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A TECHNIQUE FOR DESIGNING A COMPOUND HYDROSTATIC BEARING,(U)

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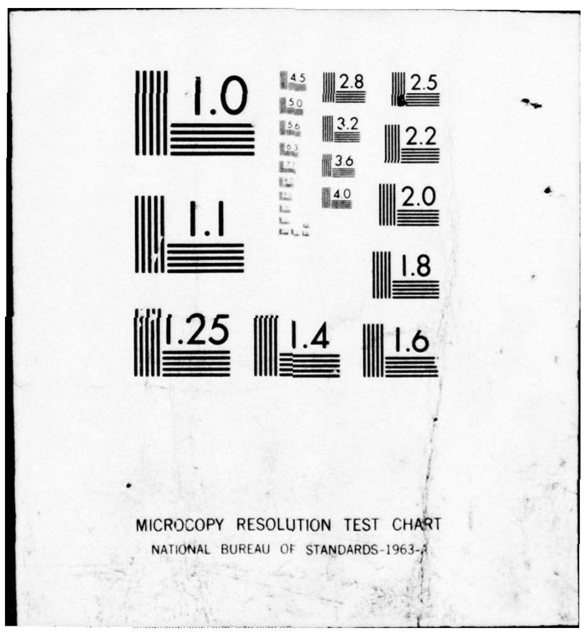
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A TECHNIQUE FOR DESIGNING A COMPOUND HYDROSTATIC BEARING

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Traditionally, the resultant radial load associated with radial-piston hydraulic pumps and motors has been supported by rolling-element bearings. For most applications, a single bearing cannot be located in the plane containing the centerlines of all the pistons because of excessive race peripheral velocities. A compound hydrostatic bearing has been invented; placed in the plane containing the centerlines of all the pistons, it reduces the axial length and volume required by the machine.

This paper describes the design approach and the selection of critical design parameters. System mathematical modeling and design trade-off techniques developed to facilitate application of the hydrostatic-bearing design theory are presented, and procedures for selecting specific bearing geometry are explained.

ABSTRACT

Traditionally, the resultant radial load associated with radial-piston hydraulic pumps and motors has been supported by rolling-element bearings. For most applications, a single bearing cannot be located in the plane containing the centerlines of all the pistons because of excessive race peripheral velocities. A compound hydrostatic bearing has been invented; placed in the plane containing the centerlines of all the pistons, it reduces the axial length and volume required by the machine.

This paper describes the design approach and the selection of critical design parameters. System mathematical modeling and design trade-off techniques developed to facilitate application of the hydrostatic-bearing design theory are presented, and procedures for selecting specific bearing geometry are explained.

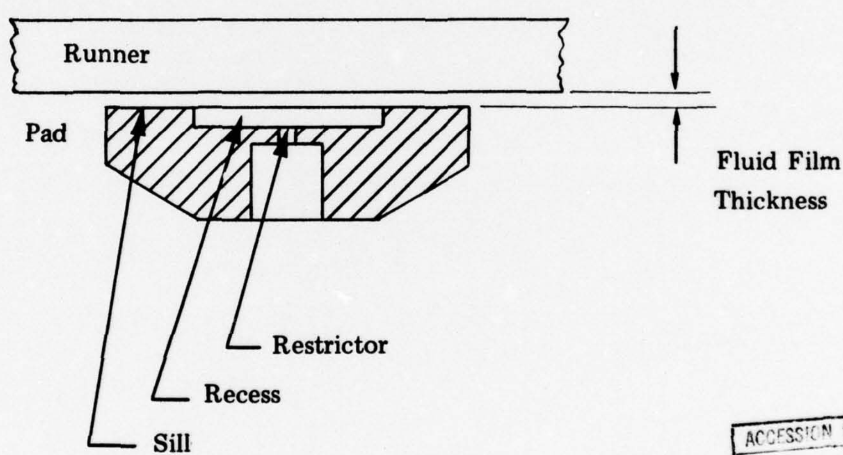
INTRODUCTION

Manufacturers of radial piston hydraulic pumps and motors have traditionally used rolling-element bearings to support the radial loads created by pressure and centrifugal forces on the pistons. Locating the rolling elements in the plane of the pistons caused high peripheral speed which, in combination with high loading, shortened service life. An alternative approach was to use two smaller rolling-element bearings located on the cylinder-block drive shaft, one on each side of the cylinder block. This design, although increasing bearing life, resulted in a larger, heavier machine.

The compound hydrostatic bearing (two simple hydrostatic bearings in the same fluid-circuit branch) can be located in the same plane as the pistons, thereby minimizing the axial length of the machine. In addition, the compound hydrostatic bearing offers the same operational advantages as the simple hydrostatic bearing. These include high load capacity with low friction, reduced vibration and noise, and longer service life. Hydrostatic bearings are particularly suited to hydraulic pumps and motors because these machines provide an inherent fluid power source. Their use is limited by the system's fluid cleanliness, fluid temperature stability, and available power as well as manufacturing costs.

A hydrostatic bearing in its simplest form consists of a pad and runner. The pad incorporates three basic parts: recess, sill, and restrictor (Figure 1). Pressurized fluid is fed through the restrictor

Figure 1. **SIMPLE HYDROSTATIC BEARING**



into the recess. When the product of the recess pressure and recess area becomes greater than the load holding the pad and runner together, the pad and runner will separate and fluid will flow from the source through the restrictor and recess and out over the sill, establishing the load-supporting fluid film. The thickness of the fluid film is determined by the bearing supply pressure, restrictor size and type, fluid type and temperature, recess and sill areas, and applied load.

Figure 2 illustrates the essential parts of a compound hydrostatic bearing. The particular application shown is the positive-displacement radial piston pump from an aircraft hydraulic constant speed drive. In contrast to the simple hydrostatic bearing, a free-floating ring containing an array of bearing elements is added to create two hydrostatic bearings in the same fluid-circuit branch. The first bearing is formed between the ring inner surface and the piston pad. This bearing is specifically designed to operate with zero fluid-film thickness (direct contact between pad and ring) and contains no flow restrictor. The second bearing is formed between the ring outer surface and the wobbler. Pressurized fluid is fed into the individual pad recesses, each of which in turn communicates at all times with at least one of the ring-bearing elements. Fluid pressure acts on the individual ring-bearing elements to support the applied loads transmitted to the ring by the pads. In this case, a fluid-film thickness is established that prevents metal-to-metal contact between ring and wobbler.

