EXPERIMENTAL INVESTIGATION OF THE INFLUENCE OF ECCENTRICITY ON END LEAKAGE OF A FULL JOURNAL BEARING

by

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TABLE OF CONTENTS

THEORETICAL INVESTIGATION

Boyd and Robertson,[1*] applying a series method to the differential equation for hydrodynamic flow, developed the following equation for lubricant flow in a concentric bearing.

$$Q_B = 1.875 \times 10^{-4} \frac{P_i \, (2\,c')^3}{z_1 \left[\frac{l'}{D} - 2 \ln \frac{d}{D} - 4 \sum_{n=1}^{n=\infty} \frac{1}{n(1 + e^{\frac{2nl'}{D}})} \right]} \qquad (1)$$

(Terms are identified in Appendix I)

The ratio R_B, of the flow with eccentricity ratio ϵ, to the flow with zero eccentricity, as suggested by Boyd and Robertson, is expressed by an approximate formula,

$$R_B = \frac{(1 + \epsilon)^3 \left[\frac{l'}{D} - 2 \ln \frac{d}{D} \right]}{\left[\frac{l'}{D} - 2 \ln \frac{d}{D} + 3\epsilon \left(\frac{l'}{D} + 2 - \ln 16 \right) \right]}$$

These equations are not recommended for eccentricity ratios greater than 0.5.

To express the end leakage as a proportion of the volume of lubricant flowing through the clearance space of a concentric journal bearing, Needs[2] determined the constant of proportionality by using an electrical analogy to solve the differential equation for hydrodynamic flow. The equation established is,

$$Q = k.\ u.\ r.\ m.\ l \qquad (2)$$

The values of k at different eccentricity ratios for various $\frac{l}{D}$ ratios are tabulated in Table 2, Appendix III. This work was done for 120° journal bearing. However, Fuller[3] suggests that these values can also be used for bearings with other arc lengths.

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*Superscripts refer to bibliography references.

Shaw and Macks[4] developed expressions for the end leakage through journal bearings by consideration of the predominant effect of supply pressure on flow. It is assumed that the rate of lubricant flow through loaded journal bearings depends mainly on the flow through the regions where the thickness of the oil film is relatively large and pressure in the film is relatively low.

The expression developed by Shaw and Macks (See Appendix II) for oil flow in a full journal bearing with a single inlet hole is,

$$Q = \frac{c^3 p_i}{3\mu} \left[\tan^{-1}\left(\frac{2\pi r}{l}\right)\right] (1 + 1.5\epsilon^2) \qquad (3)$$

These three equations differ considerably in form and also in the calculated flow rate (Fig. 2). Because of this, an experimental investigation was made.

EXPERIMENTAL INVESTIGATION

Test Equipment:

The test fixture is shown in Fig. 1. A nominal diameter of 1 3/4 inches was selected for journal and bearing. A dial indicator having 0.0001 inch divisions was used to measure the eccentricity between the journal and the bearing. In usual cases, the bearing is fixed and the journal rotates inside it, with eccentricity depending on the load. In this experiment, the shaft was kept fixed and the bearing was movable. The details of the apparatus are shown in Fig. 6, Appendix III.

Procedure:

Oil used for the experiment was Mobile No. 797 D.T.E. The viscosity of the oil, measured by Saybolt Universal Viscometer, was 7.6850×10^{-6} Reynolds at 75°F. Lubricating oil was introduced through tubing from an overhead tank at a height of 3 feet above the bearing center. The oil flow corresponding to concentric position as well as different eccentric ratios was determined by measuring the volume of liquid required to maintain the liquid level in the tank for each experiment. Fifteen minute test runs were made at each eccentricity ratio. Two tests were run at each setting to check reproducibility of results. The duration of each run was such that the flow would be sufficient for good accuracy in determining the flow rates.

The experimental data recorded during the experiment are tabulated below.

Fig. 1. Test setup.

