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STUDIES ON THE DYNAMIC CHARACTERISTICS OF GAS FILM BEARINGS AND DAMPERS

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Studies on the Dynamic Characteristics

of Gas Film Bearings and Dampers

by

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Final Report

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Preface

This final report contains only a brief summary of several independent studies conducted over the last three years on the general topic: stiffness and damping characteristics of inherently compensated gas film bearings. The major portion of the work has been presented in greater detail as master of science theses by David M. Smith and Ricardo R. Tapia under the direction of A. Kent Stiffler, principal investigator. All published material that resulted from this grant is denoted by an astrisk in the reference section of this report.

Summary

The dynamic characteristics of inherently compensated gas film bearings have been investigated for small excursion ratios. Both circular and rectangular cases have been solved for the stiffness and damping as a function of supply pressure, restrictor coefficient, and squeeze number. The effect of disturbance amplitude has been studied for the inherently compensated strip. Analytical solutions for the simple gas film damper problem have established the effect of disturbance amplitude at low squeeze numbers. These results are applicable to pressurized bearings as limiting case of the restrictor coefficient.

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Introduction

The dynamic response of a flexible rotor in rigid supports is certainly not a new problem. However, the solution to the more realistic problem of elastic rotors in elastic bearing supports is somewhat more recent. Given the stiffness and damping properties of the supports, computer programs can be developed to predict the response of unbalanced rotors with some certainty [1][2]. In general the support stiffness can serve to reduce the critical speeds below operating conditions and reduce transmitted forces to the bearings [3]. Support damping can serve to attenuate the response when operated through critical speeds and to reduce when operated through critical speeds and to reduce the effect of shock loading.

Although lubricating oils are commonly employed in bearings and dampers, they are inadequate for high temperature environments. Thus, there is a growing interest in air lubricated systems. A limited amount of information is available to the designer concerned with the dynamic properties of externally pressurized gas films. A literature review suggests that analyses pertain to two types of bearings: (a) pocket-type, orifice compensated [4-7] and (b) annular-type, inherently compensated [7,8]. Both lumped and distributed parameter methods are employed. In all cases the analyses are presented for several one-dimensional bearings. In general, the pocket type exhibits poor stability and high stiffness while the inherent type exhibits good stability and low stiffness. It is the superior stability qualities of the inherently compensated design that has lead to its greater acceptance.

The purpose of the research effort was to provide more design information on inherently compensated, gas film bearings and dampers. When the film is externally pressurized the designation: "bearing" as opposed to "damper" is generally prevalent in the literature. However, it is understood that pressurized films may be used primarily as dampers and labeled as such. The bearings are a thrust type with a series of inherent feed holes. These holes are usually replaced by a line source. The film gap is displaced periodically and the resulting forces analyzed. The particular geometries of interest are shown in Figure 1. The results of this research can be found in references [9-15]. The following brief summary is presented.

Summary

Gas film pressures are described by the non-linear Reynolds equation:

$$\frac{\partial^2 p^2}{\partial x^2} + \frac{\partial^2 p^2}{\partial z^2} = \frac{2\sigma}{h^3} \frac{\partial}{\partial t} (ph) \tag{1}$$

where the variables are normalized by the ambient pressure p_a^*, bearing characteristic length L^*, mean film thickness h_o^* and the excitation frequency ω^*. The squeeze number is defined by

$$\sigma = \frac{12\mu\omega^* L^{*2}}{h_o^{*2} p_a^*} \tag{2}$$

and the film disphacement h is given as

$$h = 1 + \varepsilon \sin t \quad . \tag{3}$$

Integration of the pressure field establishes the time dependent load capacity. The component of force in phase with motion h is defined as the stiffness; the component of force 90° out of phase with the motion is defined as the damping.

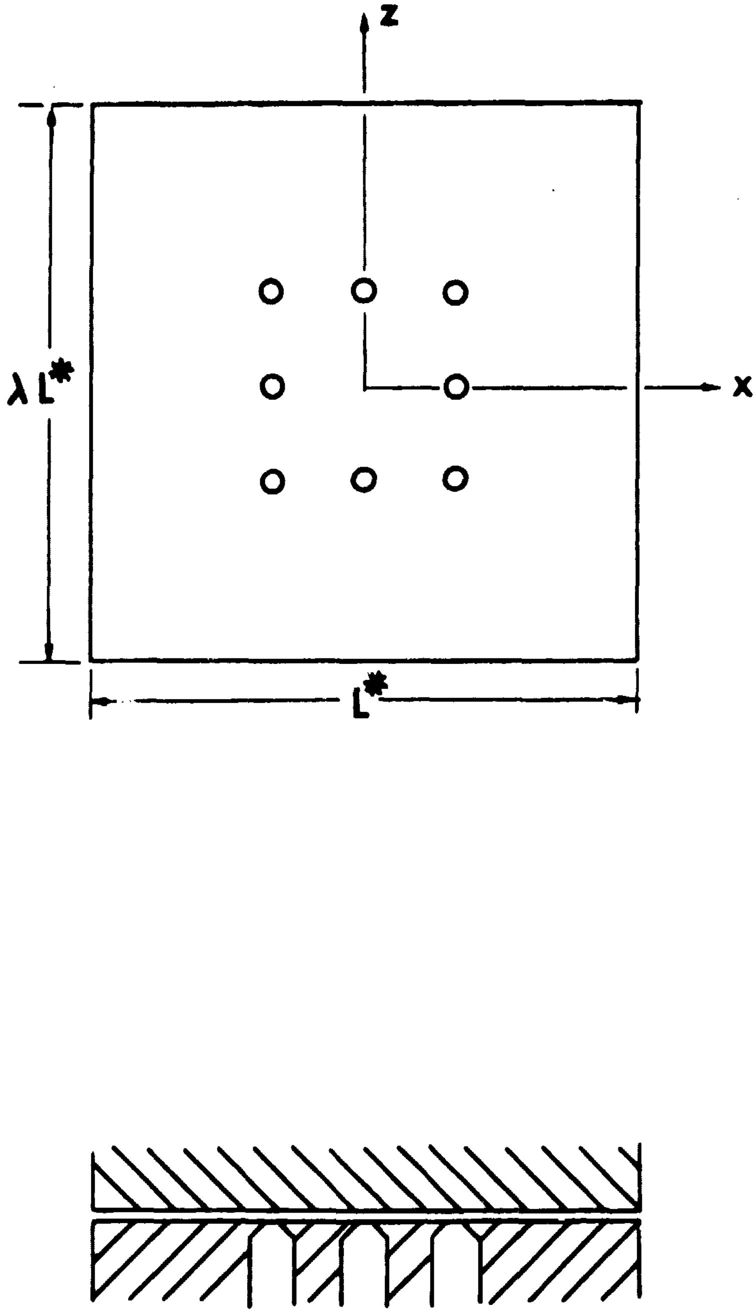

Figure 1. Inherently Compensated, Multiple-Inlet, Rectangular Thrust Bearing.

No analytical solutions to Equation (1) exist, even for the one-dimensional spacial problem. Thus, one must resort to finite-difference iterative solutions using digital computers. Non-linear, three-dimensional (time and space) partial differential equations can have iterative convergence problems. Furthermore, the boundary conditions include the time dependent flow of compressible gases through orifices. The orifice equations are not only non-linear but during one cycle the flow can switch from sub-critical to critical and can switch directions! Then of course the following parameters must be investigated: λ, ratio of bearing length to width; r, ratio of central length to characteristic length, p_s, supply pressure; σ, squeeze number; ε, excursion ratio; Λ, restrictor coefficient, a measure of the flow resistance through the bearing to the flow resistance through the bearing to the flow resistance through the orifice. The problem can be made manageable by (a) parameter perturbation solutions or (b) reducing the dimensions.

