ANALYSIS OF THE GAS-LUBRICATED FLAT-SECTOR-PAD THRUST BEARING

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A flat sector-shaped pad geometry for a gas-lubricated thrust bearing is analyzed considering both the pitch and roll of the pad. It is shown that maximum load capacity is achieved when the pad is tilted so as to create uniform minimum film thickness along the pad trailing edge. Performance characteristics for various geometries and operating conditions of gas thrust bearings are presented in the form of design curves, and a comparison is made with the rectangular slider approximation. It is found that this approximation is unsafe for practical design, since it always overestimates load capacity.
ANALYSIS OF THE GAS-LUBRICATED FLAT-SECTOR-PAD

THRUST BEARING

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SUMMARY

A flat sector-shaped pad geometry for a gas-lubricated thrust bearing is analyzed considering both the pitch and roll of the pad. The analysis uses only two parameters (rather than the conventional four) to completely describe pad pitch and roll about a certain point. This was achieved by transforming the pitch and roll about the point to a corresponding pure pitch about a certain radial pivot line. This enables performance optimization to be based on the true geometry. Maximum load capacity is achieved when the pad is tilted so as to create uniform minimum film thickness along the pad trailing edge. Performance characteristics for sector angles of $30^\circ$, $45^\circ$, and $60^\circ$ and radius ratios of 0.3, 0.5, and 0.7 are presented, over the range from almost incompressible lubrication to a compressibility number as high as 100. A comparison is made with the rectangular approximation. It is found that this approximation is unsafe for practical design, since it always overestimates load capacity.

INTRODUCTION

Of all the existing thrust bearing configurations the flat sector pad (fig. 1) is the simplest to produce and probably the most commonly used for practical applications. It is therefore astonishing to find that so little has been published in the literature on this sort of bearing.

Taylor and Saffman (ref. 1) present an analytical solution for a flat circular plate misaligned about a radial line and spinning opposite a fixed stator disk that is perpendicular to the axis of rotation. Although the actual film thickness distribution in the

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lubricating film was used, the solution is inapplicable to thrust bearings that consist of individual flat sector pads. While for the circular plate the film is converging-diverging, for the sector pad it is usually wholly converging. Gross (ref. 2) in his book presents a solution, in the form of an infinite series, for a sector pad with exponential film thickness distribution. An approximate solution then follows in which he assumes that the film thickness is independent of radius. This type of bearing, known as a Michell bearing, is by no means an actual flat sector pad. For a flat sector pad tilted about any point the clearance varies in both the radial and circumferential directions, with the circumferential variation being sinusoidal rather than linear.

In two other references (refs. 3 and 4) the sector-shaped pad is approximately by a rectangular slider. The transformation from a sector shape is made by assuming that the rectangular width is equal to the mean circumference of the sector and that the length is equal to the difference between the inner and outer radii. This approximation not only distorts the sector shape, but also introduces linear film thickness and velocity profiles that are independent of radius.

Extensive work on pad bearings has been done by Shapiro and Colsher (ref. 5). They investigated the crowned sector pad and included optimization criteria, design information, and procedures. The flat pad was treated as a particular case of zero crown height. Although the problem was generally posed, solutions were limited to a pad inclined about a radial line through the center of the pad.

In the course of developing a new type of gas-lubricated thrust bearing (ref. 6), it was found that solutions that cover a wide range of flat-sector-pad operating conditions are desired. Such solutions are not available in the open literature. It is the intent of this report to provide the necessary data for design of flat-sector-pad configurations. A solution will be sought for bearings of various pad sector angles and radius ratios operating over a wide range of compressibility numbers. Bearing load capacity, center-of-pressure location, and friction loss will be given.

**ANALYSIS**

**Film Thickness Distribution**

One reason for approximating the sector-shaped pad as a rectangular slider is the large number of parameters involved in the exact solution. The film thickness distribution for a tilted pad is defined by four parameters. Two of them describe the location of the pivot, while the other two measure the amount of tilt about radial and tangential lines through the pivot point, referred to as pitch and roll.