Table 1

Observation Number	Time Interval, Minutes	Eccentricity Ratio	Flow Milliliters	Flow Rate cu. in./min.
1	15	0	56.0	0.228
2	15	0.25	62.5	0.254
3	15	0.50	76.0	0.309
4	15	0.75	102.5	0.419
5	15	1.0	140.0	0.569

DISCUSSION

Figure 2 compares the flow rate obtained experimentally with that calculated by the equations proposed by Boyd and Robertson, Needs, and Shaw and Macks. The data are tabulated in Table 3, Appendix III.

The experimental curve indicates that the end leakage increases with increasing eccentricity. For eccentricity ratios from zero to 0.25, the flow rate increases by about 10 per cent. At eccentricity ratio 0.5, the flow rate attains a value of 35 per cent more than that at zero eccentricity. For eccentricity ratio 0.75, the flow rate becomes 1.85 times the initial and for maximum attainable eccentricity ratio 1, the flow rate is about 2.5 times the end leakage rate with zero eccentricity. This shows that the end leakage rate increases more rapidly at the range of high eccentricity ratio than at low eccentricity. This means the flow rate will increase as the load increases which is desirable because of the increased cooling requirement. Frictional losses are directly proportional to the load on the bearing.

Graph of theoretical results on the basis of Shaw and Macks' equation is found to be parabolic and the flow rate increasing with increasing eccentricity. This equation is derived on the assumption that the oil supply pressure has a predominant effect on the end leakage. Calculated flow rates very nearly parallel experimental values, although slightly higher.

Calculated results for eccentricity ratios from zero to 0.5, on the basis of Boyd and Robertson's formulae were plotted. The portion of the curve from eccentricity ratio 0.5 to 1 could not be plotted, since this equation is not applicable for eccentricity ratios greater than 0.5. For

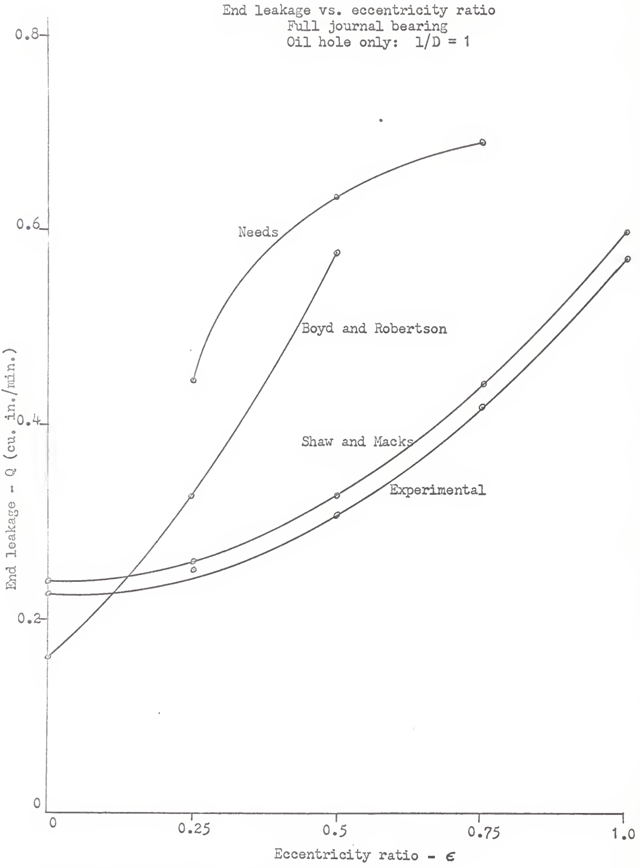

Fig. 2. End leakage vs. eccentricity ratio.

eccentricity ratios from zero to 0.25, flow rates are of the same order of magnitude as the experimental. However, the predicted flow rate increases much more rapidly with higher eccentricity than observed experimentally. This indicates that the end leakage evaluated on the basis of Boyd and Robertson's equation is unsuitable for operating conditions of low speed and low inlet pressure. As suggested by Boyd and Robertson, their predictions may be better for high speed with high inlet pressure.

