PAPER NO. 552

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ABSTRACT

In this paper, the use of externally pressurized gas lubricated bearings in modern high technology fields is presented. In recent years gas lubricated bearings have been employed in many high speed machine applications or where extreme precision is required. Thus, it has become very important to predict accurately the performance characteristics of these bearings. The steady state characteristics of externally pressurized (EP) gas lubricated journal bearings are presented employing superposition analysis. In this theoretical analysis, the load capacity and the attitude angles are predicted. The comparison with experiments within the employed range of speed show that the load capacity and the attitude angle are consistent provided an empirical factor of 0.74 is employed.

Key Words: Externally Pressurized, Gas Bearings

NOTATION

c	radial clearance
D	bearing diameter
e	eccentricity

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Kg	gauge pressure ratio
Kg_o	gauge pressure ratio for concentric bearing
L	bearing length
M_1	mass flow rate through the top orifice
M_2	mass flow rate through the bottom orifice
O	bearing centre
O^1	rotor centre
p	pressure
P_{i1}	pressure downstream of the top orifice
p_{i2}	pressure downstream of the bottom orifice
R	journal radius
W	load
W_{hr}	hybrid radial force
W_{ht}	hybrid tangential force
W_{dr}	aerodynamic radial force
W_{dt}	aerodynamic tangential force
W_{st}	aerostatic force
W	dimensionless load, $W/(\pi p_m DL)$
x	displacement in x-direction
y	displacement in y-direction
ϵ	eccentricity ratio, e/c
ϕ	attitude angle
μ	lubricant viscosity
ω	rotational speed

$\Lambda = \dfrac{6\mu\omega\ R^2}{P_a\ c^2}$ compressibility number

Subscripts

O	Zero eccentricity condition
a	condition referring to atmosphere
d	aerodynamic component
m	mean condition
s	condition upstream of orifice

Abbreviations

EP	externally pressurized
equ.	equation
LSA	linearized superposition analysis

1. INTRODUCTION

The use of gas as a lubricant has special characteristics which encourage their use over conventional lubricants in many engineering situations. The primary applications of EP gas lubricated bearings include experimental rotors for closed-cycle gas turbines, refrigeration machines, gyroscopes, high speed dental drills and grinders, precision machine tools, jigs, fixtures, elements of computing machines, measuring instruments, textile spindles and chemical plants (Grassam and Powell, 1964).

The EP bearing has an external supply of high pressure which is fed into the bearing through small holes located on the circumference of the bearing. The load carrying capacity is primarily dependent on the supply pressure of the gas. Many industrial applications demand large loads or small loads with virtually no relative movement of the surfaces. Self-acting bearings cannot operate under such conditions, so externally pressurized bearings may be required. External pressurisation also alleviates the start-up and shut-down wear in the case of aerodynamic bearings by permitting jacking of the rotor in the absence of rotation.

The linear superposition theory postulates that for the analysis of an externally pressurized hybrid gas journal bearing, it is possible to consider the aerostatic and aerodynamic pressures to be generated independently of one another and their cumulative effect is additive. The finite width solution of Ausman (1957) provides the aerodynamic load component for eccentricity ratios of less than 0.5 ($\epsilon < 0.5$) whereas the aerostatic load component derived from the axial flow theory in the form of design charts (Robinson and Sterry, 1958; Shires, 1963; Grassam and Powell, 1964; Powell, 1970) provides the first estimate of the static stiffness. In case of aerostatic bearing performance, the effect of non axial flow is an important consideration and a semi-empirical correction factor by Shires (1963) derived from a series of experiments by Robinson (1958) is included in the design charts for the aerostatic performance. As far

as the aerodynamic effect is concerned, it is not fully accomplished in the presence of external pressurisation. The experiments carried out by Powell (1964) shows that factor of 0.74 on the aerodynamic load component can be used on L/D ratios of upto 2.0. The same factor is used in the present analysis to achieve the net aerodynamic load capacity. By using an empirical factor of 0.74, the load capacity predicted, within the employed range of speed, gives a better estimate. The attitude angle gives qualitative agreement. A reducing eccentricity ratio with speed indicates increasing load capacity as self-acting effects begin to dominate.

