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Interim Report

I-A2049-2

**THE INFLUENCE OF THE MOLECULAR MEAN FREE PATH
ON THE PERFORMANCE OF HYDRODYNAMIC GAS LUBRICATED BEARINGS**

by
Albert Burgdorfer

June 1958

Prepared under
Contract Nonr-2342(00)
Task NR 097-343

Jointly supported by
DEPARTMENT OF DEFENSE
✓ ATOMIC ENERGY COMMISSION
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FOREWORD

This report, I-A2049-2, describes one item of investigation from the general research program being conducted at The Franklin Institute Laboratories on the fundamental aspects of gas lubricated bearings.

NOMENCLATURE

x, y, z = cartesian coordinates

h = film thickness, in.

h_1, h_2 = film thickness of inclined plane slider bearing at inlet and outlet of slider, in.

l = slider length, in.

r = radius of journal, in.

ϕ = angle

u, v = flow velocity of lubricant in x, y -direction, in./sec

u_r = velocity of runner, in./sec

p = pressure (absolute), lb/in.²

ρ = mass density, lb sec²/in.⁴

μ = absolute viscosity, lb sec/in.²

T = temperature, °deg. abs.

γ = ratio of specific heats (c_p/c_v)

τ = shear stress, lb/in.²

F = friction force on runner, lb

W = load carrying capacity, lb

σ = numerical constant

α = thermal accomodation coefficient

f = reflection coefficient

λ = molecular mean free path, in.

n = number of molecules per in.³

s = mutual collision cross section, in.²

Pr = Prandtl number

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$$X = \frac{x}{l} = \text{dimensionless length}$$

$$Y = \frac{y}{l} = \text{dimensionless width}$$

$$P = \frac{p}{p_{bi}} = \text{dimensionless pressure}$$

$$H = \frac{h}{h_{bi}} = \text{dimensionless film thickness}$$

$$m = \frac{\lambda_{bi}}{h_{bi}} = \text{ratio of molecular mean free path to film thickness at a reference location}$$

$$c = \frac{6\mu_r l^2}{p_{bi} h_{bi}} = \text{bearing parameter}$$

$$C_f = \frac{F_x}{\frac{\mu_r l^2}{h_{bi}}} = \text{friction coefficient}$$

$$C_L = \frac{W}{p_{bi} l^2} = \text{load coefficient}$$

SUBSCRIPTS

b = refers to a value at outer perimeter of slider

bi = refers to the value at a reference location on outer perimeter of slider

0,1,2... = order of perturbation terms

THE INFLUENCE OF THE MOLECULAR MEAN FREE PATH
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HYDRODYNAMIC GAS LUBRICATED BEARINGS

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ABSTRACT

A modified Reynolds equation is derived for gas-lubricated hydrodynamic bearings operating under "slip flow" conditions. Closed analytical solutions are given for a Rayleigh-type step-bearing and an inclined plane slider bearing for the case of two-dimensional flow.

The influence of the molecular mean free path on the performance of bearings of arbitrary form is obtained by means of a small parameter, perturbation technique.

INTRODUCTION

The present theory of gas-lubricated bearings assumes that the gas can be considered as a continuous fluid. A closer analysis, however, has shown that this assumption is no longer valid when the bearing gap-width becomes comparable in magnitude to the molecular mean free path of the gas. The breakdown of this assumption is not totally unexpected since for relatively small gaps the experimental data are rarely in agreement with the predicted bearing performance.

An example where the influence of the molecular mean free path should be taken into consideration is the case of hydrodynamic gas bearings operating at low ambient pressures. Gas bearings in missiles and those in aircraft flying at high altitude are representatives of this

class. Another example are gas lubricated bearings with extremely small clearances of the type used in some instruments.