The plot of results of Needs' formula suggests the highest flow rates and does not agree with any of the above curves. Needs' formula takes into account hydrodynamic pressure distribution in the bearing in evaluating k. Moreover, the values of k used for calculations, were determined for 120 degree partial bearings. This may be the reason why Needs' results do not agree with experimental data, although Fuller maintained the values were satisfactory for other than 120 degree bearings.

It may be concluded that for bearings with $\frac{l}{D}$ ratio 1, operating at low speed and low inlet pressure, the end leakage calculated using Shaw and Macks' analysis is satisfactory.

Several factors may have contributed to the experimental data being smaller than Shaw and Macks' theoretical predictions by essentially a constant magnitude. The phenomenon of surface tension at the sides of the ends of the bearing, causes the formation of a meniscus, which has a sealing effect on the flow of the lubricant out of the bearing. This sealing mechanism causes the lubricant to flow backwards along the sides of the bearing to re-enter in the low-pressure region to be recirculated. Another phenomenon is the formation of a vena-contracta around the annular outlet which restricts the free outflow of the oil, thus making the end leakage less than the predicted value.

Any misalignment in the bearing assembly would have an effect on the flow characteristics inside the bearing. Under conditions of operation at higher eccentricity ratios, the bearing and journal surfaces are liable to elastic deformation. This produces a different film shape and causes a change in the bearing performance. The surface roughness of the journal and bearing materials, which may be of the same order of magnitude as the minimum clearance under heavy loading, would also influence the end leakage.

Shaw and Macks' equation was developed on the assumption that the oil hole is placed at the unloaded region. In this experiment, the oil hole in the bearing was located just on the dividing line between the loaded and the unloaded region. The internal pressure in the bearing might cause a reduced flow into the bearing.

SUMMARY AND RECOMMENDATIONS

An experimental investigation of the effect that eccentricity has on end leakage through a full journal bearing under steady load, was conducted to provide data for comparison with analytical solutions. A 1 3/4 inch nominal diameter journal bearing with 0.004 inch radial clearance was tested. A length-diameter ratio of 1 and a speed of 140 rpm was used. The eccentricity or the relative displacement of the shaft in the bearing clearance, was varied from zero to the maximum obtainable in a bearing.

End leakage at different eccentricity ratios was calculated using three different theoretical approaches. 1) Boyd and Robertson's approximate formulae. 2) Needs' analysis on the basis of a solution of Reynolds' equation by using an electrical analogy. 3) Shaw and Macks' analysis on the assumption of stationary journal bearing with predominant effect of inlet pressure on the flow.

Comparison of the results of the theoretical approaches with the experimental data indicates that Shaw and Macks' equation more nearly agrees with the experimental findings.

This experiment was limited in scope and it is recommended that the following areas be considered as regions for further checking of theoretical predictions by experimental investigations.

(1) For the same rotational speed and inlet pressure as used in this experiment, the relationship between eccentricity ratio and the end leakage can be determined for various $\frac{l}{D}$ ratios. Ratios from 0.5 to 4 would be sufficient to cover the normal range used.

(2) For constant $\frac{l}{D}$ ratio and rotational speed, the effect of inlet pressure on oil flow can be studied.

(3) An experimental investigation can also be made to determine the effect speed will have on the oil flow.

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APPENDICES

APPENDIX I

Nomenclature

c - bearing radial clearance in inches.

c' - bearing radial clearance in milli-inches.

d - diameter of oil supply hole in inches.

D - diameter of bearing in inches.

e - bearing eccentricity in inches.

h - film thickness in inches. (Subscript denotes position.)

k - Needs' constant.

l - axial length of bearing in inches.

l' - half the axial length of bearing in inches.

m - clearance modulus (Ratio of radial clearance to radius of journal.)

N - shaft rotational speed in rpm.

p - pressure at any point.

p_i - oil supply pressure in psi.

Q - lubricant flow rate in cubic inches per minute.

Q_B - lubricant flow rate in gallons per minute using Boyd and Robertson's equation.

