NOTES 12(b)

HYDROSTATIC JOURNAL BEARINGS

Summary

In a hydrostatic bearing an external source of pressurized fluid forces lubricant between two surfaces; thus enabling non-contacting operation and the ability to support a load. Hydrostatic bearings can support large loads without journal rotation and provide large (accurate and controllable) direct stiffness as well as damping (energy dissipation) coefficients.

Hydrostatic bearings rely on external fluid pressurization to generate load support and a large centering stiffness, even in the absence of journal rotation. The load capacity and direct stiffness of hydrostatic bearings do not depend on fluid viscosity, thus making them ideal rotor support elements in process fluid pumps. Current applications intend to replace oil lubricated bearing with hybrid bearings to improve efficiency with shorten rotor spans and less mechanical complexity. Current cryogenic liquid turbopumps implement hydrostatic bearings enabling an all fluid film bearing technology with very low number of parts and no DN limit operation. Details on the bulk-flow analysis of turbulent flow hydrostatic bearings are given along with the discussion of performance characteristics, static and dynamic, for hydrostatic bearings supporting a water pump. Angled liquid injection produces a hydrostatic bearing with unsurpassed dynamic force and stability characteristics.

Introduction

Hydrostatic bearings derive their load capacity not from shear flow driven effects (hydrodynamic wedge and surface sliding) but rather from the combination of pressure versus flow resistance effects through a feed restrictor and in the film lands. Figure 1 depicts thrust and radial hydrostatic bearing configurations for process fluid lubrication turbopumps. Table 1 presents the major advantages and disadvantages of hydrostatic bearings.

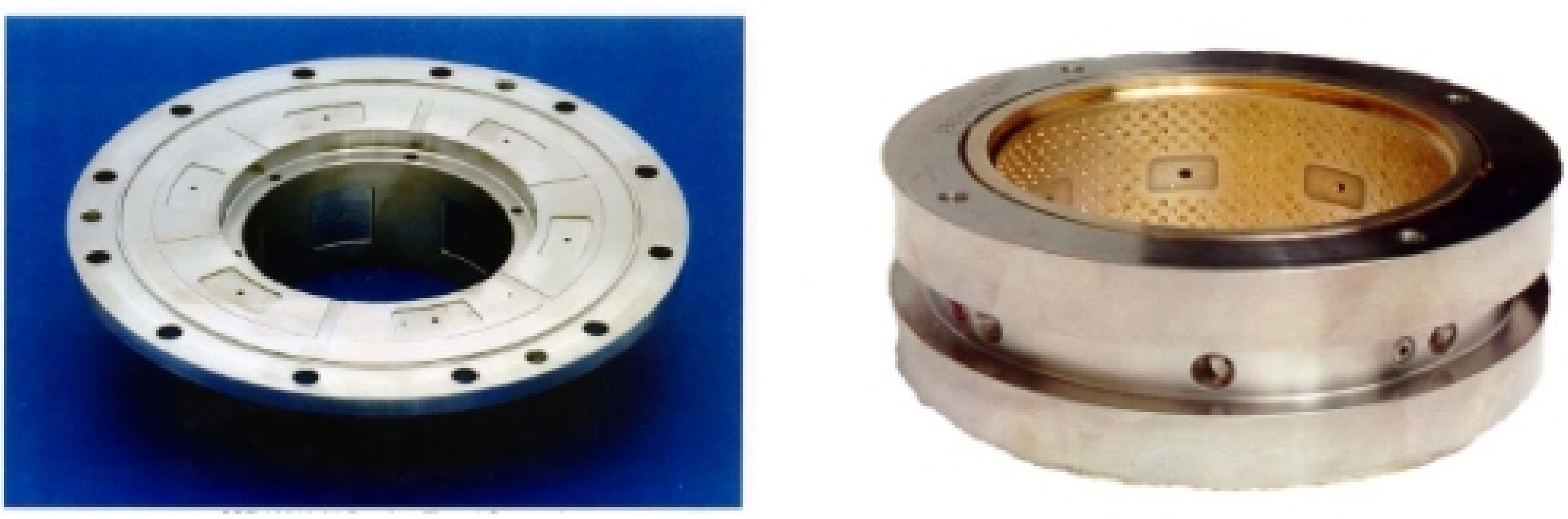

Fig 1. Hydrostatic radial and thrust bearings for process fluid rotating machinery

The hydrostatic stiffness is of unique importance for the centering of high-precision milling machines, gyroscopes, large arena movable seating areas, telescope bearings, and even cryogenic fluid turbo pumps for rocket engines. Note that hydrostatic bearings require an external pressurized supply system and some type of flow restrictor. Also, under dynamic motions, hydrostatic bearings may display a pneumatic hammer effect due to fluid compressibility. However, and most importantly, the load and static stiffness of a hydrostatic bearing are independent of fluid viscosity; thus making this bearing type very attractive for application with non-viscous fluids, including gases and cryogens.

Table 1. Hydrostatic Bearings: Advantages and Disadvantages

Advantages	**Disadvantages**
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.	Require ancillary equipment. Larger installation and maintenance costs. Need of fluid filtration equipment. Loss of performance with fluid contamination. High power consumption because of pumping losses. Potential to induce hydrodynamic instability in hybrid mode operation. Potential to show pneumatic hammer instability for highly compressible fluids, i.e. loss of damping at low and high frequencies of operation due to compliance and time lag of trapped fluid volumes.

