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GAS LUBRICATED FOIL BEARING TECHNOLOGY DEVELOPMENT FOR
PROPULSION AND POWER SYSTEMS

MECHANICAL TECHNOLOGY, INCORPORATED

PREPARED FOR
AIR FORCE AERONAUTICS PROPULSION LABORATORY

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An investigation of air-lubricated foil bearing technology for application in advanced propulsion and power systems was performed. The work conducted under the program comprised: (a) establishment and documentation of the state-of-the-art of foil bearings, (b) development of analytical tools and design guides to permit calculation of the performance of bending-dominated foil bearings, (c) investigation of potential substrate and coating materials, as well as conduct of start-stop and high speed rub tests with two air-lubricated foil bearings at 1000°F, (d) preparation of a layout drawing of a dynamic simulator test vehicle based upon an advanced engine (1980 or later time frame) in the 1000 to 3000 pounds thrust class, as well as a description of the test facility requirements, (e) description of the technology segments of air-lubricated foil bearings that need to be advanced to meet the requirements of the advanced engine application, and definition of the parameters for a long range test effort. These work items are all described in this report. In particular, analyses of two types of bending-dominated foil bearings in both their journal and thrust bearing configurations were developed. These analyses coupled the foil elasticity and the hydrodynamic pressure relations to determine the equilibrium film shapes and, hence, the bearing performance. The analyses included compressibility effects, but did not incorporate side leakage. Substrate and surface coating materials for operation at up to 1000°F were identified. The unit load capacities necessary to meet the maneuver load requirements in MIL-E-5007C were established, based upon an advanced Teledyne-DAE engine. It was concluded that the bearing load capacity requirements, although they exceed currently demonstrated performance, are realistic technology targets that should be readily achievable before 1980, by following an integrated analytical and test effort.			

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TECHNOLOGY DEVELOPMENT FOR
PROPULSION AND POWER SYSTEMS

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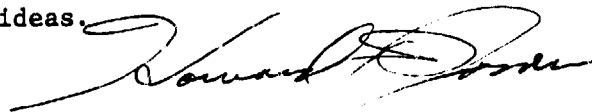
FOREWORD

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This report covers work conducted by the authors from 1 May 1972 - 31 August 1973. This report was submitted by the authors in September 1973.

The layout of the MAT-1 engine supported on air-lubricated foil bearings, discussed in Section VI of this report, was prepared by Teledyne-CAE. A first layout of the dynamic simulator test vehicle based on this advanced engine concept was also critically reviewed by Teledyne-CAE. In particular, the authors wish to acknowledge the assistance provided by Mr. S. Mathews, Mr. T. Moyer and Mr. P. Smith of Teledyne-CAE.

Publication of this report does not constitute Air Force approval of the report's findings or conclusions. It is published only for the exchange and stimulation of ideas.



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ABSTRACT

An investigation of air-lubricated foil bearing technology for application in advanced propulsion and power systems was performed. The work conducted under the program comprised: (a) establishment and documentation of the state-of-the-art of foil bearings, (b) development of analytical tools and design guides to permit calculation of the performance of bending-dominated foil bearings, (c) investigation of potential substrate and coating materials, as well as conduct of start-stop and high speed rub tests with two air-lubricated foil bearings at 1000°F, (d) preparation of a layout drawing of a dynamic simulator test vehicle based upon an advanced engine (1980 or later time frame) in the 1000 to 3000 pounds thrust class, as well as a description of the test facility requirements, (e) description of the technology segments of air-lubricated foil bearings that need to be advanced to meet the requirements of the advanced engine application, and definition of the parameters for a long range test effort. These work items are all described in this report. In particular, analyses of two types of bending-dominated foil bearings in both their journal and thrust bearing configurations were developed. These analyses coupled the foil elasticity and the hydrodynamic pressure relations to determine the equilibrium film shapes and, hence, the bearing performance. The analyses included compressibility effects, but did not incorporate side leakage. Substrate and surface coating materials for operation at up to 1000°F were identified. The unit load capacities necessary to meet the maneuver load requirements in MIL-E-5007C were established, based upon an advanced Teledyne-CAE engine. It was concluded that the bearing load capacity requirements although they exceed currently demonstrated performance, are realistic technology targets that should be readily achievable before 1980, by following an integrated analytical and test effort.

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LIST OF ABBREVIATIONS AND SYMBOLS
FOR
SECTION II

A_p	Total pad area, in ²
C	Clearance, inch
D_B	Flexural rigidity of bump foil, in-lb.
D_T	Flexural rigidity of top foil, in-lb.
E_B	Elastic modulus of bump foil, lb-in ⁻²
e	Shaft displacement, inch
$F_n(\chi)$	Dimensionless function, Equation (52)
H	Horizontal reaction, lb-in ⁻¹
$\tilde{H}$	Dimensionless horizontal reaction, Equation (11)
h	Film thickness, inch
h_B	Bump height, inch
h_o	Nominal film thickness, inch
h_{oI}	Incompressible nominal film thickness, inch
$\tilde{h}$	Dimensionless film thickness, Equation (49)
k	Integration constant appearing in Reynolds Equation
K_B	Unit bump stiffness, lb-in ⁻²
L	Transverse length perpendicular to direction of motion, inch
ℓ_i	Inlet length, inch
ℓ_o	Half length of bump, inch
ℓ_p	Pad length, inch
M	Bending moment, lb
p	Pressure, lb-in ⁻²
$\bar{p}$	Global pressure, lb-in ⁻²
p'	Pressure undulation, lb-in ⁻²

LIST OF ABBREVIATIONS AND SYMBOLS (Continued)
FOR
SECTION II

$\tilde{p}$	Dimensionless pressure, Equation (50)
$\hat{p}$	Dimensionless pressure, Equation (65)
p_a	Ambient pressure, lb-in ⁻²
p_o	Pressure, lb-in ⁻²
$\hat{p}_a$	Dimensionless ambient pressure, p_a/p_o
R	Bearing radius, inch
R_B	Bump radius, inch
R_m	Mean thrust bearing radius, inch
S	Bump pitch, inch
$\tilde{T}$	Dimensionless torque parameter, Equation (47)
T	Torque per unit transverse length, lb.
T^*	Total torque, in-lb.
t_B	Bump foil thickness, inch
u_o	Surface velocity, in-sec ⁻¹
v_B, w_B	Bump deflection, inch
$\tilde{v}_B, \tilde{w}_B$	Dimensionless deflection, Equation (11)
W	Load on bearing, lb-in ⁻¹
$\tilde{W}$	Dimensionless load parameter, Equation (46)
W^*	Total applied load, lb.
W_B	Bump load, lb-in ⁻¹
$\tilde{w}_R, \tilde{w}_o$	Dimensionless radial and tangential load components
$\hat{w}$	Dimensionless deflection, Equation (21)
w_T	Top foil deflection, inch
$\tilde{w}_{BO}$	Dimensionless deflection $\tilde{w}_B$ at $\theta = 0$

LIST OF ABBREVIATIONS AND SYMBOLS (Continued)
FOR
SECTION II

x	Coordinate along foil, inch
α	Dimensionless parameter, Equation (11)
α_B	Dimensionless bump foil compliance, Equation (44)
α_T	Dimensionless top foil compliance, Equation (43)
ϵ	Shaft displacement to clearance ratio, e/C
ϵ_θ	Tangential strain
ζ	Dimensionless coordinate, Equation (66)
θ	Angular coordinate
θ_a	Attitude angle
θ_o	Half angle subtended by bump, radians
θ_1	Angular position of start of foil
Λ	Compressibility number
Λ_1	Dimensionless parameter, Equation (68)
μ	Viscosity, lb-sec-in^{-2}
η	Coefficient of friction
ν_B	Poisson's ratio for bump foil
ξ	Dimensionless coordinate, θ/π
ξ_1	Dimensionless coordinate, θ_1/π
σ_θ	Tangential stress, lb-in^{-2}
ϕ_c	Compressibility correction factor, Figure 25
X	Dimensionless parameter, Equation (53)
Ω	Angular velocity, radians-sec^{-1}

LIST OF ABBREVIATIONS AND SYMBOLS
FOR
SECTIONS III & IV

a	Film thickness ratio, h_1/h_2
$a_{\max}$	Value of a , for maximum load capacity, inch
D_{eff}	Effective flexural rigidity of leaf thrust bearing, in-lb
D_L	Flexural rigidity of leaf, in-lb
e	Shaft displacement, inches
F_o, F_1	Forces of leaf, lb-in ⁻¹
$\tilde{F}_o, \tilde{F}_1$	Dimensionless forces scaled on W_i
$g(x)$	Static deflection profile, inches
h	Film thickness, inch
h_o	Minimum film thickness, inch
h_{oI}	Incompressible minimum film thickness, inch
h_1	Inlet film thickness, inch
h_2	Exit film thickness, inch
I_{jk}	Integral defined by Equation (135)
K	Bearing stiffness, lb-in ⁻²
K_i	Unit stiffness of i th leaf, lb-in ⁻²
l	Length of leaf chord, inch
l^*	Length, inch, Equation (132)
l_a, l_b	Coordinates of start and end of wrapped region, inch
$\tilde{l}_a, \tilde{l}_b$	Dimensionless lengths scaled on l
l_w	Distance between support and line between centers of shaft and leaf, inch
l_1	Overlap length, inch
$\tilde{l}_1, \tilde{l}_v$	Dimensionless lengths scaled on l
M, M_o	Bending moments, lb.

LIST OF ABBREVIATIONS AND SYMBOLS (Continued)
FOR
SECTIONS III & IV

$\tilde{M}, \tilde{M}_O$	Dimensionless moments, Equations (87), (88)
N	Number of leaves
P	Power per unit length, lb-sec ⁻¹
p	Pressure, lb-in ⁻²
p_a	Ambient pressure, lb-in ⁻²
p_{max}	Maximum pressure, lb-in ⁻²
R	Shaft radius, inch
R_{eff}	Effective radius, inch, Equation (100)
R_L	Preformed leaf radius, inch
R_M	Mean thrust bearing radius, inch
S	Leaf pitch, inch
T	Torque per unit length, lb.
$\bar{T}$	Dimensionless torque parameter, Equation (146)
u_O	Surface speed, in-sec ⁻¹
W	Applied load, lb-in ⁻¹
$\bar{W}$	Dimensionless load, Equation (142)
W_i	Load on ith leaf, lb-in ⁻¹
W_i^*	Load, lb-in ⁻¹ , Equation (133)
W_{ai}, W_{bi}	Reactions at beginning and end of wrapped region, lb-in ⁻¹
$\tilde{W}_{ai}, \tilde{W}_{bi}$	Dimensionless reactions scaled on W_i
w	Leaf deflection, inch
$\tilde{w}$	Dimensionless deflection, Equation (85)
w_i	Deflection of ith leaf at distance l_w from support
x	Coordinate along leaf, inch

LIST OF ABBREVIATIONS AND SYMBOLS (Continued)
FOR
SECTIONS III & IV

α_c	Fraction of pitch of deflected leaf bearing that is convergent, Figure 45b
β	Dimensionless parameter, Equation (102)
δ_0	Preload deflection, inch
Λ	Compressibility number, Equation (43)
μ	Viscosity, lb-sec-in ⁻²
ξ	Dimensionless coordinate, x/l
ϕ_c	Compressibility correction factor, Figure 25
ϕ_1	Angle defined in Figure 26 and Equation (77)
ϕ_2	Angle between successive leaf support

LIST OF ABBREVIATIONS AND SYMBOLS
FOR
APPENDIX I

C	Compressibility parameter (See Equation I-3)
D	Flexural rigidity of the foil per unit width, in-lbs
	$\left[D = \frac{Et^3}{12(1-\nu^2)} \right]$
E	Young's modulus of the foil, lbs-in ⁻²
f _N	Natural frequency of rotor bearing system, cycles per second
	$\left[f_N = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \right]$
h _o	Film thickness in central, uniform gap width section, inches
I	Fluid inertia parameter (See Equation I-4)
K	Dimensionless parameter in film thickness equation (See Equation I-1)
k	Bearing stiffness per unit width of foil, lbs-in ⁻²
L	Foil width, inches
ℓ	Foil length between supports, inches
m	Rotor mass per unit width of foil, lb-sec ² -inches ⁻²
p _a	Ambient pressure, psia
R	Bearing radius, inches
S	Foil Stiffness parameter (See Equation I-2)
T	Tension per unit width of foil, lbs-in ⁻¹
t	Foil thickness, inches
U	Journal surface velocity, inches-second ⁻¹
δℓ	Elongation of foil between supports beyond the stretch caused by the initial tension (e.g., due to thermal growth or relaxation of initial tension), inches
θ	Wrap angle, radians

LIST OF ABBREVIATIONS AND SYMBOLS (Continued)
FOR
APPENDIX I

λ	Molecular mean free path, inches
μ	Absolute viscosity, lb-sec-inch ⁻²
ν	Poisson's ratio
ρ_a	Density of ambient fluid, lb-sec ² -in ⁻⁴

SECTION I

INTRODUCTION

1. Background

High performance, complex cycle engines and power systems for 1980 and later time frames will demand sharp performance increases from their bearings and lubrication systems while subjecting them to very severe environments. The U.S. Air Force Aero Propulsion Laboratory has initiated a program directed at developing the technology of air-lubricated foil bearings for their application in advanced propulsion engines in the 1000 to 3000 pounds thrust class and equivalent power systems. The long-range goal of this and other efforts is to remove or lift the penalties imposed by the temperature, cooling-requirement and speed limitations of conventionally lubricated rolling element bearings. The program described herein is an initial step in the development of the foil bearing technology required for application of air-lubricated foil bearings in the engine environment.

This emphasis on air-lubricated foil bearings is prompted by the fact that they combine the performance capabilities of other, self-acting, air-lubricated bearings, together with certain additional advantages that are uniquely their own. Thus, in common with rigid-surface, air-lubricated bearings, the foil bearings have no high temperature limitations beyond those imposed by the structural and surface materials, and they are not life-limited when operating on a full hydrodynamic film. Compared with rigid-surface bearings, they have the potential for higher load capacity, stability margin, tolerance of rotor and housing distortions, and of particulate contaminants in the lubricant gas stream. Most importantly, they combine these benefits with mechanical simplicity, ease of assembly and installation, and low fabrication cost.

Air-lubricated foil bearings are currently being used to support airborne, high-speed turbomachines, notably air-cycle machines. Such bearings, lubricated with air or other gases are being evaluated for possible use in the hot (turbine-end) location or small aircraft engines, in turbochargers and in high-speed motors, as well as for support of turbine-alternator-compressor rotors of dynamic space power machinery.

2. Program Approach

The program comprised several interrelated work items as follows:

a. Analysis of Foil Bearing Performance

A survey of the published work on foil bearings was made to establish the state-of-the-art from the standpoints of theoretical development, test and practice. The state-of-the-art was then extended by initiating analyses of three types of bending-dominated foil bearings, in both their journal and thrust bearing configurations. The analyses of two of these bearing types were pursued through the preparation of computer programs and design aids for performance calculations. These two bearing types are the

ridged foil bearing where a compliant tape is supported on a corrugated foil, and the cantilevered leaf foil bearing which is made of compliant strips (which overlap in the journal bearing arrangement), fixed at one end. The third concept, which comprised a single foil strip pleated to form a series of convergent wedges was found to have lower load capacity than the first two. It was not pursued beyond a preliminary calculation of unit load versus film thickness.

b. Foil Bearing Materials

Surveys were made of substrate and surface coating materials suitable for use in foil bearings operating in the engine environment at temperatures up to 1000°F. Start-stop and high-speed rub tests were conducted at 1000°F on two, full 360° foil bearing arrangements. The two bearings were of the same design and used the same substrate material, but had two different surface coatings.

c. Dynamic Simulator Test Vehicle

A requirement of the present program was to prepare a layout of a dynamic simulator test vehicle in which a long-term test program on foil bearings could be conducted in the future. The advanced engine that served as the model for the test vehicle was selected in consultation with Teledyne CAE. This engine is an advanced gas generator that Teledyne CAE is developing for the Air Force, for powering drone and missile craft. The engine under development uses conventionally lubricated, rolling-element bearings. However, for the purpose of the present program, Teledyne CAE prepared a layout of the engine adapted for support on air-lubricated foil bearings. This concept, designated the MAT-1 engine, was also used to establish the bearing loads corresponding to the maneuver conditions specified in MIL E-5007C. The dynamic simulator layout was then prepared with its rotor mass and stiffness distributions modelled after those of the engine. A test facility schematic and description of necessary hot gas flow, controls and instrumentation were also made.

d. Technology Segments and Long-Range Test Effort

Based on the results of the program, the basic technology segments of air-lubricated foil bearings that need to be advanced to meet the operational and environmental requirements of the advanced engine were identified and test parameters for the long-range test effort were defined.

3. Report Organization

In this report, the state-of-the-art survey is provided in Appendix I. The analyses, equations and graphs that deal with the performance on the ridged and the cantilevered-leaf type of foil bearings are given in Sections II and III respectively of this report. The computer programs written for these two bearing types in both their journal and thrust bearing configurations are included in Appendix II of this report. The programs were written in Fortran IV for

use with the General Electric Mark II system. Section IV of the report briefly describes the pleated type of foil bearing.

The materials investigation, including the survey of substrate materials and surface coatings, as well as the start-stop and high-speed rub tests is described in Section V.

Section VI of the report describes the MAT-1 engine, as well as the layout and analysis of the dynamic simulator test rig and its test facility.

The identification and discussion of the technology segments as well as recommended values for test parameters in the long-range test program are provided in Section VII. Conclusions and Recommendations resulting from the program are given in Section VIII.

SECTION II

ANALYSIS OF RIDGED BEARINGS

1. Elasticity Considerations

Both the ridged journal and thrust bearings consist of a smooth upper tape riding over a ridged lower tape. The stiffness of the bearing will depend both upon the stiffness of the gas film and the stiffness of the lower ridged or bump tape. The deflection of the bump tape is particularly important when the gas film is stiff compared with the foil which is the situation which makes optimum use of the compliance of the bending dominated foil bearing.

a. Bump Deflection and Stiffness

A schematic of the coordinate system and loading used to analyze an individual bump is shown in Figure 1. The bump is assumed to be loaded at the center with a load W_B and a local deflection under the load is w_B . The horizontal reactions, which will be more difficult to evaluate, are denoted by H . This horizontal load will be dependent on the friction characteristics and could also have a component resulting from compression in the tape between bumps. If there is no friction and the bumps do not interact, then H will equal 0. If there is friction between the bump tape and the housing, of a magnitude less than that necessary to hold the ends of the bump pinned in place and there are no interactions between bumps, then the quantity $2H/W_B$ will be equal to the coefficient of sliding friction η . In this case, the load W_B necessary to produce a given deflection will be somewhat higher than that which would produce the same deflection in the frictionless case. The difference between the two values of W_B provides an index of the frictional damping within the system. This will be the dominant mode of damping for applications where the stiffness of the bump is much lower than that of the film.

If the friction is extremely high, then the ends of the bump will be held in place and there may be no sliding or friction damping occurring. The value of $2H/W_B$ required to hold the bump in place can be obtained by solving the problem for the deflection of a bump with pinned ends. The remainder of this section is devoted toward developing the equations for the load deflection relationships for various coefficients of friction, η , and various bump angles, θ_0 .

To begin with, one may equate the bending moment, M , per unit transverse length, resulting from forces occurring to the left of any position θ along the bump to the change in curvature of the bump multiplied by the flexural rigidity of the bump in a manner directly analogous to a beam. Hence, we have the relationship

$$\frac{D_B}{R_B} \frac{1}{2} \left[\left(\frac{d^2 w_B}{d\theta^2} \right) + w_B \right] = M \quad (1)$$

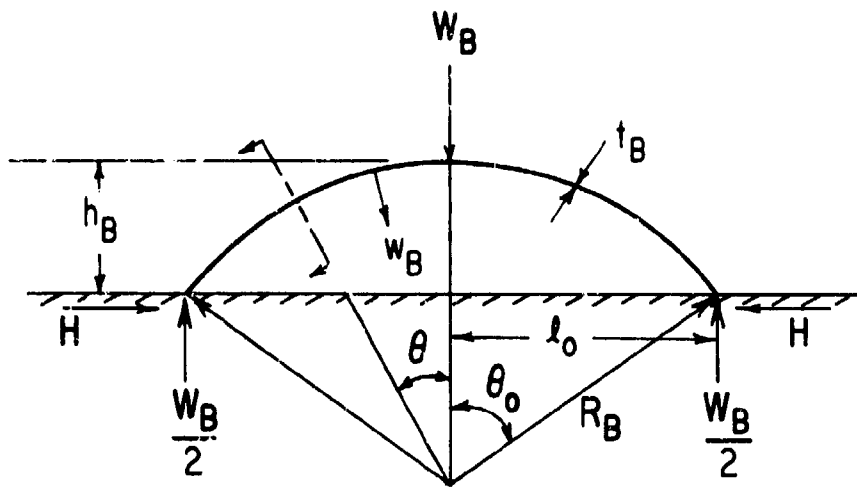


Fig. 1 Schematic for Analysis of Bump Deflection

In the above equation, the flexural rigidity of the bump tape D_B is defined by

$$D_B = \frac{E_B t_B^3}{12(1-\nu_B^2)} \quad (2)$$

where E_B , t_B , ν_B represent the elastic modulus, thickness, and Poisson's ratio of the bump tape respectively. The radial deflection w_B is taken to be positive and the radially inward direction as shown in Figure 1 and the bending moment M about the section shown in Figure 1 is taken to be positive when forces to the left of that section produce a counter-clockwise moment. The quantity w_B appearing next to the second derivative term on the left of Equation (1) results from the fact that the beam is initially circular rather than straight and the curvature term is the difference in curvature resulting from elastic bending. One may evaluate moments from the forces acting to the left of the section shown in Figure 1 to obtain the relationship

$$\frac{D_B}{R_B^2} \left(\frac{d^2 w_B}{d\theta^2} + w_B \right) = \frac{R_B W_B}{2} (\sin \theta - \sin \theta_0) + R_B H (\cos \theta - \cos \theta_0) \quad (3)$$

Equation (3) corresponds to the deflection of an inextensible beam without centerline elongation. The effect of longitudinal compression on the value of H required to hold the bump in a pinned position can be very significant when θ is small and should be considered at least in an approximate manner.

To obtain this correction, one may first calculate the average tangential stress acting on a plane cut by the section shown in Figure 1 balancing the forces at the support to the left of that section. Hence,

$$\sigma_\theta = - \left(\frac{W_B}{2t_B} \sin \theta + \frac{H}{t_B} \cos \theta \right) \quad (4)$$

where σ_θ denotes the tangential tensile stress and the minus sign appears since positive values of W_B and H will produce a compressive stress. The tangential strain, ϵ_θ , is given by

$$\epsilon_\theta = \frac{1}{R_B} \frac{d v_B}{d\theta} - \frac{w_B}{R_B} \quad (5)$$

where v_B is the tangential elongation of the bump.

For a thin circular arc in plane strain, ϵ_θ is related to σ_θ approximately by

$$\epsilon_\theta = -\frac{1 - \nu_B^2}{E_B} \sigma_\theta = \frac{t_B^3}{12 D_B} \sigma_\theta \quad (6)$$

One may now combine Equations (4) through (6) to obtain the relationship

$$\frac{dv_B}{d\theta} - w_B = -\frac{t_B^2}{12 D_B} \left(\frac{W_B}{2} \sin \theta + H \cos \theta \right) \quad (7)$$

which constitutes the correction to Equation (3) for extensibility of the bump foil.

If H is known, then three boundary conditions are required to solve the system of differential equations given by Equations (3) and (7); two of these are needed for w_B and one for v_B . When the friction is high enough to hold the ends pinned, a fourth condition is required to determine H . Two boundary conditions applicable at the center of the bump resulting from symmetry and valid for either case are

$$v_B = \frac{dw_B}{d\theta} = 0 \quad \text{at } \theta = 0 \quad (8)$$

The remaining two conditions for the case where the ends are pinned are simply

$$v_B = w_B = 0 \quad \text{at } \theta = \theta_0 \quad \text{for pinned ends.} \quad (9)$$

When $2H/W_B$, as calculated from the solution to Equations (3), (7), (8) and (9), is less than the coefficient of friction, the ends will be held in place. When the friction is low enough so that the ends are not held pinned, then $2H/W_B$ will equal the coefficient of friction and there will be a horizontal deflection at the boundaries but no vertical deflection. These conditions can be expressed mathematically as

$$v_B \sin \theta_0 + w_B \cos \theta_0 = 0 \quad \text{at } \theta = \theta_0 \quad (\text{no vertical deflection}) \quad (10a)$$

$$2H/W_B = \eta \quad (\text{sliding friction}) \quad (10b)$$

Equations (3) and (7) may be expressed in terms of the dimensionless variables and parameters

$$\tilde{w}_B = \frac{2 D_B}{W_B R_B^3} w_B, \quad \tilde{v}_B = \frac{2 D_B}{W_B R_B^3} v_B, \quad \alpha = \frac{1}{12} \left(\frac{t_B}{R_B} \right)^2, \quad \tilde{H} = 2 \frac{H}{W_B}, \quad (11)$$

as follows

$$\frac{d^2 \tilde{w}_B}{d\theta^2} + \tilde{w}_B = \sin \theta - \sin \theta_0 + \tilde{H} (\cos \theta - \cos \theta_0) \quad (12)$$

$$\frac{d \tilde{v}_B}{d\theta} - \tilde{w}_B = -\alpha (\sin \theta + \tilde{H} \cos \theta) \quad (13)$$

The boundary conditions on $\tilde{v}_B$ and $\tilde{w}_B$ are the same as those on v_B and w_B given in Equations (8) through (10). The solution to Equation (12) satisfying $d\tilde{w}_B/d\theta = 0$ at $\theta = 0$ is given by

$$\tilde{w}_B = A \cos \theta + \frac{1}{2} \sin \theta - \sin \theta_0 - \tilde{H} \cos \theta_0 - \frac{1}{2} \theta \cos \theta + \frac{\tilde{H}}{2} \theta \sin \theta \quad (14)$$

where A is a constant to be determined.

One may now substitute Equation (14) for $\tilde{w}_B$ into Equation (13) which may in turn be integrated subject to the boundary condition given by Equation (8) to determine $\tilde{v}_B$ which may then be evaluated at θ_0 to give the relationship

$$\begin{aligned} \tilde{v}_B(\theta_0) = & A \sin \theta_0 + (1-\alpha) (1-\cos \theta_0) - \frac{3}{2} \theta_0 \sin \theta_0 \\ & + \frac{\tilde{H}}{2} \left[-3 \theta_0 \cos \theta_0 + (1-2\alpha) \sin \theta_0 \right] \end{aligned} \quad (15)$$

Equation (9) may now be used to evaluate A and $\tilde{H}$ when the ends are pinned and Equation (10) may be used to evaluate A when the ends are not pinned.

The quantities $\tilde{H}$ and A for the case of pinned ends are as follows:

$$\left. \begin{aligned} \tilde{H} &= \frac{s^2 - 2c (\theta_o s + c - 1) - 2\alpha c(1 - c)}{-3sc + \theta_o (2c^2 + 1) + 2\alpha sc} \\ A &= \frac{1}{2} \frac{s}{c} + \frac{1}{2} \theta_o - \frac{1}{2} \tilde{H} \left[\theta_o \frac{s}{c} - 2 \right] \end{aligned} \right\} \text{Pinned Ends} \quad (16)$$

$$(17)$$

where the abbreviations $s = \sin \theta_o$ and $c = \cos \theta_o$ are used. If the coefficient of friction is less than the value of $\tilde{H}$ calculated from Equation (16), then the ends will not be held in place and $\tilde{H}$ and A are given by the following equations:

$$\tilde{H} = \eta \quad (18)$$

$$\begin{aligned} A &= -s + \frac{3}{2} sc + \frac{1}{2} \theta_o + \theta_o s^2 + \alpha s(1 - c) \\ &+ \eta \left(\theta_o sc - \frac{1}{2} s^2 + c^2 + \alpha s^2 \right) \end{aligned} \quad (19)$$

For either set of conditions, (sliding friction or pinned ends), the deflection at the center of the bump $\tilde{w}_{Bo}$ is given as follows:

$$\tilde{w}_{Bo} = A - s - \tilde{H} c \quad (20)$$

where $\tilde{H}$ and A are defined by Equations (16) and (17) for pinned ends and Equations (18) and (19) for sliding friction.

Center deflections are plotted in terms of the dimensionless quantity $\hat{w}$ defined below

$$\hat{w} = \frac{2D_B w_{Bo}}{w_B \ell_c^3} = \frac{\tilde{w}_{Bo}}{\sin^3 \theta_o} \quad (21)$$

as functions of θ_o for various values for the coefficient of friction η for the case $\alpha = 0$ in Figure 2 and $\alpha = 10^{-2}$ in Figure 3. This latter case represents an upper limit for values of α one could expect to obtain in practice.

The unit bump stiffness, K_B , (per unit transverse length L) is defined in terms of $\hat{w}$ below

$$K_B = \frac{w_B}{w_{Bo}} = \frac{2 D_B}{\hat{w} \ell_o^3} \quad (22)$$

It can be seen from Figures 2 and 3 that when $\eta = 0$, $\hat{w}$ is very nearly independent of both θ_o and α . For $0 < \theta_o < \pi/2$ and $\alpha < .01$, $\hat{w}$ assumes a nearly

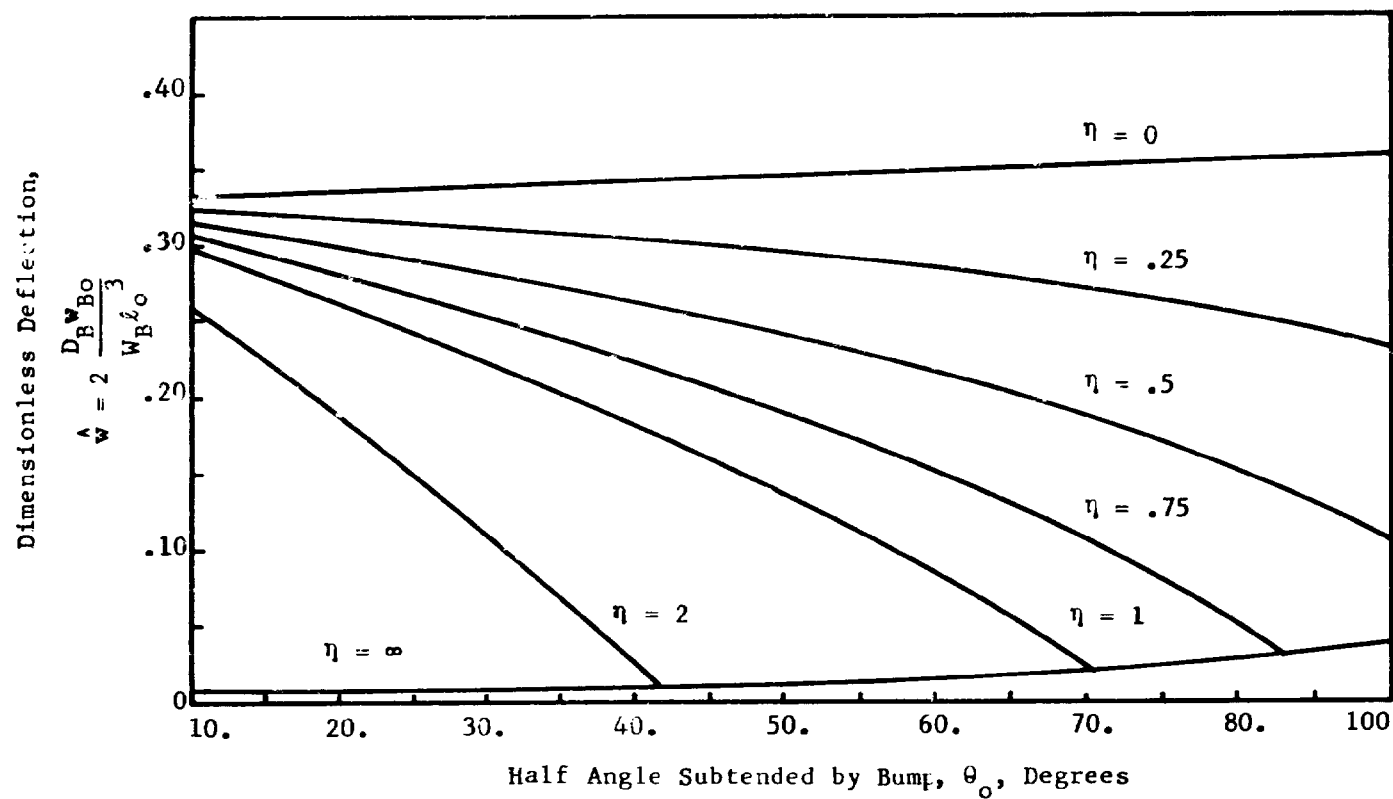


Fig. 2 Dimensionless Deflection for an Inextensible Bump ($\alpha = 0$)

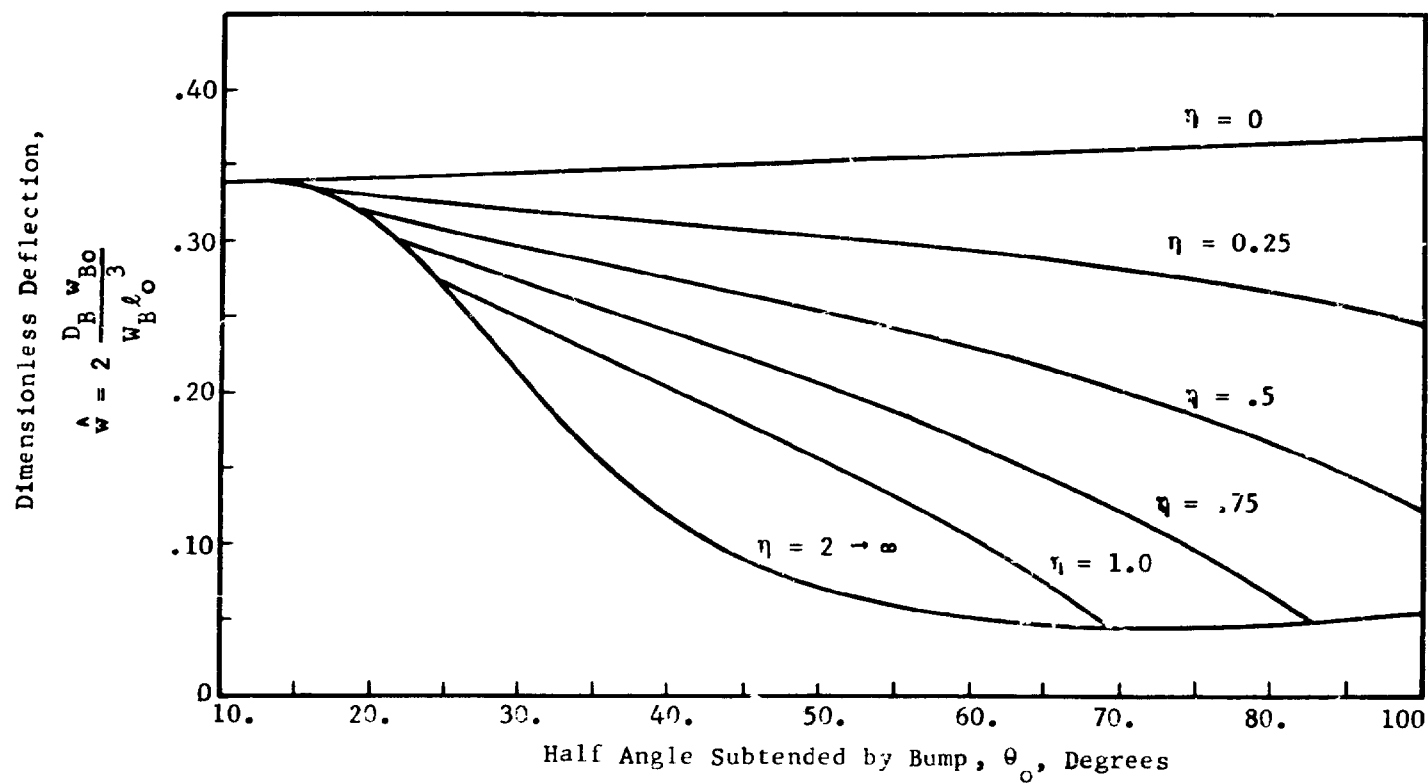


Fig. 3 Dimensionless Bump Deflection, $\alpha = .01$

constant value of $\hat{w} = \frac{1}{3}$. For this case, we have the approximate relationship

$$K_3 \approx \frac{6 D_B}{\ell_0^3} \quad \text{for } \eta = 0 \quad (23)$$

which is the result that one would obtain by treating the bump as a simply supported beam of length $2\ell_0$ loaded at the center with a force W_B .

It can be seen that for large values of θ_0 , friction can have a marked effect on the bearing stiffness.

b. Elastic Deflections of the Top Foil

In a ridged bearing, pressure distribution will be shown later to consist of a global pressure*, together with a small undulating pressure. The deflection profile between bumps, for the ridged bearing analysis, is the deflection resulting from the global pressure. The situation of interest for the top foil deflection analysis is shown schematically in Figure 4. Here, the top foil is exposed to a uniform pressure p_0 which could vary around the bearing but stays constant between consecutive bumps. A variety of different deflection characteristics could arise under this loading. For example, the top foil could wrap around each bump and drape on the land between bumps or various combinations of wrapping and draping could occur. It will be shown later that from a practical design point of view it will be desirable to have the top foil sufficiently rigid so that it does not wrap around the bumps or drape in between the bumps but hangs in the manner illustrated in Figure 4. Under these circumstances, the top foil deflection behaves exactly like a beam that is fixed at both ends and subjected to a uniform pressure p_0 . Solutions to this problem can be found in a variety of texts (e.g., Reference 1) and will not be derived here. The top foil deflection, w_T , is given below

$$w_T = \frac{S^4 p_0}{24 D_T} \left(\frac{x}{S}\right)^2 \left(1 - \frac{x}{S}\right)^2 \quad (24)$$

where D_T denotes the flexural rigidity of the top foil. It can be seen from the above equation the maximum deflection occurs when $x/S = 1/2$ and is given by

$$w_T = \frac{S^4 p_0}{384 D_T} \quad (25)$$

The above equation provides the deflection of the top foil relative to the tops of the bumps on the lower foil. The analysis presented in the preceding section provides the deflections of the tops of the lower foil

* The global or large scale pressure is the part of the pressure profile that varies around the bearing but does not oscillate from bump to bump.

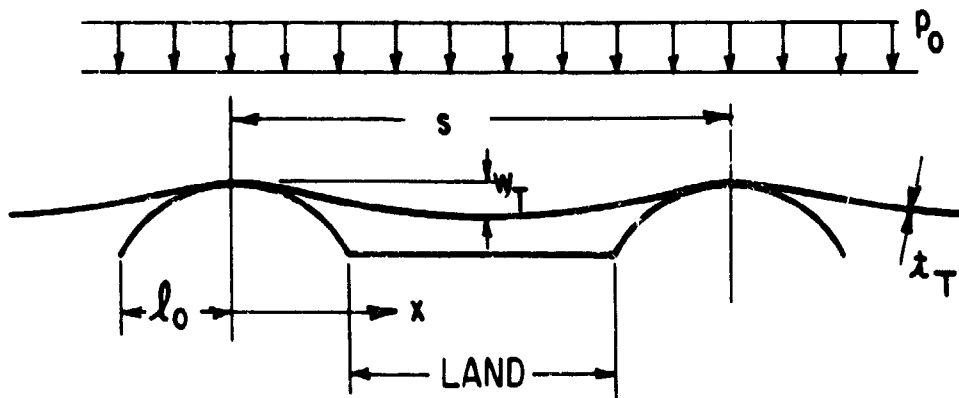


Fig. 4 Schematic for Analysis of Top Foil Deflection

with respect to the housing. The two analyses together provide all the elasticity information that will be required in treating the ridged journal and thrust bearings. The remaining portion of the ridged bearing analysis involves hydrodynamic considerations and the coupling of hydrodynamic theory with the elasticity equations presented above.

2. Performance Analysis of Ridged Journal Bearings

The ridged journal bearing incorporates two modes of compliance upon the conventional rigid journal bearing. These are manifested in the form of grooves which develop when the top foil deflects between bumps and in the change in bearing geometry developing from the deflection of the bumps themselves. A schematic of a ridged journal bearing is shown in Figure 5. The principle qualitative differences in operation of the ridged bearing from the conventional one can be seen from Figure 5 in that the shaft displacement e can be many times larger than the clearance C as the result of bump compliance. The bump compliance will, of course, affect the stiffness of the bearing but it will also affect the load capacity, the attitude angle θ_a , and the overall stability of the bearing. These effects will, in general, be advantageous.

The effect of the top foil deflection is less obvious. It will, in fact, be shown that the grooves that result from the top foil deflection will, in general, be detrimental to bearing operation. These grooves are in the form of axial grooves and hence provide no pumping to generate pressures but do provide a source for escape of fluid and can result in somewhat reduced load capacity if the grooving is too deep. In order to evaluate these effects simultaneously, an analysis has been developed which considers both the deflection of the bumps and the deflection of the top foil. This analysis has been developed for a one dimensional infinitely long journal bearing. It makes use of approximations based on the assumption that the bump spacing S is small compared to the circumference of the journal bearing. Therefore, the chord length may be considered to be a close approximation to the arc length.

In that the overall geometry is very similar to that of a conventional journal bearing, the notation in the equations used will be similar to that used in the analysis of conventional journal bearings.

The governing equation for the hydrodynamic behavior of the ridged journal bearing is the Reynolds Equation which may be written as

$$p \frac{dp}{d\theta} = 6\mu u_o R \frac{(ph - k)}{h^3}, \quad (26)$$

where p is pressure, μ is viscosity and u_o is the linear speed of the bearing. The constant k and the necessary constant of integration for the first order differential equation given above are determined from the boundary conditions given below

$$p = p_a \text{ at } \theta = \theta_1, \quad \theta_1 = 2\pi \quad (27)$$

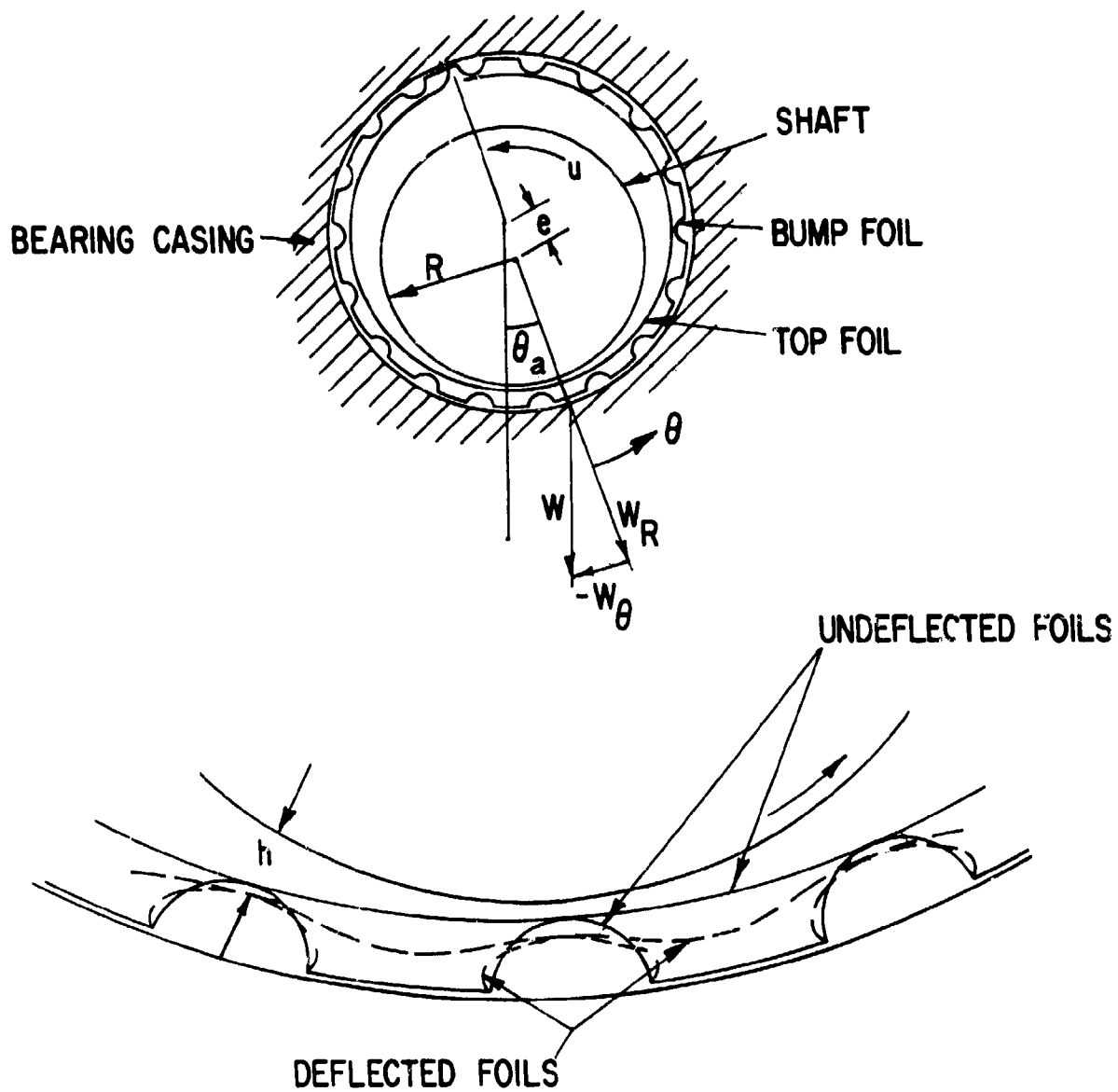


Fig. 5 Schematic of Ridged Journal Bearing

which assumes that the tape is in a form of a 360° bearing that starts at a value of $\theta = \theta_1$ and that the pressure is ambient at $\theta = \theta_1$. If the bearing were completely closed and there was no such starting groove, then the boundary condition given by Equation (27) would be replaced by

$$\left. \begin{aligned} p(-\pi) &= p(\pi) \\ \int_{-\pi}^{\pi} p^2 h^3 d\theta &= p_a^2 \int_{-\pi}^{\pi} h^3 d\theta \end{aligned} \right\} \begin{pmatrix} \text{For Closed} \\ 360^\circ \text{ Bearing} \end{pmatrix} \quad (28)$$

In the above equations, p_a denotes the ambient pressure. Equation (27) will apply most frequently to conventional bearing designs and it assumes that the bearing is coupled to the ambient pressure at the start of the foil, whereas Equation (28) assumes the coupling to ambient to be along the sides of the bearing.

Equation (26) is somewhat difficult to solve in its present form as a result of the relationship between h and the pressure p . Unlike the conventional bearing h is a highly undulating function of p as a result of the many grooves which form. The situation, however, is not unlike that of a spiral groove bearing and, in fact, some of the precepts developed for the spiral groove bearing are used to treat the effect of the grooves. In particular, the narrow groove theory developed by Whipple for a spiral groove bearing is used here, simplified further by the fact that the grooves of the ridged bearing are purely axial. On the other hand, the analysis is complicated by the fact that the groove geometry itself is not flat and is, in fact, a function of the local pressure. We will proceed in a manner directly analogous to that used in developing the narrow groove bearing theory by considering the pressure p to consist of two components as follows:

$$p = \bar{p} + p' \quad (29)$$

where $\bar{p}$ denotes the global pressure and p' represents the small undulation in the pressure at each groove. Hence p' represents a wave of small amplitude but high frequency.

Whereas p' will be of the order of S/R with respect to $\bar{p}$, derivatives of p' will be the same order of magnitude as the derivative of $\bar{p}$ with respect to θ .

A function f may be broken into its global and undulating components as follows:

$$\bar{f} = \frac{1}{S} \int_{R\theta}^{R\theta + S} f(x) dx, \quad f' = f - \bar{f} \quad (30)$$

where x is a position variable varying from $R\theta$ to $R\theta + S$. $\bar{p}$ is then integral of the pressure over one bump spacing divided by the bump spacing and p' is the

difference between p and $\bar{p}$.

If we substitute Equation (29) for p , appearing in Equation (26) and linearize by neglecting terms of order S/R , we obtain the following relationship:

$$\bar{p} \frac{d\bar{p}}{d\theta} + \bar{p} \frac{dp'}{d\theta} = 6\mu u_o R \frac{(\bar{p} h - k)}{h^3} . \quad (31)$$

One may now perform the operation indicated by Equation (30) on Equation (31) making use of the fact that p' is periodic to obtain the relationship for the global pressure

$$\bar{p} \frac{d\bar{p}}{d\theta} = 6\mu u_o R \left[\bar{p} \left(\frac{1}{h^2} \right) - k \left(\frac{1}{h^3} \right) \right] . \quad (32)$$

The film thickness profile h , consists of the undeformed profile plus the bump deflection plus the top foil deflection and is given by

$$h = C - e \cos \theta + w_{Bo} + w_T . \quad (33)$$

The deflection of the tops of the bumps w_{Bo} is obtained from Equation (22) with w_B replaced by $(\bar{p} - p_a) S$ and is given below

$$w_{Bo} = \frac{(\bar{p} - p_a) S}{K_B} . \quad (34)$$

Here the pressure under the bumps is assumed to be ambient and the load on the bump is taken to be the net applied pressure over one span multiplied by the length of the span between bumps. w_{Bo} will be taken to be 0 when the global pressure $\bar{p}$ is less than the ambient pressure p_a .

In a similar manner, one may obtain the expression for w_T by replacing the uniform pressure p_o appearing in Equation (24) by the quantity $\bar{p} - p_a$ to obtain

$$w_T = \frac{S^4 (\bar{p} - p_a)}{24 D_T} \left(\frac{x}{S} \right)^2 \left(1 - \frac{x}{S} \right)^2 . \quad (35)$$

Here again, w_T will be taken to be 0 if $\bar{p}$ is less than p_a .

The various average film thickness terms appearing in Equation (32) may be obtained by combining Equations (33) through (35) and averaging in accordance with Equation (30) to obtain the result given below:

$$\left(\frac{\bar{1}}{h^n}\right) = \frac{1}{S} \int_{R\theta}^{R\theta + S} \frac{dx}{\left[C - e \cos \theta + \frac{(\bar{p}-p_a) S}{K_B} + \frac{S^4 (\bar{p}-p_a)}{24 D_T} \left(\frac{x}{S}\right)^2 \left(1 - \frac{x}{S}\right)^2 \right]^n} \quad (36)$$

The only term which varies significantly with local position is the last term on the right in the denominator of Equation (36). All the other terms appearing in the integrand may be treated as constants when evaluating the integral. Hence, the use of Equation (36) together with Equation (32) provides the full global Reynolds equation for the ridged journal bearing. The boundary conditions for a 360° bearing that starts at a value of $\theta = \theta_1$ is the same as that given by Equation (27) with p replaced by $\bar{p}$. For the case of a closed circular 360° bearing, the boundary conditions are given below

$$\left. \begin{aligned} p(-\pi) &= p(\pi) \\ \int_{-\pi}^{\pi} \bar{p}^2 \bar{h}^3 d\theta &= p_a^2 \int_{-\pi}^{\pi} \bar{h}^3 d\theta \end{aligned} \right\} \quad \left(\begin{array}{l} \text{For Closed} \\ 360^\circ \text{ Bearing} \end{array} \right) \quad (37)$$

Equations (32) and (36) together with the boundary conditions given by Equations (27) or (37) are sufficient to mathematically determine the pressure and film thickness profiles $\bar{p}$ and $\bar{h}$. One may now compute the loads W_R and W_θ as shown in Figure 5 from the relationships

$$W_R = R \int_{\theta_1 - 2\pi}^{\theta_1} \bar{p} \cos \theta d\theta \quad ; \quad W_\theta = -R \int_{\theta_1 - 2\pi}^{\theta_1} \bar{p} \sin \theta d\theta \quad (38)$$

and the resulting load W and attitude angle θ_a by the relationship

$$W = \sqrt{W_R^2 + W_\theta^2} \quad ; \quad \theta_a = \tan^{-1} (W_\theta / W_R) \quad (39)$$

Finally, the bearing torque, resulting from the Couette portion of the velocity profile, will be given by the relationship

$$T = \mu u_o R^2 \int_{\theta_1 - 2\pi}^{\theta_1} \left(\frac{\bar{1}}{h}\right) d\theta \quad (40)$$

which can be used to estimate power requirements.

A complete dimensional formulation of the ridged journal bearing problem has now been presented. The remainder of this section deals with the nondimensional equations used in obtaining and presenting solutions to the problem.

We may express the ridged journal bearing equations in terms of the dimensionless parameters given below:

$$\epsilon = e/C \quad \text{displacement to initial clearance ratio} \quad (41)$$

$$\Lambda = \frac{6\mu u_o R}{C^2 p_a} \quad \text{compressibility number} \quad (42)$$

$$\alpha_T = \frac{6\mu u_o RS^4}{384D_T C^3} \quad \text{dimensionless top foil compliance} \quad (43)$$

$$\alpha_B = \frac{6\mu u_o RS}{K_B C^3} \quad \text{dimensionless bump foil compliance} \quad (44)$$

$$\xi_1 = \frac{\theta_1}{\pi} \quad \text{dimensionless position at start of foil} \quad (45)$$

$$\tilde{W} = \frac{W}{6\mu u_o} \left(\frac{C}{R}\right)^2 \quad \text{dimensionless load parameter} \quad (46)$$

$$\tilde{T} = \frac{TC}{\mu u_o R^2} \quad \text{dimensionless torque parameter} \quad (47)$$

At present ϵ , Λ , α_B , α_T and ξ_1 are treated as input parameters and $\tilde{W}$, θ_a and $\tilde{T}$ are output parameters to be determined.

The dimensionless variables to be introduced for position, film thickness and pressure are:

$$\xi = \theta/\pi \quad (48)$$

$$\tilde{h} = \frac{h}{C} \quad (\text{over top of bump}) \quad (49)$$

$$\tilde{p} = \frac{C^2 \bar{p}}{6\mu u_o R} \quad (50)$$

The global Reynolds Equation (Equation (32)) may now be written as

$$\tilde{p} \frac{d\tilde{p}}{d\xi} = \frac{\pi \tilde{p} F_2(\xi)}{\tilde{h}^2} - \frac{\pi k' F_3(\xi)}{\tilde{h}^3} \quad (51)$$

where k' is a constant to be determined from the boundary conditions and the functions $F_n(\xi)$ are given by

$$F_n(\chi) = \begin{cases} \int_0^1 \frac{d\zeta'}{\left(1 + \chi \zeta'^2 (2-\zeta)^2\right)^n} & \text{if } \chi \geq 0 \\ 1 & \text{if } \chi < 0 \end{cases} \quad (52)$$

$$\text{where } \chi = \alpha_T \left(\tilde{p} - \frac{1}{\Lambda}\right) / \tilde{h} \quad , \quad \zeta' \text{ is a dummy variable of integration} \quad (53)$$

and

$$\tilde{h} = \begin{cases} 1 - \epsilon \cos \pi \xi + \alpha_B \left(\tilde{p} - \frac{1}{\Lambda}\right), & \text{if } \tilde{p} \geq \frac{1}{\Lambda} \\ 1 - \epsilon \cos \pi \xi, & \text{if } \tilde{p} < \frac{1}{\Lambda} \end{cases} \quad (54)$$

The boundary conditions given by Equation (27) become

$$\tilde{p}(\xi_1 - 2) = \tilde{p}(\xi_1) = \frac{1}{\Lambda} \quad (55)$$

and the boundary condition given by Equations (37) become:

$$\tilde{p}(-1) = \tilde{p}(1)$$

$$\Lambda^2 \int_{-1}^1 \tilde{p}^2 (1 - \epsilon \cos \pi \xi) F_{-3}(\chi) d\xi = \int_{-1}^1 (1 - \epsilon \cos \pi \xi)^3 F_{-3}(\chi) d\xi \quad (56)$$

The dimensionless load capacity $\tilde{W}$ and attitude angle θ_a are determined from the radial and tangential load integrals

$$\tilde{W}_R = \pi \int_{-1}^1 \tilde{p} \cos \pi \xi d\xi \quad \text{and} \quad \tilde{W}_\theta = -\pi \int_{-1}^1 \tilde{p} \sin \pi \xi d\xi$$

as follows:

$$\tilde{W} = \sqrt{\tilde{W}_R^2 + \tilde{W}_\theta^2} \quad (57)$$

and

$$\theta_a = \tan^{-1} (\tilde{W}_\theta / \tilde{W}_r) \quad (58)$$

and the dimensionless torque parameter $\tilde{T}$ is given by

$$\tilde{T} = \pi \int_{-1}^1 \frac{F_1(\chi)}{\tilde{h}} d\chi \quad (59)$$

The functions $F_n(\chi)$ have been evaluated numerically as functions of (χ) for $n = 2$ and 3 and F_n are evaluated analytically for $n = 1$ and -3 . In the cases where numerical evaluation is used, the values are tabulated in the computer and sub-routines are used for rapid interpolation. With the values $F_n(\chi)$ so determined, the Reynolds Equation (Equation (51)) is solved numerically with a Runge-Kutta scheme with various nesting procedures to satisfy the boundary conditions given by Equation (55) for $\xi_1 = 1$ or the closed foil boundary conditions given by Equation (56).

This concludes the description of the analysis of the ridged journal bearing. Solutions will be presented in terms of the dimensionless variables and parameters given by Equations (41) through (50) in the following section. A description and sample output of the program to calculate dimensional pressures, loads and power requirements will be given in Appendix I.

3. Design Curves for Ridged Journal Bearings

A number of parametric studies have been performed to predict the influence of the various journal bearing parameters on load capacity, minimum film thickness, attitude angles and power requirements. All curves have been generated for an infinitely long journal bearing as analyzed in the preceding section with an ambient pressure condition at $\xi_1 = 1$.

For purposes of comparison, the rigid bearing load capacity and torque parameters are plotted as a function of the shaft displacement to clearance ratio, ϵ , (this is simply the eccentricity ratio for a rigid bearing) at various values of the compressibility number Λ in Figure 6. The dimensionless load capacity $\tilde{W}$ is shown to increase with increasing eccentricity and to decrease with increasing compressibility number, Λ . Since the torque has been approximated by its Couette component, $\tilde{T}$ will be independent of Λ for a rigid journal bearing and is represented by the single curve in Figure 6 which increases with increasing eccentricity. The minimum film thickness, which is not shown, will be equal to $1 - \epsilon$ for a rigid bearing.

The effect of the bump foil compliance on load capacity is shown in Figures 7 through 10 where the top foil is kept rigid ($\alpha_T = 0$) and the dimensionless compliance parameter for the bump foil is varied from $\alpha_B = 1$ to $\alpha_B = 20$. The load capacity curves shown which are in the form of $\tilde{W}$ versus ϵ represent the load-deflection curves for the shaft and the slopes of these curves represent the combined stiffness of the gas film together with the foil. It can be seen that at large shaft displacements, the curves obtained at different values of the compressibility number become very nearly parallel indicating that the stiffness in that region is governed largely by the foil stiffness.