R_B - ratio of flow with certain eccentric ratio to the flow with no eccentricity.

u - velocity of lubricant along x direction.

v - velocity of lubricant along y direction.

w - velocity of lubricant along z direction.

x - coordinate along the circumference of the developed surface of bearing or journal.

y - coordinate along the normal to the developed surface.

z - coordinate along the axes of the bearing.

z_1 - viscosity of lubricant in centipoise.

μ - viscosity of lubricant in Reynolds.

ϵ - eccentricity ratio (Ratio of eccentricity to the radial clearance.)

γ - shear stress.

APPENDIX II

The expression for end leakage of a journal bearing having a central oil supply groove will be derived first and then by analogy, the expression for a hole only will be obtained.

Figure 3(a) represents an eccentric journal bearing. Coordinates x and y have been shown at the point of maximum radial clearance, along the circumferential and radial direction, respectively. Figure 3(b) shows the developed view of the bearing surface with the central oil groove.

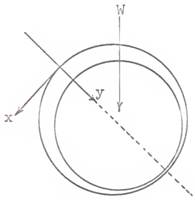


Fig. 3(a)
Eccentric journal bearing.

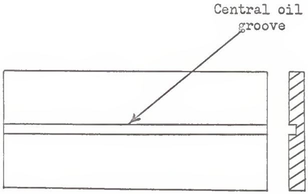


Fig. 3(b)
Developed view of bearing surface.

Figure 4(a) shows the general shape of the oil film for one-half of the bearing length with the coordinate axes. Figure 4(b) is a free body diagram of a small elemental volume of oil.

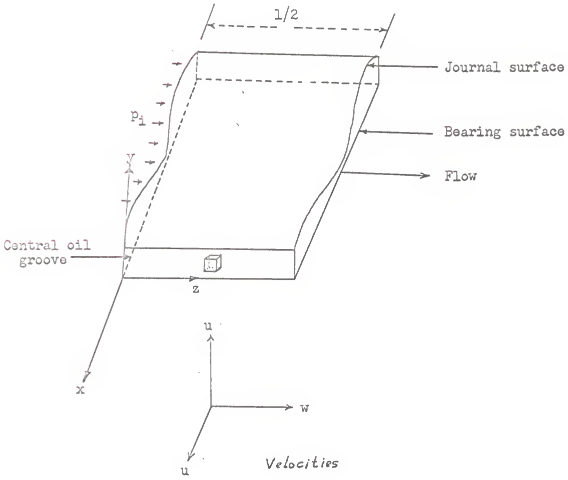

1/2
Journal surface
Bearing surface
Flow
p_i
y
Central oil groove
z
x
u
w
u
Velocities

Fig. 4(a)

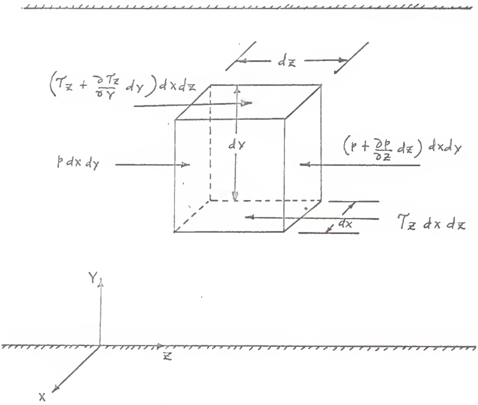


Fig. 4(b)

Static equilibrium of the element requires

$$\left(\tau_z + \frac{\partial \tau_z}{\partial y} dy\right) dx dz - \tau_z \, dx \cdot dz - \left(p + \frac{\partial p}{\partial z} dz\right) dx dy + p \, dx dy = 0$$

or, $$\frac{\partial \tau_z}{\partial y} dx dy dz - \frac{\partial p}{\partial z} dx dy dz = 0$$

or, $$\frac{\partial p}{\partial z} = \frac{\partial \tau_z}{\partial y} \qquad (1)$$

By Newton's law of viscous flow,

$$\frac{\partial p}{\partial z} = \mu \frac{\partial^2 \omega}{\partial y^2} \qquad \text{(page 191 equation 6-7, Shaw and Macks)}$$

$$\therefore \quad \frac{\partial^2 \omega}{\partial y^2} = \frac{1}{\mu} \frac{\partial p}{\partial z}$$

Integrating with respect to y

$$\frac{\partial \omega}{\partial y} = \frac{1}{\mu} \frac{\partial p}{\partial z} y + c_1$$

or, $$\omega = \frac{1}{\mu} \frac{\partial p}{\partial z} \frac{y^2}{2} + c_1 y + c_2$$

$\omega = 0$ when $y = 0$ so $c_2 = 0$.