Similar problems also are associated with high-speed, high-altitude flight. As soon as the boundary layer thickness becomes comparable to the molecular mean free path, effects occur which no longer can be explained by the continuum flow theory. For small ratios ($\ll 1$) of the mean free path and the boundary layer thickness, the flow is known as the so called "slip flow". An exact evaluation of the flow phenomena in this region is extremely difficult^{1,2,3} and, in order to get approximate results, some simplifying assumptions have to be made.⁴

In order to bring the predictions based on the continuum theory and experimental results into better agreement, we account for the "slip flow" phenomena by modifying the boundary conditions. Instead of vanishing velocities at the boundaries, slip velocities are introduced. The temperature profile is also modified in assuming a temperature discontinuity at the boundary between the wall and the gas.

The magnitude of the slip velocity and of the temperature discontinuity can be evaluated from the kinetic theory of gases. Neglecting higher order terms, the slip velocity is given by:

$$u_{\text{slip}} = \sigma \cdot \frac{(2-f)}{f} \lambda \frac{\partial u}{\partial z} + \dots \quad (1)$$

$$\lambda = \frac{1}{\sqrt{2} \, n s} \quad (2)$$

The temperature jump on the basis of the kinetic theory of gases is given by:

$$T_{\text{gas}} - T_{\text{wall}} = \sigma \frac{(2-\alpha)}{\alpha} \frac{\gamma}{\gamma-1} \cdot \frac{\lambda}{\text{Pr}} \frac{\partial T}{\partial z} + \dots \quad (3)$$

A theoretical estimate of the temperature differences between the bearing surfaces and the gas film shows that they remain small. Consequently the film temperature is essentially governed by the bearing surface temperatures.

ANALYSIS

1. Derivation of a Modified Reynolds Equation

The differential equation for the pressure distribution in a slider bearing operating under "slip flow" conditions is derived below. The equations are written with reference to a rectangular coordinate system as shown in Figure 1. The stationary slider may have an arbitrary form, $h = h(x,y)$, and the flat runner moves with velocity u_r in the positive x-direction. The temperatures of the bearing surfaces are assumed to be equal and constant. As previously mentioned, the temperature of the gas film will vary but little from the surface temperature. As a first approximation, therefore, an isothermal relationship

$$\frac{p}{p_{bi}} = \frac{\rho}{\rho_{bi}} \quad (4)$$

is assumed.

The appropriate relationship is obtained by combining the Navier-Stokes equations and the continuity equation, both used in a simplified form⁵:

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial z^2} \quad (5)$$

$$\frac{\partial p}{\partial y} = \mu \frac{\partial^2 v}{\partial z^2} \quad (6)$$

$$\frac{\partial p}{\partial z} = 0 \quad (7)$$

$$\frac{\partial}{\partial x} \left[\rho \int_0^h u dz \right] + \frac{\partial}{\partial y} \left[\rho \int_0^h v dz \right] = 0 \quad (8)$$

Boundary conditions are:

$$u(z=0) = u_r + \lambda \left. \frac{\partial u}{\partial z} \right|_{z=0} \quad (9)$$

$$u(z=h) = -\lambda \left. \frac{\partial u}{\partial z} \right|_{z=h} \quad (10)$$

$$v(z=0) = \lambda \left. \frac{\partial v}{\partial z} \right|_{z=0} \quad (11)$$

$$v(z=h) = -\lambda \left. \frac{\partial v}{\partial z} \right|_{z=h} \quad (12)$$

$$p[x(t), y(t), z=0] = p_b [x(t), y(t)] \quad (13)$$

Because the values of the constants σ and f in equation (1) are close to unity, $\sigma \frac{(2-f)}{f}$ in (9) through (12) was set equal to 1.

From equation (7) it follows that the pressure p is independent of z . Integrating equation (5) twice with respect to z gives:

$$u = \frac{1}{2\mu} \frac{\partial p}{\partial x} z^2 + f_1(x, y) z + f_2(x, y)$$

$f_1(x, y)$ and $f_2(x, y)$ can be determined by substituting the boundary conditions (9) and (10) in the equation given above. This yields eventually:

$$u = -\frac{1}{2\mu} \frac{\partial p}{\partial x} [h\lambda + hz - z^2] + u_r \left[1 - \frac{z+\lambda}{h+2\lambda} \right] \quad (14)$$

From equation (6) we obtain in a similar manner,

$$v = -\frac{1}{2\mu} \frac{\partial p}{\partial y} [h\lambda + hz - z^2] \quad (15)$$