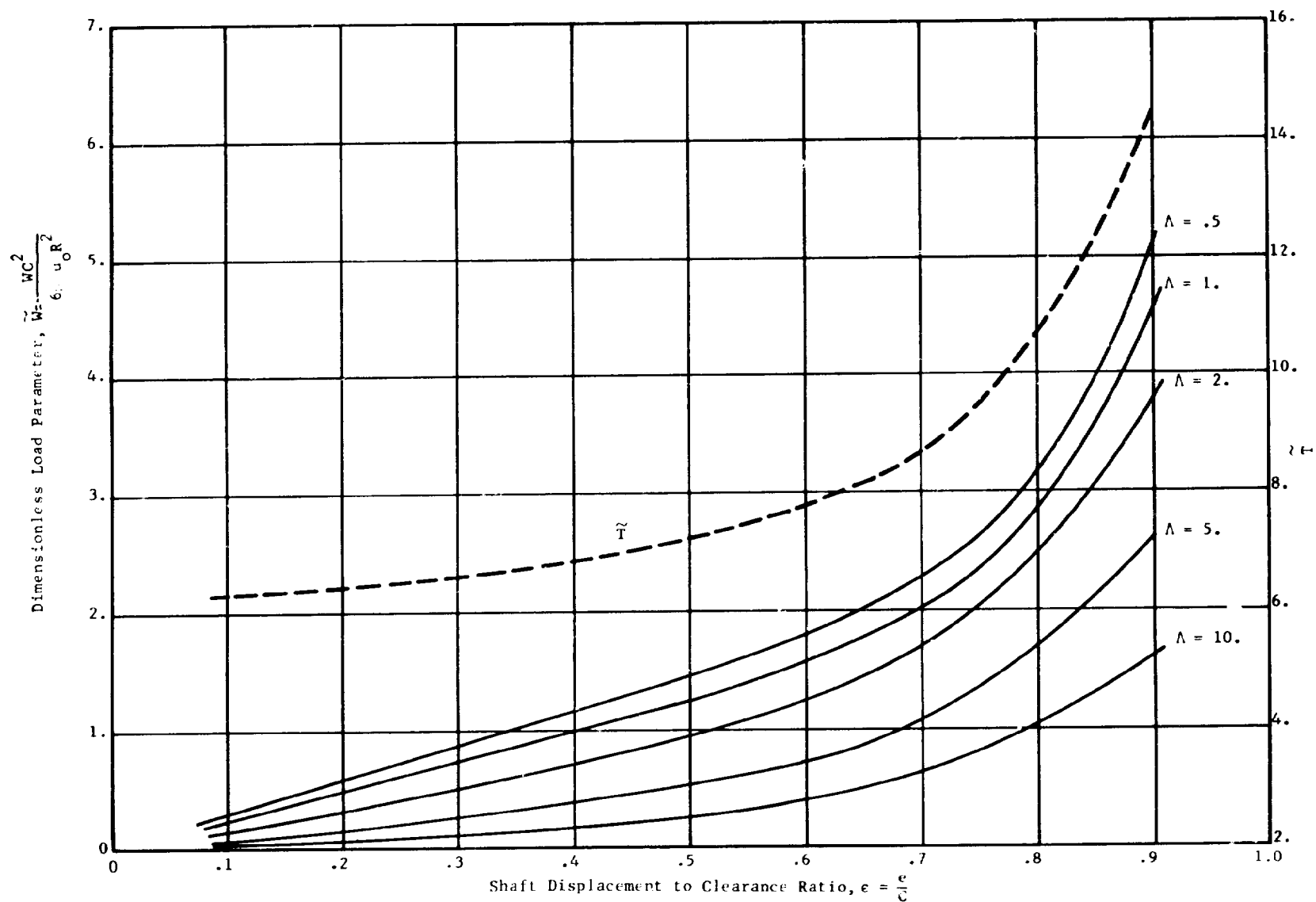


Fig. 6 Dimensionless Load Capacity and Torque for Ridged Journal Bearing, $\alpha_T = \alpha_B = 0$

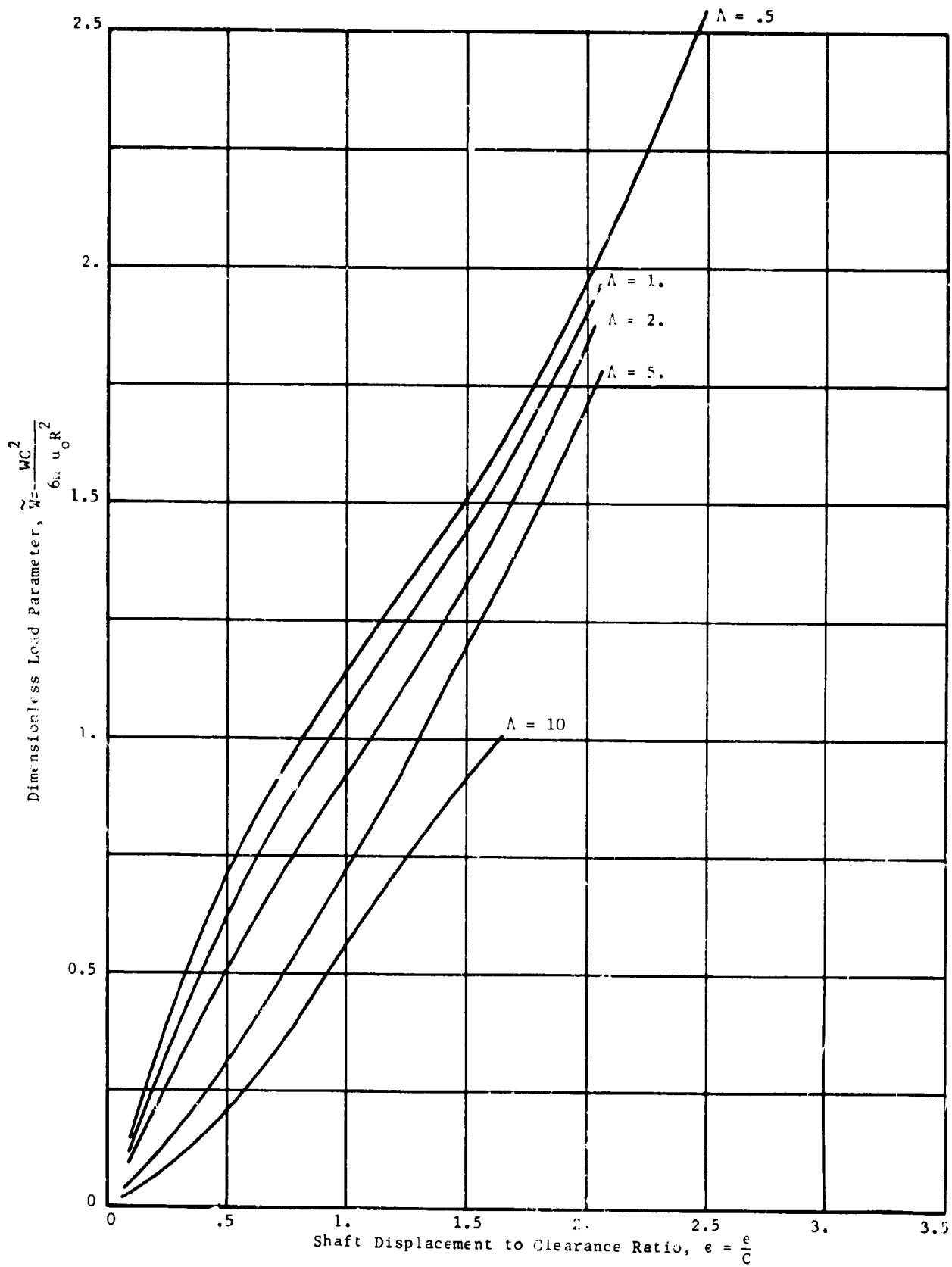


Fig. 7 Dimensionless Load Capacity Vs. Shaft Displacement for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 1$

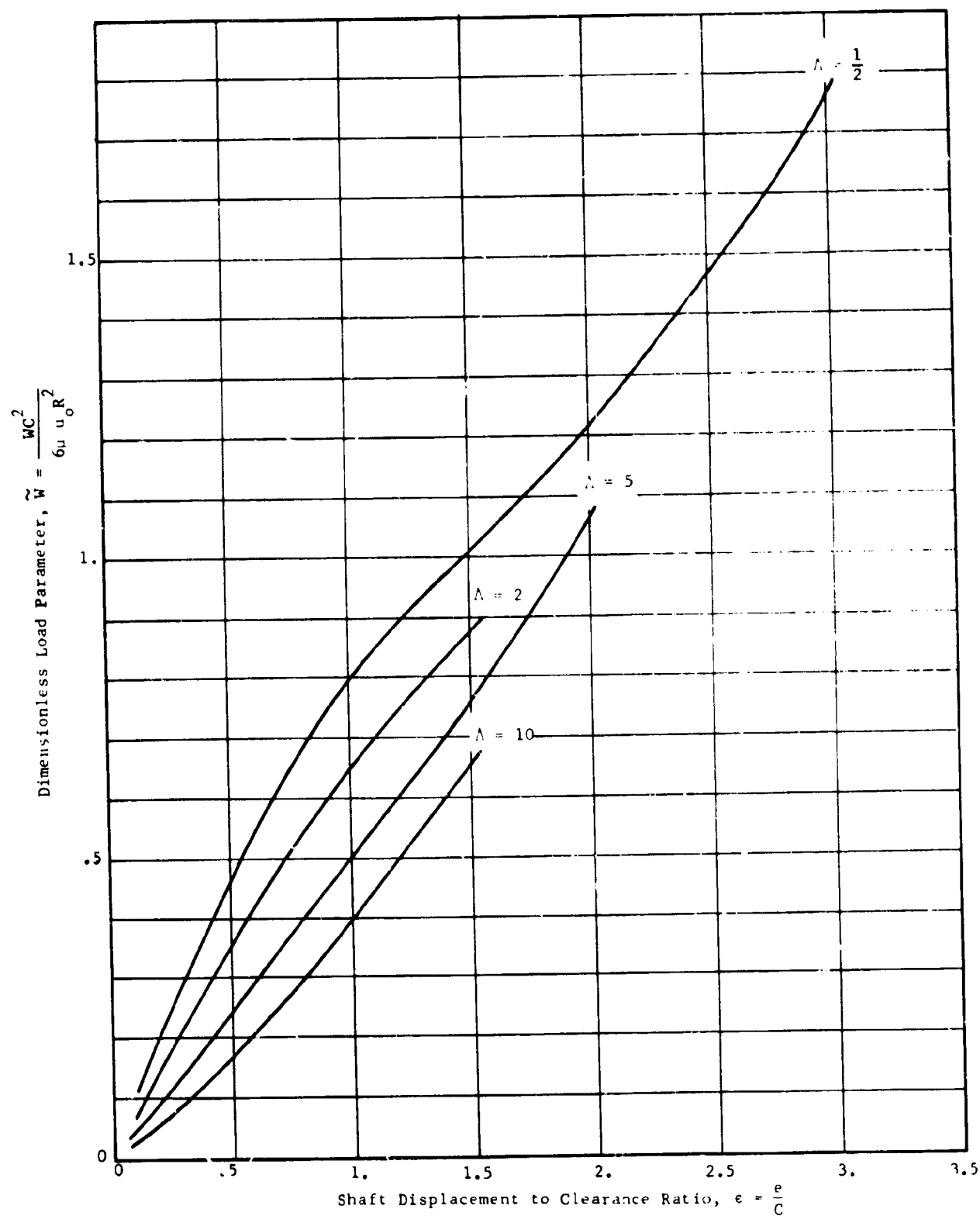


Fig. 8 Dimensionless Load Capacity Vs. Shaft Displacement
for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 2$

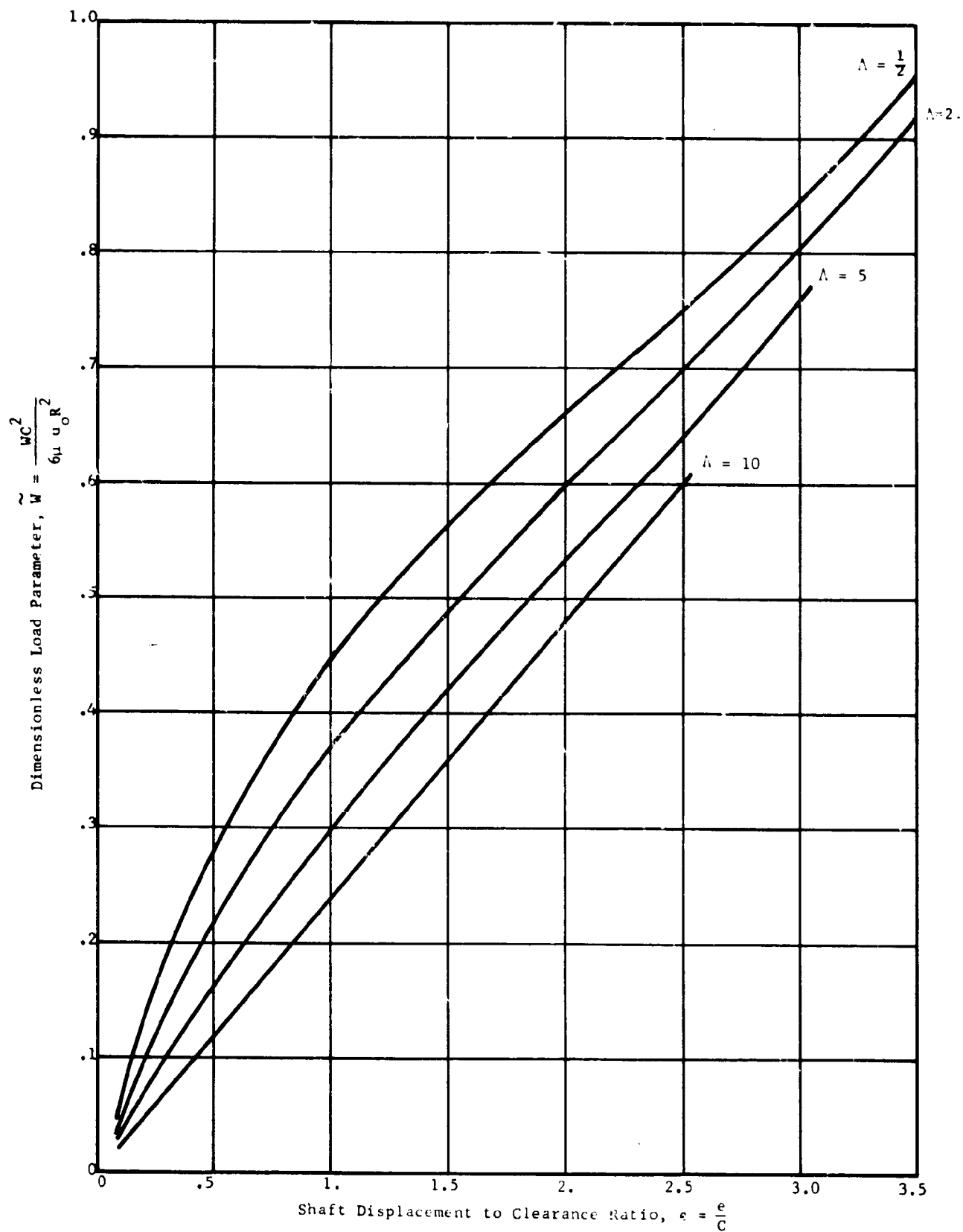


Fig. 9 Dimensionless Load Capacity Vs. Shaft Displacement for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 5$

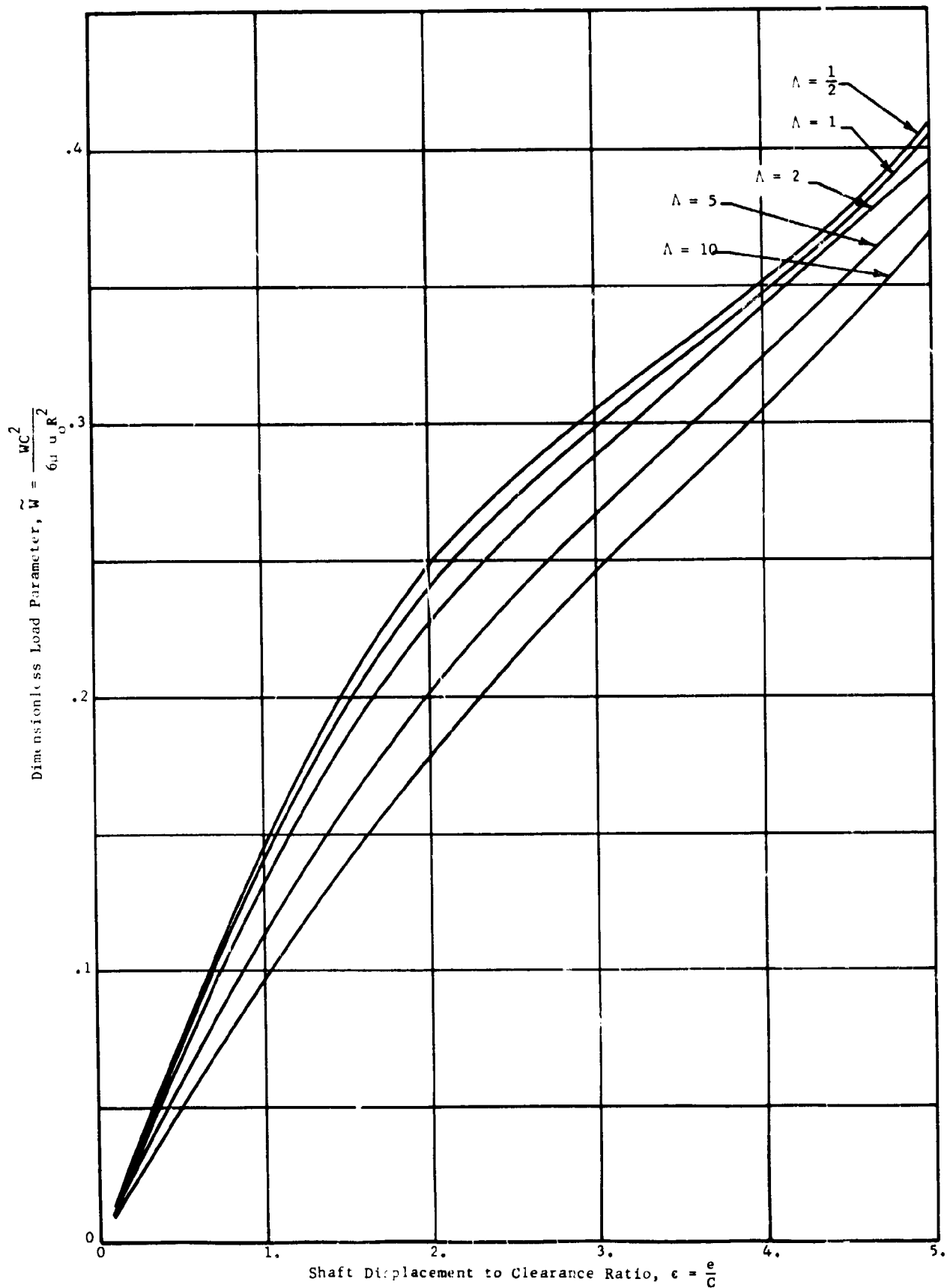


Fig. 10 Dimensionless Load Capacity Vs. Shaft Displacement for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 20$

The effect of the upper tape compliance on the load capacity, is shown in Figure 11 for α_T and α_B both equal to 2, and it can be seen by comparison with Figure 8 that the compliance of the top foil results in a slightly reduced load capacity.

The variation of the minimum film thickness $\tilde{h}_{min}$ with ϵ for various values of the compressibility number are shown in Figures 12-15 for $\alpha_T = 0$ and from α_B varying from 1 to 20. It can be seen from these figures that the minimum film thickness is not a function of the shaft displacement ϵ alone but will also depend on the compressibility number as a result of the compliance of the bearing.

When α_B becomes sufficiently large, it is actually possible to have a situation where the film thickness increases with shaft displacement. This will occur for highly compliant foil bearings generating sufficient pressures to force back the foil by amounts greater than the shaft displacement resulting in an increase in film thickness. This should not be construed to imply negative stiffness since the stiffness relates to the slope of the $\tilde{h}$ vs. ϵ curves which were shown previously to always be positive. When the α_B becomes as high as 20, as shown in Figure 15, a substantial peaking in the $\tilde{h}_{min}$ vs. ϵ curves is seen.

The effect of α_T on the minimum film thickness may be seen by comparing the results of Figure 16 with those in Figure 13, and again it is shown that the compliance of the upper foil results in a small decrease in film thickness.

Power requirements may be estimated from the dimensionless torque curves given in Figures 17-22. It can be seen that the power can dip slightly with increasing ϵ for small values α_B and dip markedly for larger values of α_B . The reason for this drop in power with increasing load is again caused by the pressure pushing back the foils, thereby opening the clearance and reducing the power. The effect of the upper tape compliance on power may be observed by comparing the curves in Figure 21 with those in Figure 18 to see that the compliance of the upper tape results in a slight increase in power.

The effect of the bump foil compliance on the attitude angle is shown in Figure 22 for $\alpha_T = 0$ and $\Lambda = 1$. It can be seen that increasing the bump foil compliance tends to decrease the attitude angle, thereby possibly providing more stability. This decrease will be lower at higher values of Λ where the attitude angles, in general, are substantially lower.

The effect of the top foil compliance on the attitude angle for $\alpha_B = 2$ is shown in Figure 23 for $\Lambda = 1$ and $\Lambda = 5$. In either case no significant effect on attitude angle appears to result from the compliance of the top foil.

The design curves thus indicate that the ridged bearing can be designed to provide independent control of stiffness with perhaps improved stability over a rigid bearing and can provide equivalent hydrodynamic film thickness. The compliance of the bearing will be dictated by the lower foil at large displacements. The upper foil should be made sufficiently rigid so that α_T is not excessively large.

The most serious omission in the analyses with regard to their quantitative accuracy is believed to be the effects of side leakage which should be taken into account as a next step in improving the accuracy of the ridged bearing analyses.

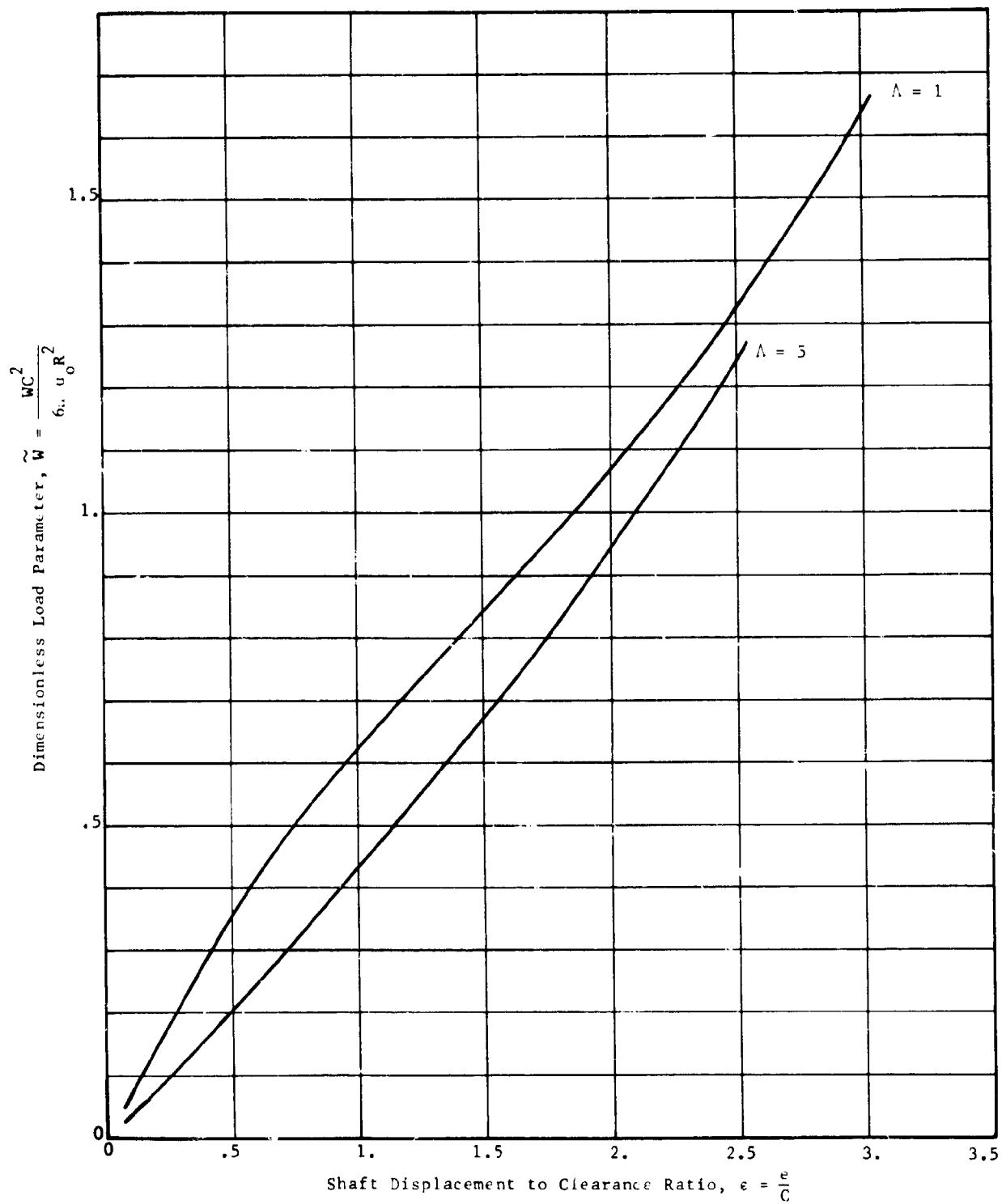


Fig. 11 Dimensionless Load Capacity Vs. Shaft Displacement
for Ridged Journal Bearing, $\alpha_T = 2$, $\alpha_B = 2$

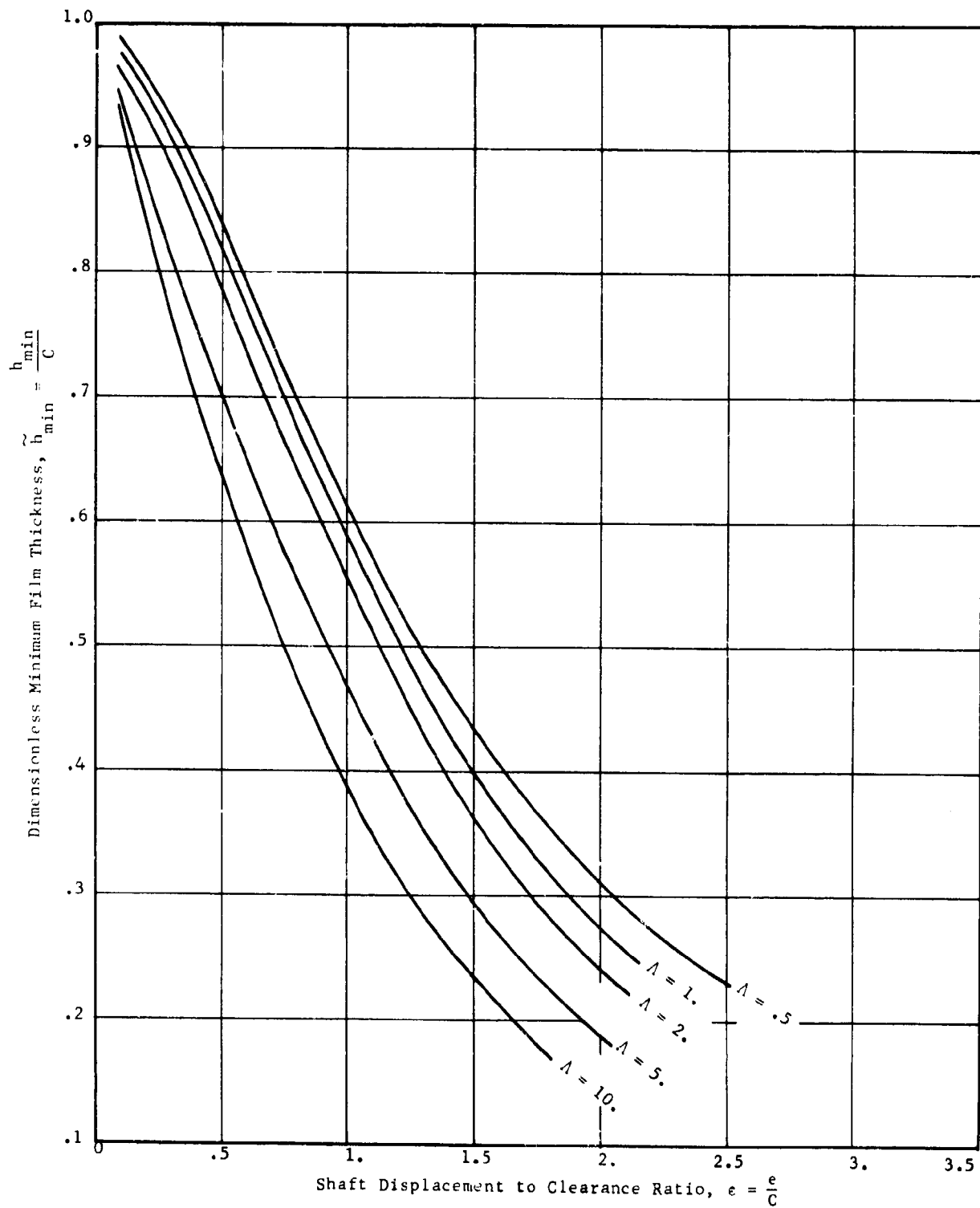


Fig. 12 Dimensionless Minimum Film Thickness Curves for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 1$

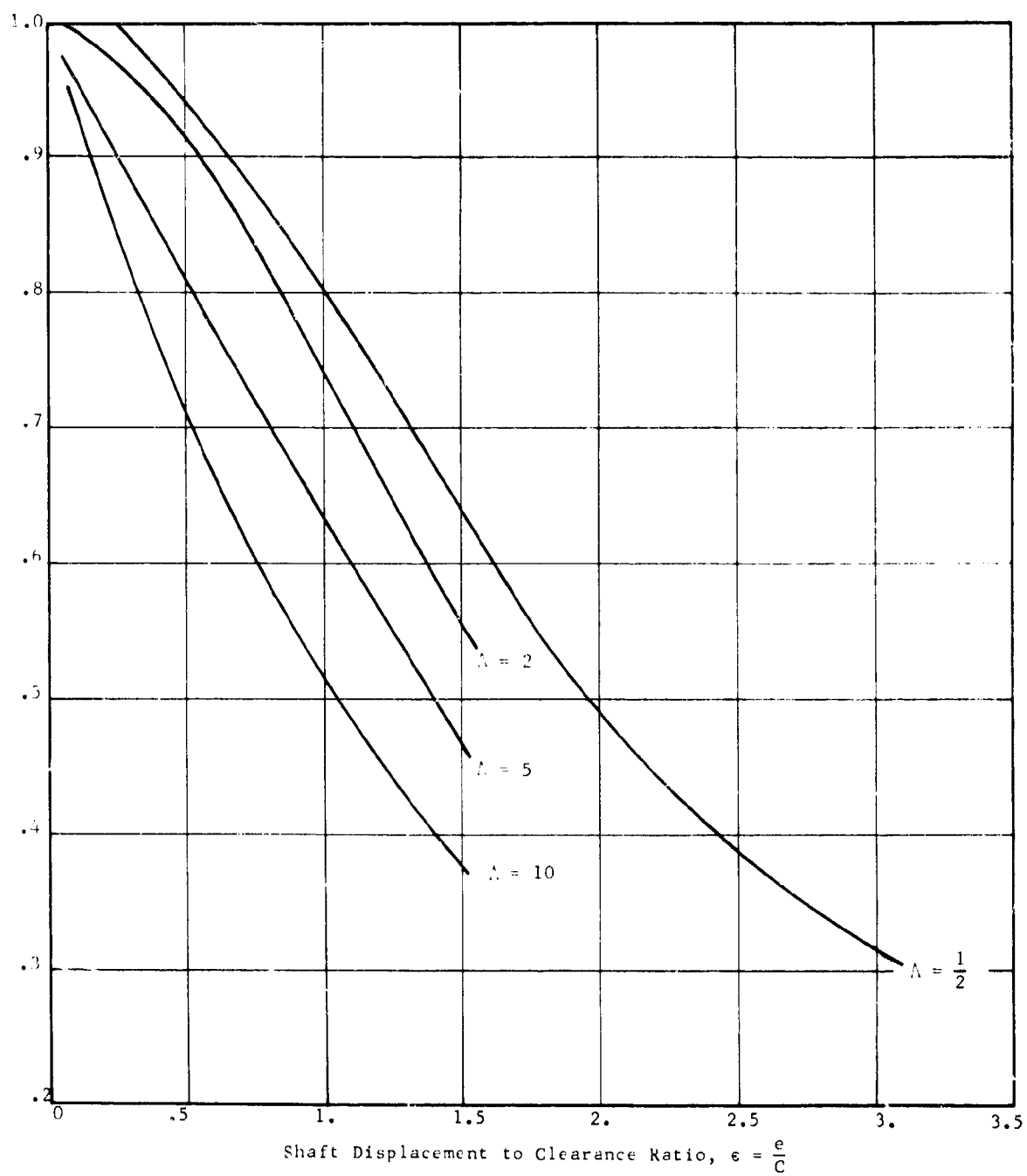


Fig. 13 Dimensionless Minimum Film Thickness Curves for
Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 2$

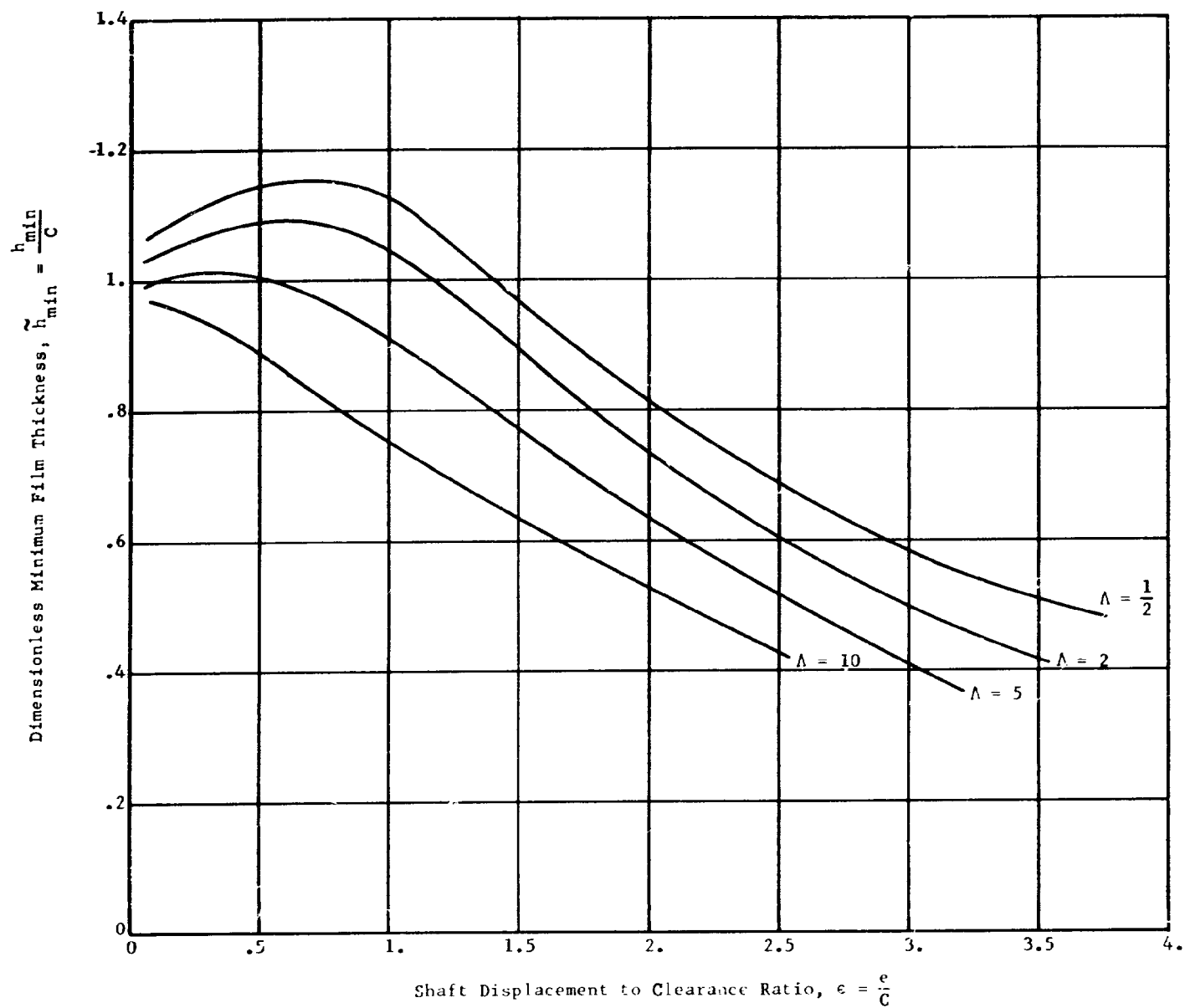


Fig. 14 Dimensionless Minimum Film Thickness Curves for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 5$

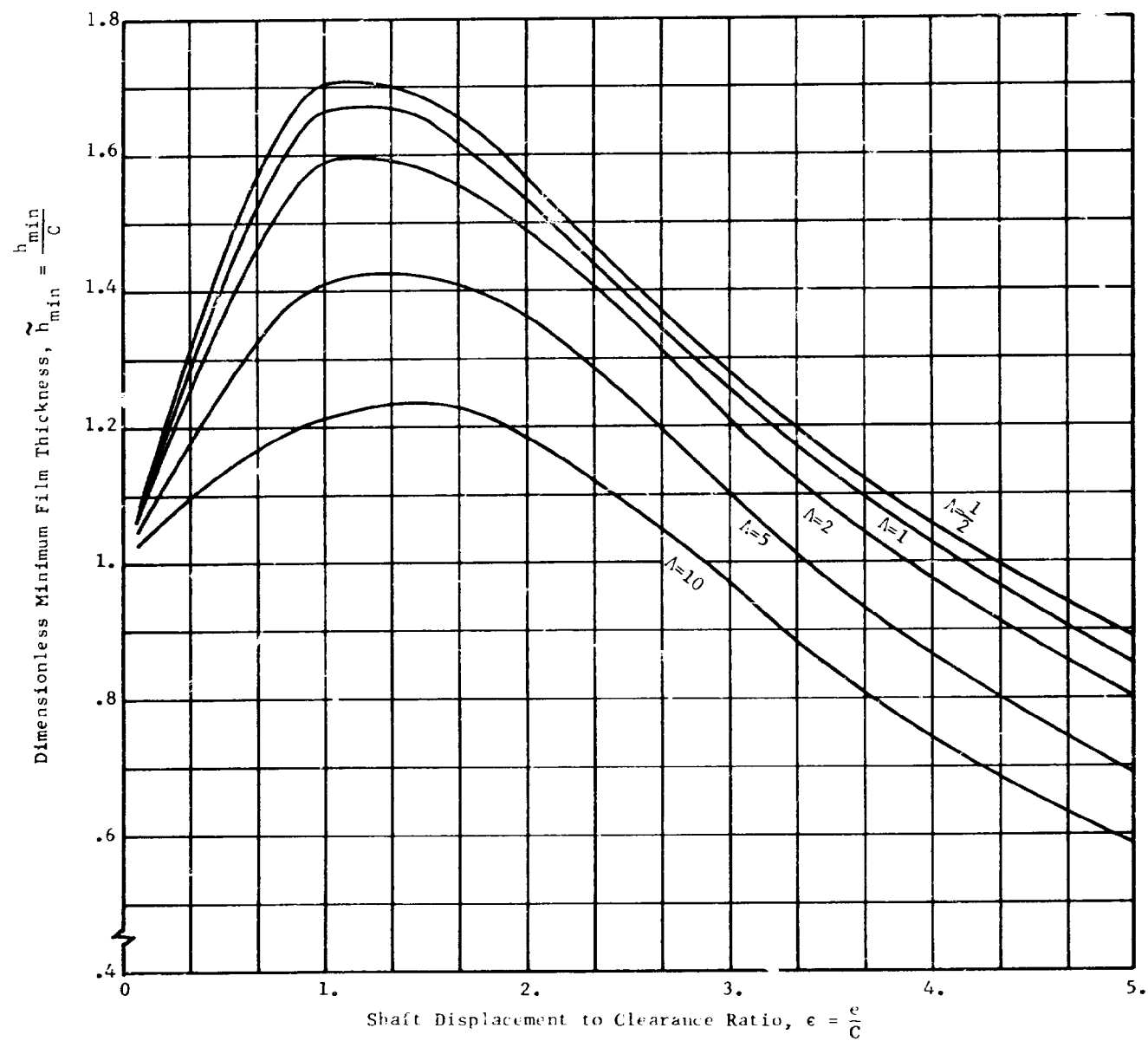


Fig. 15 Dimensionless Minimum Film Thickness Curves for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 20$

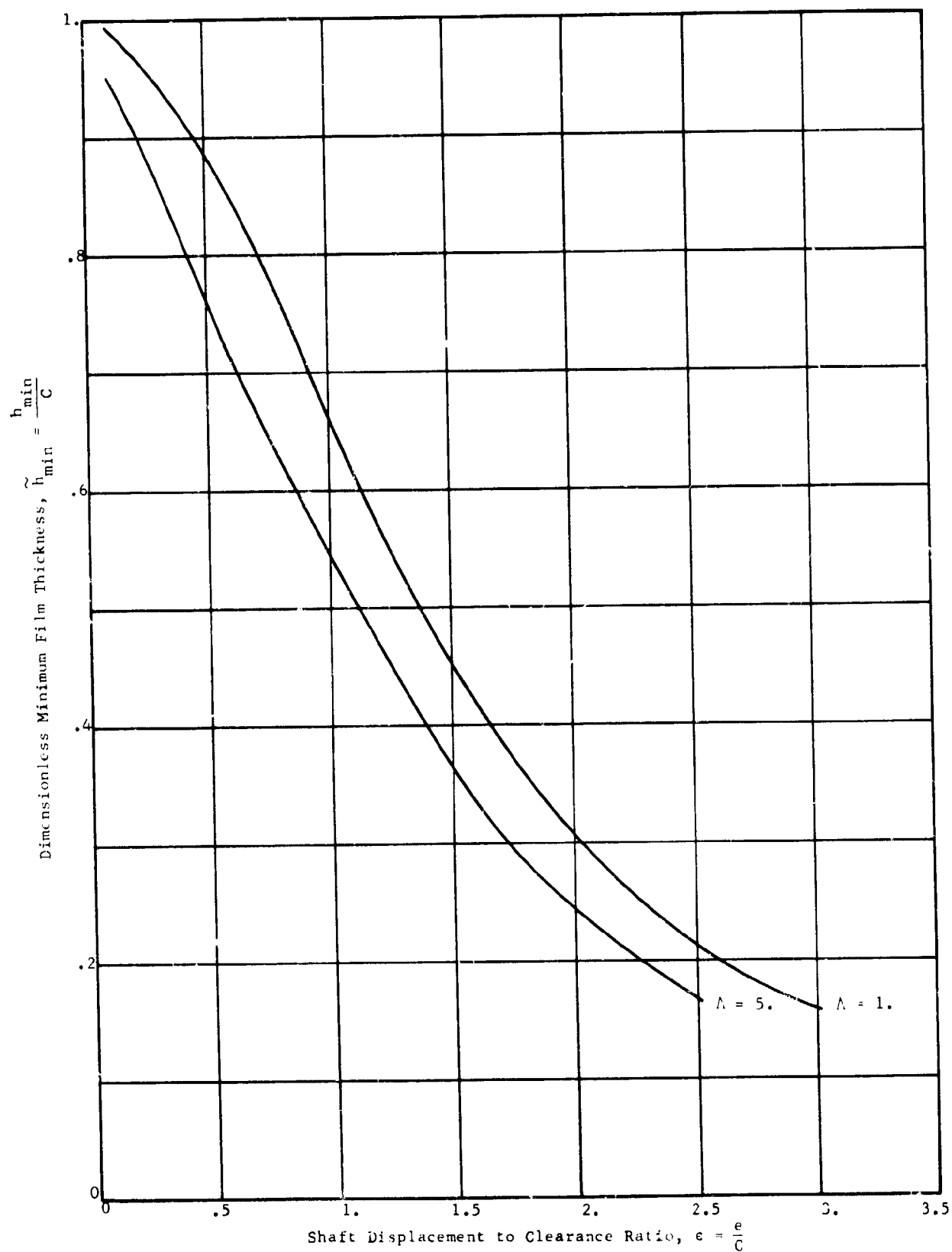


Fig. 16 Dimensionless Minimum Film Thickness Curves for Ridged Journal Bearing, $\alpha_T = 2$, $\alpha_B = 2$

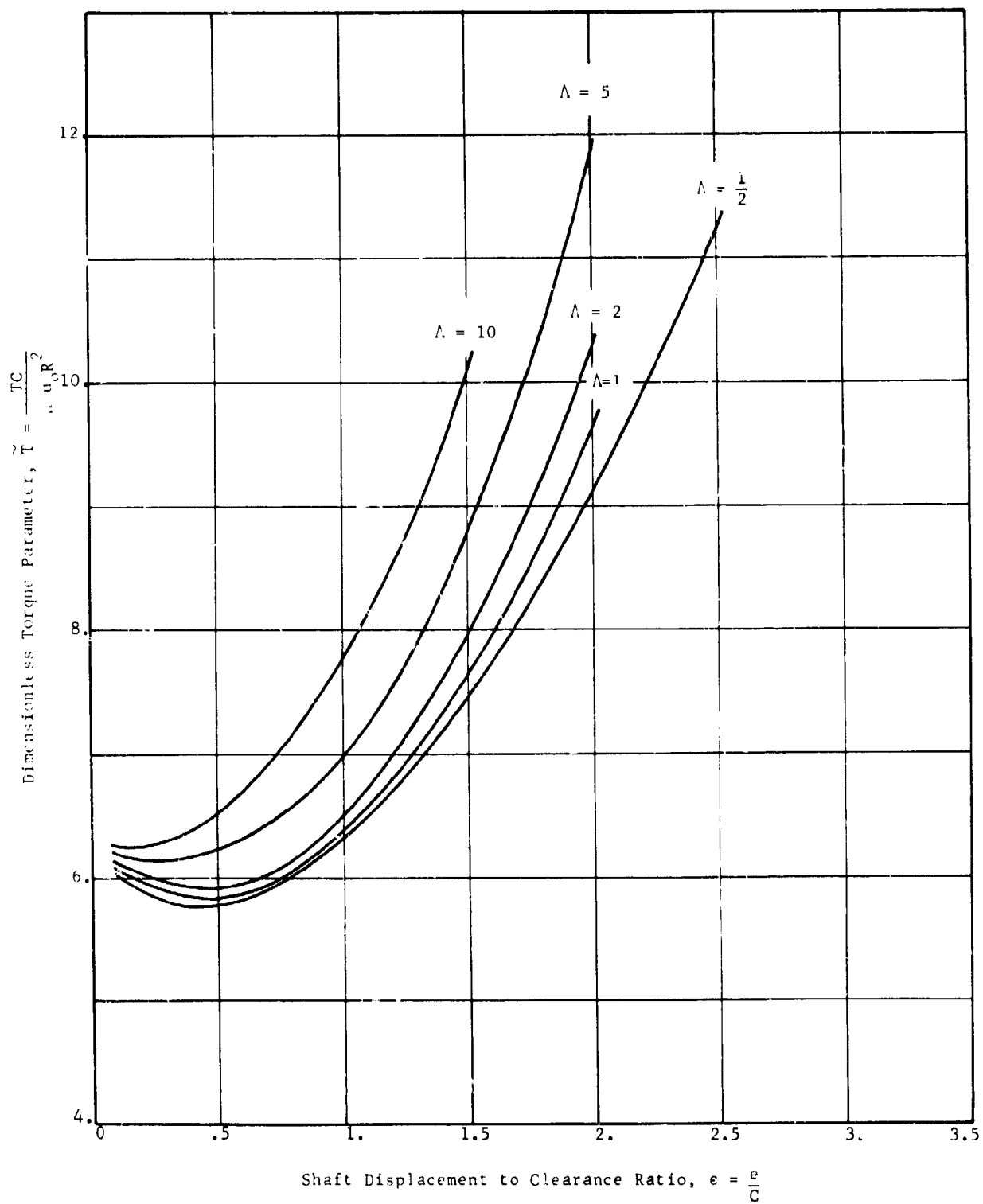


Fig. 17 Dimensionless Torque Curves for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 1$

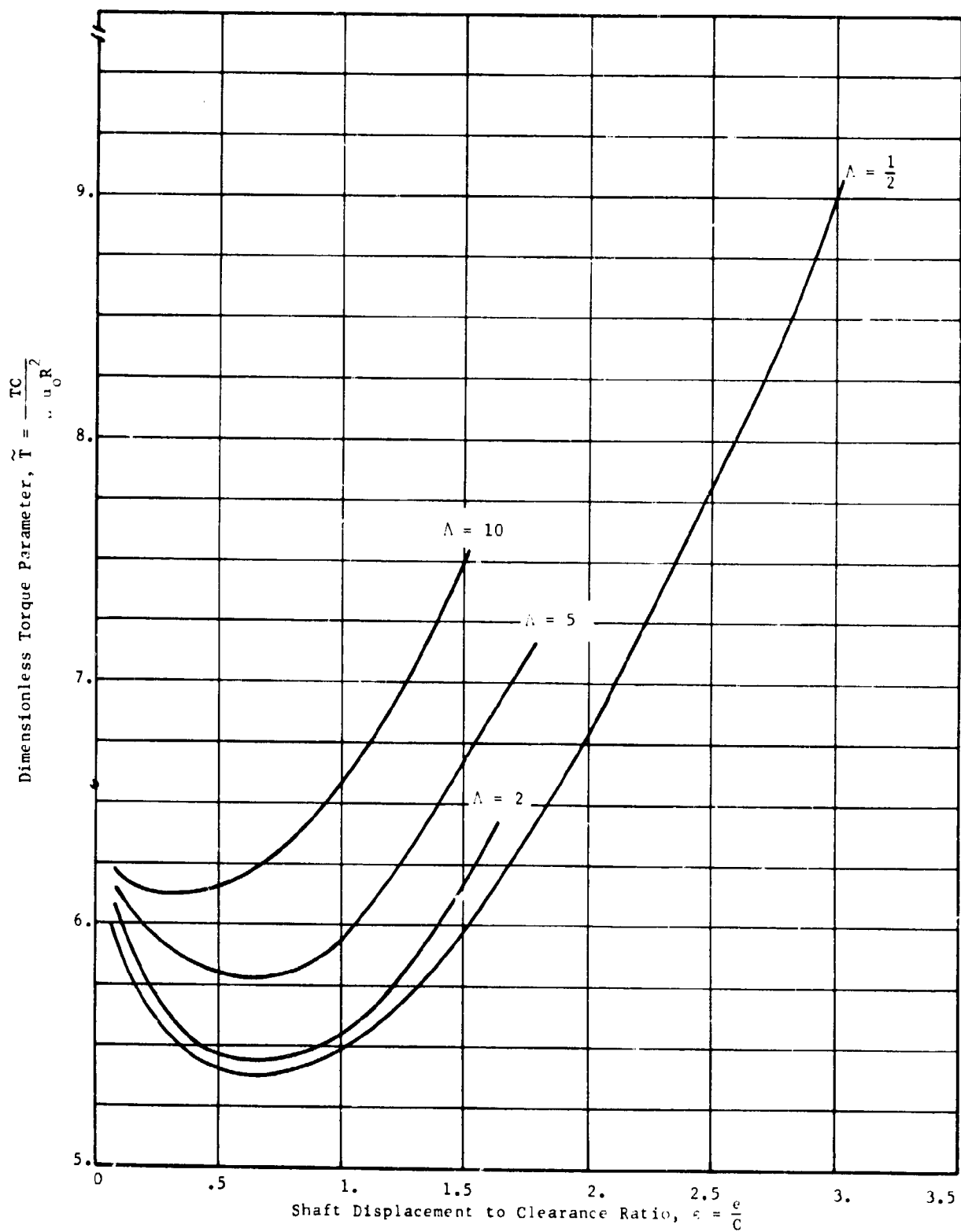


Fig. 18 Dimensionless Torque Curves for Rided Journal Bearing, $\alpha_T = 0$, $\alpha_B = 2$

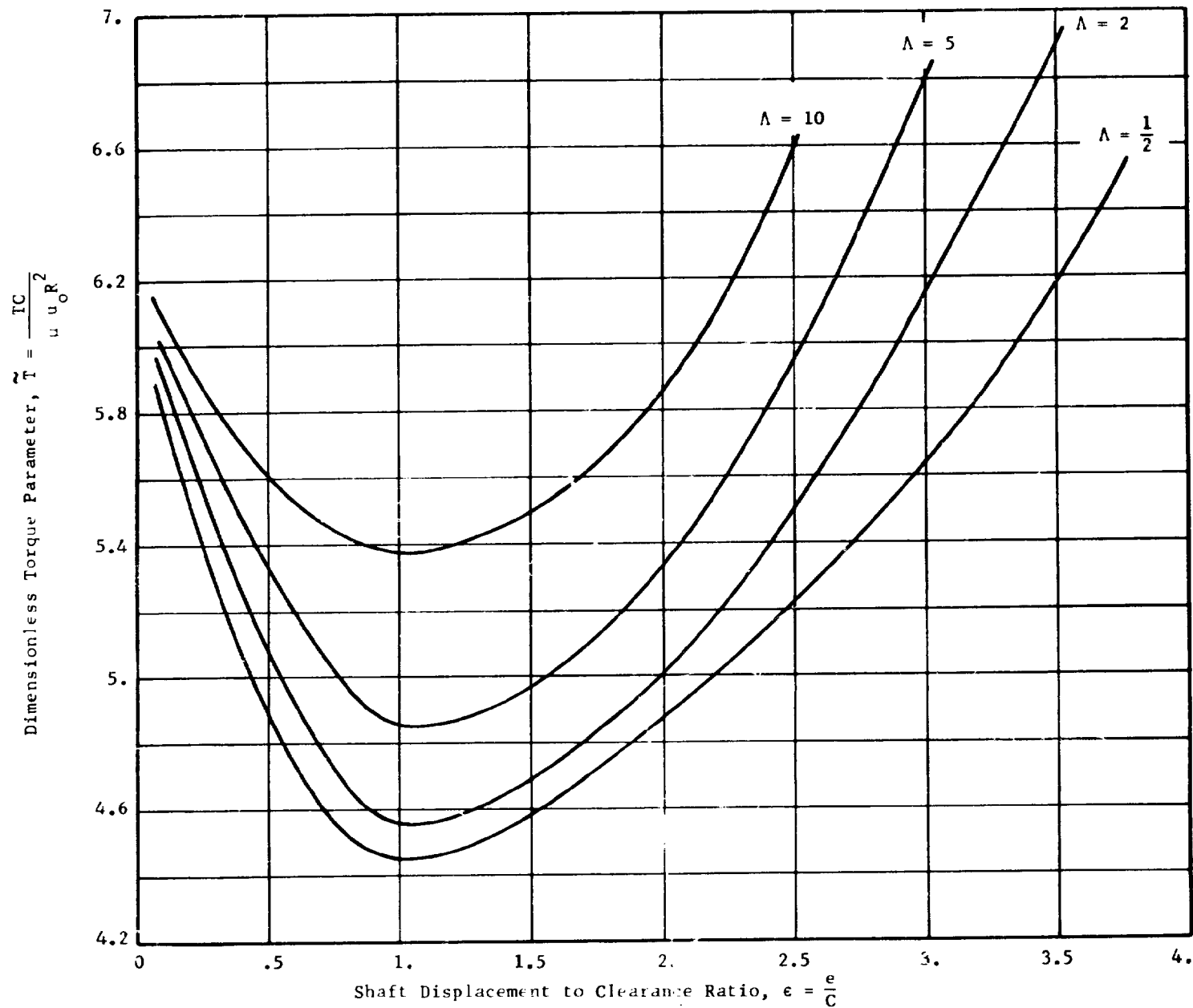


Fig. 19 Dimensionless Torque Curves for Ridged Journal Bearing, $\alpha_T = 0$, $\alpha_B = 5$

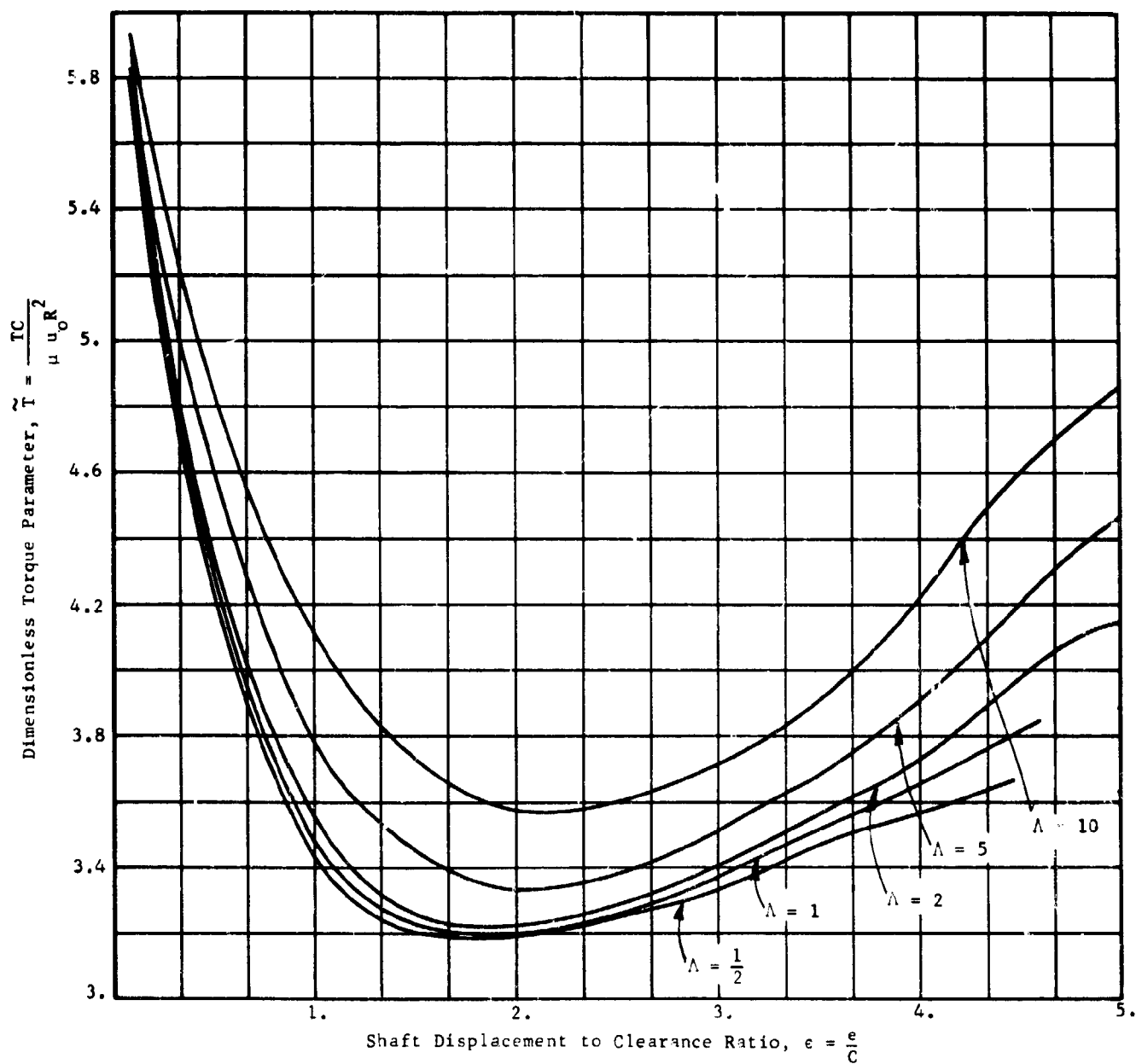


Fig. 20 Dimensionless Torque Curves for Rided Journal Bearing, $\alpha_T = 0$, $\alpha_B = 20$

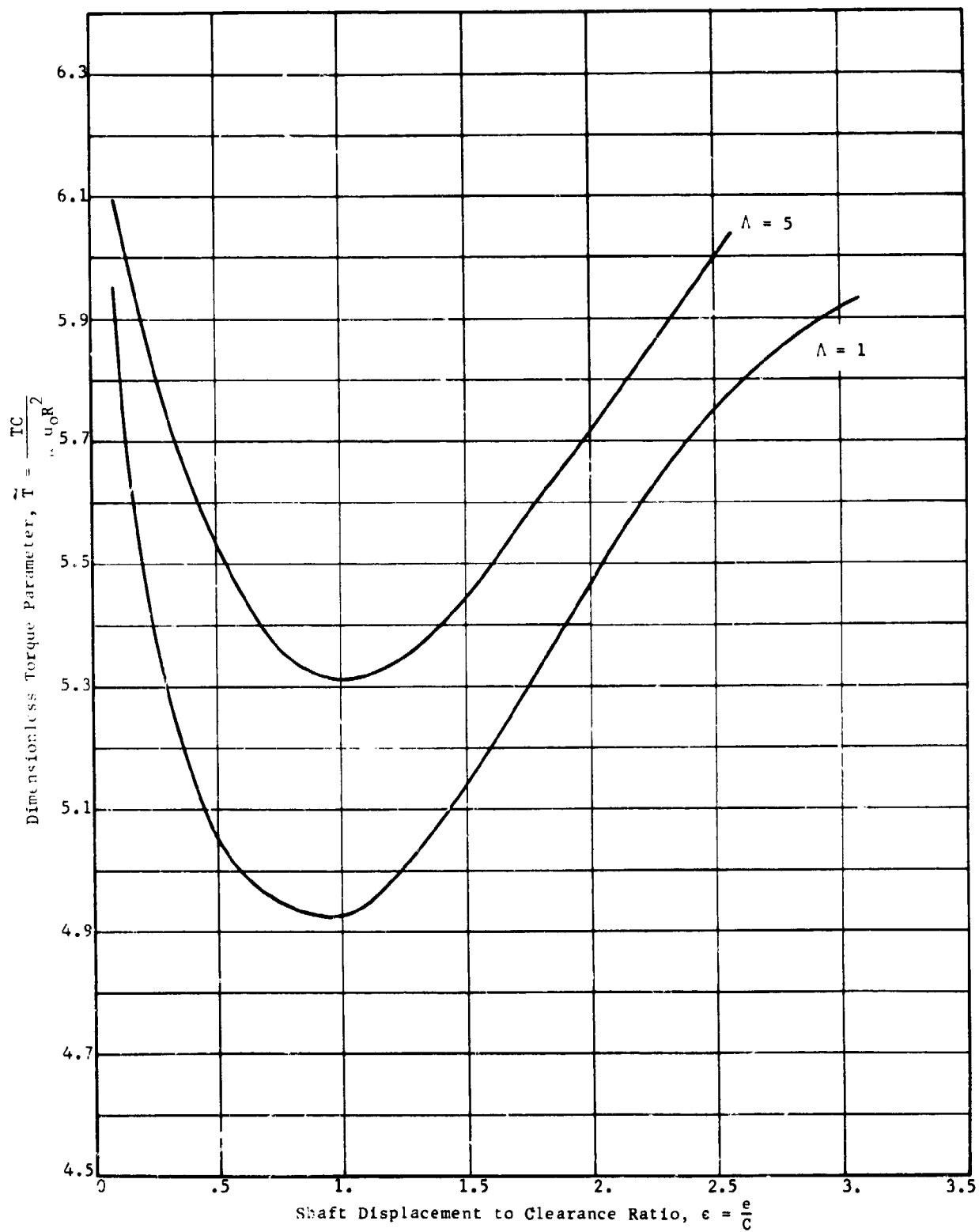


Fig. 21 Dimensionless Torque Curves for Ridged Journal Bearing, $\alpha_T = 2$, $\alpha_B = 2$

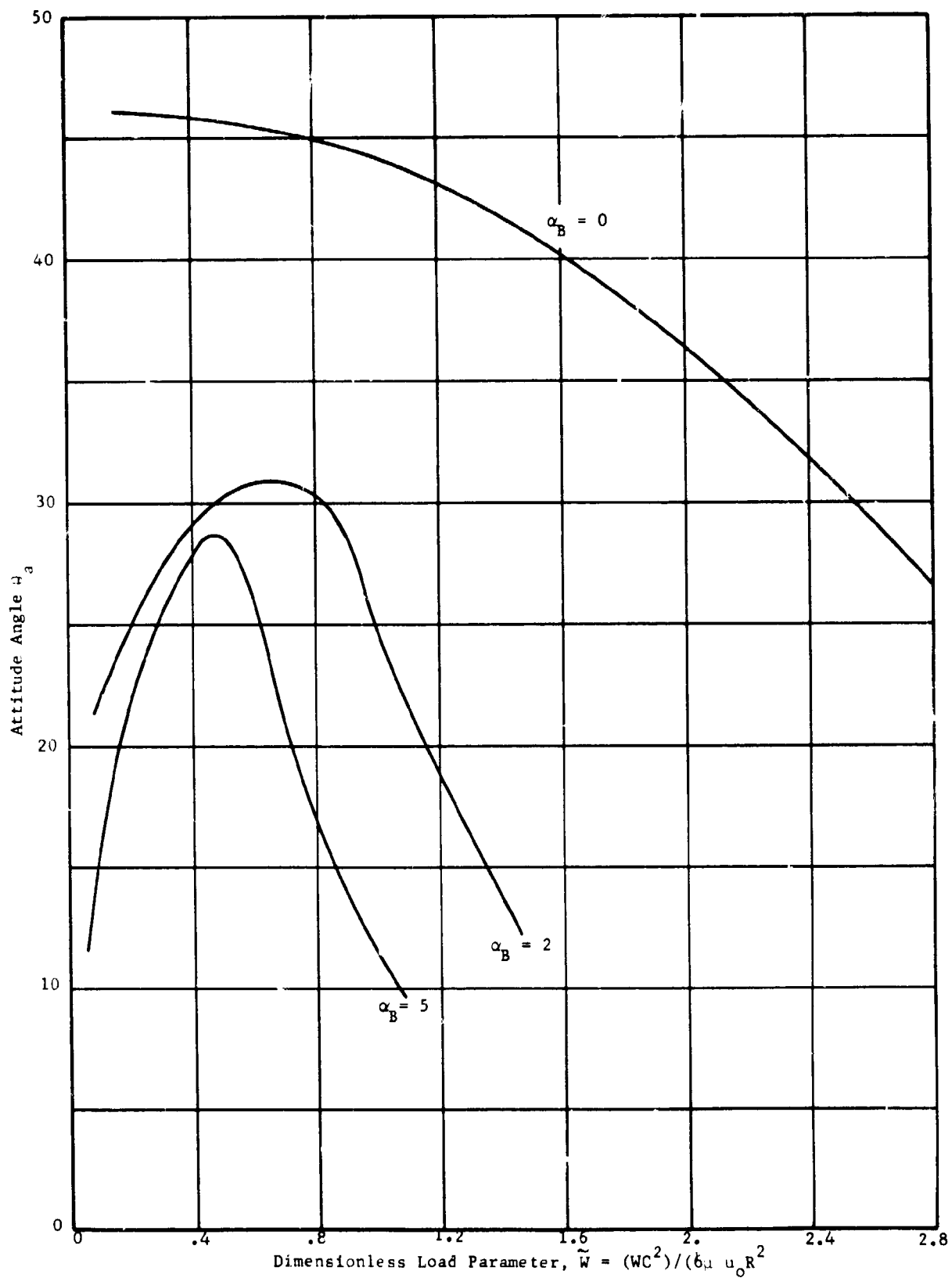


Fig. 22 The Effect of Bottom Foil Compliance on Attitude Angle for a Ridged Journal Bearing, $\alpha_T = 0$, $\Lambda = 1$

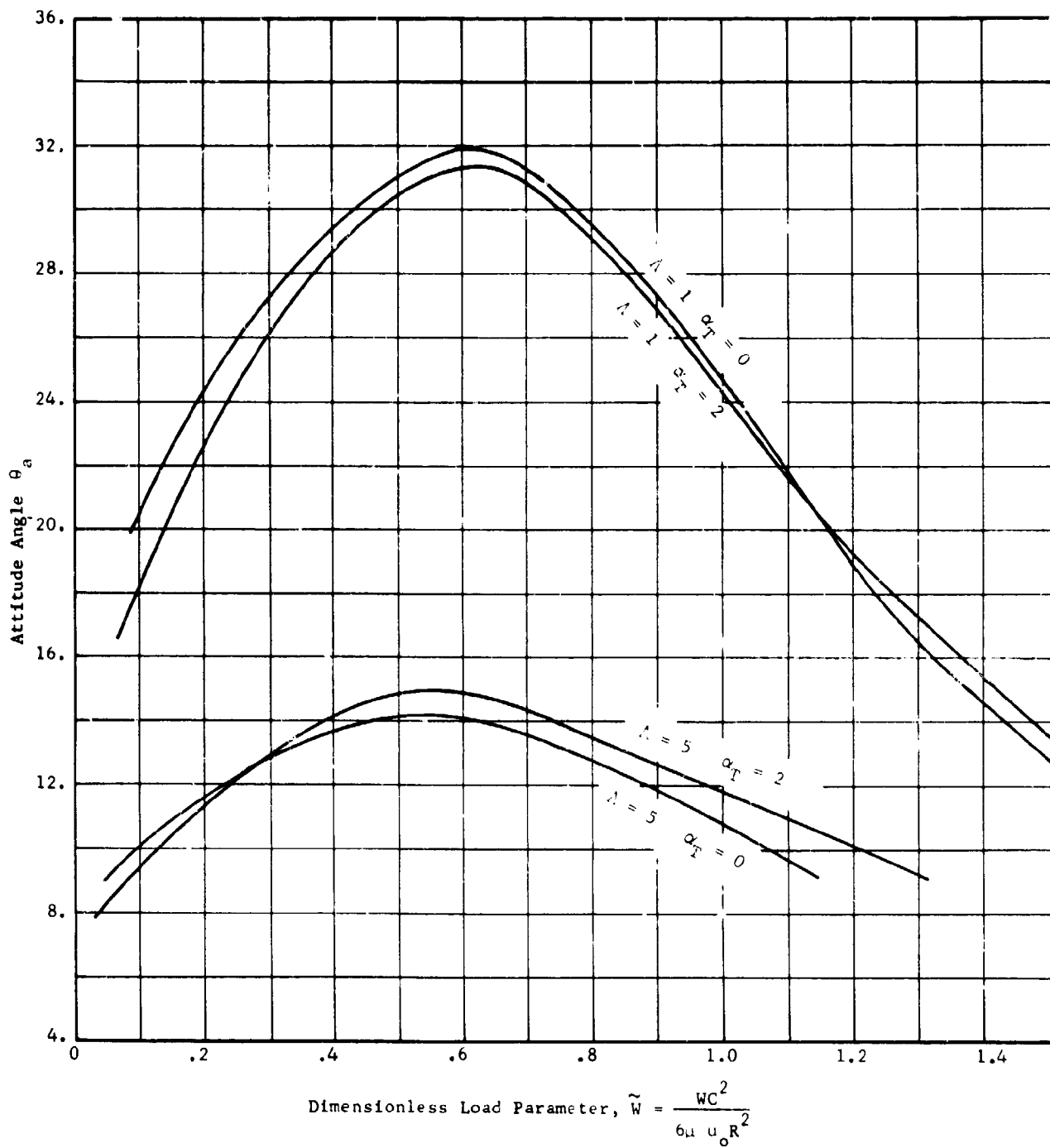


Fig. 23 The Effect of Top Foil Compliance on Attitude Angle for a Ridged Journal Bearing, $\alpha_B = 2$

4. Performance Analysis of Ridged Thrust Bearing

The convergent wedge necessary for hydrodynamic action is generated automatically in the ridged journal bearing as the shaft center moves away from the bearing center. However, in the ridged thrust bearing, the basic geometry is initially flat, and does not automatically develop a hydrodynamic wedge. One can think of a series of pads with flat top foils resting over the bump foils. In such situations, the necessary hydrodynamic wedge will be absent and under "ideal" circumstances of perfectly smooth parallel surfaces, one would anticipate that no load support would result. A mechanism must thus be provided for the development of a hydrodynamic wedge. Such a mechanism would be prevalent near the leading edge of each pad if the bump heights were varied to form a tapered inlet.