Figure 2 shows a flat sector pad tilted about some point \( O' \) with a pitch \( \alpha_r \) and
roll $\alpha_{\theta}$. (All symbols are defined in the appendix.) If the pad is initially parallel to some plane at a distance $h'_o$, the expression for the clearance at any point $(r, \theta)$ is

$$h = h'_o + \alpha_r r \sin(\theta' - \theta) + \alpha_{\theta}[r' - r \cos(\theta' - \theta)]$$  \hfill (1)

Using the clearance at the center $O$ as a reference, we have

$$h = h_o + \alpha_r r \sin(\theta' - \theta) - \alpha_{\theta} r' \cos(\theta' - \theta)$$

where

$$h_o = h'_o + r' \alpha_{\theta}$$  \hfill (2)

A radial line at an angle $\theta_p$ can now be found along which the clearance is constant and equal to $h_o$. This radial pivot line is defined by the equation

$$\alpha_r r \sin(\theta' - \theta_p) - \alpha_{\theta} r \cos(\theta' - \theta_p) = 0$$

where the angle $\theta_p$ is given by

$$\tan(\theta' - \theta_p) = \frac{\alpha_{\theta}}{\alpha_r}$$  \hfill (3)

The pitch about the pivot line is

$$\gamma = -\frac{1}{r} \left( \frac{dh}{d\theta} \right)_{\theta = \theta_p} = \alpha_r \cos(\theta' - \theta_p) + \alpha_{\theta} \sin(\theta' - \theta_p)$$  \hfill (4)

The preceding analysis shows that any tilt about some given point is equivalent to a pure pitch about a certain radial line. This can be easily understood by visualizing a plane parallel to the runner that goes through the origin of the sector (point $O$ in Fig. 2). The radial pivot line is the intersection between this parallel plane and the plane of the tilted sector. Equations (3) and (4) represent the proper transformation. By considering the clearance $h_o$ along this pivot line as a reference, we can give the film thickness at any point $(r, \theta)$ by

$$h = h_o + \gamma r \sin(\theta_p - \theta)$$  \hfill (5)

where $\gamma$ is the amount of tilt about the pivot line.
Describing the film thickness distribution in terms of $\theta_p$ and $\gamma$ reduces the number of involved parameters from four in equation (1) to two in equation (5). We normalize the clearance by $h_0$ and the radial coordinate by the outer radius $r_o$. The nondimensional film thickness then becomes

$$H = 1 + \varepsilon R \sin(\theta_p - \theta)$$  \hspace{1cm} (6)

where

$$\varepsilon = \gamma \frac{r_o}{h_0}$$

$$R = \frac{r}{r_o}$$

It will be instructive, prior to obtaining explicit solutions, to examine equation (5) in order to understand some features of the flat pad. When dealing with an actual flat pad, the location $\theta_p$ of the pivot line plays an important role in regard to the convergence or divergence of the fluid film. Differentiating equation (5) yields the circumferential and radial gradients

$$\frac{1}{r} \frac{dh}{d\theta} = -\gamma \cos(\theta_p - \theta)$$  \hspace{1cm} (7a)

$$\frac{dh}{dr} = \gamma \sin(\theta_p - \theta)$$  \hspace{1cm} (7b)

From this we can see that whenever $\gamma > 0$ the condition for circumferential clearance convergence is

$$-\frac{\pi}{2} < \theta_p - \theta < \frac{\pi}{2}$$

To meet this demand for any $\theta$ within the pad boundaries,

$$\theta_{\text{max}} - \frac{\pi}{2} < \theta_p - \frac{\pi}{2} < \theta_{\text{min}} + \frac{\pi}{2}$$

and since $\theta_{\text{max}} = \beta$ and $\theta_{\text{min}} = 0$ we have

$$\beta - \frac{\pi}{2} < \theta_p - \frac{\pi}{2}$$  \hspace{1cm} (8)
The range of pivot location for circumferential film thickness variation as described by expression (8) can be divided into three different regions corresponding to three different modes of radial clearance variation (fig. 3).