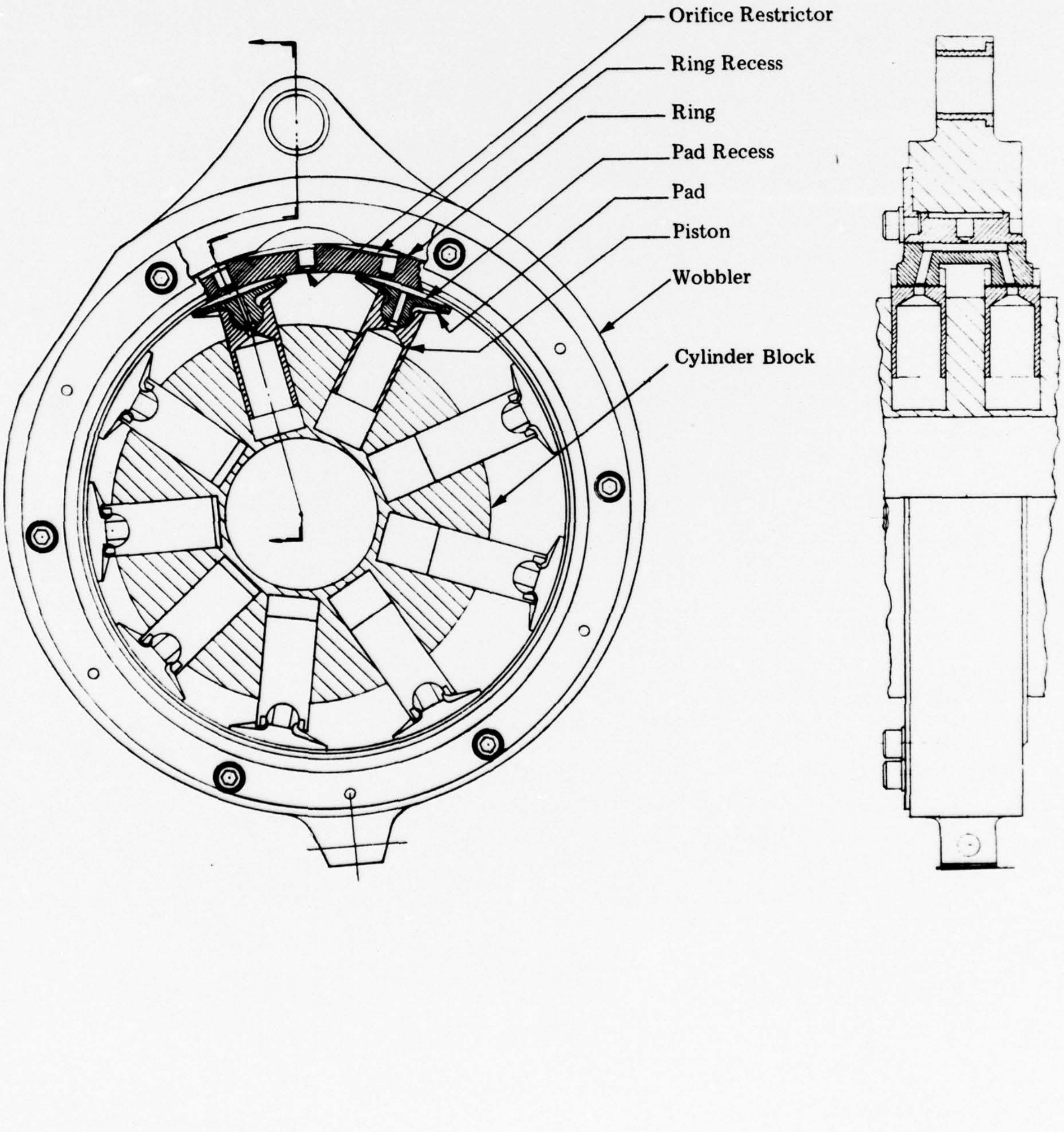
Although restrictors could be added to the pads and the pads designed to run directly against the wobbler without the ring, this approach is not practical for pumps whose pistons generate high centrifugal loads. Abnormally large pads would be required to support the loads, and these in turn would significantly increase the centrifugal loading, thereby necessitating a further increase in pad size.

The basic purpose of using the ring is to support the combined piston radial loads (i.e., the resultant force vector) rather than each load individually as just described. Since the lines of piston action are radial, their forces have a mutual canceling effect, permitting support of the resultant load by a series of relatively small ring-bearing elements.

RING DESIGN

Under the influence of nonuniform piston loads, the ring will tend to move eccentrically within the wobbler. This motion will be consistent with the direction of the resultant applied force vector. The ring will continue to move eccentrically until its outer surface contacts the wobbler (eccentricity = radial clearance) or until the hydrostatic forces at the ring outer surface exactly

Figure 2. **COMPOUND HYDROSTATIC BEARING USED IN RADIAL PISTON HYDRAULIC PUMP**



balance the applied force (dynamic equilibrium). The design of the ring thus involves selecting specific design parameters that will cause the ring to achieve dynamic equilibrium at eccentricity values lower than the radial clearance under all conditions of speed, load, and temperature experienced by the pump. The specific design parameters include radial clearance between ring and wobbler, restrictor diameter, ring recess size, ring outside diameter, and the number of ring-bearing elements. In addition to meeting these fundamental design criteria, the combination of parameters selected should tend to optimize bearing performance characteristics, including bearing flow, fluid temperature rise, and power required.

Figure 3 is a free-body diagram of a single piston/pad assembly. To calculate the net radial load, the following assumptions are made:

- The piston side-wall friction is zero.
- The cylinder-block rotational speed is constant.
- There is instantaneous transition from discharge to inlet pressure.
- The piston/pad assembly center of gravity coincides with the pad pivot center.

Applying Newton's second law to the piston/pad assembly yields

$$p_s A_p + \frac{W}{g} r \left(\frac{d\theta}{dt}\right)^2 - F_r \cos \delta = \frac{W}{g} \left(\frac{d^2 r}{dt^2}\right) \tag{1}$$

where

$$p_s = p_1 \quad (0 \leqslant \theta < \pi) \tag{2}$$

$$p_s = p_2 \quad (\pi \leqslant \theta \leqslant 2\pi) \tag{3}$$

$$r = C \left(\frac{\sin \alpha}{\sin \theta}\right) \tag{4}$$

$$\alpha = \theta \pm \delta \tag{5}$$

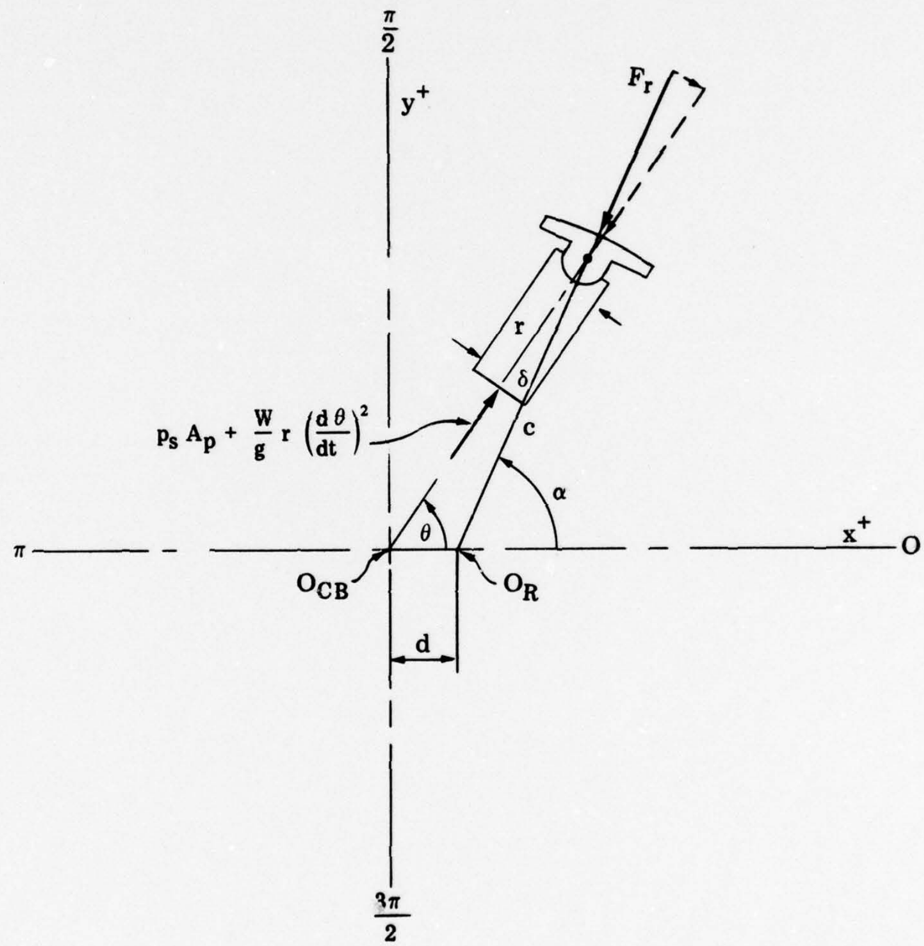
$$\delta = \sin^{-1} \left[\pm \left(\frac{d}{C}\right) \sin \theta\right] \tag{6}$$

$$\frac{d\theta}{dt} = \left(\frac{\pi}{30}\right) N \tag{7}$$

and, by using the relationship

$$\frac{dr}{dt} = \left(\frac{dr}{d\theta}\right) \left(\frac{d\theta}{dt}\right)$$

Figure 3. **FREE BODY DIAGRAM OF PISTON/PAD ASSEMBLY**



the radial acceleration is found to be

$$\frac{d^2 r}{dt^2} = \left(\frac{d\theta}{dt}\right)^2 \frac{d^2 r}{d\theta^2} \tag{8}$$

Plus signs are used in equations 5 and 6 when $0 \leq \theta < \pi$; minus signs are used when $\pi \leq \theta \leq 2\pi$. The $\frac{d^2 r}{d\theta^2}$ term of Equation 8 can be evaluated by differentiating Equations 4, 5, and 6 with respect to θ.

