EMPIRICAL CORRELATIONS FOR THE PRESSURE DEPRESSION IN EXTERNALLY PRESSURIZED GAS BEARINGS

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ABSTRACT

The pressure depression phenomenon, usually observed in the inlet region of externally pressurized gas bearing, is predicted by constructing a simple empirical model. The same sets of experimental data, used previously by other investigators, were used to obtain the empirical equations which formulate the proposed model. Three correlations are proposed. These are the inlet pressure correlation, the minimum pressure correlation, and the position of the minimum pressure within the inlet region. Comparisons between the values obtained using these correlations and available experimental data show good agreements. Further, a comparison between the present model and another previously proposed model shows that the present one gives better results. In addition, comparison between the proposed model and available experimental data shows that the model predicts correctly the pressure variations, which occur due to the effect of pressure depression, within the inlet region.

Keywords: Lubrication, Gas Bearings, Externally Pressurized.

Nomenclature

B_o	Bearing outer width
B_r	Recess width
d_o	Supply hole diameter
H	Film, thickness
L_o	Bearing length
L_r	Recess length
P_a	Atmospheric pressure
P_i	Inlet pressure
P_{min}	Minimum pressure in the inlet region
P_r	Pressure at the outer edge of the recess
P_s	Supply pressure
P_x	Pressure at any distance x
x	Cartesian coordinate starting from the centerline of the bearing width
χ_{min}	Position where P_x=P_{min}

1. INTRODUCTION

Minimizing the differences between experimental and theoretical results is one of the most important goals in many areas of scientific as well as applied research. The approach to reach this goal is usually branched into two parallel ways. First, improving the experimental results by using updated and advanced measuring techniques in order to reduce the uncertainty of the obtained results. Second, improving the theoretical results by minimizing the assumptions used and making them more realistic. Therefore, the present study is a step towards the achievement of this goal. It is a continuation of a programme, started long time ago, aimed to minimize the deviation between the experimental and theoretical results obtained for the performance of externally pressurized gas bearing. A recent review concerning this programme is given by Kassab [1]. The present study specifically deals with the study of the pressure depression phenomenon observed in the inlet region of the thrust air lubricated bearings.

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It is well known that the pressure distribution along the bearing is one of the main parameters in calculating all other bearing characteristics (such as load carrying capacity and stiffness). The pressure depression observed in the inlet region of circular thrust bearing was explained by Mori [2]. His explanation was based on the concept of sonic velocity at the periphery of the inlet orifice followed by supersonic flow in the recess region where a shock wave can result to raise the pressure again. Since Mori investigation [2] of the supersonic depression in the feeding region of externally pressurized bearings, there have been a number of comprehensive studies of this phenomenon [3-8]. A review of these studies can be found in Gross et al. [9] .

Stahler [3] used a free surface water table analogue to study the shock waves in externally pressurized circular and rectangular gas bearings. He showed that, in a rectangular bearing with circular source, the existed shock boundary is slightly elliptical in shape. Further, in the field of externally pressurized circular bearings, Salem [10] investigated theoretically and experimentally the stepped bearing using incompressible (oil) and compressible (air) lubricants. In the comparison between air and oil bearings, he found that if the viscous flow condition prevails, the stepped air bearing has a higher load carrying capacity than of an identical one, using oil as the lubricating fluid, when both bearings are working under the same film thickness and recess pressure. However, with the increase in the film thickness and inlet recess pressure the inertia effects become more important in the case of air bearings, resulting in pressure depression and giving the oil bearings the advantage of carrying more load.

For circular thrust bearing with single central supply hole Mori and Miyamatsu [6] constructed several mathematical flow models by making proper chains of fundamentals and elemental flow patterns and pressure changes. Their models were quite effective in explaining the pressure distribution, the load capacity, and the rate of flow observed experimentally over wide operating conditions. In addition Vohr [7] studied experimentally the pressure profiles across a feeding hole in the center of a circular thrust bearing. He reported a sudden depression in pressure at the entrance to the film followed by a partial pressure recovery. He also showed that the values of pressure at the entrance of the film were considerably lower than the values predicted by the viscous theory. Moreover, Kawaguchi [11] constituted an empirical equation by using nondimensional quantities which have large influences on the entrance loss. He found that the larger gap and the smaller roundness of inlet corner give the minimum pressure at the enterance region.

On the other hand, Kamal [12] suggested a tapered recess in circular bearings to decrease the bearing air storage capacity, leading to a better stability. This modified bearing was found to work at a larger film thickness than the plane recessed bearing without shock wave formation but with a smaller load carrying capacity. Meanwhile, Shawky [13] studied theoretically the effect of the surface roughness on the performance of externally pressurized rectangular air bearings. He found that the surface roughness of the bearing has a major effect of increasing the flow rate and pressure depression in the recess region. In addition, Salem and Shawky [14] found that introducing a recess, and/or step in the bearing surface of a plane rectangular bearing has an increasing effect on the pressure depression following the inlet supply hole.

Gross et al. [9] demonstrated that although inertia effects can be negligible in the bearing film away from entrance regions, they can be significant near the entrance region owing to the development of the velocity profiles, and to the existence of a separation bubble when flow enters the film around a sharp-edged corner. Downstream of the separation bubble, the flow must reexpand to fill the channel with an accompanying loss of pressure head. In addition, for the case of radial flow, the recovery pressure beyond the shock is strongly affected by both the radius ratio and the film thickness.

