Hydrodynamic Bearings and Seals

Hydrodynamic bearings, seals, and squeeze-film dampers constitute a class of devices that involve the flow in an annulus between two cylinders; the inner cylinder is generally the shaft (radius, R) which is rotating at a frequency, Ω, and may also be whirling with an amplitude or eccentricity, ϵ, and a frequency, ω. The outer cylinder is generally static and fixed to the support structure. The mean clearance (width of the annulus) will be denoted by δ, and the axial length by L. In both hydrodynamic bearings and seals, the basic fluid motion is caused by the rotation of the shaft. In a seal, there is an additional axial flow due to the imposed axial pressure difference. In a squeeze-film damper, there is no rotational motion, but forces are generated by the whirl motion of the "rotor".

The Reynolds number is an important parameter in these flows, and it is useful to evaluate three different Reynolds numbers based on the rotational velocity, on the mean axial velocity, V (given by $V = Q/2\pi R\delta$ where Q is the volumetric axial flow rate), and on the velocity associated with the whirl motion. These are termed the rotational, axial and whirl Reynolds numbers and are defined, respectively, by

$$Re_\Omega = \Omega R\delta/\nu, \quad Re_V = V\delta/\nu, \quad Re_\omega = \omega R\delta/\nu \qquad \text{(Mcc1)}$$

where ν is the kinematic viscosity of the fluid in the annulus. In a hydrodynamic bearing, the fluid must be of sufficiently high viscosity so that $Re_\Omega \ll 1$. This is because the bearing depends for its operation on a large fluid restoring force or stiffness occurring when the shaft or rotor is displaced from a concentric position. Typically a bearing will run with a mean eccentricity that produces the fluid forces that counteract the rotor weight or other radial forces. It is important to recognize that the fluid only yields such a restoring force or stiffness when the flow in the annulus is dominated by viscous effects. For this to be the case, it is necessary that $Re_\Omega \ll 1$. If this is not the case, and $Re_\Omega \gg 1$ then, as we shall discuss later, the sign of the fluid force is reversed, and, instead of tending to decrease eccentricity, the fluid force tends to magnify it. This is called the "Bernoulli effect" or "inertia effect", and can be simply

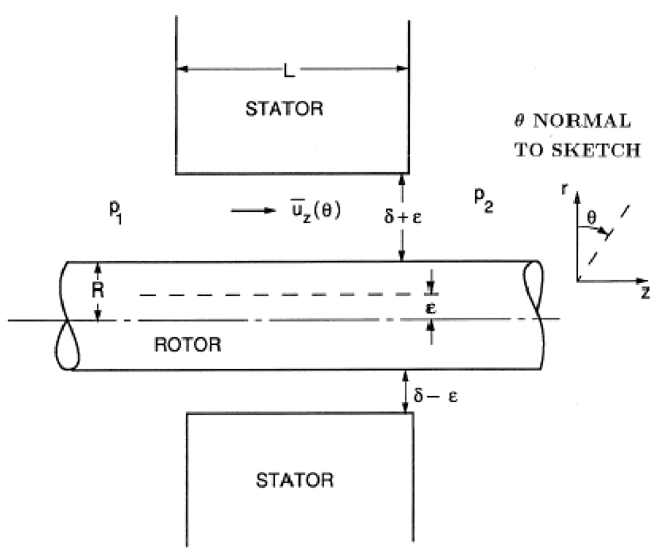

Figure 1: Schematic of a short seal demonstrating the Lomakin effect.

explained as follows. When an eccentricity is introduced, the fluid velocities will be increased over that part of the rotor circumference where the clearance has been reduced. At Reynolds numbers much larger than unity, the Bernoulli equation is applicable, and higher velocities imply lower pressure. Therefore the pressure in the fluid is decreased where the clearance is small and, consequently, there will be a net force on the rotor in the direction of the displacement. This "negative stiffness" ($K < 0$) is important in the rotordynamics of seals and impellers.

Another parameter of importance is the ratio of the axial length to radius, L/R, of the bearing or seal. For large L/R, the predominant fluid motions caused by the rotordynamic perturbations occur in the circumferential direction. On the other hand, in a short seal or bearing, the predominant effect of the rotordynamic perturbation is to cause circumferential variation in the axial fluid velocity. This gives rise to the so-called "Lomakin effect" in short seals operating at high Reynolds numbers (Lomakin 1958). The circumstances are sketched in figure 1, in which we use a cylindrical coordinate system, (r, θ, z), to depict a plain annular seal with a clearance, δ. The fluid velocity, u_z, is caused by the pressure difference, $\Delta p = (p_1 - p_2)$. We denote the axial velocity averaged over the clearance by $\bar{u}_z$, and this will be a function of θ when the rotor is displaced by an eccentricity, ϵ. The Lomakin effect is caused by circumferential variations in the entrance losses in this flow. On the side with the smaller clearance, the entrance losses are smaller because $\bar{u}_z$ is smaller. Consequently, the mean pressure is larger on the side with the smaller clearance, and the result is a restoring force due to this circumferential pressure distribution. This is known as the Lomakin effect, and gives rise to a positive fluid-induced stiffness, K. Note that the competing Bernoulli and Lomakin effects can cause the sign of the fluid-induced stiffness of a seal to change as the geometry changes.

In the following sections we examine more closely some of the fluid-induced rotordynamic effects in bearings, seals, and impellers.