From Equation 1, the radial force exerted on the ring (center, 0_R) by an individual pad is

$$F_r = \left\{ p_s A_p + \frac{W}{g} \left[r \left(\frac{d\theta}{dt}\right)^2 - \frac{d^2 r}{dt^2} \right] \right\} \Big/ \cos \delta \tag{9}$$

which is resolved into the components

$$(F_r)_x = F_r \cos \alpha \tag{10}$$

$$(F_r)_y = F_r \sin \alpha \tag{11}$$

The resultant applied force acting on the ring, illustrated in Figure 4, is given by

$$R = \left[(R_x)^2 + (R_y)^2 \right]^{1/2} \tag{12}$$

where R_x and R_y are the resultant components given by

$$R_x = \sum_{i=1}^{n_p} (F_r)_{ix} \tag{13}$$

and

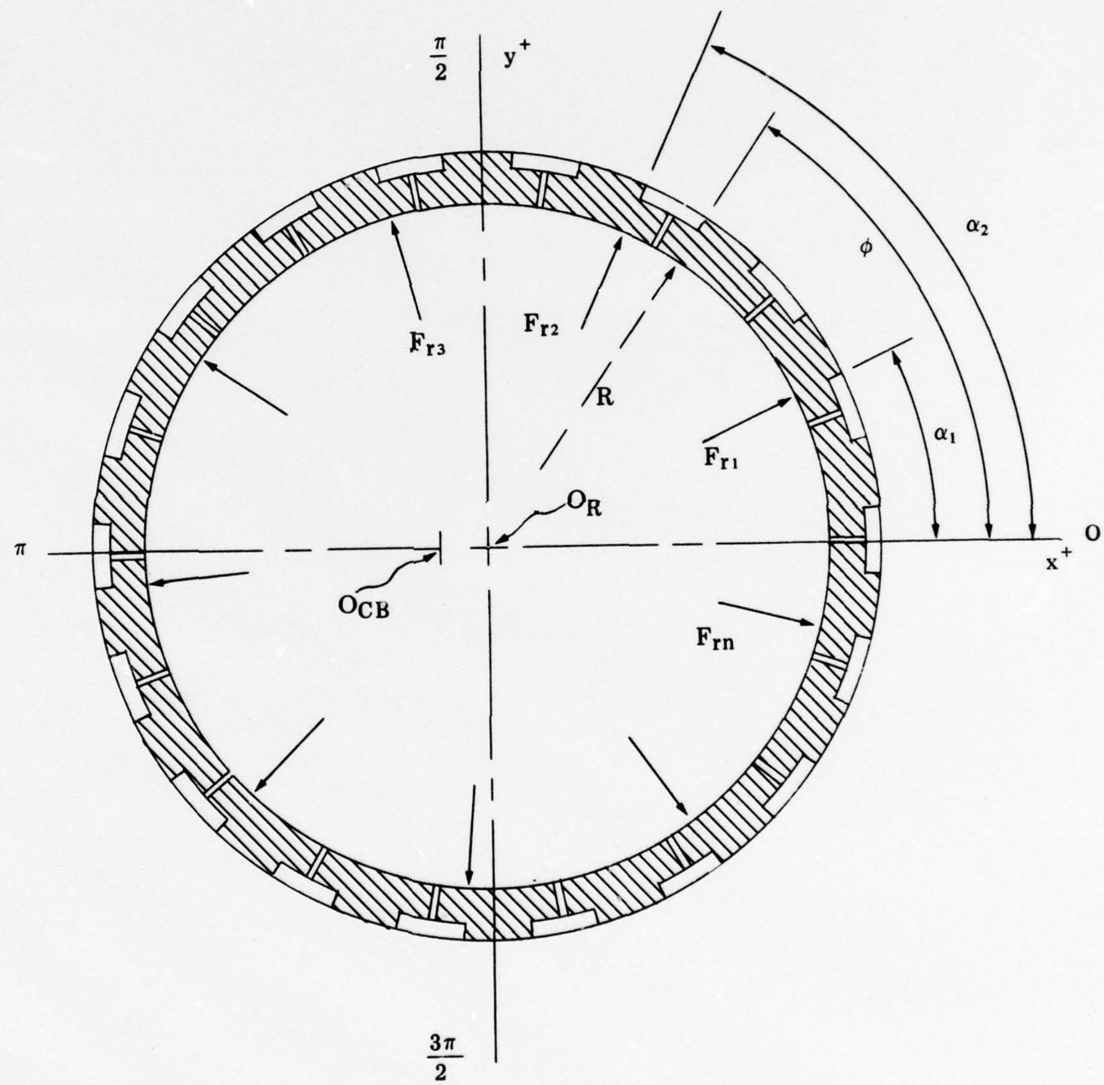
$$R_y = \sum_{i=1}^{n_p} (F_r)_{iy} \tag{14}$$

In these calculations, the angular position of each individual piston/pad assembly is established from

$$\theta_i = \theta_1 + (i-1) \frac{2\pi}{n_p} \Bigg|_{i=1}^{n_p} \tag{15}$$

where θ_1 is an arbitrarily specified angular location for the first piston/pad assembly. The direction of the resultant applied force (ϕ, Figure 4) is a function of both the magnitude and the direction of this force's components; it is given by the following equation:

Figure 4. **RING RADIAL LOADING**



$$\phi = \begin{vmatrix} \psi & \text{for} & (+R_x, +R_y) \\ \pi - \psi & \text{for} & (-R_x, +R_y) \\ \pi + \psi & \text{for} & (-R_x, -R_y) \\ 2\pi - \psi & \text{for} & (+R_x, -R_y) \end{vmatrix} \tag{16}$$