Equations (14) and (15) when substituted in equation (8) give:

$$\frac{\partial}{\partial x} \left\{ \rho \left[\frac{h^3}{12\mu} \frac{\partial p}{\partial x} \left(1 + 6 \frac{\lambda}{h} \right) - \frac{u_r h}{2} \right] \right\} + \frac{\partial}{\partial y} \left\{ \rho \frac{h^3}{12\mu} \frac{\partial p}{\partial y} \left(1 + 6 \frac{\lambda}{h} \right) \right\} = 0 \quad (16)$$

Because the viscosity is nearly independent of the pressure in the range considered, and since the temperature variations across the gas film are small, the viscosity can be assumed to be constant. Using the expression

for the density as given in equation (4), we obtain from equation (16):

$$\frac{\partial}{\partial x} \left[ph^3 \frac{\partial p}{\partial x} \left(1 + 6 \frac{\lambda}{h} \right) \right] + \frac{\partial}{\partial y} \left[ph^3 \frac{\partial p}{\partial y} \left(1 + 6 \frac{\lambda}{h} \right) \right] = 6\mu u_r \frac{\partial}{\partial x} (ph) \quad (17)$$

Equations (16) and (17) are modified forms of Reynolds' equation of lubrication applicable in the "slip-flow" region ($0 < \frac{\lambda}{h} \ll 1$). The boundary condition is given by equation (13).

To write equation (17) in dimensionless form, we set;

$$p = p_{bi} \cdot P \quad (18)$$

$$x = 1 \cdot X \quad (19)$$

$$y = 1 \cdot Y \quad (20)$$

$$h = h_{bi} \cdot H \quad (21)$$

where p_{bi} and h_{bi} are arbitrarily chosen, but fixed values of p and h on the outer perimeter of the slider. For constant temperature, equation (2) gives:

$$\frac{\lambda}{\lambda_{bi}} = \frac{p_{bi}}{p} \quad (22)$$

Combining equations (17) to (22) results in the following equation:

$$\frac{\partial}{\partial X} \left[PH^3 \frac{\partial P}{\partial X} \left(1 + \frac{6m}{PH} \right) \right] + \frac{\partial}{\partial Y} \left[PH^3 \frac{\partial P}{\partial Y} \left(1 + \frac{6m}{PH} \right) \right] = c \cdot \frac{\partial}{\partial X} (PH) \quad (23)$$

where

$$c = \frac{6\mu u_r}{p_{bi} h_{bi}^2} \quad (24)$$

$$m = \frac{\lambda_{bi}}{h_{bi}} \quad (25)$$

The corresponding boundary condition is:

$$P[X(t), Y(t)] = P_b[X(t), Y(t)] \quad (26)$$

where P_b is a given function along the curve $f[X(t), Y(t)] = 0$. This curve represents the projection of the perimeter of the slider on the XY-plane. Equation (23) is a non-linear differential equation in P which cannot be solved analytically for arbitrary slider geometries.

If, for a given slider profile, the pressure distribution for $m = 0$ is known, then the influence of m on the bearing performance can be evaluated by applying a perturbation technique to equation (23). The procedure is given in Appendix I.

With the pressure distribution of the bearing known, the velocity field in the gap can be computed from equations (14) and (15). The friction force on the runner can be determined from the knowledge of the shear stress at the boundary. The expression for the friction coefficient is derived in Appendix II.

Although equation (16) has been derived for the case of a slider bearing it can easily be modified to apply to journal bearings. The only change required is to replace the x -coordinate by $r\phi$.

