HYDROSTATIC LUBRICATION IN TURBOPUMPS OF ROCKET ENGINES

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Abstract

The paper presents a computational procedure that can be useful in assessing the main performance parameters (load, flow rate, attitude angle) of hydrostatic journal bearings, without resorting to full numerical computing. The procedure is based on the "lumped resistance method„ which has been extended to take into account turbulence and inertia effects. Comparison with available experimental data shows a pretty reasonable agreement, in spite of the heavy approximations that have been introduced.

Introduction

Cryogenic fluid turbopumps of rocket engines used in space transport systems need special bearings. Advanced rolling bearings for cryogenic environments are very costly and present serious limitations because of the high DN values and of the lack of conventional lubrication. Hydrostatic bearings very often represent an attractive alternative to rolling bearings: their main drawback, the need of an external high-pressure supply system, is not a problem in turbopumps, which are themselves a source of fluid at a rather high pressure, the lost work being compensated by the very low friction losses. These bearings have typically a very high stiffness, which ensures accurate positioning and subcritical rotor operation. On the other hand, an important disadvantage of this intrinsic supply system is that during start-stop operations the bearing is badly lubricated until the pressure of the pumped fluid is high enough: hence the bearing parts must be built of materials whit suitable tribological properties.

Hydrostatic bearings have been used for decades in conventional machinery and hence a wide literature exists about their design [1]. In usual applications (oil lubrication and low Reynolds number) a first assessment of their behaviour can be easily performed, since a number of design charts are available (usually obtained solving the appropriate Reynolds equation with numerical methods); approximate formulas and simplified computing methods are also available. Unfortunately, most of this works may prove to be useless in cryogenic applications: due to low viscosity and high speed, these bearings are likely to work in a turbulent regime. Moreover, also inertia effects may be not negligible.

A number of algorithms have been developed which are aimed to the full numerical solution of problems of this kind, incorporating turbulence and inertia effects and also taking into account the flow pattern in the inlet recesses (e.g. see [2]). The aim of the present work is, instead, the development of a numerical procedure able to give a reasonably approximate assessment of the bearing performance, but simple enough to be implemented within a computing ambient (such as "*M W Matlab*„) or even with a spreadsheet program (such as "*MS Excel*„) with a limited programming effort.

The work is framed in a research activity promoted by the Italian Space Agency (ASI) in the FAST2 (Future Advanced Space Transportation Technologies - Propulsion Technologies) Contract, where FiatAvio is Prime Contractor, and is aimed to appraise the replacement of ball bearings with hydrostatic bearings in liquid oxygen turbopumps for rocket engines.

Analysis

A hydrostatic journal bearing is schematically shown in Fig. 1. A number of recesses are hollowed out in the inner surface of the bush: the lubricating fluid is fed into each recess, from a constant-pressure source, through a compensating restrictor and then flows out of the bearing passing through the narrow clearance between the land surfaces of journal and bush.

In conventional applications the bearing is fed with oil and the high value of viscosity ensures laminar flow, up to high values of rotational speed: in this case the pressure field in the film can be obtained by means of a numerical solution of the Reynolds equation. On the contrary, when low-viscosity fluids must be used the flow is likely to become turbulent, even for $\Omega=0$. A more comprehensive model is therefore needed.

Returning to the Navier-Stokes equations for a turbulent thin film and disregarding the inertia terms, after integration one may express the bulk-flow speed components in the following form

$$v_x = \frac{\Omega D}{4} - G_x \frac{h^2}{\mu}\frac{\partial p}{\partial x} \quad , \qquad v_z = -G_z \frac{h^2}{\mu}\frac{\partial p}{\partial z} \qquad (1)$$

where "pressure„ p is intended as the stochastic average of the actual local pressure which in facts, due to turbulence, fluctuates in time around p. Similarly, v_x and v_z are the mean values across the film of the stochastic averages of the fluid velocity components. The turbulent coefficients G_x and G_z depend on the fluid velocity field and are smaller than 1/12, which is the value they take in the case of laminar flow.