where

$$\psi = \text{Tan}^{-1} \frac{|R_y|}{|R_x|} \tag{17}$$

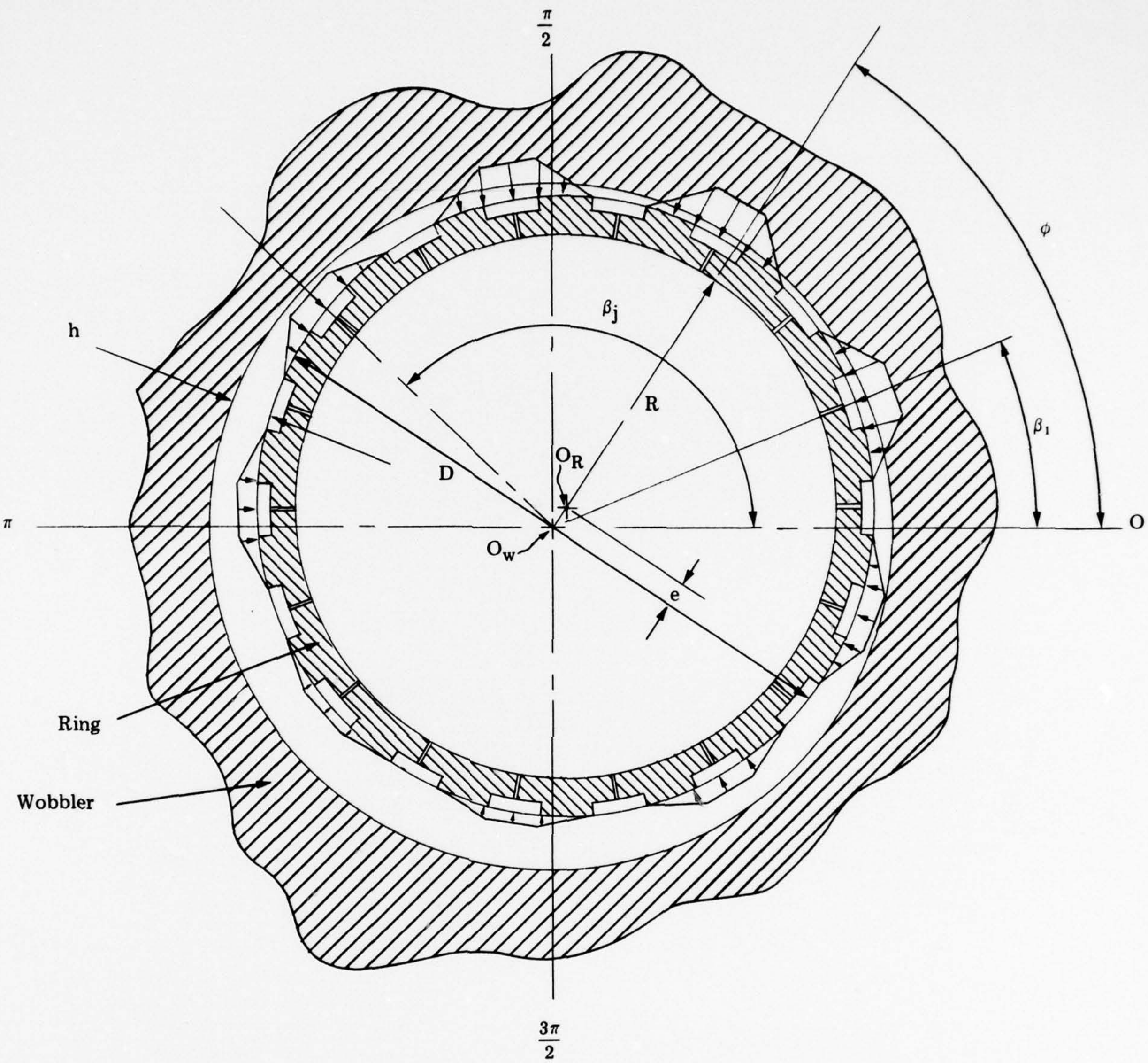
and the plus and minus signs indicate the sense of the particular component in accordance with the coordinate system shown in Figure 4.

It is now necessary to determine the hydrostatic forces on the ring outer surface. Figure 5a illustrates typical hydrostatic-pressure distributions acting on the various ring-bearing elements. A single bearing element includes one recess, one restrictor, and the surrounding sill area between adjacent recesses and the edge of the ring. The projected total area for a single element is A_b, as shown in Figure 5b. It is emphasized that the number of ring recesses is chosen, consistent with pad-recess geometry, so that each alternate ring-recess restrictor will be exposed to pump pressure at any given instant for any wobbler displacement.

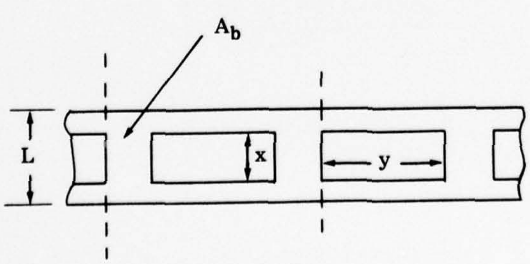
As indicated earlier, the resultant applied load, R, tends to move the ring eccentrically in accordance with its direction. It is therefore necessary to investigate the reactive hydrostatic-pressure forces as a function of ring-to-wobbler eccentricity (e, Figure 5a) for various combinations of design parameters over all expected operating modes. Equations developed for this purpose are based on the following simplifying assumptions:

- O_R is coincident with O_W since $e << D$.
- A steady-state operating condition prevails; all transient dynamic effects are zero.
- Each ring-bearing element functions as a pure hydrostatic bearing with no hydrodynamic effects.
- The pressure distribution across the bearing-element sill is linear.
- Full pump pressure (either p_1 or p_2) is available at the restrictor independent of fluid flow.
- Pump case pressure is zero psig.
- The restrictor is an ideal orifice.

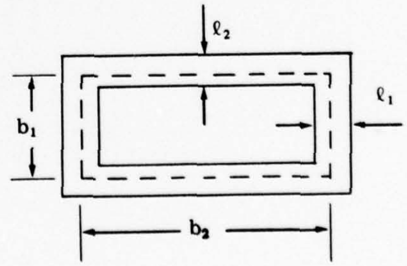
Figure 5. COMPOUND HYDROSTATIC BEARING RING



(a) Hydrostatic Pressure Distributions Acting on Ring Periphery



(b) View of Ring Outer Surface



(c) Idealized Recess/Sill Geometry

- Fluid leakage over the pad sill is zero.
- Heat transfer between pump walls and the fluid is zero.

On the basis of the dimensions illustrated in Figure 5, individual bearing-element load and flow coefficients are determined to be*

$$a_f = \frac{1}{3}\left(1 + \lambda + \lambda^{1/2}\right) \tag{18}$$

$$q_f = \frac{1}{6a_f}\left(\frac{b_1}{\ell_1} + \frac{b_2}{\ell_2}\right) \tag{19}$$

where

$$\lambda = \frac{xy}{A_b} \tag{20}$$

and

$$A_b = L\left[\frac{2\pi D}{n_r} - y\right] \tag{21}$$

Fluid volume flow through an individual bearing-element orifice restrictor is given by

$$Q_{oj} = k_o (2)^{1/2} (p_s - p_{rj})^{1/2} \tag{22}$$

where

$$k_o = \frac{C_d \pi d_o^2}{4 \rho^{1/2}} \tag{23}$$

and

$$p_s = \begin{cases} p_1 & (0 \leqslant \beta < \pi) \\ p_2 & (\pi \leqslant \beta \leqslant 2\pi) \end{cases} \tag{24}$$

Similarly, the fluid volume flow over the sill area of a single bearing element is

$$Q_{sj} = \left(\frac{a_f q_f}{\mu}\right) h_j^3 p_{rj} \tag{25}$$

where the radial clearance between the particular bearing element considered and the wobbler (h_j) is given by

$$h_j = h_o - e \cos(\beta_j - \phi) \tag{26}$$

---

*Reference 2.

with β_j, the angular position of the particular bearing element, calculated by

$$\beta_j = \beta_1 + (j-1)\,\frac{4\pi}{n_r}\Bigg|_{j=1}^{n_r/2} \qquad (27)$$

As was the case with θ_1 in Equation 15, β_1 is the arbitrarily specified position of the first pressurized bearing element counterclockwise from zero (see Figure 5a).