2. THE PRINCIPLE OF OPERATION OF EP BEARINGS

The principle of operation of an aerostatic journal bearing is explained with reference to fig. 1. The bearing consists of a cylindrical bush into which are drilled one or two rows of gas feed holes spaced evenly around the bearing circumference. In its simplest form, only two inlet orifices on opposite sides may be considered. Compressed gas from the reservoir surrounding the bearing flows through the feed holes into the clearance and exhausts to the atmosphere. The pressure in the reservoir is the gas supply pressure Ps. When the gas flows through the feed holes and enters the bearing clearance, the pressure falls to P_i, finally exhausting to atmosphere at P_a. The rotor adopts a concentric position in the bush when no load is applied to the rotor. The pressure forces on the rotor are balanced with $P_{i1}=P_{i2}$, resulting in no variation of pressure circumferentially around the bearing.

When a load is applied to the rotor in the vertically downward direction, the rotor deflects in that direction so that the clearance is increased at the top and decreased at the bottom

The resistance to the flow of gas escaping to the atmosphere is reduced at the top and increased at the bottom with the result that more gas flows through the top feed holes and less gas flows through the bottom feed holes. The pressure at the top of the bearing, decreases while P_{i2} at the bottom of the bearing increases. A difference of pressure now exists across the rotor to balance the applied load. There is an equilibrium position of the rotor for all loads within its capacity. The eccentricity ratio ($\epsilon = e/c$), relevant to this position is known as the 'equilibrium eccentricity' and is an important parameter in all journal bearings.

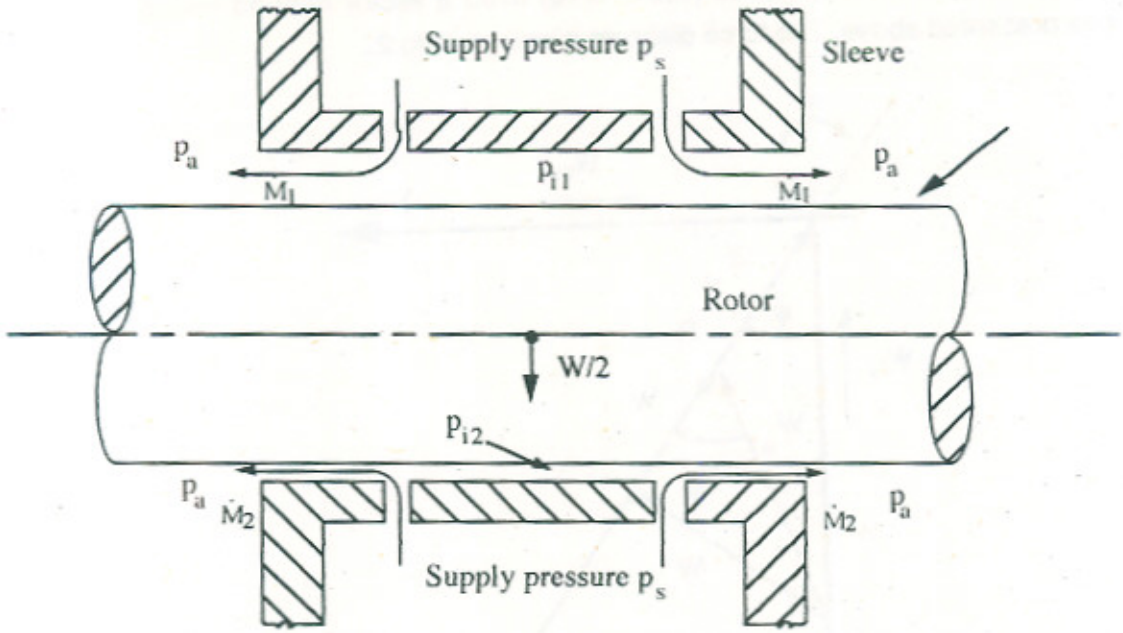


Fig. 1 Principle of operation for externally pressurised gas journal bearing.

In order that an aerostatic bearing with constant supply pressure has stiffness, it is necessary that there be an external flow restrictor between the supply and point where the gas enters the bearing film. In addition to being necessary for static stability of a bearing, stiffness is an important parameter regarding the dynamic stability of rotor bearing system. Hence the proper method of compensation is primarily of critical importance in the design of these bearings.