A few turbulence models, based on theoretical as well as on experimental considerations, have been developed that permit to evaluate the turbulent coefficients, in some case also taking into account the effect of the surface roughness. Following the bulk flow the-

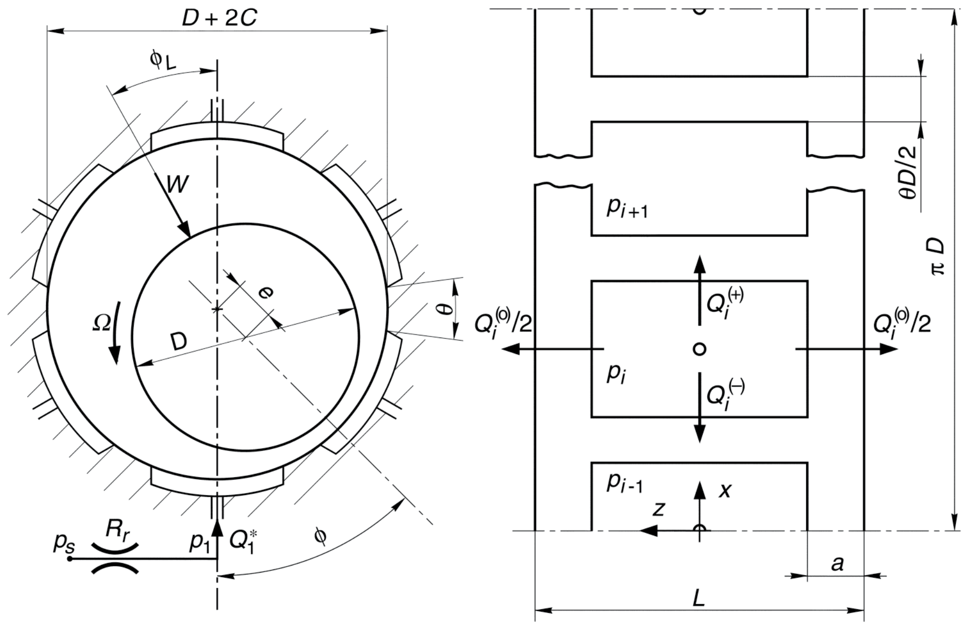

Fig. 1: Geometry of a hydrostatic multirecess journal bearing.

ory of Hirs [3], the pressure gradient depends on the bulk flow velocity

$$\begin{aligned} \frac{\partial p}{\partial x} &= -\frac{\mu}{2h^2}\left[\left(k_J + k_B\right)v_x - k_J\,\frac{\Omega D}{2}\right] \\ \frac{\partial p}{\partial z} &= -\frac{\mu}{2h^2}\left(k_J + k_B\right)v_z \end{aligned} \tag{2}$$

The shear parameters k_J and k_B depend on local Reynolds numbers and hence on flow, again. Disregarding the effect of surface roughness, the following formulae can be used

$$k_J = 0.066\,Re_J^{0.75} \qquad k_B = 0.066\,Re_B^{0.75} \tag{3}$$

$$Re_J = \frac{\rho}{\mu}h\sqrt{\left(v_x - \Omega D/2\right)^2 + v_z^2} \quad Re_B = \frac{\rho}{\mu}h\sqrt{v_x^2 + v_z^2} \tag{4}$$

Comparing equations (1) and (2) it is readily obtained that

$$\begin{aligned} G_x &= \min\left[\frac{1}{12}\ ;\ \frac{2v_x - \Omega D/2}{\left(k_J + k_B\right)v_x - k_J\,\Omega D/2}\right] \\ G_z &= \min\left[\frac{1}{12}\ ;\ \frac{2}{k_J + k_B}\right] \end{aligned} \tag{5}$$

When lubricant speed becomes very high, effect of inertia forces may become not negligible in the lubricant film as well as in the recess. However, a full inertial solution would require a simultaneous solution of Navier-Stokes and continuity equations (see for instance [2]) and is clearly beyond the purpose of the present work. Hence, the only inertia effect considered in what follows is the pressure drop at the recess edges.