While full experimental data is lacking, it has been found experimentally that the load support characteristics are very sensitive to the inlet geometry. On this basis, a mechanism has been postulated as shown schematically in Figure 24.

An analysis has been developed starting with the following set of assumptions:

1. The bump height is large compared with the film thickness.
2. The film thickness is small compared with the elastic deflection of the bumps.
3. No side leakage.
4. The inlet length ℓ_1 is small compared with the pad length ℓ_p .
5. The top foil is designed with sufficient rigidity so that deflection of the top foil between bumps is small.
6. The thrust pads are approximated by rectangular pads having mean circumferential length.

If the deflections in the pad region are large compared with the film thickness, then the pressure in the pad region will be very nearly equal to the static contact pressure which will have a constant value of p_0 . The deflection in the pad region would then be $p_0 S/K_B$. Much like other elastohydrodynamic systems, the film will be developed in the inlet region ($x < 0$) and the pressures are relatively small and the film thicknesses are relatively large. The Reynold's Equation must be solved to determine the pressure profile. The situation is thus much like the usual situation of inverse hydrodynamics where an inlet zone of known shape is used in Reynold's Equation to determine the film thickness that then prevails in an inverse region which, in this case, would be the pad region.

Reynold's Equation for the inlet zone is given by

$$\frac{dp}{dx} = 6 \mu u_0 \frac{(h - k/p)}{h^3}, \quad x < 0. \quad (60)$$

The boundary conditions on Equation (60) at $x = 0$ are obtained from the consideration that the film thickness and pressure as well as the flow rate must be continuous at $x = 0$. These conditions will be satisfied if the following boundary

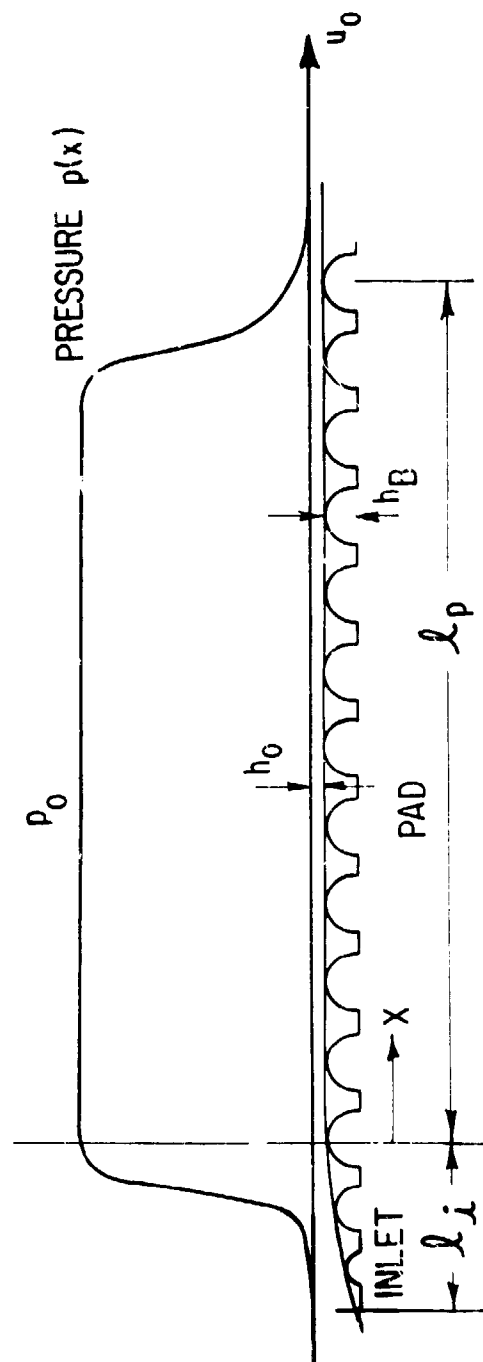


Fig. 24 Schematic of ridged thrust bearing

conditions are satisfied:

$$h = h_o, \quad dp/dx = 0, \quad p = p_o \quad \text{at} \quad x = 0 \quad (61)$$

The vanishing of the pressure gradient at $x = 0$ insures continuity of the pressure gradient between the inlet and pad regions and hence continuity of flow. When one combines Equation (61) with (60) to express the constant k in terms of p_o and h_o , the relationship

$$\frac{dp}{dx} = 6 \mu u_o \frac{(h - p_o h_o / p)}{h^3}, \quad x < 0 \quad (62)$$

is obtained which automatically satisfies the pressure gradient condition.

There are a variety of ways the inlet land could be fabricated. Almost any shape could, in principle, be obtained with suitable tooling or stamping techniques. It is assumed here that the inlet zone can be expressed by a quadratic function that joins smoothly with the pad region. If the rise in the inlet zone is equal to the bump height h_B and occurs over a length l_i and the film thickness is displaced by an amount h_o from the inlet shape, then the film thickness in the inlet zone will be given by the expression

$$h = h_o + h_B \left(\frac{x}{l_i} \right)^2, \quad x < 0. \quad (63)$$

Since the film thickness is very small, compared with the bump height, the pressure will become ambient long before the start of the inlet land, hence rather than requiring the pressure to be ambient at $x = -l_i$ one can with little loss of accuracy take the pressure to be ambient far upstream ($x = -\infty$), and we have the upstream condition given by

$$p = p_a \quad \text{at} \quad x \rightarrow -\infty. \quad (64)$$

Equations (61) through (64) are sufficient to determine the film thickness, h_o as functions of the various input variables.

The above system of equations may be expressed in terms of the following dimensionless variables and parameters:

$$\frac{h}{p} = \frac{p - p_a}{p_o - p_a} \quad (65)$$

$$\zeta = -\sqrt{\frac{h_B}{h_O}} \frac{x}{\ell_i} \quad (66)$$

$$\hat{p}_a = p_a/p_O \quad (67)$$

$$\Lambda_1 = \frac{6\mu u_O \ell_i}{h_O^2 (p_O - p_a)} \sqrt{\frac{h_O}{h_B}} \quad (68)$$

The dimensionless Reynold's Equation may then be written as

$$\frac{d\hat{p}}{d\zeta} = -\frac{\Lambda_1}{(1 + \zeta^2)^3} \left[1 + \zeta^2 - \frac{1}{\hat{p}_a + (1 - \hat{p}_a) \hat{p}} \right] \quad (69)$$

with boundary conditions given by

$$\hat{p} = 1 \text{ at } \zeta = 0, \quad \hat{p} = 0 \text{ at } \zeta \rightarrow \infty \quad (70)$$

The solution to Equations (69) and (70) constitutes the determination of Λ_1 as a function of $\hat{p}_a$. One may obtain the solution by numerically solving Equation (69) by Runge-Kutta procedure. This has been done, and the results are expressed in terms of the incompressible film thickness as discussed below.

For the case of incompressible flow, $\hat{p}_a$ will be equal to unity and Equation (69) will be linear. One may then integrate both sides of Equation (69) for the incompressible case, between the limits of $\zeta = 0$ and $\zeta = \infty$ to obtain the results Λ_1 (incompressible) = $16/\pi$ which may be expressed in terms of the incompressible film thickness h_{OI} as follows:

$$\frac{6\mu u_O \ell_i}{h_{OI}^2 (p_O - p_a)} \sqrt{\frac{h_{OI}}{h_B}} = \frac{16}{\pi} \quad (71)$$

The solution to Equations (69) and (70) for the compressible case is presented in the form of h_O/h_{OI} as a function of $\hat{p}_a$ in Figure 25.

One may calculate p_O in terms of the total applied load, W^* , and total pad area, A_p , which may be considered design input variables as follows:

$$p_O = W^*/A_p + p_a \quad (72)$$

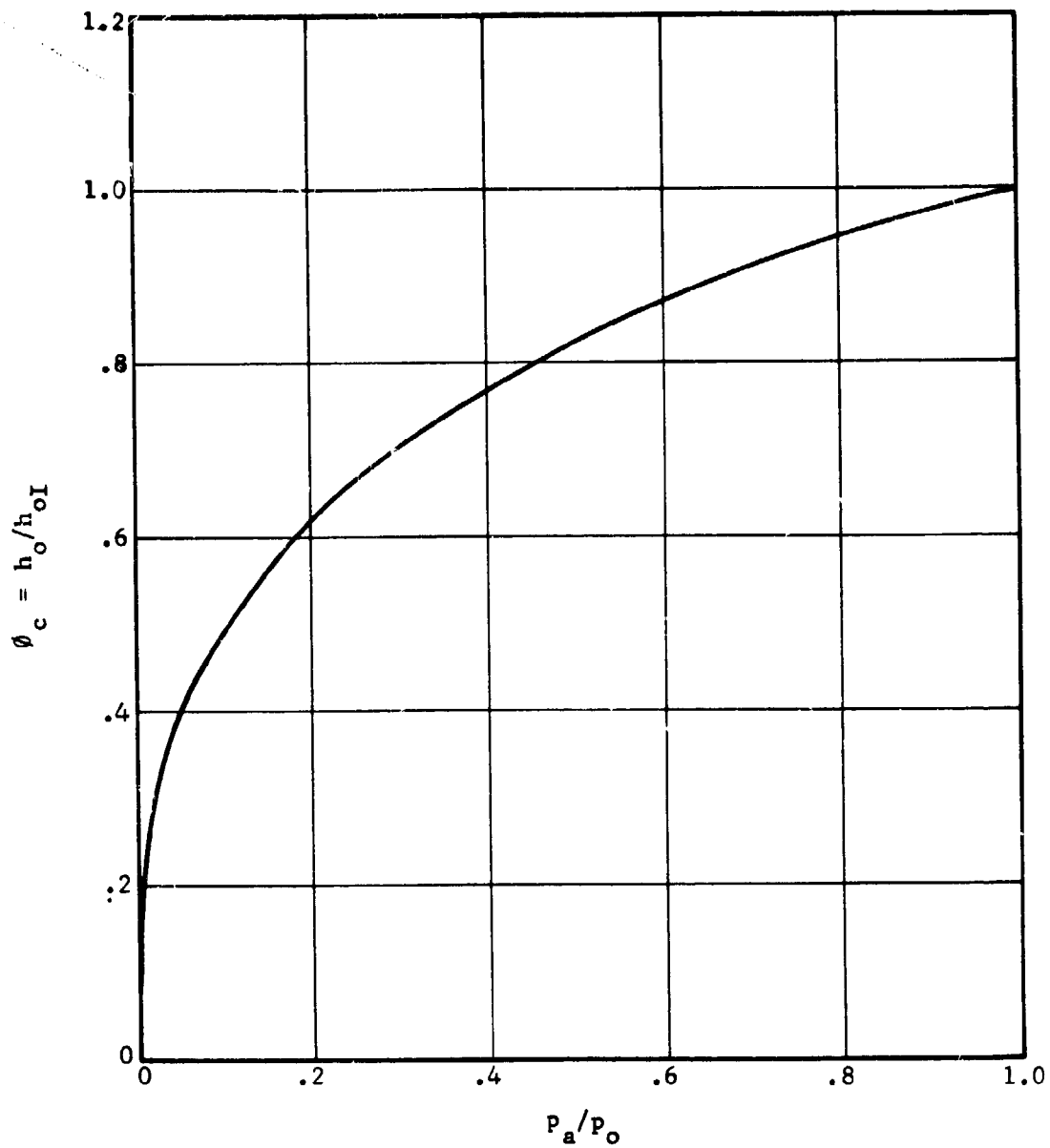


Fig. 25 Effect of Compressibility on Thrust Bearing Film Thickness

One may then calculate the mean velocity u_o by

$$u_o = R_m \Omega \quad (73)$$

where R_m is the mean radius and Ω the angular velocity and compute h_{OI} from Equation (71). The film thickness h_o is then determined from Figure 27. The torque may then be calculated from the equation

$$T^* = \frac{\mu u_o R_m A_p}{h_o} \quad (74)$$

which may be used to determine the power requirements. The above design procedure is straightforward and can be readily done by hand but has been programmed for rapid use. The computer program is given in Appendix II.

SECTION III

ANALYSIS OF THE CANTILEVERED, LEAF TYPE BEARING

The preceding sections dealt with the ridged journal and thrust bearing. Another type of bending dominated foil bearing that has been demonstrated successfully is the cantilevered leaf type bearing. This section deals with an analysis of both leaf journal and thrust bearings.

1. Elasticity Equations for Leaf Journal Bearing Assembly

A schematic of a cantilevered leaf journal bearing assembly is shown in Figure 26. Each leaf is preformed with a radius, R_L and the leaves are equally spaced at an angle θ_2 with respect to each other. For a given number of leaves and hence a given value of θ_2 , the degree of overlap will be dependent upon θ_1 .

The leaf radius will significantly affect the film thickness and must be considered when calculating the film profile. For purposes of calculating elastic deflection however, one may approximate the preformed leaf arcs with chords perpendicular to the lines between centers of the leaves and the housing as shown in Figure 26. The chord to arc approximations that are made here would be strictly valid either in the case of large leaf radii or a relatively large number of leaves. Such approximations greatly reduce the complexity of the analysis and will be used throughout. In terms of the chords then, leaf $i-1$ may be considered to contact leaf i at a distance ℓ_1 from the cantilever support. The leaf is then loaded with load W_1 per unit length at a distance ℓ_w from the support and rests on the following leaf at a distance ℓ from the cantilever support.

The angle θ_2 is related to the number of leaves by

$$\theta_2 = 2\pi/N \quad (76)$$

and the angle between the chord and the housing θ_1 is related to the fractional overlap (ℓ_1/ℓ) by

$$\tan \theta_1 = \frac{1 + \ell_1/\ell}{1 - \ell_1/\ell} \tan \left(\frac{\pi}{N} \right) \quad (77)$$

where the ratio of quantities ℓ_1 , ℓ_w and ℓ to the shaft radius R are expressed in terms of θ_1 and θ_2 as follows:

$$\frac{\ell_1}{R} = \frac{2 \sin (\theta_2/2)}{\sin \theta_2} \sin (\theta_1 - \theta_2/2) \quad (78)$$

$$\frac{\ell_w}{R} = \sin \theta_1 \quad (79)$$

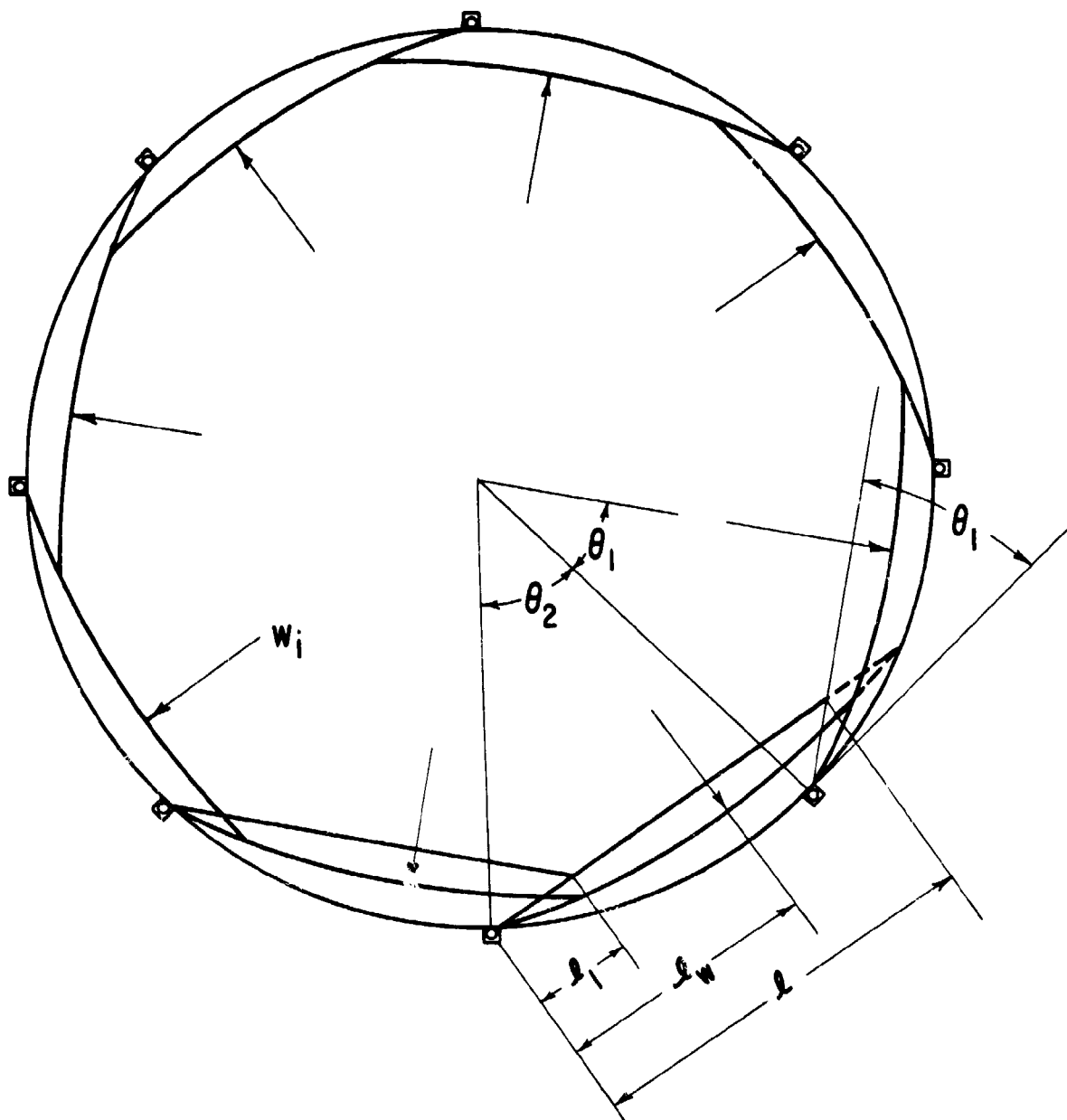


Fig. 26 Cantilevered Leaf Journal Bearing

$$\frac{\rho}{R} = \frac{2 \sin (\theta_2/2)}{\sin \theta_2} \sin (\theta_1 + \theta_2/2) \quad (80)$$

a. Deflection of Unwrapped Leaf

A schematic of the coordinate system and the forces under consideration in the analysis of the deflection of the i th cantilevered leaf is shown in Figure 27. The leaf is cantilevered at $x = 0$ and a shear force F_0 acts at the fixed end. At a distance ℓ_1 from the fixed end, a force F_1 is exerted on the leaf by the preceding leaf.

It is to be assumed for purposes of arriving at boundary conditions that the loading is periodic (that is to say that the deflections of each leaf are the same). This would strictly be true only if the load W_i was the same for each leaf. However, this periodicity condition will be applied under all circumstances and then W_i will be allowed to differ from leaf to leaf and will be determined from consideration of the over-all assembly.

The force F_1 exerted on the i th leaf by the end of leaf $i-1$ will, as a result of periodicity, be identical to the force exerted by the end of the i th leaf on leaf $i+1$; however, bending will only be influenced by the component of that force normal to the i th leaf which will have the value $F_1 \cos \theta_2$. We thus have the force $F_1 \cos \theta_2$ acting at the end of the i th leaf at $x = \ell$ as shown in Figure 27.

In a similar manner, let us take the deflection at $x = \ell_1$ to be w_1 . Since leaf $i-1$ rests upon leaf i , the deflection of the end of leaf $i-1$ normal to itself will be $w(\ell_1)/\cos \theta_2$ which must equal the deflection of the leaf at $x = \ell$, hence we have the condition

$$w(\ell_1) = w(\ell) \cos \theta_2 \quad (81)$$

The other boundary conditions governing the leaf deflections are the usual cantilever conditions at $x = 0$ which are

$$w = \frac{dw}{dx} = 0 \quad (82)$$

The elasticity equations may now be expressed in terms of the following dimensionless variables and parameters:

$$\tilde{F}_1 = F_1/W_i \quad (83)$$

$$\tilde{F}_0 = \frac{F_0}{W_i} \quad (84)$$

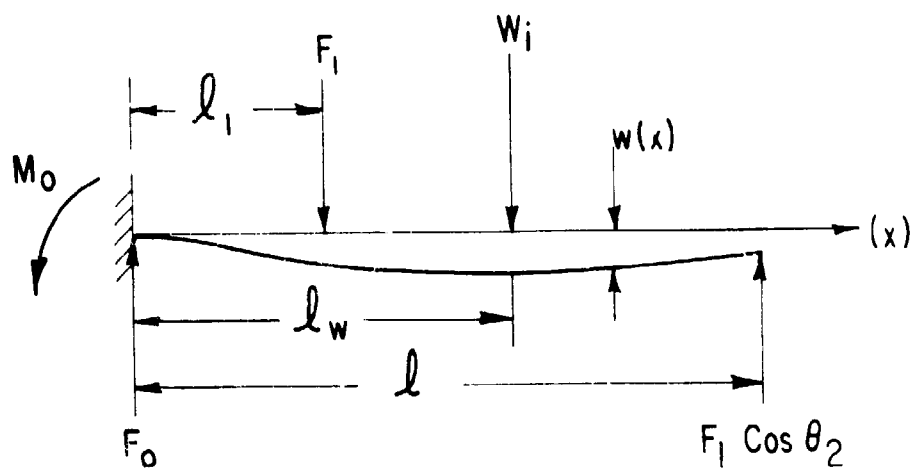


Fig. 27 Projected Forces on Chord of Leaf

$$\tilde{w} = \frac{D_L w}{\ell^3 w_i} \quad (85)$$

$$\xi = \frac{x}{\ell} \quad (86)$$

$$\tilde{M} = M/w_i \ell \quad (87)$$

$$\tilde{M}_O = M_O/w_i \ell \quad (88)$$

$$\tilde{\ell}_1 = \ell_1/\ell \quad (89)$$

$$\tilde{\ell}_w = \ell_w/\ell \quad (90)$$

The dimensionless beam equation thus becomes

$$\frac{d^2 \tilde{w}}{d\xi^2} = \tilde{M} \quad (91)$$

with boundary conditions

$$\tilde{w} = \frac{d\tilde{w}}{d\xi} = 0 \quad \text{at } x = 0 \quad (92)$$

$$\tilde{w}(\tilde{\ell}_1) = \tilde{w}(1) \cos \theta_2 \quad (93)$$

Taking moments about $x = 0$, we may relate $\tilde{M}_O$ to $\tilde{F}_1$ by

$$\tilde{M}_O = \tilde{F}_1 \tilde{\ell}_1 + \tilde{\ell}_w - \tilde{F}_1 \cos \theta_2 = \tilde{\ell}_w + \tilde{F}_1 (\tilde{\ell}_1 - \cos \theta_2) \quad (94)$$

A vertical force balance will relate $\tilde{F}_O$ to $\tilde{F}_1$ by

$$\tilde{F}_O = 1 + \tilde{F}_1 (1 - \cos \theta_2) \quad (95)$$

Taking moments resulting from forces to the left of any point ξ along leaf, one may write the dimensionless bending moment $\tilde{M}$ as

$$\tilde{M} = \begin{cases} \tilde{M}_0 - \tilde{F}_0 \xi & , \quad \text{if } 0 \leq \xi < \tilde{\ell}_1 \\ \tilde{M}_0 - \tilde{F}_0 \xi + \tilde{F}_1 (\xi - \tilde{\ell}_1) & , \quad \text{if } \tilde{\ell}_1 \leq \xi < \tilde{\ell}_w \\ \tilde{M}_0 - \tilde{F}_0 \xi + \tilde{F}_1 (\xi - \tilde{\ell}_1) + (\xi - \tilde{\ell}_w) & , \quad \text{if } \tilde{\ell}_w \leq \xi < 1 \end{cases} \quad (96)$$

The above equation may be integrated once subject to the conditions given in Equation (92) to obtain the relationship

$$\frac{d\tilde{w}}{d\xi} = \begin{cases} \tilde{M}_0 \xi - \frac{1}{2} \tilde{F}_0 \xi^2 & , \quad \text{if } 0 \leq \xi < \tilde{\ell}_1 \\ \tilde{M}_0 \xi - \frac{1}{2} \tilde{F}_0 \xi^2 + \frac{1}{2} \tilde{F}_1 (\xi - \tilde{\ell}_1)^2 & , \quad \text{if } \tilde{\ell}_1 \leq \xi < \tilde{\ell}_w \\ \tilde{M}_0 \xi - \frac{1}{2} \tilde{F}_0 \xi^2 + \frac{1}{2} \tilde{F}_1 (\xi - \tilde{\ell}_1)^2 + \frac{1}{2} (\xi - \tilde{\ell}_w)^2 & , \quad \text{if } \tilde{\ell}_w \leq \xi < 1 \end{cases} \quad (97)$$

which may be integrated again to obtain the dimensionless deflection

$$\tilde{w} = \begin{cases} \frac{1}{2} \tilde{M}_0 \xi^2 - \frac{1}{6} \tilde{F}_0 \xi^3 & , \quad \text{if } 0 \leq \xi < \tilde{\ell}_1 \\ \frac{1}{2} \tilde{M}_0 \xi^2 - \frac{1}{6} \tilde{F}_0 \xi^3 + \frac{1}{6} \tilde{F}_1 (\xi - \tilde{\ell}_1)^3 & , \quad \text{if } \tilde{\ell}_1 \leq \xi < \tilde{\ell}_w \\ \frac{1}{2} \tilde{M}_0 \xi^2 - \frac{1}{6} \tilde{F}_0 \xi^3 + \frac{1}{6} \tilde{F}_1 (\xi - \tilde{\ell}_1)^3 + \frac{1}{6} (\xi - \tilde{\ell}_w)^3 & , \quad \text{if } \tilde{\ell}_w \leq \xi < 1 \end{cases} \quad (98)$$

Equations (96) through (98) are all expressed in terms of $\tilde{M}_0$, $\tilde{F}_0$ and $\tilde{F}_1$. These may be combined with Equations (94) and (95) to eliminate $\tilde{M}_0$ and $\tilde{F}_0$ resulting in expressions for $\tilde{M}$, $d\tilde{w}/d\xi$ and $\tilde{w}$ all in terms of $\tilde{F}_1$. The final expression for $\tilde{F}_1$ is obtained by satisfying the boundary condition given by Equation (93). When the deflections at $\xi = \tilde{\ell}_1$ and $\xi = 1$ given by Equation (98) are substituted into Equation (93), the following expression for $\tilde{F}_1$ is obtained:

$$\tilde{F}_1 = \frac{\frac{1}{6} (1 - \tilde{\ell}_w)^3 \cos \theta_2 + \frac{1}{2} \tilde{\ell}_w (\cos \theta_2 - \tilde{\ell}_1^2) - \frac{1}{6} (\cos \theta_2 - \tilde{\ell}_1^3)}{\frac{1}{6} (\cos \theta_2 - \tilde{\ell}_1^3)(1 - \cos \theta_2) - \frac{1}{6} (1 - \tilde{\ell}_1)^3 \cos \theta_2 + \frac{1}{2} (\cos \theta_2 - \tilde{\ell}_1)(\cos \theta_2 - \tilde{\ell}_1^2)} \quad (99)$$

All of the information required to calculate the deflections of a

cantilever foil in line contact are thus provided.

Dimensionless deflection curves in the form of $\tilde{w}$ vs. ξ for various fractional overlaps are shown in Figures 28 through 30 for values of $N = 8, 12$ and 16 . The locations of $\tilde{\ell}_1$ and $\tilde{\ell}_w$ are marked on each curve. These curves may be used to estimate individual foil stiffness for line contact and may be used in conjunction with the load distribution and stiffness analyses discussed later to estimate the overall bearing.

The elasticity equations do not account for the effect of distributions in pressure resulting from lubrication and further assume that line contact will occur at the line between centers of the shaft and the leaf. When the curvature of the deformed leaf is equal to the curvature of the shaft, line contact will not occur but the leaf will wrap the shaft over a finite area. Incipient wrapping is predicted to occur when the curvature of the deformed leaf is exactly equal to the curvature of the shaft which may be expressed mathematically as

$$-\left. \frac{d^2 w}{dx^2} \right|_{x=\ell_w} = \frac{1}{R_{eff}} = \frac{1}{R} - \frac{1}{R_L} \quad (100)$$

The second derivative appearing in Equation (100) may be evaluated from Equation (96) at $\xi = \tilde{\ell}_w$ and the dimensionless expression for incipient wrap is given by

$$\tilde{M}_0 - \tilde{F}_0 \tilde{\ell}_w + \tilde{F}_1 (\tilde{\ell}_w - \tilde{\ell}_1) = -\frac{D_L}{W_i \ell R_{eff}} \quad (101)$$

In the following section, deflection equations for a wrapped leaf will be presented.

b. Elastic Deflection of Leaf Under Wrapped Conditions

Conditions for incipient wrapping are given by Equation (101). We may thus define a parameter β as

$$\beta = \frac{D_L}{W_i \ell R_{eff}} \quad (102)$$

and wrapping will occur when

$$R < -\left(\tilde{M}_0 - \tilde{F}_0 \tilde{\ell}_w + \tilde{F}_1 (\tilde{\ell}_w - \tilde{\ell}_1) \right) = (1 - \tilde{\ell}_w) \tilde{F}_1 \cos \theta_2 \quad (103)$$

where $\tilde{F}_1$ is calculated from Equation (99). When β is sufficiently

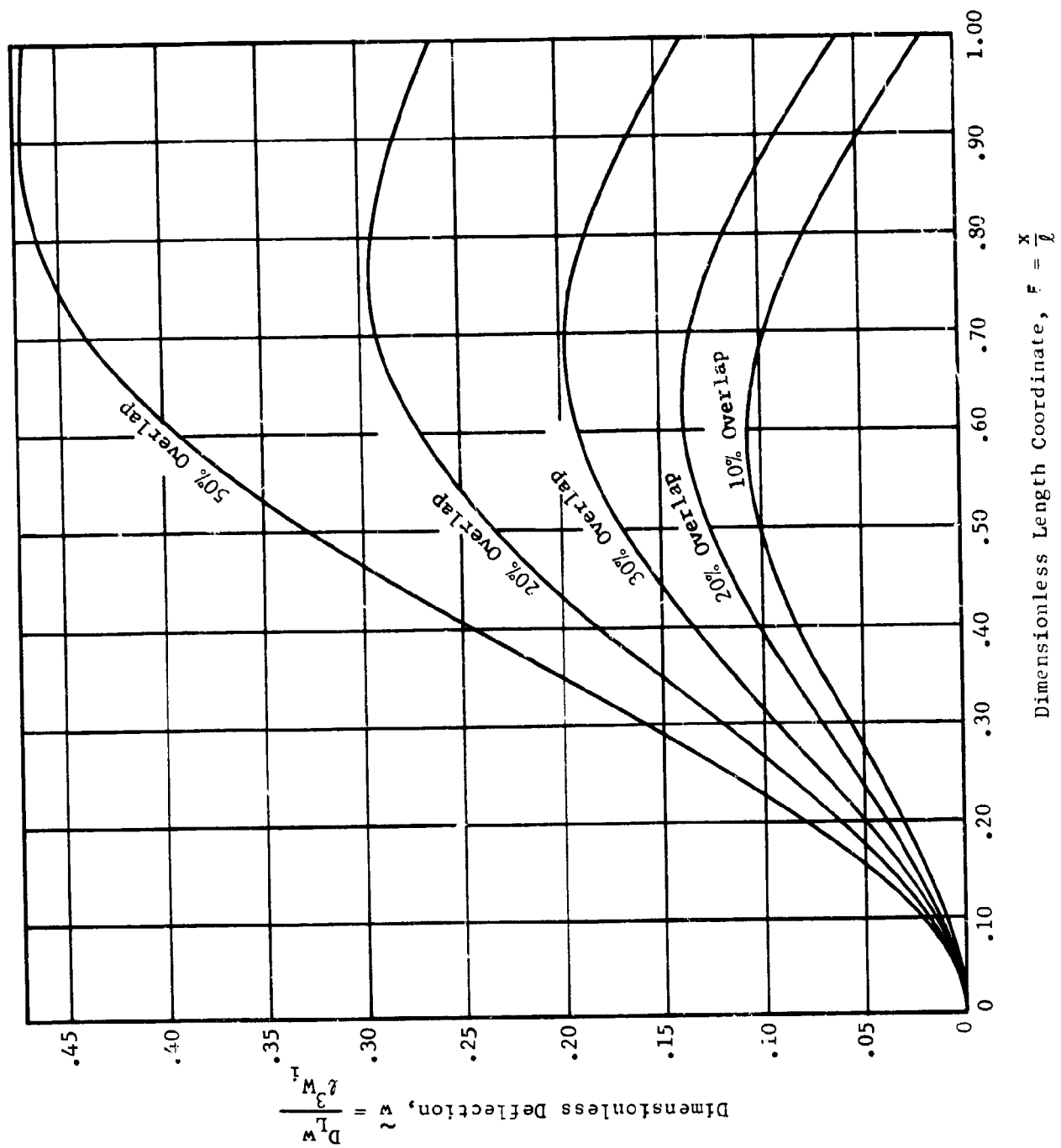


Fig. 28 Line Contact Leaf Deflection Curves, $N = 8$

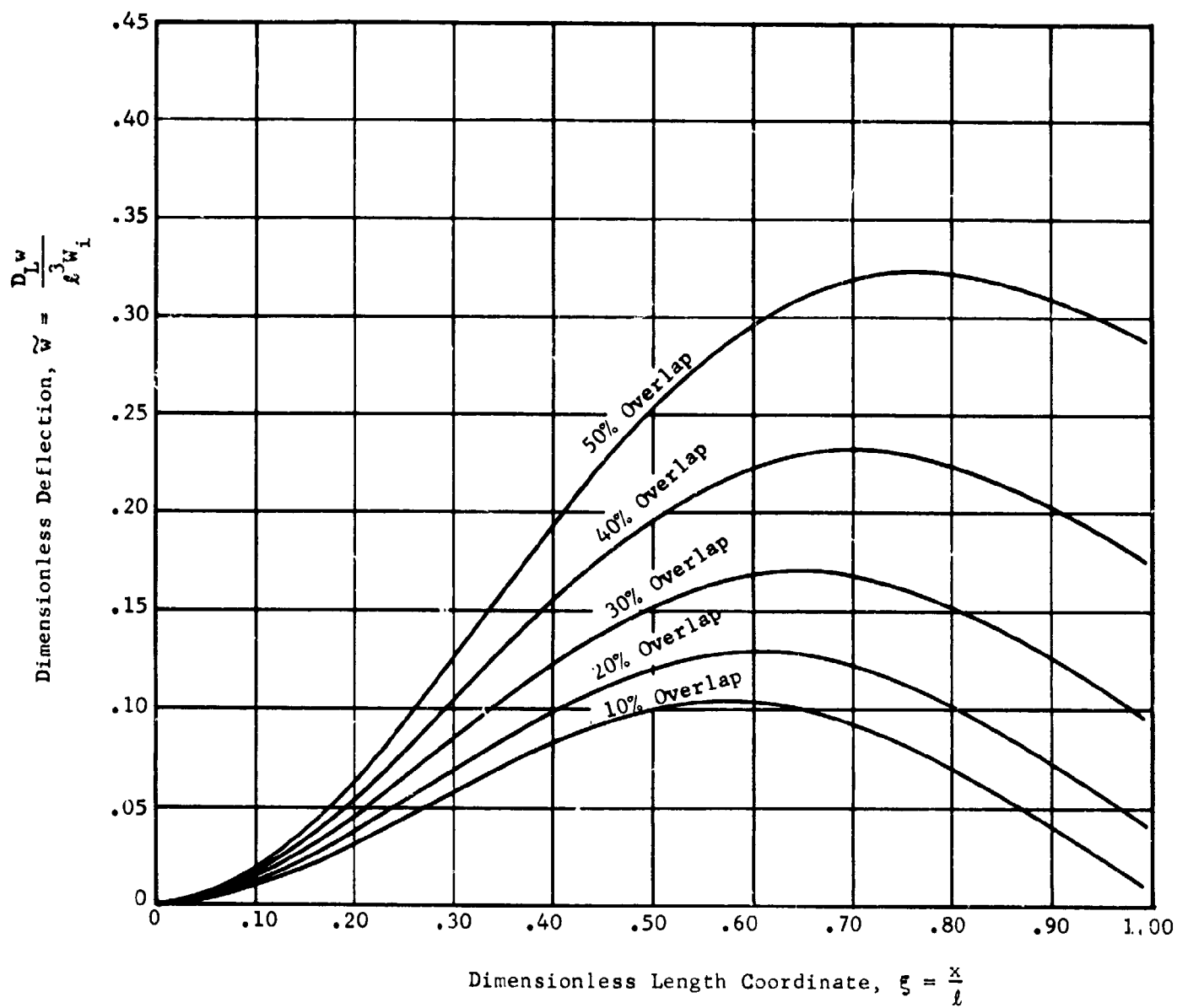


Fig. 29 Line Contact Leaf Deflection Curves, $N = 12$

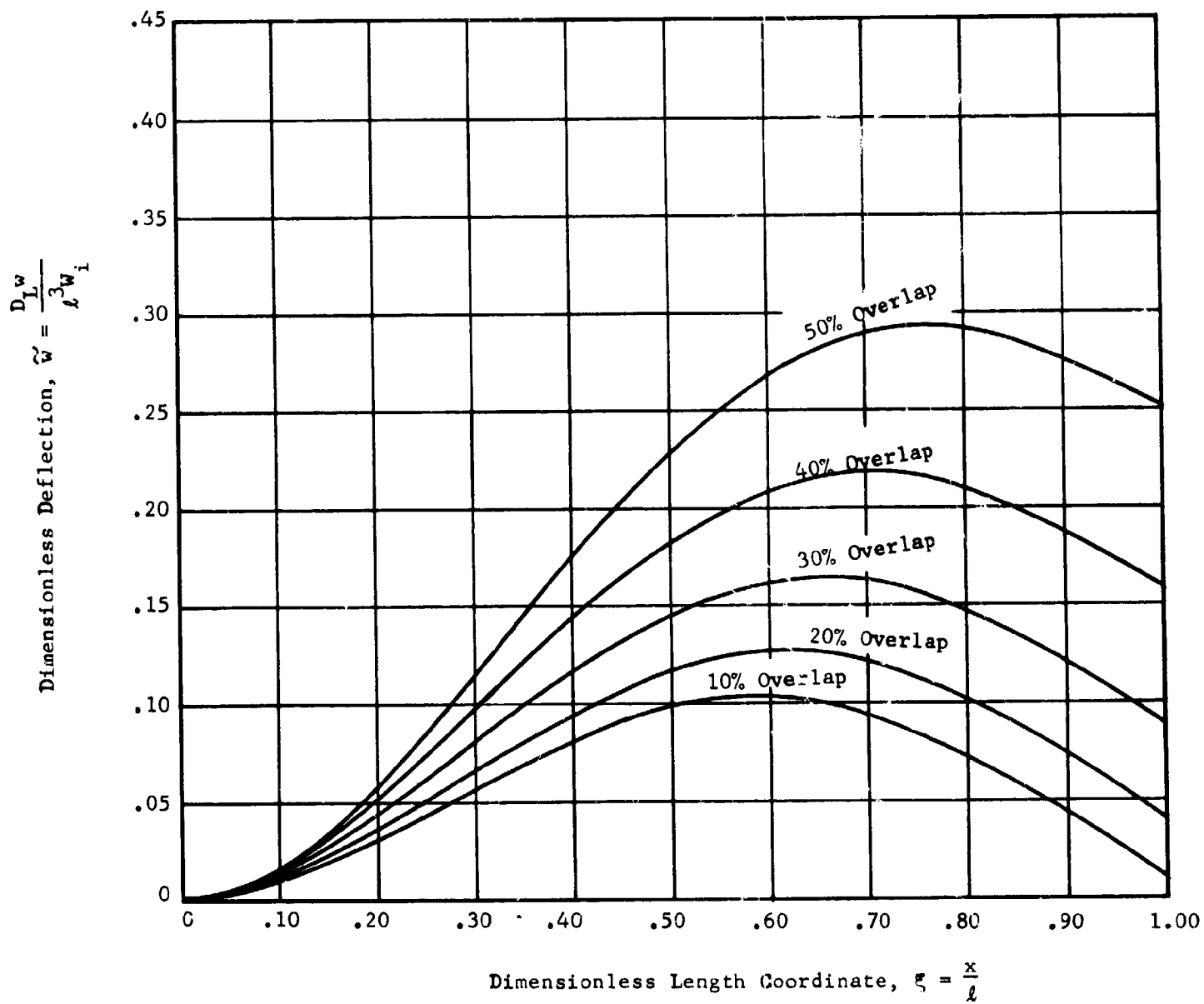


Fig. 30 Line Contact Leaf Deflection Curves, $N = 16$

small so that the above inequality is satisfied, the leaf and the shaft will be in contact over finite length between $x = l_b$ and $x = l_a$ as shown on Figure 31. These lengths do not necessarily coincide with the line between centers of the shaft and leaf located at l_w . There will be forces exerted at l_a and l_b of magnitude W_{ai} and W_{bi} respectively. Between these two points, the curvature of the leaf will be constant and, of course, equal to the curvature of the shaft. Since the curvature is constant, the bending moment will be constant and there will be no vertical shear force in the interval $l_a < x < l_b$.

One may write equations for the bending moment and deflections of the leaf in terms of the dimensionless forms of the forces and lengths shown in Figure 31 by using the dimensionless parameters defined previously along with the additional parameters given below:

$$\tilde{W}_{ai} = W_{ai}/W_i \quad (104)$$

$$\tilde{W}_{bi} = W_{bi}/W_i \quad (105)$$

$$\tilde{l}_a = l_a/l \quad (106)$$

$$\tilde{l}_b = l_b/l \quad (107)$$

The dimensionless bending moment $\tilde{M}$ is then given by

$$\tilde{M} = \begin{cases} \tilde{M}_0 - \tilde{F}_0 \xi & , \quad \text{if } 0 \leq \xi < \tilde{l}_1 \\ \tilde{M}_0 - \tilde{F}_0 \xi + \tilde{F}_1 (\xi - \tilde{l}_1) & , \quad \text{if } \tilde{l}_1 \leq \xi < \tilde{l}_a \\ -\beta & , \quad \text{if } \tilde{l}_a \leq \xi < \tilde{l}_b \\ -\beta + \tilde{W}_{bi} (\xi - \tilde{l}_b) & , \quad \text{if } \tilde{l}_b \leq \xi < 1 \end{cases} \quad (108)$$

which may be integrated once subject to the condition that $d\tilde{w}/d\xi = 0$ at $\xi = 0$ to obtain the relationship

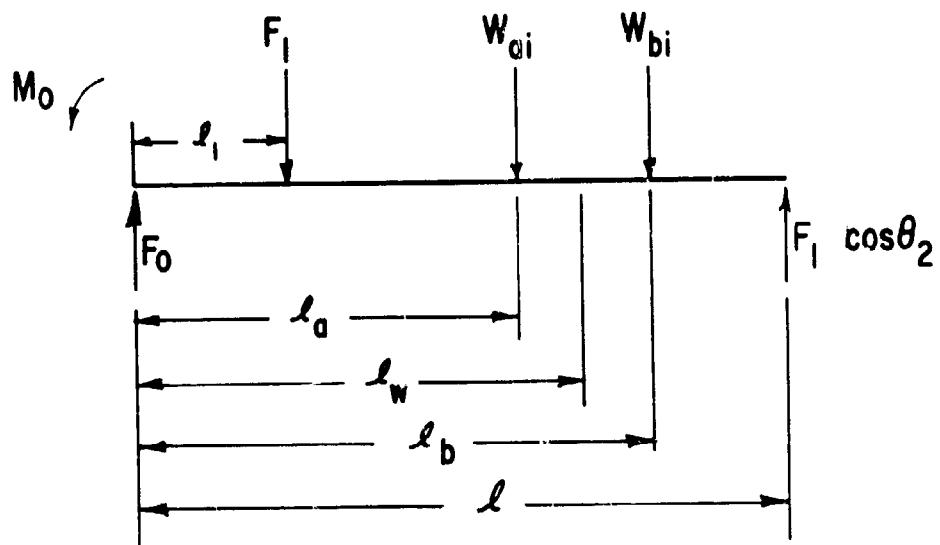


Fig. 31 Schematic of Projected Forces on Chord of Wrapped Leaf

$$\frac{d\tilde{w}}{d\xi} = \begin{cases} \tilde{M}_0 \xi - \frac{1}{2} \tilde{F}_0 \xi^2, & \text{if } 0 \leq \xi < \tilde{\ell}_1 \\ \tilde{M}_0 \xi - \frac{1}{2} \tilde{F}_0 \xi^2 + \frac{1}{2} \tilde{F}_1 (\xi - \tilde{\ell}_1)^2, & \text{if } \tilde{\ell}_1 \leq \xi < \tilde{\ell}_a \\ \tilde{M}_0 \tilde{\ell}_a - \frac{1}{2} \tilde{F}_0 \tilde{\ell}_a^2 + \frac{1}{2} \tilde{F}_1 (\tilde{\ell}_a - \tilde{\ell}_1)^2 - \beta (\xi - \tilde{\ell}_a), & \text{if } \tilde{\ell}_a \leq \xi < \tilde{\ell}_b \\ \tilde{M}_0 \tilde{\ell}_a - \frac{1}{2} \tilde{F}_0 \tilde{\ell}_a^2 + \frac{1}{2} \tilde{F}_1 (\tilde{\ell}_a - \tilde{\ell}_1)^2 - \beta (\xi - \tilde{\ell}_a) \\ \quad + \frac{1}{2} \tilde{W}_{bi} (\xi - \tilde{\ell}_b)^2, & \text{if } \tilde{\ell}_b \leq \xi < 1 \end{cases} \quad (109)$$

which, in turn, may be integrated to obtain the dimensionless deflection given below

$$\tilde{w} = \begin{cases} \frac{1}{2} \tilde{M}_0 \xi^2 - \frac{1}{6} \tilde{F}_0 \xi^3, & \text{if } 0 \leq \xi < \tilde{\ell}_1 \\ \frac{1}{2} \tilde{M}_0 \xi^2 - \frac{1}{6} \tilde{F}_0 \xi^3 + \frac{1}{6} \tilde{F}_1 (\xi - \tilde{\ell}_1)^3, & \text{if } \tilde{\ell}_1 \leq \xi < \tilde{\ell}_a \\ \frac{1}{2} \tilde{M}_0 \tilde{\ell}_a^2 - \frac{1}{6} \tilde{F}_0 \tilde{\ell}_a^3 + \frac{1}{6} \tilde{F}_1 (\tilde{\ell}_a - \tilde{\ell}_1)^3 \\ \quad + \left[\tilde{M}_0 \tilde{\ell}_a - \frac{1}{2} \tilde{F}_0 \tilde{\ell}_a^2 + \frac{1}{2} \tilde{F}_1 (\tilde{\ell}_a - \tilde{\ell}_1)^2 \right] (\xi - \tilde{\ell}_a) - \frac{1}{2} \beta (\xi - \tilde{\ell}_a)^2, & \text{if } \tilde{\ell}_a \leq \xi < \tilde{\ell}_b \\ \frac{1}{2} \tilde{M}_0 \tilde{\ell}_a^2 - \frac{1}{6} \tilde{F}_0 \tilde{\ell}_a^3 - \frac{1}{6} \tilde{F}_1 (\tilde{\ell}_a - \tilde{\ell}_1)^3 \\ \quad + \left[\tilde{M}_0 \tilde{\ell}_a - \frac{1}{2} \tilde{F}_0 \tilde{\ell}_a^2 + \frac{1}{2} \tilde{F}_1 (\tilde{\ell}_a - \tilde{\ell}_1)^2 \right] (\xi - \tilde{\ell}_a) - \frac{1}{2} \beta (\xi - \tilde{\ell}_a)^2 \\ \quad + \frac{1}{6} \tilde{W}_{bi} (\xi - \tilde{\ell}_b)^3, & \text{if } \tilde{\ell}_b \leq \xi < 1 \end{cases} \quad (110)$$

which satisfies the condition that $\tilde{w} = 0$ at $\xi = 0$.

The determination of the dimensionless forces corresponding to those shown in Figure 31, which, with the exception of $\tilde{W}_{ai}$, are required to calculate deflections from Equation (110) involve seven unknowns. These are, $\tilde{M}_0$, $\tilde{F}_0$, $\tilde{F}_1$, $\tilde{W}_{ai}$, $\tilde{W}_{bi}$, $\tilde{\ell}_a$ and $\tilde{\ell}_b$. The seven equations required to determine these constants are obtained from considerations given below.

Continuity of the bending moment given by Equation (108) at $\xi = \tilde{\ell}_a$

provides the relationship

$$\tilde{M}_0 - \tilde{F}_0 \tilde{x}_a + \tilde{F}_1 (\tilde{x}_a - \tilde{x}_1) = -\beta \quad (111)$$

Since there is no vertical shear in the wrapped region, one may sum forces to the left of that to obtain

$$\tilde{W}_{ai} + \tilde{F}_1 = \tilde{F}_0 \quad (112)$$

and sum forces to the right of the wrapped region to obtain

$$\tilde{W}_{bi} = \tilde{F}_1 \cos \theta_2 \quad (113)$$

Since the sum of the two forces at the edges of the wrapped region will equal the force supplied by the shaft to the leaf, one may write

$$\tilde{W}_{ai} + \tilde{W}_{bi} = 1 \quad (114)$$

The dimensionless bending moment in the region of wrap is $-\beta$, hence one may take moments about $\xi = 1$ to obtain the relationship

$$\beta = \tilde{W}_{bi} (1 - \tilde{x}_b) \quad (115)$$

The dimensionless curvature of the shaft will be equal to β , hence the dimensionless slope of the shaft at the start of wrap ($\xi = \tilde{x}_a$) will have the magnitude $\beta (\tilde{x}_w - \tilde{x}_a)$. It should be noted here that we refer to the curvature of the shaft relative to the curvature of the preformed leaf. The slope of the deflection curve as given by Equation (109) must match the slope of the shaft relative to that of the preformed leaf, hence we have the relationship

$$\tilde{M}_0 \tilde{x}_a - \frac{1}{2} \tilde{F}_0 \tilde{x}_a^2 + \frac{1}{2} \tilde{F}_1 (\tilde{x}_a - \tilde{x}_1^2) = \beta (\tilde{x}_w - \tilde{x}_a) \quad (116)$$

Finally, the remaining equation needed to determine the seven constants is the boundary condition given by Equation (93).

The quantities $\tilde{W}_{bi}$, $\tilde{W}_{ai}$, $\tilde{F}_1$, $\tilde{F}_0$, and $\tilde{x}_a$ may all be expressed in terms of $\tilde{x}_b$ as follows:

$$\tilde{W}_{bi} = \frac{\beta}{1 - \tilde{\ell}_b} \quad (117)$$

$$\tilde{W}_{ai} = 1 - \frac{\beta}{1 - \tilde{\ell}_b} \quad (118)$$

$$\tilde{F}_1 = \frac{\beta}{(1 - \tilde{\ell}_b) \cos \theta_2} \quad (119)$$

$$\tilde{F}_o = \frac{\beta}{1 - \tilde{\ell}_b} \left(\frac{1}{\cos \theta_2} - 1 \right) + 1 \quad (120)$$

$$\tilde{\ell}_a = \left\{ \frac{\beta}{(1 - \tilde{\ell}_b - \beta)} \left[2 (1 - \tilde{\ell}_b) \tilde{\ell}_w - \frac{\tilde{\ell}_1^2}{\cos \theta_2} \right] \right\}^{1/2} \quad (121)$$

The above expressions for $\tilde{F}_o$, $\tilde{F}_1$ and $\tilde{\ell}_a$ may be substituted in Equation (111) to determine $\tilde{M}_o$ in terms of $\tilde{\ell}_b$. With the use of the above expressions, the deflection equation (Equation (110)) may be expressed in terms of $\tilde{\ell}_b$. The substitution of this equation for $\tilde{w}$ into the boundary condition given by Equation (93) will result in one equation with one unknown, $\tilde{\ell}_b$. This equation has been obtained and is solved numerically by a nesting procedure.

A comparison of deflection curves for a leaf in line contact with a wrapped leaf is shown in Figure 32. A value of $\beta = .0789$ has been used which is a factor of 2 less than that required for incipient wrap. The foil is shown to wrap the shaft somewhat to the left of the center of loading, ℓ_w . The deflection of the foil can be seen to be less when wrapping occurs, hence inclusion of the effect of wrapping will result in prediction of higher stiffnesses than those predicted under conditions of line contact.

The above considerations thus provide all the necessary information for determining the deflection characteristics of a single leaf in either a wrapped or unwrapped mode in terms of the load on the leaf. The next section will deal with determining the distribution of loads on the leaves in terms of the total load applied to the journal bearing.

c. Load Distribution and Stiffness in a Cantilevered Leaf Journal Bearing

In the preceding section, elastic deflections have been determined in terms of the load on an individual leaf, W_i . Let us assume now that the applied load W acts along the line between centers of the shaft and the N th leaf and that the shaft has displaced an amount e . If the pre-load is δ_o , then the deflection of the i th leaf at ℓ_w is given by

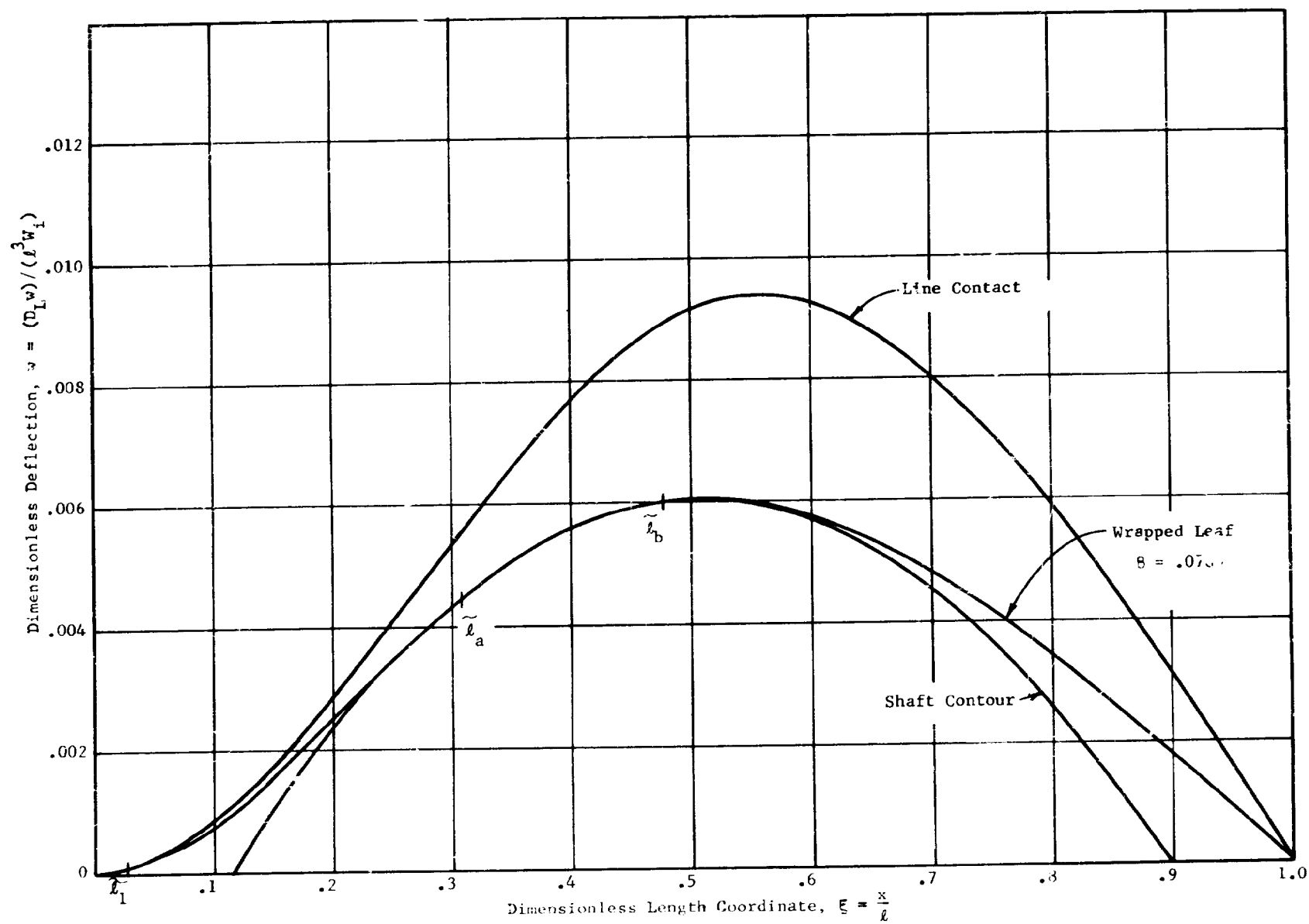


Fig. 32 Comparison of Deformed Leaf in Wrapped and Line Contact Modes, $\theta_1 = 23^\circ$ $\theta_2 = 45^\circ$

$$w_i = \begin{cases} \delta_o + e \cos i \theta_2 & , \quad \text{if } \delta_o + e \cos i \theta_2 > 0 \\ 0 & , \quad \text{if } \delta_o + e \cos i \theta_2 < 0 \end{cases} \quad (122)$$

$i = 1, 2, \dots, N$

The load on the i th leaf is related to the deflection given above through the stiffness K_i by the relationship

$$W_i = K_i w_i \quad (123)$$

The stiffness K_i is related to the dimensionless deflection $\tilde{w}$ by

$$K_i = \frac{\ell^3}{D_L \tilde{w}(\tilde{\ell}_w)} \quad (124)$$

where $\tilde{w}$ is calculated from Equation (98) for the unwrapped case and by the portion of Equation (110) valid in the interval $\tilde{\ell}_a \leq \tilde{\ell} < \tilde{\ell}_b$ for the wrapped case. The total load W may then be calculated from the vector sum of the individual loads by

$$W = \sum_{i=1}^N w_i \cos i \theta_2 \quad (125)$$

In the case where no wrapping occurs, the stiffnesses K_i will be independent of the load and one may calculate them immediately from Equation (98) and then calculate the deflections w_i from Equation (123) and the total load from Equation (125) directly. In the case where wrapping occurs in some of the leaves, the individual leaf stiffnesses K_i will be dependent upon the loads since the parameter $\mathcal{B}$ is load dependent.

An iterative procedure outlined below has been used for determining the load distribution when wrapping is indicated to occur.

1. Calculate $\tilde{w}(\tilde{\ell}_w)$ from Equation (98) (on Figures 28-30) which neglects wrapping.
2. Calculate K_i from Equation (124), and W_i from Equation (123).
3. Calculate $\mathcal{B}$ for each leaf and check Equation (103) for wrapping.
4. For cases where wrapping is indicated, re-evaluate $\tilde{w}(\tilde{\ell}_w)$ from Equation (110)
5. Go back to step 2 and carry out the iteration until the total

load W as calculated from Equation (125) converges to the desired tolerance.

The total stiffness of the bearing K may then be calculated from the relationship

$$K = W/e \quad (126)$$

2. Film Thickness and Power Estimates in a Leaf Journal Bearing

In the preceding section, equations were presented for computing the load distribution and static deflection profiles of a leaf journal bearing assembly. In this section, the formulas will be presented for determination of the minimum film thickness and power requirements of a leaf journal bearing assembly. These formulas will be approximations to provide reasonable estimates of these quantities.

We will start out by assuming that the shape of the film thickness profile in the region where the film thickness is formed is the same as that of the static profile but displaced by an amount h_o . The static deflection profile $g(x)$ is given below

$$g(x) = \frac{(x - l_w)^2}{2 R_{eff}} + w(x) - w(l_w) \quad (127)$$

In the above equation, the first term on the right denotes the undeformed profile of the leaf relative to the shaft. The second term is the elastic deflection of the leaf and the third term is the displacement of the shaft.

We will start out by assuming incompressible flow and then apply a compressibility correction factor similar to that used in the ridged thrust bearing analysis. The Reynold's equation for steady incompressible flow is given by

$$\frac{dp}{dx} = 6\mu u_o \frac{h - k}{h^3} \quad (128)$$

where the film thickness h is in accordance with the assumptions stated above and given as

$$h = h_{oI} + g(x) \quad (129)$$

where h_{oI} denotes the incompressible minimum film thickness and constant k appearing in Equation (128) is a constant of integration to be determined.

Far upstream from the center of contact the pressure will become ambient. As in the case of the ridged thrust bearing, the film thickness will become sufficiently

large upstream that one may use the condition

$$p - p_a = 0 \quad \text{as } x \rightarrow -\infty \quad (130)$$

If we make the conservative assumption neglecting pressures developed in the divergent portion of the film, then for the case of line contact we have the downstream condition given by

$$p = p_a \quad \text{at } x = l_w.$$

In the case where wrapping occurs, under static conditions there will be no pressure in the wrapped region since the curvature and hence the bending moment must be constant there. Any hydrodynamic pressure developed there will then cause pocketing to occur. A schematic of line contact, wrap and pocketing are shown in Figure 33. Under the conditions of wrapping, a very small hydrodynamic pressure occurring in the wrapped region can cause a large pocket (large in terms of the film thickness) to develop.