3. LINEARIZED SUPERPOSITION ANALYSIS (LSA)

The simplified, but not so rigorous, method proposed by Powell (1964,1970) postulates that the resultant pressure force acting on a rotor supported by externally pressurized bearings operating in a hybrid mode, has its aerostatic and aerodynamic components treated separately. The aerodynamic gas film force is composed of its radial film force component (W_{dr}) and tangential film force component (W_{dt}) while the aerostatic component is due to external pressurisation without any contribution from hydrodynamic effects. Now the aerostatic film force, W_{st}, acts along the line of centres and thus it is postulated that a first order estimate of hybrid radial force W_{hr} is the sum of the independent aerostatic and aerodynamic radial forces, and the hybrid tangential

force W_{ht} is the same as the aerodynamic tangential force. For calculation of hybrid steady state load capacity, Powell (1964, 1970) used a vector method encompassing the idea presented above. The force diagram is shown in fig.2.

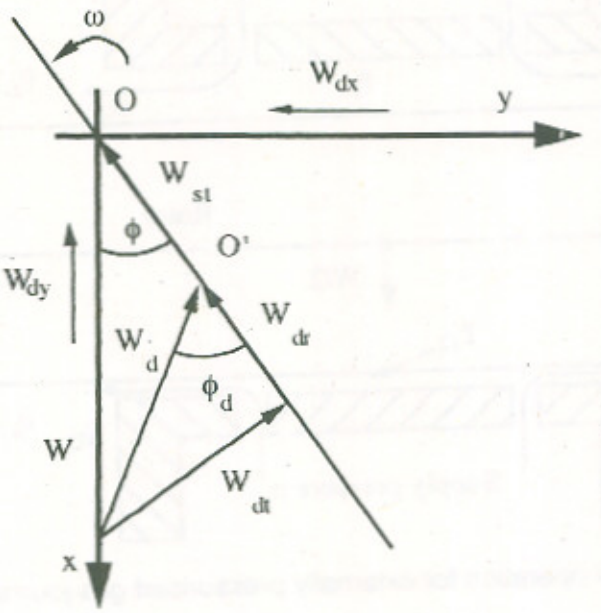


Fig. 2 Force diagram for linearised superposition method

The important consideration regarding the aerodynamic load contribution is the estimation of hybrid compressibility number (Λ) based on mean pressure instead of ambient pressure. The idea behind this is, that the added mass of gas in the film due to the aerostatic supply has the same effect on the aerodynamic component as increasing the ambient pressure. The mean pressure is defined as:

$$P_m = \frac{1}{2}(P_s + P_a) \qquad (1)$$

The argument for using the mean pressure based on supply pressure is that since in hybrid bearings, the compressibility numbers tend to be small compared to aerodynamic bearings, the bearing operation takes place in the incompressible region where the load capacity is almost independent of P_m. Consequently, small variations in P_m will have little effect on the load capacity. Tully (1966) proposed to use the mean

pressure based on an intermediate pressure instead of the supply pressure. Pink (1981) suggested to use P_i as means pressure for the central section and $(P_i+P_a)/2$ for the outboard sections of the 1/4 station double plane admission bearing. His results showed that the load capacity is underestimated by using such method. The mean pressure based on supply pressure overestimates the aerodynamic load component. As a consequence, Powell (1964, 1970) presented an empirical correction of 0.74 on the aerodynamic load component for L/D ratios up to 2. In the present work therefore, it is felt appropriate to use the mean pressure based on the supply pressure with the empirical correction of 0.74.