Since the recess depth is much greater than the film thickness, the fluid velocity in the film is much greater than in the recess; when the lubricant crosses the recess boundaries, it experiences a sudden acceleration that must be compensated by a sudden pressure drop, for energy conservation. The problem is studied in detail in [4], where it is shown that the pressure drop across the recess edges parallel to the bearing axis depends on flow rate, film thickness, recess depth, journal speed and rotational Reynolds number. However, as a first approximation it may be taken simply

$$\Delta p = 0.5\,\rho\,v_x^2 \tag{6}$$

(only when flow rate actually leaves the recess to enter the bearing lands). A similar equation holds for the side edges.

Lumped resistance method

In steady conditions, the net flow rate leaving the boundary of each recess must be equal to the flow rate supplied through the compensating restrictor, that is

$$Q_i^* = Q_i^{(0)} + Q_i^{(+)} + Q_i^{(-)} \tag{7}$$

where Q_i^* is the supplied flow rate, $Q_i^{(0)}$ is the flow rate crossing the lateral boundaries of the recess; $Q_i^{(+)}$ and $Q_i^{(-)}$, which can be negative, are the flow rates leaving the recess toward the following and the preceding recesses, respectively.

When all the geometrical parameters, the journal speed and the lubricant properties are fixed, these flow rates depend only on the recess pressures P_i. The method consists in obtaining an approximate expression of each flow rate as a function of the recess pressures; the set of n equations (7) is then solved to obtain the recess pressures, from which load capacity and total flow rate can be readily evaluated.

Supply flow rate

If the compensating devices were laminar-flow restrictors (e.g. capillary pipes), their hydraulic resistance $R_r=(p_s-p_i)/q_i^*$ would be constant. With low-viscosity fluids laminar flow cannot be obtained in practical applications and the flow rate-pressure relationship is better represented by the following law

$$q_i^* = A_0 C_d \sqrt{2(p_s - p_i)/\rho} \tag{8}$$

where A_0 is the section area of the restrictor (in general an orifice) and C_d an empirical discharge coefficient. For “true„ sharp-edge orifices, it can be taken $C_d \approx 0.6$. More generally, C_d depends on the Reynolds number (that is on the flow rate itself) and on the geometry of the orifice and of the preceding and following ducts. Commonly it is in the range 0.6÷0.85.

It is convenient to substitute equation (8) with a linear approximation, obtaining (in dimensionless form)

$$Q_i^* = \kappa_0 \sqrt{1-\overline{P}_i} - \kappa_0 \frac{P_i - \overline{P}_i}{2\sqrt{1-\overline{P}_i}} \tag{9}$$

where the $\overline{P}_i$ constitute a set of trial values that must be updated in the course of iterations. The dimensionless orifice parameter

$$\kappa_0 = A_0 C_d \frac{\mu}{C^3} \sqrt{\frac{2}{\rho p_s}} \tag{10}$$

is considered here as an input data though, strictly speaking, it should vary slightly for each recess when the relevant flow rate changes.

Inter-recess lands

Most part of the flow rate $Q_i^{(+)}$ flows into the next recess i+1, crossing the relevant inter-recess land. Over this land the bulk velocity component in the axial direction may hence be disregarded and the land can be treated like an infinite-length slide.

Under this assumption v_x is constant on the whole recess edge and the flow rate

$$q_i^{(+)} = v_x h(L-2a) \tag{11}$$

remains constant for all the land width θ. If v_x is eliminated between equations (1) and (11), the pressure gradient is obtained. Integration over the whole land finally leads to obtain the flow rates exchanged with the neighbouring recesses

$$Q_i^{(+)} = \kappa_2 \frac{J_i^{(2)}}{J_i^{(3)}} + \kappa_3 \frac{1}{J_i^{(3)}} (P_i - P_{i+1} - \delta_i) \qquad Q_i^{(-)} = -Q_{i-1}^{(+)} \tag{12}$$