Consider the fundamental operation of a simple one dimensional hydrostatic bearing [Rowe 1983, San Andrés 2002]. The flow is laminar and fluid inertia effects are not accounted for; i.e. a classical lubrication example. Figure 2 depicts a 1D bearing of very large width (B). A hydrostatic bearing combines two flow restrictions in series, one at the feed or supply port, and the other through the film lands. In the feed restrictor (orifice, capillary, etc.) the fluid drops its pressure from the supply value (P_S) to a magnitude (P_R) within a recess or pocket of typically large volume (see Figure 3). Since the recess is deep, the pocket pressure is regarded as uniform over the entire recess area A_R=bB. The fluid then flows from the recess into the film lands of small thickness h, and discharges to ambient pressure through the bearing sides, say P_a=0 for simplicity.

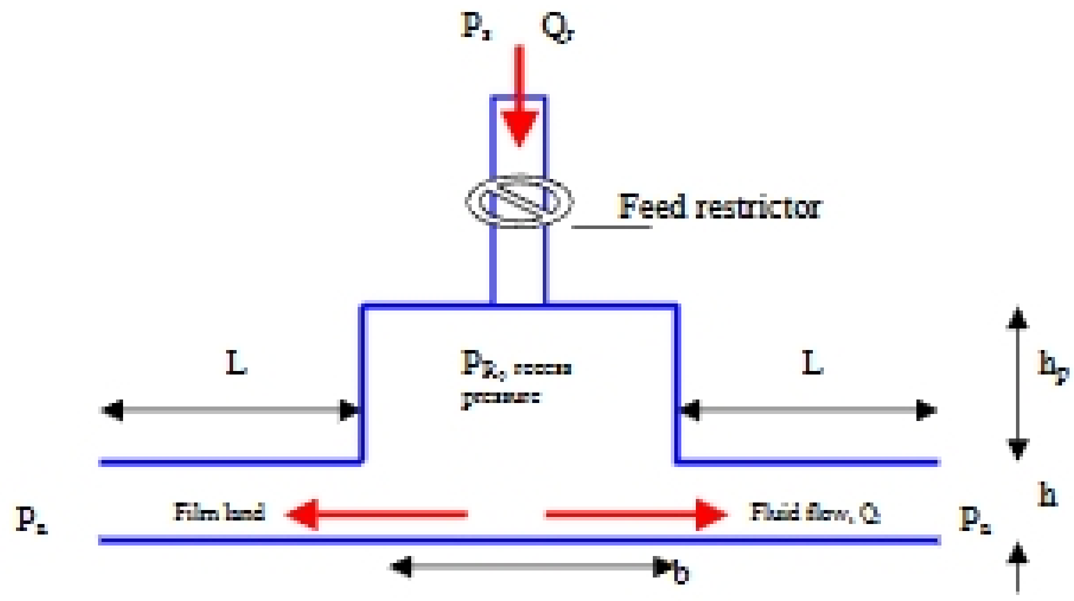

Fig. 2 Geometry of a simple 1-D hydrostatic bearing

The flow rate (Q_r) across the restrictor is a function of the pressure drop, $Q_r=f(P_S-P_R)$. For an orifice and capillary feeding,

$$Q_r = Q_o = A_o C_d \sqrt{\frac{2}{\rho}(P_s - P_R)} \;;\quad Q_r = Q_c = \frac{\pi d^4}{128 \mu \ell_c}(P_s - P_R) \tag{1}$$

with A_o and C_d as the orifice area and empirical discharge coefficient, respectively. (d, ℓ_c) are the diameter and length of the capillary tube, typically ℓ_c » 20 d. The orifice coefficient (C_d) ranges from 0.6 to 1.0, depending on the flow condition (Reynolds number), the orifice geometry and even the film thickness. Under turbulent flow conditions, tests and CFD analysis evidence C_d ~0.80.

Across the bearing film lands the fluid drops in pressure from (P_R) to ambient pressure, P_a. In the laminar flow of an incompressible fluid, the flow rate is a function of the pressure drop and equals

$$Q_\ell = -\frac{B h^3}{12\mu}\frac{\partial P}{\partial x} = +\frac{B h^3 (P_R - P_a)}{12 \mu L} \tag{2}$$

where B is the bearing width and L is the film length with thickness h. Presently, no surface motion along the x-axis is accounted for, i.e. the bearing is stationary. Under steady state conditions, the flow through the restrictor equals the flow through the film lands, i.e.

$$Q_r = f(P_s - P_R) = 2\,C_l (P_R - P_a) = 2\,Q_l \tag{3}$$

with $C_l = B\,h^3/(12\,\mu\,L)$ as a flow-conductance along the film land. Eqn. (3) permits the determination of the recess pressure (P_R) given the film conductance (C_l) and feed restrictor parameters. For bearing design, a value of pocket pressure (P_R) is desired, and Eqn. (3) serves to size the diameter of the supply restrictor.

For the simple bearing considered, the pressure field on the bearing surface takes the shape shown in Figure 3. Note that the recess pressure is assumed uniform or constant