Under these conditions, minimum film thicknesses would occur in two places, one at the inlet to the wrap region at $x = l_a$ and one at the exit from the pocketed region at $x = l_b$. The film thickness at l_a will, in general, be slightly smaller than that at l_b in that W_{ai} is generally larger than W_{bi} . Since the film will start to diverge at l_a , we will use the boundary condition

$$p = p_a \quad \text{at } x = l_a$$

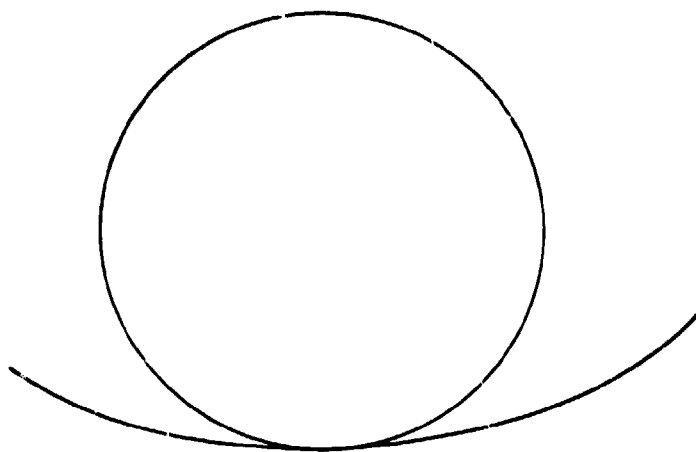
and depict the load to be W_{ai} in determining h_{OI} at $x = l_{ai}$. We may summarize the downstream boundary conditions for either the wrapped or line contact modes by the following relation:

$$p = p_a \quad \text{at } x = l^* \quad (131)$$

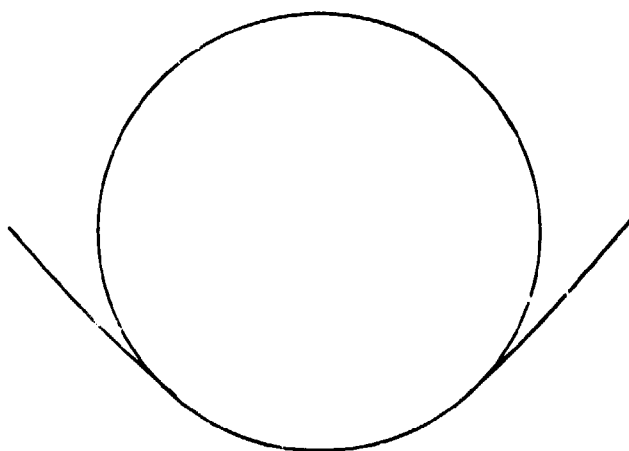
where

$$l^* = \begin{cases} l_w, & \text{for line contact} \\ l_a, & \text{for wrapping} \end{cases} \quad (132)$$

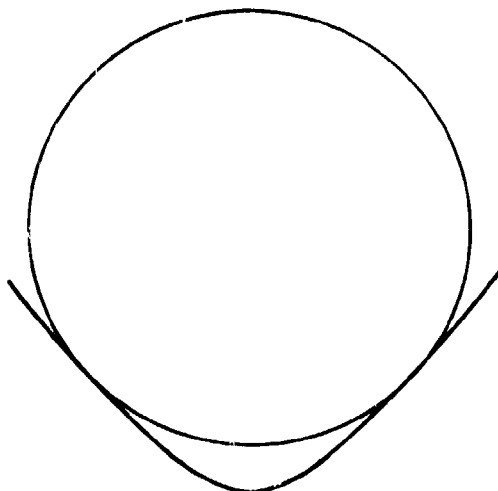
Similarly, the relevant load on the i th leaf used for determining h_{OI} may be summarized as



a) Line Contact



b) Wrap



c) Pocketing

Fig. 33 Contact Modes for Leaf Journal Bearing

$$W_i^* = \begin{cases} W_i & , \text{ for line contact} \\ W_{ai} & , \text{ for wrapping} \end{cases} \quad (133)$$

Equation (128) may be integrated subject to the boundary conditions as given by Equations (130) through (132) to give the following expression

$$p - p_a = I_{02}(x) - \frac{I_{02}(\ell^*)}{I_{03}(\ell^*)} I_{03}(x) \quad (134)$$

where the integrals I_{jk} are defined as follows:

$$I_{jk}(x) = \int_{-\infty}^x \frac{x'^m}{h^n(x')} dx' \quad (135)$$

The load on the i th leaf can be calculated by integrating Equation (134) over the interval $-\infty < x < \ell^*$ to obtain the relationship

$$W_i^* = \int_{-\infty}^{\ell^*} (p - p_a) dx = \frac{I_{02}(\ell^*)}{I_{03}(\ell^*)} I_{13}(\ell^*) - I_{12}(\ell^*) \quad (136)$$

which relates the minimum film thickness, h_{0I} , to the load for incompressible flow. The geometry of the flow channel under consideration as seen from Equation (127) may be expressed as the sum of a second order polynomial for the rigid profile and a third order polynomial for the elastic distortion. A compressibility factor has already been worked out for a quadratic inlet region similar to that presently under consideration for the ridged thrust bearing (Figure 25). One may obtain an estimate of the effect of compressibility on film thickness by assuming a value of p_0 defined as follows:

$$p_0 = p_a + (p_{\max} - p_a)/2 \quad (137)$$

which takes p_0 to be the ambient pressure plus half the maximum pressure rise. The maximum pressure rise will occur at the value of x where the pressure gradient is equal to 0, which from Equations (128) and (134) is given by the solution to

$$h(x_{\max}) = \frac{I_{02}(\ell^*)}{I_{03}(\ell^*)} \quad (138)$$

Having calculated $x_{\max}$, one can then substitute it into Equation (134) to obtain the maximum pressure rise relative to ambient which is given by

$$P_{\max} - P_a = I_{02}(x_{\max}) - \frac{I_{02}(\ell^*)}{I_{03}(\ell^*)} I_{03}(x_{\max}) \quad (139)$$

The film thickness h_o may be obtained with the use of the correction factor ϕ_c given in Figure 25

$$h_o = h_{oI} \phi_c (p_a/p_o) \quad . \quad (\text{See Figure 25}) \quad (140)$$

The minimum film thickness in the system is obtained with a load per unit length of W_N^* (for the leaf directly under the load).

An estimate of the power requirement can be obtained simply by using the minimum value of h_o obtained from Equation (140) together with the use of the rigid quadratic film thickness shape contour corresponding to the first term on the right in Equation (127). If one integrates the reciprocal of the film thickness obtained in this manner over the entire leaf and then multiplies it by the number of leaves and the appropriate scale factors, one obtains the relationship for the power given by

$$P = \mu u_o^2 N \pi \sqrt{\frac{2 R_{\text{eff}}}{h_o}} \quad (141)$$

All of the above equations have been programmed so that dimensional input can be used to determine stiffness, film thickness and power requirements. (See Appendix II).

3. Design Data For Cantilevered, Leaf-Type Journal Bearings

Computations have been performed about a central case corresponding to a 1.5" diameter x 1.6" long bearing with eight (8) equally spaced foils or leaves. The input parameters for the central case are shown in Table I. All parameters on the graphs correspond to the data given in Table I unless otherwise specified.

The effect of bearing speed on the film thickness-load relationship is shown in Figure 34. These relationships are characteristic of hydrodynamic bearing performance, although the film thicknesses are somewhat lower than those predicted to occur in ridged bearings as a result of the higher degree of conformity associated with the ridged bearing. The conformity is somewhat limited in the leaf bearing by the fact that it would be undesirable to have leaf edges in the contact zone. The breaks in the curves are a result of the onset of wrapping. The pocketing assumption used in the analyses will be substantially in error until significant wrapping has occurred, hence the breaks in the curves will not be real and should be ignored.

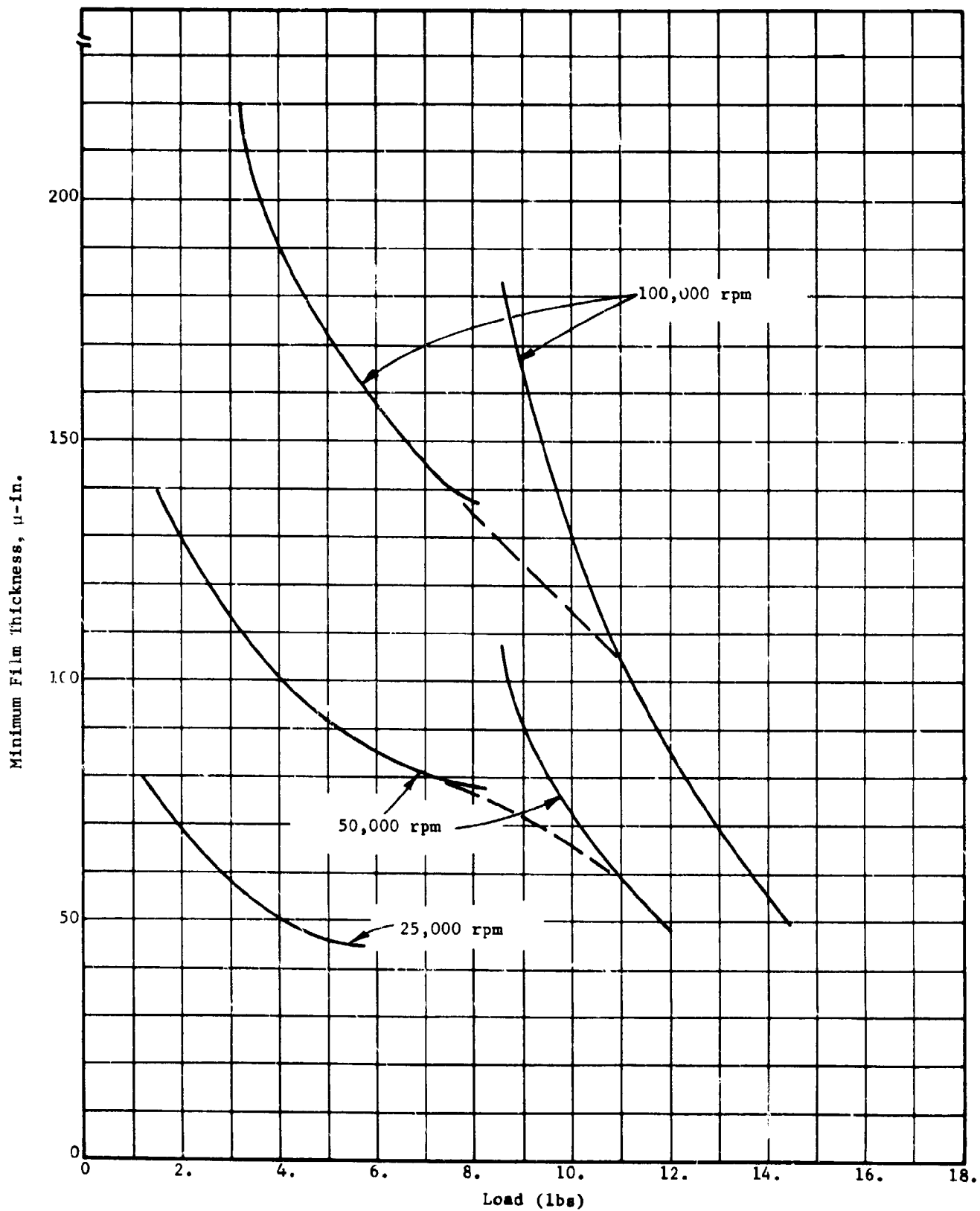


Fig. 34 Effect of Speed on Film Thickness for Leaf Journal Bearing

TABLE I
INPUT VALUES FOR PARAMETERS USED IN CENTRAL CASE

<u>Parameters</u>	<u>Central Case Values</u>	<u>Range Covered</u>
Length (inches)	1.6	1.1 to 1.6
Diameter (inches)	1.5	1.0 to 1.5
Number of Leaves	8	
Overlap (percent)	17	17 to 42
Leaf Radius (inches)	0.85	0.85 to 0.9
Leaf Thickness (inches)	0.010	0.010 to 0.015
Ambient Pressure (psia)	14.7	14.7 to 50
Speed (rpm)	50,000	25,000 to 100,000
Viscosity (reyns)	2.9×10^{-9}	2 to 5.5
Preload (inches)	0.001	0 to 0.002

The effect of viscosity on film thickness between the temperature range of -65°F and 1000°F is shown in Figure 35. The effect of viscosity on film thickness will be the same as the effect of a proportionate variation in speed.

The effect of ambient pressure as shown in Figure 36 is not overly large between atmospheric pressure and 50 psi.

The effect of preload on film thickness is shown in Figure 37. The preload has the effect of providing a more constant film-thickness versus load relationships. For a given preload in mils, the load per leaf will be higher and the film thickness correspondingly smaller with thicker leaves, as seen in Figure 38.

The effect of leaf radius on film thickness is quite strong as shown in Figure 39. A small change in leaf radius from 0.85 to 0.9 inches results in a very significant change in load-film relationship.

The effect of overlap is shown in Figure 40 where the overlap is increased from 17 percent to 42 percent. With the larger value of overlap, the conformity is better, hence, the film thickness is somewhat higher. However, as the extent of overlap is increased, consideration has to be given to the possibility of contact between the shaft and the edge of the preceding foil.

The effect of bearing size on load capacity is shown in Figure 41 for conditions of zero preload. A 1 inch diameter bearing 1.1 inches long is compared with the 1.5 inch diameter bearing 1.6 inches long used in the central case.

The variation in predicted power with speed is shown in Figure 42. These power

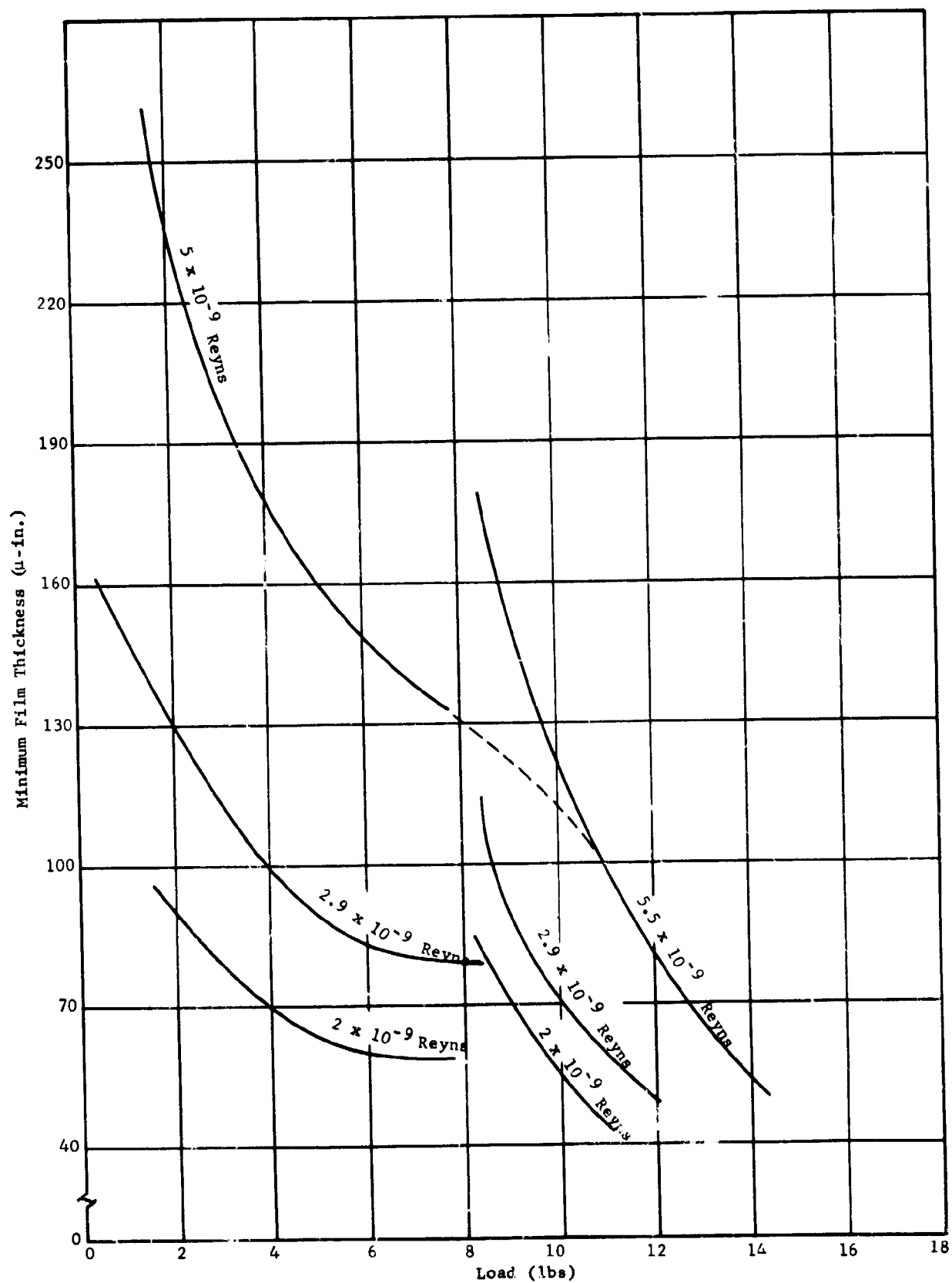


Fig. 35 Effect of Viscosity on Film Thickness for Leaf Journal Bearing

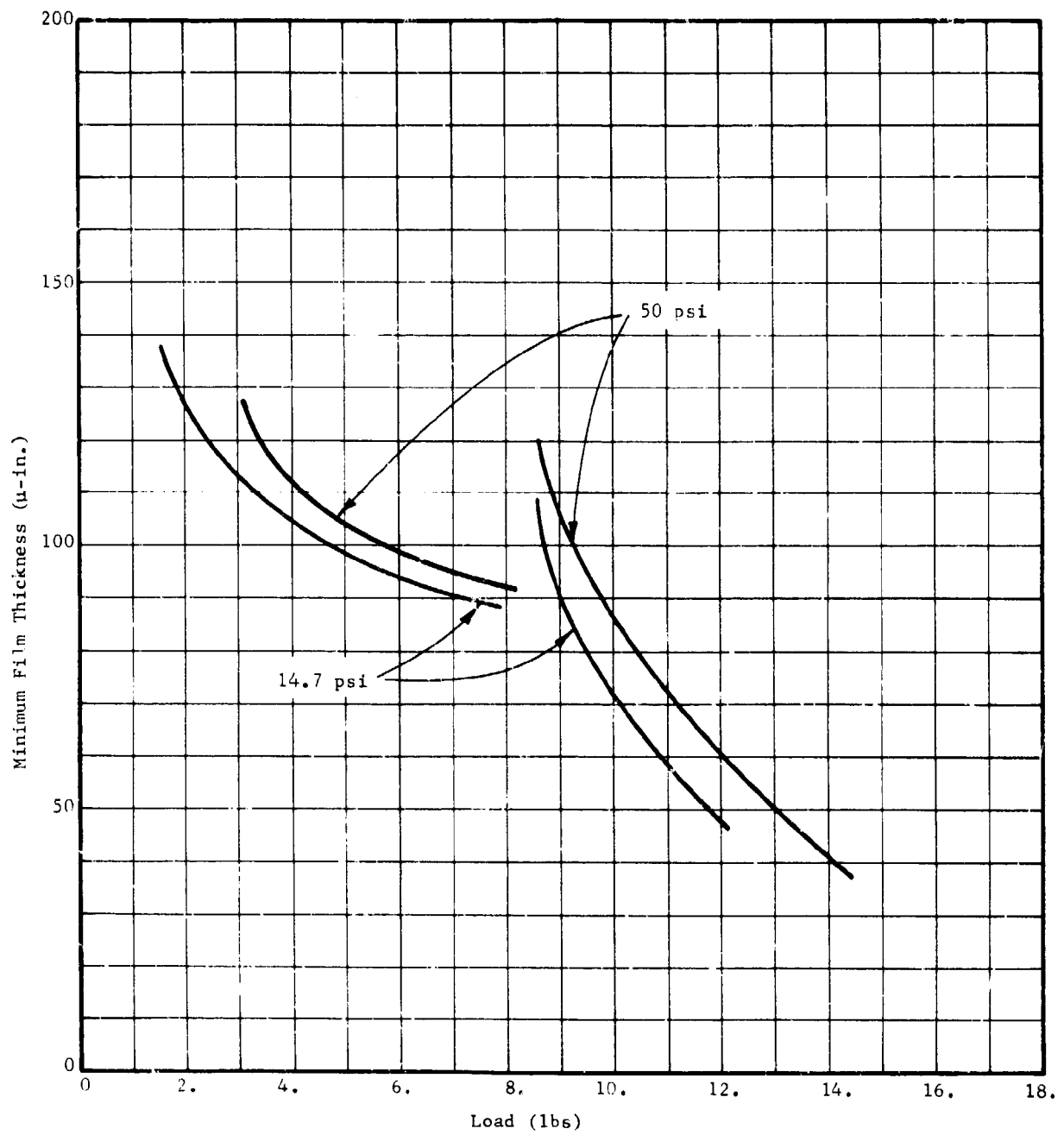


Fig. 36 Effect of Ambient Pressure on Film Thickness for Leaf Journal Bearing

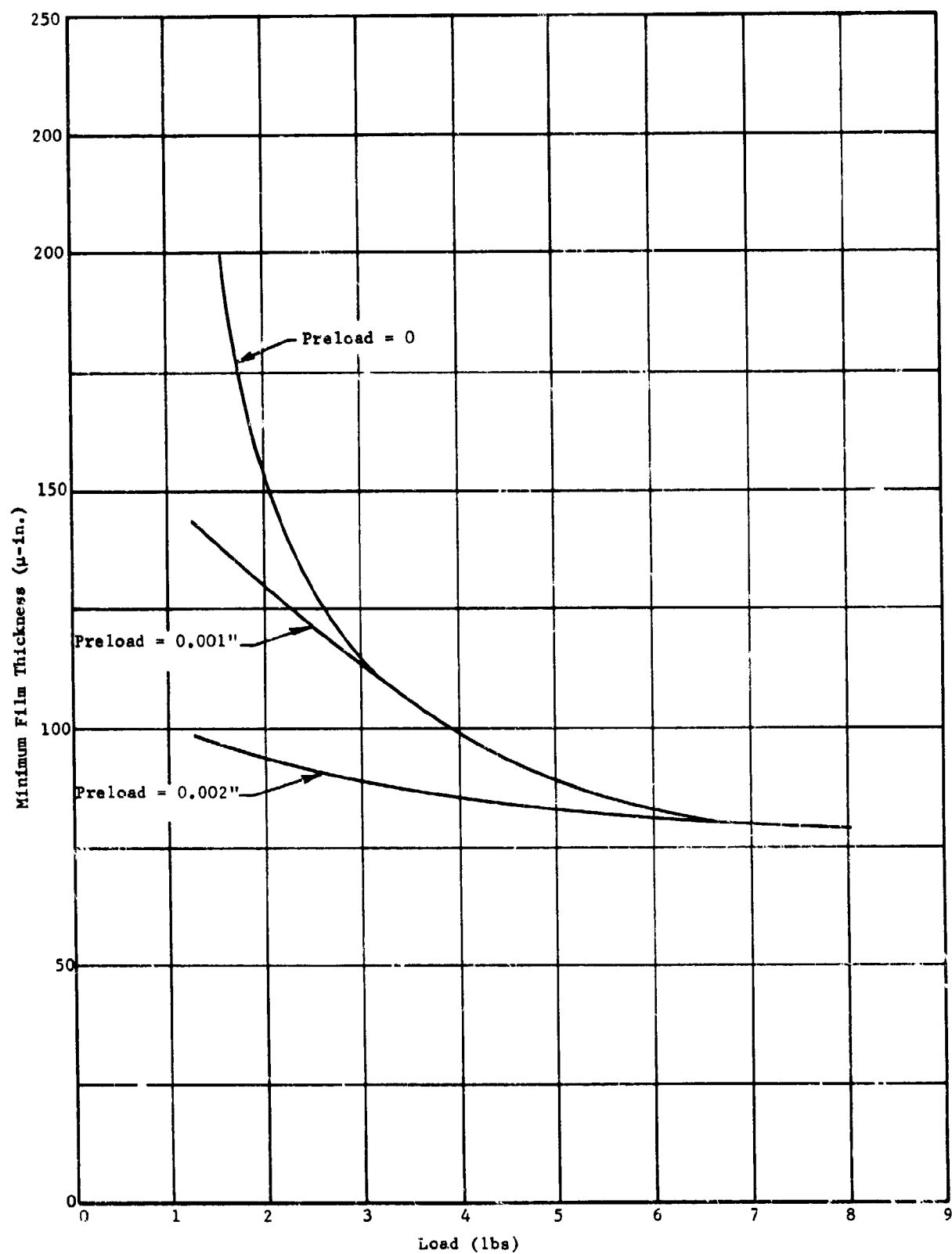


Fig. 37 Effect of Preload on Film Thickness for Leaf Journal Bearing

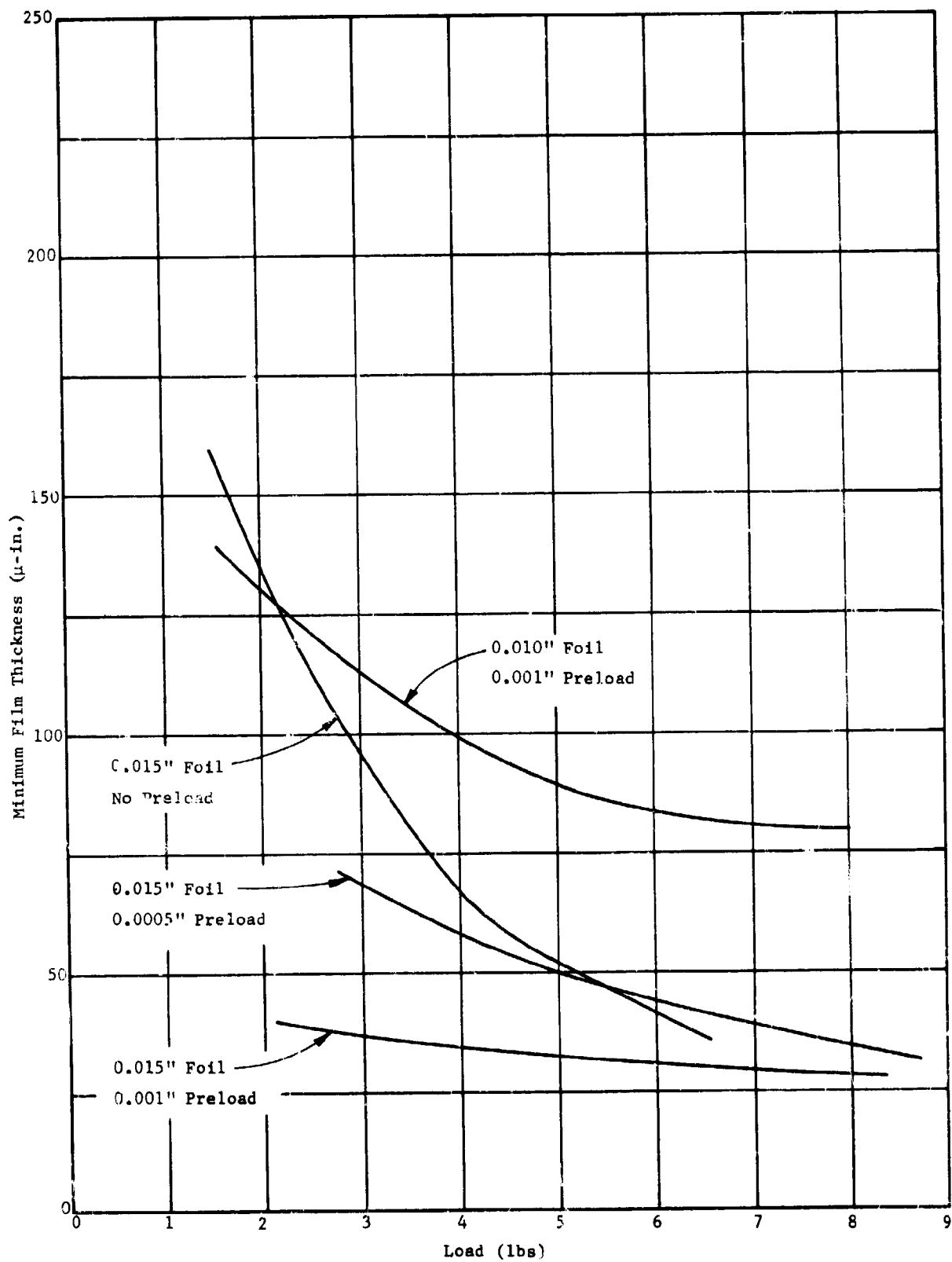


Fig. 38 Effect of Leaf Thickness on Film Thickness for Leaf Journal Bearing

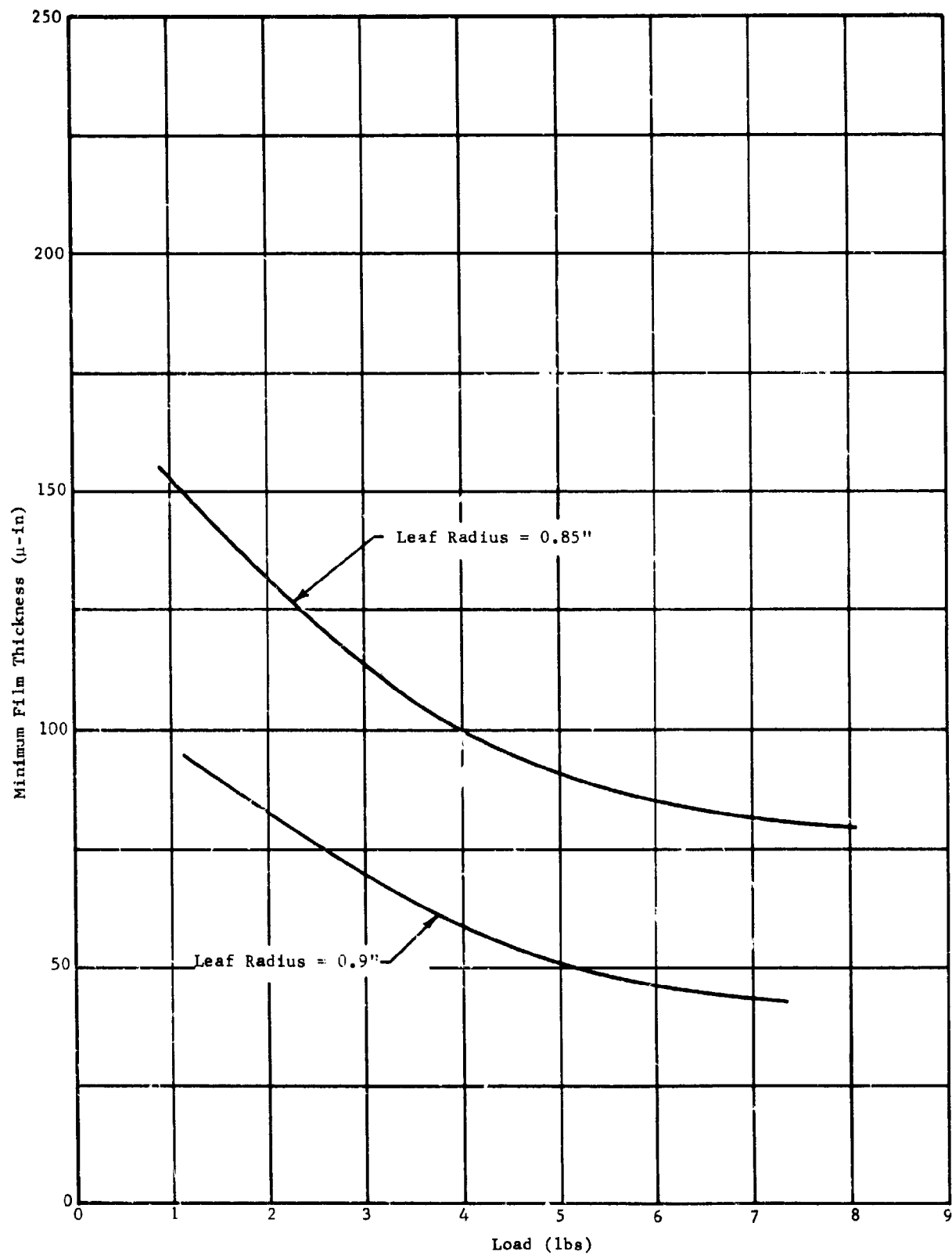


Fig. 39 The Effect of Leaf Radius on Film Thickness for Leaf Journal Bearing

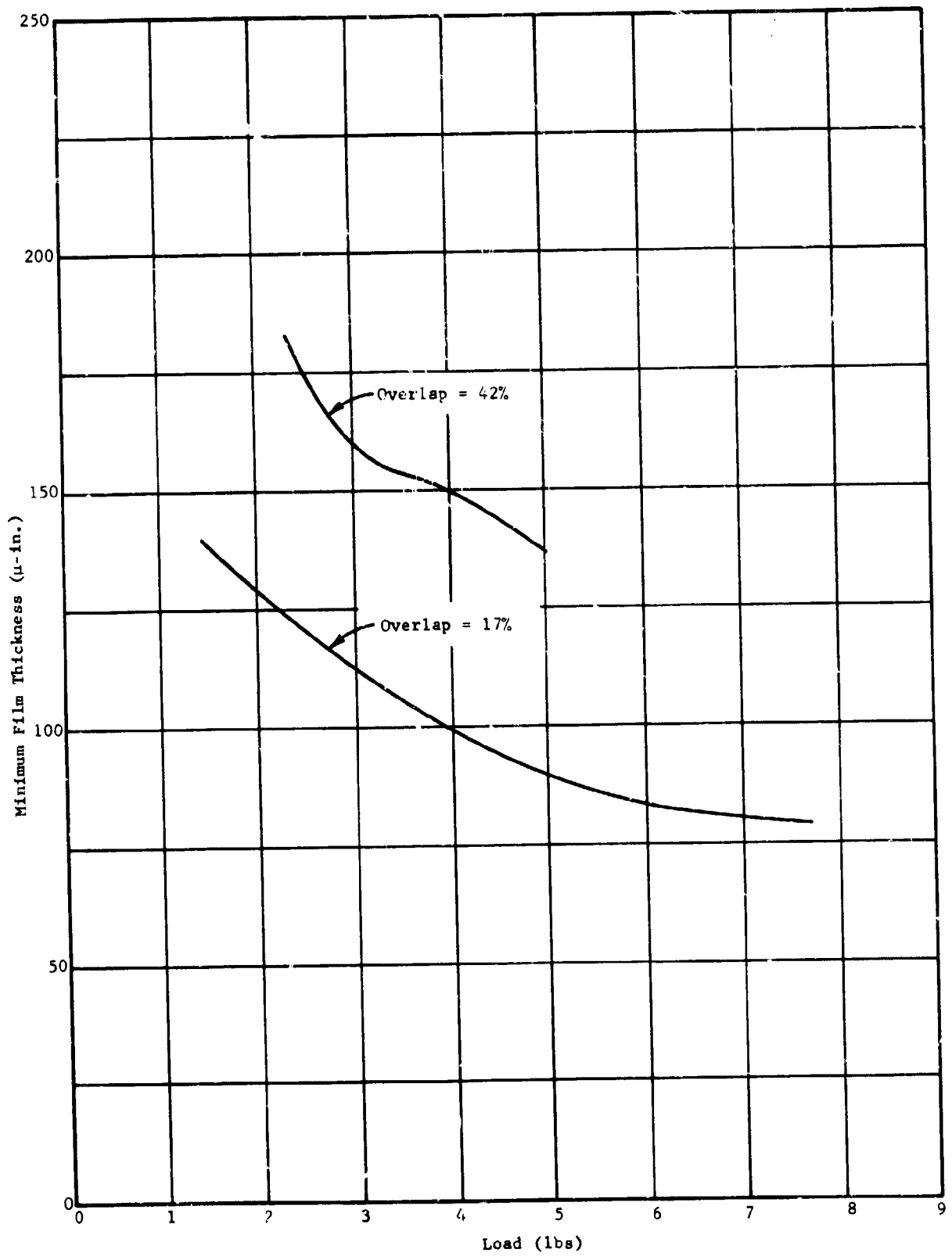


Fig. 40 Effect of Overlap on Film Thickness for Leaf Journal Bearing

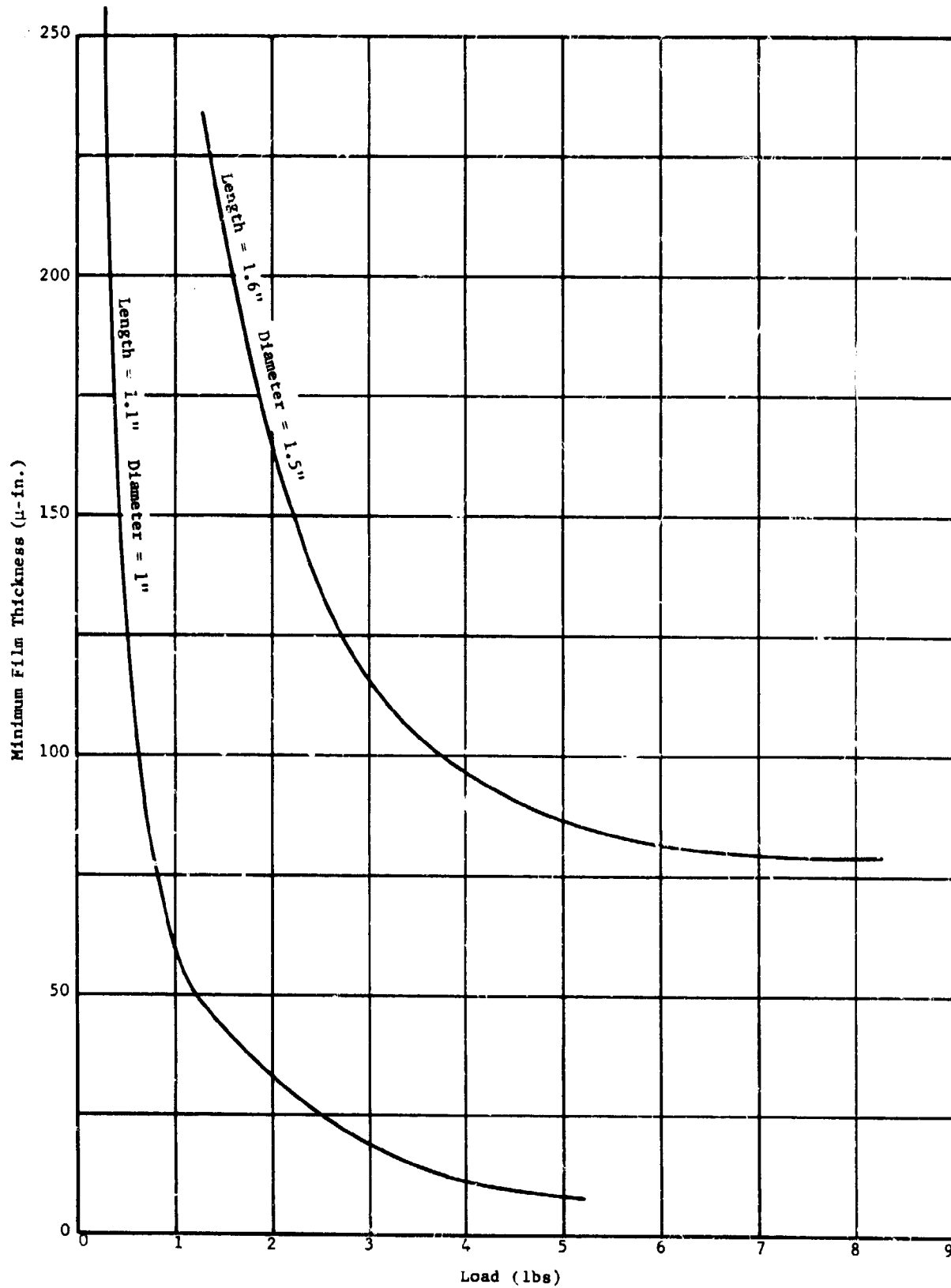


Fig. 41 Effect of Bearing Size on Minimum Film Thickness (0 Preload) for Leaf Journal Bearing

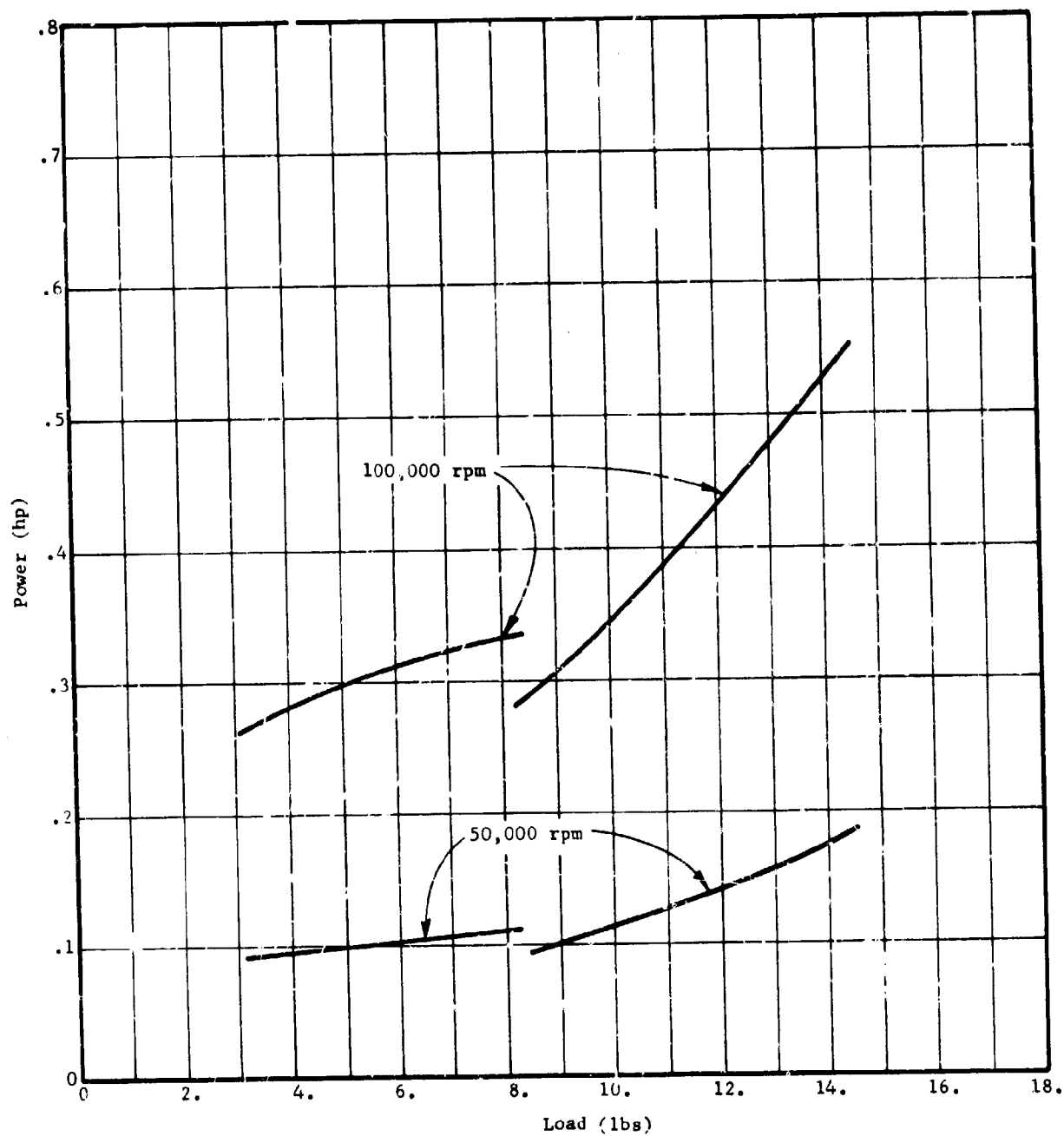


Fig. 42 Effect of Speed on Power for Leaf Journal Bearing

estimates are highly approximate and the breaks in the curves at incipient wrapping should be disregarded.

Finally, the variation of stiffness with load for the 10 and 15 mil leaf thicknesses at different preloads is shown in Figure 43. In general, if the preload is greater than the shaft eccentricity, the stiffness in the preloaded bearing will be constant and approximately twice that of the unpreloaded value. As the radial deflection exceeds the preload value, the shaft pulls away from the upper leaves and the stiffness gradually falls to the zero preload value. The thickness of the leaf, of course, has a very marked effect on the stiffness.

4. Analysis and Design Data for the Cantilevered, Leaf-Type Thrust Bearing

A schematic of the leaf thrust bearing under consideration is shown in Figure 44. The bearing consists of a spring assembly which is essentially a plate with springs attached below it which is separated by a backing plate from a foil assembly plate on which the leaves are mounted. The plates will, of course, be in the form of annular rings but for purposes of the one dimensional analysis to be used here, Figure 44 shows the bearing in unraveled form. Reference 3 describes a similar type of thrust bearing.

If we take the springs to be in the form of circular bumps as shown in Figure 44, then the elasticity considerations for bump deflections that have already been developed may be used for calculating stiffness. One may calculate the stiffness per spring per unit length from Equation (23) for the frictionless case or from Equation (22) and Figure 2 when friction is considered. The total bearing stiffness would then be this unit stiffness multiplied by the land width of the bearing and by the number of springs.

In considering the deflection of the bearing between springs, one may lump the plate containing the springs, the plate containing the leaves and the backing plate conceptually into a single plate having an effective flexural rigidity D_{eff} upon which very flexible leaves are mounted. A schematic showing the conceptual deflection of the loaded bearing is shown in Figure 45. The plates deflect between bumps forming a both convergent and divergent sections. The leaves which are very flexible, rest on the top plate when the film thickness is convergent and the pressures are above ambient. The leaves, however, will tend to resist the formation of large divergent sections under conditions of sub-ambient pressures as they will lift off the top plate as shown in Figure 45. In addition to eliminating sub-ambient pressures, the leaves serve a second purpose of providing ambient pressure supply grooves. The leaves should be thin enough to permit considerable flexibility but still be fairly thick in comparison with the film thickness, hence the land between leaves will serve as ambient pressure sources. The third effect of the thin leaves could be to trigger the pressure generation mechanism which causes the deflections necessary to produce the resultant convergent sections. This triggering could result from an initial preformed radius of the leaves or from slight step bearing action near the start of each leaf.

The precise operation of a leaf thrust bearing is somewhat open to conjecture in that necessary experimental data are not presently available for purposes of analysis. It is possible, however, to obtain bounds on the operation of

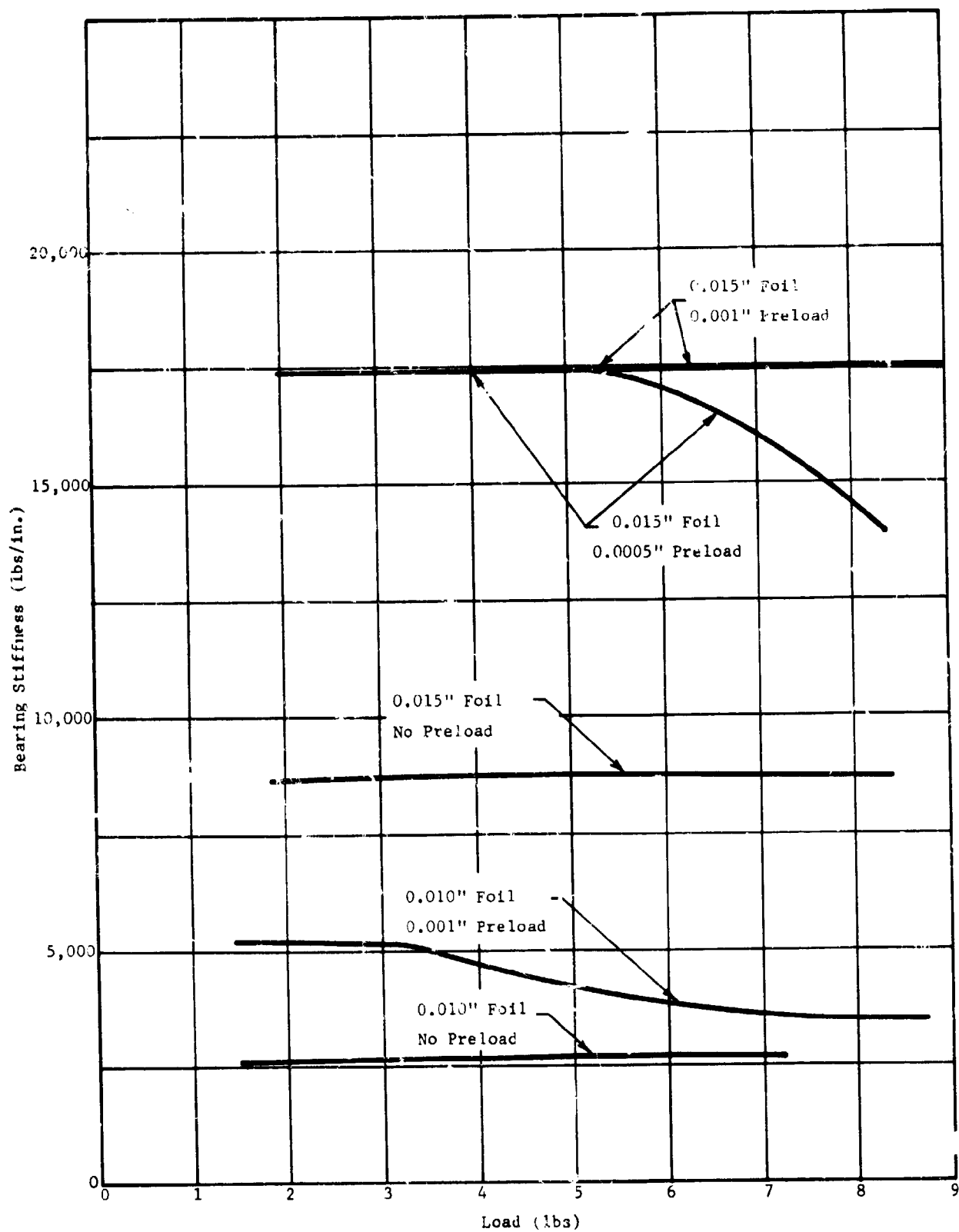


Fig. 43 Leaf Bearing Stiffness Vs. Load

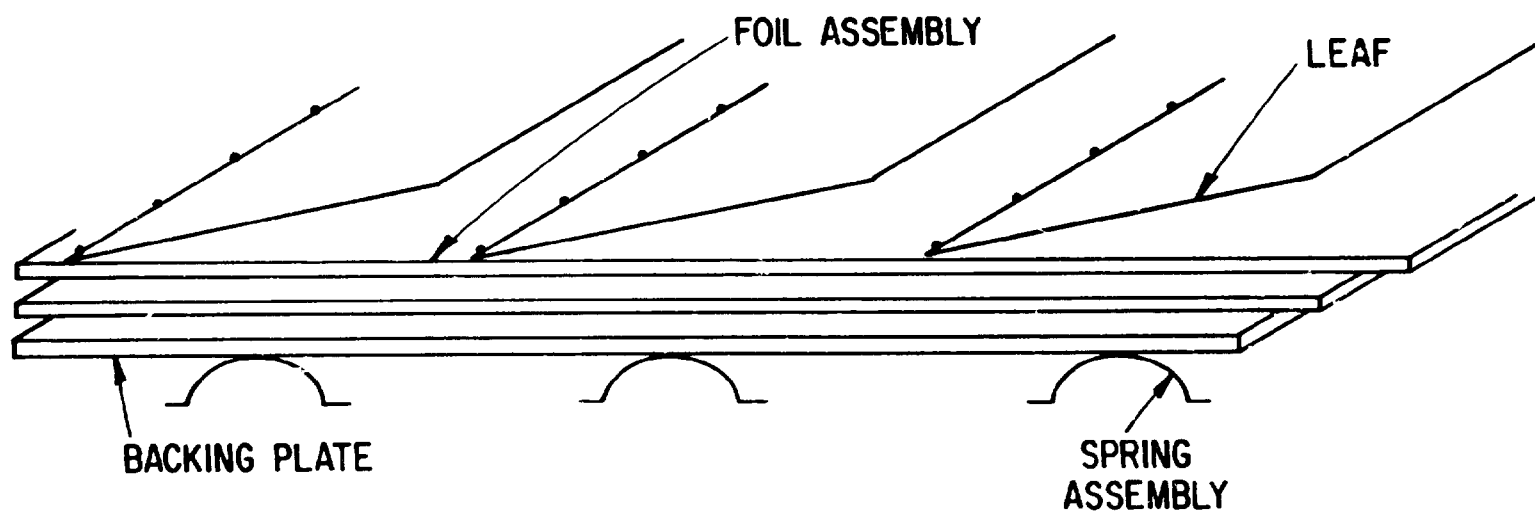


Fig. 44 Schematic of One Dimensional Leaf Thrust Bearing

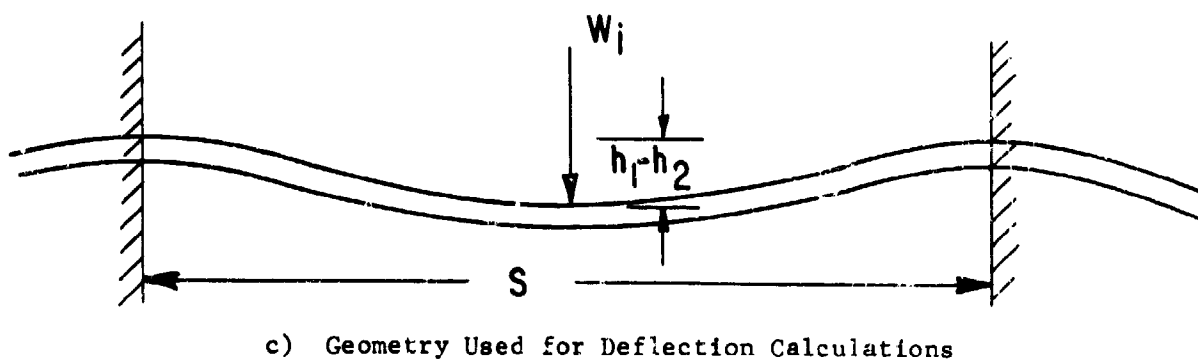
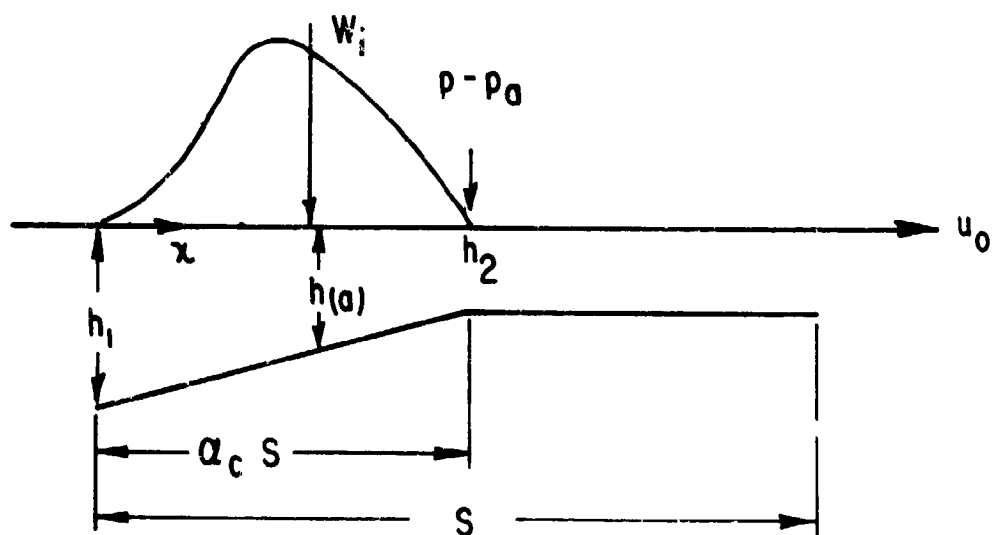
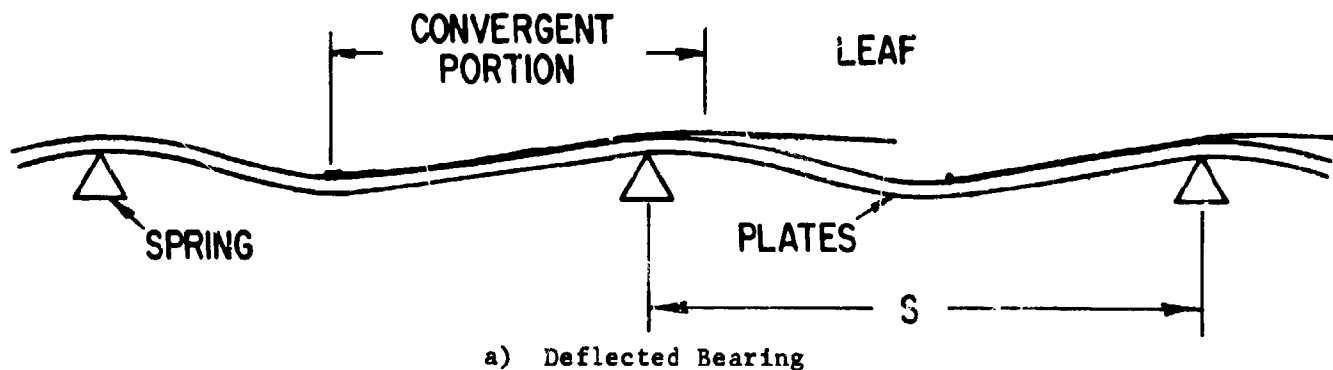


Fig. 45 Schematic of Geometry Used for Leaf Thrust Bearing Analysis

the leaf thrust bearing by analyzing the mechanism stated above in an approximate manner. A conceptual drawing, showing the geometry to be used for the hydrodynamic analysis, is shown in Figure 45. If we take S to be the spacing between springs, and we arbitrarily assume that a fraction of that region α_c is convergent, we have a mechanism for calculating load capacity. By suitably designing the bearing, it should be possible to make α_c at least equal to one half but, of course, no greater than one. We will assume that the non-converging portion of the bearing does not contribute to the load capacity (due to the flexibility of the leaf) but does contribute to traction. We will for simplification further treat the convergent portion as a linear slider bearing as shown in Figure 45.

Load capacity and power requirements for an infinitely long linear slider bearing have been well worked out in the literature. The results have been extracted from Reference (2) and put into a form commensurate with the notation in this report.* The dimensionless load capacity

$$\bar{W} = \frac{W_i}{p_a \alpha_c S} \quad (142)$$

may be expressed in terms of a compressibility number Λ

$$\Lambda = \frac{6\mu u_o \alpha_c S}{p_a h_2^2} \quad (143)$$

and the film thickness ratio, a ,

$$a = h_2/h_1 \quad (144)$$

In the above equations, the load per leaf W_i would be the same on each leaf and equal to the total load W divided by the number of leaves N .

For a given value of Λ , one may plot $\bar{W}$ as a function of a to find the magnitude a_{max} for which $\bar{W}$ is a maximum. This procedure has been carried out over a range of Λ 's and cross plots have been obtained of a_{max} as a function of $\bar{W}$ in Figure 46 and Λ as a function of $\bar{W}$ corresponding to $a = a_{max}$ in Figure 47.

The suggested design procedure would be to assume a value of $\alpha_c = 1/2$ and calculate W from Equation (142). One would then determine Λ from Figure 47 which

* Numerical solutions were carried out for evaluating Λ beyond those given accurately in Reference (2). Solutions in Reference (2) have been used for values of Λ up to 50. Further solutions have been obtained for Λ up to 400. Extrapolations are used for values of Λ greater than 400.

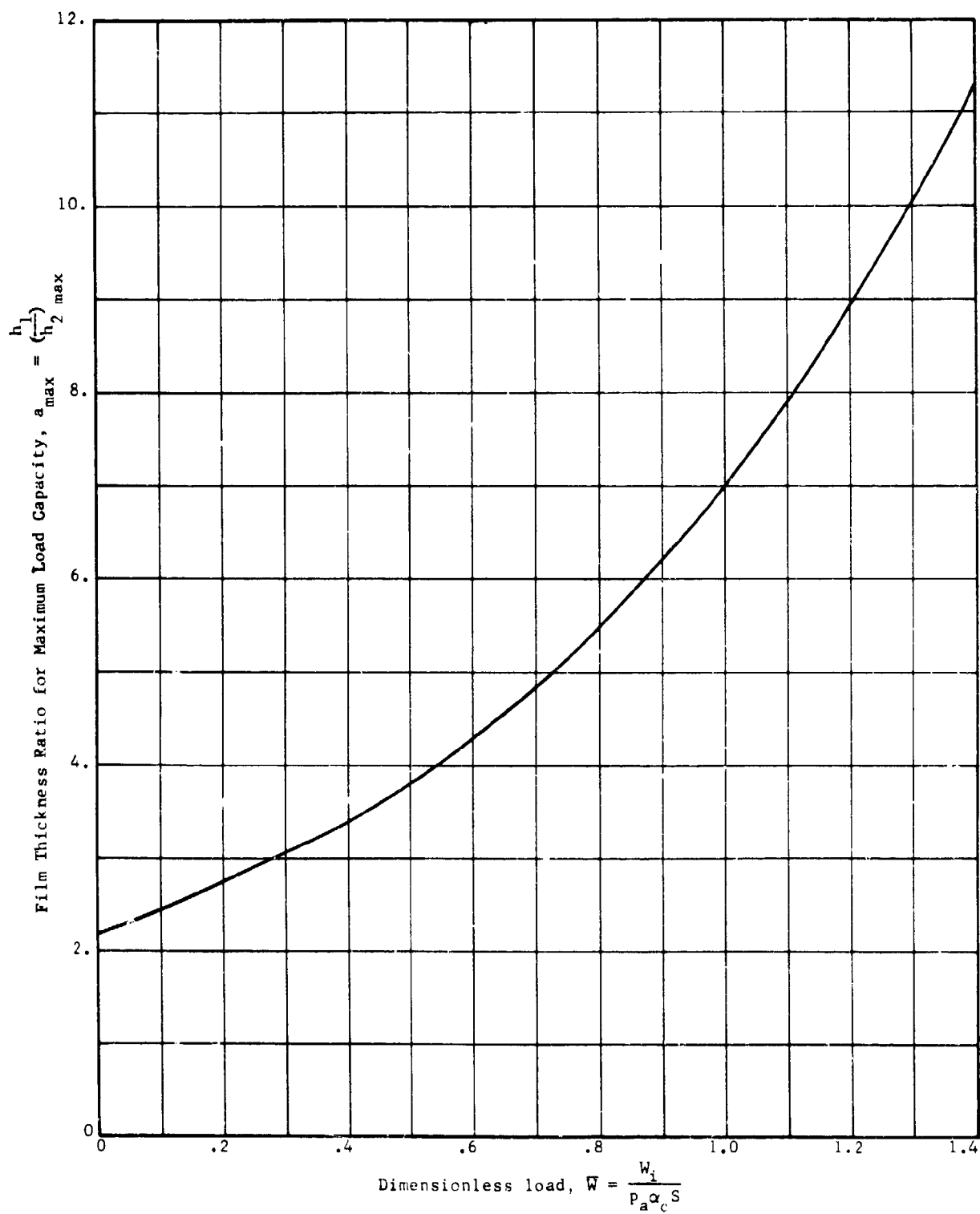


Fig. 46 Optimum Film Thickness Ratio Vs. Dimensionless Load

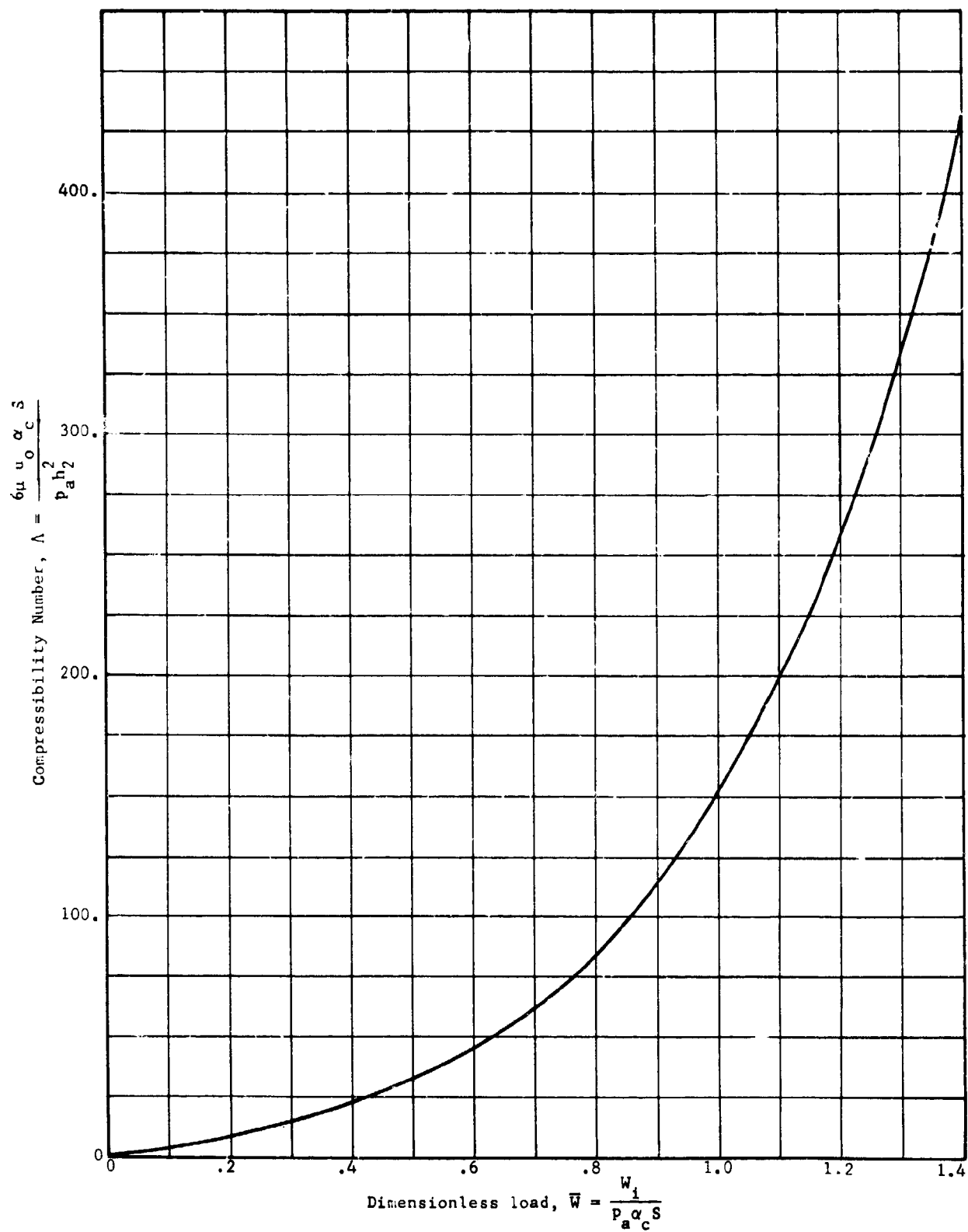


Fig. 47 Leaf Thrust Bearing Compressibility Number Vs. Dimensionless Load

would be used together with Equation (143) to calculate h_2 . $a_{\max}$ would then be determined from Figure 46.