Parameter Perturbation

Both the circular [9] and the rectangular [10][11] shaped bearings have been solved by seeking a solution in the form:

$$p(x,z,t) = p_1(x,z) + \varepsilon p_2(x,z,t) \quad , \tag{4}$$

The effect of the perturbed solution in ε is to linearize the equations although the solution is valid only for "small" excursion ratios. A brief account is given for the rectangular bearing since it is more complete.

When Equations (3) and (4) are substituted into Equation (1), two equations are obtained from the terms of order O(1) and O(ε), respectively:

$$O(1) \qquad \frac{\partial^2 p_1^2}{\partial x^2} + \frac{\partial^2 p_1^2}{\partial z^2} = 0 \tag{5}$$

$$O(\varepsilon) \qquad \frac{\partial^2 g}{\partial x^2} + \frac{\partial^2 g}{\partial z^2} = \frac{\sigma}{p_1} [p_1{}^2 \cos t + \frac{\partial g}{\partial t}] \tag{6}$$

where

$$g(x,z,t) = p_1 p_2 \quad . \tag{7}$$

The load disturbance is assumed to be periodic. Thus, time may be eliminated from Equation (6) by assuming the solution:

$$g = g_1(x,z) \sin t + g_2(x,z) \cos t \quad . \tag{8}$$

The equations to be solved are coupled linear second order partial differential equations in g_1 and g_2 subject to the appropriate boundary conditions [10].

The dynamic load is defined by

$$W_2 = \iint p_2(x,z,t) dx dz \tag{9}$$

and can be written in the form

$$W_2 = C \sin t + B \cos t \quad . \tag{10}$$

If the bearing executes small harmonic motion,

$$\varepsilon W_2{}^* = -K^* y^* - D^* dy^*/dt \tag{11}$$

where K^* and D^* are the stiffness and damping constants, respectively and $y^* = \varepsilon h_o \sin \omega^* t^*$. Thus,

$$W_2 = \frac{W_2{}^*}{\lambda p_a{}^* (L^*)^2} = - \frac{K^* h_o}{\lambda p_a{}^* (L^*)^2} \sin t - \frac{D^* h_o \omega^*}{\lambda p_a{}^* (L^*)^2} \cos t \tag{12}$$

a dimensionless stiffness and damping can be defined by

$$K_s = - \frac{C}{p_s - 1} = \frac{K^* h_o}{\lambda p_a{}^* (L^*)^2 (p_s - 1)} \tag{13}$$

$$D = -\frac{12B}{\sigma} = \frac{D^*}{\lambda L^*(L^*/h_o)^3\mu} \qquad (14)$$

Typical results are shown in Figures 2-5. The restrictor coefficient

$$\Lambda = \frac{6C_d N\pi d_o \mu(2g_o kRT)^{1/2}}{p_a{}^* p_s h_o{}^{*2} F(k-1)^{1/2}}$$

where F = .83, 3.48, 8.44 for r = .4, .6, .8 respectively. Experimental studies of this bearing configuration have been reported by Cunningham [16].

Summary and Optimum Design

The first decision to be made in the design of the bearing is the choice between optimum stiffness and optimum damping. Stiffness is usually the first choice since a load disturbance can lead to closure of the bearing if it is too "soft". Furthermore, natural frequencies of the bearing-load system, which depend on the stiffness, must be avoided. However, damping is necessary when disturbances are present, and many bearings are designed primarily as film dampers. The design procedure is as follows:

1. For maximum stiffness, select a restrictor coefficient in the range $1 < \Lambda < 2$. Generally, a larger ratio of central length to bearing length r and a higher supply pressure increase stiffness, but these choices must be weighed against the supply requirements associated with the increased mass flow and against stability considerations.

2. The choice of damping is dependent upon the minimum allowable stiffness. Low supply pressures (p_s = 1.5,2) provide high damping for low values of the restrictor coefficient; however, there is a considerable decrease in stiffness. At high supply pressures, damping increases for larger values of the restrictor coefficient. Supply pressure has little effect on damping at higher values of restrictor coefficient, but a higher supply pressure will improve the corresponding stiffness. There are two choices of damping which can be made:

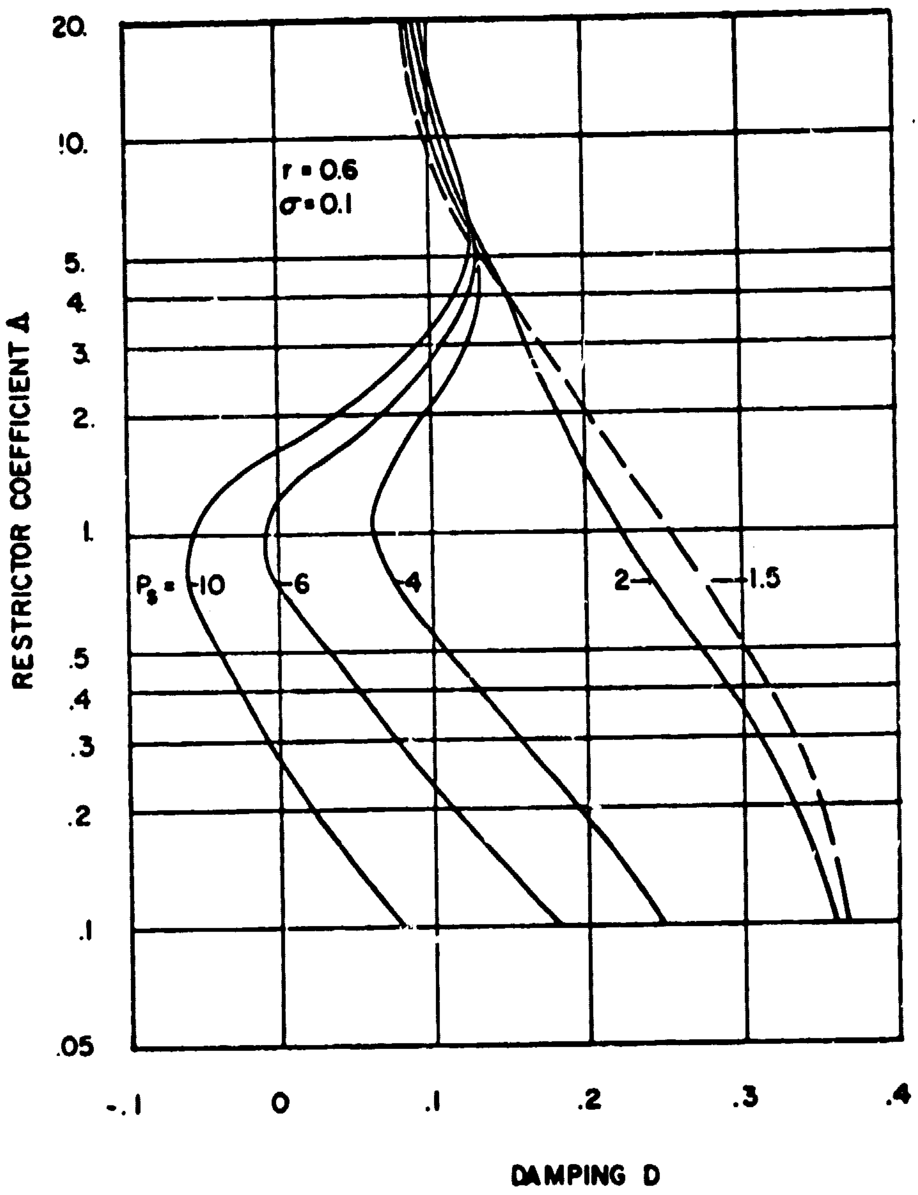

Figure 2. Dimensionless Damping versus Restrictor Coefficient ($r = 0.6, \lambda = 1, \sigma = 0.1$)

