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JOURNAL BEARING ANALYSIS WITH SOFT, POROUS LUBRICATION

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Abstract: *A: Widely used in industrial machines and road vehicles, journal bearings play a crucial role in the functionality and longevity of these systems. These cylindrical hydrodynamic bearings rely on pressure-induced fluid films to support radial loads. However, this lubrication mechanism exhibits reduced effectiveness at lower rotational speed. Lubrication of soft, porous medium can address this weakness as it was found that it is able to generate a much higher fluid phase pressure, regardless of the speed. The fundamental idea of this mechanism involves replacing the lubricant film which separates the contacting surfaces with a low elastic modulus, porous layer imbibed with fluid. This study introduces an analytical model of journal bearing's fluid pressure generation, adapted with the compressible porous theory.*

Key words: *XPHD, Compressible, Porous, Journal Bearing, Theoretical Model.*

1. INTRODUCTION

The mechanism of compressible porous lubrication was studied relatively intensely over the last few decades, because of its high load capacity nature and undeveloped potential. Dedicated studies have been looking at different aspects of this regime, under various approaches. The initial research was conducted separately by two distinct research groups led by Pascovici [1-3] and Weinbaum [4]. Both groups formulated respective theoretical models predicting a lift generation phenomenon surpassing traditional hydrodynamic mechanisms by 2-4 orders of magnitude [3, 4]. Feng and Weinbaum [4] described the motion of blood cells where contribution of a soft, porous structure that coat the inner layer of blood vessels, so called Endothelial Glycocalyx was highlighted. Following the paper, a series of experimental and theoretical studies [5-10] were published, looking at pressure generation of planar gliding surfaces, capturing the mechanics of skiing and snowboarding. On the other hand, Pascovici invented a fluid pumping device utilizing the dislocation effect of a step surface [1]. A Reynolds-type equation was introduced and then adapted for both step and inclined sliders, under 1-D flow condition [3]. Despite of being proposed for varied scenarios, these studies converge on the same working principle of the regime, explained by reduced permeability within the pores caused by compression, resulting in elevated resistance to fluid flow, thus generating unmatched lift pressure. The most fitting term to characterize the mechanism is XPHD, an abbreviation coined by Pascovici in 2001, meaning Ex-poro-hydrodynamic [2].

There have been limited practical applications of XPHD, with notable examples including a comprehensive investigation by Kunik et al [11], introducing a thrust bearing configuration with parallel step slider. A train track proposal was suggested by Wu et al [6] back in 2004 but did not progress further. Despite its well-recognized lift advantage, literature review indicates minimal impact of the mechanism, showing the need of further development. The paper herein presents a theoretical configuration for a journal bearing, focusing on radial movements and loads. Through this analysis, ideal pressure estimations were derived, enabling calculations of carried load. The findings constitute an important addition to the understanding and advancement of bearings based on this method of lubrication.

The presence of a pre-compressed porous layer is essential during shaft installation to guarantee complete XPHD lubrication coverage. However, without actual experimental data, the parameters associated with this requirement remain undetermined. Thus, the analysis results presented here are based on an unspecified initial thickness of the porous layer and an indeterminate level of pre-compression. Consequently, the characteristic parameters of the porous layer were chosen as achievable values rather than being precisely tailored to any particular material.

2. MODELLING OF XPHD JOURNAL BEARING

For traditional hydrodynamic theory, the modelling of conformal contact bearing stood out as the most important and challenging aspect due to the intricate nature of the film's shape [4]. Hence, having a fully analytical model for journal geometry would facilitate the creation of a brand-new journal porous bearing that considers the drop of permeability of compressed layer. It's worth noting that the pressure estimated with the following calculations is the fluid phase pressure. The total load capacity of the bearing would be the sum of fluid phase's support and the reaction force from solid structure of the porous layer.