$\omega = 0$ when $y = h$

$$\therefore 0 = \frac{1}{2\mu} \frac{\partial p}{\partial z} h^2 + c_1 h$$

or, $$c_1 = -\frac{1}{2\mu} \frac{\partial p}{\partial z} h$$

so, $$\omega = \frac{1}{\mu} \frac{\partial p}{\partial z} \frac{y^2}{2} - \frac{1}{2\mu} \frac{\partial p}{\partial z} h y$$

$$= \frac{1}{2\mu} \frac{\partial p}{\partial z} (y - h) y \qquad (2)$$

Quantity of liquid flowing across an elementary area is

$$dQ = \omega \, dx \, dy$$

$$= \frac{1}{2\mu} \frac{\partial p}{\partial z} (y - h)y \; dy \, dx$$

For a journal bearing,

$$h = C(1 + \epsilon \cos\theta)$$

and $dx = r d\theta$.

$$\therefore \quad dQ = \frac{r}{2\mu} \frac{\partial p}{\partial z} \left[y^2 dy d\theta - hy dy d\theta \right]$$

$$= \frac{r}{2\mu} \frac{\partial p}{\partial z} \left[y^2 dy d\theta - C(1 + \epsilon \cos\theta) y dy d\theta \right]$$

$$Q = \int_0^{2\pi} \int_0^{h} \frac{r}{2\mu} \frac{\partial p}{\partial z} \left[y^2 dy dQ - c(1 + \epsilon \cos\theta) y dy d\theta \right]$$

$$= \frac{r}{2\mu} \frac{\partial p}{\partial z} \int_0^{2\pi} \int_0^{h} \left[y^2 - C(1 + \epsilon \cos\theta) y \right] dy d\theta$$

Integrating with respect to y gives,

$$Q = \frac{r}{2\mu}\frac{\partial p}{\partial z}\int_0^{2\pi}\left[\frac{1}{3}h^3 - \frac{c}{2}(1 + \epsilon\cos\theta)h^2\right]d\theta$$

But $h = c(1 + \epsilon\cos\theta)$

Substituting,

$$Q = \frac{r}{2\mu}\frac{\partial p}{\partial z}\int_0^{2\pi}\left[\frac{1}{3}c^3(1 + \epsilon\cos\theta)^3 - \frac{c^3}{2}(1 + \epsilon\cos\theta)^3\right]d\theta$$

$$= -\frac{rc^3}{12}\frac{\partial p}{\partial z}\int_0^{2\pi}(1 + \epsilon\cos\theta)^3\, d\theta$$

$$= -\frac{rc^3}{12}\frac{\partial p}{\partial z}\left[\int_0^{2\pi}(1 + 3\epsilon^2\cos^2\theta)d\theta + \int_0^{2\pi}(3\epsilon\cos\theta + \epsilon^3\cos^3\theta)d\theta\right]$$