The value of $a_{\max}$ would only be obtained in practice if the effective flexural rigidity of the plates (D_{eff}) were of the right magnitude. An estimate of the appropriate magnitude for D_{eff} may be obtained with the use of the model shown in Figure 45c, which shows the entire leaf load to act at the center of the span between springs. The validity of this assumption would be dependent upon the placing of the leaves relative to the springs and, in general, this assumption will only provide a rough estimate. If we adopt this assumption, then the elastic deflection of the leaf would correspond to that for a beam with fixed ends loaded at the center. The solution to this problem has been taken from Reference (1) and is calculated based on a maximum deflection of $h_1 - h_2 = (a_{\max} - 1)\tilde{h}_2$ to give the following result

$$D_{\text{eff}} = \frac{S^3 W_i}{24 (a_{\max} - 1) h_2} \quad (145)$$

Finally, the power requirements can be expressed in terms of a dimensionless torque parameter $\bar{T}$ defined as

$$\bar{T} = \frac{T h_2}{\mu u_o R_M \alpha_c S} \quad (146)$$

The Couette portion of the torque for an infinitely long slider bearing is related to α_c and a , by

$$\bar{T} = \frac{(1 - \alpha_c)}{\alpha_c} + \frac{\ln a}{a - 1} \quad (147)$$

The quantity T would be the torque per leaf per unit land width. The power would thus be the product of T and the number of leaves N , the land width, and the angular velocity Ω .

These calculations, if carried out with a value of $\alpha_c = 1/2$, should provide a reasonably conservative estimate of the load capacity and torque. The calculations if carried out with a value $\alpha_c = 1$, on the other hand, should provide an estimate of the highest load capacity that one could ever hope to obtain from a leaf bearing of the present design.

Equations (142) through (147) are readily amenable to hand calculation and have also been computerized and the program is given in Appendix II.

As an illustration, the friction torques of leaf type bearings have been calculated for the following condition:

Bearing Outside Diameter: 3.5 inches
 Radial Land Width : 0.7 inches
 Number of Leaves : 12
 Speed : 60,000 rpm
 Air Viscosity : 2.9×10^{-9} reyns

The calculations were made for two values of the convergent portion of the leaf, i.e., for $\alpha_c = 0.5$ and 1.0 . The resulting friction torques are plotted in Figure 48, as functions of load. For this geometry, convergence of the leaves over their full circumferential lengths results in much lower power loss due to operation on a thicker film.

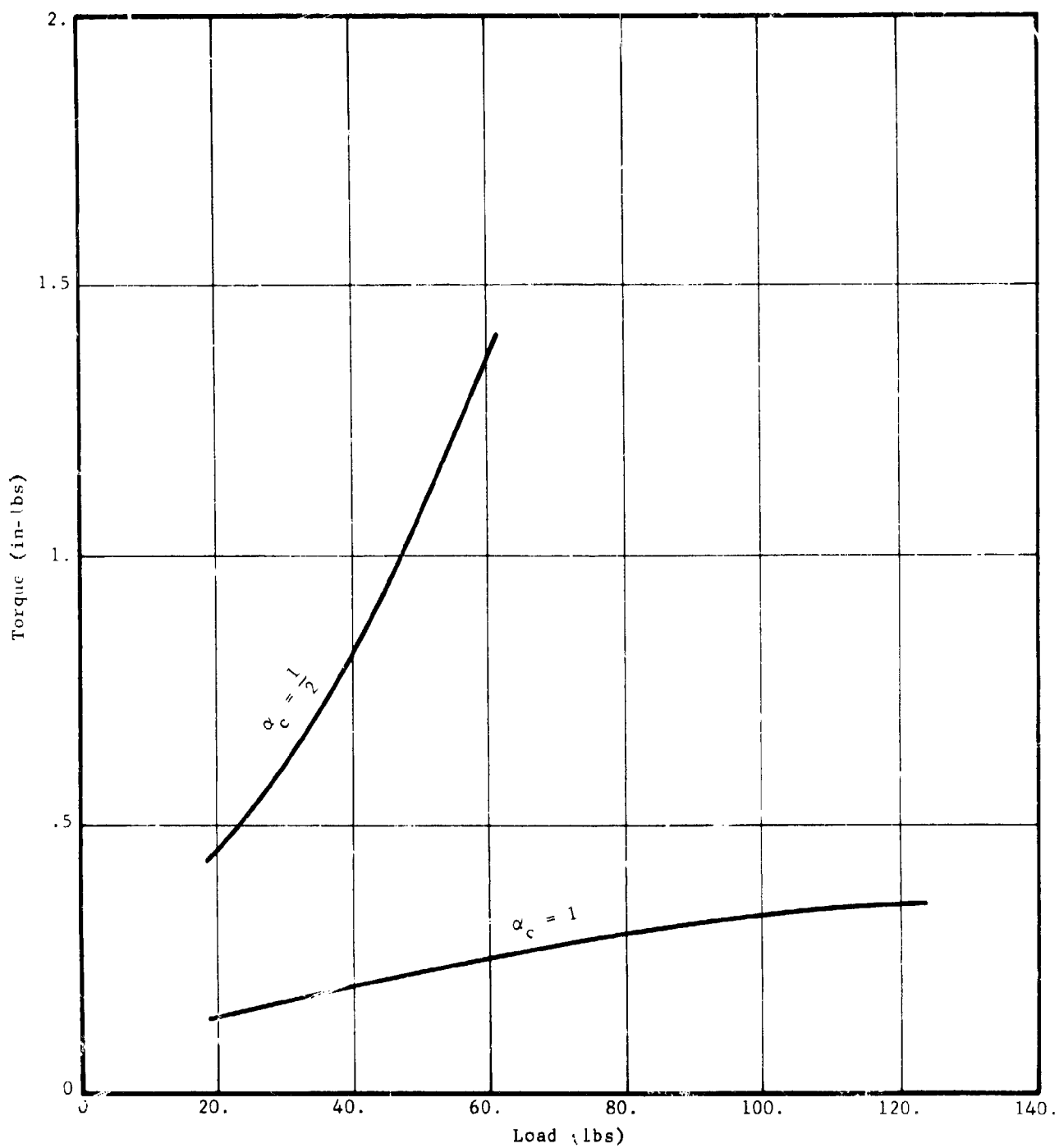


Fig. 48 Comparison of Torque Between Leaf Thrust Bearings

SECTION IV

ANALYSIS OF THE PLEATED BEARING

A third foil bearing configuration was initially considered. This is the pleated bearing illustrated schematically in Figure 49. Conceptually, this is the simplest of the bending-dominated foil bearing types as it can be made, for example, from a single foil folded into a number of pleats. Prior to the present study, the pleated bearing had been operated satisfactorily in light duty applications where the load capacity requirements were very small, of the order of 1 or 2 psi.

Further, further experience with the pleated bearing obtained during the course of the present program showed that, in its present configuration, it exhibited substantially lower load capacity than the other two types of foil bearings that were discussed in Sections II and III of this report. In particular, as discussed in Section V of this report, an air-lubricated pleated bearing did not survive the 5 psi load that was applied in the start-stop tests. The experience in this test confirmed also that, when loaded to the point where rotor-to-stator rubs occurred during operation of the rotor, the foil tended to break at the pleats, because of the high stresses in these regions. As noted in Section V, the pleated bearing was replaced by a ridged type of foil bearing for the start-stop and high-speed rub tests.

The low load capacity problem associated with single-foil, pleated bearing in gas-lubricated bearing applications is related to the fact that the taper of each pleat is relatively large in comparison with the trailing edge film thickness. This results in values of h_1/h_2 , which are far greater than the optimum value and, hence, in load capacities that are inadequate for engine applications. This was demonstrated in a simple analysis that was undertaken to estimate the maximum load capacity obtainable from the pleated bearing, to compare it with the load capacities calculated for other, bending-dominated foil bearings. It was deemed desirable to make such a comparison before undertaking any more elaborate analysis of the pleated bearing in its present, single foil configuration. The results of the comparison are illustrated in Figure 50 and discussed below.

The upper limit on the load capacity of the pleated bearing in its present, single foil configuration can be obtained by treating each section as an incompressible linear sliding bearing and neglecting side-leakage effects. The load capacity for such a bearing, taken from Reference 2, is given below:

$$W = \frac{6\mu u_o S^2}{h_2^2} \frac{1}{(a-1)^2} \left[\ln(a) - \frac{2(a-1)}{a+1} \right] \quad (148)$$

If the thickness of the pleated foil is 3 mils, it will be impractical to make the taper, h_1-h_2 any smaller than say 9 or 10 mils. The comparison of load capacities for a pleated thrust bearing containing 12 pleats and a leaf bearing containing 12 leaves is shown in Figure 50. The bearings have outside diameters of 3-1/2 inches, land widths of 0.7 inches and operate at a speed of 60,000 rpm in air having a viscosity of 2.9×10^{-9} Reyns. (These are the same conditions used in obtaining Figure 48). Film thicknesses are shown for the leaf bearing

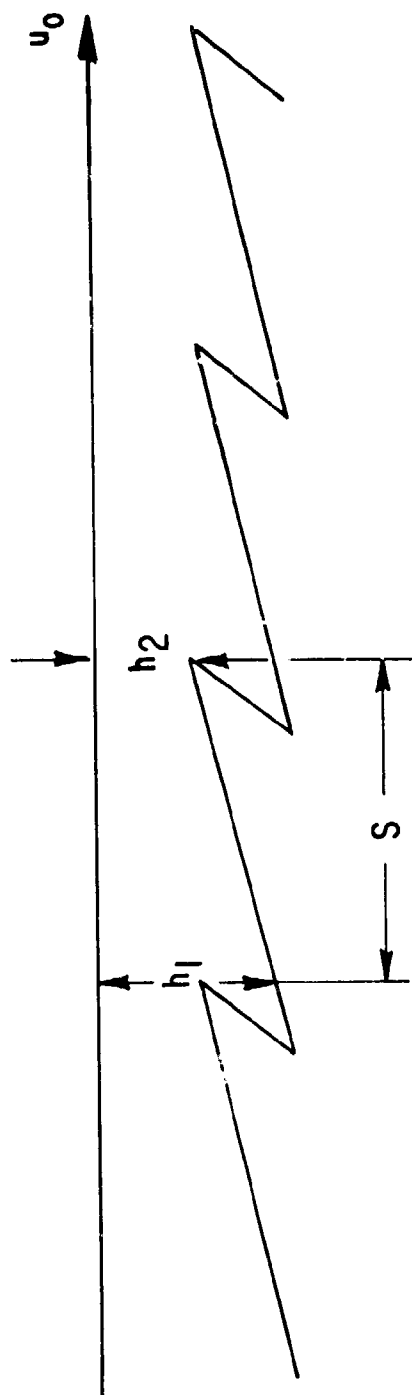


Fig. 49 One Dimensional Pleated Bearing

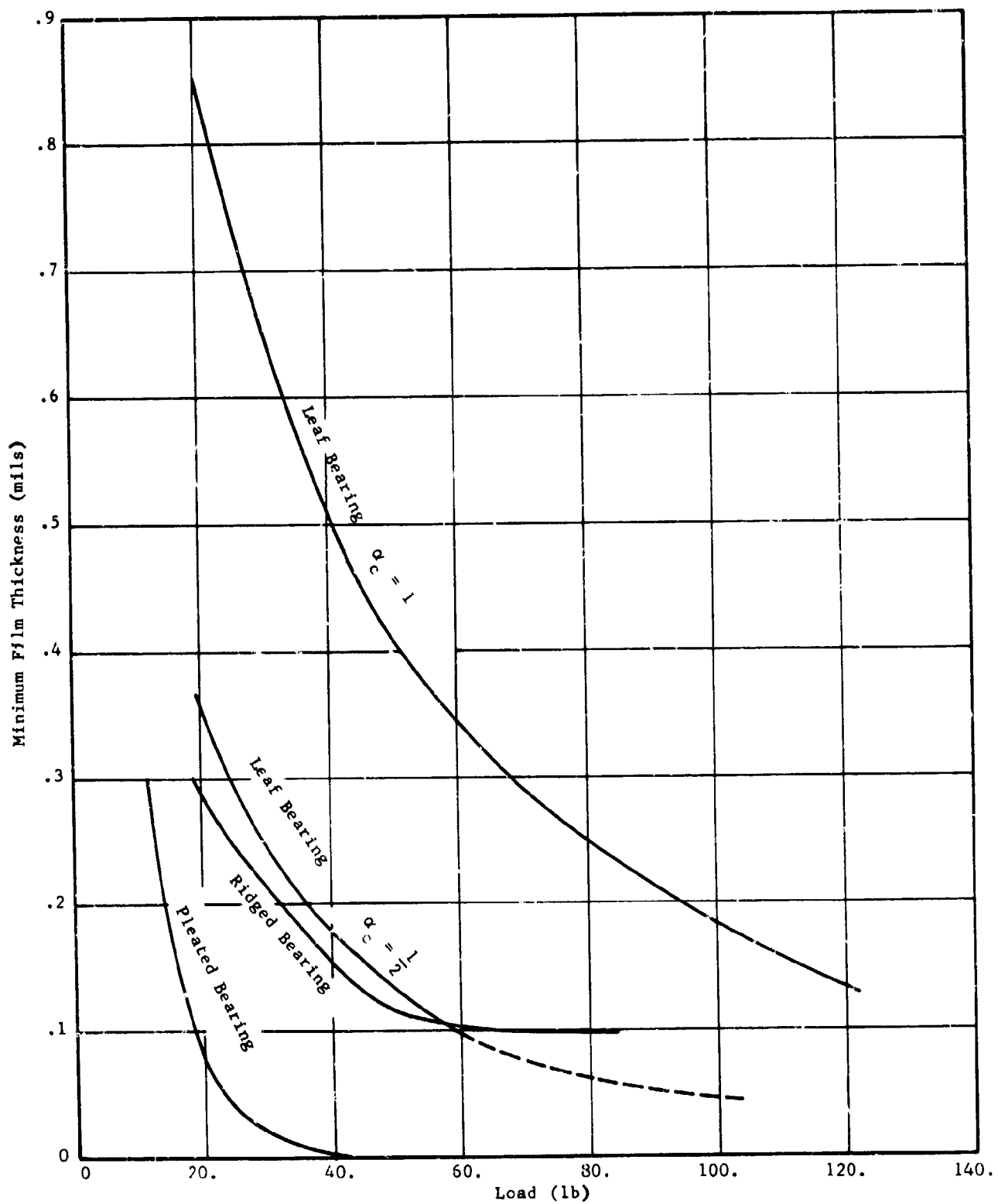


Fig. 50 Comparison of Bending Dominated Foil Bearings

corresponding to $\alpha_c = .5$ and $\alpha_c = 1$. Bump heights of $1/32$ inch and an inlet length of $1/4$ inch^c were used in obtaining the ridged bearing curve. The results confirm the relatively poor load capacity of the single-foil, pleated bearing compared to the other types. Moreover, compressibility effects are included in the calculations for the ridged and leaf type thrust bearings, but neglected for the pleated thrust bearing, so that the comparison is optimistic, as regards the pleated bearing.

In light of the above analytical results, as well as the low load capacity experienced in the early start-stop tests and the stress concentration problem at the pleats, further analysis of the pleated bearing was not performed. Substantial modifications of the geometry of the pleated bearing are clearly necessary, before it can be considered with air-lubrication in aircraft engine applications.

SECTION V

FOIL BEARING MATERIALS

The overall objective of the materials phase of this program was to investigate and evaluate promising foil bearing materials and surface treatments for use in aircraft engines and power systems.

Although the basic concept of the foil bearing is established, there is very little useful data available on the performance of foil bearing materials. Wear rates, material compatibility and anticipated life as a function of stress are all unknown factors. Such information is needed in order to advance the state-of-the-art.

The selection of materials for use in these high temperature foil bearings is based on the following considerations:

- Bulk Material Properties
- Bearing Surface Properties
- Utilization Aspects

Bulk material properties are those properties that are important to the bearing as a load support structure. For example, what stress can the material sustain without permanent deformation? Will the material fatigue as the result of the large number of bending cycles which it must sustain? Is it possible for the foil material to be fabricated and heat-treated without inducing permanent distortions that will affect bearing geometry and performance? These questions relate to the suitability of the bearing as a machine element.

Bearing surface properties are those properties which relate to the ability of the bearing to sustain a number of start-stop contacts as well as high-speed rubs without surface damage or failure. This is critical to achieving successful high temperature gas bearing operation.

Finally, the utilization aspects have to be considered. For example, is the material available in strip form in a range of thicknesses? Can the strips be heat-treated to achieve required physical properties? Can welding attachments to the housing be made? Such questions relate to processing of the material to arrive at desired foil bearing geometries.

In prior developments of rigid surface gas bearings, one design philosophy has been to select suitable base alloys as the bearing and shaft materials, without regard for their sliding behavior. This permits the designer to choose the best alloys for bulk properties, such as: strength, density, thermal expansion characteristics, etc. Then, surface coatings or surface modifications are used to provide the desired surface properties including good sliding behavior, erosion resistance, etc. In this way, a wide variety of both substrate materials and surface coatings can be considered on their own merits without imposing an unreasonable number of requirements on any one material. This design philosophy has given very successful results in a large number of gas bearing systems.

The same approach was taken in the foil-bearing application, although it was recognized that the required flexibility of the foils may present problems with certain coatings. The approach which was used to select and investigate materials in the present program was broken down into the following tasks:

- Survey of Material Requirements for Foil Bearings
- Survey of Foil Bearing Substrate Materials
- Survey of Foil Bearing Surface Treatments
- Experimental Evaluation of Two Foil Journal Bearing Assemblies

The results of the work in each of these tasks is discussed in this section of the report.

1. Survey of Material Requirements for Foil Bearings

Regardless of the actual design, the basic function of the foil bearing is to act as a flexible, compliant surface which will conform to a certain shape in response to some particular applied stress and which will recover elastically when this stress is removed. Thus, the physical properties of materials which appear to be of most concern include the following:

- Modulus of Elasticity
- Yield Strength
- Tensile Strength
- Fatigue Endurance Limit
- Ductility

In addition, the metallurgical characteristics and condition of the material will have a marked effect on the results obtained. Table II shows some of the interrelationships among the general material requirements and properties.

However, it should be noted that the stiffness of a bending-dominated foil bearing will vary as the cube of the thickness of the foil section. Thus, for any given geometry, both the elastic deflections and the bending stresses are largely controlled by the foil thickness used. This can overshadow the effect of variations in the material properties identified above.

Among the aspects to be considered in the selection of materials for engine foil bearings, the following appear to be of greatest importance:

- Operating Temperature Range (-65°F to 1000°F)
- Oxidation Resistance
- Surface Damage Resistance

TABLE II

GENERAL REQUIREMENTS FOR BENDING-
DOMINATED FOIL BEARING MATERIALS

	Requirement	Criteria for Selection of Materials
Bulk Foil Material Properties. (Mechanical, Physical and Chemical)	Flexural Stiffness	Modulus of Elasticity Foil Thickness Yield Strength over the temperature range
	Ductility	Metallurgical Condition Work and Age Hardening Characteristics Poisson's Ratio Yield and Tensile Strength
	Notch Toughness	Crystal Structure Chemical Composition Metallurgical Condition
	High Fatigue Strength	Tensile Strength Endurance Limit
	Metallurgical and Dimensional Stability	No metallurgical change within the operating temperature range (-65°F to 1,000°F).
Bearing Surface Properties (Friction and Wear)	Abrasion Resistance, Low Wear Characteristics, Low Friction Coefficient	Coatings or surface modifications will be required. These can either be hard, wear-resistant coatings, such as ceramics or cermets or low friction, solid lubricant films. Past experience and available test data are the only criteria available for the selection of these coatings.
Utilization Aspects (Manufacture and Use)	Availability Ease of Forming Suitable high temperature strength and high fatigue endurance limits	Availability in particular sizes and thicknesses. Ductility and formability. Availability of pertinent background information and data on processing. Available data and information from the literature.

- Abrasion Resistance
- Coefficient of Friction
- Pressure (altitude) Range
- Pressure Gradient
- Temperature Gradient
- Manufacturing Techniques
- Quality Control Procedures

Table III summarizes the state-of-the-art of materials technology as it applies to each of these aspects.

The specified operating temperature range (-65°F to 1000°F) has a greater influence on the selection of foil materials than the other environmental aspects or operating variables, because of its effect on corrosion rates and on the physical properties of materials. Consider, for example, some of the candidate foil materials as listed in Table IV. None of the steels or refractory alloys have adequate oxidation resistance for long term operation in air at 1000°F. In addition, the requirement for operation at -65°F will rule out many alloys which have poor ductile to brittle transformation temperatures. Finally, many materials will lose strength at high temperatures.

Sliding compatibility, which encompasses surface damage and abrasion resistance as well as frictional behavior, can also be influenced by temperature. However, since protective coatings are required to minimize sliding damage and to reduce friction, the primary concern is to find coating materials and binders which can tolerate exposure to a temperature of 1000°F for reasonably long periods of time and which will provide the desired friction and wear behavior in the bearing.

While both temperature and pressure gradients can influence bearing performance, there is little that can be done to counter these conditions, as far as materials selection is concerned, except to anticipate such gradients in the design phase, and to insure that they are not so severe as to cause local buckling.

Manufacturing techniques and quality control procedures influence material selection primarily to the extent that it may be impractical to test some new alloy composition until suitable background data is generated to define its strength characteristics, high temperature properties and metallurgical processing techniques.

2. Survey of Substrate Materials for Foil Bearings

A survey has been made of candidate substrate materials for use as foil bearings in the temperature range from -65°F to 1000°F. Consideration was given to the following classes of alloys:

TABLE III
OPERATING PARAMETERS WHICH WILL GOVERN
THE SELECTION OF FOIL BEARING MATERIALS

Operating Parameters	Comments
Operating Temperature	This is a governing factor as regards materials selection. Present technology is sufficient to permit selection of suitable materials for the temperature range of -65°F to $1,000^{\circ}\text{F}$. The problem of materials selection will become more acute for temperatures above $1,200^{\circ}\text{F}$.
Sliding Competitibility • Friction • Surface Damage • Abrasion Resistance	This is a governing factor, particularly as regards selection of surface coatings. Present technology is at least partially adequate to make preliminary selections of surface coatings for the temperature range of -65°F to $1,000^{\circ}\text{F}$. However, tests are necessary to verify approaches. The high end of this temperature range will limit the choice of protective coatings. The problem will become increasingly severe as temperatures are raised above $1,000^{\circ}\text{F}$.
Absolute Pressure and Gradients of Temperature and Pressure	These parameters are more important from the standpoints of bearing load capacity. They will have little influence on materials selection. The major effect will be an increase in liftoff speed and, hence, of rubbing contact between the bearing and the journal.
Corrosion Resistance	Present technology permits selection of available foil materials that will not have corrosion problems at temperatures below about $1,300^{\circ}\text{F}$.
Manufacturing and Processing	Specific processing techniques will need to be developed for individual foil bearing geometries particularly from the standpoint of foils retention.
Quality Control	Present technology is adequate as far as the substrate materials are concerned. Quality control procedures are poorly defined as far as surface coatings are concerned. Specific quality control tests and procedures for the coatings will need to be generated at a future stage in the development of foil bearings for high-temperature engine applications.

TABLE IV
POTENTIAL SUBSTRATE MATERIALS FOR USE AS FOIL
BEARINGS OPERATING IN SPECIFIED TEMPERATURE RANGES

<u>Ambient to 700°F</u>	<u>Ambient to 1200°F</u>	<u>Ambient to 1600°F</u>	<u>Above 1600°F</u>
Steels:	Stainless Steels:	Nickel Alloys:	Oxidation is the primary barrier. If suitable materials or surface coatings are found, then strength and metallurgical stability will be the major consideration.
AISI 4340	Type 316*	Inconel 718**	
Type 410	Type 347*	Nickel-Chrome Alloys	
Mod. H-11	17-7 PH		
M-50	Cobalt Alloys:	Cobalt Alloys:	
Molybdenum:	Haynes 25	S-816	
IZM	Nickel Alloys:		
	Inconel 750X*		
	Hastelloy X		

* Upper end of temperature range should be limited to 1000°F

** Upper end of temperature range should be limited to 1300°F

- Low-alloy, high-strength steels
- Martensitic and ferritic stainless steels
- Austenitic stainless steels
- Nickel-base superalloys
- Cobalt-base superalloys
- Refractory metals

At the present time this list is believed to include all serious candidates that could be used for this application. (See also Table IV).

As noted previously, one particular environmental factor dominates the selection of substrate materials. That is, the need for operation over the entire temperature range from -65°F to 1000°F . The two extremes, both low and high temperature, will impose severe limitations on which classes of materials can be used with confidence for foil bearings. At low temperatures, the ductile-brittle transition temperature of metals, especially the low-alloy steels and the ferritic stainless steels, could present a major problem. If the transition temperature of the alloy was above -65°F , then even light impacts against folds, bends, or other high-stress points in the foil at this low temperature could result in cracking. The safest candidates for low-temperature use are those alloys with a cubic structure, such as the austenitic stainless steels, or alloys which contain more than 8 percent nickel. The nickel-base superalloys are, therefore, very promising.

At the high end of the temperature range up to 1000°F , strength, metallurgical stability, and corrosion resistance will govern the selection of materials. Again, the nickel-base superalloys and the stainless steels seemed to offer the most promise, at least from the standpoint of corrosion resistance. Special consideration was given to the stainless steels, especially the austenitic types, because of cost and availability. For example, cold-rolled type 305 stainless is commonly used as a shim stock material and is readily available in a variety of thicknesses. However, prolonged exposure of these alloys to temperatures between 800°F and 1200°F can result in carbide precipitation and decreased ductility. For this reason, the cold-worked austenitic stainless steels are limited to temperatures below 750°F for long term operation.

The martensitic and ferritic stainless steels have many questionable characteristics including the fact that many of these alloys do not have favorable ductile to brittle transition temperatures, and are susceptible to carbide precipitation at temperatures above 800°F . Like the low alloy carbon steels, there are certain alloys in these two classes which could meet the required ductile to brittle transition temperature and they do have the necessary corrosion resistance for use in air to 1000°F . However, before any of these alloys could be used as foil bearings, it would be necessary to demonstrate their suitability and, within the limited scope of this present work, there was no justification for expending effort on candidate materials which might not meet the requirements of the application when there were other alloys available that did not have any obvious disadvantages.

Certain of the nickel-base superalloys are particularly promising for foil bearing applications since they have excellent low-temperature properties, good corrosion resistance, and the necessary high-temperature strength for an aircraft engine application. Inconel X-750, for example, appears to be one of the best candidates. It is suitable as a high-temperature spring material for continuous use at 1000°F. Because of the high nickel content of this alloy, low-temperature embrittlement will not be a problem. At high temperatures, to at least 1000°F, corrosion resistance and strength characteristics are satisfactory. One of the chief advantages of this alloy is the fact that it can be age-hardened. The part can, therefore, be formed in the annealed condition and then age-hardened to develop full strength.

As an alternate material, one cobalt-base superalloy, Haynes 25, was also selected. This alloy must be cold-worked to develop strength and will retain the beneficial effects of this cold working to a temperature of about 1200°F.

To sum up this discussion, Inconel X-750 is the leading choice for the foil bearing material at the present time. The one particular advantage of this alloy is the fact that it can be age-hardened after forming. Haynes 25 appears to be a reasonable second choice, but the alloy must be cold-worked for optimum strength. Cold-worked austenitic stainless steels could be used at temperatures below 750°F, but carbide precipitation could be a problem at temperatures above 800°F. Pertinent physical property data on these candidate alloys are given in Table V.

3. Survey of Surface Coatings for Foil Bearings

Surface coatings are required on foil bearings for several reasons. First, most foil bearings have characteristically higher lift-off speeds than rigid gas bearings, partly because of initial preload. Thus, the time in sliding contact during starts and stops is significantly higher with the foil bearings, and any surface treatment that will reduce the amount of heat generated by friction will be beneficial. Secondly, the starting torque of the foil bearings may be higher than that of rigid surface gas bearings, because the elasticity of the foil bearings favors some initial stick-slip. Coatings which can reduce this starting friction will be essential for many applications. Even when improvements in the starting torque and lift-off characteristics of the foil bearings are realized as the result of design improvements, the fact that these bearings will be used in aircraft applications means that they will be subjected to a variety of maneuvering loads and should be tolerant of short-term, high-speed rubbing contacts. In addition to the above considerations, it should be noted that one of the most promising substrate foil materials which has been found, to date, for the temperature range to 1000°F is Inconel X-750. Previous experience with this alloy in sliding applications has shown that, at temperatures below the point where visible oxide films are formed, Inconel X-750 has relatively poor sliding compatibility and is susceptible to welding and material transfer. Finally, the likelihood of abrasive particle contamination being present in the bearing clearances makes the use of protective coatings a necessity.

A preliminary review was made to determine which surface coatings or surface modifications could be used to provide protection for the surface of foil

TABLE V
TYPICAL PHYSICAL PROPERTIES OF SHEET MATERIAL AS
A FUNCTION OF TEMPERATURE AND METALLURGICAL CONDITION

<u>Alloy</u>	<u>Condition</u>	<u>Temperature</u>	<u>Ultimate Tensile Strength</u>	<u>Yield Strength (0.2%)</u>	<u>Elongation</u>
Inconel X-750 (0.062" Sheet)	Cold rolled and	75°F	110,000 psi	46,500	51%
	Annealed	1000°F	91,000	35,000	55%
	Cold rolled,	75°F	177,000	122,500	2%
	annealed and				
	age-hardened	1000°F	154,000	112,000	26%
	20 Hrs. at 1300°F A.C.				
Haynes 25 (0.050" Sheet)	Annealed	75°F	146,000	67,000	64%
		1000°F	115,700	35,800	59%
	20% cold	75°F	183,000	141,000	19%
	reduced	1000°F	156,000	133,000	18%
Type 301 Stainless	Annealed	75°F	90,000	40,000	60%
		1000°F	61,000	24,000	--
	1/2 Hard	75°F	155,000	106,000	22%
	(20% cold reduction)	1000°F	82,000	65,000	10%

bearings operating in the temperature range from -65°F to 1000°F. For satisfactory performance, these surface coatings must meet a number of requirements including the following:

1. Capability of being applied to nickel-base superalloy foils, such as Inconel X-750.
2. No detrimental chemical interaction with the foil.
3. Satisfactory adherence over the entire operating temperature range.
4. Good flexural fatigue resistance.
5. Suitable frictional properties.
6. Good wear life.
7. Resistance to oxidative or thermal degradation in a high temperature oxidizing environment.

A number of protective films and coatings can be identified as possible candidates for consideration and evaluation for aircraft engine applications. These include:

- Conventional, high temperature, bonded, solid lubricant films
- Complex oxide systems, such as K_2MoO_4 or $NiWO_4$
- Thin, soft, noble metals such as gold or silver
- Hard, diffusion coatings, e.g., siliconizing, chromizing, etc.
- Glasses, such as $PbO.SiO_2$
- Hard coatings, e.g., chrome oxide or silicon nitride

Table VI lists some of the more promising candidates, ranked according to their effective temperature ranges.

The main drawbacks to the use of most of the high temperature solid lubricant coatings (those developed for use above 1000°F) is the fact that they are only effective at high temperatures. Below 1000°F, most of these films have poor friction and wear characteristics.

Previous experience in protecting the surfaces of rigid surface gas bearings from damage during sliding contacts has shown that hard coatings, such as chrome oxide or silicon nitride, are superior to any other type of surface treatment as far as resistance to surface damage or wear is concerned. For example, the results of gas bearing tests, which were reported in Reference 4, showed that chrome oxide coatings gave much better results than resin-bonded MoS_2 films. Conservatively, it is estimated that the wear life of a thin film of chrome oxide on the bearing surfaces will be at least ten times the life of the best conventional solid lubricant film, such as an epoxy-bonded MoS_2 coating.

TABLE VI
CANDIDATE LUBRICANTS AND PROTECTIVE COATINGS

<u>-65°F to 700°F</u>	<u>-65°F to 1000°F</u>	<u>To 1500°F</u>	<u>Above 1500°F</u>
Conventional MoS ₂ Bonded Films	Ceramic Bonded MoS ₂ -Graphite Films	BaF ₂ /CaF ₂ Eutectic Films	Evaluation of 1500°F Candi- dates
Hard Coatings. Chrome Oxide Tungsten Carbide	Gold Silver Hard Coatings: Chrome Carbide Metal-Bonded Chrome Oxide Silicon Nitride	Silicon Nitride Chromizing Siliconizing Siliciding	

However, hard coatings have inherently higher friction coefficients (0.25 - 0.35) than solid lubricant films. In addition, there is a question as to how the hard coatings will affect the flexibility of the foil substrate material. At the present time, it is believed that suitable hard coatings, with a thickness on the order of 0.0001", can be applied to foil surfaces by sputtering or ion-plating techniques. These coated foils should have sufficient flexibility to perform satisfactorily in bearing applications and still provide suitable wear life. Improvements could be made in the frictional characteristics of these hard coatings by applying a solid lubricant film over the coating and burnishing this film down to a minimal thickness. A study of the effectiveness of this approach of using solid lubricant films over hard, plasma-sprayed coatings has been completed (Reference 5).

It should be noted that previous evaluations of plasma-sprayed chrome-oxide at temperatures up to 1000°F have shown that coatings of this oxide, 0.002" to 0.003" thick, are subject to thermal spalling in the 1000°F range. This may not be the case when very thin, sputtered or ion-plated films are used. However, even if chrome-oxide is not suitable for use at 1000°F, silicon nitride should perform equally well as far as wear resistance is concerned.

In spite of the fact that sputtered films of these hard coating materials showed promise for this type of application, the scope of this program did not allow for the materials development effort that would be required to apply and evaluate these types of coatings. The program scope was limited to review and selection between commercially available solid lubricant coatings. This was a reasonable limitation for a preliminary study, since there was a good possibility that a solid lubricant film could satisfy the requirements of the tests and, in any event, such data would be useful in determining how much improvement would be necessary to operate foil bearings with a high degree of reliability.

Therefore, a search was made to select two of the most promising solid lubricant films that could be used, in air, at temperatures up to 1000°F. It was found that most of the currently available high temperature films contain MoS₂, MoS₂ and graphite, or MoS₂ and graphite plus certain soft metals. The binder phase is generally sodium silicate, silicone resins, or it is a proprietary material identified only as an organic or inorganic binder.

One promising exception to these MoS₂-based solid lubricant films is the fused calcium fluoride-barium fluoride coatings. For example, the Air Force has developed a film with the composition: calcium fluoride-barium fluoride - aluminum phosphate, which is designated as AFSL-28. This lubricant film is useful in air, at temperatures up to 1500°F. At temperatures below 1000°F, it will give relatively high friction, although it will still protect the surfaces. One of its major advantages is the fact that it was developed particularly for use on nickel-base alloys, such as Rene-41 or Inconel X-750. This fluoride film was one of the two coatings selected for evaluation in a foil bearing under start-stop and high-speed rub conditions.

The second choice for high temperature solid lubricant film was the NASA MLF-5 film which is a sodium silicate-bonded coating containing MoS₂, graphite and gold. This film has been studied extensively and is known to be effective over

a wide temperature range. The major drawback to this coating is the fact that 1000°F is above the oxidation limit for MoS₂ and, although this film has performed effectively at 1000°F in air, it should be anticipated that the useful life of the material may be limited.

4. Experimental Evaluation of Two Foil Journal Bearing Assemblies

Based on the results of the surveys of candidate substrate materials and surface coatings, which were described in the preceding sections, the following combinations were selected for experimental evaluation:

<u>Substrate Foil Material</u>	<u>Surface Coating</u>
Inconel X-750	Sodium silicate bonded MoS ₂ - Graphite-Gold film (MLF-5)
Inconel X-750	Fused calcium fluoride - barium fluoride - aluminum phosphate film (AFSL-28)

The Inconel X-750 foil material was purchased as annealed strip stock, six inches wide, and of the desired thicknesses. Because of the wide usage of this alloy as a high temperature spring material, it is available from a number of sources.

All of the solid lubricant coatings used in these tests were applied at the Midwest Research Institute. The extensive background and experience of that laboratory in the development and processing of these solid lubricant films made them one of the best sources for reliable and consistent coatings.

In the following subsections, the test equipment, test procedure, preparation of the foil bearings and the test results are summarized

a. Test Equipment

The experimental evaluations of the foil bearing materials were done by using the MTI Gas Bearing Materials Test Rig, which has been used previously for a number of high temperature gas bearing materials investigations, e.g., References 4 to 7. In this test rig, gas bearings, fabricated from the materials which are being investigated, are evaluated to determine the effect of sliding contacts on bearing performance. A photograph of the test rig is shown in Figure 51. A block schematic of the layout of this test rig is shown in Figure 52.

Essentially, this rig consisted of a support shaft mounted on two high-speed, preloaded, angular contact bearings. A Rene-41 heat dam is shrunk on one end of the support shaft outboard of the ball bearing mounts. The actual test shaft is shrunk on the other end of the Rene-41 heat dam and the whole assembly is pulled together with a Rene-41 tie bolt which is threaded into the support shaft. Only the overhung test shaft and the Rene-41 heat dam are in the hot zone. Heat baffles are used to interrupt the flow of heat out of this area. The entire support shaft and ball bearing system is enclosed in a water-cooled

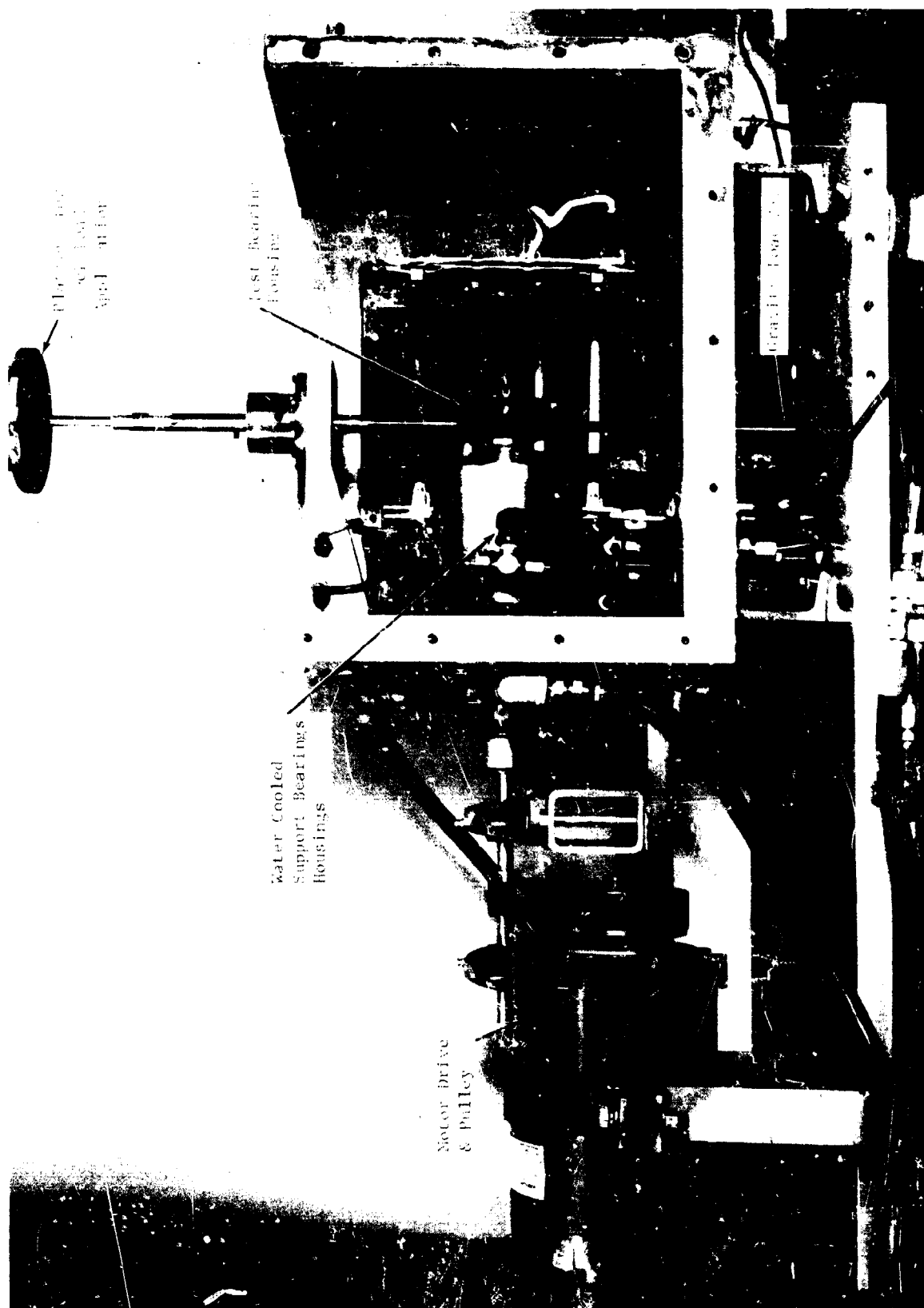


Fig. 51 Test Rig for 1000°F Start-Stop and High-Speed Rub Tests

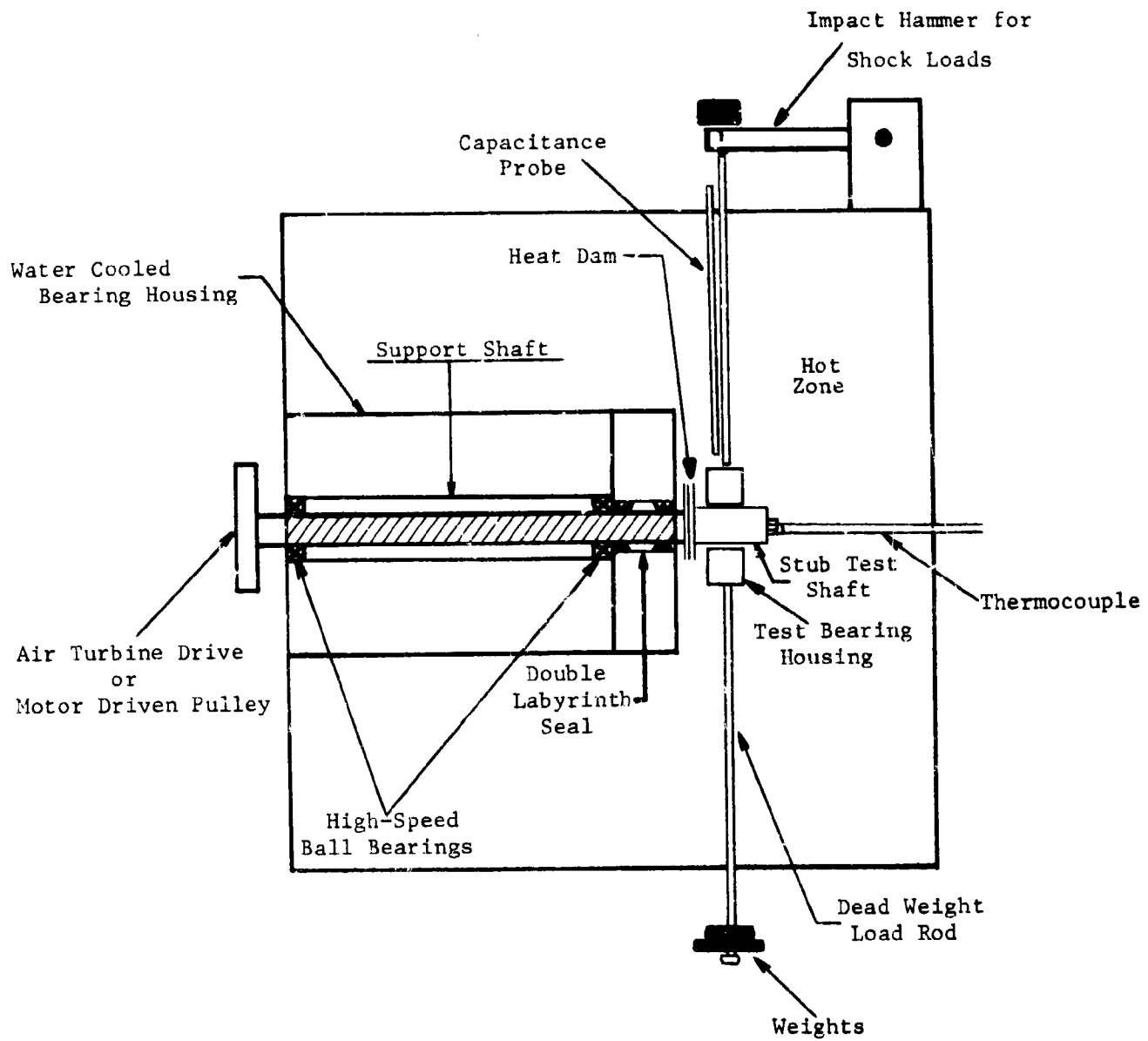


Fig. 52 Schematic of Bearing Test Rig

housing. Lubrication for the ball bearings is provided by an air-oil mist. Double labyrinth seals block the air-oil mist from traveling down the test shaft and into the test chamber. In addition, pressurized air is supplied between the double labyrinth seals at a higher pressure than the air-oil mist, thus creating an additional barrier to oil leakage into the hot zone.

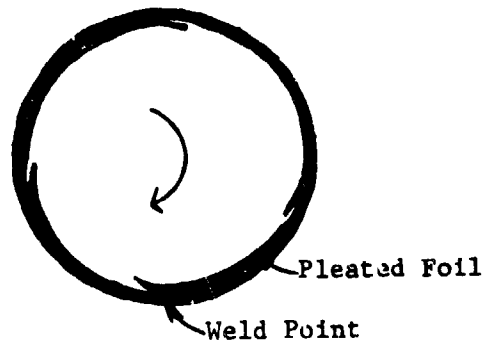
In the tests, the foil bearing was mounted in the bore of a steel housing and was spot-welded at one end as shown in Figures 53a to 53c. Where a double foil design was used, as shown in Figure 53b and 53c, the corrugated foil and the plain foil were both spot-welded at one end to the housing. Rotation of the shaft was in the direction of unloading the foil.

The test bearing was 1.5" in diameter and 1" long. Load was applied by means of dead weights suspended from a rod which was threaded into the bottom of the bearing housing. For the high-speed impact tests, a vertical steel rod, which penetrated the top of the furnace and was connected to the bearing housing through a pivot arrangement, was used to transmit the shock loads. To impose an impact load on the bearing, a known load was dropped from a known height by using a pivoted hammer assembly and impacting the top of the vertical steel rod. The rod transmitted the impact load to the housing and the test bearing.

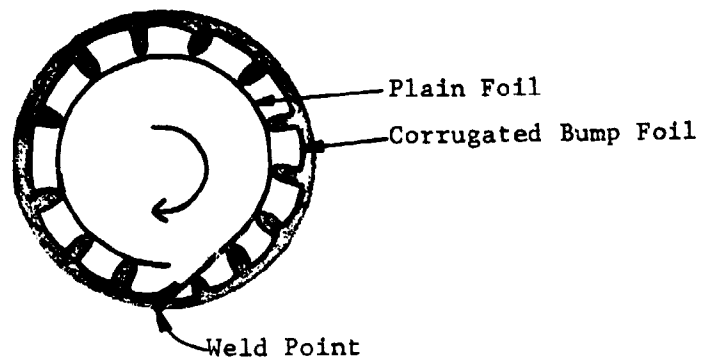
The shaft was driven by two systems, depending on the type of test. For the start-stop tests, an electric motor and belt driven pulleys were used to rotate the shaft at 13,800 rpm. In the high-speed tests, a compressed air turbine, mounted in a nozzle box on the cold end of the support shaft, provided the drive. A magnetic speed pickup located at the cold end of the support shaft, was used to measure shaft speed. This pickup produced six pulses for each revolution of the shaft. The signal from the pickup was fed into a Hewlett-Packard frequency meter for visual observation. The accuracy of the shaft speed readings was on the order of $\pm 2\%$.

Heat was obtained in the test area by means of a clam shell type furnace with quartz infrared heating tubes. A chromel-alumel thermocouple, located about 1/8" from the end of the test shaft was used to measure the temperature. It has been earlier determined that when the test rig is brought up to temperature and allowed to soak at that temperature for about thirty minutes, the temperature measured with the thermocouple is essentially the same as the bearing temperature within $\pm 20^\circ\text{F}$.

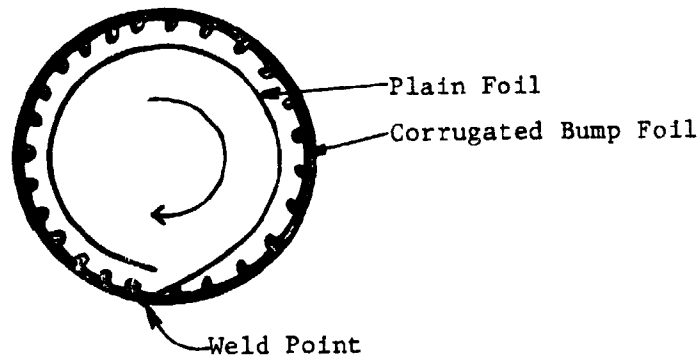
A high temperature capacitance probe was used to determine lift-off and running film thicknesses. This probe was located so as to look at the back of the bearing housing directly above the loaded zone. As the bearing lifts off the journal during start-up, the housing moves closer to the probe and the change in position can be read out on an oscilloscope.



a) Schematic of Pleated Foil Design



b) Double Foil Design with Wide Spacing
(0.25" Between Corrugations)



c) Double Foil Design with Close Spacing
(0.125" Between Corrugations)

Fig. 53 Foil Bearing Designs Evaluated in Bearing Tests

b. Test Procedure

The purpose of the tests was to determine the effect of sliding contacts on subsequent hydrodynamic bearing performance. To achieve this goal, the foil bearing was first subjected to 1000 starts and stops, in air, at 1000°F. A one horsepower electric motor and belt driven pulleys were used to rotate the shaft at 13,800 rpm. A cycle timer turned the motor on and allowed it to run for forty seconds. Then the timer shut the motor down for forty seconds. This start-stop cycle was repeated 1000 times. Lift-off speed and running film thicknesses were monitored to determine what changes, if any, were taking place in the performance of the bearing as a result of the sliding contacts during each start-up and coastdown.

Load was applied to the bearing by means of dead weights suspended from the bearing housing. The unit load was calculated by taking the product of the bearing length, 1.0", and the bearing diameter, 1.5", and dividing this value into the load:

$$(\text{Unit Load} = \frac{\text{Total Applied Load}}{\text{Length} \times \text{Diameter}})$$

Following the start-stop tests, the bearing was subjected to a predetermined series of high-speed rubs by bringing the shaft up to a stable operating speed of 35,000 rpm and applying impact loads of increasing magnitude to the bearing so that the bearing actually rubbed against the journal during each impact. Performance was evaluated by monitoring the shaft speed and the running film thickness. Failure was manifested by the fact that the bearing would not operate on a hydrodynamic film once the surface condition had deteriorated beyond a certain level. If the bearing was no longer operating on a hydrodynamic film, frictional drag would also result in a loss in speed because of the limited torque capability of the air turbine.

For the start-stop tests, the following procedure was used:

1. Assemble the foil bearing and photograph the surface.
2. Take surface profilometer readings on the bearing surface.
3. Assemble the bearing on the test shaft and measure the breakaway torque at preselected loads.
4. Run the test at room temperature for at least 20 cycles to determine that the bearing will support the 5 psi unit loading.
5. Heat the test rig to 1000°F and allow the temperature to stabilize for 30 minutes.
6. Measure the breakaway torque at the maximum unit load that the bearing will carry (up to 5 psi), without chatter or excessive torque.

7. Run the initial start-stop tests at this load.
8. After 500 cycles, cool the rig down and examine the bearing surface. Photograph.
9. Re-assemble the bearing and measure torque at room temperature.
10. Run 20 cycles at room temperature while monitoring the probe signals.
11. Heat the test rig to 1000°F and repeat Item 6, then run until the 1000 cycles have been completed or the bearing has failed.
12. Cool the test rig and measure the torque.
13. Disassemble the test rig, inspect the bearing, make surface profilometer readings and photograph the bearing surfaces.

For the high-speed rub tests, the procedure was as follows:

1. Assemble the foil bearing and photograph the surface.
2. Make surface profilometer readings of the bearing surface.
3. Assemble the bearing on the test shaft and measure the breakaway torque at preselected loads.
4. Run the bearing at room temperature and at a shaft speed of 35,000 rpm to determine if the bearing can operate satisfactorily.
5. Heat the test rig to 1000°F and allow the temperature to stabilize for thirty minutes.
6. Measure the breakaway torque to check preload and to be sure that the bearing is operable.
7. Bring the shaft up to a stable operating speed of 35,000 rpm and impact the bearing against the journal fifty times by dropping a one pound dead weight from a height of one inch against the top of the vertical steel rod which is coupled to the bearing housing. Monitor the probe signals to determine that the bearing actually contacts the journal during each impact. If the bearing is still operating satisfactorily, impact the bearing twenty more times by dropping a three pound dead weight from a height of one inch.
8. Stop the test, cool the rig and examine and photograph the bearing surface.
9. Re-assemble the bearing in the rig and run the bearing at room temperature at a shaft speed of 35,000 rpm to check performance.

10. Heat the test rig to 1000°F and allow the temperature to stabilize for thirty minutes.
11. Measure the breakaway torque to check preload.
12. Bring the shaft up to a stable operating speed of 35,000 rpm and impact the bearing against the journal ten times by dropping a five pound weight from a height of one inch. If the bearing is still operating satisfactorily, impact the bearing ten more times by dropping a seven pound weight from a height of one inch.
13. Stop the test, cool the rig and examine and photograph the bearing surfaces. Take profilometer measurements of the surfaces.

c. Preparation of Foil Bearing Specimens

The solid lubricant films used in these tests were sprayed as coatings, about 0.0015" thick, on one side of the foils after they had been grit-blasted to provide a good substrate surface for the coating. The sodium silicate bonded film was cured by air drying the film for thirty minutes and then curing at 180°F for two hours and 300°F for eight hours. The fluoride film was cured by fusion at 1697°F for one minute. No coatings were applied to the corrugated bump tapes.

Only the foil bearing surfaces were coated. For the journal surfaces, plasma-sprayed, nickel-chrome bonded chrome carbide coatings, 0.003" thick, were used throughout these tests. The hard-coated journal surface was ground and lapped to a finish of about 20 microinches CLA.

Three types of foil bearing designs were evaluated in these tests. These designs are shown schematically in Figures 53a through 53c. The foil used for the pleated foil design (Figure 53a) was 0.002" thick. In subsequent tests on the double foil design with the corrugated bump foil, the bump foil was 0.003" thick, while the plain bearing foil was 0.002" thick.

The analysis of foil bearing designs (Sections II and III of this report), which was completed after the tests described here had been run, showed subsequently that the use of a thicker plain foil, say 0.003 - 0.004" thick, would have given substantially higher load carrying capacity to the test bearing. Thus, the results of these bearing tests are on the pessimistic side.

The foil materials used in the tests on the pleated foil design, and on the corrugated double foil design with the wide bump spacing, were heat-treated by the following procedure:

- Solution treat at 2100°F for 2 hours
- Air cool

- Stabilize at 1550°F for 24 hours
- Air cool
- Precipitation treat at 1300°F for 20 hours
- Air cool

Some distortion of the foil occurred as the result of this heat treatment, but the material still appeared to be flexible enough to function properly as a bearing. The solid lubricant films were then applied to the bearing surfaces.

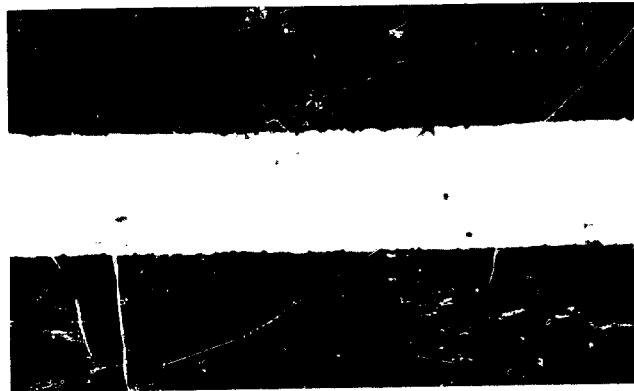
In subsequent tests, where the corrugated foils were made with close spacing between the corrugations, as shown in Figure 53c, the foil was used in the annealed condition "as received" from the vendor. Static load tests, which are described briefly in the next section, showed that the annealed, corrugated foil could support stresses up to at least 6 psi without undergoing permanent deformation. In these tests, Inconel X-750 foils in the annealed condition were used. Subsequent experience has shown that the foils should be age-hardened after forming, for greater strength.

Microstructures of the annealed foil materials are shown in Figures 54a and 54b. The photomicrographs in Figure 54a show fine grained "step" structures, typical of nickel-base alloys that have been work-hardened and annealed. The microstructures of the foils which were made after the start-stop tests had been run are shown in Figure 54b, which indicates that some incipient age-hardening had taken place because of the long time exposure of the foils to the 1000°F test temperature.

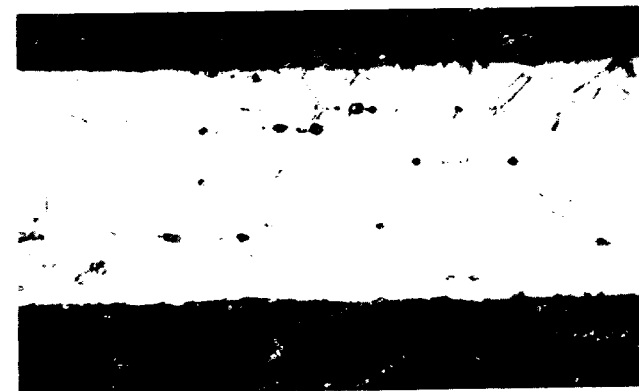
d. Results of the Start-Stop Tests

As noted in the preceding section, the first start-stop test was attempted with the pleated foil design shown in Figure 53a. The foil was formed in the annealed condition, heat-treated, and then coated with the MLF-5 film (sodium silicate-bonded MoS₂-graphite-gold). Before heating the test rig to 1000°F, the performance of this bearing was checked out at room temperature. After a few starts and stops at a stress level of 5 psi, it was found that the bearing would no longer lift off. The test rig was disassembled and the bearing was examined. The foil had cracked through at two points where the foil had been doubled over to form the pleats. Both of these points were in the loaded zone of the bearing.

The second start-stop test was made using the corrugated foil design shown in Figure 53b. In that design, the corrugations had a spacing of 0.25" between each corrugation. The bearing foil was formed, heat-treated, and then coated with the AFSL-28 film (fused CaF₂-BaF₂-aluminum phosphate). This bearing ran for a small number of starts and stops at 1000°F, with a stress of 5 psi on the bearing, but performance deteriorated rapidly. The test was shut down and the bearing was

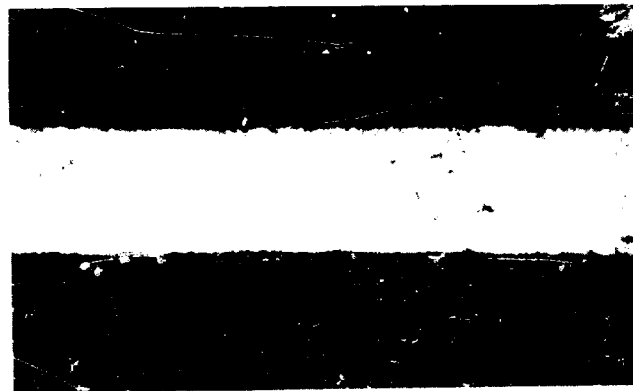


250X

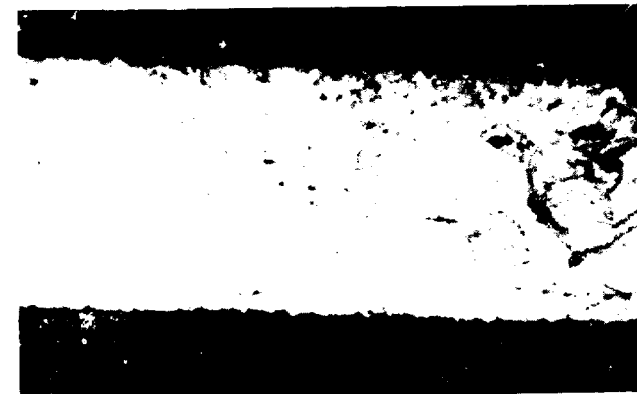


500X

(a) Annealed Inconel X-750 Foil Material "As Received"



250X



500X

(b) Annealed Inconel X-750 Foil Material After Start-Stop Tests at 1000°F, Showing Incipient Age-Hardening

Fig. 54 Microstructure of Annealed Inconel X-750 Foil Material

examined. It was found that the corrugations had partially collapsed in the loaded zone of the bearing.

The theoretical analysis that was being done at the same time that these tests were in progress confirmed the fact that the initially used spacing of 0.25" between corrugations provided poor hydrodynamic load carrying capacity and imposed high loads on the individual ridges. This condition was corrected by using a 0.125" spacing for the corrugations, as shown in Figure 53c. All subsequent tests were run with this close spacing geometry.

The results of the start-stop tests on the MLF-5 and AFSL-28 coated foils are summarized in Tables VII and VIII, and are discussed below:

MLF-5 Film

Initially, the MLF-5 film could not operate effectively at stress levels above 4 psi at the 13,800 rpm motor speed. At the end of the first 500 start-stop cycles, the bearing was removed from the test rig and was examined. It was found that considerable wear debris from the solid lubricant film had been trapped in the bearing clearance. This debris was brushed out. Photographs of this bearing are shown in Figures 55a through 55d. No deformation could be observed on the corrugated bump foil, as shown in Figure 55a. Figure 55b shows the appearance of the bearing at the weld joint. Figures 55c and 55d show the polished appearance of the bearing surface in the loaded zone.

The bearing was set up in the rig again and it was then able to support a unit load of 5 psi at the motor speed, without difficulty. The last 500 start-stop cycles were therefore run at a unit load level of 5 psi. Figures 56a through 56d show the appearance of the bearing at the end of the test. The corrugated bump foil was still in good condition (Figure 56a). Figure 56b shows the appearance of the bearing at the weld joint (the unloaded area), while Figures 56c and 56d show the polished appearance of the bearing in the loaded zone.

The lubricant film was extremely thin and highly polished in the loaded zone, especially at those points where the plain bearing foil was supported on the peaks of the corrugations. Although the film still appears to be intact, it is difficult to estimate how many more cycles it could tolerate before metal-to-metal contact started to take place.

The results of this test indicated that some sort of a run-in procedure should be used with the MLF-5 film to remove any excess solid lubricant coating before the bearing is put in service. Most of the debris is probably generated in the first 50 cycles.

AFSL-28 Film

At the beginning of the test, the AFSL-28 film could not be run at stress levels above 2.5 psi because of "stick-slip" chatter. After running at a stress level of 2.5 psi for 250 cycles, it was found

TABLE VII

SUMMARY OF START-STOP TEST RESULTS ON MLF-5 FILM

Initial Surface Roughness: 50 μ in. (Centerline average)
 Initial Torque Readings At: 4 psi 5 psi
 Room Temperature 2.4 inch pounds 3.6 inch pounds
 1000°F 2.4 inch pounds 2.4 inch pounds

Maximum Unit Load for Effective Operation: 4 psi

Ran 500 Cycles at 4 psi and 1000°F

Lift-off Clearances: 0.0002 to 0.0003"

Torque Readings After 500 Cycles:

	4 psi	5 psi
Room Temperature	3.6 inch pounds	3.6 inch pounds
1000°F	2.4 inch pounds	2.4 inch pounds

Appearance of Foil:

Foil polished in loaded zone, especially in areas where the corrugations backed the foil. There was a large quantity of wear debris in the unloaded zone which was apparently solid lubricant particles. These particles were cleaned out. No damage was visible on the foil bearing surface and the corrugations were not deformed.