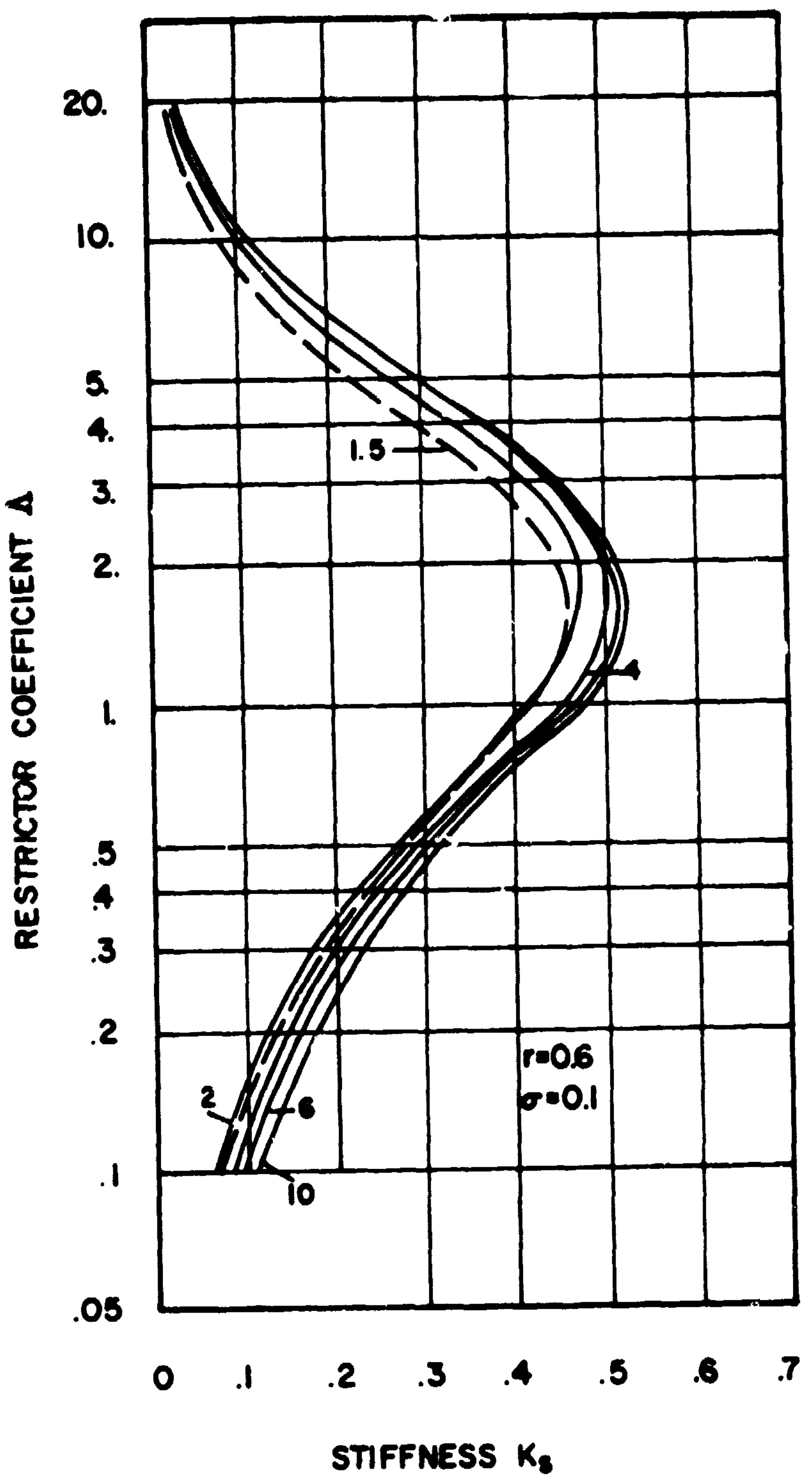

Figure 3. Dimensionless Stiffness versus Restrictor Coefficient ($r = 0.6$, $\lambda = 1$, $\sigma = 0.1$)

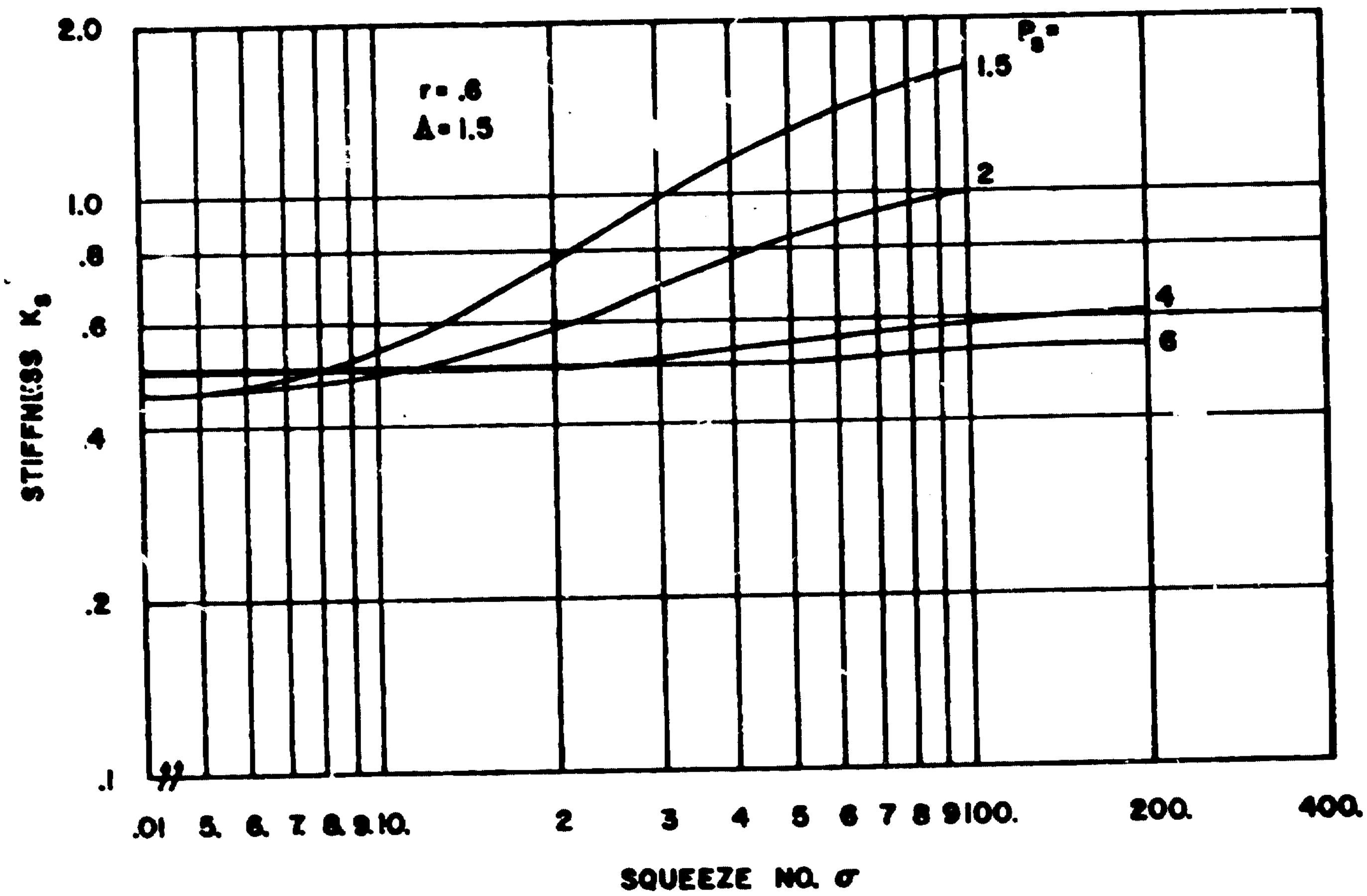

Figure 4. Normalized Stiffness versus Squeeze Number
(Λ = 1.5, r = 0.6, λ = 1)