Kassab [15] investigated experimentally the performance characteristics of an externally pressurized rectangular air bearing working under various operating and geometrical conditions. He concluded that the dimensionless pressure profiles in the inlet zone of air bearings are characterized by a pressure depression which causes the pressure to drop to a minimum value after which it re-increases to a maximum value less than the inlet pressure. Moreover, the point of minimum pressure is always at the same distance from the supply hole no matter the bearing recess and outer dimensions are. Meanwhile, Shawky and Kassab [16] concluded that the non-dimensional load drops by increasing the operating film thickness and/or increasing the working supply pressure as a result of the increase in pressure depression. They also found that decreasing the bearing width to length ratio (B_o/L_o), keeping the supply hole diameter and the recess to outer width ratio (B_r/B_o) constant, increases the pressure depression following the supply hole and consequently lowers the pressure profile at the same working conditions.

For externally pressurized air lubricated circular thrust bearing, Boffey and Wilson [17] observed a sharp fall-off in pressure as the flow enters the film which is consistent with an increase in velocity at this point. Further, in their comparison of experimental results with theoretical findings, they found that the theory overestimates the measured steady state static pressure at the entrance of the film. Meanwhile, Wark and Foss [18] determined experimentally the magnitude of the force caused by the radial out flow between parallel disks. They compared their experimental results with the analytical results of Hayashi et al. [19]. They found that the empirical information of the length and thickness of separation zone, that are required in Hayashi et al. model, is at least one factor that contributes to the disagreement between the two-results.

For the case of elastohydrostatic lubrication of externally pressurized rectangular gas bearing, Khalil et al. [20] reported that when the pressure depression due to inertia effects near the recess edge and supply hole is minimal, the experimental pressure distribution along the film was found to be in a close agreement with the theoretically obtained distribution.

Shawky [21] used Kassab's [15] results to obtain an empirical formula which relates the maximum value of pressure recovered after depression (effective recess pressure, P_r) with the outer supply pressure, P_s, for different operating conditions and outer dimensions. Meanwhile, the numerical results of San Andres and Velthuis [22] show that the pressure field in the recess is not uniform. For pure hydrostatic effects, the pressure decreases at the recess region and a large local pressure drop occurs at the transition recess edge/film land.

Noureldeen [23] investigated experimentally the effect of many geometrical and operating parameters on the performance of externally pressurized rectangular air bearing. He found that the dimensionless pressure profile drops in the inlet region to a minimum value after which it re-increases to a maximum value less than the inlet pressure. Moreover, the pressure depression increases by the increase of the film thickness and or/the supply pressure.

Finally, Kassab [1] observed that the pressure inside the recess, of externally pressurized rectangular bearing is not constant as assumed theoretically. The pressure distribution in the inlet region is characterized by a pressure depression zone followed by an increase to a certain maximum. Kassab also discussed thoroughly the limits and applicable ranges of Shawky's [21] empirical equation. He found that Shawky's formula may be used effectively under the following design and operating conditions.

1. When the bearing has a longitudinal recess with a length close to the bearing outer length.
2. For the condition of maximum dimensionless load and minimum mass flow rate at certain operating conditions.
3. When using a sharp edged supply hole.

In addition, Kassab [1] reported that, in general, the theoretical results of the pressure distribution and load carrying capacity are higher than the experimental data. This difference is attributed to be mainly to the simplifying assumptions used in his analysis.

From the above review of literature concerning the pressure depression phenomenon, it seemed that the pressure distribution in the inlet region to externally pressurized gas bearings still needs a further investigation. Therefore, the present study aims at predicting the behaviour of air flow, at the inlet region to a recessed bearing, in a simple way. It depends on analyzing available experimental data to deduce correlations between pressures in various positions within the inlet region. In addition, a comparison is also included between the proposed model and available experimental data, to check the validity of the model.

2. THE TESTED DATA

The analyses of Shawky [21] and Kassab [1] regarding the externally pressurized recessed rectangular air bearing shown in Figure (1), were both based on the experimental data obtained by Kassab [15] and Shawky and Kassab [16]. For two reasons, in the present study the same sets of data, used previously by these two authors, were used to obtain the empirical equations which formulate the proposed model. First, in order to compare and combine the results obtained from the present study with the results obtained by Shawky [21] and Kassab [1], all the three studies should be based on the same sets of data. Second, Kassab's [15] data were examined thoroughly by many other investigators. Elgayar [24] and Kandil [25] showed, through comparison with their experimental results, that the experimental results obtained by Kassab [15] are repeatable and reliable. In addition, recently Noureldeen [23] repeated some of the experiments done previously by Kassab [15] to obtain new sets of data for the same operating and geometrical conditions used in Kassab's data. A comparison between these two sets of experimental data is shown in Figure (2). Both sets of data coincided with each other and this enhances the confidence in both of them. Moreover, credence is added to the measurements of Kassab [15] and Noureldeen [23] since these two sets of experimental results were obtained by two different investigators independently over a fourteen year period.

3. THE PROPOSED MODEL

The flow in the inlet region of a gas bearing is extremely complicated. As complete analysis may need to allow for shock waves, vortex formation, and boundary layer growth. Gross et al. [9] pointed out that the stokes assumption regarding the bulk viscosity does not apply at a shock front. Thus, the conventional Navier-Stokes equations are not applicable throughout the film. On the other hand, for a fundamental understanding of externally pressurized gas bearing, it is important to know the flow patterns and pressure profiles in the inlet region of such bearing. However, as emphasized by Gross et al. [9], it may not be necessary to take accurate account of such details in design calculations. This is due to the fact that the pressure depression region is confined to an area around the inlet hole which is small compared with the rest of the bearing area.