Appearance of Journal:

A black film completely covered the contact area on the journal. This was smooth and polished.

After Reassembly the Data was:

Unit Load: 5 psi

Ran Cycles (500) at 5 psi and 1000°F

Lift-off Clearances: 0.0002-0.0003"

Torque Readings After 1000 Cycles (Total):

	1 psi	4 psi	5 psi
Room Temperature	3.6 in. lbs.	4.8 in. lbs.	7.2 in. lbs.
1000°F		2.4 in. lbs.	2.4 in. lbs.

Appearance of Foil:

Same as after first 500 cycles except that no further loose particles or solid lubricant or other debris were observed.

Shaft Appearance:

Same as after first 500 cycles.

Final Surface Roughness of Foil:

Polished contact areas 10 μ in.

Non-contacting areas 50 μ in.

TABLE VIII
SUMMARY OF START-STOP TEST RESULTS ON AFSL-28 FILM

Initial Surface Roughness of Foil: 40 μ in. (centerline average)

Initial Torque Readings:

	2.5 psi	4 psi	5 psi
Room Temperature	7.2 in. lbs.	*	*
1000 ^o F	2.4 in. lbs.	*	*

* Severe chatter, no measurements possible.

Maximum Unit Load for Effective Operation: 2.5 psi

Ran 250 Cycles at 2.5 psi and 1000^oF

Increased Stress to 4.0 psi Without Problems

Ran 340 Cycles at 4.0 psi and 1000^oF

Lift-off Clearances: 0.0001 - 0.0003"

Torque Readings After 590 Cycles:

	2.5 psi
Room Temperature	4.8 in. lbs.

Appearance of Foil:

Polished in loaded zone. Lubricant film appeared to be almost gone. There was no wear debris visible. The foil was crinkled near the weld in the unloaded zone.

Appearance of Journal:

The contact area on the journal was completely covered by a dark film. No damage could be observed.

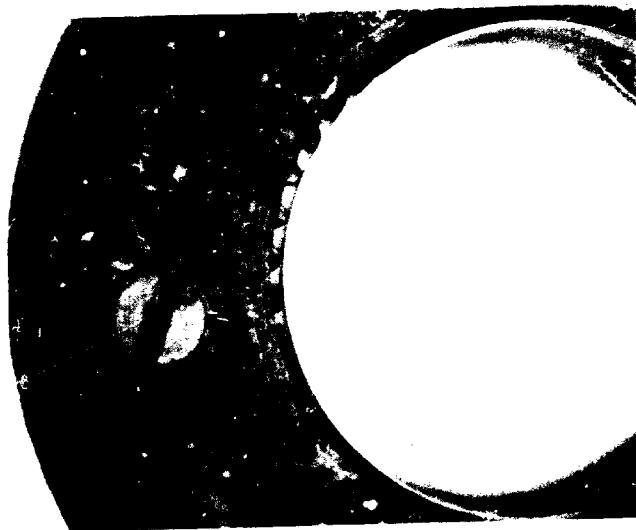
After reassembly, the bearing would not lift under a unit load of 4 psi and 1000^oF. After 20 cycles, the bearing would not lift under 2.5 psi. Cooled test rig. Checked torque readings at room temperature.

- 1.0 psi - 2.4 inch pounds
- 2.3 psi - 4.8 inch pounds
- 3.7 psi - Severe chatter
- 5.0 psi - Severe chatter

Reinspection of the bearing showed that the lubricant film appeared to be worn through in the loaded zone, and a circumferential scratch was noted on the foil, running from the loaded zone into the unloaded zone. The foil was crinkled near the weld in the unloaded zone. The contact area appeared to be shifting. No damage was visible on the journal.

Reassembled test and ran with 1 psi load at 1000^oF for balance of the 1000 starts and stops. Lift-off clearance 0.0001-0.0002". Final surface roughness of foil - 50 to 100 μ in. (CLA).

(Increased roughness due, in part, to local deformations of foil).



(a)



(b)

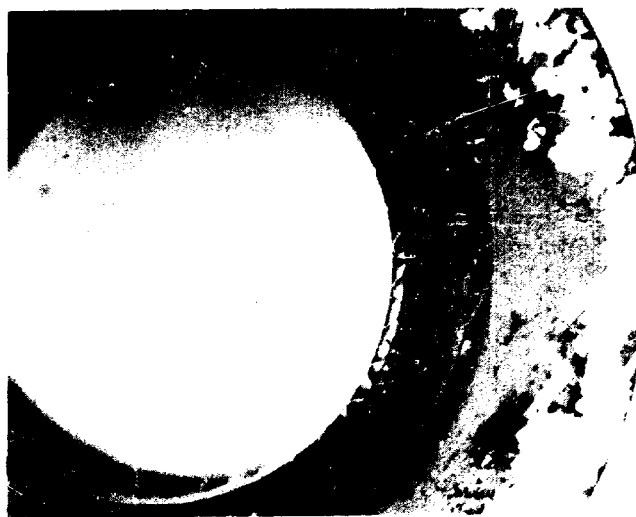


(c)



(d)

Fig. 55 Appearance of MLF-5 Coated Bearing After 500 Start-Stop Cycles at 4 psi Stress



(a)



(b)



(c)



(d)

Fig. 56 Appearance of MLF-5 Coated Bearing After 1000 Start-Stop Cycles.
500 Cycles at 4 psi and 500 Cycles at 5 psi Stress.

that the stress could be increased to 4 psi without any difficulty. Again, this demonstrated the importance of run-in. After running for 340 cycles at 4 psi, the test rig was shut down and the bearing was examined.

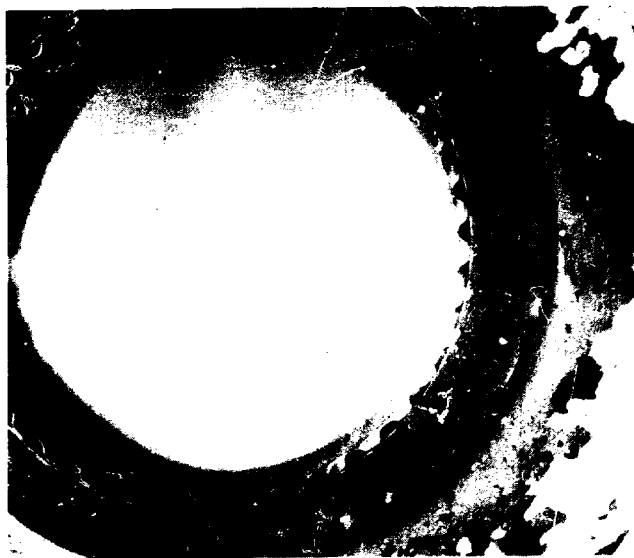
No deformation was observed on the corrugated bump foil (Figure 57a). However, it was found that the foil had crinkled near the weld in the unloaded zone. Figure 57b is a photograph of this area. Bearing in mind the fact that the shaft is rotating in the direction of unwinding the foil, it appears likely that the frictional drag between the foil and the shaft was enough to crinkle the foil at the weld point. As shown in Figures 57c and 57d, the surface was polished in the loaded zone. No loose wear debris was detected.

The bearing was reinstalled in the test rig and an attempt was made to continue the test at 4 psi. The bearing would not lift off, so the stress was reduced to 2.5 psi. After twenty cycles, the bearing was found to be continuously rubbing in contact. The test was again disassembled and the bearing examined. No lubricant film could be seen in the loaded zone, but it should be noted that the fluoride film is almost transparent when the coating has been worn to a thin film. A circumferential scratch could be seen on the foil running from the loaded zone into the unloaded zone, but the poor performance of the bearing appeared to be mainly due to the fact that the foil had crinkled at the weld. This conclusion was reinforced by the observation that the contact area appeared to be shifting after the bearing had been removed and reinstalled in the test rig.

The bearing was put back in the test rig and the balance of the 1000 starts and stops were run at a stress level of 1 psi. This was the maximum stress that the bearing would carry. Figure 58 shows the appearance of the bearing at this point.

Although the results of these start-stop tests indicate that the MLF-5 film does have lower friction and, apparently, better wear life than the AFSL-28 film, it should be noted that the crinkling of the foil with the AFSL-28 film was an obscuring factor. As noted previously, analytical studies had predicted that a thicker plain foil, say 0.003" to 0.004", would have given better load carrying capacity. If the thicker foil had been used in these tests, this problem of crinkling would certainly have been reduced or even eliminated. Thus, although the MLF-5 film is the better of these two solid lubricant coatings at temperatures not exceeding 1000°F, the difference is probably not as great as the test results indicate.

Post-test examinations of the corrugated annealed foils used in these start-stop tests disclosed that the MLF-5 coated bearing was still in good condition with no evidence of corrugation collapse. The corrugated foil used with the AFSL-28 film showed slight evidence of permanent deformation.



(a)



(b)

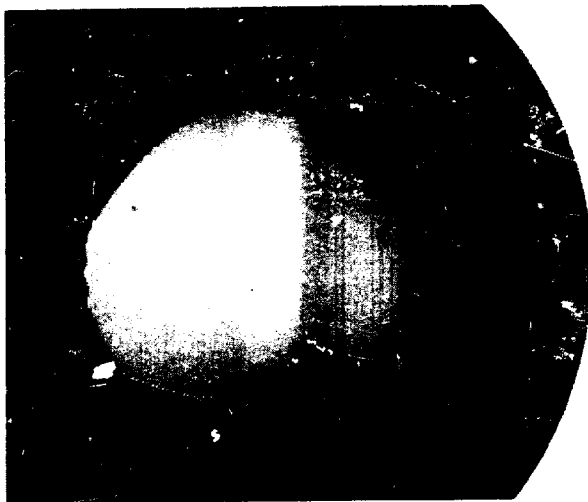


(c)



(d)

Fig. 57 Appearance of AFSL-28 Coated Bearing After 590 Start-Stop Cycles. 250 Cycles at 2.5 psi and 340 Cycles at 4 psi (note crinkling of foil at weld joint in Fig. 57b).



(a)



(b)



(c)



(d)

Fig. 58 Appearance of AFSL-28 Coated Bearing After 1000 Start-Stop Cycles.
250 Cycles at 2.5 psi, 340 Cycles at 4 psi and 410 Cycles at 1 psi.

e. Results of the High-Speed Rub-Tests

The results of the high-speed rub tests on the MLF-5 film and the AFSL-28 coating are summarized in Tables IX and X. Both coatings protected the bearings from failure through the full sequence of shock loads. Comparison of the results of these high-speed foil bearing tests with similar tests on rigid bearings showed that the resilience of the foil bearings is especially advantageous under the shock load conditions used in the tests, as the elasticity of the foil support serves to "cushion" the applied shock load.

MLF-5 Film

The MLF-5 coated bearing completed the first series of high-speed rubs without any difficulty. Examination of the bearing and journal at this point disclosed that there was considerable transfer of solid lubricant from the bearing to the journal surface, but no loose wear debris was observed. It was noted that the corrugated foil was slightly deformed in an arc about 15° to 30° beyond the point of maximum load.

The MLF-5 coated bearing survived the second series of high-speed rubs, although the torque increased and the probe signals at the end of the test showed that the bearing was making intermittent contact with the journal, even during steady-state operation. The final torque measurement was about twice the value obtained at the beginning of the test.

Initial examination of the foil surface, after the test, indicated that the coating had worn through at the high spots in the most heavily loaded area, but a more careful study of the surface showed that a film of gold was still present. Apparently, the gold powder which was dispersed in the solid lubricant film had been "plated" out on the bare spots on the surface during the high-speed rubs. Photographs of the bearing and journal surfaces are shown in Figures 59 and 60. The journal surface was covered with a thin, adherent and smooth layer of solid lubricant which had been transferred from the bearing surface. No damage was observed on the journal. Figure 61 shows the condition of the foil bearing surface in greater detail. The light areas in the photomicrographs are the points where gold has been friction plated on the bare metal surface. The higher torque values measured at the end of the test are apparently due to the fact that the solid lubricant film has been worn away at the contact peaks, and we are measuring the friction of the thin, plated film.

The corrugated bump foil was slightly deformed in the most heavily loaded zone.

AFSL-28 Film

The AFSL-28 coated bearing gave erratic torque values during the initial check-out, but operated satisfactorily at high speed. This bearing ran very well during the first series of high-speed rubs. However, when the test was disassembled, and the bearing was examined,

TABLE IX
TEST RESULTS ON MLF-5 FILM
USED IN HIGH-SPEED RUB TESTS

Initial Surface Roughness:

Foil	40-50 microinches (centerline average)
Journal	18 microinches (centerline average)

Initial Starting Torque Readings:

	<u>3 psi</u>	<u>4.7 psi</u>
1000°F	2.1 inch-pounds	3.4 inch-pounds

Ran at 35,000 rpm; 50 drops of one pound from one inch height, 20 drops of three pounds from one inch height. Bearing appears to be running well at speed; bearing lifts about one mil at 35,000 rpm. Coastdown time is 13 seconds. Spikes on scope indicate that bearing was touching lightly at speeds below 30,000 rpm. Torque at 1000°F was 2 inch-pounds at 3 psi. Cooled rig and inspected bearing. Considerable transfer of solid lubricant to journal, but no loose debris. Corrugations appeared to be slightly deformed in arc about 15 to 30 degrees beyond center of load point of bearing.

After reassembly, the starting torque at 1000°F was 3.2 inch-pounds at 3 psi.

Ran at 35,000 rpm; 10 drops of five pounds from one inch height and 10 drops of seven pounds from one inch height. Coastdown time was 14 seconds. Probe signals indicate that bearing was touching lightly, even at speed after these impacts. Final starting torque at 1000°F was 6 inch-pounds at 4.7 psi.

Final Surface Roughness:

Bearing Foil	- 28 microinches (centerline average)
Journal	- 20 microinches

The coating appeared to have worn through, but microscopic examination of the contact areas showed that a thin film of gold was preventing bare metal-to-metal contact.

TABLE X
TEST RESULTS ON AFSL-28 FILM
USED IN HIGH-SPEED RUB TESTS

Initial Surface Roughness:

Foil Bearing	50 microinches (centerline average)
Journal Surface	20 microinches (centerline average)

Initial Starting Torque Reading:

2.6 to 3.4 inch-pounds at 3 psi

Ran at 35,000 rpm; 50 drops of one pound from one inch height and 20 drops of three pounds from one inch height. Running film thickness was about one mil. Coastdown time was 18 seconds. Touchdown speed estimated at about 10,000 rpm after this test.

Cooled rig and inspected bearing. Surfaces appear to be in good condition, but corrugated backing foil is severely distorted, starting at the top of the loaded zone and extending through an arc of about 120 degrees.

After reassembly, the starting torque at 1000°F was 1.3 inch-pounds at 3 psi.

Ran at 35,000 rpm; 10 drops of five pounds from one inch height and 10 drops of seven pounds from one inch height. Probe signal indicates light. intermittent contact was taking place after these impacts. Coastdown time was 18 seconds. Touchdown speed estimated at about 12,000 rpm. Final torque at 1000°F was 1.7 to 2.1 inch-pounds at 4.7 psi.

Final Surface Roughness:

Foil Bearing	- 45 microinches (centerline average)
Journal	- 130 microinches (centerline average)

Considerable lubricant transferred from high spots on foil (over the corrugations) onto the journal.



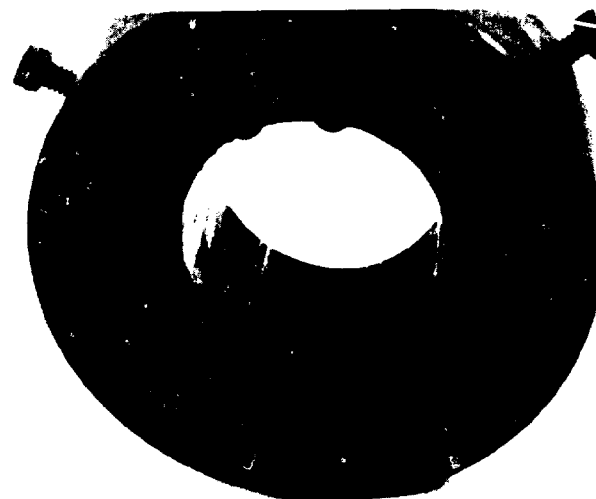
(a)



(b)



(c)



(d)

Fig. 59 Appearance of MLF-5 Coated Foil Bearing After High Speed Rub Tests



1.25X

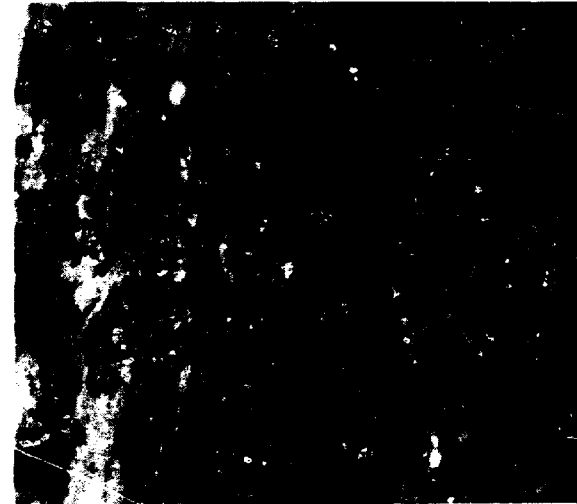
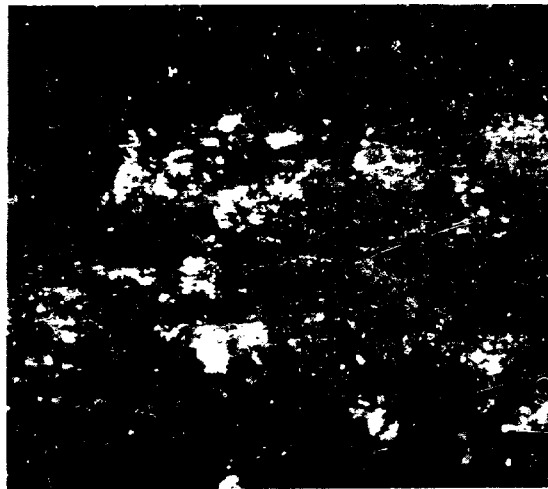


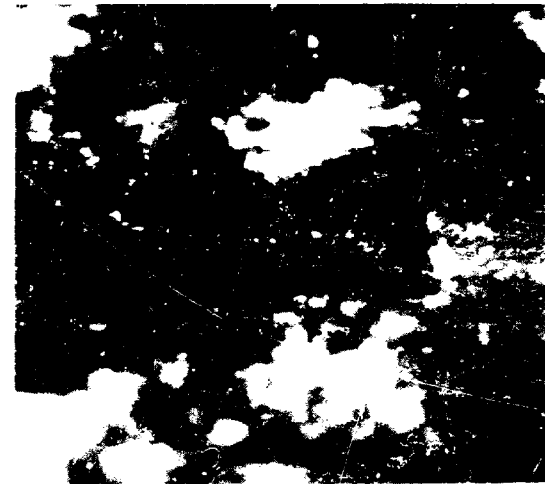
Fig. 60 Appearance of Journal Surface Used with MLF-5 Coated Foil After High Speed Rub Tests



5X



100X



500X

(White Areas are "Friction Plated" Gold)

Fig. 61 Photomicrographs of the Surface of the MLF-5 Coated Foil After the High Speed Rub Tests.

it was found that the corrugated bump foil had deformed, particularly in that section of the bearing which was just beyond the center of the load line through the bearing. Photographs of the foil bearing are shown in Figure 62.

When the bearing was reassembled in the test rig, low and consistent torque values were measured. The bearing survived the second series of high-speed rubs without any unusual problems. Again, some light, intermittent contacts were taking place through the gas film, even under stead-state conditions.

Photographs of the bearing and journal surfaces are shown in Figures 63, 64 and 65. There was considerable transfer of the fluoride solid lubricant film from the bearing to the journal surface. This transferred material was not smooth and continuous (as shown by the final surface roughness measurements), but rather was in the form of islands of material. No damage was observed on the journal surface underneath this transferred material.

It was difficult to determine if the lubricant on the foil bearing surface had worn through. The surface appeared to be smooth and glassy to the eye. Under the microscope, the film was discontinuous in the contact areas as though some of the material had spalled off the surface. One of these areas is shown in Figure 64. Since the fluoride film is transparent after it has been worn to a thin layer, and since the torque values were so low at the end of the test, it appears likely that most of the film was still intact.

The most significant conclusions that can be drawn from the results of these tests are as follows:

1. Foil bearings can survive a large number of start-stop cycles and high-speed rubs, even at fairly high stress levels. The foil bearing is particularly tolerant toward shock loads at high speeds.
2. A solid lubricant film is essential to reduce frictional drag and to minimize surface damage.
3. In these tests, the MLF-5 film was generally more effective under start-stop conditions, but the AFSL-28 film appeared to have a better wear life in the high-speed rub tests.
4. The annealed, corrugated Inconel X-750 bump foil was not strong enough to withstand the stresses to which it was subjected, especially during the high-speed rubs. It must be age-hardened to develop higher strength.

There is one other consideration which must be factored into the results. That is, the optimum thickness of the plain bearing foil. The results of the analytical work done in this program indicate

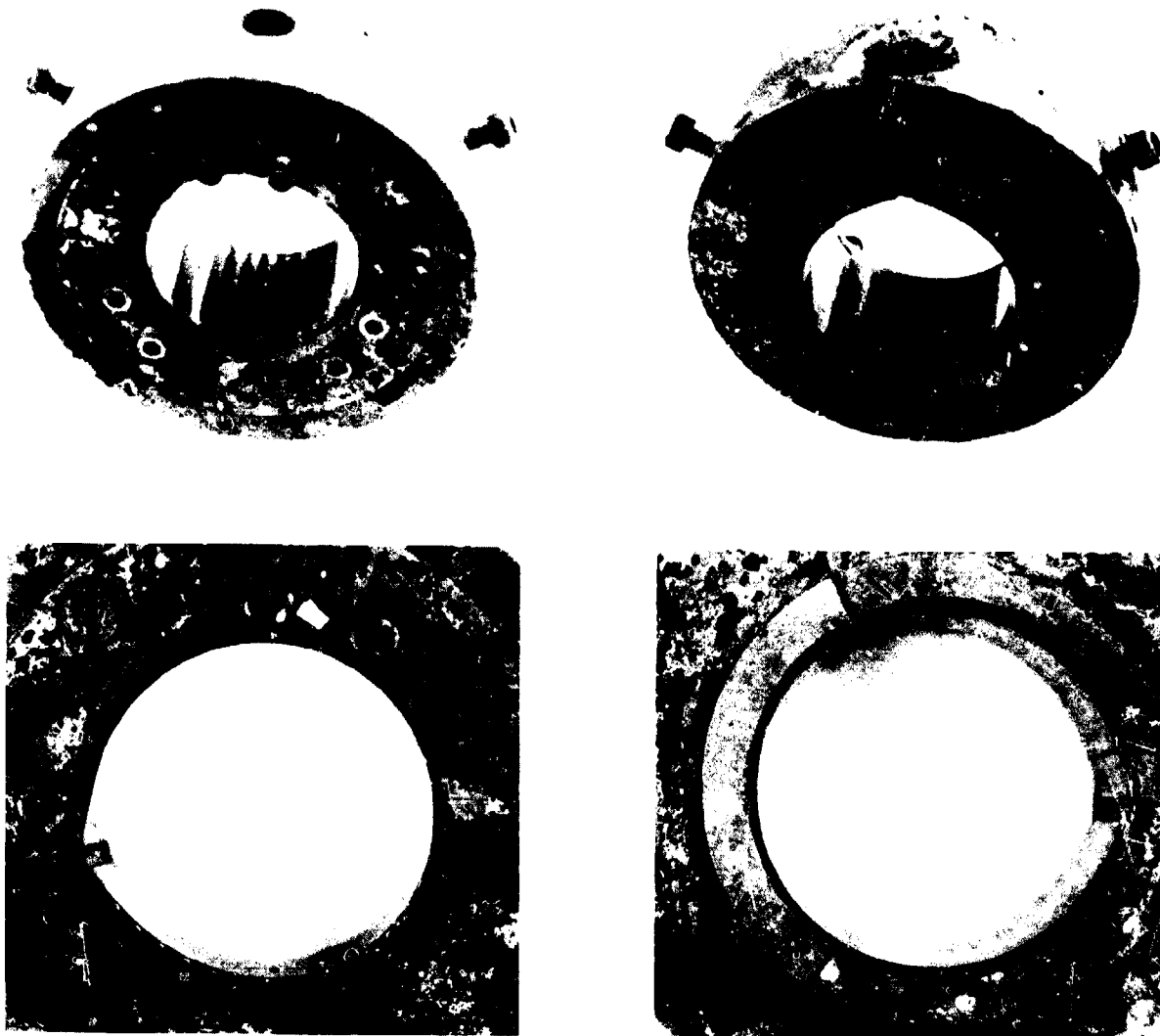


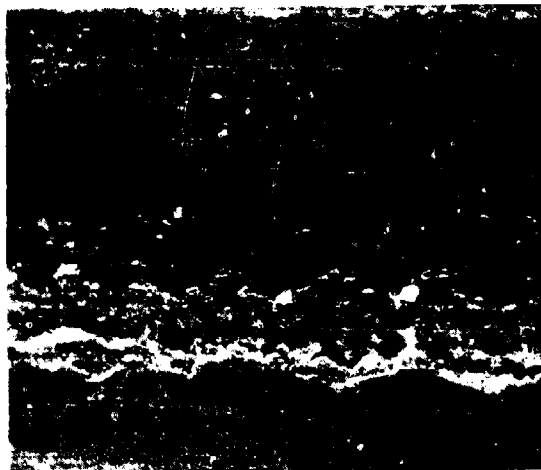
Fig. 62 AFSL-28 Coated Foil Bearing After First Series
of High Speed Rubs



Fig. 63 Appearance of AFSL-28 Coated Bearing After Complete Series of High Speed Rub Tests

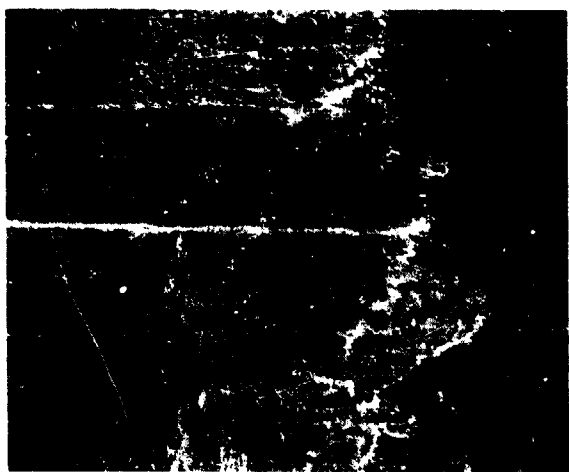


30X Mag

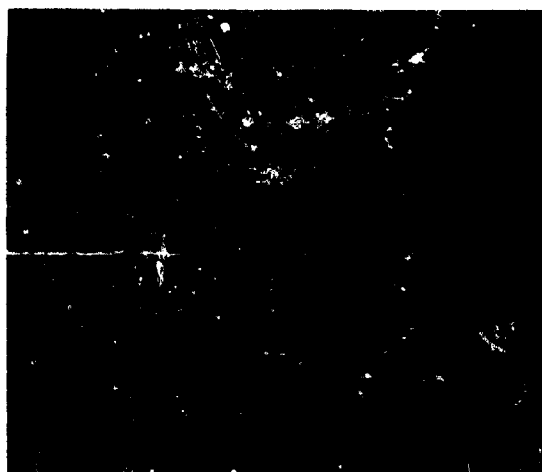


30X Mag

Fig. 64 Photomicrographs of AFSL-28 Coating on Foil Surface
After Complete Series of High Speed Rub Tests



100X



100X

Fig. 65 Appearance of the Journal Surface Which Was Run Against the AFSL-28 Coated Foil Bearing After Complete Series of High Speed Rubs

that a thicker foil than the 0.002" foil that was used in these tests would have given better performance as a hydrodynamic bearing. This is an important point because the use of a thicker foil would also decrease the possibility that the foil would crinkle at the weld point - as it did in the start-stop tests on the AFSL-28 film. In addition, a thicker foil would distribute the load more evenly over the bearing surface. In these tests, the wear on the plain foil took place at the points where the peaks of the corrugated bump foil actually contacted and supported the plain foil.

SECTION VI

DYNAMIC SIMULATOR TEST VEHICLE

As the technology of gas-lubricated foil bearings is advanced and such bearings are developed for eventual use in an advanced aircraft engine, they will need to be evaluated in a test vehicle that is a dynamic simulator of the engine.

Accordingly, as part of the foil bearing technology program, a layout and a facility schematic of a dynamic simulator test vehicle were prepared and are described in this section.

1. Selection of the Engine Model

Consultations were held with Teledyne CAE, to select an advanced engine (1980 and later time frame), that could serve as a model for the dynamic simulator test vehicle. In accordance with the requirements of the present technology program such an engine was to be in the 1000 lbs. to 3000 lbs. thrust class. Clearly also, the engine needed to be one that could be adapted for rotor support by means of air-lubricated foil bearings, and one that would benefit from such an adaptation.

The selected engine was an advanced gas generator under development by Teledyne CAE for the U. S. Air Force, for use in unmanned drone and missile craft. A derivative version incorporating a turbofan was also being studied at Teledyne CAE, for eventual use in manned aircraft.

For the purpose of the current program, Teledyne CAE prepared a layout of the advanced gas generator, incorporating air lubricated foil bearings. This layout of the engine (designated MAT-1), is shown in Figure 66. The performance characteristics of the engine are listed in Table XI for the drone or missile version and in Table XII for the manned turbofan derivative.

This engine was selected to serve as model for the dynamic simulation test rig as it offered several advantages, notably:

- It is in the thrust class defined in the RFP.
- It is an advanced engine, currently being developed for use in the 1980 and later time frame. Accordingly, there will be benefit of interaction between the future engine development work and the foil bearing technology effort.
- It is intended for use in powering unmanned, drone or missile craft, where low-cost considerations are especially important. Use of foil bearings is, potentially, a significant step in the low-cost engine area.
- Utilization of foil bearings is also, potentially, of considerable benefit in raising present limits on bearing operating temperature and greatly facilitate cooling requirements, particularly the turbine-end bearing location.

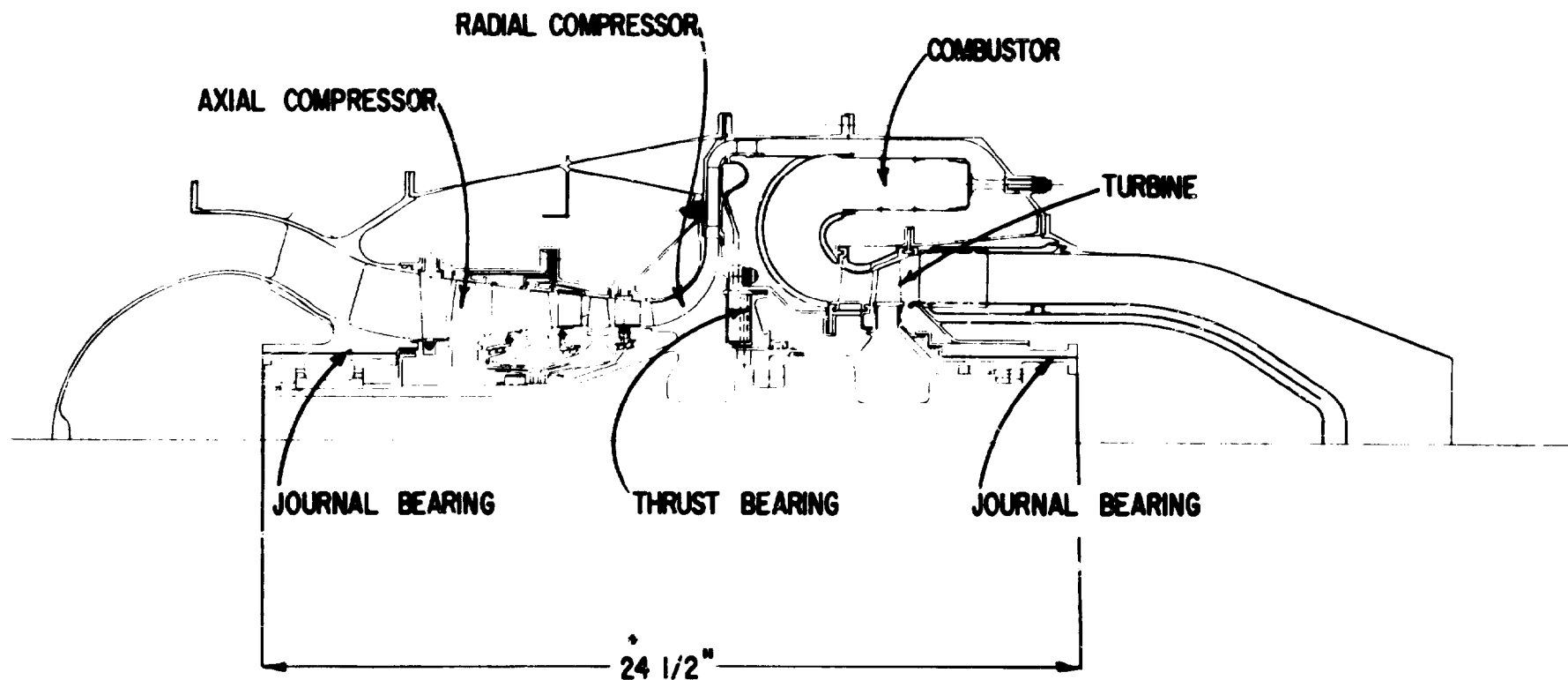


Fig. 66 Layout of MAT-1, Advanced Gas Generator Engine

TABLE XI
DRONE OR MISSILE MODEL ENGINE PERFORMANCE CHARACTERISTICS

	Sea-Level Static			Altitude 60,000 Feet		
	100% Speed	95% Speed	90% Speed	100% Speed	95% Speed	90% Speed
Rotor Speed, (rpm)	35,000	33,250	31,500	35,000	33,250	31,500
Compressor Inlet Pressure (psia)	14.7	14.7	14.7	1.51	1.51	1.51
Compressor Discharge Pressure (psia)	188.7	160.7	134.2	21.9	20.8	18.6
Thrust (lbs)	1259	1013	783.0	124	115	98
Turbine Inlet Temperature ($^{\circ}$ F)	> 2000	> 2000	> 1900	> 2000	> 2000	> 1900
Compressor Discharge Temperature ($^{\circ}$ F)	766	680	619	615	585	535

TABLE XII
TURBOFAN DERIVATIVE (MANNED) ENGINE PERFORMANCE CHARACTERISTICS

	Sea-Level Static			Altitude 45,000 Feet		
	100% Speed	95% Speed	90% Speed	100% Speed	95% Speed	90% Speed
Rotor Speed, (rpm)	35,000	33,250	31,500	35,000	33,250	31,500
Compressor Inlet Pressure (psia)	14.7	14.7	14.7	3.12	3.12	3.12
Compressor Discharge Pressure (psia)	234	191	156	60	56	47
Thrust (lbs)	2630	2083	1627	448	399	313
Turbine Inlet Temperature (^o F)	> 2000	> 2000	> 2000	> 2000	> 2000	> 1900
Compressor Discharge Temperature (^o F)	820	748	683	722	670	620

- Turbofan derivatives of this engine, which are under study at Teledyne CAE, will provide a framework for assessing foil bearing applications in two-spool arrangements.

As part of their consultation support under the current program, Teledyne CAE provided the data on the rotor characteristics of the engine, they also determined the bearing loads in accordance with the specifications of MIL E-5007C and computed the critical speeds of the rotor-bearing system for support stiffnesses ranging from 10^4 to 10^6 lbs/inch.

Specifically, the rotor characteristics of the MAT-1 engine are as follows:

Shaft Weight	99.2 lbs.
Bearing Span	20.2 inches
Shaft Mass Polar Moment of Inertia	1.7 lb-in-sec ²
Shaft Mass Transverse Moment of Inertia about CG	7.7 lb-in-sec ²
Shaft CG (From Centerline of Front Bearing)	11.1 inches
Maximum Engine Speed	35,000 rpm

The calculated critical speeds of the engine's rotor-bearing system are listed below for the range of bearing stiffnesses.

Bearing Stiffness (lbs/inch)	1st Critical Speed (rpm)	2nd Critical Speed (rpm)	3rd Critical Speed (rpm)
10^4	3000	5000	51,000
10^5	8000	16,000	54,000
10^6	18,000	45,000	78,000

The stiffness of the air-lubricated foil bearings will depend on the size and geometry selected. However, for 5" diameter journal bearings, this stiffness will be in the range of 10^5 to 10^6 lbs/inch. Thus, the 35,000 rpm maximum operating speed of the MAT-1 engine is well below the third critical speed of the rotor-bearings system. However, the above data also shows that, over the range of bearing stiffnesses from 10^5 to 10^6 lbs/inch, the second critical speed may be below, coincident with or above the second critical speed of the rotor-bearings system. Thus, in the actual application, the bearing stiffness would have to be adjusted to maintain an adequate margin (say about 25 percent) between the operating speed and the critical speed. This could be done, for example, by proper selection of foil compliance, bearing clearance and preload.

Table XIII shows the compressor inlet and outlet conditions for three flight conditions for both the turbojet engine concept and its turbofan derivative. The design provides for maintaining the journal and thrust bearing cavities at compressor discharge conditions. The last two columns of Table XIII show, respectively, the calculated thrust load and the corresponding unit loading of the foil thrust bearing. These thrust loads were calculated for the current gas generator design condition (sea level flight, CDP = 189 psia), based upon the predicted pressure distributions. The loads for the other conditions were ratioed as the geometry for the turbofan derivative and the pressures at all

TABLE XIII
CALCULATED COMPRESSOR INLET AND
OUTLET CONDITIONS AND THRUST BEARING LOADING

MAT-1 TURBOJET ENGINE

Altitude Ft.	Mach No.	Speed %	Temp. °F	Press. psia	Temp. °F	Press. psia	Bearing Thrust Load lb.	Equivalent* Thrust Bearing Unit Loading psi
Sea-Level	0	100.	59	14.7	766	188.7	1020	33
		95.	59	14.7	680	160.7	870	28
		90.	59	14.7	619	134.2	720	23
		50.	59	14.7	218	29.5	160	5
	.6	100.	100	18.7	780	218.	1180	38
60,000	0.75	100.	-22	1.51	615	21.9	120	4
		95.	-22	1.51	585	20.8	110	4
		90.	-22	1.51	535	18.6	100	3

TURBOFAN DERIVATIVE OF MAT-1 ENGINE

Sea-Level	0	100.	59	14.7	820	234.	1260	41
		95.	59	14.7	748	191.	1030	33
		90.	59	14.7	683	156	840	27
		50.	59	14.7	350	37	200	6
	.6	100.	100	18.7	850	260	1400	45
45,000	0.75	100.	-26	3.12	722	60	320	10
		95.	-26	3.12	670	56	300	10
		90.	-26	3.12	620	47	250	8

* Based on an effective thrust bearing area of 31 square inches.

points throughout the engine were not available when this data was generated.

The calculated thrust bearing unit loadings on the turbojet engine are everywhere below 40 psi and constitute a challenging but reasonable technology target for the 1980 time frame. The turbofan derivative engine concept has slightly higher thrust loads, but the engine design allows room for increased thrust bearing area, if it is needed.

Table XIV shows the calculated journal bearing loads for level flight, landing and 3.5 radians/second yaw maneuver. The present design allows for 5" diameter journal bearings. Thus, the start-up unit loads are very small, about 2 psi, assuming a length-to-diameter ratio of unity. The most critical condition, is that associated with the 3.5 radians/second yaw maneuver. Under this condition, the maximum bearing load would approach 50 psi which is two to three times the unit loading demonstrated to date on foil bearings. Although this is only a short duration maneuver load, it indicates that some increase in bearing span, to reduce the bearing reactions to the gyroscopic moment induced in the maneuver, as well as an increase in bearing size may need to be considered, with the objective of reducing the maximum radial unit loading to about 35 psi.

In summary, the Teledyne CAE MAT-1 engine is a reasonable choice for use as a model for the dynamic simulator test rig. The brief reviews of such an engine supported on air-lubricated bearings indicates that the priority in air-lubricated, foil bearing development needs to be directed primarily at enhancing load capacity at least to the point where safe operation under unit loads of 35 to 40 psi can be demonstrated. Materials suitable for continuous operation at up to 1000°F will be adequate.

2. Dynamic Simulator Layout

The layout drawing of the dynamic simulator, modelled after the MAT-1 engine was prepared and is shown in Figure 67.

Actually, this figure shows two distinct rotor configurations, with a common housing. The rotor detail shown above the axial centerline in Figure 67 is the dynamic simulator rotor. The rotor detail shown below the axial centerline is a simpler rotor that is fully rigid over the operating speed range. It has the same overall dimensions and bearing span as the dynamic simulator rotor, is interchangeable with it and is accommodated by the same housing.

A simple, fully rigid rotor is desirable for initial evaluation of air-lubricated foil journal and thrust bearings without introducing rotor-dynamic influences. This is accomplished most economically, by using the same test facility and test housing that would be eventually used in the dynamic simulator tests, but conducting the early bearing tests with a simpler configuration.

In this context, the layout drawing shows pressurized gas provisions at each of the two journal bearings so that hydrostatic bearings could be used at first, for test rig de-bugging purposes and during initial tests of the double-acting, air-lubricated foil thrust bearing. Subsequently, the air-lubricated foil journal bearings could be tested with the rigid rotor. The final testing would then be conducted with the foil journal and thrust bearings, using the dynamic simulator rotor so as to include rotor-dynamic effects.

TABLE XIV
JOURNAL BEARING LOADS IN MAT-1 TURBOJET ENGINE

	Front Bearing Load (lbs.)	Rear Bearing Load (lbs.)
<u>Level Flight</u>		
Non-Rotating Load Component:	45	55
Rotating Load Component:	100	100
<u>Landing</u>		
Non-Rotating Load Component:		
10G Down = 992.2 lb.	447	545.2
2G Side = 198.4 lb.	89.6	108.8
$\pm 14 \text{ Rad/sec}^2$ Pitch Acceleration = 107 in/lb.	5.31	5.31
$\pm 6 \text{ Rad/sec}^2$ Yaw Acceleration = 46 in/lb.	2.28	2.28
Vertical	452	550
Horizontal	<u>91.9</u>	<u>111</u>
Combined	461	560
Rotating Load Component:	100	100
<u>3.5 Radians/second Yaw Manuever</u>		
Non-Rotating Load Component:		
Gyro: 3.5 Rad/sec in Yaw = 21,806 in/lb.	1081	1081
$\pm 1G$ Vertical = 99.21 lb.	<u>45</u>	<u>55</u>
Combined	1126	1136
Rotating Load Component:	100	100

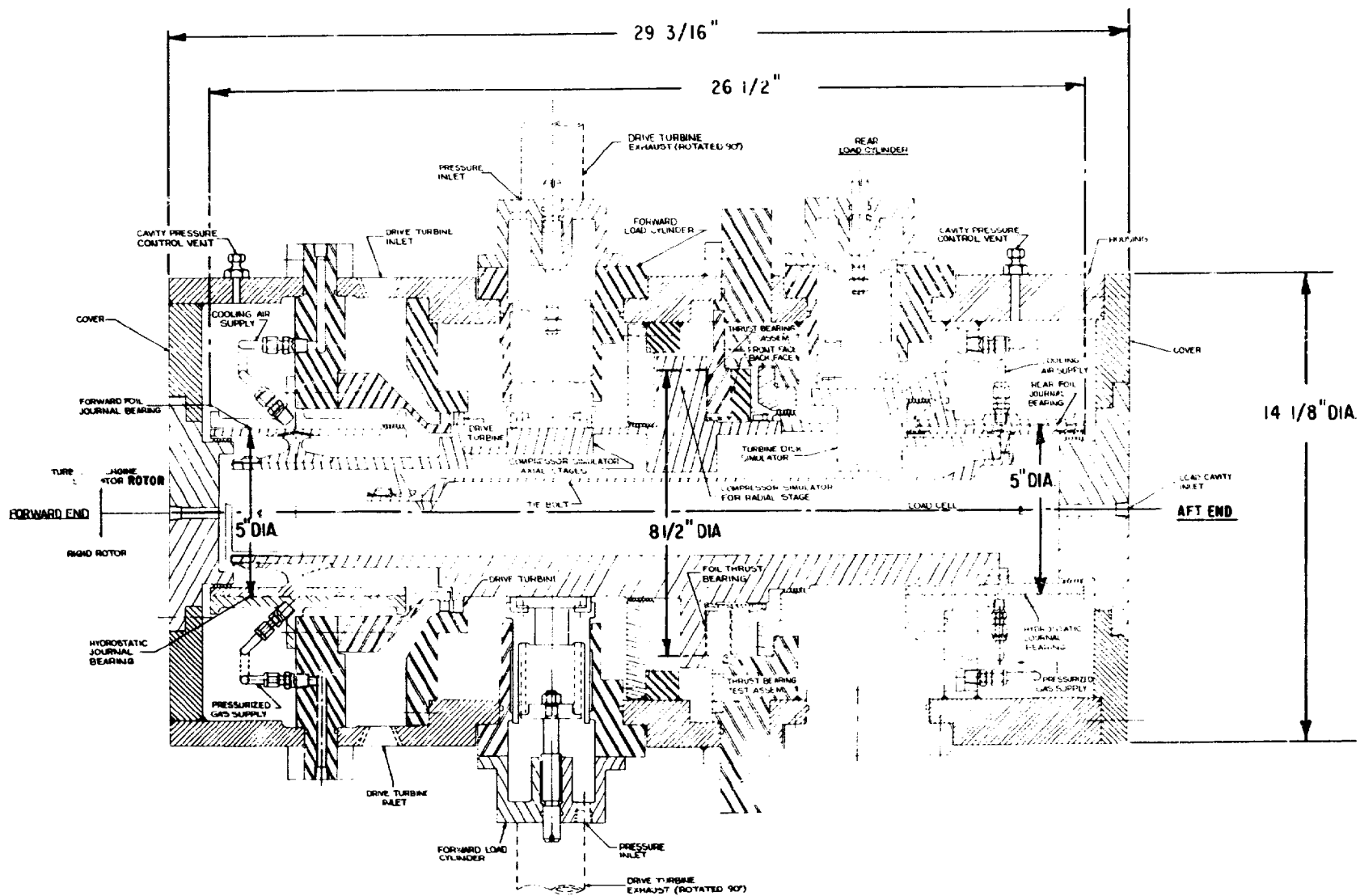


Fig. 67 Layout of Dynamic Simulator

The diameters of journal and thrust bearings shown in the test vehicle drawing (Figure 67), are the same size as those that could be accommodated by the MAT-1 engine. Each of the two journal bearings is 5 inches in diameter, and can be up to 5 inches long. The dimensions of the double-acting thrust bearing are 8-1/2 inches maximum O.D. by 5- 7/8 inches minimum I.D. Each bearing cavity is sealed by means of labyrinths at each end to permit its operation over a range of ambient pressures.

In the actual engine, convective cooling of the bearings using leakage flow from the compressor will be required to control the bearing temperature and temperature gradients. This is necessary to minimize thermally induced changes of bearing clearance. In the test vehicle illustrated in Figure 67, cooling air provisions are shown for each bearing cavity.

Radial loading is applied by means of pneumatic load cylinders inboard of each journal bearing. Thrust load application is achieved pneumatically by applying a pressurized gas source to the rotor bore (designated "load cell" in the drawing), through the inlet at the aft end of the housing.

The rigid rotor is essentially a single piece construction, with the two journals attached to the center portion of the rotor. The forward journal is piloted and shouldered against the rotor and held in place by the lock-nut. The aft journal is piloted and bolted to the rotor.

The dynamic simulator rotor is fabricated in three sections. These sections are piloted and shouldered and are retained by means of the tie-bolt as illustrated. The forward section of the rotor has the air turbine drive and the forward journal. As in the case of the rigid rotor, this forward journal is piloted and shouldered and it is retained by the lock-nut. The center rotor section has the masses simulating the axial compression stages and the radial compression stage. It also has the double-acting thrust faces. The aft rotor section has the turbine disk simulator mass. The aft journal bearing is again shouldered and bolted to this rotor section.

3. Dynamic Characteristics of the Simulator Rotor

The shaft mass and stiffness distributions in the dynamic simulator follow quite closely those of the rotor of the MAT-1 engine. However, the bearing span of the dynamic simulator rotor was made one inch longer than that of the engine rotor in order to insure that journal bearings having a length-to-diameter ratio of unity could be accommodated. Table XV on the next page shows the weights and geometrical characteristics of the two rotors.

The dynamic simulator is seen from Table XV to be heavier and to have larger moments of inertia than the engine. In part, these differences are due to the fact that the engine design emphasized minimum weight. The test vehicle design, on the other hand, places greater emphasis on ease of disassembly and reassembly, as well as accessibility of bearings and instrumentation.

TABLE XV
COMPARISON OF ROTOR CHARACTERISTICS

	MAT-1 ENGINE ROTOR	DYNAMIC SIMULATOR ROTOR
Rotor Weight (lbs)	99.2	109.2
Bearing Span (inches)	20.2	21.3
Shaft mass polar moment of inertia (in-lb-sec ²)	1.7	2.9
Shaft mass transverse moment of inertia about C.G. (lb-in-sec ²)	7.7	12.7
Distance from centerline of forward bearing to the center of gravity of the rotor (inches)	11.1	12.2

Figure 68 shows the modeling of the dynamic simulator rotor that was generated for the calculation of critical speeds and unbalance response. Figure 69 shows the calculated critical speed map of the rotor-bearings system of the dynamic simulator rotor. The stiffness of each of two 5 inches diameter air-lubricated, foil journal bearings will be in the range of 10^5 to 10^6 lbs/inch, depending on the bearing type and foil geometry used in each test. Thus, as seen from Figure 69, the full operating speed range up to 35,000 rpm is well below the third critical speed.

Table XVI compares the calculated rotor-bearing system critical speeds of the dynamic simulator and of the MAT-1 engine, for the two limiting values of bearing stiffness. One difference between them is that the third critical speed of the engine rotor is more sensitive to bearing stiffness, than is that of the dynamic simulator. However, this third critical speed is well above the operating speed range.

TABLE XVI
COMPARISON OF CALCULATED CRITICAL SPEEDS

Bearing Stiffness (lbs/inch)	1st Critical Speed (rpm)		2nd Critical Speed (rpm)		3rd Critical Speed (rpm)	
	MAT-1	Simulator	MAT-1	Simulator	MAT-1	Simulator
100,000	8000	7700	16000	13800	54000	65000
1,000,000	18000	21000	45000	45000	78000	77000

Figures 70 and 71 show the calculated dynamic response of the system at the two

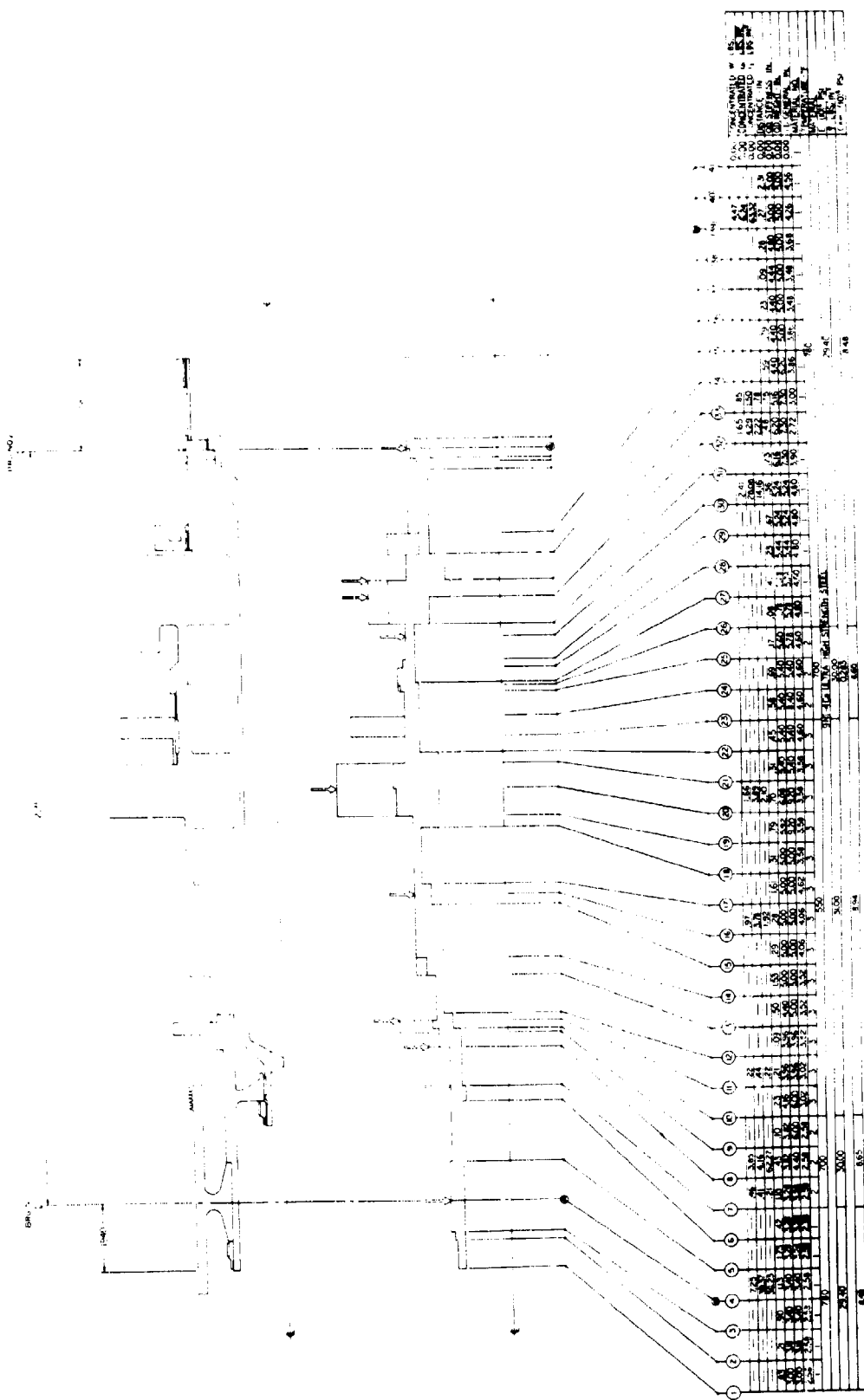


Fig. 68 Model of Dynamic Simulator

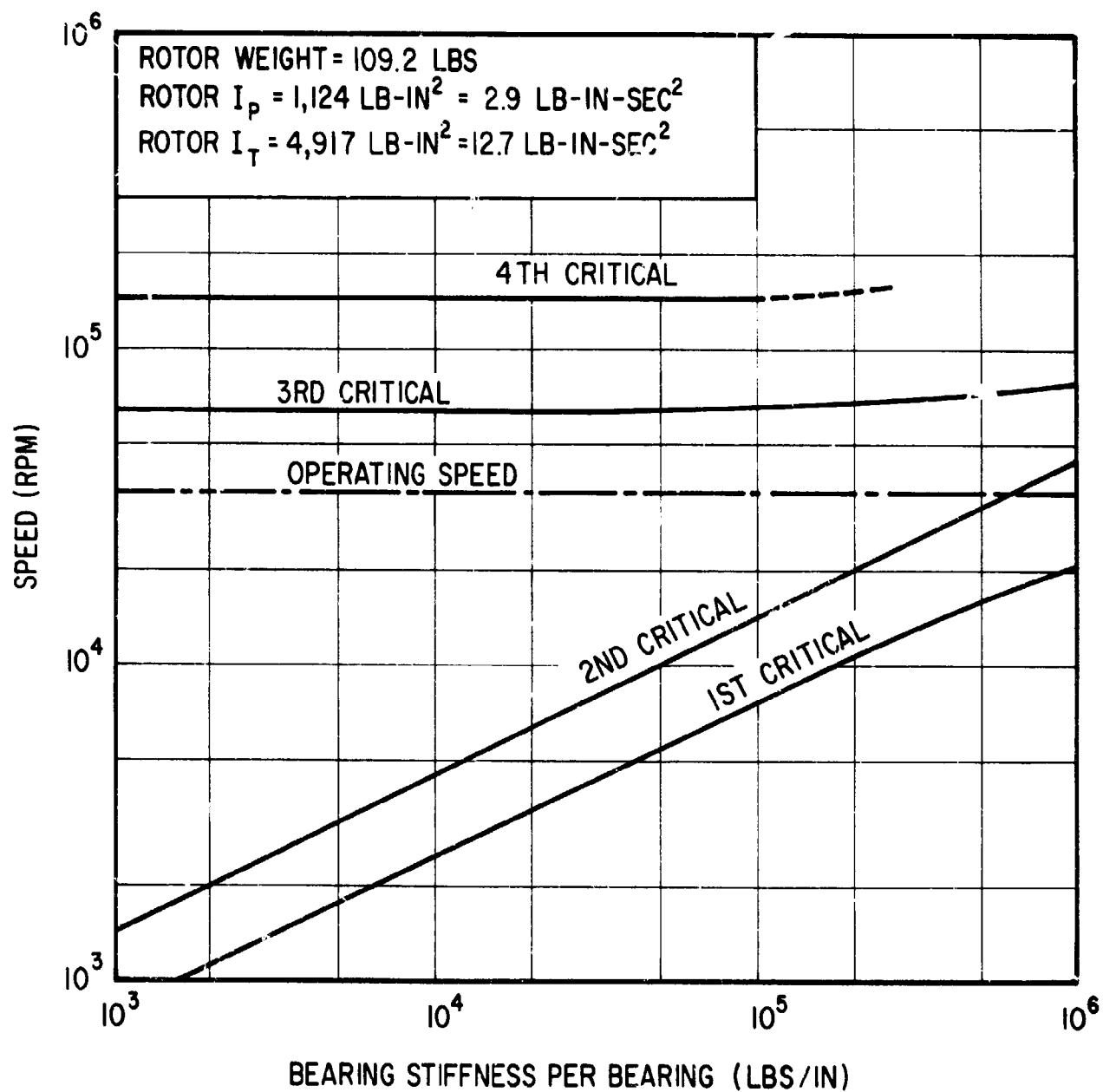


Fig. 69 Calculated Critical Speed Map of the Dynamic Simulator Rotor

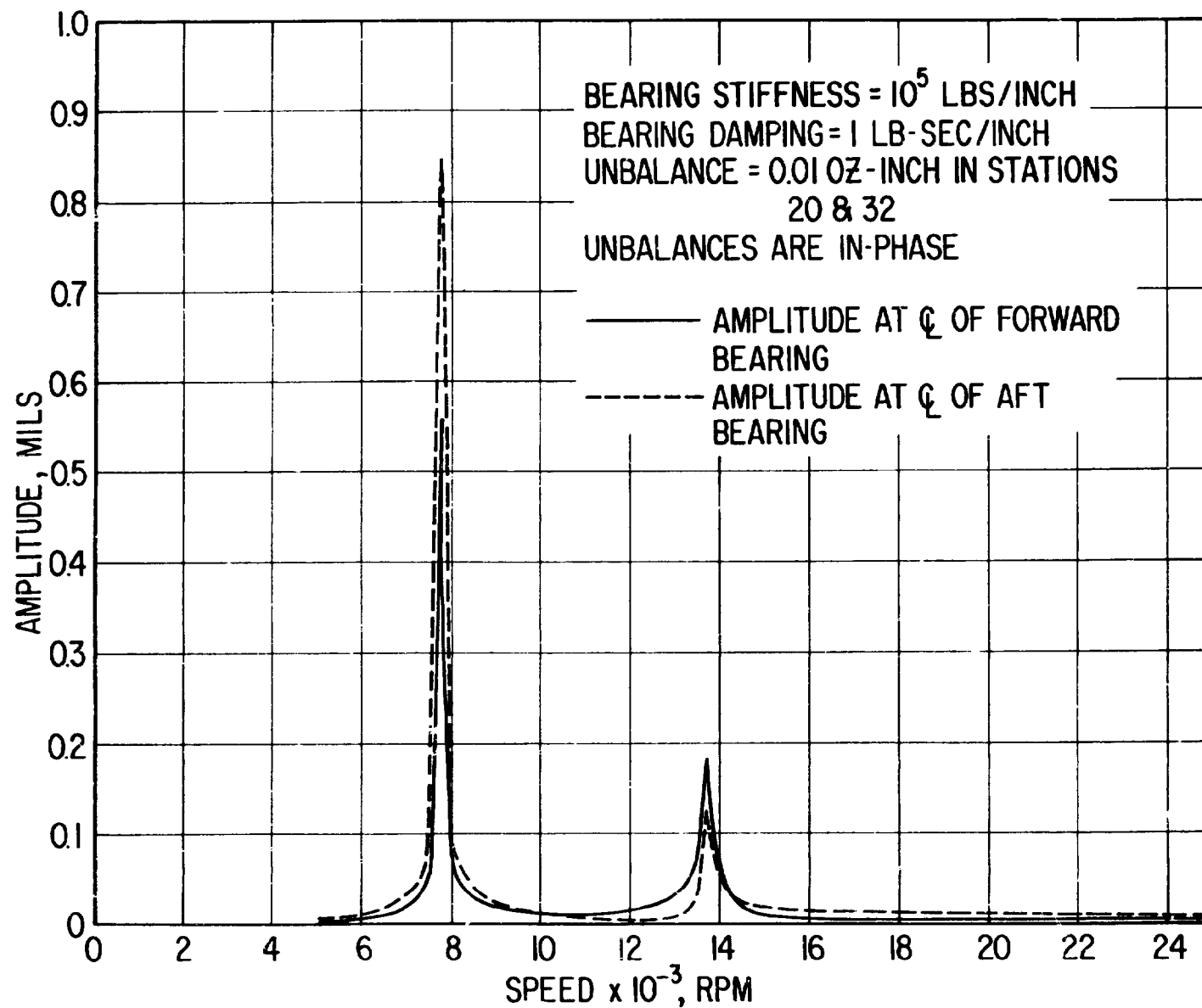


Fig. 70 Calculated Unbalance Response on the Dynamic Simulator Rotor,
 for 0.02 ounce-inches, Total Static Unbalance

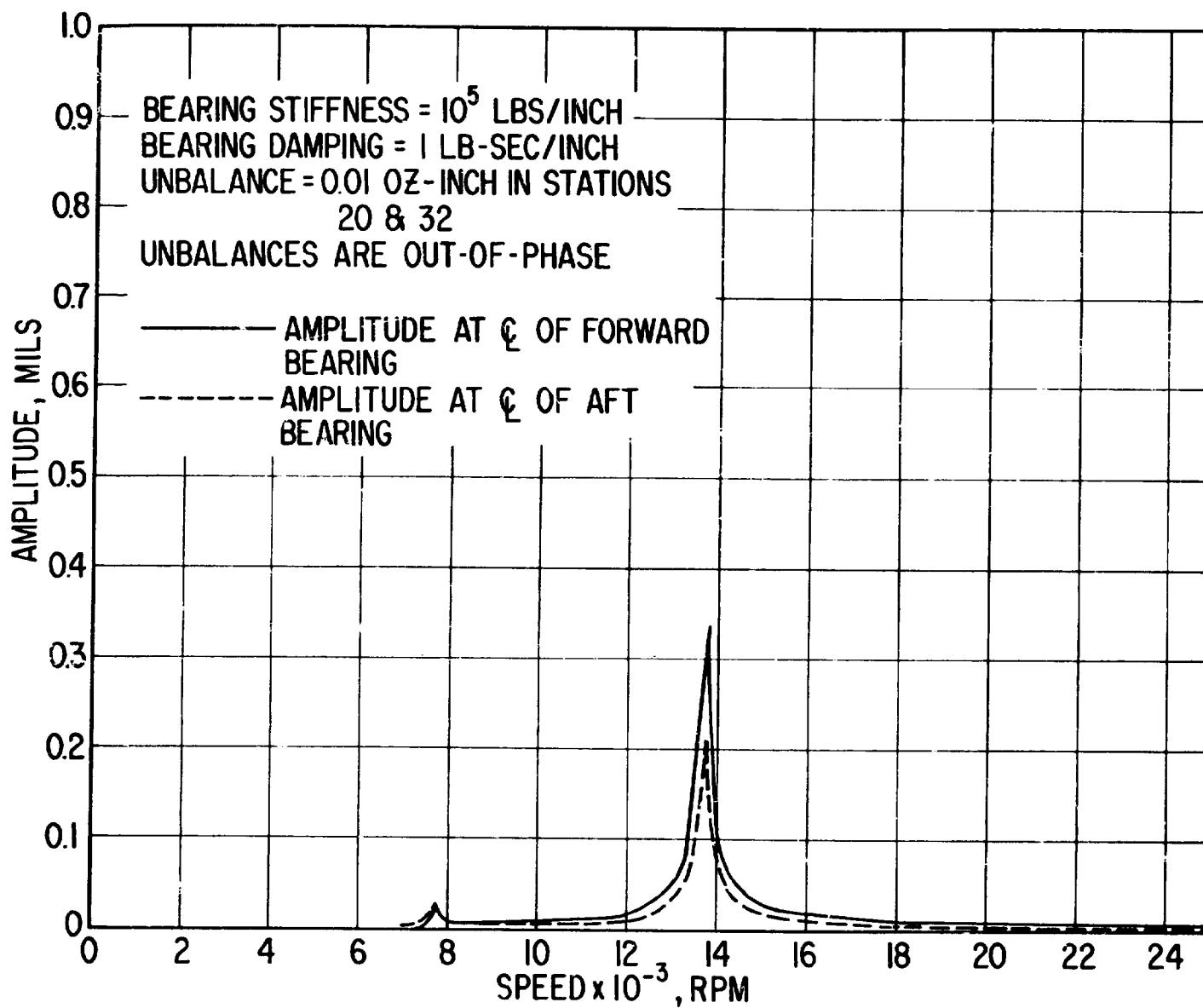


Fig. 71 Calculated Unbalance Response of the Dynamic Simulator Rotor at the Bearing Locations, 0.0539 ounce-inches², Total Dynamic Unbalance

bearing center planes, for unbalances of 0.01 ounce-inches in each of two planes. Figure 70 shows the response when two mass unbalances are in-phase and Figure 71 shows when the two unbalances are 180° out-of-phase. These calculations were made for a bearing stiffness of 100,000 lbs/in and with the small damping value of 1 lb-sec/inch in each journal bearing. The amplitudes at the bearings are less than 1 mil, at the critical speeds and less than 0.1 mils past the second critical speed. The curves are plotted only up to 25,000 rpm, but the data did not show any increase in amplitude up to 35,000 rpm.