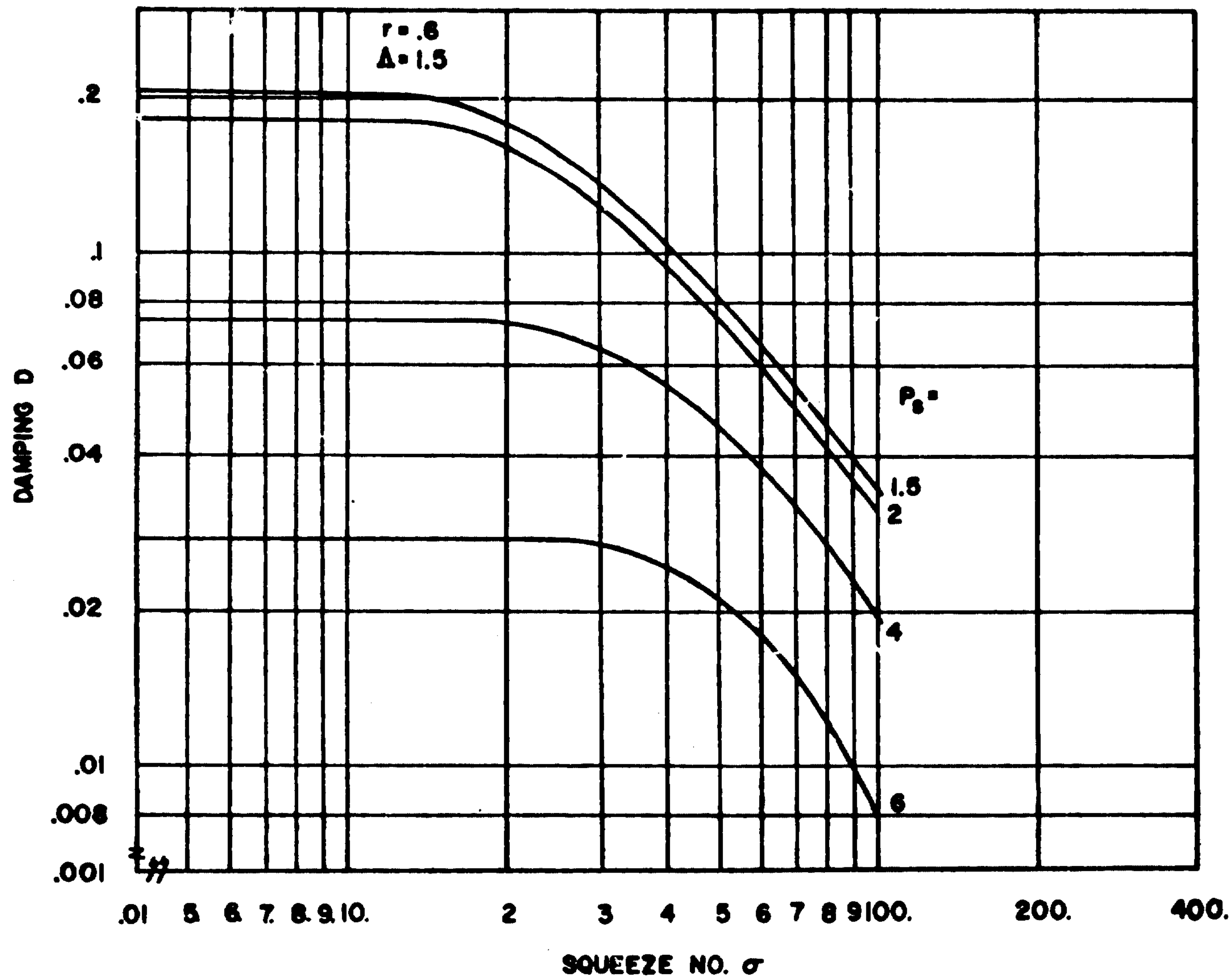

Figure 5. Normalized Damping versus Squeeze Number
(Λ = 1.5, r = 0.6, λ = 1)

(a) if low values of stiffness are acceptable, choose a low supply pressure and a restrictor coefficient in the range $1 \leq \Lambda \leq 2$;

(b) if higher values of stiffness are needed, choose a high supply pressure and the restrictor coefficient, $\Lambda = 5$.

3. The choice of restrictor coefficient fixes the dimensionless load capacity. Once the load is specified, the supp[illegible] pressure determines the bearing dimensions.

It is observed that the dimensionless stiffness and damping are a function of the film thickness. The actual stiffness and damping are improved by the selection of small film thicknesses. Thus, for a fixed restri[illegible]r coefficient the film thickness can be made arbitrarily small by reducing the inlet area of the orifices. In this respect the designer is limited by the minimum allowable clearance for the bearing.

Amplitude Effects

The effect of excursion amplitude ratio ε on gas thrust type bearings similar to Figure 1 is not understood. In fact the non-linear nature of the Reynolds equation has prevented a complete analytical solution for the simple non-pressurized gas film damper. Salbu [17] investigated the mean load capacity of an unpressurized disk subjected to large excursions in film thickness. The time dependent load for any shape damper approaches

$$W(t) = \frac{W^*(t)}{p_a^* \text{ Area}} = \frac{(1 + \frac{3}{2}\varepsilon)^{1/2}}{h} - 1 \tag{19}$$

as the squeeze number $\sigma \to \infty$ [17]. It was only recently that the author [12] solved the case in which $\sigma \ll 1$. For the strip

$$W(t) = -\frac{1}{3}\frac{\dot{h}}{h^3}\sigma + \frac{1}{15}\left[\frac{\ddot{h}}{h^5} - \frac{5}{2}\frac{\dot{h}^2}{h^6}\right]\sigma^2 \quad , \tag{20}$$

The above results are important since they provide limits $(0 \leftarrow \Lambda \to \infty)$ to the damping performance of externally-pressurized gas bearings [14].

In order to understand the effect of disturbance amplitude on the dynamic performance of inherently compensated gas bearings, a study of Equation (1), as it pertains to the one-dimensional strip, was undertaken, Figure 6. It was necessary to retain the non-linear form of Equation (1) together with the non-linear boundary conditions associated with the severe gas flow switching sequences referred to previously. These non-linearities produced converge difficulties for the finite difference methods used; thus, excursion ratios were limited to $\varepsilon \leq 0.5$.

The time dependent load capacity is given by

$$W(t) = \int [p(x,t) - 1] dx \tag{21}$$

The load capacity as a function of time can be expressed in the form of the Fourier series:

$$W(t) = A_o + A_1 \cos t + B_1 \sin t + A_2 \cos 2t + B_2 \sin 2t + \ldots \tag{22}$$

Using the orthogonality properties the coefficients are given by

$$\begin{aligned} A_o &= \frac{1}{2\pi} \int_0^{2\pi} W(t)dt \\ A_N &= \frac{1}{\pi} \int_0^{2\pi} W(t) \cos(Nt)dt \\ B_N &= \frac{1}{\pi} \int_0^{2\pi} W(t) \sin(Nt)dt \quad . \end{aligned} \tag{23}$$

These coefficients are determined by numerical integration of Equations (21)(23) over space and time. The results are tabulated in reference [13].