Region a: \( \beta - \pi/2 < \theta_p < 0 \). In this region the pivot line is ahead of the leading edge. For any \( \theta \) within the pad, \( \theta_p - \theta < 0 \). Hence, from equation (7b) it is clear that radial convergence exists all over the pad.

The maximum and minimum clearances are at the points \((r_i, 0)\) and \((r_o, \beta)\), respectively, and the clearance ratio is

\[
\frac{h_1}{h_2} = \frac{1 + \epsilon \frac{r_i}{r_o} \sin \theta_p}{1 + \epsilon \sin(\theta_p - \beta)}
\]  

Note that region a can exist only for \( \beta < \pi/2 \).

Region b: \( 0 \leq \theta_p \leq \beta \). For any \( \theta > \theta_p \) the film thickness still converges radially, but the portion of the pad between the leading edge and the pivot line \( (\theta < \theta_p) \) assumes radial divergence. The maximum and minimum clearances are at \((r_o, 0)\) and \((r_o, \beta)\), respectively. The clearance ratio is

\[
\frac{h_1}{h_2} = \frac{1 + \epsilon \sin \theta_p}{1 + \epsilon \sin(\theta_p - \beta)}
\]  

Region c: \( \beta < \theta_p < \pi/2 \). Here, radial divergence exists over the entire pad. The minimum clearance occurs at \((r_i, \beta)\), the maximum clearance occurs at \((r_o, 0)\), and the clearance ratio is

\[
\frac{h_1}{h_2} = \frac{1 + \epsilon \sin \theta_p}{1 + \epsilon \frac{r_i}{r_o} \sin(\theta_p - \beta)}
\]  

Solution of the Reynolds Equation

For isothermal films the compressible Reynolds equation in polar coordinates is

\[
\frac{\partial}{\partial r} \left( r \phi h^3 \frac{\partial p}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left( \phi h^3 \frac{\partial p}{\partial \theta} \right) = 6 r \omega \frac{\partial (\phi h)}{\partial \theta}
\]

(10)
To transform equation (10) into dimensionless form, the following substitutions are used:

\[ R = \frac{r}{r_o} \]
\[ p = \frac{P}{p_a} \]
\[ H = \frac{h}{h_o} \]
\[ \Lambda = \frac{6\mu \omega r^2}{p_a h_o^2} \]

With these transformations the Reynolds equation becomes

\[ \frac{\partial}{\partial R} \left( H^3 R P \frac{\partial P}{\partial R} \right) + \frac{1}{R} \frac{\partial}{\partial \theta} \left( H^3 P \frac{\partial P}{\partial \theta} \right) = \Lambda H^2 R \frac{\partial}{\partial \theta} (PH) \tag{11} \]

The boundary conditions are \( P = 1 \) on the boundaries of the pad.

Equation (11) can be written in terms of the variable \( Q \), where \( Q = (PH)^2 \) (ref. 7):

\[ \frac{\partial}{\partial R} \left[ R \left( \frac{H}{2} \frac{\partial Q}{\partial R} - Q \frac{\partial H}{\partial R} \right) \right] + \frac{1}{R} \frac{\partial}{\partial \theta} \left( \frac{H}{2} \frac{\partial Q}{\partial \theta} - Q \frac{\partial H}{\partial \theta} \right) = \frac{\Lambda H^2 R}{2\sqrt{Q}} \frac{\partial Q}{\partial \theta} \tag{12} \]

We now differentiate the left side and rearrange, noting that for the film thickness distribution as given by equation (6),

\[ \frac{\partial^2 H}{\partial R^2} = 0 \]

and

\[ \frac{\partial H}{\partial R} + \frac{1}{R} \frac{\partial^2 H}{\partial \theta^2} = 0 \]