In summary, the dynamic simulator appears to be well behaved dynamically over its operating speed range.

4. Test Facility and Instrumentation

In this section, the facility required for operating the dynamic simulator test vehicle is briefly discussed, and the required instrumentation, gas flow and controls are noted.

The envelope dimensions of the housing of the dynamic simulator are noted in Figure 67. In the test cell, this unit should be secured through vibration isolators to an appropriate bedplate. The rotor weighs about 109 lbs. and the housing weight is about 700 lbs. Appropriate overhead crane and other lifting provisions will be needed in the test cell.

Figure 72 is a schematic of the test facility that identifies the following auxilliary equipment:

- Loaders
- Shaft position monitors
- Air pressure monitors
- Heaters and temperature monitors
- Speed indicators
- Accoustic and vibration monitors
- Hot gas supply
- Control room equipment

These are briefly discussed below.

a. Loaders

Radial load is applied by two vertically mounted, cylindrical load cells (RL-1) and RL-2). These load cells are supplied with pressurized gas which is monitored by (PE-8) and (PE-10). The gas exits the cell across a narrow compliant curtain that is positioned in close proximity to the shaft surface. Thus, the pressure inside the curtain is virtually constant and equal to the cell supply pressure. This type of load cell design permits application of the required load with minimum influence on the natural dynamic response of the shaft.

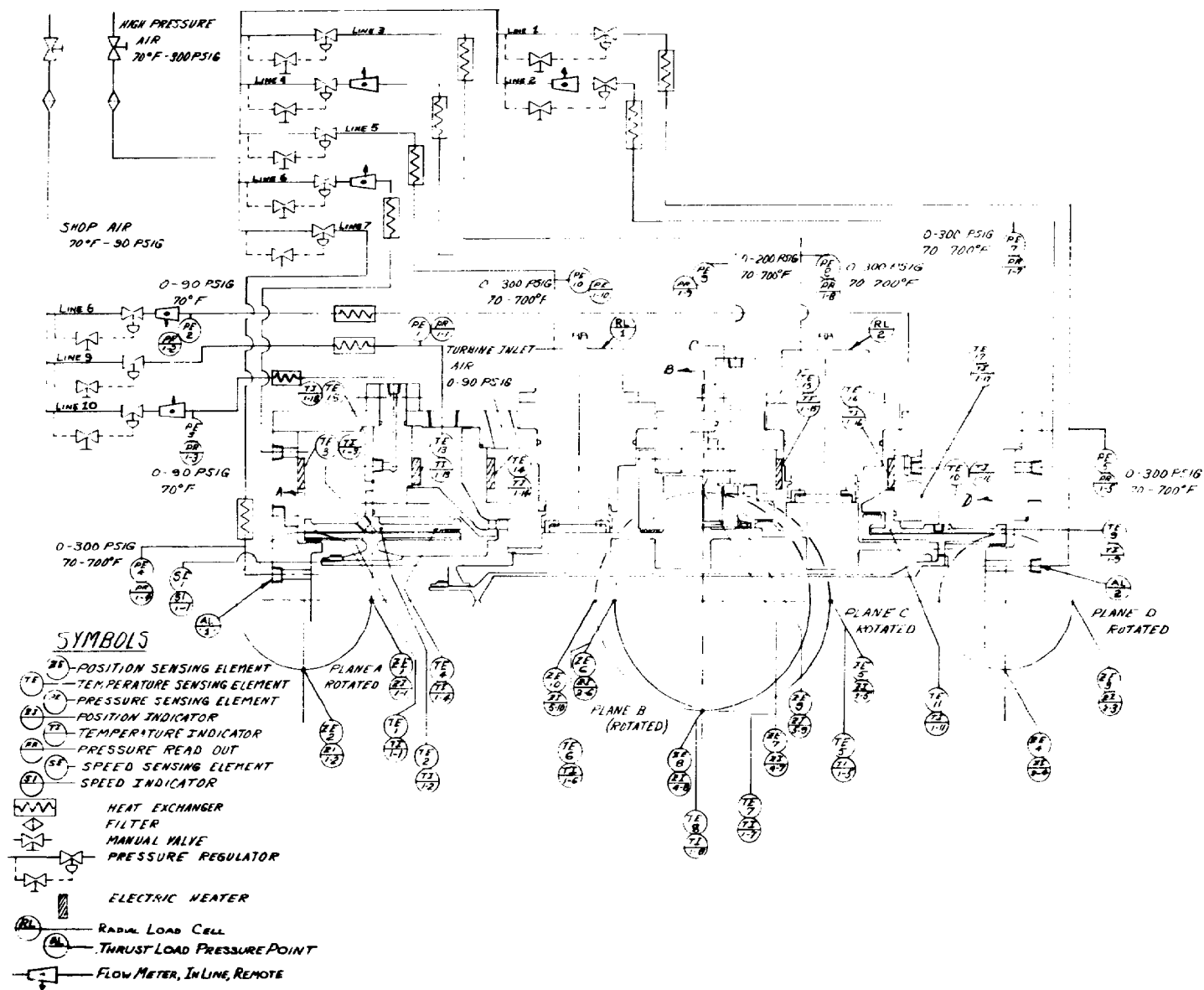


Fig. 72 Schematic of the Dynamic Simulator Test Facility

Thrust load application is achieved pneumatically, by applying a pressurized gas source to the rotor bore (designated "load cell" in Figure 67), through the inlet at the aft end of the housing, thus creating a pressure differential between (AL-1) and (AL-2). Note that controlled thrust pulses can be established by oscillating the pressure differential. Note also that shaft conical motion can be similarly induced by controlling the pressure differential between the two radial load cells.

b. Shaft Position Monitors

High temperature capacitance probes should be used to determine the shaft motion relative to the housing. Since the shaft is very stiff compared to the gas bearing stiffness, and since the maximum operating speed is well below the first bending critical speed, the shaft can be considered rigid. Therefore, two sets of two probes spaced 90° apart located in two separate planes that are perpendicular to the shaft axis are sufficient to completely determine translatory and conical shaft motions. These planes are labeled "A" and "D" in Figure 72 and the position sensors in these planes are labeled (ZE-1), (ZE-2), (ZE-3) and (ZE-4).

Unlike the shaft, the thrust bearing and thrust runner members should not be considered rigid because of the high thrust loads and structural mounting characteristics of the members. Therefore, two sets of three probes should be installed in the stationary thrust bearing. As indicated in Figure 72, the probes (ZE-7), (ZE-8) and (ZE-10) are located in plane "B" and (ZE-5), (ZE-6) and (ZE-9) are located in plane "C".

c. Air Pressure Monitors

Remote reading pressure gages should be used to monitor the gas pressure supplied to each of the bearing cavities since the load capacity of the gas bearings is influenced by pressure and thus, the bearing ambient pressure is an important test parameter. Note that, in the case of the journal bearings, the gas supply is introduced directly against the journal for cooling purposes and that leakage rate through the labyrinth seals will influence the bearing ambient pressure as well as the journal bearing temperature distribution. As shown in Figure 72, the supply pressure to the compressor end journal is sensed by (PE-3), the pressure to the turbine end journal by (PE-2), and the pressure to the thrust bearing by (PE-9).

In addition to the bearing cavity pressures, the following pressures should also be monitored:

- Drive Turbine Inlet: (PE-1)
- Radial Load Cells: (PE-10) and (PE-8)
- Thrust Load Cavities: (PE-4) and (PE-5)

d. Heaters and Temperature Monitors

Electric heaters with the capacities shown in Tables XVII and XVIII should be incorporated to control the test rig temperature distribution. (These heaters are sized to raise the temperature of the rig components and supply air to a maximum test temperature of 1400°F).

Five (5) heaters are shown in Figure 72 and these are accompanied by the temperature sensing elements (TE-3), (TE-13), (TE-14), (TE-15) and (TE-16). In addition, the following temperatures should also be monitored:

- Compressor-end journal bearing: (TE-1), (TE-2), (TE-4) and (TE-18).
- Thrust bearing: (TE-5), (TE-6), (TE-7) and (TE-8).
- Turbine-end journal bearing: (TE-11), (TE-10), (TE-9) and (TE-17).

e. Speed Indicator

As indicated in Figure 72, an optical speed sensor (SE-1) should be incorporated.

f. Accoustic and Vibration Monitors

A directional microphone should be located in the test cell and pointed towards the turbine-end section of the simulator housing, which should also be fitted with at least three accelerometers for sensing vibrations.

g. Hot Gas Supply

Operation of the dynamic simulator test vehicle will also require a sizable air supply, as shown in Figure 72 and in Tables XVII and XVIII. A hot gas test facility will be required, preferably located outside, but adjacent to the test cell of the dynamic simulator. This hot gas test facility will require the following items:

- A high pressure air source capable of providing 1700 lbs/hr of air at 400 psig and 70°F
- A high pressure header with take-off and heater requirements as noted in Table XVII
- A low pressure air source capable of providing 2400 lbs/hr of air at 90 psig and 70°F
- A low pressure header with take-off and heater requirements as noted in Table XVIII
- Remotely operated pressure regulators with local pressure gages for each line identified in Tables XVII and XVIII
- Flow meters in the cooling air flow lines to each bearing

TABLE XVII

TAKE-OFF AND HEATER REQUIREMENTS
ASSOCIATED WITH HIGH-PRESSURE HEADER *

Line No.	Description	Flow (lb/hr)	Heater Size (KW)	Comments
1	Thrust Cavity Pressurizer	Negligible	0.5	Filter not required.
2	Turbine End Journal Cavity	585	33	Include in-line flow meter and filter.
3	Load Cell	Negligible	0.1	Filter not required.
4	Thrust Bearing Cavity	530	28	Include in-line flow meter and filter.
5	Load Cell	Negligible	0.1	Filter not required.
6	Compressor End Journal Cavity	585	33	Include in-line flow meter and filter.
7	Thrust Balance Cavity	Negligible	0.5	Filter not required.
TOTAL		1700	95.2	

* 400 psi, 700°F air

TABLE XVIII

TAKE-OFF AND HEATER REQUIREMENTS
ASSOCIATED WITH LOW-PRESSURE HEADER *

Line No.	Description	Flow (lb/hr)	Heater Size (KW)	Comments
8	Cooling Air Turbine End	620	7.0	Include in-line flow meter.
9	Turbine Drive Supply	1125	22.7	
10	Cooling Air Compressor End	600	6.8	Include in-line flow meter.
TOTAL		2345	36.5	

* 90 psi, 200°F air

For short term testing, tank trucks of compressed air can be used. However, for long term testing in a permanent test facility, it would be more economical to incorporate a large capacity, positive displacement air compressor coupled to accumulators to provide the necessary air flow capacity.

h. Control Room Equipment

A control room adjacent to the test cell will house the test personnel as well as the controls, display and recording equipment.

The controls schematic should be engraved or drawn on the panel board, with the operation knobs, switches and buttons appropriately located in the flow lines of the schematic. Visual displays such as pressure gages, temperature read-outs and associated alarm lights should be mounted directly above each control actuator. The control items must include:

- Air pressure and temperature to inlet of drive turbine
- Air pressure to radial load cylinders
- Air pressure to rotor bore (thrust load applicator)
- Air pressure and temperature to the bearing cavities
- Air pressure and temperature of cooling air to the bearings
- Current flow to the electric heaters in the dynamic simulator

In addition to the above noted controls, there should be a section of the control room devoted to the monitoring and recording of shaft motion measurements. The signals from the capacitance probes should be displayed on cathode ray oscilloscopes for monitoring and simultaneously recorded on magnetic tape. This shaft motion display and recording section should comprise:

- Four dual beam oscilloscopes to display the orbital or component motions as sensed by the displacement probes located in planes A, B, C, and D of Figure 72.
- Circuit and additional oscilloscope to determine and display phase relations between motions in planes A and D.
- Circuit to determine differences between any of the thrust bearing displacement sensors and a dual beam oscilloscope to display axial shaft position and distortion of the thrust bearing surfaces.
- Magnetic tape recording of all displacement sensors together with a voice channel for the test director, a sound channel for the microphone in the test cell, a channel for recording the shaft speed and four channels for recording load cell pressures.

Finally, the control room needs to have a section devoted to the alarms and to the balance of the monitoring and display equipment comprising:

- Alarms as shown on Table XIX below
- Temperature indicator-recorder for in-line heaters and electric heater located inside the simulator
- Shaft speed
- High pressure header, pressure and temperature
- Low pressure header, pressure and temperature
- Radial and axial loader pressures
- Drive turbine supply gas pressure
- Journal and thrust bearing cavity pressures
- Cooling gas supply bearings, pressure

TABLE XIX
CONTROL FACILITY ALARMS

No.	Description	Type	Action
1	Shaft Over Speed	Sound & Light	Automatic shutdown line 9; turbine inlet and heater
2	Shaft Under Speed	Light	At discretion of operator.
3	Loss of High Pressure Air Supply	Sound & Light	Automatic shutdown line 9 and heaters in lines 1, 2, 3, 4, 5, 6, 7, and 9.
4	Loss of Low-Pressure Air Supply	Light	Automatic shutdown lines 1, 3, 5, 7 and heaters in lines 1, 3, 5, 7, 8, 9, and 10.
5	Temperature in Simulator Exceeded	Light	At discretion of operator.
6	Bearing Overload	Flashing Light	At discretion of operator, or automatic shutdown lines 1, 3, 5, 7; load cells.
7	Bearing Over Temperature	Flashing Light	At discretion of operator, or automatic shutdown heaters in lines 8 and 10.
8	Oversize Orbit	Light	At discretion of operator, or automatic shutdown lines 1, 3, 5, 7, 9 and associated heaters.

SECTION VII

PRINCIPAL TECHNOLOGY SEGMENTS AND LONG RANGE TEST EFFORT

Advancement of foil bearing technology, leading to the development of air-lubricated foil bearings that will be applied in advanced engines and power systems requires attention to the following, basic elements of the technology:

1. Extension and verification of design tools for prediction of foil-bearing performance.
2. Optimization of foil bearing designs to meet the operational and environmental requirements of advanced engines and power systems.
3. Selection and verification tests of foil bearing materials and surface treatments for the engine and power system environments.

The developed foil bearings will need to be evaluated first in a dynamic simulator of an advanced engine and subsequently in acceptance tests conducted in actual engines or power systems. This is illustrated in diagrammatic form in the Figure 73 flow chart.

Here, the basic technology items are briefly discussed, based on the results of the present program to identify needed advances.

1. Extension and Verification of Design Tools

An analysis of two types of air-lubricated foil bearings, in both their journal and thrust bearing configurations, were provided earlier in Sections II and III of this report. This analysis coupled the elasticity characteristics of the particular foil geometries with the hydrodynamic pressure generation properties of the film to arrive at equilibrium film shapes and, hence, to compute bearing performance. The analysis included compressibility effects, but neglected side leakage.

From the charts and equations provided in Sections II and III, as well as by using the computer programs described in Appendix II, performance calculations for individual bearings and operating conditions can be effected. In particular, the geometrical features (such as foil thicknesses, initial clearance and pre-load, as well as bump height and spacing in the case of ridge type foils, or curvature and overlap in the case of leaf type bearings) can be optimized for maximum load capacity, using the data and programs. Application of this analysis has already shown that some initial, intuitive assumptions regarding desirable foil properties were incorrect. For example, as noted in Section V of this report, the initial approach to the design of ridge-type foil-bearings had used thin and highly conformable upper tapes. The analysis of Section II of this report, subsequently showed this to be detrimental to load capacity. Tests conducted outside the present program, confirmed the analytical predictions that the upper tape should be sufficiently rigid that it does not form pockets between support bumps.

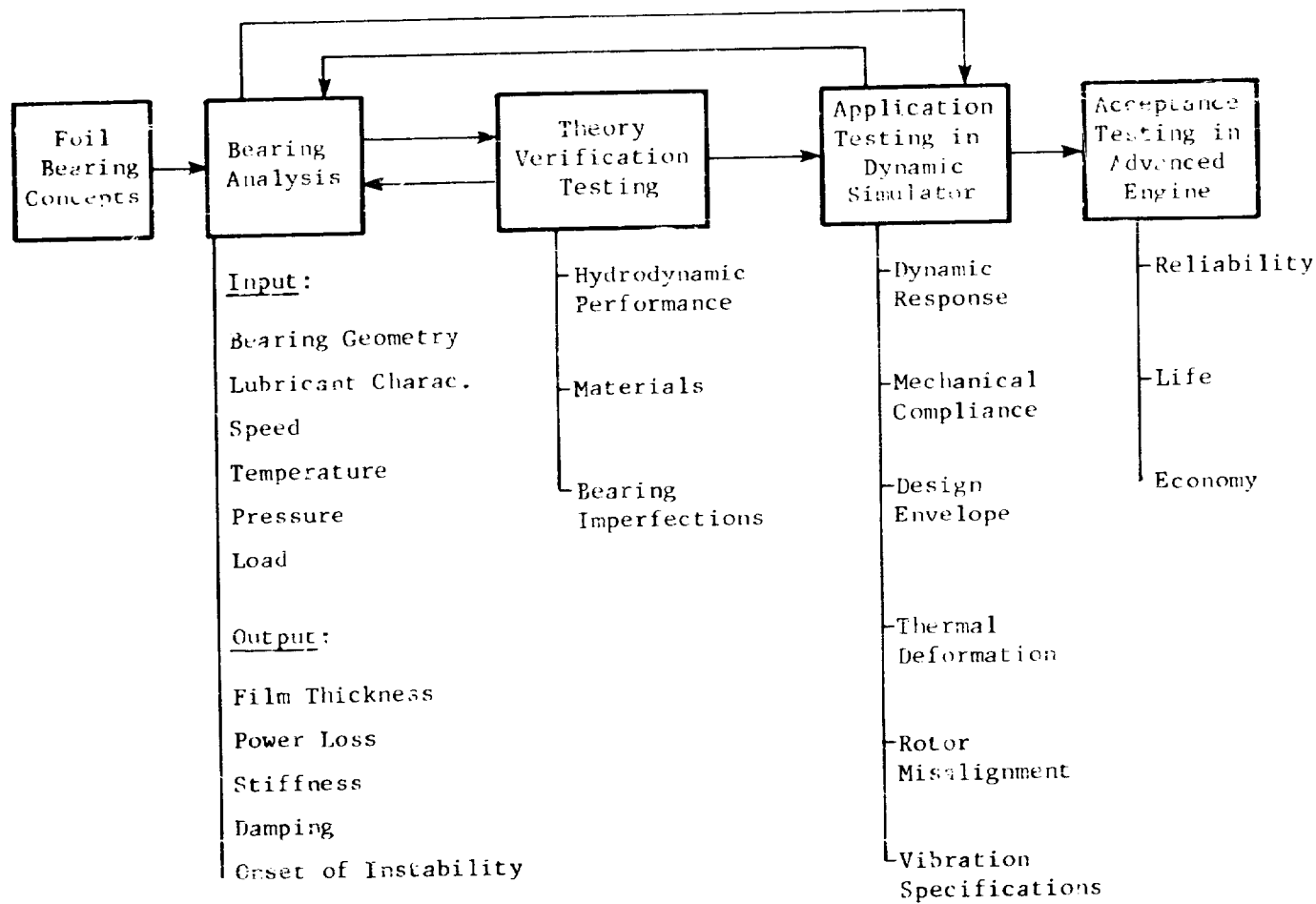


Fig. 73 Foil-Bearing Development Flow Chart

To advance this segment of the technology, complementary bearing tests and refinements of the present analysis are needed. The tests made for purposes of checking the validity of present analysis and to direct future refinements may be conducted in existing air-bearing test rigs where possible. Such tests should include measurements of:

- Film Thickness,
- Attitude Angle (in the case of journal bearings),
- Static and Dynamic Stiffnesses,
- Damping,
- Instability Threshold and,
- Power Loss,

all as functions of size, foil geometry and operating conditions.

Film thickness measurement is especially important, as regards correlations of theory and test. However, it is also a difficult measurement to make in compliant surface bearings. A promising approach is to incorporate the capacitance probe in the shaft, viewing the foil bearing surface, and transmission of the signal via slip rings or air-borne signals.

In parallel with such tests and guided by the test data, the present analysis should be extended to include:

- Side-Leakage Effects
- Foil-to-Foil and Foil-to-Housing, Coulomb Damping
- Turbulence and Fluid Inertia Effects
(where applicable in high ambient pressure and high-speed applications)
- Mean-Free Path Effects (where applicable in low ambient pressure applications with very thin bearing films)
- Misalignment and thermal distortion effects

2. Optimization of Foil Bearing Designs

The review of the MAT-1 engine described in Section VI of this report has shown the need for load capacities of the order of 35 or 40 psi unit loading. Although reported unit loads carried to date on air-lubricated foil bearings have been less than 20 psi (see, e.g. Reference 3), the required doubling of this figure is believed to be a realistic technology target that should be achieved well before 1980. For example, in tests with a (rigid surface) air-lubricated tilting pad (Reference 8), support of unit loads up to 35 psi without hydrostatic assist was already demonstrated.

Present foil bearing designs will need to be reviewed with a view to insuring that the foil deflections serve to generate a near optimum hydrodynamic wedge. More importantly perhaps, emphasis will need to be placed on mechanical design from the standpoint of minimizing loss of load capacity from thermally induced distortions and from misalignment.

Perhaps the principal limitation on the load capacity of air-lubricated bearings in high-speed engine applications, is differential thermal growths between the mating elements. These are induced by thermal gradients that are developed in the bearing region as a result of the heat generated in the films as well as heat influx from engine gases. In the case of a journal bearing, the frequent consequence is loss of clearance* while, in an annular thrust bearing, dishing type distortions of the runner and stator cause relatively large separation at the outside diameter and rubbing contact at or near the inside diameter. Earlier analytical and experimental studies of rigid surface gas-lubricated bearings for engine applications (e.g., References 8 and 9), have shown the need to provide convective cooling by means of leakage air from the compressor, for well-designed heat exchange paths in the bearing regions, and for thrust plate supports that are maintained nearly isothermal by virtue of the convection cooling and which serve to limit thrust stator dishing. Another factor that must be considered is heat soak back from the turbine, following engine shut down. This can produce high rotor temperatures in the turbine end journal. This journal temperature must be determined under this transient condition, to establish its effect on clearance and on the bearing surface materials.

Foil bearings are, inherently, more tolerant of distortions than are rigid-surface bearings. The analysis, for example, shows foil journal bearing film thickness to be less sensitive to increased preload than is the case with rigid-surface journal bearings, and this has been confirmed by experience. Nevertheless, when considering 5 inches diameter journal bearings and up to 8-1/2 inches diameter thrust bearings operating at 35,000 rpm as in the dynamic simulator previously discussed, convective cooling provisions and optimization of heat transfer paths will be necessary to achieve the desired 35 to 40 psi unit load capacity.

The other key mechanical design factor as regards reliable operation under high loads is tolerance of misalignment. Here, the foil elasticity provides inherent self-alignment capability. However, with present designs, the self-alignment capability decreases under high loads in both the ridged type and the cantilevered leaf type of foil bearings. In the former case, the bump stiffness rises after initial deflection. In the latter case, the bottoming of the foils precludes further self-alignment.

In summary, then, a high priority item in the development of foil bearings for applications in advanced engines and power systems in the range of 1000 pounds to 3000 pounds thrust, is to optimize the bearing design so as to achieve reliable operation under unit loads in the range of 35 or 40 psi. To achieve this goal, emphasis in mechanical design of the bearings has to be placed on

- Foil designs that deflect so as to provide a near optimum hydrodynamic wedge.
- Cooling provisions.
- Extending the self-alignment range of foil designs.

* This is in addition to clearance reduction due to centrifugal growth of the rotor.

3. Materials Development

The review of the MAT-1 engine in Section VI of this report showed bearing cavity temperatures to be of the order of 700°F. Thus, a bearing temperature capability of 1000°F appears to be a reasonable, intermediate goal. For this temperature range, the study of foil bearing substrate materials in Section V of this report has shown that Inconel X-750 is satisfactory. The materials technology emphasis should be concentrated on surface coatings and/or treatments or provide the necessary protection against starts, stops and high-speed rubs.

Even though both of the solid lubricant coatings that were used in the tests described in Section V survived the start-stop and high-speed rubs, the appearance of the bearings at the end of the test showed that the films were approaching the limit of their endurance. In fact, the MLF-5 film was in the last stage of effectiveness at the end of the high-speed rub tests with only a thin coating of gold left in the contact areas. The AFSL-28 film had been worn to a thin, transparent layer of material which probably would not have survived many more sliding cycles. Although these tests were unduly severe because the bearing design had not been optimized, the fact remains that improvements in surface coatings or treatments will be required for foil bearings in engine applications.

Past experience in evaluating coatings for tilting pad bearings, has shown that certain plasma-sprayed coatings, such as chrome-oxide, have much better wear resistance than the conventional epoxy-bonded MoS₂ coatings. For example, a 0.0005 inch thick coating of resin bonded MoS₂ on a steel substrate survived 1000 starts and stops at a unit load level of 4 psi, but examination of the surfaces after the test showed that the film was beginning to wear through and permit metal to metal contact. In contrast, a 0.003 inch thick plasma coating of chrome-oxide showed about 50 microinches of smooth wear under the same conditions. It will probably be difficult to plasma-spray the foil bearings with any hard surfacing materials because of the difficulty of controlling coating thickness. However, coatings such as chrome-oxide or silicon nitride can be sputtered on the foil surfaces with little difficulty. A sputtered coating about 100 microinches thick should be adequate for wear resistance, and still leave the foil flexible enough to act as a conformable bearing surface. To combat the problem of the high friction of hard, wear-resistant coatings, a coating of a solid lubricant, say MoS₂ (if permitted by the operating temperature) or the AFSL-28 fluoride composition could then be sputtered on top of the hard coating to reduce the coefficient of friction. A similar approach has been evaluated using plasma-sprayed chrome-oxide and a sputtered top coating of MoS₂. Reference 5, and it has been found that this is a promising combination with high wear resistance and good sliding characteristics.

In one promising approach that should be evaluated, the hard coating is sputtered on the surface of the foil in a pattern so that the surface is interrupted by shallow grooves. Then the solid lubricant coating is sputtered in the grooves. When the surfaces are sliding in contact, the load is being

carried on the hard coating and the grooves simply act as reservoirs which feed small quantities of solid lubricant to the surface as the hard coating wears. Since the solid lubricants function best as very thin films, this should provide sufficient lubrication for the bearing. Alternatively, the Inconel X-750 foil could be etched to form grooves about 0.0001 inch deep. Then, the hard coating could be sputtered onto the surface, leaving a grooved pattern, and then the whole surface could be coated with a sputtered MoS_2 film in sufficient depth to fill the grooves. This would be relatively easy to do since MoS_2 sputters at a fast rate. Following this, the whole surface could be lapped down until the MoS_2 coating was a very thin film, about 50 microinches thick on the hard coating surface, and 150 microinches thick in the grooves. Even after the 50 microinch coating had been worn off the hard coating, the grooves would still continue to supply some lubricant as the hard coating was worn away.

The most promising hard coatings that are known at the present time are chrome-oxide and silicon nitride. Both of these are worthy of consideration. For the sputtered solid lubricant films, MoS_2 or the AFSL-28 should be evaluated. Although the frictional behavior of the fluoride film is not completely satisfactory at low temperatures, this is the most promising coating for applications which must operate above 1000°F. All of these materials can be sputtered without any unusual difficulty.

4 Long Range Test Effort

The basic elements of bending-dominated foil bearing technology that are in need of advancement were noted in the earlier parts of this section. Testing for the purposes of verification and guidance of the analysis, as well as to determine the capability of the surface materials will be needed. This can be conducted initially in existing test rigs and completed by tests with larger bearings (5 inches diameter journal bearings and up to 8-1/2 inches O.D. thrust bearings) in the test vehicle described in Section VI of this report.

Table XX shows recommended intermediate and final targets for the operational capabilities of the bearings that should be demonstrated in the dynamic simulator tests. These targets are based upon the review of the MAI-1 engine and our current estimate of the potential capacity of air-lubricated foil bearings. This section of the report is concluded with a brief discussion of the targets given in Table XX.

a. Ambient Pressure

Air-lubricated bearings operate in the compressible flow regime and their performance is therefore affected by the ambient pressure in the bearing cavities. The extent of compressibility effects in a given application can be gauged by computing the ratio or the unit load that the bearing supports, to the ambient pressure. In current practice with high-speed,

TABLE XX

INTERMEDIATE AND FINAL TEST TARGETS FOR BENDING-DOMINATED FOIL BEARINGS FOR 1980
TIME FRAME ADVANCED ENGINES, AND POWER SYSTEMS IN THE 1000 TO 3000 POUNDS THRUST CLASS

Parameter	Intermediate Target	Final Target
Ambient Pressure, Minimum (psia)	15	15
Ambient Temperature, Maximum (°F)	700	1000*
Surface Speed - Journal Bearings (ft/sec)	770	770
Surface Speed - Thrust Bearings (ft/sec)	1100	1300
Unit Loading - Journal Bearings (psi)	21 (or 60% of MIL E-5007C)	35 (or 100% of MIL E-5007C)
Unit Loading - Thrust Bearings (psi)	24 (or 60% of MIL E-5007C)	40 (or 100% of MIL E-5007C)
Start-Up Unit Loads, (psi)	5	5
Number of Start-Stop Cycles	1000	10,000

*This bearing temperature is adequate for the MAT-1 engine. However, substantially higher temperature capability may well be required, or at least desirable, for other advanced engines in the 1980 and later time frames.

air-lubricated bearings, this ratio is generally less than 2:1*, for reliable operation. Well designed foil bearings should be capable of somewhat higher values, probably up to about 2.5 or 3:1.

We have noted that the advanced engine bearings of interest here will need to support unit loads up to 35 or 40 psi. Accordingly, a realistic limiting value for ambient pressure is about 15 psi, as noted in Table XX. This value is adequate for the MAT-1 engine since, as shown in Table XI, the compressor discharge pressure available at 60,000 feet altitude is 18.6 psi, minimum. For higher altitude operation, the compressor discharge flow will probably need to be boosted before admission to the bearing cavity.

b. Ambient Temperature

From the standpoint of air-lubricated bearings, the only limitation here is materials capability. In the case of the MAT-1 engine, bearing cavity temperatures will be less than 1000°F. The bearing substrate and coating materials investigated in the present program are suitable for this range and can be developed for continuous operation at 1000°F.

c. Surface Speeds

The surface speeds shown in Table XX are those of the journal and thrust bearings sized for the MAT-1 engine. Because of the high heat generation and the high stresses associated with the thrust bearing, an intermediate target on 1100 feet/second is recommended to be followed by operation at 1300 feet/second, which corresponds to an 8-1/2 inch outside diameter thrust collar rotating at 35,000 rpm.

d. Unit Loads

As noted earlier in this report, the unit loads of 35 to 40 psi for the journal and thrust bearings respectively, which are required to meet the maneuver loads. In the case of the MAT-1 engine, for example, the start-up load was only about 2 psi. Thus, the value of 5 psi noted in Table XX is a conservatively high figure for start-stop evaluations, but it is one that should be readily achievable with development effort in surface coatings.

Initially such coatings should be capable of at least 1000 starts and stops without significant degradation. Ultimately, a capability of 10,000 starts and stops should be sought.

* Except for some spiral grooved bearings that are less severely affected by compressibility effects.

SECTION VIII

CONCLUSIONS AND RECOMMENDATIONS

1. Conclusions

The analysis of air-lubricated foil bearings requires coupling of the elasticity relations for the foils with the compressible flow Reynolds equation. This was done in the present program for two types of foil bearings in both their journal and thrust bearing versions. The comparison of the data generated showed that, in the journal bearing version, the ridged type of foil bearing appears to have superior load carrying capacity. In the thrust bearing version, on the other hand, the cantilevered leaf type of foil bearing appears to have the better load capacity.

The charts and computer programs provided permit optimization of each of the types of foil bearings analyzed for any specific set of operating conditions. However, the parametric study of the ridged type of foil journal bearing indicated that the effect of upper tape compliance is to reduce load capacity. Conversely, a high compliance of the support (or bump) foil is necessary to achieve high film thickness and to minimize attitude angle. This latter is desirable from stability considerations. However, in an engine application, the selection of the support foil compliance (and, hence, bearing stiffness) will generally need to be determined on the basis of critical speed and dynamic response considerations.

In the case of the ridged thrust bearing, a converging inlet zone must be provided to serve as the mechanism for hydrodynamic pressure generation. Again, the rigidity of the upper tape must be high enough to prevent pocketing between adjacent supports.

In the cantilevered leaf type journal bearing, load capacity is improved by increasing the degree of conformity between the leaves and the shaft. This can be effected by proper selection of leaf radius and degree of overlap.

The calculations made for the leaf type thrust bearing indicated that, as would be expected, load capacity is improved by achieving film convergence over the full circumferential length of each leaf.

The survey of foil substrate materials showed that for operation at temperatures up to 1000°F, Inconel X750 is the preferred choice at this time. It has the advantage that it can be air hardened after forming. Haynes 25 appears to be a reasonable second choice but the alloy must be cold-worked for optimum strength. Cold-worked austenitic stainless steels could be used at temperatures below 750°F, but carbide precipitation could be a problem at temperatures above 800°F. All those materials are available in strip form in a range of thicknesses.

The calcium fluoride-barium fluoride-aluminum phosphate solid lubricant film developed by the Air Force and designated AFSL-28 is a candidate surface coating for engine foil bearings. This film is useful in air at temperatures up

to 1500°F, and was developed particularly for use on nickel base alloys, such as RENE 41 and Inconel X750. Another potential solid film lubricant is the NASA developed MLF-5 film which is a sodium silicate bonded coating containing MoS₂, graphite and gold. It is effective over a wide temperature range, but its life in air at 1000°F needs to be determined in long duration tests, since this is above the oxidation limit of MoS₂.

Start-stop and high-speed rub tests were conducted at 1000°F with full 360° journal bearings coated with the AFSL-28 coating as well as with identical bearings coated with the MLF-5 film. In these tests, the MLF-5 film was generally more effective under start-stop conditions, but the AFSL-28 coating appeared to have a better wear life in the high-speed rub tests. Both of these coatings should receive further consideration for application in the advanced engine bearings.

The high speed rub tests also showed that foil bearings appear to be more tolerant toward shock loads than are rigid surface bearings, as the foil flexibility seems to "cushion" the applied shocks.

The bearing load analysis of the MAT-1 engine, including the maneuver loads defined in MIL E-5007C, showed that unit loads can be kept to the order of 35 psi in the journal bearings with some increase in journal bearing span. Thrust unit loading is of the order of 40 psi and this may be reduced by improved thrust balance. This level of bearing loading, while it is substantially higher than has been realized to date on air-lubricated foil bearings, constitutes a realistic technology target that can be achieved before 1980 through an integrated analytical and test program.

A layout drawing of a dynamic simulator of the MAT-1 engine was prepared. The critical speed and unbalance response analysis of this test vehicle showed that its operating speed range is well below the third rotor-bearing system critical speed and that the unit should be well-behaved dynamically.

The most important factors as regards the application of foil bearings in the engine environment appear to be: (a) the required increase in load capacity, (b) high thrust bearing surface speeds, possibly up to 1300 feet/second in the case of the MAT-1 engine without improved thrust balance, with consequent high heat generation and thermal distortions, (c) surface materials that provide protection during starts, stops and high speed rubs.

Accordingly, the technology segments that must be emphasized comprise: (a) foil designs that deflect to provide a near optimum hydrodynamic wedge, (b) cooling provisions and bearing stator designs to minimize thermal distortions, (c) extending the self-alignment range of present foil bearing designs and, (d) improved approaches to surface coatings, possibly through the use of sputtered hard coatings with a solid lubricant surfacing.

2. Recommendations

An integrated analytical and test program is required to advance foil bearing technology to permit rational development of such bearings for reliable operation in the engine environment. The test requirements may be viewed in three categories: (a) for the purpose of verifying the analysis and providing

guidance for its extension, (b) application testing in the dynamic simulator test vehicle over the full ranges of load, speed, temperature and ambient pressure of the application as identified in Section VII and, (c) acceptance tests in the engine.

Direct film thickness measurement is especially important with regard to correlation of theory and test data. This is also a difficult measurement to make in compliant surface journal bearings. A method such as incorporation of capacitance probes in the rotor viewing the bearing surface and transmission of the readings via slip-rings or air-borne signals should be evaluated at an early date.

The current analysis of foil bearings also requires extension in several areas of which the most important are inclusion of: (a) side-leakage effects (b) Coulomb damping between the bearing elements and (c) misalignment and thermal distortions. However, it is cautioned that accurate experimental data, notably direct measurement of film thickness distribution as noted above, should be obtained to guide such future analysis.

As noted in the conclusions above, the principal technology requirement related to the use of materials for foil bearings in environments up to 1000°F, is improved surface treatments and coatings. Here, the use of sputtered hard coatings of silicon nitride (or, possibly, chrome oxide if it can be shown to be suitable for high temperature operation when applied in a thin, sputtered layer), with a surfacing of solid film lubricant need to be evaluated. A promising approach involves sputtering the hard coating in a pattern of the surface, interrupted by shallow grooves containing the solid film lubricant.

Mechanical design studies on the bearings also should be conducted emphasizing optimization of foil geometry for maximum load capacity, control of thermal distortions and extension of the self-alignment capabilities of the current bearings.

APPENDIX I

STATE-OF-THE-ART SURVEY OF GAS-LUBRICATED FOIL BEARINGS

1. INTRODUCTION AND SUMMARY

There are two basic categories of foil bearings as follows:

1. Tension-dominated foil bearings
2. Bending-dominated foil bearings

The tension dominated foil bearing uses an assembly of thin metal strips that are held in tension around a rotating journal to support it. The bending-dominated foil bearing consists of various types of preformed leaf, pleated or ridged-type, resilient metallic inserts contained in journal or thrust bearing housings that support the rotor. A fundamental difference between the two foil bearing categories is that the tension-dominated bearing relies on the wrap effect where the pressure is governed by the tension in the foils. The bending-dominated foil bearing, on the other hand, achieves its compliance from the resilience (low flexural rigidity) of the insert and can operate with little or no tension applied. The resilience of the bearing elements makes them accommodate manufacturing and assembly tolerances. It also allows safe operation under smaller film-thicknesses and, hence, higher loads than would be possible with rigid surface bearings. Finally, the low mass of the resilient, foil-type bearing elements permits tracking of rotor orbits, thus providing a high margin against self-excited whirl instabilities.

Although it was initially conceived as a method for rotor support, much of the technology on tension-dominated foil bearings has been developed in the computer industry, as a means of moving tape over stationary heads, while precisely maintaining a small air gap between them. Subsequently, tension-dominated journal bearing assemblies have been built and used in laboratory tests to support high-speed rotors.

The theory governing the tension-dominated foil bearing consists of gas-bearing hydrodynamic analysis, coupled with the relationships for elastic deflections of the foils. There exists an extensive body of technical literature, cited in this section of the report, which incorporates the coupled gas-film and foil elasticity relations in a fairly complete manner, so that the level of development and sophistication of tension-dominated foil bearing theory is comparable to that of rigid-surface gas bearings.

Computational methods are available for tension-dominated foil bearings, starting with the prediction of film thickness for a perfectly flexible foil in relative motion with an infinitely long, circular head or journal. These computational methods have been extended to include effects of elasticity of the tape, compressibility of the fluid, side leakage, external pressurization by means of orifices or porous materials, and dynamic response. Experimental confirmation is also available for many of these tension-dominated foil bearings.

Based on the analytical and test results, design criteria for various types of tension-dominated foil bearings have been worked out and the validity of the basic mathematical tools has been verified. Using the design criteria and the design tools, gas bearing designers can size tension-dominated foil bearings with good confidence for particular operating conditions.

The bending-dominated foil bearing is a more recent concept. Although successes have been reported both as regards operating stability and load capacity at high speeds with bending-dominated foil bearing inserts, the published technology behind their design is very sparse. Various types of bending-dominated foil bearings for thrust and journal bearings, as well as foil retention and surface preparation schemes can be found in several industrial patents. However, design criteria of any sort for controlling the various parameters associated with these bearings, as well as design aids for calculating their performance characteristics, are virtually nonexistent in the open literature.

The need for extensions in tension-dominated foil bearing technology primarily involves design and test of specific foil bearing configurations for use in real turbomachines. The bending-dominated foil bearings, on the other hand, require full development of design criteria and design data for the various types of bearing configurations. Experimental verification of the design criteria must be performed along with generation of test data to establish the limits of the various bending-dominated foil bearing concepts.

2. SURVEY OF THE LITERATURE

This section of the report provides a comprehensive bibliography of the published literature on foil bearing technology. This literature has been grouped in six (6) categories as listed below for ease of reference:

- a. State-of-the-art surveys and design manuals
- b. Self-Acting bearing theory and experiments
- c. Externally-pressurized bearing theory and experiments
- d. Dynamic bearing analysis
- e. Experimental studies of the dynamics of rotor-bearing systems.
- f. Bending-dominated foil bearings

The publications listed in the first five categories relate almost exclusively to tension-dominated foil bearings. The references on the subject of bending-dominated foil bearings are grouped in the sixth category.

a. State-of-the-Art Surveys and Design Manuals

The Foil Bearing Design Manual prepared by Eschel and Licht in Ampex Corporation (10), identifies the relevant design parameters and provides design curves for both self-acting and externally-pressurized, tension-dominated foil bearings. It discusses also the dynamic characteristics of such bearings that are important in rotating machinery applications. A good review and summary of the dominant foil bearing lubrication equations is given in (11), with a useful interpretation of the various terms of the equations. The state-of-the-art of tension-dominated foil bearing technology, as it existed at the particular publication times, is summarized in (12 and 13).

b. Self-Acting Bearing Theory and Experiments

Foil bearing technology was initiated by Blok (14). In this classical paper, he observed the foil bearing as a potentially whirl resistant hydrodynamic bearing and presented the first theory for predicting its film thickness. Blok's analysis assumed the development of a wrap region where the film thickness is constant. He calculated this film thickness by means of an inlet zone analysis and verified it experimentally.

There was some additional experimental study of foil bearings conducted in England (15), while in this country it was first used as a loader bearing for a high-speed test rig (16). However, interest in foil bearing technology remained slight for a number of years until the computer industry started to develop it as a means of achieving controlled head to tape separation in recording and pick-up equipment. The lightly tensioned foil with small wrap angle was analyzed (17). Subsequently, Baumeister (18, 19, 20) identified the tension-dominated foil bearing parameter:

$$K = \left(\frac{h_o}{R} \right) \bigg/ \left(\frac{6\mu U}{T} \right)^{2/3}$$

where h_o is the film thickness in the wrap region (inches), R is the radius of the head or journal (inches), T is the foil tension per unit width (lbs/inch), μ is the absolute viscosity of the lubricant (lb.-sec/in^2) and U is the sum of the foil and head or journal speeds (inches/second). The dimensionless parameter K was found by Baumeister to have the value 0.643.

A complete analysis of the performance of fully flexible foils was made by Eschel and Elrod (21, 22), verifying Baumeister's results and also showing the full pressure profile and the details of the film thickness profile including: the inlet region, the constant film thickness wrap region and the exit region where the film thickness exhibits a turning edge dip. This analysis was subsequently extended to include the effects of side-leakage (23), stiffness (24), compressibility (25), and inertia, (26). Additional analysis and design data on self-acting, tension-

dominated foil bearings are provided in (27, 28, and 29)

Paralleling this analytical effort, experimental studies of tension-dominated foil bearings were conducted. The more pertinent experimental results are covered in (30, 31 and 32). These results verified experimentally the theoretical predictions of film thickness, trailing edge dip and side edge dips in self-acting, tension-dominated foil bearings.

c. Externally-Pressurized Bearing Theory and Experiments

Externally-pressurized, tension-dominated foil bearings have also been extensively studied, primarily in connection with tape transport in magnetic recording equipment. Analyses have been developed for bearings fed through orifices and for designs that use porous elements in the heads or journals. An applied external vacuum can also be used to provide further means for controlling film thickness. The results of these studies (33 through 40), showed that some characteristics of externally-pressurized foil bearings resemble those of self-acting bearings including, under some circumstances, the development of a wrap region with uniform film thickness.

The first paper devoted to externally-pressurized foil bearings that has appeared in the open literature is (33). Here a linearized analysis was presented which showed the first order effects of external-pressurization on a simply-supported tape, moving over a flat head. External pressurization and hybrid, (i.e., combined self-acting and externally-pressurized) effects are further discussed in (34, 35 and 36). A comprehensive, non-linear analysis of the full 360°, flexible, externally-pressurized, axisymmetric foil bearing was provided in (37), which showed that the nominal film thickness and edge-dip behavior is similar to that of self-acting bearings.

The use of external-pressurization for film thickness control in otherwise self-acting bearings required the location of pressure ports in the relatively small inlet zone where film thickness is developed and which is very difficult to control. An alternate approach was the use of porous bearing which were studied in (38 and 40). It should be emphasized that externally-pressurized foil bearings are primarily of interest for gap control in magnetic tape-head systems. Because it introduces complexity, external-pressurization should be avoided in foil bearings for rotating machinery, particularly in cases where high temperatures and particulate contamination of the gas stream can be present, as in aircraft engine applications.

d. Dynamic Foil Bearing Analysis

As foil bearing technology progressed, more emphasis was placed on the practical aspects of foil bearings both from the point of view of transport of tape over a recording head, as well as a device for supporting load. Emphasis has been placed on the dynamic characteristic of foil bearings, and a number of dynamic analyses were prepared, as noted below.

The stiffness and damping characteristics of the gas-film, as well as the presence of local resonances in the individual foils and the propagation of waves and other disturbances along the foils were analyzed in (41, 42, and 43).

A special design of a tension-dominated foil bearing intended for rotating machinery application in a non-isothermal, high-temperature environment, by using a loading spring to maintain nearly constant foil tension, is described in (44). Design charts and comparison with tilting-pad bearings from the standpoints of both steady state and dynamic bearing characteristics are also provided in this reference. Further dynamic analyses of tension-dominated foil bearings, including operation in a zero gravity environment, are provided in (45, 46).

The evolution of the tension-dominated foil bearing concept for support of rotating journals, is also illustrated by the two patents (47, 48) both of which were issued in 1970, but which had been filed in 1964 and 1968, respectively. The latter patent (48), describes some practical methods for aligning, tensioning and clamping the foils. It describes also the possible use of hydrostatic-jacking during starts and stops.

e. Experimental Studies of the Dynamics of Rotor-Bearing Systems

Some of the early attempts to support high-speed rotors on tension-dominated foil bearings are reported in (49, and 50). This work was resumed by Licht at Ampex Corporation who conducted detailed experimental studies of rotors supported in tension-dominated foil journal bearings and operated at speeds up to 100,000 rpm (51, 52, 53 and 54).

Initially, this work was conducted with small (2 to 5 lb. test rotors), but it was then extended under NASA sponsorship to dynamic simulators of Brayton-Cycle machinery. A test evaluation of the dynamic simulator of the 2 to 15 KW Brayton-Rotating-Unit, comprising a single shaft mounted turbine, alternator and compressor has been completed, with the report scheduled for mid- 1973. The rotor weight is 20 lbs. and the unit operates at 36,000 rpm in argon at ambient pressures down to 15 psia. The feasibility of supporting high speed rotating machinery in tension-dominated foil bearings and the tolerance of these bearings to starts and stops, temperature gradients and particulate contamination were demonstrated in this work. A parallel analysis for this investigation is described in (55).

An additional experimental study of the dynamic behavior of foil bearings is reported in (56).

f. Bending-Dominated Foil Bearings

Although the gas-lubricated bending-dominated foil bearing is already in use in airborne and land based equipment, there is very little information contained in the literature regarding its performance characteristics. Several geometries have been developed to some extent. Most

of the experience to date has been with the cantilevered, leaf-type of foil bearing pioneered by AiResearch Division of Garrett Corporation. This bearing type is noted in the company's literature. Some of its potential applications are discussed in (57). This type of bearing is in use in air-cycle machines and it is currently being evaluated in a jet fuel starter (3). This reference reports tests on journal and thrust bearing configurations at a speed of 60,000 rpm. A 3.5" O.D. x 2.1" I.D. foil thrust bearing was operated successfully under loads exceeding 120 lbs. The application of this type of bearing in air cycle machines is also described in (58).

Currently, the use of the cantilevered leaf-type or foil bearing is being evaluated experimentally in a dynamic simulator of the NASA's TAC (Turbine-Alternator-Compressor), which was intended for dynamic power conversion in space vehicles. In this application, a 150 lb., 24,000 rpm rotor was supported on two (2), 3-1/2" diameter journal bearings and a 7" O.D. x 4" I.D. thrust bearing. The thrust load was about 700 lbs. and the lubricant (in the simulator tests) was argon gas at 15 to 100 psia bearing ambient pressure and 400 to 700°F bearing ambient temperature. The report on this work was recently published (59).

The cantilevered, leaf-type or bending-dominated foil bearing and some other concepts, as well as methods of foil retention and surface preparation and coating schemes are described in several U.S. patents assigned to the Garrett Corporation (60 through 68).

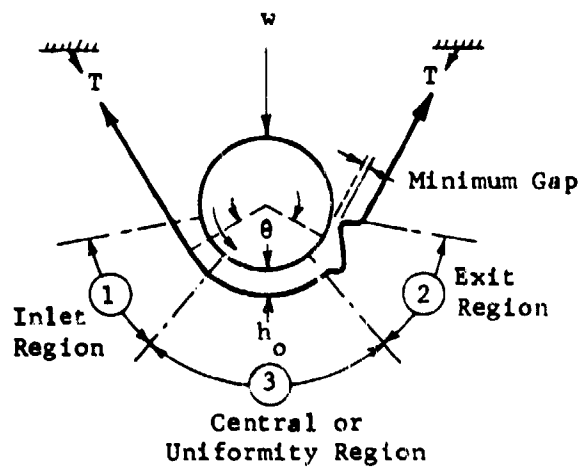
3. CLASSIFICATION OF THE STATE-OF-THE-ART

As noted earlier in this section, there exists as yet very little published design data on bending-dominated foil bearings. Leaf-type bearings are sized in (3) for a particular application and some test data is provided. Bearing sizes for a range of rotor weights and compressibility numbers are indicated in (58). The present report provides the first published analysis on bending-dominated foil bearings and it includes design charts and general purpose calculation procedures from which bearing performance can be determined. However, detailed experimental verification of such calculated performance, including direct measurements of film thickness and stiffness, is not yet available.

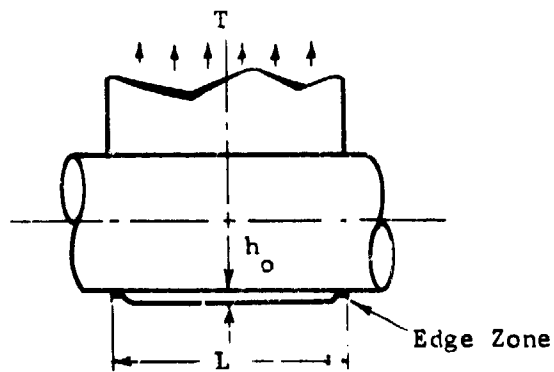
In the case of the tension-dominated foil bearing, there is a larger body of published data, as discussed above. The published design manuals, notably (10) provide useful design data. However, the published comparisons of theoretical and test results are largely limited to vertical rotor operation. Furthermore, this work is limited to journal bearings and there are as yet no published concepts or data on tension-dominated foil thrust bearings.

Following is a brief discussion that indicates the tools available for film thickness and stiffness calculations in tension-dominated foil journal bearings and the degree to which measured and calculated stiffnesses agree.

Figure I-1 illustrates schematically a single, tension-dominated foil bearing pad. The pad has an inlet region, followed by a region where the film thickness is constant and an exit region where the foil undulates and forms a



(a) Tension-Dominated
Foil Bearing - Longitudinal
Tape Contour



(b) Tension-Dominated
Foil Bearing- Lateral Tape
Contour

Fig. I-1 Schematic Illustration of Clearance
Distribution in a Tension-Dominated
Foil Bearing

smaller film thickness. In the axial direction, the film thickness is essentially constant over the middle part of the bearing, but forms edge curls at the ends that minimize side leakage.

In the region of constant film thickness, the gap width is defined by:

$$h_o = K R \left(\frac{6\mu U}{T} \right)^{2/3} \quad (I-1)$$

As earlier noted, the constant "K" was found by Baumeister (19) to have the value 0.643. Licht and Eshel (10) substantiate this value for the journal bearing configuration, provided that the following inequalities prevail:

$$\text{Stiffness:} \quad S = \frac{n}{TR^2 \left(\frac{6\mu U}{T} \right)^{2/3}} < 0.8 \quad (I-2)$$

$$\text{Compressibility:} \quad \frac{1}{C} = \frac{T}{p_a R} < 0.12 \quad (I-3)$$

$$\text{Inertia:} \quad I = \frac{\frac{1}{2} \rho_a U^2}{(T/R)} < 0.05 \quad (I-4)$$

$$\text{Wrap Angle:} \quad \frac{\theta}{\left(\frac{6\mu U}{T} \right)^{1/3}} > 6 \quad (I-5)$$

$$\text{Foil Width:} \quad \frac{\sqrt{tR}}{L} \ll 1 \quad (I-6)$$

$$\text{Mean Free Path:} \quad \frac{\lambda}{R \left(\frac{6\mu U}{T} \right)^{2/3}} < 0.005 \quad (I-7)$$

In the conventional foil journal bearing, 3 pads similar to that illustrated in Figure I-1, retain the journal. Figure I-2, shows schematically, such a three-pad, tension-dominated, foil journal bearing.

In (10), the stiffness of such a bearing, estimated on a quasi-static basis, is provided for a vertical rotor subject to the following assumptions:

- (a) Idealized, pin-support type clamps at the ends of each pad, as illustrated in Figure I-2.
- (b) Fluid inertia is neglected.
- (c) Relaxation of foil tension due to temperature increase with speed is neglected.

Point O is the
No-Load Position on
the Journal Center

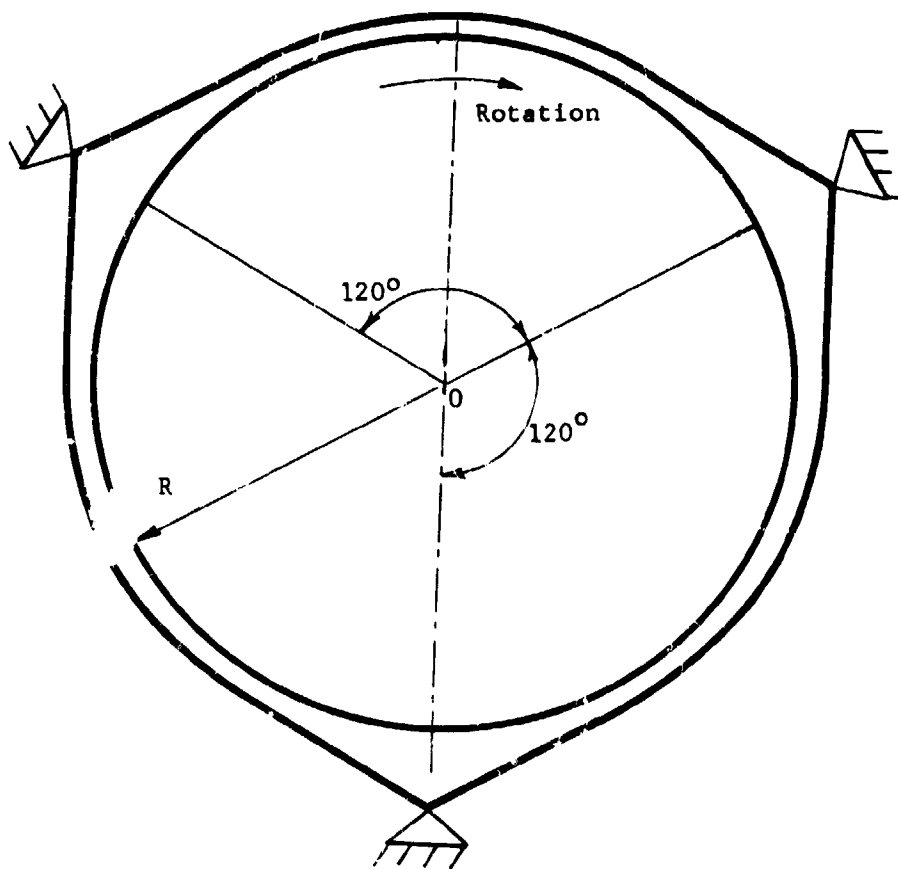


Fig. I-2 Schematic Diagram on a Three-Pad Tension-Dominated Foil Journal Bearing

(d) Slack remaining in the foil due to its finite stiffness is neglected.

The following stiffness equation is then given in (10) for the above conditions:

$$k = \frac{Et}{R} \cdot \frac{6 \sin^2 (\theta/2)}{\frac{\lambda}{R} + \frac{Et}{R} \frac{\partial \delta \lambda}{\partial T} + \frac{\theta Et}{T} \left(\frac{6\mu U}{T} \right)^{2/3} \left(\frac{2}{3} K - \frac{\partial K}{\partial T} \cdot T \right)} \quad (I-8)$$

From this stiffness equation, the natural frequency of a 2.414 lb. rigid rotor support in a tension-dominated foil bearing was calculated and compared with measured data. Figure I-3, also reproduced here from (10) shows this comparison. The curve labelled "Theoretical I" is based on equation I-8, with $\partial \delta \lambda / \partial T$ and $\partial K / \partial T$ taken as zero. The subsequent three curves in Figure I-3 were obtained by sequentially removing the assumptions (a), (b) and (c) noted above. It is seen that the more complete treatment that removes the assumptions shows fairly good agreement with the measurements, at least as regards vertical rotor operation.

Additional design data on tension-dominated foil bearings is provided in (10), including plots of calculated damping and a discussion of the foil resonance phenomenon, which is encountered in this type of bearing at certain excitation frequencies, when the damping goes to zero. This data permits the sizing of tension-dominated foil bearings for some applications but, as previously noted it is presently limited to journal bearings only.

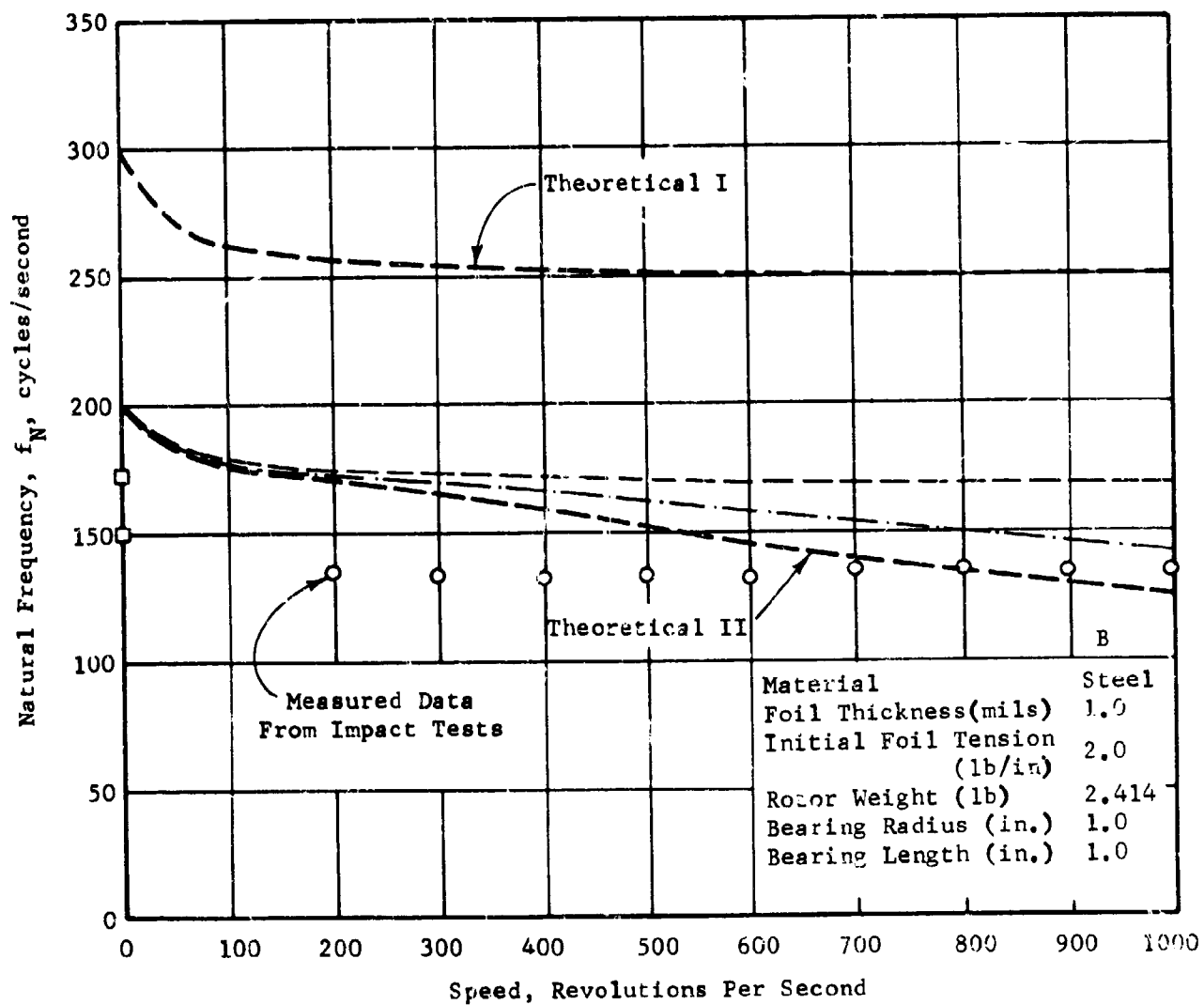


Fig. I-3 "Natural Frequencies" of Foil-Bearing Supported Rotor; - Comparison of Theory and Experiment

APPENDIX II

FOIL BEARING COMPUTER PROGRAMS

Four bearing programs are described in this report. They are the Ridged Journal Bearing Program (HYDRE), the Ridged Thrust Bearing Program (HYDRET), the Leaf Journal Bearing Program (RAP4) and the Leaf Thrust Bearing Program (LTHRST1). They are written for use on the GE Mark II Time Sharing System. They all make use of standard G.E. routines along with some special routines that must be compiled separately. These special routines comprise an integration routine QIN, and interpolation routine YLAG, and a nesting routine RTMI. These are listed in source form as QIN1, YLAG1, and RTMI1, and should be compiled and saved under the names of QIN, YLAG and RTMI respectively.

1. Notes for Using Ridged Journal Bearing Program (HYDRE)

a. Preliminary Preparations

The Ridged Journal Bearing Program makes use of a random binary file named "JBDATA" of dimension (100 x 19) which must be created and saved prior to the initial usage of the program. It is also necessary to create a sequential binary file with the name "JØUFIL". This file is loaded once and for all by running the program STARTUP whose listing is included in this Appendix.

Listing of programs HYDRE and FJØURN2 are given under the source program names HYDKEL and FJØURN3 respectively. These must be compiled and saved in binary forms under the names of HYDRE and FJØURN2. Compilation is necessary as the programs chain to one another.

b. Use of the Program

From a given foil bearing configuration, this program computes load capacity (excluding side-leakage considerations), minimum film thickness, attitude angle and the horsepower needed using air as lubricant.