For a particular Λ and ϵ, the hybrid radial and tangential load components becomes:

$$W_r = W_{dr} + W_{st} \tag{2}$$

$$W_t = W_{dt} \tag{3}$$

where from fig. 2, $W_{dr} = W_d \cos\phi_d$ and $W_{dt} = W_d \sin\phi_d$

Therefore, total hybrid load and the attitude angle become:

$$W = W^2_r + W^2_t \tag{4}$$

$$\phi = \tan^{-1}\frac{W_t}{W_r} \tag{5}$$

For eccentricity ratios less than 0.5(ϵ <0.5), the load-displacement curve is close to linear and the aerostatic load component can be used either by theoretical work of shires (1963) or by experimental evaluation e.g. (Tully, 1966). Similarly the aerodynamic component can be obtained for example by the theoretical work of Ausman (1957). The finite width solution of Ausman (1957) provides the aerodynamic load component for eccentricity ratios of less than 0.5(ϵ <0.5) whereas the aerostatic load component derived from the axial flow theory in the form of design charts (Robinson and Sterry, 1958; Shires, 1963; Grassam and Powell, 1964; Powell, 1970) provides the first estimate of the static stiffness. In case of aerostatic bearing performance, the effect of non axial

flow is an important consideration and a semi-empirical correction factor by Shires (1963) derived from a series of experiments by Robinson (1958) is included in the design charts for the aerostatic performance. As far as the aerodynamic effect is concerned, it is not fully accomplished in the presence of external pressurisation. The experiments carried out by Powell (1964) shows that a factor of 0.74 on the aerodynamic load component can be used on L/D ratios of upto 2.0. The same factor is used in the present analysis to achieve the net aerodynamic load capacity. The non-dimensional aerodynamic load components in the radial and tangential direction, as given by Ausman (1957) are:

$$\frac{W_{dr}}{\pi p_m LD} = \frac{\varepsilon\Lambda}{2\left(1+\Lambda^2\right)}\left[\Lambda + \frac{(\alpha-\beta\Lambda)\sin\left(2\beta\frac{L}{D}\right) - (\alpha\Lambda+\beta)\sinh\left(2\alpha\frac{L}{D}\right)}{\frac{L}{D}\sqrt{1+\Lambda^2}\left(\cosh 2\alpha\frac{L}{D} + \cos 2\beta\frac{L}{D}\right)}\right] \tag{6}$$

$$\frac{W_{dt}}{\pi p_m LD} = \frac{\varepsilon\Lambda}{2\left(1+\Lambda^2\right)}\left[1 - \frac{(\alpha-\beta\Lambda)\sinh\left(2\alpha\frac{L}{D}\right) + (\alpha\Lambda+\beta)\sin\left(2\alpha\frac{L}{D}\right)}{\frac{L}{D}\sqrt{1+\Lambda^2}\left(\cosh 2\alpha\frac{L}{D} + \cos 2\beta\frac{L}{D}\right)}\right] \tag{7}$$

where

$$\alpha^2 = \frac{\sqrt{\left(1+\Lambda^2\right)}+1}{2} \tag{8}$$

$$\beta^2 = \frac{\sqrt{\left(1+\Lambda^2\right)}-1}{2} \tag{9}$$

Once the aerodynamic components are established by equs.6-7, the aerostatic component (W_{st}) can be estimated from design charts described above. In order to compare the present theoretical results with experimental results of Powell (1964, 1970), design curves for simple orifice bearings are used. In this present work, the simple orifices are employed. Referring to fig. 2, the aerodynamic load components, equs. 6-7, are transformed onto fixed xy axis. The aerostatic component is also estimated along the fixed xy axis. The resulting equations are explicit and solved to obtain the steady state characteristics.

4. RESULTS AND DISCUSSIONS

While comparing the present results with Powell (1970), it is shown in fig. 3 that within the range of the operating speeds, the agreement becomes good for the load capacity after a correction factor of 0.74 is used on aerodynamic load component, as suggested by Powell (1964). The load capacity increases with increasing compressibility number, i.e. increase in speed from 524 radians/sec., due to aerodynamic contribution. The predicted attitude angles are also in approximate agreement with the measurements and are shown in fig. 4. Both theoretical and experimental results show the trend of increasing attitude angle with speed confirming the aerodynamic contribution at higher speeds.