As shown in Figure 6, flow continuity makes $Q_{sj} = Q_{oj}$. Therefore, substituting Equation 25 into Equation 22 and solving for the individual bearing-element recess pressure, p_{rj}, yields

$$p_{rj} = (K_j^2 + 2\,K_j\,p_s)^{1/2} - K_j \qquad (28)$$

where

$$K_j = \left(\frac{\mu k_o}{a_f\,q_f}\right)^2 \left(\frac{1}{h_j}\right)^6 \qquad (29)$$

The hydrostatic pressure acting on each individual pressurized ring-bearing element produces a force directed toward the center of the ring. The magnitude of this force for any given ring-bearing element is

$$W_j = a_f\,A_b\,p_{rj} \qquad (30)$$

Each of the individual hydrostatic forces can be resolved into components, one acting in the same direction as the resultant applied load (R) and the other acting perpendicular to the direction of R. The component of W_j acting in the same direction as R is given by

$$W_{jR} = W_j \cos(\beta_j - \phi) \qquad (31)$$

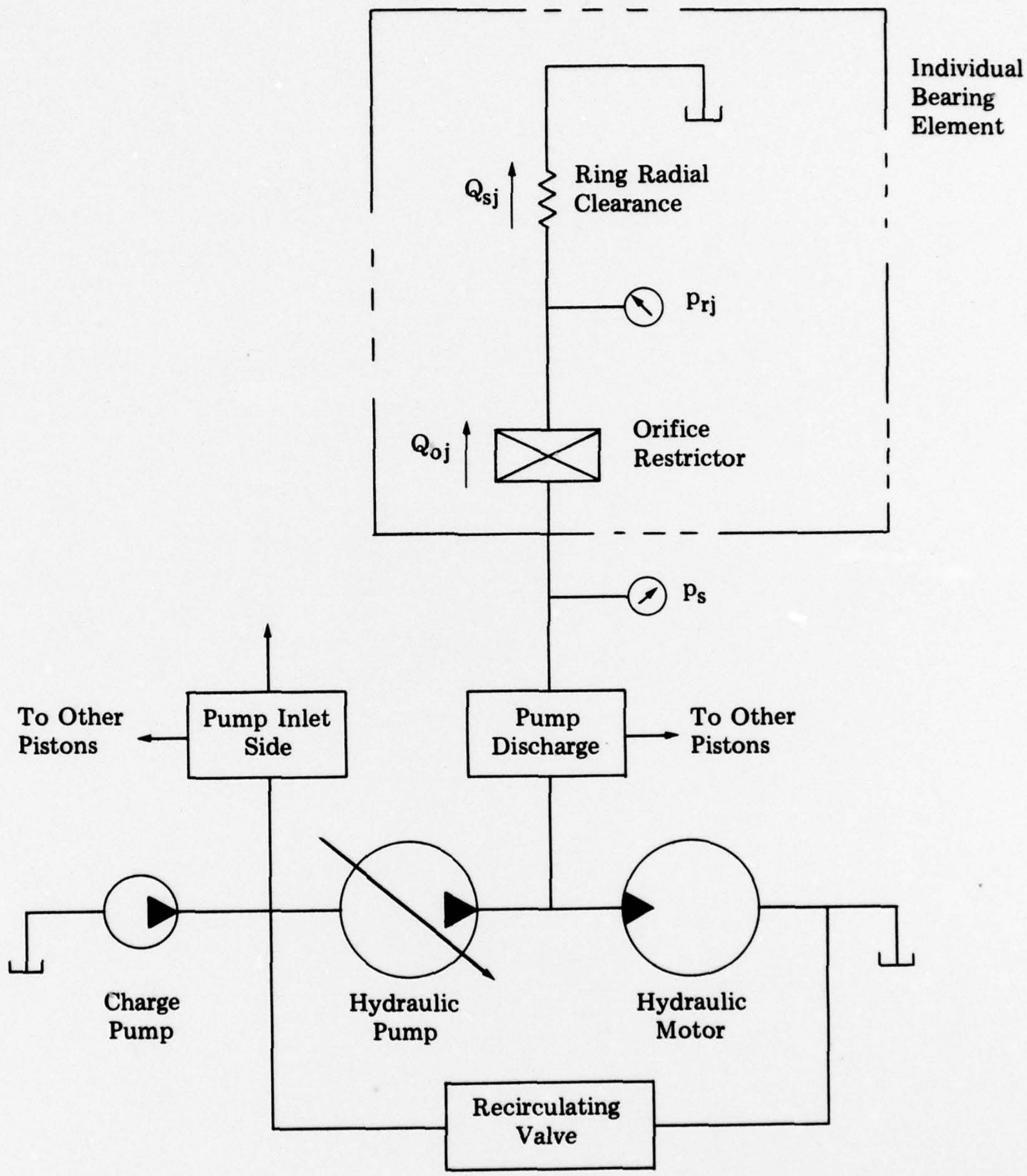
and is the only component of interest here; the vector summation of these components for each pressurized ring-bearing element is the net hydrostatic force available to balance (support) the applied load:

$$W_R = \sum_{j=1}^{n_r/2} W_{jR} \qquad (32)$$

The total horsepower required to overcome viscous shear forces acting on the ring is

$$HP_s = \left(\frac{\mu A_s V^2}{6600}\right) \sum_{j=1}^{n_r/2} \left(\frac{1}{h_j}\right) \qquad (33)$$

Figure 6. **IDEALIZED COMPOUND HYDROSTATIC BEARING SYSTEM**



where

$$V = \frac{\pi DN}{60} \tag{34}$$

and

$$A_s = A_b - xy \tag{35}$$

As the fluid enters the ring recess through the orifice restrictor, its temperature will be increased because of the accompanying loss of pressure. The increase in temperature sustained by the fluid entering an individual ring recess is

$$\Delta T_j = \left(\frac{1}{\rho g \, C_p \, J}\right) \left(p_s - p_{rj}\right) \tag{36}$$

and the average temperature increase for all $n_r/2$ pressurized ring recesses is

$$\Delta T_{AV} = \frac{2 \sum_{j=1}^{n_r/2} \Delta T_j}{n_r} \tag{37}$$

The total fluid volume flow required for the complete compound hydrostatic bearing system is the summation of the flows required by the individual bearing elements; it is given as

$$Q = \sum_{j=1}^{n_r/2} Q_{sj} \tag{38}$$

Equations 18 through 38 are the mathematical means to investigate the quantitative effect of design-parameter and operating-condition variation on the bearing performance characteristics. Assuming that the number of ring recesses (n_r), the ring outside diameter (D), and the ring axial length (L) have already been established, the design parameters to be determined include the following:

- d_o, the orifice restrictor diameter
- h_o, the ring-to-wobbler radial clearance
- x and y, *the recess dimensions*

These parameters must be determined so that, with all possible variation in pump operating modes, the bearing performance characteristics can be optimized within acceptable limits. The pump operating modes are defined by:

- μ, the fluid absolute viscosity that is specified for the expected average recess fluid temperature
- p_1 and p_2, the discharge and inlet fluid pressures
- N, the pump rotational speed

The bearing performance characteristics are defined as follows:

- e', eccentricity at which ring forces are in dynamic equilibrium
- HP_s, power to overcome viscous friction
- Q, total fluid volume flow through the bearing
- ΔT_{AV}, average increase in fluid temperature

The procedure for performing the analysis is as follows:

1. Select trial values for d_o, h_o, x and y in Equations 20, 21, 23, 26, and 35.

2. Substitute upper-limit values for N, p_1, p_2, and μ in Equations 2, 3, 7, 24, 25, and 34.

3. Calculate R (Equation 12) and ϕ (Equation 16).

4. Calculate W_R, HP_s, Q, and ΔT_{AV} as a function of e by varying the value of e in Equation 26 from $-h_o$ to $+h_o$ and solving Equations 32, 33, 37, and 38 each time e is varied.

5. Compare the function W_R (e) calculated in Step 4 with R calculated in Step 3. Ring-force dynamic equilibrium is established at the point where W_R = R. The value of e used to calculate the point of dynamic equilibrium is noted as the ring steady-state operating eccentricity.

6. Review the performance-characteristics functions, $HP_s(e)$, Q(e), and $T_{AV}(e)$ calculated in Step 4. Note those performance-characteristics values that were calculated on the basis of the ring steady-state operating eccentricity found in Step 5.

7. Proceed through Steps 2, 3, 4, 5, and 6 again on the basis of average values in Step 2, and then again on the basis of the lower-limit values in Step 2.