Second integral in the bracket is

$$\int_0^{2\pi}(3\epsilon\cos\theta + \epsilon^3\cos^3\theta)d\theta = 3\epsilon\left[\sin\theta\right]_0^{2\pi} + \frac{\epsilon^3}{4}\left[3\sin\theta + \frac{\sin 3\theta}{3}\right]_0^{2} = 0$$

$$\therefore Q = -\frac{rc^3}{12\mu}\frac{\partial p}{\partial z}\left[\int_0^{2\pi}(1 + 3\epsilon^2\cos^2\theta)d\theta\right]$$

$$= -\frac{rc^3}{12\mu}\frac{\partial p}{\partial z}\left\{\left[\theta\right]_0^{2\pi} + 3\epsilon^2\left[\theta + \frac{\sin 2\theta}{2}\right]_0^{2}\right\}$$

$$= -\frac{rc^3}{12\mu}\frac{\partial p}{\partial z}\left[2\pi + 3\epsilon^2\pi\right]$$

$$= -\frac{\pi rc^3}{6\mu}\frac{\partial p}{\partial z}(1 + \frac{3}{2}\epsilon^2) \qquad (3)$$

But $\frac{\partial p}{\partial z} = \frac{2\mu\omega}{y(y - h)}$ (equation 2)

Integrating with respect to z and evaluating the constant of integration by noting that $p = p_i$ when $z = 0$, where p_i is the supply pressure. Then

$$p = \frac{2\mu\omega}{y(y - h)}z + p_i$$

$$\therefore (p - p_i) = \frac{2 \mu \omega}{y(y - h)} z$$

When $p = 0$, $Z = 1/2$.

Then, $$-p_i = \frac{2 \mu \omega}{y(y - h)} \frac{1}{2}$$

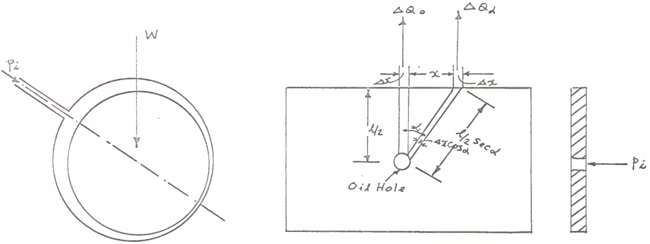
$$\therefore \frac{2 \mu \omega}{y(y - h)} = \frac{\partial p}{\partial z} = - \frac{2p_i}{l}$$

$$\therefore Q = \frac{\pi r c^3 p_i}{3 \mu l} (1 + 1.5 \epsilon^2)$$

For both sides of the bearing

$$Q = \frac{2 \pi r c^3 p_i}{3 \mu l} (1 + 1.5 \epsilon^2) \qquad (4)$$

OIL FLOW WITH SINGLE HOLE.



a) Eccentric journal bearing.

b) Developed view of bearing surface.

Fig. 5.

From equation 4, the flow per unit circumferential length through a concentric bearing with a central circumferential oil groove is seen to be

$$Q' = \frac{c^3 \, p_i}{3\mu l}$$

From the above figure, for width Δx, $Q_o = \frac{c^3 \, p_i}{3\mu l} \Delta x$

At any other point along the bearing, to a first approximation,

$$\Delta Q_\alpha = \frac{c^3 \, p_i}{3\mu l \sec\alpha} \Delta x \cos\alpha = \frac{c^3 \, p_i}{3\mu l} \Delta x \cos^2\alpha$$

$$\text{or, } \lim_{\Delta x \to 0} \Delta Q_\alpha = dQ = \frac{c^3 \, p_i}{3\mu l} \cos^2\alpha \; dx$$

$$\therefore Q = 2 \int_0^{\pi r} \frac{c^3 \, p_i}{3\mu l} \cos^2\alpha \; dx$$

$$= \frac{c^3 \, p_i}{3\mu l} \int_0^{\pi r} \frac{l^2}{4x^2 + l^2} \, dx$$

Putting $x = l/2 \tan\phi$

$$dx = l/2 \sec^2\phi \; d\phi$$

when $x = 0, \quad \phi = 0$

$x = \pi r \quad \phi = \tan^{-1} \frac{2\pi r}{l}$

$$\therefore \frac{l^2}{4x^2 + l^2} = \frac{l^2}{l^2 \tan^2\phi + l^2} \frac{l}{2} \sec^2\phi \; d\phi = \frac{l}{2} \, d\phi$$

$$\therefore Q = \frac{2c^3 p_i}{3\mu l} \frac{l}{2} \int_0^{\tan^{-1} \frac{2\pi r}{l}} d\phi$$

$$= \frac{c^3 p_i}{3\mu} \tan^{-1} \left(\frac{2\pi r}{l} \right)$$

Shaw and Macks assume that the flow from a bearing fed by a single oil hole varies in the same manner with attitude as does the flow through a bearing fed by a central oil groove. Then,

$$Q = \frac{c^3 p_i}{3\mu} \left[\tan^{-1} \left(\frac{2\pi r}{l} \right) \right] (1 + 1.5\epsilon^2) \qquad (5)$$