The increase in load capacity with increasing speed is manifested by a reduced eccentricity ratio as the speed increases. This effect is shown in fig. 5. Also shown in the figure is the effect of changing speed on the attitude angle. The linear superposition theory predicts a constant attitude angle with varying loads as was shown in fig. 4, so one curve for the attitude angle is applied to all loads. However the attitude angle dose increase with increasing speed and then decreases after attaining its maximum value. It is of course zero at the aerostatic condition, when there is no rotation of the shaft.
The effect of varying clearance at a constant supply pressure and the effect of varying supply pressure with all other parameters being fixed was studied theoretically and experimentally by references (Grassam and Powell, 1964; Powell, 1964, 1970, Tully, 1966). Using the axial flow theory, the aerostatic performance is established as a function of gauge pressure ratio defined as:

$$K_{go} = \frac{p_i - p_s}{p_s - p_a} \qquad (10)$$

The above references give a good description of linear superposition theory. Here, a brief discussion of the importance of gauge pressure ratio is given because of the dependence of load and stiffness upon it. Using the same approach as used by the author to calculate the linear superposition results, Powell (1964), who poineered the linear superposition method, provided the results of the variation of load capacity and

stiffness with gauge pressure ratio which are reproduced here. The load coefficient was defined as below:

$$C_L = \frac{W}{(p_s - p_a)LD} \qquad (11)$$

In machines using EP journal bearings, the onset of destructive vibration, known as fractional speed whirl, is in direct proportion to the square root of the aerostatic stiffness. Thus in order to achieve high whirl onset speeds, the bearing aerostatic stiffness has to be maximized. Often in the design of hybrid bearings, the mean radial clearance is the only variable left to the designer in order to optimise the bearing performance. The length to diameter ratio is controlled by the available space, supply pressure is restricted to a specific value due to the available compressor limitations. The size of an orifice and its location are restricted by flow considerations and by the limitation of the smallest feed hole that can be drilled by the available manufacturing conditions. Thus it is of value to consider the variation of radial stiffness with mean radial clearance. Fig. 6 shows theoretical and experimental curves of aerostatic stiffness and load coefficient against the gauge pressure ratio. The stiffness increases with reducing clearance, i.e. increasing gauge pressure ratio. At higher gauge pressure ratio, the stiffness curve starts reducing and will become zero at a gauge pressure ratio of 1.0. It can be seen that the maximum stiffness can be realized at a gauge-pressure ratio of 0.8. The stiffness at this gauge pressure ratio is 28 % higher than would be obtained at the point of maximum load coefficient. At the same time, the increase in bearing load capacity resulting from aerodynamic effect become greater as the

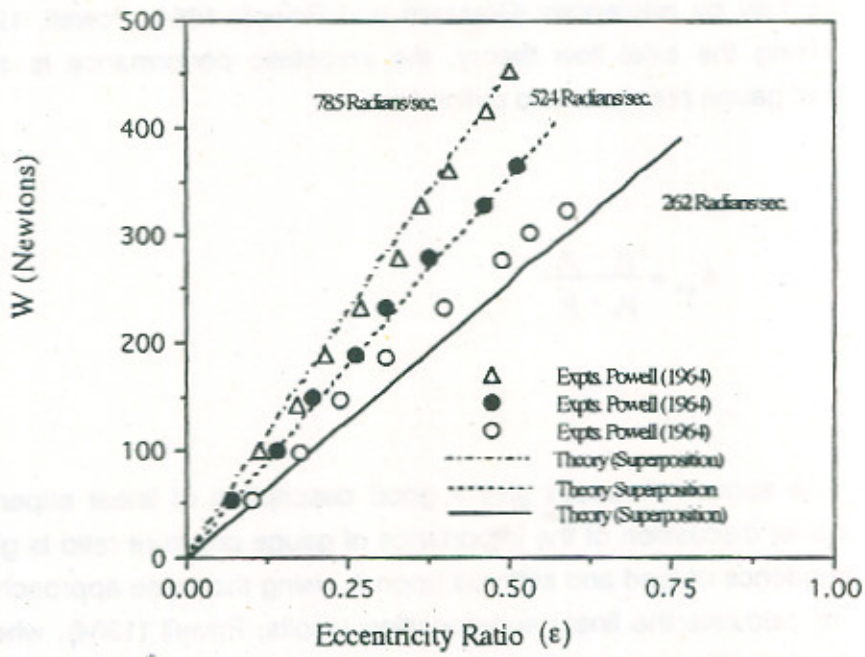


Fig. 3 Comparison of measured and the predicted load capacity by linear superposition with change in eccentricity; L/D=2, c=1.57E-5 m, n=8, d=1.47E-4 m, l/L=0.5, P_s=4.82E5 N/m², k=28.33E6 N/m,Kg_o=0.75