8. Now repeat Steps 1 through 7 a finite number of times, first holding h_o, x, and y constant and varying d_o; and then holding d_o, x, and y constant and varying h_o; etc.

9. From the results of the preceding calculations, plot families of curves that will facilitate selecting the optimum design parameters. Figures 7, 8, and 9 illustrate several such plots.

In addition to the curves plotted in Step 9, it is also useful to plot such curves as the following:

$Q(\beta_1) \mid$ all other variables constant

$Q(p_1, p_2, \mu, N) \mid$ all other variables constant

$e'(p_1, p_2, \mu, N) \mid$ all other variables constant

These will show, respectively, (a) total variation in flow through the bearing as a function of the arbitrarily selected location of the first pressurized ring recess, (b) total flow variations as a function of changes in operating modes, and (c) variations in ring steady-state eccentricity as a function of operating-mode changes to prevent metal-to-metal contact under any mode of operation.

The technique just described is extremely laborious and can be handled practically only by the computer. Figure 10 is the flow diagram for a computer program to perform the numerous iterative calculations. This particular mathematical model is designed to perform only Step 4 of the technique. The resultant applied load and its direction of action (Step 3), as well as the ring steady-state operating eccentricity (Step 5), must be calculated separately. Figure 10 could be modified to include not only Step 3 but the entire analytical process, including the selection of optimum design values for d_o, h_o, x, and y. However, in the present procedure, the engineer using the program must first calculate the resultant applied load (manually or by a separate program) and visually compare it with the range of computer-calculated load capacities [W_R (e)] to determine, for a ring-force equilibrium condition, the steady-state ring eccentricity, total leakage, and other performance characteristics.

In the selection of optimum design parameters by means of the curves developed in Step 9, the following general criteria should be observed:

Figure 7. **RING STEADY-STATE OPERATING ECCENTRICITY AS A FUNCTION OF RING-TO-WOBBLER RADIAL CLEARANCE FOR SEVERAL OIL INLET TEMPERATURES**

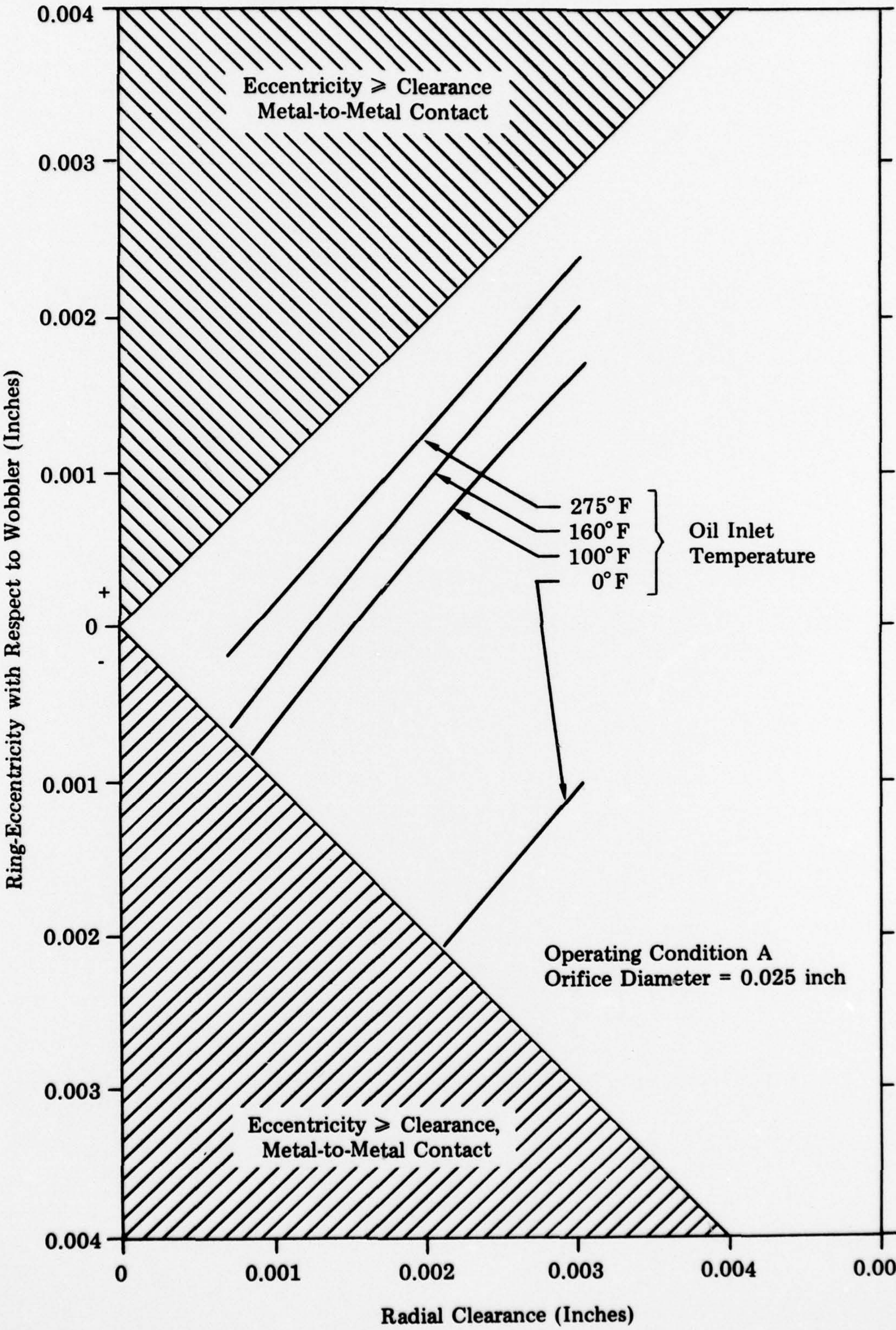


Figure 8. **TOTAL BEARING FLOW AT RING STEADY STATE OPERATING ECCENTRICITY AS A FUNCTION OF RING-TO-WOBBLER RADIAL CLEARANCE FOR SEVERAL ORIFICE DIAMETERS**