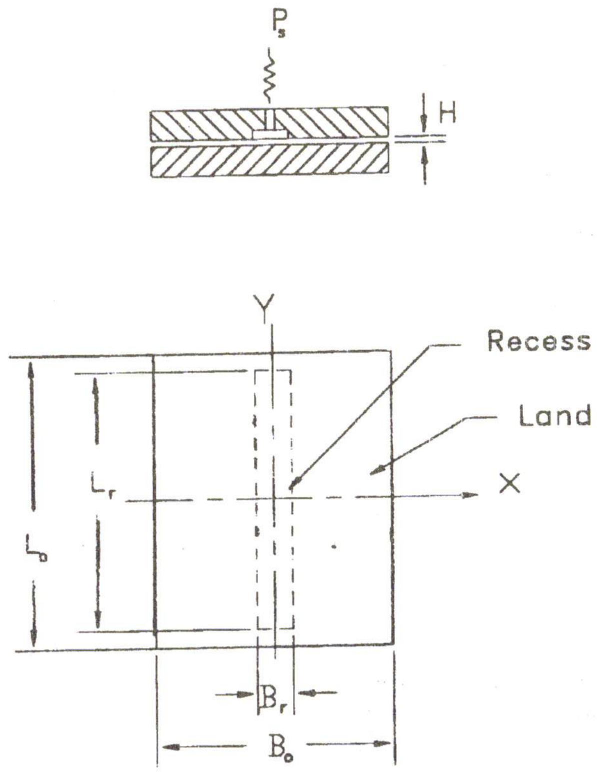

Figure 1. Recessed rectangular bearing.

Taking the above mentioned points into account, the present study is confined to predict the pressure depression by a simple but effective empirical model. Consequently, the pressure distribution inside the recess is predicted through the following steps:

1. Pressure inside the supply hole region is assumed to be constant and equal to the inlet pressure, P_i.
2. Following Shawky's [21] procedure and using the same sets of data used by Shawky [21] and Kassab [1] (i.e. Kassab's [15] data) the empirical equations which relate the inlet pressure, P_i, and the minimum pressure, P_{min}, with the supply pressure, P_s, for different operating conditions and outer dimensions are obtained as follows:

$$\frac{(P_i-P_a)}{(P_s-P_a)}=3.95(\frac{B_o}{L_o})-2.5(\frac{B_o}{L_o})^2-725(\frac{H}{L_o})-0.16\frac{(P_s-P_a)}{P_a}-0.31 \quad (1)$$

$$\frac{(P_{min}-P_a)}{(P_s-P_a)}=3.95(\frac{B_o}{L_o})-2.5(\frac{B_o}{L_o})^2-1800(\frac{H}{L_o})-0.16\frac{(P_s-P_a)}{P_a}-0.22 \quad (2)$$

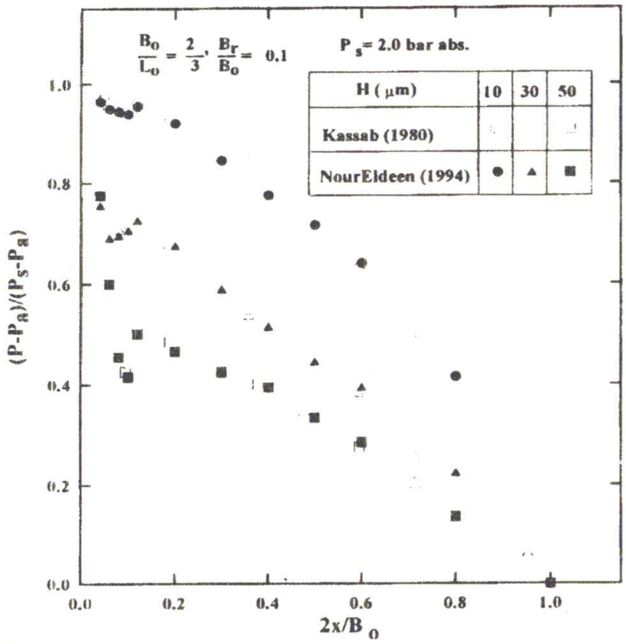

Figure 2. Comparison between experimental results Kassab (1980) and Noureldeen (1994).

3. The pressure, P_r, at the end of the recess region, $x = B_r/2$, is assumed to be equal to the pressure at the beginning of the land region. The value of P_r is obtained using Shawky's [21] empirical equation:

$$\frac{(P_r-P_a)}{(P_s-P_a)}=3.95(\frac{B_o}{L_o})^2-2.5(\frac{B_o}{L_o})^2-1386(\frac{H}{L_o})-0.16\frac{(P_s-Pa)}{P_a}-0.29 \quad (3)$$

4. The position where the pressure is minimum, X_{min}, is taken as

$$\chi_{min} = 2.8(\frac{d_o}{2}) \quad (4)$$

5. Pressure variations between P_i and P_{min} and between P_{min} and P_r are assumed to be linear.

From the above steps the present empirical model is formulated as follows:

$$P_x=P_i \quad 0 \le x \le \frac{d_o}{2}$$

$$P_s=P_i-(P_i-P_{min})\frac{(x-\frac{d_o}{2})}{(\chi_{min}-\frac{d_o}{2})} \quad \frac{d_o}{2} \le x \le \chi_{min}$$

$$P_s=P_r-(P_r-P_{min})\frac{(\frac{B_r}{2}-x)}{(\frac{B_r}{2}-\chi_{min})} \quad \chi_{min} \le x \le \frac{B_r}{2}$$

4. RESULTS AND DISCUSSION

The empirical model, which predicts the pressure depression phenomenon, is based on three new correlations. These are the inlet pressure correlation, equation 1, the minimum pressure correlation, equation 2, and the position of the minimum pressure, X_{min}. In addition to these correlations, Shawky's formula, equation 3, is also used. It is important to demonstrate first how close are the values obtained using the proposed correlations with respect to the actual experimental results.