(in dimensionless form), where

$$J_i^{(2)} = \int_{\psi_i^+}^{\psi_i^+ + \theta} \frac{d\psi}{G_x H^2} \quad , \quad J_i^{(3)} = \int_{\psi_i^+}^{\psi_i^+ + \theta} \frac{d\psi}{G_x H^3} \tag{13}$$

and δ_i is the pressure drop due to the edge effect:

$$\delta_i = \pm \frac{\rho p_s}{2} \left[\frac{C^2}{D\mu\lambda(1-2a')} \right]^2 \left(\frac{Q_i^{(+)}}{H} \right)^2 \tag{14}$$

When the flow rate is positive (directed from recess i to recess i+1) δ_i is positive, too, and must be evaluated for $\psi = \psi_i^+$; in the opposite case, δ_i is negative and the film thickness is calculated at $\psi = \psi_i^+ + \theta$.

Integrals (13) can be calculated by means of a 3-points Gauss method, although for the simplest approach they can be substituted with $\theta/(G_x H^2)$ and $\theta/(G_x H^3)$, evaluated at the centre of the land.

Axial lands

In the centred configuration (e=0) the pressure on the side lands is almost constant in the circumferential direction (especially when θ is small) and varies linearly along the axis. From equations (1) the bulk-flow velocity components are

$$v_x = \frac{\Omega D}{4} \quad , \quad v_z = \pm G_z \frac{h^2}{\mu} \frac{p_i - \delta_{0i}}{a} \tag{15}$$

where δ_{0i} indicates the inertia pressure drop at the side edges. If now it is assumed that equations (15) remain approximately valid for e>0, at least for moderate eccentricities, the side flow rates from each recess can be calculated:

$$Q_i^{(0)} = \kappa_1 J_i^{(1)} (P_i - \delta_{0i}) \tag{16}$$

$$J_i^{(1)} = \int_{\psi_i - \pi/n}^{\psi_i + \pi/n} G_z H^3 d\psi \tag{17}$$

The pressure drop depends on the bulk velocity and, hence, it is not generally constant along the side edge of the recess; however, for the sake of simplicity, it can be substituted by a mean value for each recess:

$$\delta_{0i} = \frac{\rho p_s}{8} \left(\frac{nC^2}{\pi D \mu} \right)^2 \left[\frac{Q_i^{(0)}}{H(\psi_i)} \right]^2 \tag{18}$$

Again, further simplification can be introduced substituting equation (17) with

$$J_i^{(1)} = \frac{2\pi}{n} G_z H^3 \Big|_{\psi = \psi_i}$$

Iterative solution

When equations (9), (16), and (12) are substituted into equations (7), a set of n equations is obtained that can be solved yielding the recess pressures. Since most coefficients depend on the flow rates (and, hence, on the recess pressures), an iterative solution is needed. Using a simple Eulerian approach, solution procedure may be the following:

- set trial values for recess pressures, for G_x and G_z (in all the integration points of each recess), for the pressure drops δ_{0i} and δ_i.;
- calculate the coefficients J from equations (13) and (17); assemble and invert the system matrix; obtain new values for recess pressures;
- update the pressure values (with a under-relaxation technique) and the other trial parameters.

Procedure must be repeated until a satisfactory convergence is achieved.

Bearing performance parameters

Total flow rate is readily obtained by summing the side flow rates $Q_i^{(0)}$. Load capacity should be obtained by integration of the whole pressure field. For the sake of simplicity, to each recess is attributed a force vector $\mathbf{w}_i$ directed toward the recess centre:

$$w_i = DL p_i \left[(\xi - 1) a' \sin\left(\frac{\pi}{n} - \frac{\theta}{2} \right) + 2 \frac{1 - \xi a'}{\theta} \sin\frac{\pi}{n} \sin\frac{\theta}{2} \right]$$

For parameter ξ, a value ξ=1.5 can be used. Otherwise, it can be used as a correction factor: for instance, a value ξ=0.92+2.7a/D seems to produce improved results (at least for laminar-flow bearings). Composition of all these vectors gives the external force that balances the fluid force on the journal in case of steady loading.