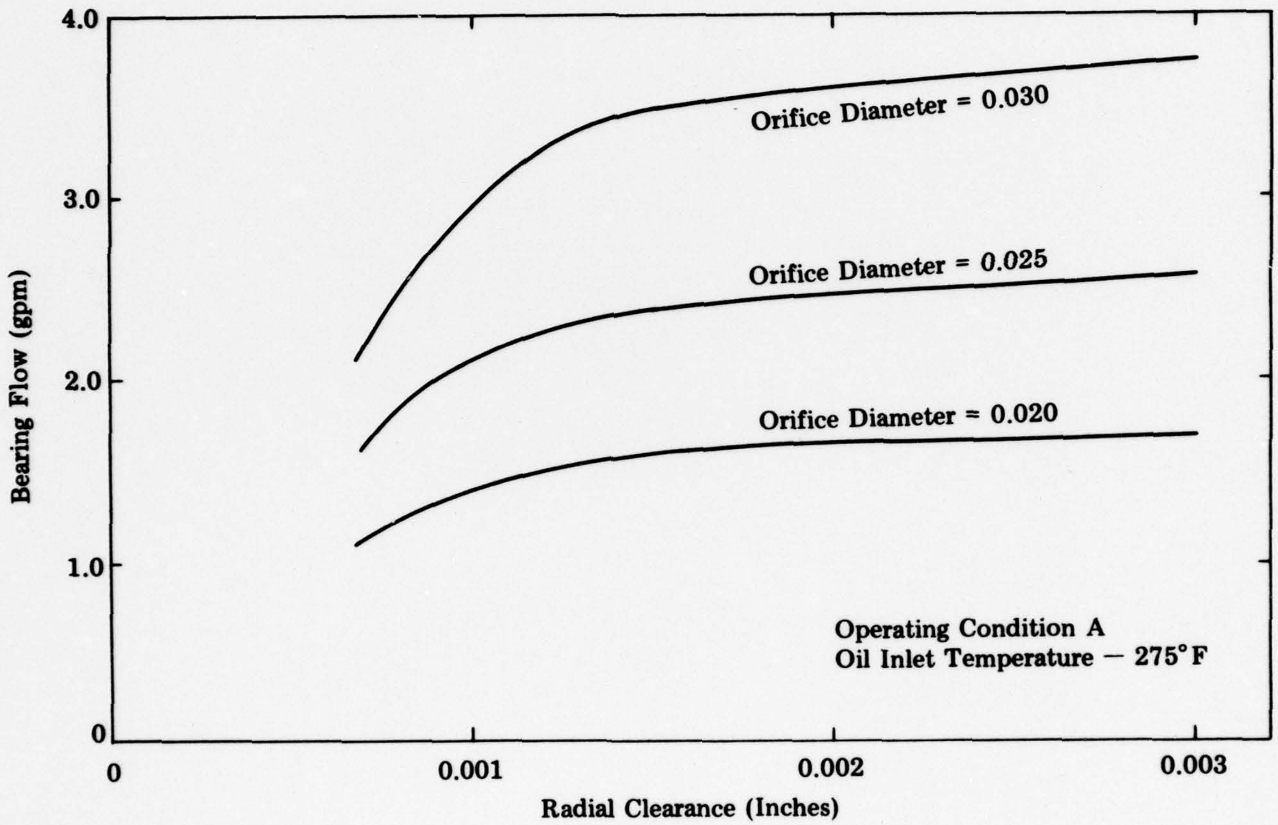


Figure 9. **TOTAL BEARING FLOW AT RING STEADY STATE OPERATING ECCENTRICITY AS A FUNCTION OF RING-TO-WOBBLER RADIAL CLEARANCE FOR SEVERAL OIL INLET TEMPERATURES**

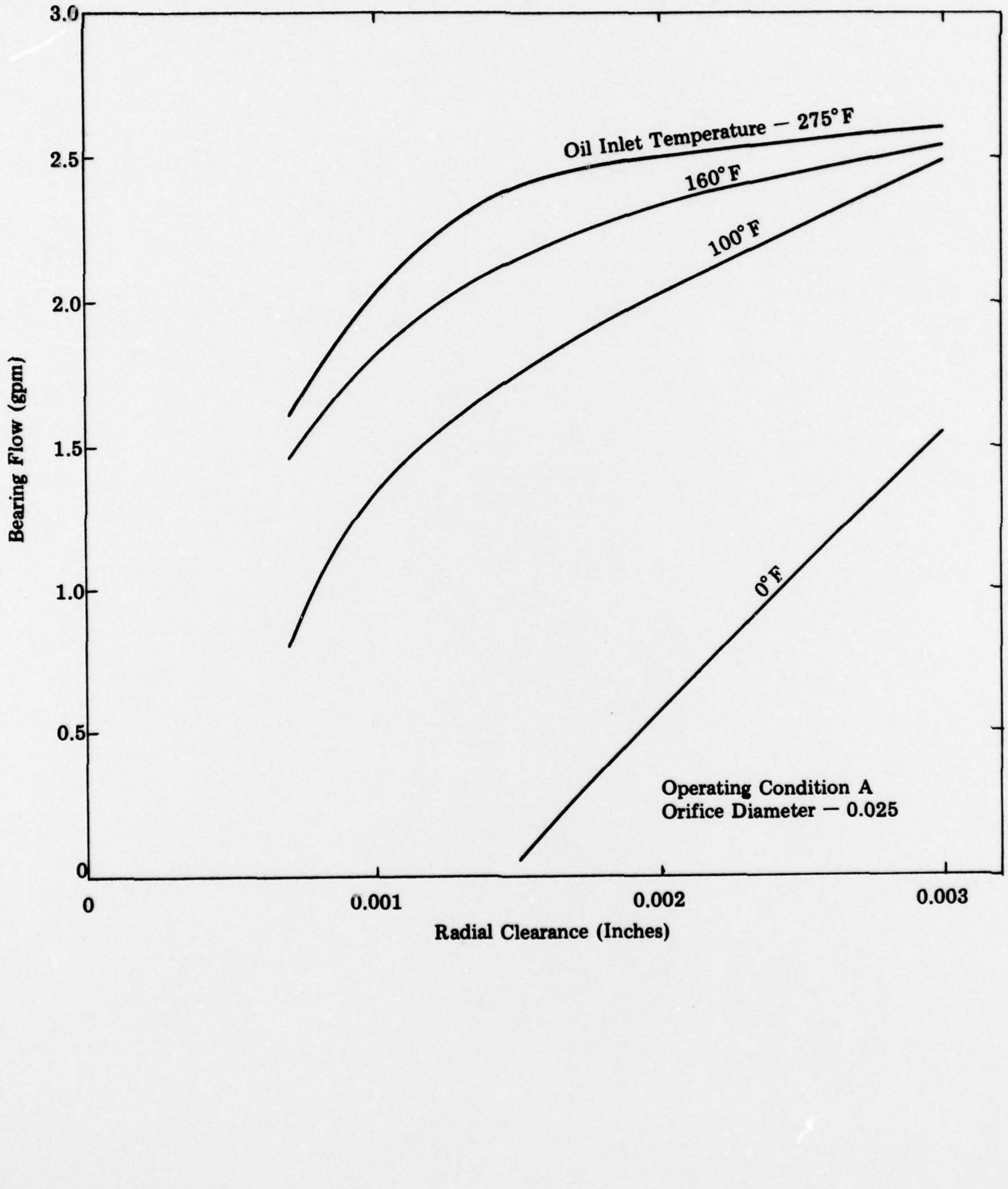
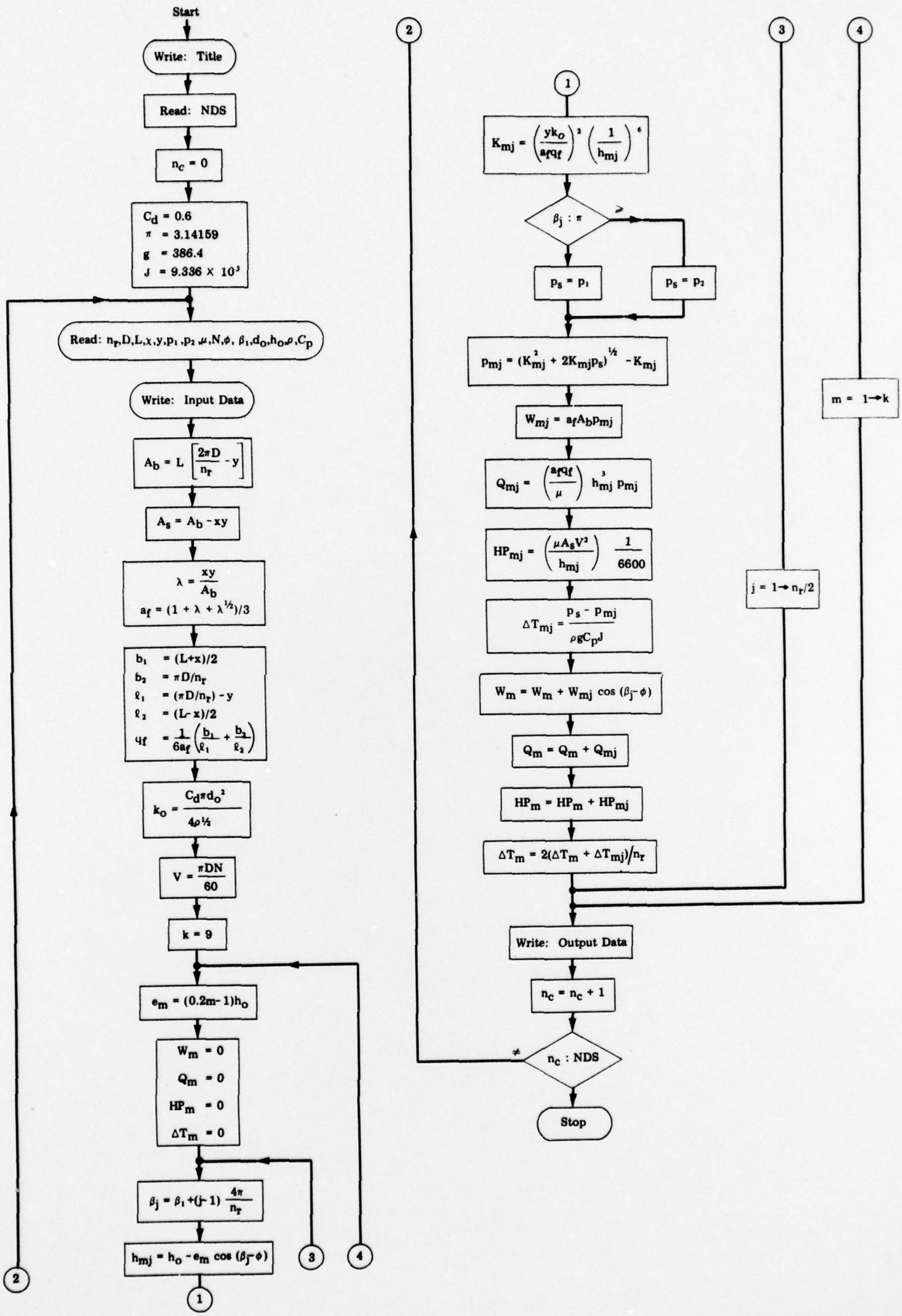