Figure (3) shows a comparison between the results obtained using the empirical equation of the inlet pressure, Eq. 1, and the experimental results of Kassab [15]. One should note that each experimental point shown in this figure represents the value of the inlet pressure for a certain operating condition, and is extracted from a complete survey of pressure distribution similar to that shown in Figure (2). It is clear from Figure (3-a) that both the empirical and experimental results have the same trend; the dimensionless inlet pressure is linearly decreased as the film thickness increased, for various values of dimensionless supply pressure. Further, the differences between empirical and experimental values are small and random. Consequently, there is a qualitative as well as quantitative agreement between the two results. Moreover, the comparison shown in Figure (3-b) for the variation of the inlet pressure, P_i, with the supply pressure, P_s, at various values of film thickness, H, would also support the above findings.

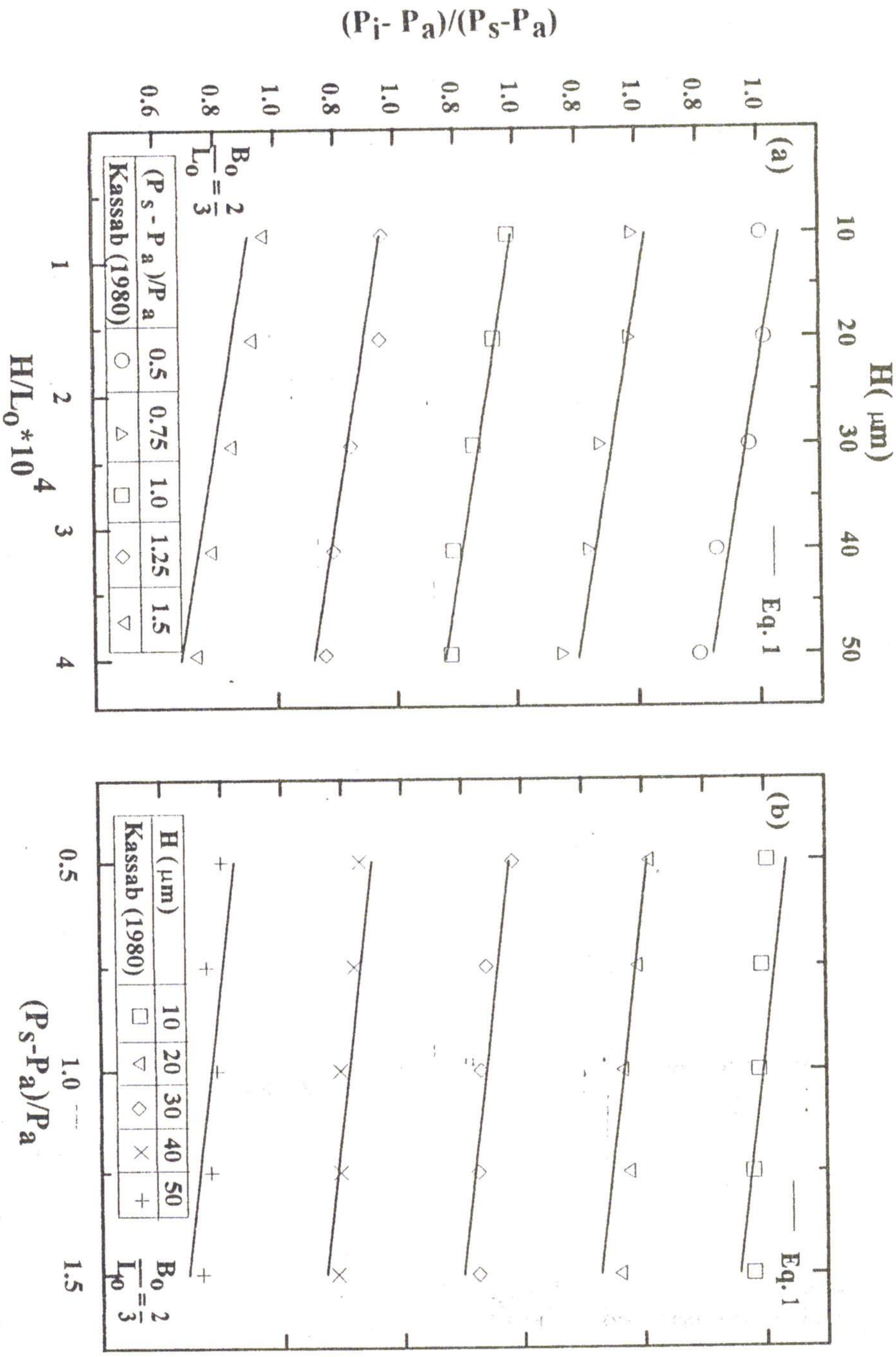

Fig. 3 Comparison between the inlet pressure, P_i, obtained using equation 1 and experimental results of Kassab (1980).
(a) For different values of supply pressure, P_s. (b) For different Values of film thickness, H.