The Reynolds equation then becomes
Equation (13) is expanded by finite differences and solved numerically by using the Gauss-Seidel iteration method. After the pressure distribution is known, the total load capacity is obtained from

\[ \overline{W} = \frac{W}{pa r^2_o} \int_{r_1/r_o}^{1} \int_{0}^{\beta} (P - 1)R d\theta dR \]  

(14)

The pad area is given by

\[ A = \frac{\beta}{2} \left( r^2_o - r^2_1 \right) = \frac{1}{2} \beta r^2_o \left[ 1 - \left( \frac{r_1}{r_o} \right)^2 \right] \]

and the nondimensional unit load of the bearing is

\[ \frac{W}{pa A} = \frac{\overline{W}}{2W} \beta \left[ 1 - \left( \frac{r_1}{r_o} \right)^2 \right] \]  

(15)

The nondimensional radial and angular coordinates of the center of pressure are given by

\[ R_{cp} = \frac{1}{W} \int_{r_1/r_o}^{1} \int_{0}^{\beta} (P - 1)R^2 d\theta dR \]  

(16a)

\[ \sin \theta_{cp} = \frac{1}{WR_{cp}} \int_{r_1/r_o}^{1} \int_{0}^{\beta} (P - 1)R^2 \sin \theta d\theta dR \]  

(16b)

The shear stress on the runner is

\[ \tau = \frac{\mu \omega R}{h} + \frac{h}{2r} \frac{\partial \theta}{\partial \theta} \]
and the power loss is

\[ F = \int \int \tau \omega R^2 \, d\theta \, dR \]

Defining

\[ \overline{F} = \frac{F}{p_a h_2 \omega r_0^2} \]

gives

\[ \overline{F} = \frac{1}{H_2} \int_{r_1/r_o}^{1} \int_{0}^{\beta} \left( \frac{\Lambda H_2^2 R}{6} + \frac{HR}{2} \frac{\partial P}{\partial \theta} \right) \, d\theta \, dR \]  \hspace{1cm} (17)

The previous equations are for one pad. When multiple pads are considered, each pad is handled separately and the results added.

RESULTS

Optimum Pivot Location and Bearing Performance

In total, 12 different geometries were analyzed. Inner to outer radius ratios were 0.3, 0.5, and 0.7 at pad angles of 30°, 45°, 60°, and 90°. Each pad was run at compressibility numbers from 1 to 100 with various pivot locations. Performance characteristics such as unit load, center-of-pressure location, and power loss were found as a function of clearance ratio and pivot location. Figure 4 is a typical plot describing unit load variation. Two important results are the existence of an optimum clearance ratio for maximum unit load and the effect of pivot location on the load capacity. The highest unit load is achieved when the radial pivot line is placed at the pad trailing edge. By shifting the pivot line from the leading to the trailing edge, the maximum available unit load for the particular pad of figure 4 is increased by 55 percent. In figure 5 the maximum available unit load of four different pads is plotted against pivot location. Figure 5(a) presents the low compressibility range (\( \Lambda = 10 \)), while figure 5(b) is for \( \Lambda = 100 \). It is clear that for the whole range of pad angles and operating conditions there is a sharp maximum at the pivot location \( \theta_P / \beta = 1 \). The same results were obtained for the other radius ratios. Thus, to get the maximum load capacity...
from a given pad, the pad should be tilted in a way that maintains a uniform minimum film thickness along the trailing edge.

A physical explanation for this is that the pressure buildup in the lubricating film is affected by the resistance to lubricant outflow. Some of this flow occurs along the inner and outer circumferences of the pad, but the larger portion leaks across the trailing edge. Hence it would be advantageous to decrease the escape area along this boundary. For the three regions of pivot location discussed in the analysis and for a given minimum film thickness \( h_2 \), the lowest escape area and hence the highest pressure buildup can be achieved by placing the pivot line on the trailing edge.