2. Analytical Solutions of Equation (23).

In the two-dimensional case, equation (23) can be integrated analytically for two slider bearing configurations. One of these is the Rayleigh step bearing and the other the inclined plane slider bearing.

a) The Rayleigh step bearing

The Rayleigh step bearing can be divided into regions with constant film thicknesses. For each of these regions H in equation (23) can be set equal to unity so that this equation reduces to:

$$\frac{d}{dX} \left[\frac{P}{c} \frac{\partial P}{\partial X} \left(1 + \frac{6m}{P} \right) - P \right] = 0 \quad (27)$$

Integrating once with respect to X we have:

$$\frac{dP}{dX} (P + 6m) - c(P - K_1) = 0 \quad (28)$$

We note that the variables of equation (28) are separable, as a consequence of which we obtain the following result:

$$P + (6m + K_1) \ln(P - K_1) = cX + K_2 \quad (29)$$

where K_1 and K_2 are constants of integration which can be determined from the boundary conditions for each region. A discussion of equation (29) is given in reference⁶ for the case of $m = 0$. Figures 2 and 3 show the pressure distribution for two bearings which differ only in their step location. Curves for three values of the parameter m are shown in each case. The friction coefficients are listed in Table 1.

b) The inclined, plane slider bearing

In this case equation (23) reduces to:

$$\frac{d}{dx} \left[\frac{PH^3}{c} \frac{dP}{dx} \left(1 + \frac{6m}{PH} \right) - PH \right] = 0 \quad (30)$$

where

$$H = 1 - \frac{(h_1 - h_2)}{1} \cdot X \quad (31)$$

If for simplicity we set;

$$\frac{dP}{dX} = - \frac{(h_1 - h_2)}{1} \frac{dP}{dH} \quad (32)$$

equation (30) can then be written as

$$\frac{(h_1 - h_2)}{1} \cdot \frac{PH^3}{c} \frac{dP}{dH} \left(1 + \frac{6m}{PH} \right) + PH = K_1 \quad (33)$$

where K_1 is a constant of integration. Substitution of

$$PH = \theta \quad (34)$$

$$\frac{1}{(h_1 - h_2)} c = c' \quad (35)$$

in equation (33) gives:

$$\theta H \frac{d\theta}{dH} \left(1 + \frac{6m}{\theta}\right) - \theta^2 \left(1 + \frac{6m}{\theta}\right) + c'(\theta - K_1) = 0 \quad (36)$$

The solution of this equation depends on the sign of K , where K is defined as

$$K = 4c'K_1 - (c' - 6m)^2 \quad (37)$$

For $K \geq 0$, the solutions of equation (36) are:

$$\underline{K > 0} \quad (38)$$

$$\frac{1}{2} \ln [\theta^2 - (c' - 6m)\theta + c'K_1] + \frac{c' + 6m}{\sqrt{K}} \operatorname{tg}^{-1} \frac{2\theta - (c' - 6m)}{\sqrt{K}} - \ln H = K_2$$

$$\underline{K = 0}$$

$$\ln \left(\theta - \frac{c' - 6m}{2} \right) - \frac{c' + 6m}{2\theta - c' + 6m} - \ln H = K_2 \quad (39)$$

$$\underline{K < 0} \quad (40)$$

$$\frac{1}{2} \ln [\theta^2 - (c' - 6m)\theta + c'K_1] + \frac{c' + 6m}{\sqrt{-K}} \ln \frac{\sqrt{-K} - 2\theta + (c' - 6m)}{\sqrt{-K} + 2\theta - (c' - 6m)} - \ln H = K_2$$

The constants of integration K_1 and K_2 can be evaluated from the boundary conditions, viz. the ambient pressure values at the inlet and the outlet of the slider.

Figure 4 shows the pressure distribution obtained from equation (40) for three values of m . The friction coefficients are listed in Table 1.

It may be noticed from equation (23) that the influence of the value of m on the load carrying capacity of a gas lubricated bearing is negligible for large values of the bearing parameter c , increasing as c decreases.

3. Conclusions

The analysis of the influence of the ratio m (molecular mean free path to a representative film thickness at a reference location) on the performance of gas lubricated bearings shows that:

- a) Values of m greater than 0.01 will have a noticeable effect on the bearing performance such as load carrying capacity and the friction coefficient.
- b) The load carrying capacity decreases with increasing value of m . This decrease is most pronounced at low speeds. On the other hand, the influence of m at very high speeds is negligible.
- c) The friction coefficient decreases with increasing values of m .