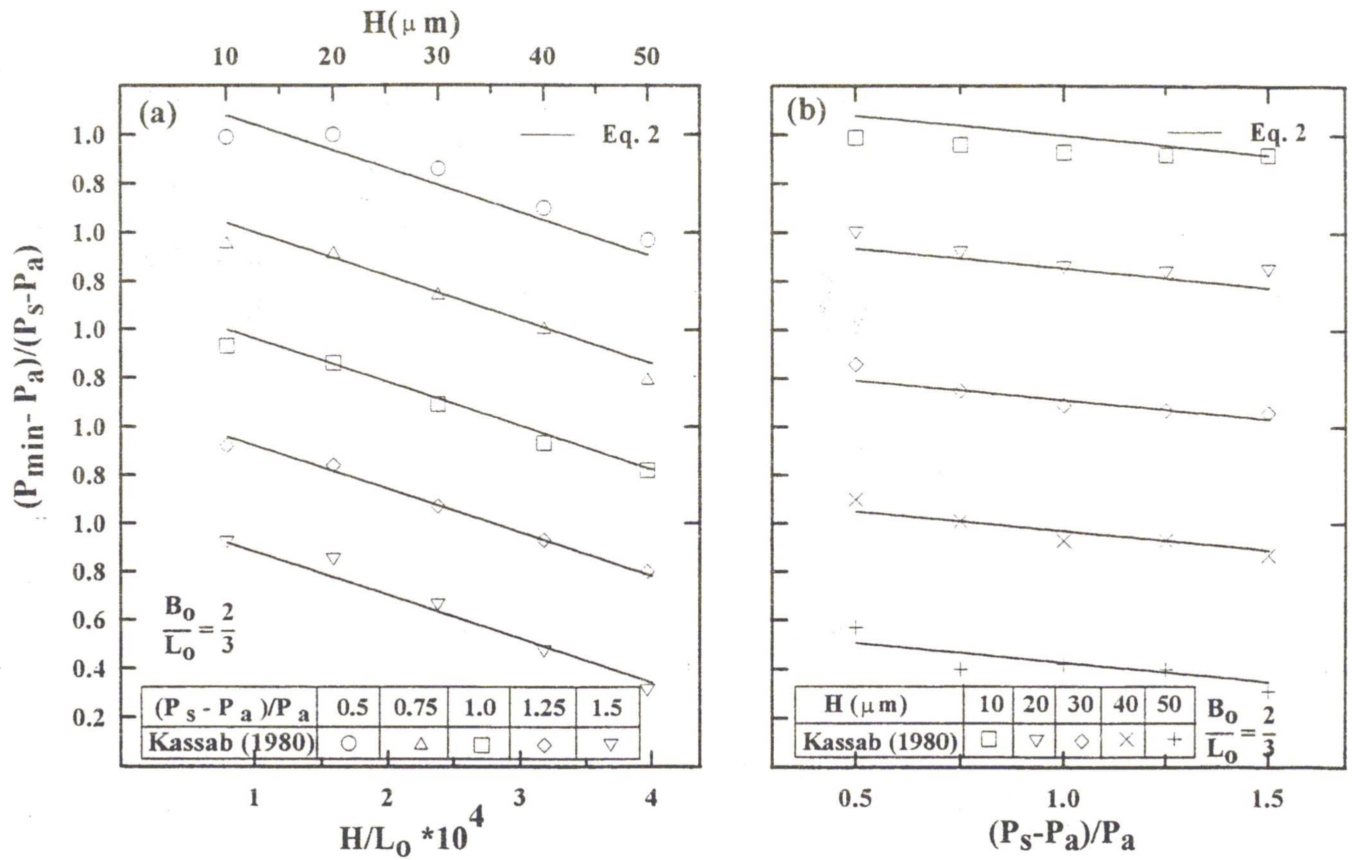

Fig. 4 Comparison between the minimum pressure, P_{min}, obtained using equation 2 and experimental results of Kassab (1980).
(a) For different values of supply pressure, P_s. (b) For different values of film thickness, H.

A comparison between the values obtained for the minimum pressure at the inlet region, P_{min}, using Eq. 3 and the experimental results of Kassab [15] is shown in Figure (4). Here, also the values obtained using the two methods are comparable for different values of film thickness and supply pressure. Consequently, the results presented in Figures (3) and (4) demonstrate that the two empirical equations, Eqs. 1 and 2, are give results in good agreement with the corresponding experimental values. This is a step forward towards the validation of the proposed model.

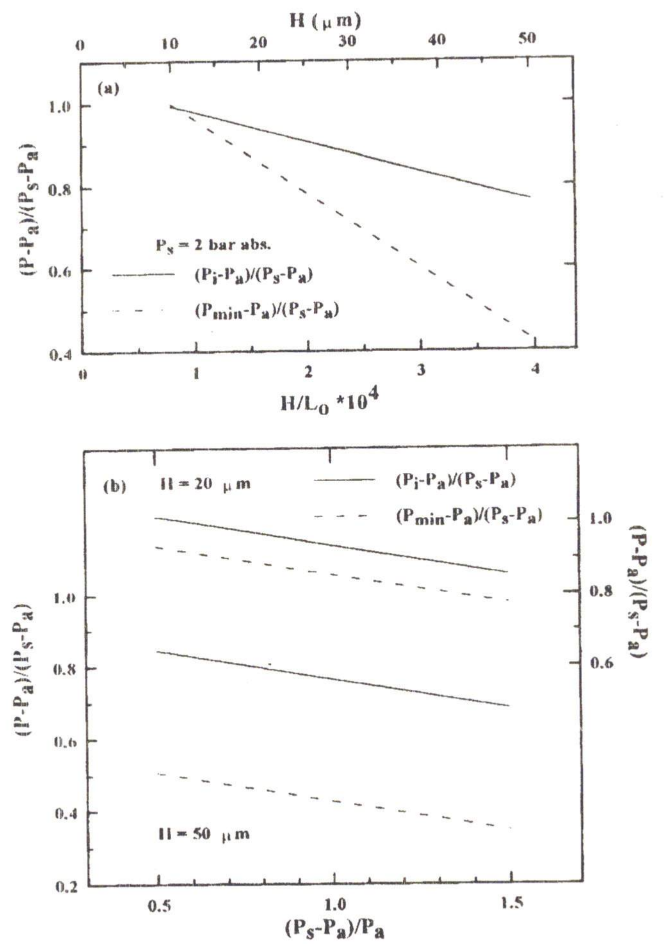

Figure 5. Graphical presentation of equation 1 and2
a- P_s = 2 bar abs   b- H = 20 and 50 μm.

