Squeeze Film Dampers

A squeeze film damper consists of a nonrotating cylinder surrounded by a fluid annulus contained by an

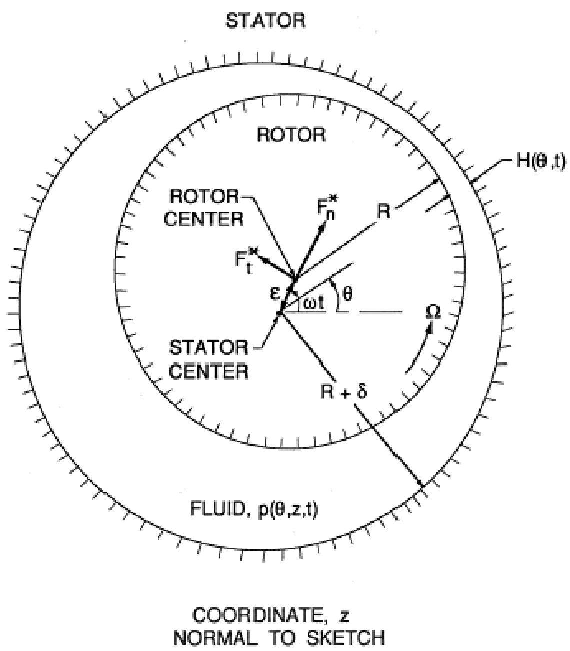

Figure 1: Schematic of fluid-filled annulus between a stator and a rotating and whirling rotor.

outer cylinder. A shaft runs within the inner, nonrotating cylinder so that the latter may perform whirl motions without rotation. The fluid annulus is intended to damp any rotordynamic motions of the shaft. It follows that the previously utilized figure 1 can also represent a squeeze film damper as long as Ω is set to zero. The device is intended to operate at low Reynolds numbers, Re_ω, and several of the results already described can be readily adopted for use in a squeeze-film damper. Clearly analyses can be generated for both long and short squeeze film dampers. The long squeeze film damper is one flow for which approximate solutions to the full Navier-Stokes equations can be found (Brennen 1976). Two sets of asymptotic results emerge, depending on whether Re_ω is much less than, or much greater than, $72R/\delta$. In the case of thin films ($\delta \ll R$), the rotordynamic forces for $Re_\omega \ll 72R/\delta$ are

$$F_n^* = 6\pi\rho R^3 L\omega^2\epsilon/5\delta \qquad \text{(Mcf1)}$$

$$F_t^* = 12\pi\mu R^3 L\omega\epsilon/\delta^3 \qquad \text{(Mcf2)}$$

where the F_t^* is the same as that for a noncavitating long bearing. On the other hand, for $Re_\omega \gg 72R/\delta$:

$$F_n^* = \frac{\pi\rho R^3 L\omega^2\epsilon}{\delta}\left[1 + \left(\frac{2\nu}{\omega\delta^2}\right)^{\frac{1}{2}}\right] \qquad \text{(Mcf3)}$$

$$F_t^* = \pi\rho R^2 L\omega^2\epsilon\left(\frac{2\nu}{\omega\delta^2}\right)^{\frac{1}{2}} \qquad \text{(Mcf4)}$$

The $\omega^{\frac{3}{2}}$ dependent terms in these relations are very unfamiliar to rotordynamicists. However, such frequency dependence is common in flows that are dominated by the diffusion of vorticity.

The relations (Mcf1) to (Mcf4) are limited to small amplitudes, $\epsilon \ll \delta$, and to values of $\omega\epsilon^2/\nu \ll 1$. At larger amplitudes and Reynolds numbers, $\omega\epsilon^2/\nu$, it is necessary to resort to lubrication analyses supplemented, where necessary, with inertial terms in the same manner as described in the last section. Vance (1988) delineates such an approach to squeeze film dampers.