Important parameters in design are the linearized stiffness and damping defined by Equation (11). The results for low squeeze numbers are presented in Figures 7-12 as a function of supply pressure and restrictor coefficient

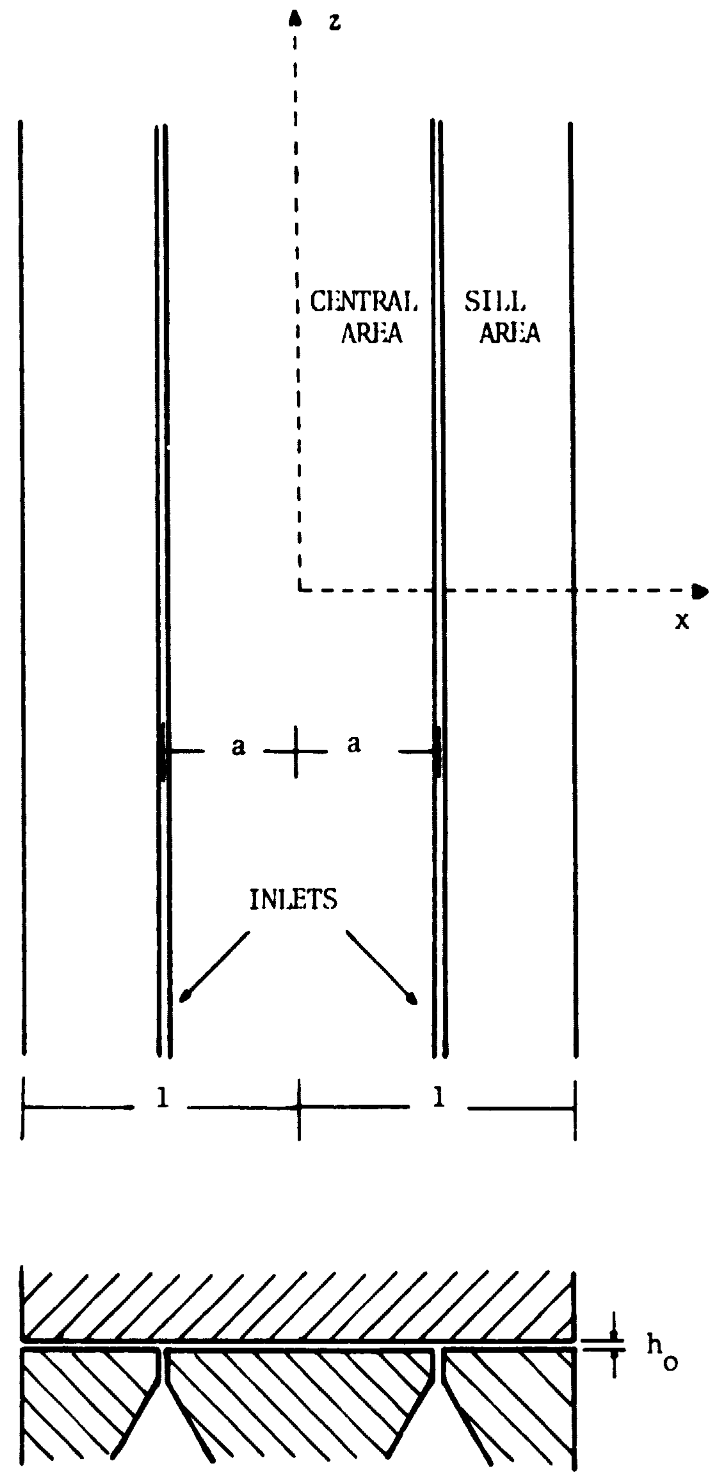

Figure 6. Inherently Compensated Strip Thrust Bearing

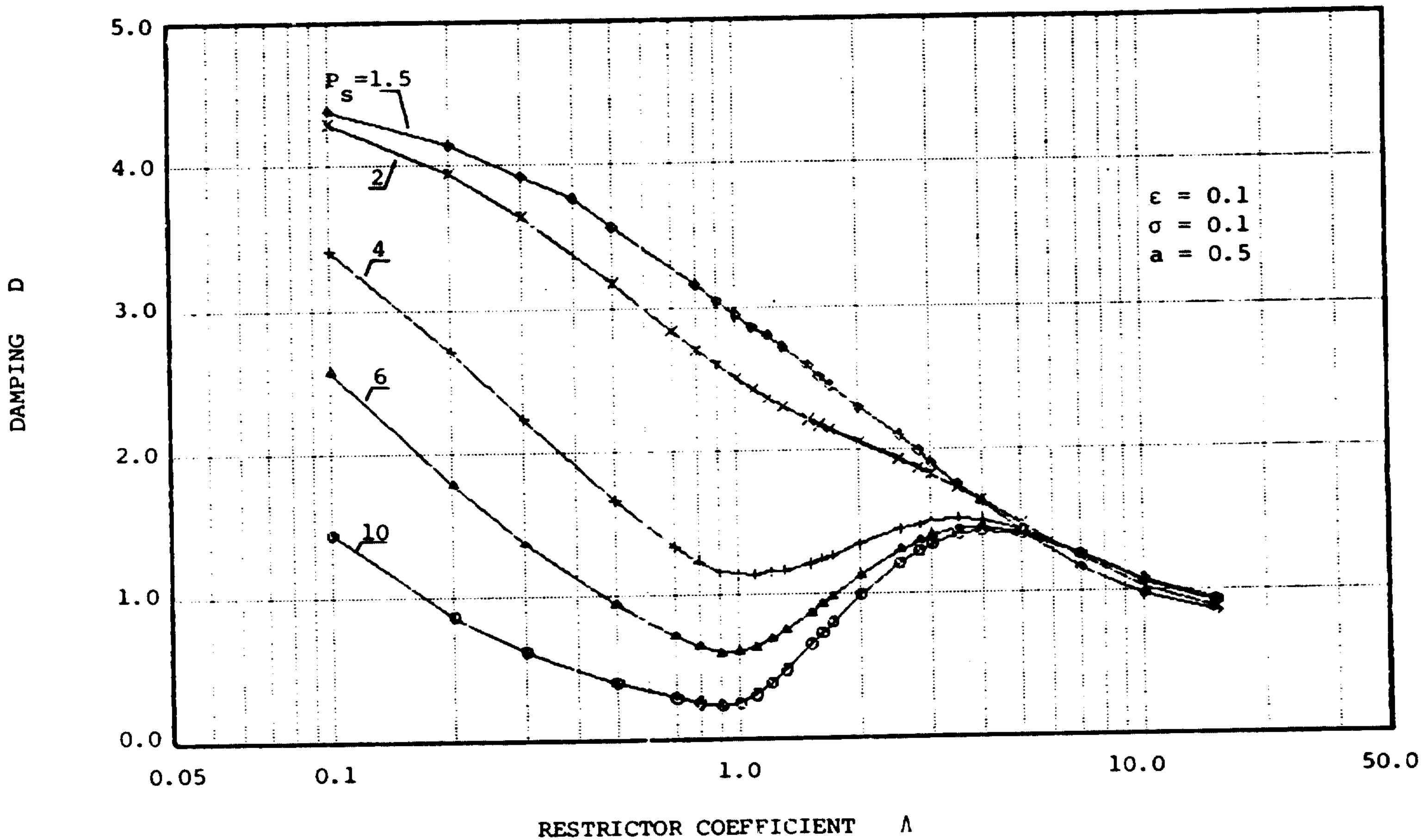

Figure 7. Dimensionless Damping versus Restrictor Coefficient (ε = 0.1, σ = 0.1, a = 0.5)