A graphical presentation of Eqs 1 and 2 is shown in Figure (5). Figure (5-a) shows that as the film thickness increases both the inlet and minimum pressures have linearly decreased. Further, for a certain supply pressure and small film thickness, the difference between inlet and minimum pressures is small. Furthermore, as the film thickness increases this difference increases. However, the trend shown in Figure (5-a) is physically correct because, for small film thickness (i.e. H = 10 μm) there is no, or very small, pressure depression (see for example Figure (2)). Consequently the difference between inlet and minimum pressures is small. However, as the film thickness increases, for a certain value of supply pressure, the pressure depression increases resulting in a decrease in the minimum pressure and an increase of the difference between the values of inlet and minimum pressures. In addition, Figure (5-b) illustrates that, for a certain value of the film thickness, as the supply pressure increases both inlet and minimum pressures are linearly decreased. Varying the supply pressure has no effect on this difference.

On the other hand, determination of the position of the minimum pressure within the inlet region, X_{min}, is important. Shawky [13] and Kassab [15] found that the point of minimum pressure is always at the same distance from the supply hole regardless of the bearing dimensions. The present proposed model will go along with their findings. An analysis of Kassab's [15] experimental data shown in Figure (6) gives a mean value of $x_{min}/(B_r/2)$ = 0.84. This value is equivalent to $x_{min}/(d_o/2)$= 2.8 for the data given in this figure. This value is adopted for X_{min} in the present model.

Using the proposed empirical model, pressure distributions inside the recess of externally pressurized rectangular air bearing are shown in Figures. (7) and (8) for different supply pressures and film thicknesses. It can be seen from Figure (7) that as the film thickness increases the pressure distribution becomes lower. Further, for certain supply pressure, the values of inlet pressure, P_i, minimum pressure, P_{min}, and maximum pressure attained after depression, P_{mad}, decrease as the film thickness increases. Meanwhile the variation of pressure distributions, for certain film thickness, at various values of supply pressure are shown in Figure (8). As the supply pressure increases the pressure decreases. Further, the rate of pressure decreasing with respect to the supply pressure is constant (i.e parallel lines in each region). Combining the observations noted from Figures (7) and (8), the pressure inside the recess, which is obtained using the present model, decreases as the film thickness and/or the supply pressure increases.

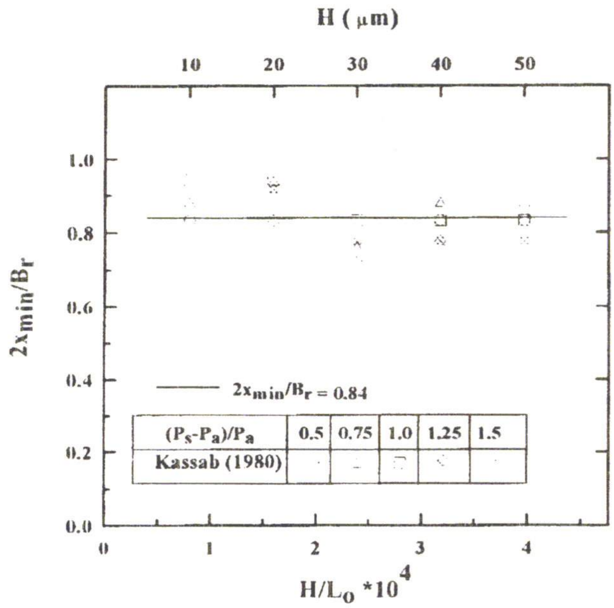

Figure 6. Position of the minimum pressure within the inlet region.

Finally, a comparison between the values obtained using the proposed empirical model, for pressure distribution in recess region, and the experimental results of Shawky and Kassab [16] is shown in Figs. 9 and 10. It is clear from these two figures that both sets of data have the same trend. In addition, the agreement between the predicted and measured pressure values are seen to be fairly good. Further, it should be mentioned here that the assumption of a constant recess pressure equal to the maximum pressure attained after depression was introduced earlier by Kassab [15], formulated empirically by Shawky [21] and used thereafter by Kassab [1]. Pressure distributions obtained using this assumption is shown in Figure (10) in comparison with the present empirical model and the experimental data of Shawky and Kassab [16]. Figure 10 shows that pressure distributions obtained using the present simple empirical model are comparable with the corresponding results obtained using the assumption of constant recess pressure (Shawky's [21] model) and the experimental data, for the case of small film thickness ($H = 10\ \mu m$) due to the absence of pressure depression. However, for the case of higher film thickness ($H = 30$ and $50\ \mu m$) the present model is seen to be better than Shawky's [21] model because it predicts correctly pressure variations within the recess which occurs due to the effect of the pressure depression.

5. CONCLUSIONS

1. Simple empirical equations relating the inlet pressure and the minimum pressure, in the inlet region, with the supply pressure, for different operating conditions and outer dimensions, are obtained. Position within the inlet region where the pressure is minimum is also obtained.
2. Comparisons between the pressure values obtained using two proposed empirical equations show that they are physically correct.
3. The values obtained using the two empirical equations are in good agreement with the corresponding experimental results.
4. Comparisons between the present proposed model and a previous one showed that the present model predicts correctly the pressure variations within the inlet region due to the effect of pressure depression.