APPENDIX III

Following Shaw and Macks' analysis, the flow rates can be calculated at various eccentricity ratios as follows:

$$Q = c^3 p_i / 3\mu \ \tan^{-1}(2\pi r/l) \quad (1 + 1.5\epsilon^2)$$

$c = 0.004$ in.

$p_i = 3 \times 0.434 \times 0.87 = 1.132$ lbs/in^2

$\mu = 7.685 \times 10^{-6}$ Reyns.

$\epsilon = 0, 0.25, 0.5, 0.75$ or 1

$l = 1.75$ inch.

$\epsilon = 1.0$

$$Q = \frac{(0.004)^3 \times 1.132}{3 \times 7.685 \times 10^{-6}} \left[\tan^{-1} \left(\frac{2\pi \times 0.875}{1.75} \right) \right] (1 + 1.5 \times 1^2)$$

$$= 0.597 \text{ in}^3/\text{min.}$$

Results are tabulated in Table 3.

Following Needs' formula the end leakage at different eccentricity ratios can be calculated as follows:

$$Q = k \ u \ r \ m \ l$$

On interpolation from Table 2, for $\frac{l}{D}$ ratio 1, the values of k are found.

ϵ	k
0.75	0.1275
0.5	0.118
0.25	0.0825

$u = \ \times 1.75 \times 140$ in/min = 770 in/min.

$m = c/r = 0.004/0.875$

$r = 0.875$

$l = 1.75$

$\epsilon = 0.75$

$Q = 0.1275 \times 770 \times 1.75/2 \times 0.004/0.875 \times 1.75$

$= 0.687\ in^3/min.$

End leakage on the basis of Boyd and Robertson's equation. For concentric journal bearing.

$$Q_B = 1.875 \times 10^{-4} \frac{p_i(2c')^3}{z_1\left[\frac{l'}{D} - 2\ln\frac{d}{D} - 4\sum_{n=1}^{n=\infty}\frac{1}{n\left(1 + e^{\frac{2nl'}{D}}\right)}\right]}$$

$p_i = 1.132$ psi

$c' = 4$ in.

$l' = 0.875$ in.

$z_1 = 53$ cp

$D = 1.75$ inch.

$n =$ the series 1, 2, 3, 4, ...

$$Q_B = \frac{1.875 \times 10^{-4} \times 1.132 \times 64 \times 8}{53(\frac{1}{2} + 3.89 - 1.4)}\ gal/min.$$

$= 6.86 \times 10^{-4}$ gal/min.

$= 0.1588\ in^3/min.$

$$R_B = \frac{(1 + \epsilon)^3\left[\frac{l'}{D} - 2\ln\frac{d}{D}\right]}{\left[\frac{l'}{D} - 2\ln\frac{d}{D} + 3\epsilon\left(\frac{l'}{D} + 2 - \ln 16\right)\right]}$$