In each case, the initial inputs are as follows:

1. Length (in.)
2. Diameter (in.)
3. Bump height (in.)
4. Bump length (in.)
5. Bump pitch (S) (in.)
6. Bump thickness (in.)
7. Tape thickness (in.)
8. Clearance (in.)
9. Temperature ($^{\circ}$ F)
10. Ambient pressure (psi)
11. Speed (rpm)
12. E/C (displacement to initial clearance ratio) (e/C)*

* Algebraic notation used here is same as that given for Section II of body of report.

If there is any change to be made in the input items 1 through 12 before the set is assigned to a case number, answer the question "Par No., Value = ?" in true values (e.g., if one wishes to change the bump height to .125 in., the answer should be 3, .125). Repeated changes can be made item by item in a similar fashion until 0 becomes the answer to Par. No in which case the program will continue to print the values of AL, ER, CB and PA, where

$$AL = \alpha_T \text{ (see Equation (43))}$$

$$ER = \text{Eccentricity ratio } E/C$$

$$CB = \alpha_B \text{ (see Equation (44))}$$

$$PA = \frac{1}{\Lambda} \text{ (see Equation (42)) input value}$$

A case number is to be assigned to the set of inputs. If there is any other changes one wishes to make in the inputs, answer YES to the question, "DO YOU WANT TO MODIFY INPUT?" It will then print, "DO YOU WANT TO PERFORM COMPUTATIONS?" If the answer is YES, the program will ask for DX, EXP and IEND, where

DX = Internal precision increment used in Runge-Kutta integration of Equation (51) ($\Delta \xi$)
(A typical value for starting is .02. If the solution is not stable, input MTRIG = 1 to change DX to .01 or .005).

EXP = Relative error on clearance in nesting process.
(A typical value of EXP is 1.E-4).

IEND = Maximum number of iterations to be taken in nesting process.
(A typical value of IEND is 30).

State the case in which the computation is to be done. The program will then ask for C and MTRIG, where C is the constant of integration in Reynold's Equation equivalent to k' in Equation (51), and is roughly of the order of PA. (See Equation (50)), and MTRIG is a control. For MTRIG = 6, the program will print out DP, which is the difference between $\tilde{p}$ at $\xi = 1$, and $\tilde{p} = -1$, where ξ is defined by Equation (48). If the solution is convergent, DP decreases as C decreases and increases as C increases. Otherwise, either the solution is too faraway, in which case make another estimate of C, or input MTRIG = 1 to reduce DX. If both trials are not successful, prepare to change the initial inputs. Obtain lower and upper limits BL and BU of C which give rise to a small negative number and a small positive number for DP in the order of 10^{-2} or less respectively. Proceed to input MTRIG = 6, and then input BL and BU. The program will now go through the nesting process and find the root C, i.e., $\tilde{p}$ for Equation (51), satisfying boundary conditions (55) for $\xi = 1$.

The above procedure may also be considered as a first iteration in satisfying the full 360° foil bearing boundary condition given by Equation (56). Further iteration may be performed by setting MTRIG = 9 which systematically changes PIN. Convergence is obtained when the output value is equal to the input value of PA.

The following parameters are printed out, where

WR = dimensionless radial load parameter (see Equation (38))
WT = dimensionless tangential load parameter (see Equation (38))
W = dimensionless load parameter (see Equation (46))
C = $\tilde{p}$
AA = Attitude angle (DEG.)
PA = $\frac{1}{\Lambda}$ (see Equation (42)) output value
T = dimensionless torque parameter $\tilde{T}$ (see Equation (47))

Input MTRIG = 10 to record the case in the permanent file. An earlier record can be replaced in the permanent file by assigning the same case number to the replacement record. There are a maximum of cases that can be recorded, i.e., from case 1 to case 100. An input of MTRIG = 0 will trigger the question, "DO YOU WANT TO LIST OUTPUT?" One can thus list any of the cases recorded in the permanent file by answering YES to the question, and specify the case number when the program asks. An input of Case No. = 0 will stop or start the program from the beginning. Complete controls of MTRIG are listed below:

MTRIG = 1, to input DX, EXP, IEND
MTRIG = 2, to input AL, ER, CB
MTRIG = 3, to input PA, PIN, where PIN = initial estimate of p at $\xi = 1$
MTRIG = 4, to print WR, WT, W, C, AA, PA bypassing nesting process and march from $\xi = -1$ to $\xi = 1$
MTRIG = 5, to print DP
MTRIG = 6, to input BL, BU
MTRIG = 7, to input NINC, where NINC = Number of tabulated points between each print out. Program then prints out table of ξ , $p(\xi)$, $h(\xi)$
MTRIG = 8, to input C, MTRIG
MTRIG = 9, to perform iteration required to satisfy Equation (56).
MTRIG = 10, to record the case in permanent file
MTRIG = 11, to print WR, WT, W, C, AA, PA and bypass nesting process and march from $\xi = 1$ to $\xi = -1$

The Reynold's Equation (Equation (51)) is generally solved by marching backwards from $\xi = 1$ to $\xi = -1$, which is found to be stable. Once C is determined, the user has the option of bypassing the nesting process and marching in the usual manner by specifying MTRIG = 11 or to march in the opposite directions by specifying MTRIG = 4.

A complete listing of the program along with a sample solution satisfying Equations (55) and (56) are given starting on the next page.

LISTINGS OF SPECIAL ROUTINES
REQUIRED BY FOIL BEARING PROGRAMS

-1-

QINI 08/08/73

```
100 FUNCTION QIN(N1,FUNC,A,B,N)
110 EXTERNAL FUNC
120 ZN=N
130 S=0.
140 H=(B-A)/ZN
150 XL=A
160 XU=A+H
170 DO 5 J=1,N
180 Y=GAUSSQ(N1,FUNC,XL,XU)
190 S=S+Y
200 XL=XL+H
210 5 XU=XU+H
220 QIN=S
230 RETURN
240 END
```

RTM11 08/08/73

```

100 SUBROUTINE RTM1(X,F,FCT,XLI,XRI,EPS,IEND,IER)
105 EXTERNAL FCT
110 IER=0
120 XL=XLI
130 XR=XRI
140 X=XL
150 TOL=X
160 F=FCT(TOL)
170 IF(F)1,16,1
180 1 FL=F
190 X=XR
200 TOL=X
210 F=FCT(TOL)
220 IF(F)2,16,2
230 2 FL
240 IF (1.,FL)-SIGN(1.,FR))25,3,25
250 3 1
260 TOLF=100.*EPS
270 4 I=I+1
280 DO 13 K=1,IEND
290 X=.5*(XL+XR)
300 TOL=X
310 F=FCT(TOL)
320 IF(F)5,16,5
330 5 IF(SIGN(1.,F)+SIGN(1.,FR))7,6,7
340 6 TOL=XL
350 XL=XR
360 XR=TOL
370 7
380 7 FR
390 7 TOL
400 7 TOL=F-FL
410 A=F*TOL
420 A=A+A
430 IF(A-FR*(FR-FL))8,9,9
440 8 IF(I-IEND)17,17,9
450 9 XR=X
460 FR=F
470 TOL=EPS
480 A=ABS(XR)
490 IF(A-1.)11,11,10
500 10 TOL=TOL*A
510 11 IF(ABS(XR-XL)-TOL)12,12,13
520 12 IF(ABS(FR-FL)-TOLF)14,14,13
530 13 CONTINUE
540 IER=1
550 14 IF(ABS(FR)-ABS(FL))16,16,15
560 15 X=XL
570 F=FL
580 16 RETURN

```

RTM11 08/08/73

```
590 17 A=FR-F
600 DX=(X-XL)*FL*(1.+F*(A-TOL)/(A*(FR-FL)))/TOL
610 XM=X
620 FM=F
630 X=XL-DX
640 TOL=X
650 F=FCT(TOL)
660 IF(F)18,16,18
670 18 TOL=EPS
680 A=ABS(X)
690 IF(A-1.)20,20,19
700 19 TOL=TOL*A
710 20 IF(ABS(DX)-TOL)21,21,22
720 21 IF(ABS(F)-TOLF) 16,16,2
730 22 IF(SIGN(1.,F)+SIGN(1.,FL))24,23,24
740 23 XR=X
750 FR=F
760 GU TU 4
770 24 XL=X
780 FL=F
790 XR=XM
800 FR=FM
810 GU TU 4
820 25 IER=2
830 RETURN
840 END
```

YLAG1 08/08/73

```
100*
110*
120*
130 FUNCTION YLAG(XI,X,Y,IND1,N1,IMAX,IEX)
140 DIMENSION X(1000),Y(1000)
150 IND=IND1
160 N=N1
170 IEX=0
180 IF (LE.IMAX) GO TO 1
190 N=IMAX
200 IEX=N
210 1 IF(IND.GT.0) GO TO
220 DO 2 J=1,IMAX
230 IF(XI-X(J))3,13,2
240 2 CONTINUE
250 IEX=1
260 GO TO 7
270 3 IND=J
280 4 CONTINUE
290 IF(IND.GT.1)GO TO 5
300 IEX=-1
310 5 INL=IND-(N+1)/2
320 IF(INL.GT.0) GO TO 6
330 INL=1
340 6 INU=INL+N-1
350 IF(INU.LE.IMAX) GO TO 8
360 7 INL=IMAX-N+1
370 INU=IMAX
380 8 S=0.
390 DO 30 J=INL,INU
400 IF(XI.EQ.X(J)) GO TO 13
410 30 CONTINUE
420 P=1.
430 DO 11 J=INL,INU
440 P=P*(XI-X(J))
450 D=1.
460 DO 10 I=INL,INU
470 IF(I.NE.J) GO TO 9
480 XD=XI
490 GO TO 10
500 9 XD=X(J)
510 10 D=D*(XD-X(I))
520 11 S=S+Y(J)/D
530 YLAG=S*P
540 12 RETURN
550 13 YLAG=Y(J)
560 GO TO 12
570 END
```

LISTING OF PROGRAMS REQUIRED FOR
RIDGED TYPE FOIL JOURNAL BEARINGS

-1-

HYDR=1 08/08/73

```

100 STRING S(16)
110 ALPHA ANS
120 DIMENSION A(19)
130 COMMON ITRIG
140 DATA S/'LENGTH (IN)', 'DIAM. (IN)', 'BUMP HEIGHT (IN)', 'BUMP LENGTH (IN)',
150 &'BUMP PITCH (IN)', 'BUMP THICK. (IN)', 'TAPE THICK. (IN)', 'CLEARANCE (IN)',
160 &'TEMP. (F)', 'AMB. PRES. (PSI)', 'SPEED (RPM)', 'E/C', 'LOAD (LB)',
170 &'MIN. FILM THICK. (IN)', 'ATT. ANG. (DEG)', 'POWER (HP)'/
180 OPENFILE "JBDATA"
190 IF(ITRIG.EQ.1) GO TO 11
200 30 PRINT,"DO YOU WANT TO WRITE INITIAL INPUT"
210 INPUT,ANS
220 IF(ANS.EQ."YES") GO TO 2
230 1 PRINT,"DO YOU WANT TO MODIFY INPUT"
240 INPUT,ANS
250 IF(ANS.NE."YES")GO TO 3
250 PRINT,"OLD CASE NO. ="
270 INPUT,NCASE
280 IF(NCASE.EQ.0) GO TO 8
290 SET("JBDATA") TO NCASE
300 READ ("JBDATA") A
310 GO TO 8
320 2 DO 5 J=1,10,3
330 PRINT 6,J,'. '+S(J)+'',J+1,'. '+S(J+1)+'',J+2,'. '+S(J+2)+' ='
340 5 INPUT,A(J),A(J+1),A(J+2)
350 8 PRINT,"PAR. NO.,VALUE ="
360 6 FORMAT(I3,A,I3,A,I3,A)
370 INPUT,NP,AAA
380 IF(NP.EQ.0) GO TO 9
390 A(NP)=AAA
400 PRINT 90,S(NP)+' ',AAA
410 90 FORMAT(IX,A,1PE13.6//)
420 GO TO 8
430 9 A(19)=0.
440 CALL PARSUB(A)
450 PRINT 20,A(13),A(12),A(14),A(15)
460 20 FORMAT(" AL =",1PE13.6,5X,"ER =",E13.6// " CB =",1PE13.6,5X,"PA =",
470 &E13.6// " NEW CASE NO. =")
480 INPUT,NCASE
490 IF(NCASE.EQ.0) GO TO 1
500 SET("JBDATA") TO NCASE
510 WRITE("JBDATA") (A(J),J=1,15)
520 GO TO 1
530 3 PRINT,"DO YOU WANT TO PERFORM COMPUTATIONS"
540 INPUT,ANS
550 IF(ANS.EQ."YES") CHAIN "FJOURN2"
560 11 PRINT,"DO YOU WANT TO LIST OUTPUT"
565 INPUT,ANS
570 IF(ANS.NE."YES") GO TO 30
580 12 PRINT 18

```

HYDREL 08/08/73

```
590 18 FORMAT(///" CASE NO. =")
600 INPUT,NCASE
610 IF(NCASE.EQ.0) GO TO 30
620 SET("JBDATA")TO NCASE
630 READ("JBDATA") A
640 CALL PARSUB(A)
650 PRINT 23,(S(J),A(J),J=1,16)
660 23 FORMAT(1X,A," =",1PE13.6,5X,A," =",E13.6,5X,A," =",E13.6//)
670 GO TO 12
680 END
690 SUBROUTINE PARSUB(A)
700 DIMENSION A(19)
710 V=AIRV(A(9))
720 DO=3.E7/12./(1.-.25**2)
730 DB=DO*A(6)**3
740 DT=DO*A(7)**3
750 CC=V*A(11)*3.141592654/120.*(A(2)/A(8))**2
760 A(15)=A(10)/6./CC
770 A(13)=6.*CC*A(5)**4/384./A(8)/DT
780 A(14)=CC*A(5)*(A(4)/2.)**3/A(8)/DB
790 IF(A(19).EQ.0.) RETURN
800 A(13)=6.*A(16)*A(2)/2.*A(1)*CC
810 A(14)=A(8)*A(17)
820 A(15)=A(18)
830 A(16)=V*(A(2)/2.)**3*(2.*3.141592654/60.*A(11))**2*A(1)/A(8)/6600.*A(19)
840 RETURN
850 END
860 FUNCTION AIRV(T)
870 TAK=(T+460.)/1.8
880 AIRV=.01709*387.1*1.45E-7/(TAK-114.)*(TAK/273.1)**1.5
890 RETURN
900 END
```

FJOURN3

08/08/73

```

100 OPTION LOAD
105 DIMENSION A(19)
110 DIMENSION P(501),X(501)
115 COMMON ITRIG
120 COMMON/BPIN/PIN,DX/BOUND/BL,BU,
130& EPS,IEND/BPAR/AL,ER,PA,CB,PI/BMT/MTRIG
140 IDM=501
150 PI=3.141592654
152 ITRIG=1
154 OPENFILE "JBDATA"
160 CALL STARTUP
170 1 PRINT,"DX,EPS,IEND ="
180 INPUT,DX,EPS,IEND
182 2 PRINT,"CASE NO. ="
184 INPUT,NCASE
186 IF(NCASE.EQ.0) GO TO 3
188 SET("JBDATA") TO NCASE
190 READ("JBDATA")A
192 PA=A(15);ER=A(12);CB=A(14);AL=A(13);PIN=PA
194 GO TO 8
200 3 PRINT,"AL,ER,CB,PA ="
210 INPUT,AL,ER,CB,PA
220 PIN=PA
230 8 PRINT,"C,MTRIG ="
240 INPUT,C,MTRIG
250 IF(MTRIG.EQ.0) CHAIN "HYDRE"
255 IF(MTRIG.EQ.10) GO TO 15
257 IF(MTRIG.EQ.11) GO TO 4
260 GO TO (1,2,3,4,5,6,7,8,9),MTRIG
270 5 PRINT,"DP =",FNEST(C)
280 PRINT," "
290 GO TO 8
300 6 PRINT,"BL,BU ="
310 INPUT,BL,BU
320 4 CALL HYDRO(X,P,C,WR,WT,W,AA,PAC,TR,IDM)
330 PRINT 10,WR,WT,W,C,AA*180./PI,PAC,TR
340 10 FORMAT(" WR =",1PE13.6,5X,"WT =",E13.6,5X,"W =",E13.6//
350& " C =",1PE13.6,5X,"A.A. =",E13.6,5X,"PA =",E13.6//
355& " T =",1PE13.6///)
360 GO TO 8
370 7 PRINT,"NINC ="
380 INPUT,NINC
390 PRINT 11
400 11 FORMAT(/7X,"X",16X,"P",18X,"H"//)
410 N9=2./DX+1.001
420 PRINT 12,(-X(J),P(J),1.-ER*COS(PI*X(J))+AMAX1(CB*(P(J)-PA),0.),J=1,N9,NINC)
430 12 FORMAT(OPF10.3,1PE20.6,E20.6)
440 PRINT," "
450 GO TO 8
452 9 PIN=PIN*PA/PAC

```

FJ00R13 08/08/73

```
454 GO TO 4
455 IS SET("JBDATA") TO NCASE
456 A(15)=N;A(18)=AA*180./PI;A(19)=TR
457 NN=1./DX+1.001
458 A(17)=1.-ER+CB*(P(NN)-PA)
459 WRITE ("JBDATA") A
460 GO TO 8
461 END
470 SUBROUTINE F2F3(X,F2,F3)
480 COMMON/BF2F3/X1(21),X2(46),X3(19),F21(21),F22(46),F23(19),F31(21)
490& ,F32(46),F33(19)
500 IF(X.GT.0.) GO TO 9
510 F2=1.;F3=1.;RETURN
520 9 IF(X.LE.1.) GO TO 1
530 IF(X.LE.10.) GO TO 2
540 U=1./SQRT(X)
550 IF(X.LE.100.) GO TO 3
560 F2=.392714*U+.123835*U*U+.184384*U**3
570 F3=.294532*U+.0620565*U*U+.0448484*U**3
580 RETURN
590 3 IND1=(X-10.)/5.+2.00001
600 IND=21-IND1
610 F2=YLAG(U,X3,F23,IND,4,19,IEX)
620 F3=YLAG(U,X3,F33,IND,4,19,IEX)
630 RETURN
640 1 IND=20.*X+1.99999
650 F2=YLAG(X,X1,F21,IND,5,21,IEX)
660 F3=YLAG(X,X1,F31,IND,5,21,IEX)
670 RETURN
680 2 IND=5.*(X-1.)+1.9999
690 F2=YLAG(X,X2,F22,IND,5,46,IEX)
700 F3=YLAG(X,X2,F32,IND,5,46,IEX)
710 RETURN
720 END
730 SUBROUTINE STARTUP
740 COMMON/BF2F3/X1(21),X2(46),X3(19),F21(21),F22(46),
750&F23(19),F31(21),F32(46),F33(19)
760 OPENFILE "JOUFIL"
770 READ("JOUFIL")X1,X2,X3,F21,F22,F23,F31,F32,F33
780 CLOSEFILE "JOUFIL"
790 RETURN
800 END
810 FUNCTION H3B(X)
820 IF(X.LE.0.) GO TO 1
830 H3B=1.+1.6*X+1.21904762*X*X+.34099234*X**3
840 RETURN
850 1 H3B=1.
860 RETURN
870 END
880 FUNCTION DIR(P,X)
```

FJOURN3 08/08/73

```

890 COMMON/BCC/CK/BPAR/AL,ER,PA,CB,PI/BITRIG/ITRIG/BMT/MTRIG
900 U=1.-ER*CUS(PI*X)
905 IF(P.GT.PA) U=U+CB*(P-PA)
910 CALL F2F3(AL*(P-PA)/U,F2,F3)
920 DIR=-PI/U/U*(F2-CK*F3/U/P)
930 IF(MTRIG.EQ.40) DIR=-DIR
940 IF(P.LE.0.) ITRIG=1
950 RETURN
960 END
970 SUBROUTINE RUK(C,DX,DP,PC,PN,XO,XN)
980 COMMON/BCC/CI
990 COMMON/BITRIG/ITRIG
1000 CI=C
1010 ITRIG=0
1020 PT=PO
1030 XT=XO
1040 N=DP/DX+.01
1050 DO 5 J=1,N
1060 CK1=DIR(PT,XT)*DX
1070 CK2=DIR(PT+CK1/2.,XT+DX/2.)*DX
1080 CK3=DIR(PT+CK2/2.,XT+DX/2.)*DX
1090 CK4=DIR(PT+CK3,XT+DX)*DX
1100 PT=PT+(CK1+2.*(CK2+CK3)+CK4)/6.
1110 XT=XT+DX
1120 IF(ITRIG.EQ.1) GO TO 6
1130 5 CONTINUE
1140 PN=PT;XN=XT
1150 RETURN
1160 6 XN=XT;PN=XT-1.
1170 RETURN
1180 END
1190 FUNCTION FNEST(C)
1200 COMMON/BPIN/PIN,DX
1210 CALL RUK(C,DX,2.,PIN,PN,-1.,XN)
1220 FNEST=PN-PIN
1230 RETURN
1240 END
1250 SUBROUTINE BACK(P,C,IDIM)
1260 DIMENSION P(IDIM),PI(501)
1270 COMMON/BITRIG/ITRIG/BMT/MTRIG/BPIN/PIN,DX
1280 MTRIG =40
1290 PI(1)=PIN
1300 N8=2./DX+.01;N=N8+1;N1=N
1310 X=-1.
1320 DO 1 J=1,N8
1330 CALL RUK(C,DX,DX,PI(J),PI(J+1),X,X)
1340 IF(ITRIG.NE.1) GO TO 1
1350 N1=J
1360 GO TO 2
1370 1 CONTINUE

```

PJOURN3 08/08/73

```

1380 2 D=1.E8;JMIN=1
1390 DO 5 J=1,N1
1400 D1=ABS(P1(J)-P(N+1-J))
1410 IF(D1.GT.D) GO TO 5
1420 JMIN=J;D=D1
1430 5 CONTINUE
1440 DO 6 K=1,JMIN
1450 6 P(N+1-K)=P1(K)
1460 PRINT,"XMIN =",(N-JMIN)*DX-1., "      DEV =",D
1470 PRINT," "
1480 RETURN
1490 END
1500 SUBROUTINE HYDRO(X,P,C,WR,WT,W,AA,PAC,TR,IDM)
1510 DIMENSION X(IDM),P(IDM),FM3(501)
1515 DIMENSION F11(501)
1520 COMMON/BBOUND/BL,BU,EPS,IEND/BPIN/PIN,DX/BMT/MTRIG/BPAR/
1530&AL,ER,PA,CB,PI
1540 EXTERNAL FNEST
1550 IF(MTRIG.EQ.4) GO TO 1
1555 IF(MTRIG.EQ.11) GO TO 1
1560 CALL RTMI(C,F,FNEST,BL,BU,EPS,IEND,IER)
1570 IF(IER.EQ.1) PRINT,"ROOT NOT CONVERGED"
1580 IF (IER.EQ.2) PRINT,"ROOT NOT NESTED"
1590 1 CONTINUE
1600 N8=2./DX+.01
1610 X(1)=-1.;P(1)=PIN
1620 DO 5 J=1,N8
1630 5 CALL RUK(C,DX,DX,P(J),P(J+1),X(J),X(J+1))
1640 IF(MTRIG.EQ.4) CALL BACK(P,C,IDM)
1650 N9=N8+1
1660 DO 7 J=1,N9
1662 U=0.
1664 IF(P(J).GT.PA) U=CB*(P(J)-PA)
1666 XI=AL*(P(J)-PA)/(1.+U-ER*COS(PI*X(J)))
1668 F11(J)=F1F(XI)
1670 7 FM3(J)=H3B(XI)
1680 P2H3=0.;H3=0.;WR=0.;WT=0.
1685 TR=0.
1690 DO 6 J=2,N8,2
1700 CM=COS(PI*X(J-1));CO=COS(PI*X(J))
1710 CP=COS(PI*X(J+1))
1715 UM=AMAX1(CB*(P(J-1)-PA),0.)
1717 UO=AMAX1(CB*(P(J)-PA),0.)
1718 UP=AMAX1(CB*(P(J+1)-PA),0.)
1720 P2H3=P2H3+P(J-1)**2*(1.-ER*CM+UM)**3*FM3(J-1)+P(J)**2
1730&*(1.-ER*CO+UO)**3*FM3(J)*4.+P(J+1)**2*(1.-ER*CP+UP)**3*FM3(J+1)
1740 H3=H3+(1.-ER*CM+UM)**3*FM3(J-1)+4.*(1.-ER*CO+UO)**3*FM3(J)
1750&+(1.-ER*CP+UP)**3*FM3(J+1)
1760 WR=WR+P(J-1)*CM+4.*P(J)*CO+P(J+1)*CP
1770 WT=WT+P(J-1)*SIN(PI*X(J-1))+4.*P(J)*SIN(PI*X(J))+P(J+1)

```

FJOURN3 08/08/73

```
1780&*SIN(PI*X(J+1))
1785 TR=TR+F11(J-1)/(1.-ER*CM+UM)+4.*F11(J)/(1.-ER*CO+UO)+F11(J+1)/(1.-ER*CP+UP)
1790 & CONTINUE
1800 WR=WR*PI*DX/3.;WT=PI*WT*DX/3.
1805 TR=TR*PI*DX/3.
1810 AA=ATAN(WT/WR)
1820 N=SQRT(WR**2+WT**2)
1830 PAC=SQRT(P2H3/H3)
1840 RETURN
1850 END
1860 FUNCTION FIF(X)
1870 IF(X.LT.1.E-7) FIF=1.
1880 IF(X.LT.1.E-7) RETURN
1890 XL=1./X
1900 R=SQRT(1.+XL)
1910 A=SQRT(2.+2.*R)
1920 Q=SQRT(2.*R-2.)
1930 FIF=XL/2./R*(.5/A*ALOG((1.+A+R)/(1.-A+R))
1940& +1./Q*(ATAN((2.+A)/Q)+ATAN((2.-A)/Q)))
1950 RETURN
1960 END
```

```

100 OPTIMIZE LOAD
110 CUM J1/BF2F3/X1(21),X2(46),X3(19),F21(21),F22(46),F23(19),F31(21),F32(46),F33(19)
120  /3*GRAND/X,N
130 EXTERNAL FGRAND
140 DO 5 J=1,21
150 X1(J)=FLOAT(J-1)*.05
160 X=X1(J)
170 N=2
180 F21(J)=QIN(16,FGRAND,0.,1.,3)
190 J=3
200 5 F31(J)=QIN(16,FGRAND,0.,1.,3)
210 DO 6 J=1,46
220 X=.2*FLOAT(J-1)+1.
230 X2(J)=X
240 J=2
250 F22(J)=QIN(16,FGRAND,0.,1.,5)
260 J=3
270 6 F32(J)=QIN(16,FGRAND,0.,1.,5)
280 DO 7 J=1,19
290 X=5.*FLOAT(J-1)+10.
300 X3(20-J)=1./SQRT(X)
310 N=2
320 F23(20-J)=QIN(16,FGRAND,0.,1.,20)
330 J=3
340 7 F33(20-J)=QIN(16,FGRAND,0.,1.,20)
350 OPENFILE "JOUFIL"
360 WRITE("JOUFIL")X1,X2,X3,F21,F22,F23,F31,F32,F33
370 CLOSEFILE "JOUFIL"
380 STOP
390 END
400 FUNCTION FGRAND(X)
410 CUM J1/BF2F3/X1(21),X2(46),X3(19),F21(21),F22(46),F23(19),F31(21),F32(46),F33(19)
420 FGRAND=1./(1.+E*X*X*(2.-X)**2)**N
430 RETURN
440 END

```

SAMPLE CASE OF RIDGED-TYPE, FOIL JOURNAL BEARING CALCULATIONS

HYDRE 16:51EDT 08/08/73

DO YOU WANT TO WRITE INITIAL INPUT?YES

1. LENGTH (IN), 2. DIAM. (IN), 3. BUMP HEIGHT (IN) =2.85 .75 .03125
 4. BUMP LENGTH (IN), 5. BUMP PITCH (IN), 6. BUMP THICK. (IN) =2.0625 .25 .002
 7. TAPE THICK. (IN), 8. CLEARANCE (IN), 9. TEMP. (F) =2.003 .0004 100.
 10. AMB. PRES. (PSI), 11. SPEED (RPM), 12. E/C =240. 125000. .5

PAR. NO.,VALUE =20 0

AL = 6.689448E+01 ER = 5.000000E-01

CB = 2.822111E+01 PA = 2.112059E-01

NEW CASE NO. =21

DO YOU WANT TO MODIFY INPUT?NO

DO YOU WANT TO PERFORM COMPUTATIONS?YES

DX, EPS, IEND =2.02 1.E-4 30

CASE NO. =21

C,MTRIG =2.2 5

DP = -7.3514385E-02

C,MTRIG =2.25 5

DP = -3.7291840E-02

C,MTRIG =2.3 5

DP = -6.1012506E-03

C,MTRIG =2.32 5

DP = 2.3152307E-04

C,MTRIG =20 6

BL,BU =2.3 .32

WR = 2.989154E-02 WT = 1.745220E-03 W = 2.994244E-02

C = 3.165755E-01 A.A. = 3.341426E+00 PA = 2.216912E-01

T = 4.132936E+00

C,MTRIG =20 10

C,MTRIG =20 0

DO YOU WANT TO LIST OUTPUT?YES

continued on following pages

CASE NO. =?1

LENGTH (IN) = 8.500000E-01 DIAM. (IN) = 7.500000E-01 BUMP HEIGHT (IN) = 3.125000E-02
BUMP LENGTH (IN) = 6.250000E-02 BUMP PITCH (IN) = 2.500000E-01 BUMP THICK. (IN) = 2.000000E-03
TAPE THICK. (IN) = 3.000000E-03 CLEARANCE (IN) = 4.000000E-04 TEMP. (F) = 1.000000E+02
AMB. PRES. (PSI) = 4.000000E+01 SPEED (RPM) = 1.250000E+05 E/C = 5.000000E-01
LOAD (LB) = 1.807554E+00 MIN. FILM THICK. (IN) = 4.155187E-04 ATT. ANG. (DEG) = 3.341426E+00
POWER (HP) = 3.298878E-02

Change Eccentricity

CASE NO. =?0

DO YOU WANT TO WRITE INITIAL INPUT?NO

DO YOU WANT TO MODIFY INPUT?YES

OLD CASE NO. =?1

PAR. NO.,VALUE =?12 .7

E/C = 7.000000E-01

PAR. NO.,VALUE =?0 0

AL = 6.689448E+01 ER = 7.000000E-01

CB = 2.822111E+01 PA = 2.112059E-01

NEW CASE NO. =?2

DO YOU WANT TO MODIFY INPUT?NO

DO YOU WANT TO PERFORM COMPUTATIONS?YES

DX,EPS,IEND =?02 1.E-4 30

CASE NO. =?2

C,MTRIG =?25 5

DP = -5.3247835E-02

C,MTRIG =?45 5

DP = 6.5091047E-03

C,MTRIG = 70 6

BL,BU = 7.25 7.45

WR = 4.419716E-02 WT = 3.858514E-03 W = 4.436528E-02

C = 3.583995E-01 A.A. = 4.989532E+00 PA = 2.278769E-01

T = 3.605566E+00

Print Pressure Profile

C,MTRIG = 70 7

NINC = 710

X	P	H
1.000	2.112059E-01	1.700000E+00
0.800	2.127103E-01	1.608768E+00
0.600	2.178993E-01	1.405206E+00
0.400	2.261571E-01	1.205677E+00
0.200	2.348744E-01	1.101638E+00
-0.000	2.393556E-01	1.094417E+00
-0.200	2.369324E-01	1.159718E+00
-0.400	2.289083E-01	1.283269E+00
-0.600	2.197947E-01	1.458698E+00
-0.800	2.135811E-01	1.633342E+00
-1.000	2.112095E-01	1.700101E+00

C,MTRIG = 70 9 - Iterate to Satisfy Closed, 360° Boundary Condition:

WR = 5.905185E-02 WT = 8.514447E-03 W = 5.966253E-02

C = 3.148587E-01 A.A. = 8.204700E+00 PA = 2.209678E-01

T = 4.022215E+00

C,MTRIG = 70 9

WR = 6.986887E-02 WT = 1.236657E-02 W = 7.095485E-02

C = 2.980918E-01 A.A. = 1.003722E+01 PA = 2.167177E-01

T = 4.188924E+00

C,MTRIG = 70 9

WR = 7.619813E-02 WT = 1.454925E-02 W = 7.757470E-02

C = 2.895460E-01 A.A. = 1.080992E+01 PA = 2.141466E-01

T = 4.276359E+00

C,MTRIG = 70 9

WR = 7.963360E-02 WT = 1.567288E-02 W = 8.116125E-02

C = 2.852036E-01 A.A. = 1.113422E+01 PA = 2.127275E-01

T = 4.323477E+00

C,MTRIG = 70 9

WR = 8.141238E-02 WT = 1.624063E-02 W = 8.301646E-02

C = 2.830018E-01 A.A. = 1.128161E+01 PA = 2.119793E-01

T = 4.347743E 00

C,MTRIG =?0 9

NR = 3.232245E-02 NT = 1.657348E-02 W = 8.397420E-02
C = 2.618557E-01 A.A. = 1.138285E+01 PA = 2.115909E-01
T = 4.359667E+00

C,MTRIG =?0 7

NINC =?10

X	P	H
1.000	1.779034E-01	1.700000E+00
0.800	1.737712E-01	1.566312E+00
0.600	2.041074E-01	1.216312E+00
0.400	2.200587E-01	1.033524E+00
0.200	2.282630E-01	9.150571E-01
-0.000	2.323314E-01	8.961858E-01
-0.200	2.293441E-01	9.455669E-01
-0.400	2.212685E-01	1.057667E+00
-0.600	2.133421E-01	1.276598E+00
-0.800	1.989128E-01	1.566312E+00
-1.000	1.779034E-01	1.700000E+00

C,MTRIG =?

USED 05.68 UNITS

BYE

0095.84 CRU 0000.32 TCH 0003.12 KC

OFF AT 09:13EDT 03/09/73

LISTING OF PROGRAM FOR RIDGED
TYPE, FOIL THRUST BEARINGS

-1-

HYDRA1 06/08/73

```

100 OPTION LOAD
110 STRING S(18)
120 ALPHA ANS
130 DIMENSION A(18)
140 DATA S//LAND WIDTH (IN)`,`U.D. (IN)`,`BUMP HEIGHT (IN)`,`BUMP LENGTH (IN)`,`
150`,`BUMP PITCH (IN)`,`BUMP THICK. (IN)`,`TAPE THICK. (IN)`,`PAD LENGTH (IN)`,`
160`,`INLET LENGTH (IN)`,`
170`,`TEMP. (F)`,`AMB. PRES. (PSI)`,`SPEED (RPM)`,`NO. PADS`,`LOAD (LB)`,`
180`,`MIN. FILM THICK. (IN)`,`BUMP DEFL. (IN)`,`CTR. TAPE DEFL. (IN)`,`POWER (HP)
190 OPENFILE "TBDATA"
200 30 PRINT,"DO YOU WANT TO WRITE INITIAL INPUT"
210 INPUT,ANS
220 IF(ANS.EQ."YES") GO TO 2
230 1 PRINT,"DO YOU WANT TO MODIFY INPUT"
240 INPUT,ANS
250 IF(ANS.NE."YES")GO TO 11
260 PRINT,"OLD CASE NO. ="
270 INPUT,NCASE
280 IF(NCASE.EQ.0) GO TO 3
290 SET("TBDATA") TO NCASE
300 READ ("TBDATA") A
310 GO TO 8
320 2 DO 5 J=1,10,3
330 PRINT 0,J,`,`'+S(J)+`,``,`J+1,`,`'+S(J+1)+`,``,`J+2,`,`'+S(J+2)+`,`' =`
340 5 INPUT,A(J),A(J+1),A(J+2)
350 PRINT,"13. ",S(13)," 14. ",S(14)," ="
360 INPUT,A(13),A(14)
370 6 PRINT,"PAR. NO.,VALUE ="
380 6 FORMAT(I3,A,I3,A,I3,A)
390 INPUT,NP,AAA
400 IF(NP.EQ.0) GO TO 9
410 A(NP)=AAA
420 PRINT 90,S(NP)+`,`' =`,`AAA
430 90 FORMAT(IX,A,1PE13.6//)
440 GO TO 8
450 9 CONTINUE
460 CALL PARSUR(A)
470 PRINT 23,(S(K),A(K),K=15,18)
480 PRINT," "
490 PRINT,"NEW CASE NO. ="
500 INPUT,NCASE
510 IF(NCASE.EQ.0) GO TO 1
520 SET("TBDATA") TO NCASE
530 WRITE("TBDATA") (A(J),J=1,18)
540 GO TO 1
550 11 PRINT,"DO YOU WANT TO LIST OUTPUT"
560 INPUT,ANS
570 IF(ANS.NE."YES") GO TO 30
580 12 PRINT 18
590 18 FORMAT(///" CASE NO. =")

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HYDRET 08/08/73

```

600 INPUT, NCASE
610 IF(NCASE.EQ.0) GO TO 30
620 SET("IBDATA") TO NCASE
630 READ("IBDATA") A
640 PRINT 23, (S(J), A(J), J=1, 18)
650 23 FORMAT(1X, A, " =", 1PE13.6, 5X, A, " =", E13.6, 5X, A, " =", E13.6//)
660 GO TO 12
670 END
680 FUNCTION AIRV(T)
690 TAK=(T+460.)/1.8
700 AIRV=.01709*387.1*1.45E-7/(TAK+114.)*(TAK/273.1)**1.5
710 RETURN
720 END
730 SUBROUTINE PARSUB(A)
740 COMMON/BCOMP/P(41), H(41)
750 DIMENSION A(18)
760 DO=3.E7/12./(1.-.25*.25)
770 PI=3.141592654
780 DT=DO*A(7)**3
790 DB=DO*A(6)**3
800 DM=A(2)-A(1)
810 DI=DM-A(1)
820 A1=(A(2)**2-DI**2)*A(8)/DM/4.
830 PO=A(14)/A(13)/A1+A(11)
840 V=AIRV(A(10))
850 U=A(12)*PI*DM/60.
860 H1=(3.*PI/8.*V*U/(PO-A(11))*A(9)/SQRT(A(3)))* (2./3.)
870 A(15)=H1*YLAG(A(11)/PO, P, H, 0, 4, 41, IEX)
880 A(18)=V/A(15)*A1*A(13)*U*U/6600.
890 A(17)=A(5)**4*(PO-A(11))/384./DT
900 A(16)=(PO-A(11))*A(5)/48./DB*A(4)**3
910 RETURN
920 END
930 BLOCK DATA
940 COMMON/BCOMP/P(41), H(41)
950 DATA P, H/
960& 0.012618, 0.013259, 0.013949, 0.014694, 0.015500, 0.016374, 0.017322, 0.018355,
970& 0.019482, 0.020714, 0.022066, 0.023552, 0.025192, 0.027006, 0.029019, 0.031263,
980& 0.033771, 0.036588, 0.039763, 0.043359, 0.047452, 0.052133, 0.057519, 0.063750,
990& 0.071007, 0.079514, 0.089559, 0.101511, 0.115845, 0.133181, 0.154323, 0.180324,
1000& 0.212555, 0.252781, 0.303241, 0.366691, 0.446383, 0.545948, 0.669140, 0.819502,
1010& 1.000000, 0.250956, 0.255117, 0.259452, 0.263973, 0.268692, 0.273624, 0.278784,
1020& 0.284189, 0.289857, 0.295809, 0.302069, 0.308660, 0.315613, 0.322957, 0.330728,
1030& 0.338967, 0.347717, 0.357029, 0.366959, 0.377572, 0.388942, 0.401152, 0.414299,
1040& 0.428491, 0.443856, 0.460537, 0.478699, 0.498532, 0.520249, 0.544090, 0.570317,
1050& 0.599211, 0.631055, 0.666111, 0.704586, 0.746582, 0.792038, 0.840691, 0.892051,
1060& 0.945429, 1.000000/
1070 END

```

SAMPLE CASE OF RIDGED-TYPE, FOIL THRUST BEARING CALCULATION

HYDRET 09:50EDT 08/09/73

DO YOU WANT TO WRITE INITIAL INPUT?YES

1. LAND WIDTH (IN), 2. O.D. (IN), 3. BUMP HEIGHT (IN) =2.7 3.5 .03125

4. BUMP LENGTH (IN), 5. BUMP PITCH (IN), 6. BUMP THICK. (IN) =2.0625 .125 .002

7. TAPE THICK. (IN), 8. PAD LENGTH (IN), 9. INLET LENGTH (IN) =2.004 .6 .125

10. TEMP. (F), 11. AMB. PRES. (PSI), 12. SPEED (RPM) =2140. 14.7 50000.

13. NO. PADS, 14. LOAD (LB) =212 20.

PAR. NO.,VALUE =20 0

MIN. FILM THICK. (IN) = 2.528857E-04 BUMP DEFL. (IN) = 1.182632E-04 CTR. TAPE DEFL. (IN) = 1.478290E-05

POWER (HP) = 4.691948E-01

NEW CASE NO. =21

DO YOU WANT TO MODIFY INPUT?YES

OLD CASE NO. =21

PAR. NO.,VALUE =214 10.

LOAD (LB) = 1.000000E+01

PAR. NO.,VALUE =20 0

MIN. FILM THICK. (IN) = 4.144061E-04 BUMP DEFL. (IN) = 5.913159E-05 CTR. TAPE DEFL. (IN) = 7.391449E-06

POWER (HP) = 2.863198E-01

NEW CASE NO. =22

DO YOU WANT TO MODIFY INPUT?NO

DO YOU WANT TO LIST OUTPUT?YES

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continued next page

CASE NO. =?1

LAND WIDTH (IN) = 7.000000E-01 U.D. (IN) = 3.500000E+00 BUMP HEIGHT (IN) = 3.125000E-02
BUMP LENGTH (IN) = 6.250000E-02 BUMP PITCH (IN) = 1.250000E-01 BUMP THICK. (IN) = 2.000000E-03
TAPE THICK. (IN) = 4.000000E-03 PAD LENGTH (IN) = 6.000000E-01 INLET LENGTH (IN) = 1.250000E-01
TEMP. (F) = 1.400000E+02 AMB. PRES. (PSI) = 1.470000E+01 SPEED (RPM) = 5.000000E+04
NO. PADS = 1.200000E+01 LOAD (LB) = 2.000000E+01 MIN. FILM THICK. (IN) = 2.528857E-04
BUMP DEFL. (IN) = 1.182632E-04 CTR. TAPE DEFL. (IN) = 1.478290E-05 POWER (HP) = 4.691948E-01

CASE NO. =?2

LAND WIDTH (IN) = 7.000000E-01 U.D. (IN) = 3.500000E+00 BUMP HEIGHT (IN) = 3.125000E-02
BUMP LENGTH (IN) = 6.250000E-02 BUMP PITCH (IN) = 1.250000E-01 BUMP THICK. (IN) = 2.000000E-03
TAPE THICK. (IN) = 4.000000E-03 PAD LENGTH (IN) = 6.000000E-01 INLET LENGTH (IN) = 1.250000E-01
TEMP. (F) = 1.400000E+02 AMB. PRES. (PSI) = 1.470000E+01 SPEED (RPM) = 5.000000E+04
NO. PADS = 1.200000E+01 LOAD (LB) = 1.000000E+01 MIN. FILM THICK. (IN) = 4.144061E-04
BUMP DEFL. (IN) = 5.913159E-05 CTR. TAPE DEFL. (IN) = 7.391449E-06 POWER (HP) = 2.863198E-01

CASE NO. =?

USED 1.98 UNITS

3. Notes for Using Leaf Bearing Programs

The leaf bearing programs are relatively easy to use. No additional preliminary preparation is required.

The inputs to the leaf journal bearing program (KAP4) are as follows*:

1. Bearing length (in.)
2. Diameter (in.)
3. θ_1 (degrees)
4. θ_2 (degrees)
5. Preformed leaf radius, R_L , (in.)
6. Leaf thickness (in.)
7. Ambient pressure (p_a) (lb-in^{-2})
8. Angular velocity, (rpm)
9. Viscosity (Reyns)
10. Preload, δ_o , (in.)
11. Eccentricity, e , (in.)

The output of the program is a repeat of the input together with the additional quantities:

12. Load (lbs)
13. Minimum film thickness (in.)
14. Contact parameter (0 for no wrap, 1 for wrap)
15. Stiffness (lb-in^{-1})
16. Joint clearance (spacing between foil and shaft under static contact at $\theta = \theta_1$).
17. Power (hp)

The inputs to the leaf thrust bearing program (LTHRST1) are as follows:

1. Land width (in.)
2. Outside diameter (in.)
3. Angle subtended by leaf (degrees)
4. Number of leaves
5. Viscosity (Reyns)
6. Ambient pressure (psi)
7. Angular velocity (rpm)
8. Thrust load (lb)
9. α_c

The output of the program consists of a repeat of the input together with the following additional quantities:

* Mathematical notation corresponds to that used for Section II of the report.

10. Film thickness, h_2 , (in.)
11. Optimum flexural rigidity, D_{max} , (in-lb)
12. Torque (in-lb)
13. Power (hp)

If $\bar{W}$ is greater than 1.4, the thrust bearing program will extrapolate and print a diagnostic to that effect.

Complete listings of the leaf journal and thrust bearing programs along with sample runs are given starting on the next page.

- 1 -

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RAP4

08/08/75

```

510 PRINT 24
513 R99=1./(2./A(2)-1./A(5))
515 A(17)=U*U*PI*A(9)*SQRT(2.*R99/A(13))*360./A(4)*A(1)/6600.
520 PRINT 23,(S(J).A(J),J=1,17)
530 23 FORMAT(1X,A,"=",1PE13.6,5X,A,"=",E13.6,5X,A,"=",E13.6//)
540 24 FORMAT(///)
550 PRINT 24
560 15 PRINT,"PAR. NO., VALUE ="
570 INPUT,NP,AAA
580 IF(NP.EQ.0) GO TO 7
590 IF(NP.EQ.-1) GO TO 1
600 IF(NP.EQ.-2) STOP
610 A(NP)=AAA
620 PRINT 90,S(NP)+'= ',AAA
630 90 FORMAT(1X,A,1PE13.6//)
640 GO TO 15
650 END
660 SUBROUTINE LHYDRO(T1,T2,D1,C1,C2,H,DEL)
670 EXTERNAL DMARCH,FUNCI
680 COMMON/BCUN/E1,EP,ER,DT,EMW,C/
690&BDM/F1,FO,PA,PB,EA,EB,EMO
700 COMMON/BFNCI/T1R,T2R,DT1
710 PI=3.141592654
720 T1R=T1*PI/180.;T2R=T2*PI/180.
730 CALL CONSTS(T1*PI/180.,T2*PI/180.,D1)
740 DT1=DT
750 EPO=EP
760 IF(DT.LE.EMW/1.19) GO TO 10
770 CALL RTMI(EP,GO,FUNCI,E1,EPO,1.E-6,30,IER)
780 IF(IER.EQ.2) PRINT,"ROOT NOT NESTED"
790 IF(IER.EQ.1) PRINT,"ROOT NOT CONVERGED"
800 CALL CMARCH
810 IF(DT.LE.PA) EP=EPO
820 IF(DT.LE.PA) GO TO 10
830 XX=EP-E1
840 C1=DT+EMO-FO*EP+F1*XX
850 C2=FO-F1
860 H=.5*C1*XX*XX+C2*XX**3/6.
870 DEL=.5*EMO*EP*EP-FO*EP**3/6.+F1*(EP-E1)**3/6.+ .5*DT*(EPO-EP)**2
880 RETURN
890 10 CONTINUE
900 BL=E1+.01;BU=.99
910 CALL RTMI(EB,FF,DMARCH,BL,BU,1.E-6,30,IER)
920 IF(IER.EQ.1) PRINT,"ROOT NOT CONVERGED"
930 IF(IER.EQ.2) PRINT,"ROOT NOT NESTED"
940 ZZZ=DMARCH(EB)
950 ZZZ=ZZZ
960 C1=0.
970 C2=PA
980 H=PA/6.*(EA-E1)**3

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KAP4 08/08/73

```

990 DEL=.5*EMO*EA*EA-FO*EA**3/6.+F1*(EA-E1)**3/6.+5*DT*(EP-EA)**2
1000 RETURN
1010 END
1020 SUBROUTINE CONSTS(T1,T2,DT)
1030 COMMON/BCUN/E1,EP,ER,DT,EMW,C
1040 C=2.*SIN(T2/2.)/SIN(T2)
1050 E1R=C*SIN(T1-T2/2.)
1060 EPR=SIN(T1)
1070 ER=C*SIN(T1+T2/2.)
1080 E1=E1R/ER
1090 EP=EPR/ER
1100 C=COS(T2)
1110 F1=((1.-EP)**3*C+3.*EP*(C-E1*E1)-(C-E1**3))/
1120& ((C-E1**3)*(1.-C)-(1.-E1)**3*C+3.*(C-E1)*(C-E1*E1))
1130 EMO=EP+F1*(E1-C)
1140 FO=1.+F1*(1.-C)
1150 EMW=-(EMO-FO*EP+F1*(EP-E1))
1160 DT=DT
1170 RETURN
1180 END
1190 FUNCTION DMARCH(EB)
1200 COMMON/BCUN/E1,EP,ER,DT,EMW,C
1210 COMMON/BDM/F1,FO,PA,PB,EA,CB1,EMO
1220 PB=DT/(1.-EB)
1230 PA=1.-PB
1240 F1=PB/C
1250 FO=PA+F1
1260 EA=SQRT((2.*DT*EP-F1*E1*E1)/(FO-F1))
1270 EMO=-DT+FO*EA-F1*(EA-E1)
1280 D11=.5*EMO*EA**2-FO*EA**3/6.+F1*(EA-E1)**3/6.
1290& +(EMO*EA-.5*FO*EA**2+.5*F1*(EA-E1)**2)*(1.-EA)
1300& -DT/2.*(1.-EA)**2+PB/6.*(1.-EB)**3
1310 DE1=.5*EMO*E1**2-FO*E1**3/6.
1320 DMARCH=C*D11-DE1
1330 RETURN
1340 END
1350 FUNCTION FUNC1(EP)
1360 COMMON/BFUNC1/T1,T2,DT
1370 C=2.*SIN(T2/2.)/SIN(T2)
1380 E1R=C*SIN(T1-T2/2.)
1390 EPR=SIN(T1)
1400 ER=C*SIN(T1+T2/2.)
1410 E1=E1R/ER
1420 EP1=EPR/ER
1430 C=COS(T2)
1440 F1=((1.-EP)**3*C+3.*EP*(C-E1*E1)-(C-E1**3))/
1450& ((C-E1**3)*(1.-C)-(1.-E1)**3*C+3.*(C-E1)*(C-E1*E1))
1460 EMO=EP+F1*(E1-C)
1470 FO=1.+F1*(1.-C)
1480 FUNC1=EMO*EP-.5*FO*EP**2+.5*F1*(EP-E1)**2-DT*(EP1-EP)

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```

1490 RETURN
1500 END
1510 SUBROUTINE CMARCH
1520 COMMON/BCUN/E1,EP,ER,DT,EMW,C
1530 COMMON/BDM/F1,FO,PA,PB,EA,EB,EMO
1540 T=((1.-EP)**3*C+3.*EP*(C-E1*E1)-(C-E1**3))/
1550 ((C-E1**3)*(1.-C)-(1.-E1)**3*C+3.*(C-E1)*(C-E1*E1))
1560 EMO=EP+F1*(E1-C)
1570 FO=1.+F1*(1.-C)
1580 PA=-(EMO-FO*EP+F1*(EP-E1))
1590 RETURN
1600 END
1610 FUNCTION GMN(X)
1620 COMMON/BMN/M,N,B
1630 GMN=1./(1.+X*X+B*X*X*X)**N
1640 IF(M.EQ.2) GMN=GMN*X
1650 RETURN
1660 END
1670 FUNCTION GEN(B)
1680 EXTERNAL GMN
1690 COMMON/BMN/M,N,B1
1695 COMMON/BCC/CC,ZB
1700 DIMENSION A(7),S(3,3)
1710 DATA A/O.,.5,1.,5.,10.,100.,1000./
1720 NI=1
1730 B1=B
1740 DO 5 M=1,2
1750 DO 5 N=2,3
1760 S(M,N)=0.
1770 DO 5 K=1,6
1780 S(M,N)=S(M,N)+QIN(16,GMN,A(K),A(K+1),NI)
1790 GEN=S(2,2)-S(1,2)/S(1,3)*S(2,3)
1795 CC=S(1,2)/S(1,3)
1800 RETURN
1810 END
1820 FUNCTION ZNEST(X)
1830 COMMON/BZNEST/CR,WS
1840 ZNEST=WS*X-GEN(CR*SQRT(X))
1850 RETURN
1860 END
1870 SUBROUTINE LOADIS(D0,D1,T1,T2,D9,D10,W1,WI)
1880 DIMENSION SK(20),DC(20),DD(20)
1890 NB=361./T2
2000 PI=3.141592654
2010 M=NB/2
2020 U=NB;V=M;DMN=0.
2030 IF(U/2-V.LE.1.E-6) DMN=1.
2040 M1=M
2050 IF(DMN.EQ.1.) M1=M+1
2070 CALL LHYDRU(T1,T2,10.,C1,C2,HH,DEL)

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10/05/73

```

3070 SK=D9/DEL
3080 DO 5 I=1,M
3090 C=COS((I-1)*PI/130.*T2)
3100 D=D+D0*C
3110 DD(I)=D
3120 C(I)=0.;SK(I)=0.
3130 IF(I.GT.0.) DC(I)=D*C
3140 IF(D.GT.0.) SK(I)=SK0
3150 M=SK(I)*DC(I)
3160 M0=M
3170 DO 6 K=2,M
3180 M=M0+2.*SK(K)*DC(K)
3190 IF(M1.GT.M) M0=M+SK(M1)*DC(M1)
3200 DO 7 I=1,30
3210 DO 8 J=1,M1
3220 IF(DD(J).LT.1.E-5) GO TO 8
3230 CALL LHYDRU(T1,T2,D10/DD(J)/SK(J),C1,C2,H1,DEL)
3240 SK(J)=D9/DEL
3250 8 CONTINUE
3260 M=SK(I)*DC(I)
3270 M0=M
3280 DO 10 K=2,M
3290 M=M0+2.*SK(K)*DC(K)
3300 IF(M1.GT.M) M0=M+SK(M1)*DC(M1)
3310 IF(ABS(M-M0).LT.1.E-2) GO TO 11
3320 M999=M0
3330 10 M0=M
3340 32 PRINT,"DIV",M,M999
3350 11 RETURN
3360 END
3370 FUNCTION PMAX(B)
3380 EXTERNAL CUFN,GMN
3390 COMMON/BCC/C,Z/BMN/M,H,ESB
3400 Z=GEN(B)
3410 Z=B;ESB=B
3420 CALL RTM1(ZM,FF,CUFN,0.,10.,1.E-6,30,IERR)
3430 IF(IERR.NE.0) PRINT,IERR
3440 M=1;N=3
3450 PMAX=C*QIN(16,GMN,0.,ZM,10)
3460 N=2
3470 PMAX=PMAX-QIN(16,GMN,0.,ZM,10)
3480 RETURN
3490 END
3500 FUNCTION CUFN(X)
3510 COMMON/BCC/C,B
3520 CUFN=1.+X*X+B*X**3-C
3530 RETURN;END
3540 BLOCK DATA
3550 COMMON/BCOMP/P(41),H(41)
3560 DATA P,H/

```

RAP4

08/08/73

34808 0.012618,0.013259,0.013949,0.014694,0.015500,0.016374,0.017322,0.018355,
34908 0.019482,0.020714,0.022066,0.023552,0.025192,0.027005,0.029019,0.031263,
35008 0.033771,0.036538,0.039763,0.043359,0.047452,0.052133,0.057519,0.063750,
35108 0.071007,0.079514,0.089559,0.101511,0.115845,0.133181,0.154323,0.180324,
35208 0.212555,0.252781,0.303241,0.366691,0.446383,0.545948,0.669140,0.819502,
35308 1.000000,0.250956,0.255117,0.259452,0.263973,0.268692,0.273624,0.278784,
35408 0.284189,0.289857,0.295809,0.302069,0.308660,0.315613,0.322957,0.330728,
35508 0.338967,0.347717,0.357029,0.366959,0.377572,0.388942,0.401152,0.414299,
35608 0.428491,0.443856,0.460537,0.478699,0.498532,0.520249,0.544090,0.570317,
35708 0.599211,0.631055,0.666111,0.704586,0.746582,0.792038,0.840591,0.892051,
35808 0.945429,1.000000/
3590 END

SAMPLE CASE OF CANTILEVERED LEAF-TYPE, FOIL JOURNAL BEARING CALCULATION

RAP4 09:57EDT 08/09/73

1. LENGTH (IN), 2. DIAM. (IN), 3. TH1 = 1.6 1.5 30.
4. TH2, 5. LEAF RAD. (IN), 6. LEAF THICK. (IN) = 245. .85 .015
7. AMB. PRES. (PSI), 8. SPEED RPH, 9. VIS. (REYNS) = 214.7 50000. 2.9E-9
10. PRELOAD (IN), 11. ECC. (IN) = 2.001 .002

LENGTH (IN) = 1.600000E+00 DIAM. (IN) = 1.500000E+00 TH1 = 3.000000E+01
 TH2 = 4.500000E+01 LEAF RAD. (IN) = 8.500000E-01 LEAF THICK. (IN) = 1.500000E-02
 AMB. PRES. (PSI) = 1.470000E+01 SPEED RPM = 5.000000E+04 VIS. (REYNS) = 2.900000E-09
 PRELOAD (IN) = 1.000000E-03 ECC. (IN) = 2.000000E-03 LOAD (LB.) = 1.750661E+01
 MIN. FILM THICK. (IN) = 2.057563E-05 CON. PAR. = 0. STIFF. (LB/IN) = 1.400529E+04
 JOINT CLEAR. (IN) = 3.319491E-03 POWER (HP) = 2.144925E-01

PAR. NO., VALUE = 211 .005
 ECC. (IN) = 5.000000E-03
 PAR. NO., VALUE = 20 0

LENGTH (IN) = 1.600000E+00 DIAM. (IN) = 1.500000E+00 TH1 = 3.000000E+01
 TH2 = 4.500000E+01 LEAF RAD. (IN) = 8.500000E-01 LEAF THICK. (IN) = 1.500000E-02
 AMB. PRES. (PSI) = 1.470000E+01 SPEED RPM = 5.000000E+04 VIS. (REYNS) = 2.900000E-09
 PRELOAD (IN) = 1.000000E-03 ECC. (IN) = 5.000000E-03 LOAD (LB.) = 3.679311E+01
 MIN. FILM THICK. (IN) = 1.640271E-05 CON. PAR. = 1.000000E+00 STIFF. (LB/IN) = 1.177380E+04
 JOINT CLEAR. (IN) = 1.091359E-03 POWER (HP) = 2.402321E-01

PAR. NO., VALUE = 2
 USED 6.11 UNITS

LISTING OF PROGRAM FOR CANTILEVERED LEAF-TYPE, FOIL THRUST BEARINGS

LTHRST1 08/08/73

```

90 OPTION LOAD
100 STRING S(13)
110 ALPHA ANS
120 DIMENSION A(13)
130 DATA S//LAND WIDTH (IN)', 'O.D. (IN)', 'PAD LENGTH (DEG)', 'NO. PADS', 'VIS. (REYN)', 'AMB. PRE. (PSI)',
140 & 'SPEED (RPM)', 'LOAD (LB)', 'CON. FRAC.', 'FILM THICK. (IN)', 'OPT. FLEX. RIG. (IN-LB)', 'TORQUE (IN-LB)', 'POWER (HP)'//
150 PI=3.141592654
160 1 CONTINUE
170 DO 5 J=1,7,3
180 PRINT 6,J,'. 'S(J)+'',J+1,'. 'S(J+1)+'',J+2,'. 'S(J+2)+' ' = '
190 5 INPUT,A(J),A(J+1),A(J+2)
200 6 FORMAT(I3,A,I3,A,I3,A)
210 7 CONTINUE
220 RM=(A(2)-A(1))/2.
230 B=RM*PI*A(3)/180.
235 CF=A(9)
240 B1=B*CF
250 N=A(8)/A(4)
260 U=PI*A(7)*2.*RM/60.
270 CALL LOPT(N/B1/A(1)/A(6),AMAX,F1,XLAM,CF)
280 A(10)=SQRT(A(5)+6.*U*B1/A(6)/XLAM)
290 DM=A(10)*(AMAX-1.)
300 A(11)=B**3/DM*N/A(1)/24.
310 A(13)=F1*U*A(5)*B1*A(1)/A(10)/6600.*A(4)
315 A(12)=A(13)*6600./U*RM
320 PRINT 24
330 24 FORMAT(///)
340 PRINT 23,(S(J),A(J),J=1,13)
350 23 FORMAT(1X,A," =",1PE13.6,5X,A," =",E13.6,5X,A," =",E13.6//)
360 PRINT 24
370 15 PRINT,"PAR. NO., VALUE ="
380 INPUT,NP,AAA
390 IF(NP.EQ.0) GO TO 7
400 IF(NP.EQ.-1) GO TO 1
410 IF(NP.EQ.-2) STOP
420 A(NP)=AAA
430 PRINT 90,S(NP)+' ' = ',AAA
440 90 FORMAT(1X,A,1PE13.6//)
450 GO TO 15
460 END
470 SUBROUTINE LOPT(W1,AMAX,F1,XLAM,CF)
480 DIMENSION W1(15),AMAX1(15),XLAM1(15)
490 DATA W1/0.,.1,.2,.3,.4,.5,.6,.7,.8,.9,1.,1.1,1.2,1.3,1.4/
500 DATA XLAM1/0.,.3,5,8.5,15.,22.5,33.,41.,67.,84.,115.,152.5,198.,257.,338.,431./
510 DATA AMAX1/2.2,2.46,2.75,3.06,3.4,3.3,4.27,4.62,5.48,6.22,7.,7.92,8.92,10.06,11.33/
540 NPT=3
550 AMAX=YLAG(W1,W1,AMAX1,0,NPT,15,1EX)
560 XLAM=YLAG(W1,W1,XLAM1,0,NPT,15,1EX)
570 F1=(1.-CF)/CF+(ALOG(AMAX))/(AMAX-1.)
590 IF(W1.GT.1.4.) PRINT, " W1 GT 1.4. EXTRAPOLATION USED"
600 RETURN
610 END

```

SAMPLE CASE OF CANTILEVERED-TYPE FOIL, THRUST BEARING CALCULATIONS

OLD LTHRST

READY
RUN

LTHRST 15:49EDT 08/16/73

1. LAND WIDTH (IN), 2. O.D. (IN), 3. PAD LENGTH (DEG) = 7.7 3.5 29.
4. NO. PADS, 5. VIS. (REYNS), 6. AMB.PRE. (PSI) = 212 2.9E-9 14.7
7. SPEED (RPM), 8. LOAD (LB), 9. CUN. FRAC. = 60000. 20. .5

LAND WIDTH (IN) = 7.000000E-01 O.D. (IN) = 3.500000E+00 PAD LENGTH (DEG) = 2.900000E+01
NO. PADS = 1.200000E+01 VIS. (REYNS) = 2.900000E-09 AMB.PRE. (PSI) = 1.470000E+01
SPEED (RPM) = 6.000000E+04 LOAD (LB) = 2.000000E+01 CUN. FRAC. = 5.000000E-01
FILM THICK. (IN) = 3.62131E-04 OPT. FLEX. RIG. (IN-LB) = 3.718737E+01 TORQUE (IN-LB) = 4.376138E-01
POWER (HP) = 4.166121E-01

PAR. NO., VALUE = 28 70.

LOAD (LB) = 7.000000E+01

PAR. NO., VALUE = 20 0

W1 ST 1.4, EXTRAPOLATION USED

LAND WIDTH (IN) = 7.000000E-01 O.D. (IN) = 3.500000E+00 PAD LENGTH (DEG) = 2.900000E+01
NO. PADS = 1.200000E+01 VIS. (REYNS) = 2.900000E-09 AMB.PRE. (PSI) = 1.470000E+01
SPEED (RPM) = 6.000000E+04 LOAD (LB) = 7.000000E+01 CUN. FRAC. = 5.000000E-01
FILM THICK. (IN) = 7.516020E-05 OPT. FLEX. RIG. (IN-LB) = 1.239549E+02 TORQUE (IN-LB) = 1.627562E+01
POWER (HP) = 1.616081E+00

PAR. NO., VALUE = 2-2 0.

PROGRAM STOP AT 410

USED 1.18 UNITS

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