Finally, it is important to reemphasize here that the empirical correlations, although valid for the test data used, are not necessarily applicable over the wide range of possible operating conditions. Therefore, the present proposed empirical model can be regarded as a step towards reducing the differences between experimental and theoretical results in the area of gas bearings.

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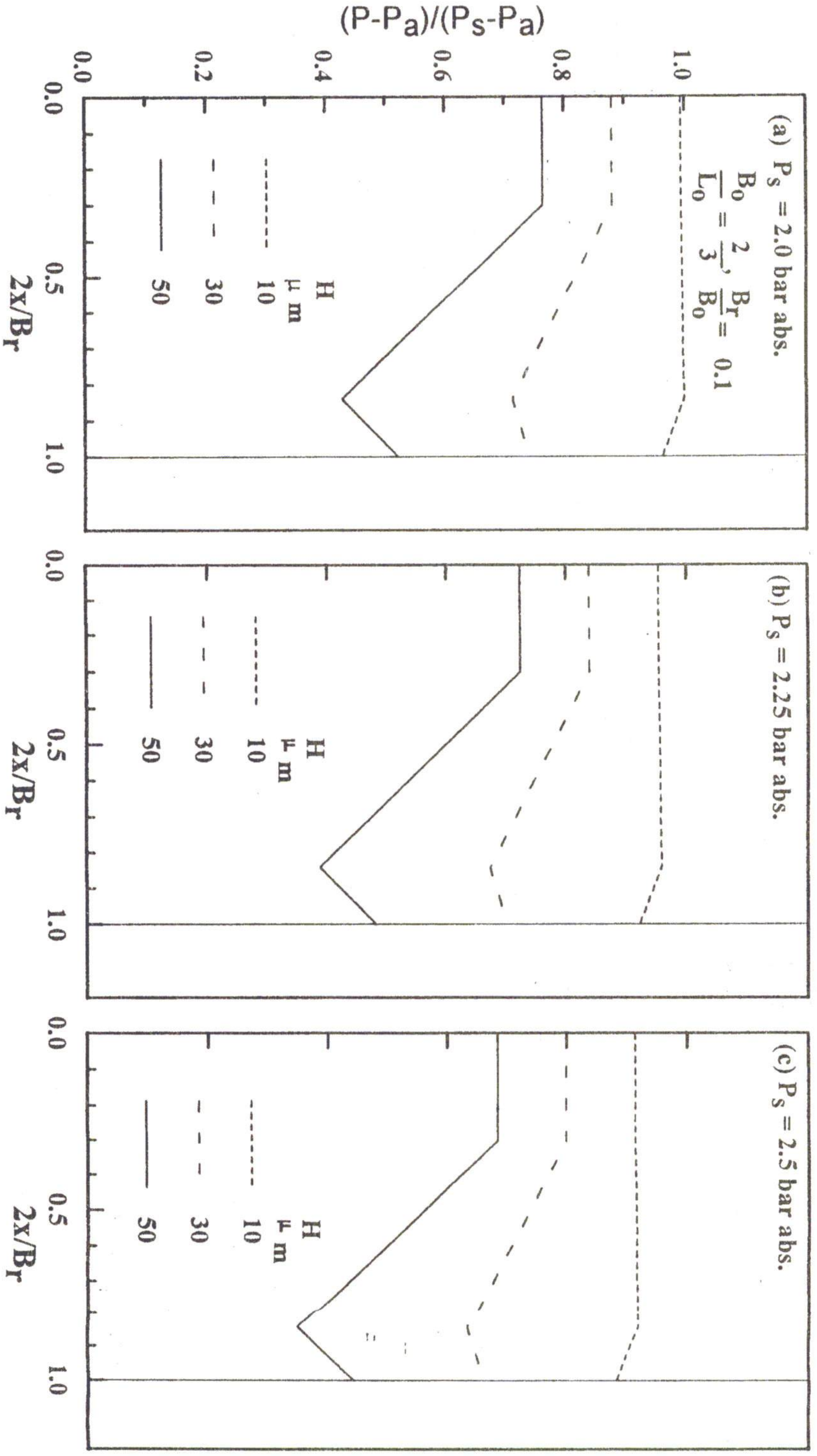

Fig. 7 Pressure distribution inside the recess using the present model.
(a) $P_s = 2$ bar abs (b) $P_s = 2.25$ bar abs (c) $P_s = 2.5$ bar abs