To examine the effect of pad geometry on load capacity, maximum available unit loads at \( \theta_p/\beta = 1 \) are presented in table I. The highest load is achieved at the lowest radius ratio. This means that for a given outer radius the lowest possible inner radius is desirable. Low radius ratio not only produces high unit load but further increases the load capacity by providing larger bearing area. From table I it is seen that at radius ratios of 0.7 the best bearing configuration of those computed is that consisting of 12 pads of \( 30^\circ \) each. The gain in maximum load capacity over the next best configuration, having \( \beta = 45^\circ \), is about 20 percent.

At a radius ratio of 0.5 there is only a slight difference between the \( 30^\circ \) and \( 45^\circ \) pads. At \( r_i/r_o = 0.3 \) the \( 60^\circ \) pad is slightly better than the \( 45^\circ \) up to \( \Lambda = 10 \) and gives 16 percent more load capacity than the \( 30^\circ \) configuration. At higher compressibilities, up to \( \Lambda = 100 \), the \( 45^\circ \) pad takes over, being slightly better than the \( 60^\circ \) at \( \Lambda = 25 \) and \( \Lambda = 50 \) and giving 11 and 7.5 percent more load than the \( 30^\circ \) pad at \( \Lambda = 25 \) and \( \Lambda = 50 \), respectively. At \( \Lambda = 100 \) the maximum available load of the \( 30^\circ \) and \( 60^\circ \) pads is the same, being only 4 percent less than that of the \( 45^\circ \) pad.

Figures 6 to 10 present performance characteristics as functions of the clearance parameter for \( \theta_p/\beta = 1 \) and various compressibility numbers. Note that for \( \theta_p/\beta = 1 \), \( h_o = h_2 \). Hence the clearance parameter is \( \epsilon = \gamma r_o/h_2 \). The clearance parameter \( \epsilon \) is considered to be more convenient for bearing construction than the film thickness ratio \( h_1/h_2 \). Equation (9b) gives the relation between \( h_1/h_2 \) and \( \epsilon \). Pad angles of \( 30^\circ \) and \( 45^\circ \) are considered, with the addition of the \( 60^\circ \) pad at \( r_i/r_o = 0.3 \). From figures 6(a) to (c) it can be seen that although at \( r_i/r_o = 0.3 \) and \( \Lambda < 25 \) the maximum available load capacity is achieved at \( \beta = 60^\circ \), the maximum load region is confined to a narrow range of clearance parameter, compared with the \( 30^\circ \) or \( 45^\circ \) configurations. On the other hand, the \( 60^\circ \) pad is advantageous because a smaller number of pads are required.

From figures 6(d) to (g) it is seen that for \( r_i/r_o = 0.5 \), like in the case of \( r_i/r_o = 0.3 \), again the \( 30^\circ \) pad has a wider range of clearance parameter around the maximum unit load. As a general rule the optimum pad angle decreases as the radius ratio and compressibility number increase. The optimum pad angle also appears to be that
which results in a pad having a mean circumferential length approximately equal to $r_o - r_i$, that is, a "square" pad.

In figures 7 to 9, the angular location of the center of pressure is presented. For all cases there is only a slight change with clearance parameter variation. This change is most pronounced at low compressibility numbers and decreases to less than 5 percent at $\Lambda = 100$. Loci of maximum load capacity are also shown; this information will be useful for tilting pad design.

Radial location of the center of pressure is much less sensitive to clearance parameter variation, and values to cover the whole range of operating conditions within an accuracy of 2 percent are given in table II. For a radius ratio of 0.7 the center of pressure is very close to the midradius of the pad. The center of pressure moves toward the outside as the radius ratio and pad angle decrease.