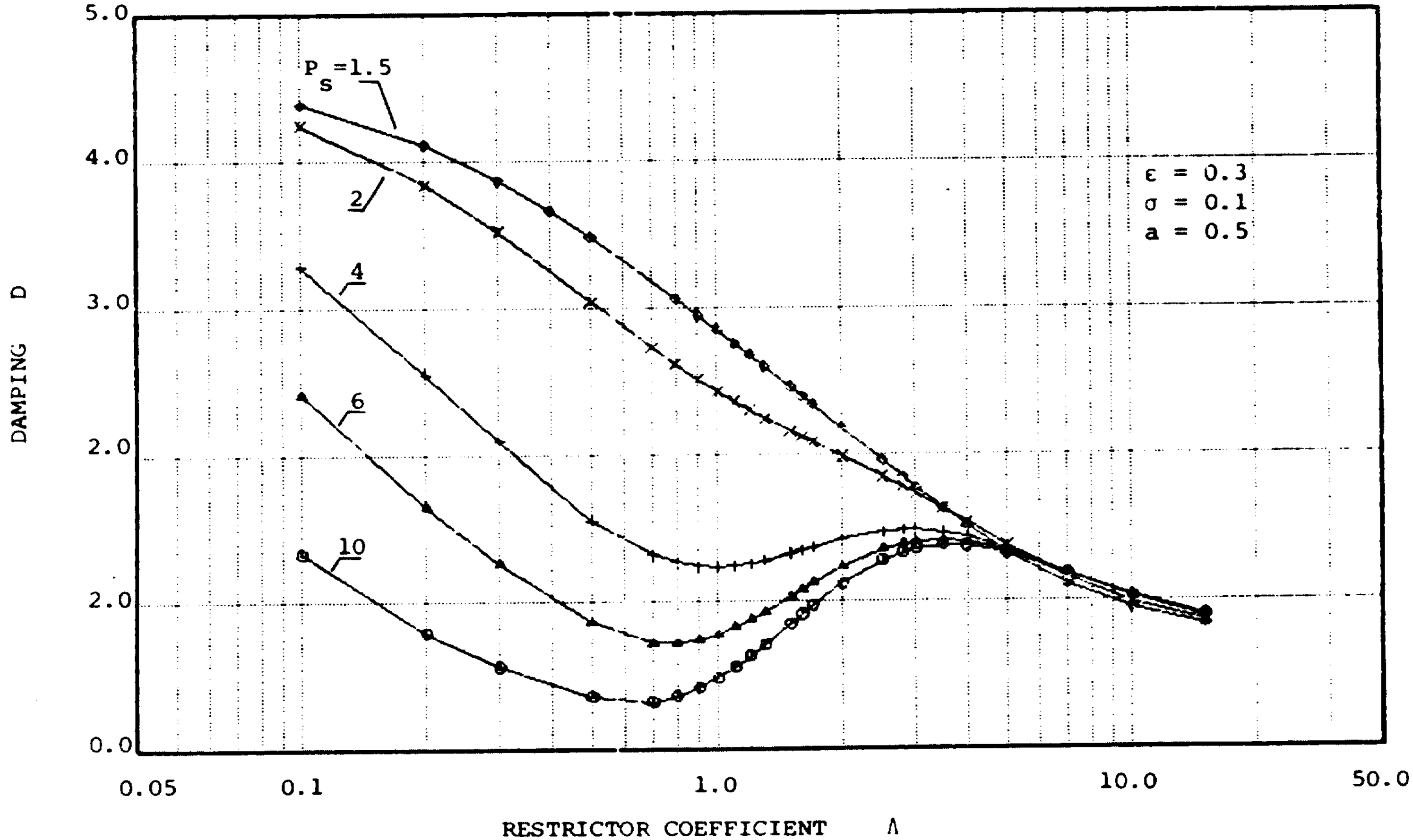

Figure 8. Dimensionless Damping versus Restrictor Coefficient (ϵ = 0.3, σ = 0.1, a = 0.5)

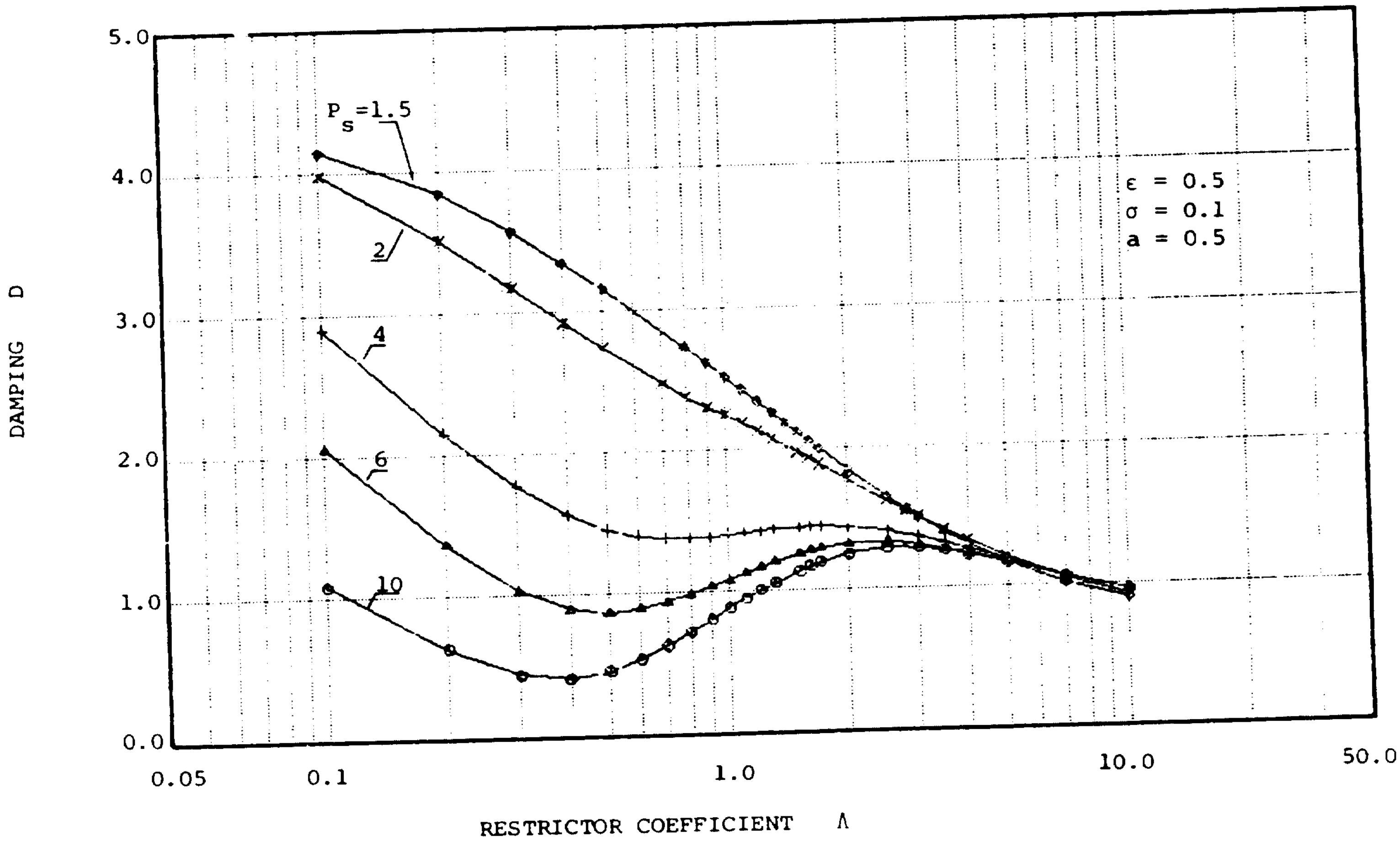

Figure 9. Dimensionless Damping versus Restrictor Coefficient (ε = 0.5, σ = 0.1, a = 0.5)