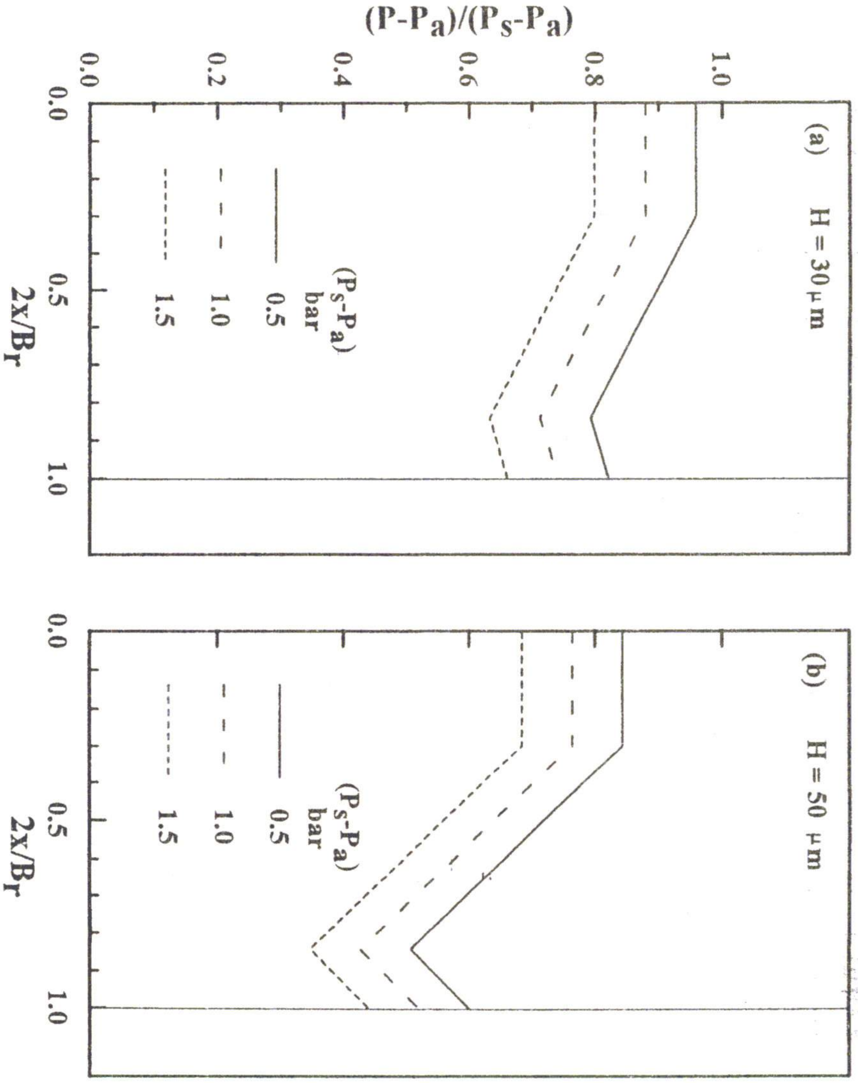

Fig. 8 Pressure distribution inside the recess using the present model.

(a) H = 30 μm (b) H = 50 μm

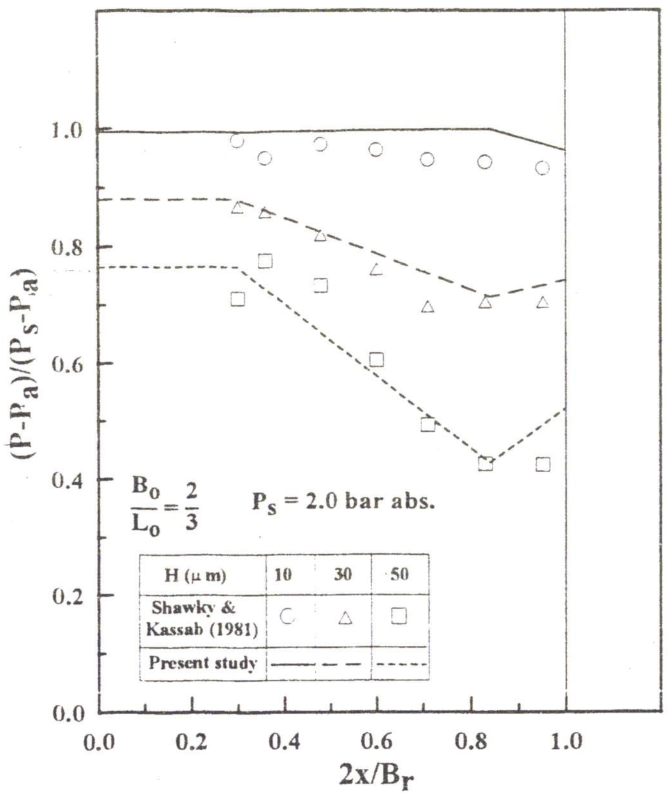

Fig. 9 Comparison between the results obtained using the present model and the experimental results of Shawky and Kassab (1981).

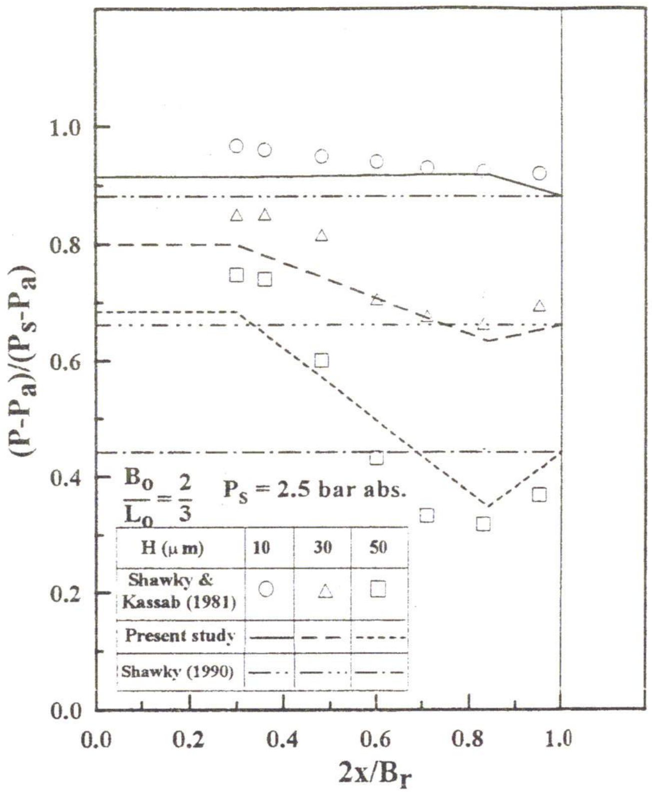

Fig. 10 Comparison between the results obtained using the present model and Shawky's (1990) model with experimental results of Shawky and Kassab (1981).

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