Figure 10 presents power loss coefficients for the various geometries and operating conditions. Power loss is shown in the form $F/W_oh_2$; thus the lower the value of $F/W_oh_2$ the higher the bearing efficiency. Around the values of clearance parameter for maximum load the power loss coefficient is not much affected by changes in the compressibility number for $\Lambda$ up to 50. The efficiency drops at higher compressibility numbers, being about 20 percent less at $\Lambda = 100$ than at $\Lambda = 1$. Pad angle $\beta$ affects the efficiency similar to the way it affects the unit load. Hence, high efficiency is related to high load capacity. Again, the low radius ratio of 0.3 is the most efficient. This, combined with its higher load capacity, makes the low radius ratio a desirable feature in bearing design.

Comparison with Rectangular Approximation

Most of the published performance information on tilted-pad thrust bearings is for rectangular sliders. The approximation of a sector shape is made by assuming that the rectangular length is equal to the difference between the inner and outer radii of the sector and that the width and velocity are equal to the mean circumferential length and velocity of the sector. The equivalent compressibility number $\Lambda_R$ of the rectangular approximation becomes

$$\Lambda_R = \beta \left[ \frac{1 + \left( \frac{r_1}{r_0} \right)}{2} \right]^2 \Lambda$$
The sector-pad analysis of reference 5 was based on a pad inclination about a radial line through the center of the pad ($\theta_p/\beta = 0.5$). This is the most common technique in sector-pad bearing analyses and is based on the assumption that it best fits the rectangular approximation. However, as will now be shown this is not the case.

Table III compares the variation in unit load, at various clearance ratios, for two sector-shaped pads and their rectangular approximations. The first sector pad, having a radius ratio of 0.9, a pad angle of 24°, and a compressibility number of 26.5, corresponds to a rectangular slider with a length-width ratio of 0.25 and an equivalent compressibility number of 10. The second sector pad has $r_1/r_0 = 0.3$, $\beta = 62^\circ$ and $\Lambda = 22$ and is approximated by a rectangular slider having $L/B = 1$ and $\Lambda_R = 10$.

As can be seen from table III, the rectangular approximation always overestimates the unit load. The error increases as $r_1/r_0$ and $\theta_p/\beta$ become less than unity. For pad 1, having $r_1/r_0 = 0.9$ and $\theta_p/\beta = 1$, the results differ by less than 3 percent (when $h_1/h_2 \leq 3$); but the difference is 11 percent for $\theta_p/\beta = 0.5$. For pad 2, having $r_1/r_0 = 0.3$, which is a more practical ratio than $r_1/r_0 = 0.9$, the rectangular approximation overestimates the load by 12 to 35 percent for $\theta_p/\beta = 1$ and by as much as 60 percent when $\theta_p/\beta = 0.5$. The unit load for the rectangular approximation was reproduced from reference 8, and the close agreement with the results for pad 1 serves as a check on the validity of the present analysis.

Another error resulting from the rectangular approximation is the center-of-pressure location. The rectangular approximation assumes the midradius of the pad as the center-of-pressure location, thus producing errors as large as 10 percent of the pad length (see table II).

CONCLUDING REMARKS

The flat sector-shaped pad was analyzed by taking into account both the pitch and roll of the pad. The highest load capacity is achieved when the pad is tilted about its trailing edge, and for most operating conditions the best bearing configuration is that with a low radius ratio. The optimum pad angle depends on the radius ratio and compressibility number.

Errors resulting from the rectangular approximation were pointed out. These are overestimation of load capacity and mislocation of the center of pressure. Design curves for actual sector-shaped pads were presented to provide bearing unit load, center-of-pressure location, and power loss for various radius ratios, pad angles,
and a wide range of operating conditions from almost incompressible lubrication to compressibility numbers as high as 100.