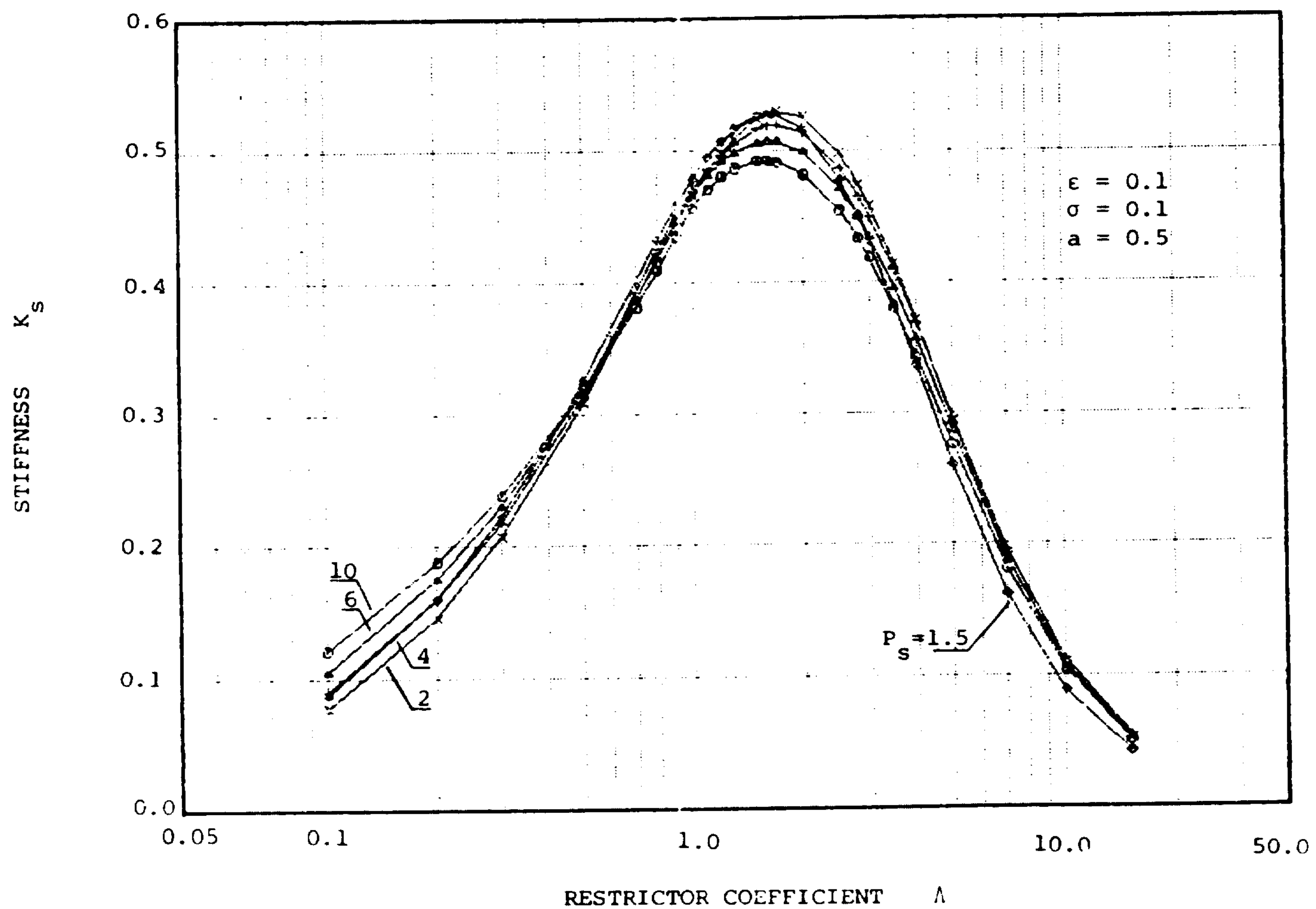

Figure 10. Dimensionless Stiffness versus Restrictor Coefficient (ε=0.1, σ=0.1, a=0.5)

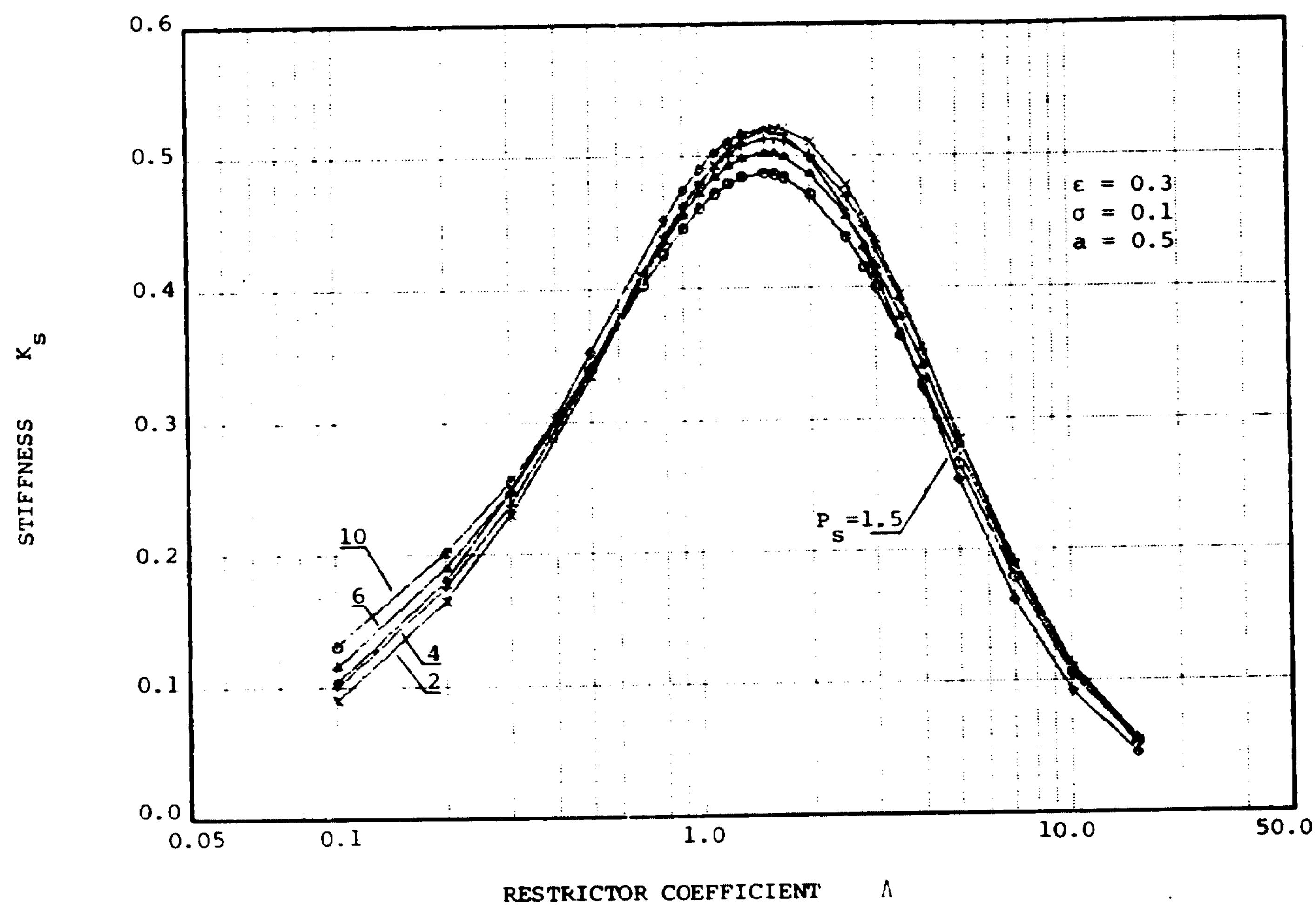

Figure 11. Dimensionless Stiffness versus Restrictor Coefficient (ε=0.3,σ=0.1,a=0.5)