Sample results

The algorithm described above has been implemented in a *"MS Excel"* spreadsheet. Program has shown to converge satisfactorily except than in case of large eccentricity and high speed, where cavitation phenomena are likely to occur. Convergence, for errors on recess pressures smaller than $10^{-4}p_s$, is usually obtained into 10÷40 iterations (strongly influenced by the algorithm that choices the initial trial values); each iteration requires less than 1 s and, of course, mostly depends on the computer employed.

In order to test the program, a comparison has been made with a number of experimental results reported in [5] for a bearing with n=5, D=L=76.2 mm, a=24.6 mm, θ=0.548 rad, lubricated with water (μ=0.429 mPas, ρ=986.3 Kg/m^3). For 5 cases with different values of supply pressure and speed, flow rate and pressure ratio p_i/p_s have been calculated in the centred position e=0; they are reported in Table 1 and compared with experimental values. Maximum errors found are about 15% for pressure ratio and about 10% for flow rate.

For 3 cases also the load has been calculated for a number of eccentricity values. Results are shown in Fig. 2. Agreement with experimental data is still reasonably good.

Table 1: test results in centred configuration

case	*1*	*2*	*3*	*4*	*5*
Speed [Krpm]	10.2	10.2	17.4	24.6	24.6
Supply pressure [MPa]	4.133	6.877	6.846	4.135	6.844
Clearance [μm]	122.8	124.9	122.7	119.4	117.1
Orifice parameter κ_0	0.0201	0.0140	0.0162	0.0283	0.0208
Pressure ratio	0.290	0.239	0.311	0.495	0.414
ref. [5]	*0.273*	*0.209*	*0.338*	*0.586*	*0.468*
Flow rate [l/m']	79.0	99.3	103.8	86.2	106.5
ref. [5]	*79.9*	*102.3*	*101.8*	*78.1*	*101.4*

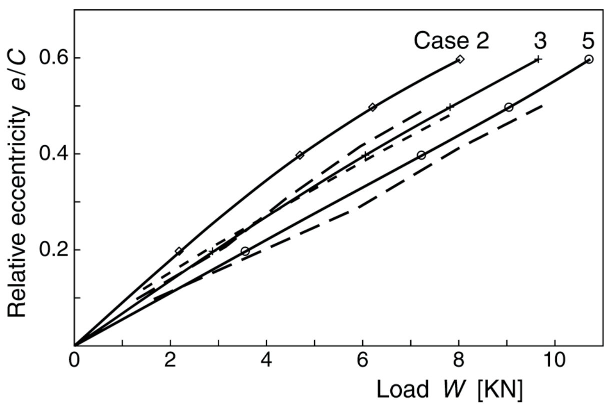

Fig. 2: Eccentricity versus load for the test bearing (dotted lines refer to results from [5]).

Conclusions

A procedure has been presented that can be used to forecast the performance of hydrostatic journal bearings, operating in turbulent flow regime. The procedure is based on the "lumped resistance method„ which has been extended to take into account turbulence and inertia edge effect, and proved to be simple enough to be implemented with the aid of a commercial spreadsheet program, with a limited macro-programming effort.

Comparison with available experimental data shows a pretty reasonable agreement, in spite of the heavy approximations on which the algorithm is based.

The procedure has been developed as an aid for designing an experimental cryogenic test rig for hydrostatic bearings, which is currently being set up.

Nomenclature

$a' = a/L$
e = journal eccentricity
$H = h/C$ dimensionless film thickness
$h = C - e\cos(\psi - \phi)$ thickness of the lubricant film
n = number of recesses
$P = p/p_s$ dimensionless pressure
P_i = dimensionless recess pressure
p = relative pressure
p_s = relative supply pressure
$Q = q\mu/(p_s C^3)$ dimensionless flow rate
q = flow rate
$\kappa_1 = 1/(\lambda a')$
$\kappa_2 = \lambda(1-2a')(\Omega\mu/p_s)/(2C/D)^2$
$\kappa_3 = 2\lambda(1-2a')$
$\lambda = L/D$
μ = lubricant viscosity
ρ = lubricant density
$\psi = 2x/D$
$\psi_i = 2(i-1)\pi/n$
$\psi_i^+ = \psi_i + (\pi/n - \theta/2)$
Ω = rotational speed of journal

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