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APPENDIX - SYMBOLS

A  pad area, \( \beta\left(r_o^2 - r_i^2\right)/2 \)

B  width of rectangular slider

F  power loss

\( \overline{F} \)  nondimensional power loss, \( F/pa \omega^2 h_0^2 r_0^2 \)

H  nondimensional film thickness, \( h/h_0 \)

h  film thickness

h_0  film thickness along pivot line

h'_0  film thickness at point where pad is tilted

h_1, h_2  maximum and minimum film thicknesses

L  length of rectangular slider (normal to direction of motion)

P  nondimensional pressure, \( p/p_a \)

p  pressure

p_a  ambient pressure

Q  \((PH)^2\)

R  nondimensional radius, \( r/r_0 \)

\( R_{cp} \)  nondimensional radius to center of pressure, \( r_{cp}/r_0 \)

r  radial coordinate

r_{cp}  radius to center of pressure

r_i  pad inner radius

r_o  pad outer radius

r'  radius to point where pad is tilted

W  pad load capacity

\( \overline{W} \)  nondimensional load, \( W/pa r_0^2 \)

\( \alpha_r \)  tilt about a radial line (pitch)

\( \alpha_\theta \)  tilt about a tangent line (roll)

\( \beta \)  angular extent of pad

\( \epsilon \)  clearance parameter, \( \gamma r_0/h_0 \)
\( \gamma \)  pitch about pivot line
\( \theta \)  angular coordinate, measured from leading edge
\( \theta' \) angle to point where pad is tilted
\( \theta_{\text{cp}} \) angle to center of pressure
\( \theta_p \) angle to pivot line
\( \Lambda \) compressibility number, \( 6\mu \omega r_0^2/p_ah_2^2 \)
\( \Lambda_R \) equivalent compressibility number for rectangular approximation
\( \mu \) viscosity
\( \tau \) shear stress on runner
\( \omega \) shaft speed
REFERENCES


### TABLE I. - MAXIMUM AVAILABLE UNIT LOAD FOR VARIOUS GEOMETRIES AND COMPRESSIBILITY NUMBERS

[Pivot location, $\theta_p/\beta$, 1.]

<table>
<thead>
<tr>
<th>Compressibility number, $\Lambda$</th>
<th>Ratio of pad inner radius to outer radius, $r_1/r_o$</th>
<th>0.3</th>
<th>0.5</th>
<th>0.7</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Angular extent of pad, $\beta$, deg</td>
<td>30</td>
<td>45</td>
<td>60</td>
</tr>
<tr>
<td>1</td>
<td>4.1</td>
<td>4.7</td>
<td>4.8</td>
<td>4.3</td>
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<td>10</td>
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<td>46</td>
<td>47</td>
<td>40</td>
</tr>
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<td>25</td>
<td>97</td>
<td>108</td>
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<td>50</td>
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<td>182</td>
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<tr>
<td>100</td>
<td>283</td>
<td>295</td>
<td>282</td>
<td>229</td>
</tr>
</tbody>
</table>

Maximum available unit load, $(W/p_d A_{max}) \times 10^3$

### TABLE II. - RADIAL LOCATION OF CENTER OF PRESSURE FOR VARIOUS GEOMETRIES AND COMPRESSIBILITY NUMBERS

[Pivot location, $\theta_p/\beta$, 1; values are for optimum load capacity.]

<table>
<thead>
<tr>
<th>Compressibility number, $\Lambda$</th>
<th>Ratio of pad inner radius to outer radius, $r_1/r_o$</th>
<th>0.3</th>
<th>0.5</th>
<th>0.7</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Angular extent of pad, $\beta$, deg</td>
<td>30</td>
<td>45</td>
<td>60</td>
</tr>
<tr>
<td>1 to 25</td>
<td>0.723</td>
<td>0.705</td>
<td>0.695</td>
<td>0.773</td>
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<tr>
<td>50</td>
<td>.722</td>
<td>.702</td>
<td>.692</td>
<td>.773</td>
</tr>
<tr>
<td>100</td>
<td>.716</td>
<td>.702</td>
<td>.690</td>
<td>.771</td>
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</table>
TABLE III. - COMPARISON BETWEEN UNIT LOAD FOR FLAT SECTOR-SHAPED PADS AND THEIR RECTANGULAR APPROXIMATION

