K_{yy}$. On the other hand, cross-coupled stiffness, such as $K_{xy}$ or $K_{yx}$, produces a force that is orthogonal to shaft center displacement, thus inducing a force that will tend to move the shaft 90 degrees from its direction of movement. Similarly, cross-coupled damping forces will behave as that of cross-coupled stiffness. Cross-coupled coefficients induce destabilizing effects depending on their strength and sign (+/-) that may physically translate into inducing higher vibration in the rotating assembly. Cross-coupled stiffnesses with the same sign are stabilizing, while cross-coupled stiffnesses with opposite signs are destabilizing [1]. It is important to note that for a given rotor/bearing system, destabilizing forces increase with operating speed and decease with increased bearing applied load.

**Figure 4.** Example Bearing Analysis output showing Dynamic Coefficients.
A rotating system is considered “stable” when stabilizing forces, such as direct stiffness and damping, exceed the destabilizing forces, i.e. the cross-coupled forces [3]. One step in predicting system stability is to determine the system natural frequencies and critical speeds. Critical speeds are the speeds where the response to an unbalance is the greatest [1]. Rotating systems can have several critical speeds, but in low- to medium-speed applications, such as fans, the operating speed is usually well below the first and subsequent critical speeds. The operating speed of a fan should not be too close to a critical speed, or the fan vibration will be unacceptably high. The most common recommendation is that the operating speed should not be within 20% above or below a critical speed.

**Figure 5.** Forces caused by A) Direct Stiffness and B) Direct Damping and their relation to shaft orbit [2].
Effects of Bore Profile on Dynamic Coefficients

If a fan is experiencing high vibration as a result of approaching a critical speed, there are three possible options to remedy the problem: (1) change the operating speed, (2) change the load, or (3) change the dynamic properties of the bearings. Often, application requirements make options (1) and (2) unfeasible. Option (3), however, can be implemented at the bearing level on hydrodynamic bearings.

The easiest way to alter dynamic properties is to adjust the clearance between the shaft and bearing. In general, increasing the clearance lowers the operating temperature of the bearing, but also lowers the stiffness and damping coefficients.

Another way to alter the dynamic properties of the system is to change the bearing bore profile. The advantage of adjusting the bearing profile, rather than simply change the clearance, is that it allows the bearing performance to be more closely tuned to the application needs.

The two most common bore profiles of hydrodynamic bearings in fans are elliptical and cylindrical, as shown in Figure 6. Notice that the two figures have the same vertical clearance, but different horizontal clearance values. Using the elliptical profile allows the bearing to retain similar, if not higher, stiffness and damping to the cylindrical profile in the vertical direction, but the lower stiffness and damping in the horizontal direction allow for more shaft movement, as shown in the example bearing selection shown in Figure 7. However, the lower stiffness results in lower first critical speeds. Depending on the fan operating speed, this could cause higher levels of vibration than desired. If this is the case, using a cylindrical bore instead may raise the first critical speed enough to substantially lower the vibration in the fan at operating speed.

Figure 6. Common journal bearing profiles for fan applications are A) elliptical and B) cylindrical.
Conclusion

Unlike rolling element bearings, hydrodynamic bearings produce both stiffness and damping forces, which provide significant flexibility for machine designers. When selecting hydrodynamic bearings for an application, it is critical to consider the dynamic effects to ensure that the system operates as desired. In some applications, the bearing load, shaft speed, and oil viscosity can be easily changed to fine tune the dynamic performance. However, in tightly constrained or existing applications, it may be preferable to alter the bearing bore profile to find the best balance between the dynamic properties and the critical speeds of the system.

References:


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