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APPENDIX I

Equation (23) has been derived on the assumption that m is a small quantity. Consequently a small parameter technique can be applied in equation (23) when evaluating the influence of the ratio m . To this effect, the pressure is expressed in the form of a power series in m :

$$P = P_0 + mP_1 + m^2P_2 + \dots = \sum_{n=0}^{\infty} m^n P_n \quad (41)$$

Substituting (41) into equation (23) and collecting terms of like power of m gives a system of equations for the evaluation of the coefficients P_n of the power series. The zeroth term equation is:

$$\frac{\partial}{\partial X} \left(H^3 P_0 \frac{\partial P_0}{\partial X} \right) + \frac{\partial}{\partial Y} \left(H^3 P_0 \frac{\partial P_0}{\partial Y} \right) = c \frac{\partial}{\partial X} (P_0 H) \quad (42)$$

The boundary condition (26) is independent of m and equation (42) is identical with equation (23) for $m = 0$. Equation (26) therefore must also be a boundary condition for equation (42).

$$P_0[X(t), Y(t)] = P_b[X(t), Y(t)] \quad (43)$$

The first order equation then becomes:

$$\begin{aligned} \frac{\partial}{\partial X} \left[H^3 P_0 \frac{\partial P_1}{\partial X} + H^2 (H P_1 + 6) \frac{\partial P_0}{\partial X} \right] + \frac{\partial}{\partial Y} \left[H^3 P_0 \frac{\partial P_1}{\partial Y} + H^2 (H P_1 + 6) \frac{\partial P_0}{\partial Y} \right] \\ = c \frac{\partial}{\partial X} (P_1 H) \end{aligned} \quad (44)$$

with the boundary condition

$$P_1[X(t), Y(t)] = 0 \quad (45)$$

Equations of higher order are more complex algebraically, but in most cases the convergence of the series is sufficiently rapid to justify the neglect of higher order terms.

APPENDIX II

The shear stresses at the runner surface are:

$$\tau_x = \mu \left. \frac{\partial u}{\partial z} \right|_{z=0} \quad (46)$$

$$\tau_y = \mu \left. \frac{\partial v}{\partial z} \right|_{z=0} \quad (47)$$

Normally the only friction forces of interest are those acting in the direction opposing motion. In the present analysis this force, acting on a surface element $dx dy$, is:

$$dF_x = -\tau_x dx dy = -\mu \left. \frac{\partial u}{\partial z} \right|_{z=0} dx dy \quad (48)$$

The total friction force can then be determined by integration over the bearing area. Using equation (14) we have:

$$F_x = -\mu \iint \left. \frac{\partial u}{\partial z} \right|_{z=0} dx dy = \mu \iint \left(\frac{h}{2\mu} \frac{\partial p}{\partial x} + \frac{u_r}{h+2\lambda} \right) dx dy \quad (49)$$

The friction coefficient can then be written as:

$$C_f = \frac{F_x}{\frac{\mu u_r^2}{h_{bi}}} = \iint \left[\frac{3}{c} \frac{\partial P}{\partial X} + \frac{1}{H \left(1 + \frac{2m}{PH} \right)} \right] dX dY \quad (50)$$

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TABLE 1.

	Step bearing (1)			Step bearing (2)			Straight slider bearing		
m	C_f	C_1	$\frac{C_f}{C_1}$	C_f	C_1	$\frac{C_f}{C_1}$	C_f	C_1	$\frac{C_f}{C_1}$
0	1.413	0.337	4.19	1.692	0.562	3.01	1.396	0.331	4.22
0.05	1.282	0.331	3.87	1.528	0.511	2.99	1.256	0.305	4.12
1/6	1.037	0.285	3.64	1.226	0.399	3.09	1.005	0.252	3.99

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FIGURE I
COORDINATE SYSTEM

FIGURE 2
PRESSURE DISTRIBUTION ALONG A STEPPED SLIDER BEARING

FIGURE 3
PRESSURE DISTRIBUTION FOR COMPRESSIBLE FLUIDS
ALONG A STEPPED SLIDER BEARING

FIGURE 4
PRESSURE DISTRIBUTION FOR COMPRESSIBLE FLUIDS
ALONG A STRAIGHT SLIDER BEARING

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