For $\epsilon = 0.25$, $R_B = 2.06$

$Q = Q_B \times R_B = 0.1588 \times 2.06 = 0.326\ in^3/min.$

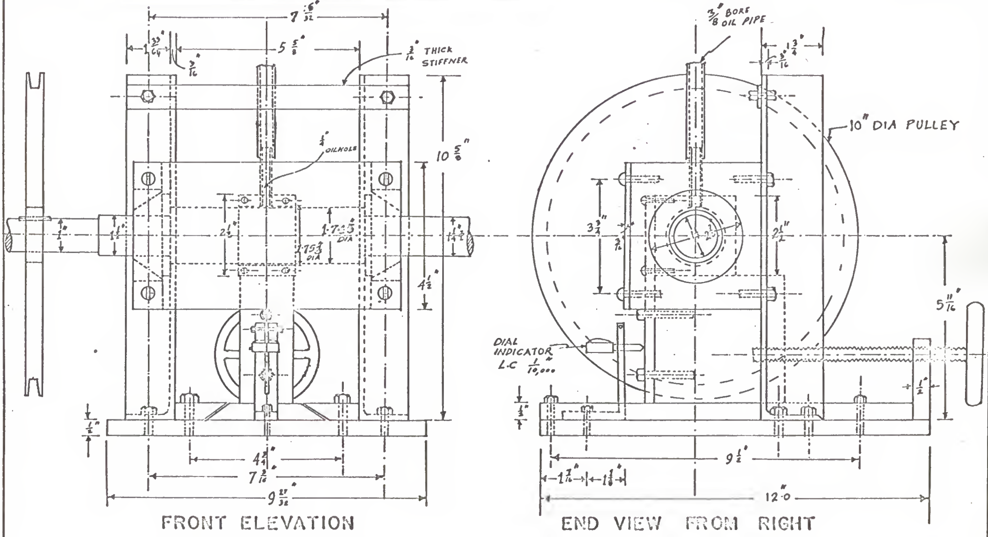
Boyd and Robertson do not recommend using the formula for $\epsilon > 0.5$.

Table 2. Values of k for Needs' formula for end leakage.

ϵ 1/D	0.9	0.8	0.6	0.4	0.2
0.25	0.15	0.22	0.234	0.192	0.096
0.5	0.18	0.225	0.195	0.186	0.104
0.75	0.13	0.153	0.162	0.133	0.870
1.0	0.109	0.126	0.132	0.104	0.075
1.25	0.09	0.108	0.11	0.086	0.064
1.50	0.078	0.093	0.096	0.072	0.054
2.0	0.054	0.069	0.078	0.054	0.042
2.5	0.04	0.051	0.06	0.040	0.03
3.0	0.027	0.034	0.039	0.025	0.021
4.0	0.0	0.0	0.0	0.0	0.0

Table 3. Flow rates based on various formulas and experimental data.

Eccentric ratio	Flow rate based on experimental investigation $in^3/min.$	Flow rate based on Needs' formula $in^3/min.$	Flow rate based on Boyd and Robertson's formula $in^3/min.$	Flow rate based on Shaw and Macks formula $in^3/min.$
0	0.228	-	0.1588	0.239
0.25	0.254	0.444	0.326	0.261
0.5	0.3085	0.635	0.575	0.328
0.75	0.419	0.687	-	0.440
1.0	0.569	-	-	0.597



TEST APPARATUS ASSEMBLY

Fig. 6.

SHAFT DIA = 1.745
BEARING DIA = 1.753

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by

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AN ABSTRACT OF A MASTER'S REPORT

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1964

The object of this report was to determine experimentally the influence of eccentricity on the end leakage of a full journal bearing operating under low inlet pressure and low rotational speed and subsequently to compare the experimental data with theoretical predictions.

The experiment was conducted on a mild steel journal and bearing, 1 3/4 inch nominal diameter, with radial clearance of 0.004 inch and $\frac{l}{D}$ ratio 1. Mobiloil No. 797 D.T.E. with viscosity 7.685×10^{-6} Reynolds at 75°F was used. The rotational speed of the journal was 140 rpm and oil was passed into the bearing through an inlet hole of 1/4 inch diameter with an inlet pressure of 1.132 lbs/in^2. A dial indicator graduated in 0.0001 inch was used to measure the eccentricity of the journal.

The oil flow data were compared with the theoretical results obtained from three different approaches.

The investigations showed that under the operating conditions stated above, the end leakage of a full journal bearing closely agrees with the analysis of Shaw and Macks for a journal bearing with single inlet hole.

The predictions of Boyd and Robertson, and Needs did not agree with experimental results or with each other.