Figure 10. **COMPUTER PROGRAM FLOW CHART**



- The steady-state ring eccentricity cannot physically exceed the radial clearance. If dynamic equilibrium cannot be established within radial-clearance limits, metal-to-metal contact will occur at the race periphery. The prime design objective is to achieve, as nearly as possible, perfect steady-state concentricity between ring and wobbler.
- The total bearing flow, fluid temperature increase, and power to overcome viscous friction should be minimized.
- There should be negligible variation in ring eccentricity and in bearing performance characteristics as the pump operating conditions change.

Of course, complete adherence to these criteria is impossible; thus a design compromise is necessary. The relative importance of the various performance characteristics will depend on the specific system for which the bearing is intended.

PAD DESIGN

The functions of the pad are to transmit the piston loads to the ring and to transfer pressurized fluid from the individual piston chambers to the ring-bearing element. Since the pad is intended to operate in contact with the ring (zero fluid flow over the pad sill), its design is relatively simple and straightforward. Guidelines for designing the pad are as follows:

- The individual pad-recess area must be smaller than the individual total piston area to ensure intimate contact at any pump operating speed.
- The pad-recess width must be chosen in conjunction with the number of ring-bearing element flow restrictors to ensure constant flow communication with at least one ring-bearing element for any wobbler displacement. This is best accomplished by graphical analysis.
- The total pad area must be such that the metal-to-metal bearing stress between pad sill and ring, as well as the sill PV factor, is within acceptable design limits for the materials used. Part of the applied load, as determined by Equation 9, is supported by the pad-recess pressure acting on the pad recess area. The sill bearing stress is calculated by

$$S_b = \frac{F_r - p_s A_m}{A_t - A_m} \qquad (39)$$

- The cross-sectional flow area of the port through the pad and the pad-recess flow area (product of recess depth and recess length) must be large enough to ensure negligible fluid-pressure drop under maximum flow through the ring-bearing element.

- The power required to overcome friction between the pads and ring must be within the capability of the system.
- The fatigue stress at the pad's neck section must be within acceptable limits for the pad material and design life.

CONCLUSION

This paper has presented the basic mathematical analysis and trade-off considerations required to design a compound hydrostatic bearing. The particular technique discussed has been applied and has yielded promising results.* It is hoped that the technique will facilitate studies by design engineers who wish to explore the feasibility of such a bearing in a specific application.

*Reference 1.

ACKNOWLEDGEMENT

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Nomenclature

A_b = Total bearing area of a single ring-bearing element, in.^{2}

A_m = Area of a single pad recess, in.^{2}

A_p = Total piston area of an individual piston/pad assembly, in.^{2}

A_r = Area of a single ring recess, in.^{2}

A_s = Sill area associated with a single ring recess, in.^{2}

A_t = Total pad projected area, in.^{2}

a_f = Bearing load coefficient

C_d = Orifice coefficient of discharge

C_p = Specific heat of pump fluid, BTU/lb/°F

C = Distance from center of ring to pad pivot point, in.

D = Ring outside diameter, in.

d = Eccentricity of ring with respect to pump cylinder block, in.

d_o = Orifice diameter, in.

e = Eccentricity of ring with respect to wobbler, in.

F_p = Net radial load acting on a single piston/pad assembly, lb.

F_r = Reactive load exerted by the ring on a single piston/pad assembly, lb.

g = Gravitational acceleration, 386 in./sec.^{2}

HP_s = Total power to overcome ring viscous frictional drag, horsepower

HP_{sj} = Power to overcome viscous friction for a particular ring-bearing element, horsepower

h_j = Radial clearance between ring and wobbler at a particular bearing element, in.

h_o = Average radial clearance between ring and wobbler, in.

J = Mechanical equivalent of heat, 9336 in·lb/BTU

L = Ring axial length, in.

N = Pump cylinder-block rotational speed, rpm

n_p = Total number of piston/pad assemblies

n_r = Total number of ring recesses

O_{CB} = Pump cylinder-block axis of rotation

O_R = Ring axis of rotation

O_W = Center of wobbler bore

p_1 = Pump discharge pressure, psi

Nomenclature (continued)

p_2 = Pump inlet pressure, psi

p_s = Fluid supply pressure available at any given ring-bearing element, psi

p_{rj} = Fluid pressure within a single particular ring recess, psi

Q = Total fluid volume flow, in.^{3}/sec.

Q_{oj} = Fluid volume flow through a single particular orifice, in.^{3}/sec.

Q_{sj} = Fluid volume flow over a single particular sill area, in.^{3}/sec.

q_f = Bearing flow coefficient

R = Resultant radial load applied to ring, lb.

r = Distance from cylinder block axis of rotation to pad pivot point, in.

ΔT_{AV} = Average increase in fluid temperature entering recess over fluid temperature in pump, °F

ΔT_j = Increase in temperature of fluid entering a particular recess over the fluid temperature in the respective pump cylinder, °F

V = Relative sliding velocity between ring and wobbler, in./sec.

W = Piston/pad assembly weight, lb.

W_j = Hydrostatic radial force exerted on a particular ring-bearing element, lb.

W_{jR} = Component of W_j in the direction of the resultant applied ring load, lb.

W_R = Net hydrostatic force acting in the direction of the resultant applied ring load, lb.

x = Ring recess axial length, in.

y = Ring recess width, in.

α = Angle defining the direction of the piston/pad reactive load

β_j = Angle defining the location of a particular ring-bearing element, degrees

λ = Ratio of single ring-recess area to single ring-bearing element area

ϕ = Angle defining the direction of the resultant applied radial load on ring, degrees

θ = Angular displacement of pump cylinder block, degrees

θ_i = Angle defining the position of a particular piston/pad assembly, degrees

ρ = Mass density of fluid, lb · sec.^{2}/in.^{4}

μ = Fluid absolute viscosity, lb · sec./in.^{2}

$\frac{d\theta}{dt}$ = Cylinder-block rotational speed, radians/sec.

$\frac{d^2r}{dt^2}$ = Piston/pad assembly radial acceleration, in./sec.^{2}

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