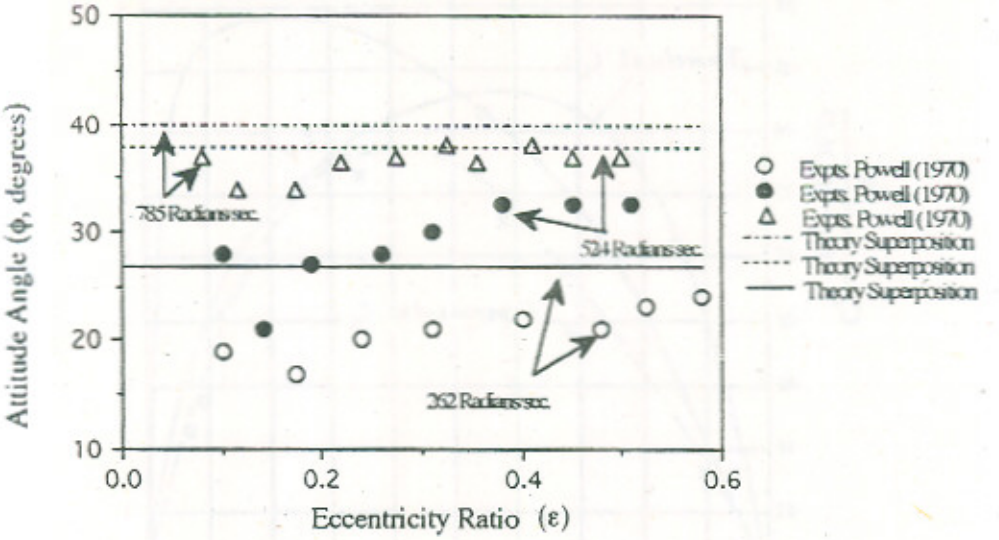


Fig. 4 Comparison of measured and the predicated attitude angle by linear superposition with change in eccentricity; L/D=2, c=1.57E-5 m, n=8, d=1.47E-4 m, l/L=0.5, P_s=4.82E5 N/m², k=28.33E6 N/m, Kg=0.75

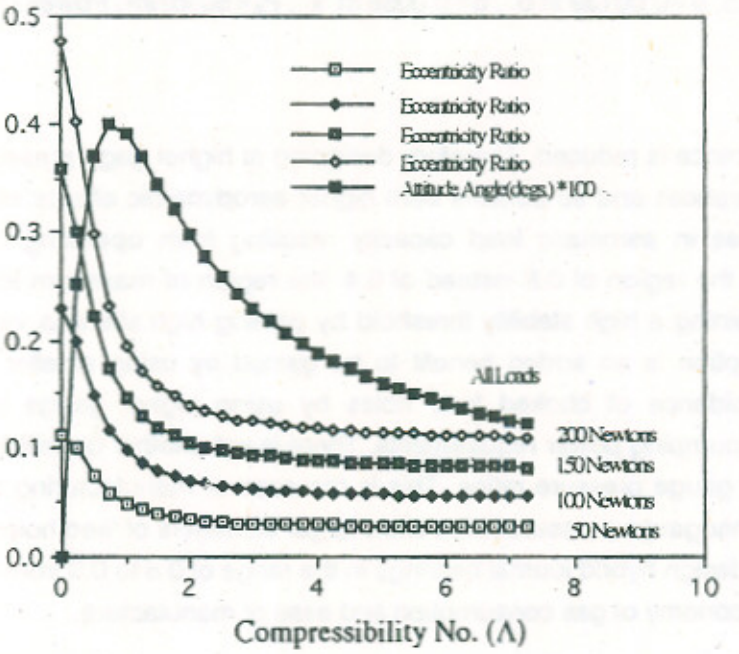


Fig. 5 Change of eccentricity and the attitude angle with the compressibility number; L/D=2, c=1.57E-5 m, n=8, d=1.47E-4m, l/L=0.5, P_s=4.82E5 N/m², k=28.33E6 N/m, Kg_o=0.75