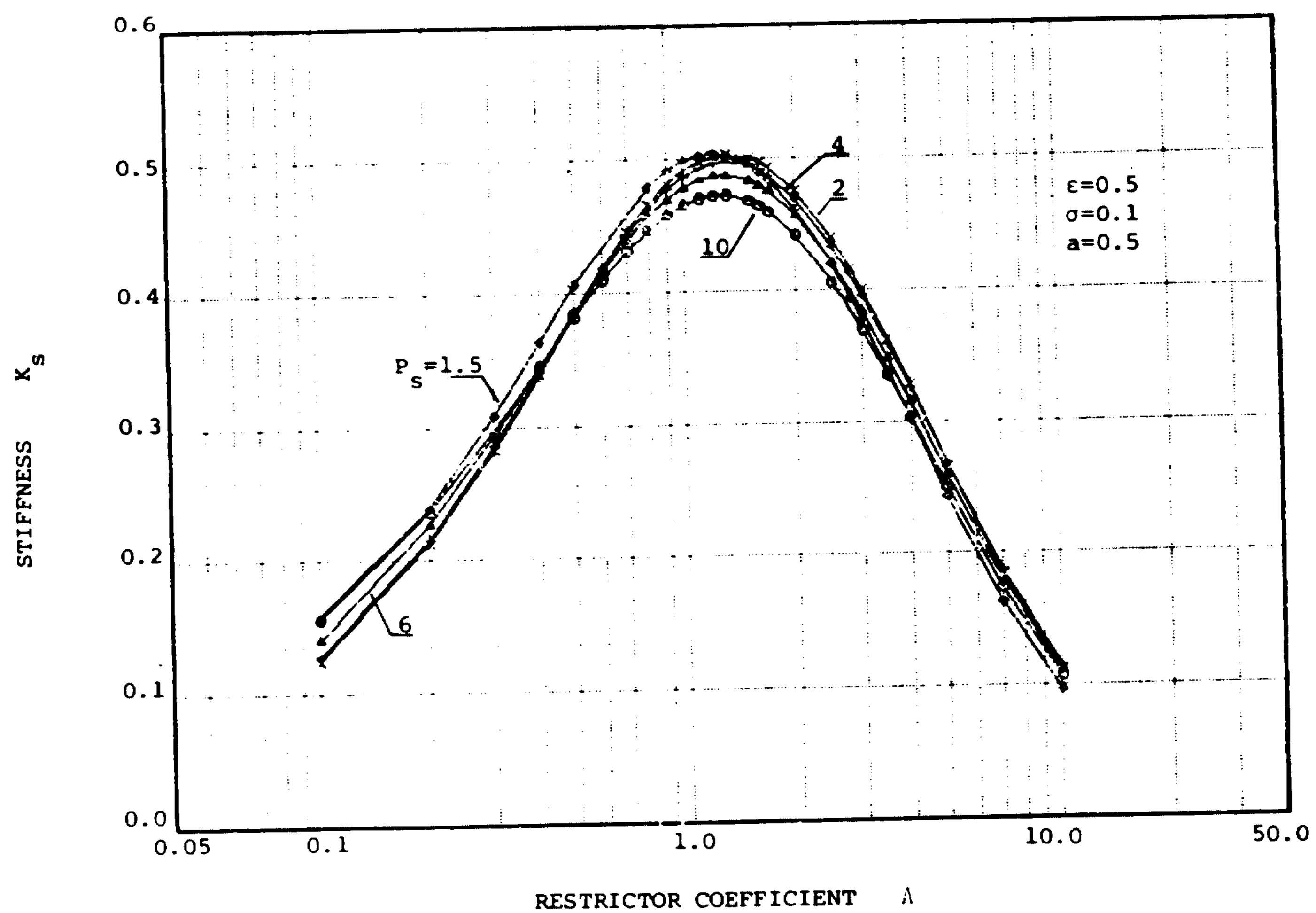

Figure 12. Dimensionless Stiffness versus Restrictor Coefficient (ε=0.5,σ=0.1,a=0.5)

$$\Lambda = \frac{24C_d \quad w^*(1-a)}{h_o^{*2} \, p_a^* \, p_s} \left[\frac{2g_o kRT}{k-1}\right]^{1/2} \quad .$$

The dimensionless stiffness and damping are defined by

$$K_s = \frac{K^* h_o^*}{2w^* L^* p_a^* (p_s - 1)}$$

$$D = \frac{D^*(1-\varepsilon^2)^{3/2}}{2\mu L^* \left(\frac{w^*}{h_o^*}\right)^3}$$

The main results are as follows:

(1) the dimensionless plots are essentially independent of the excursion ratios $\varepsilon = 0.1, 0.3, 0.5$. There is a slight shift of the minimum damping point away from the maximum stiffness point, $\Lambda \simeq 1$, as the ratio increases; thus, the restrictor coefficient is somewhat dependent on amplitude;

(2) the strip is stable for all parameters investigated;

(3) the actual stiffness, K^*, is not dependent on the amplitude;

(4) the actual damping, D^*, is inversely proportional to $(1-\varepsilon^2)^{3/2}$; thus amplitude increases the damping without affecting the stiffness.

Conclusions

The dynamic characteristics of inherently compensated gas film bearings have been investigated for small excursion ratios. Both the circular and the rectangular cases have been solved for the stiffness and damping as a function of supply pressure, restrictor coefficient, squeeze number, and geometry variations [9-11]. The effect of disturbance amplitude has been studied for the inherently compensated strip [13][14]. These results can be generalized and extended to the two-dimensional problem. Analytical solutions for the gas film damper problem have established the effect of disturbance amplitude at low squeeze numbers [12]. These results are applicable to limiting cases (restrictor coefficient approaching zero or infinity) of the inherently compensated bearing. All of the above information can be of immediate use to the designer of gas bearings.

The non-linear character of the Reynolds equation and the corresponding boundary conditions has forced the solutions technique away from analytical means to finite difference schemes with the digital computer. The method of linear equations are not available to ascertain iterative converge criteria. It has been the principal investigators experience that solution converge times can be quite long even when trial and error methods are successful. When parameter extremes were studied, converge failed more often than not. In these types of problems the situation is compounded by the large number of parameters which require independent study. The results must be abridged and presented in graphical form. It is this writer's opinion that alternate methods of solution should be explored together with the finite difference approach so that their accuracy can be deduced for future application. Such methods should be partially analytical not only to reduce computer time but to condense multi-parameter presentation. They could include:

(1) lumped parameter methods. This technique is certainly not new, but it has been rarely compared with computer solutions so that its net usefulness is not yet established. The method has been successfully applied by the author to the above one-dimensional strip [14];

(2) approximate methods such as Galerkin [15].

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