<table>
<thead>
<tr>
<th>Clearance ratio, ( h_1/h_2 )</th>
<th>Pad 1(^a)</th>
<th>Pad 2(^b)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( h_1/h_2 )</td>
<td>Pivot location, ( \theta_p/\beta )</td>
<td>Rectangular approximation (ref. 9)</td>
</tr>
<tr>
<td>1.5</td>
<td>1.122</td>
<td>1.152</td>
</tr>
<tr>
<td>1.8</td>
<td>1.355</td>
<td>1.409</td>
</tr>
<tr>
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<td>1.625</td>
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<tr>
<td>5.0</td>
<td>1.246</td>
<td>1.429</td>
</tr>
</tbody>
</table>

Unit load, \((W/p_A)\times10^2\)

\(^a\)Ratio of pad inner radius to outer radius, \( r_i/r_o \), 0.9; angular extent of pad, \( \beta \), 24\(^\circ\); compressibility number, \( \Lambda \), 26.5.

\(^b\)\( r_i/r_o = 0.3; \theta = 62^\circ; \Lambda = 22. \)

Figure 1. - Sector-pad thrust bearing.
Figure 2. - Geometry of sector pad.

Figure 3. - Different regions of pivot-line location.
Figure 4. - Unit load at various pivot locations.
Ratio of pad inner radius to outer radius,
\( r_1/r_o \), 0.5, angular extent of pad, \( \beta \), 45\(^\circ\), compressibility number, \( A \), 10.
Figure 5. - Maximum available unit load at various pad angles for low and high compressibility numbers. Ratio of pad inner radius to outer radius, $r_1/r_o$, 0.5.
Figure 6. Unit load for optimally tilted pad for various radius ratios and angular extents of pad, at various compressibility numbers.

(a) Radius ratio, $r_1/r_0$, 0.3; angular extent of pad, $\beta$, 30°.

(b) Radius ratio, $r_1/r_0$, 0.3; angular extent of pad, $\beta$, 45°.

(c) Radius ratio, $r_1/r_0$, 0.3; angular extent of pad, $\beta$, 60°.
Compressibility number, \( \Lambda \)

(d) Radius ratio, \( r_1/r_2 \), 0.5; angular extent of pad, \( \beta \), 30°.

(e) Radius ratio, \( r_1/r_2 \), 0.5; angular extent of pad, \( \beta \), 45°.

Figure 6. - Continued.
Figure 6. - Concluded.
Figure 7. - Angular coordinate of center of pressure for radius ratio of 0.3 and different angular extents of pad at various compressibility numbers.
Figure 8. - Angular coordinate of center of pressure for radius ratio of 0.5 and different angular extents of pad at various compressibility numbers.

Figure 9. - Angular coordinate of center of pressure for radius ratio of 0.7 and different angular extents of pad at various compressibility numbers.
Figure 10. - Power loss coefficient for various radius ratios and angular extents of pad, at various compressibility numbers.
Compressibility number, $\Lambda$

(d) Radius ratio, $r_1/r_o$, 0.5; angular extent of pad, $\beta$, 30°.
(e) Radius ratio, $r_1/r_o$, 0.5; angular extent of pad, $\beta$, 49°.

Figure 10. - Continued.
Compressibility number, $\Lambda$

- 100
- 50
- 25
- 10 and 1

(f) Radius ratio, $r_1/r_0$, 0.7; angular extent of pad, $\beta$, 30°.

(g) Radius ratio, $r_1/r_0$, 0.7; angular extent of pad, $\beta$, 45°.

Figure 10. Concluded.
"The aeronautical and space activities of the United States shall be conducted so as to contribute . . . to the expansion of human knowledge of phenomena in the atmosphere and space. The Administration shall provide for the widest practicable and appropriate dissemination of information concerning its activities and the results thereof."

—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

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