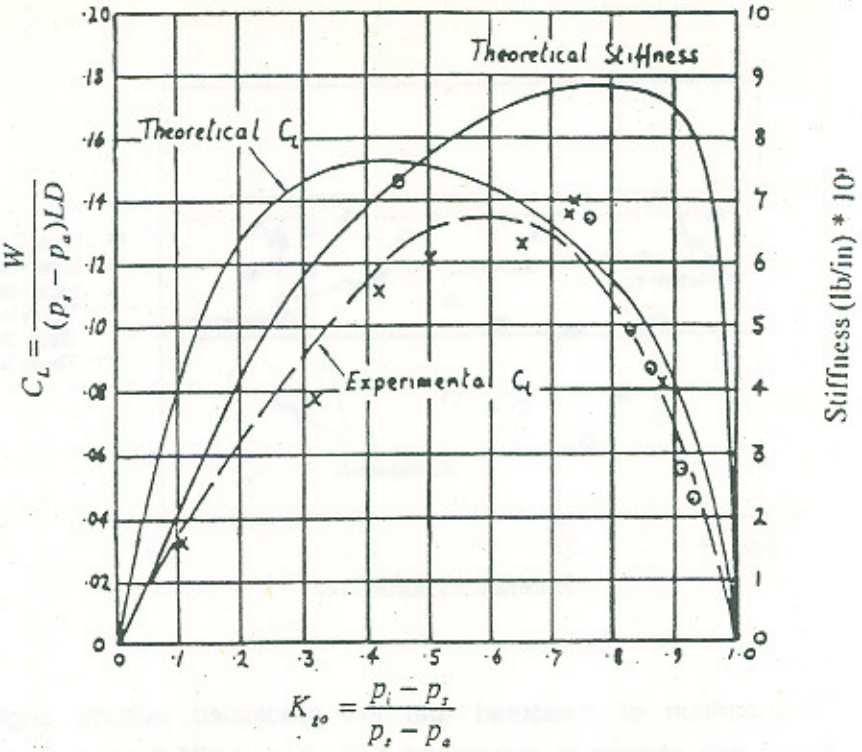


Fig. 6 Variation of stiffness and load coefficient with gauge pressure ratio; $\epsilon = 0.5$, d = 0.00139 in o , d = 0.0058 in x , $P_s = 50$ lb/in^2, Powell (1964)

bearing clearance is reduced. Therefore designing at higher gage pressure permits use of lower clearances and so benefits from higher aerodynamic effects which more than offset the loss in aerostatic load capacity resulting from operating at higher gage pressures in the region of 0.8 instead of 0.4 (the region of maximum load coefficient). Besides obtaining a high stability threshold by gaining high stiffness values, the lower gas consumption is an added benefit to be gained by using smaller clearances. In addition, avoidance of choked feed holes by using higher gauge pressure ratios ensures low pumping power requirements. There is yet another benefit to be gained by using higher gauge pressure ratios. This is the ease of manufacturing the feed holes, because higher gauge pressure ratios allow larger diameters of feed holes. It is therefore desirable to design hybrid journal bearings in the range of 0 6 to 0.9 from considerations of stability, economy of gas consumption and ease of manufacture.

5. CONCLUSIONS

In this paper, the use of externally pressurized gas bearings and steady state characteristics of EP gas journal bearings were presented. The load capacity and the attitude angle were presented as functions of varying eccentricity ratio and compressibility number.

The results shown employ the linear superposition theory to give the qualitative and approximate quantitative information about the bearing characteristics, which can be helpful in understanding the separate aerodynamic and aerostatic film force contributions. By using an empirical factor of 0.74, the load capacity predicted, within the employed range of speed, gives a better estimate and takes care of the 30 % overestimation that this method normally predicts for the load capacity. The attitude angle gives qualitative agreement.

The load capacity and the attitude angle show increase with increase in compressibility number indicating the dominance of aerodynamic effects at higher speeds. A reducing eccentricity ratio with speed also indicates increase in load capacity as aerodynamic effects begin to dominate.

In designing EP bearings, the radial clearance is an important parameter which has to be set to its optimum value in order to provide maximum stiffness and low gas consumption.

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