The Reynolds equation provided by Pascovici [3] for 1-D sliding surface under both squeezing and sliding motions for a compressible porous layer, imbibed in fluid:

$$\frac{d}{dx}\left(kh\frac{dp}{dx}\right) = \mu\left(u\frac{d(h\emptyset)}{dx} - V\emptyset\right) \qquad (1)$$

where p - pressure, x - coordinate in the direction of sliding, μ - dynamic viscosity, u -sliding velocity, V -normal velocity, k - permeability, h - separation between the solid surfaces, $\emptyset$ - porosity.

Several assumptions were required for the derivation of this formula and will remain in place for the following approach. The assumptions are:

- The fluid is Newtonian.
- The pressure changes across the thickness of the porous layer are negligible (porous layer is very thin).
- The layer only deforms in the normal direction.

In our case, only the rotation (sliding) motion is considered and hence $V = 0$. Applying this and integrate equation (1) gives:

$$\frac{dp}{dx} = \frac{\mu u\emptyset}{k} + \frac{\alpha}{kh} \qquad (2)$$

where α – arbitrary constant.
The equation that correlates the film thickness and the corresponding porosity is:

$$\frac{h}{h_0} = \frac{1-\emptyset_0}{1-\emptyset} \qquad (3)$$

where 0 denotes the parameters at highest porous layer's thickness/porosity ($\theta = 0$).

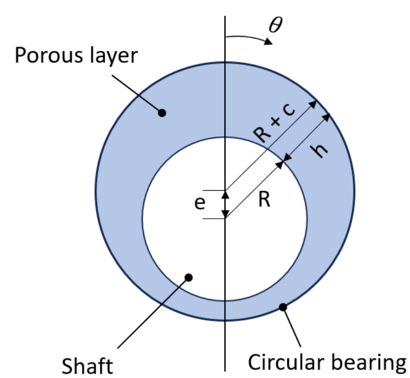

Fig. 1. Geometry of approximate porous layer thickness relation

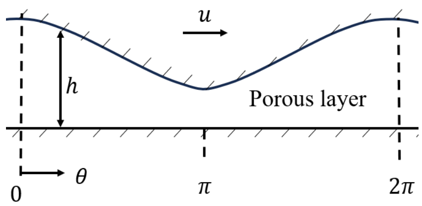

Fig. 2. Porous layer thickness unwrapped.

From the simplified geometry of journal bearing, the thickness of the porous layer corresponding to the angle coordinate is:

$h = c + e\,cos\theta = c(1 + \varepsilon cos\theta)$ (4)

c is the radial clearance of the bearing, e is the eccentricity, $\varepsilon = \frac{e}{c}$ which is the eccentricity ratio of the shaft [4].

From (3) and (4) yields the following equation of porosity with the corresponding angle coordinate:

$$\emptyset = 1 - \frac{(1+\varepsilon)(1-\emptyset_0)}{1+\varepsilon cos\theta} \quad (5)$$

The porosity and permeability for porous fiber matrix can be related by the Carman-Kozeny equation [3]:

$$k = \frac{D\emptyset^3}{(1-\emptyset)^2} \quad (6)$$

where $D = \frac{d^2}{16k_C}$ with dimensionless constant of k_C and d is the average fibre diameter.

k_C can be found via a permeability test, where the formula is curve-fitted to permeability-porosity plot. Some very detail permeability studies were presented by Turtoi et al [13] and Kunik et al [11] in which, an in plane permeameter design by Lundström et al [14] was adapted.

Substitute (5) to (6) yields:

$$k = D * \frac{[1+\varepsilon cos\theta - (1+\varepsilon)(1-\emptyset_0)]^3}{(1+\varepsilon cos\theta)[(1+\varepsilon)(1-\emptyset_0)]^2} \quad (7)$$

And hence,

$$kh = Dc\frac{[1+\varepsilon cos\theta - (1+\varepsilon)(1-\emptyset_0)]^3}{[(1+\varepsilon)(1-\emptyset_0)]^2} \quad (8)$$

By adapting (5), (7), (8) and $x = Rd\theta$, equation (2) becomes:

$$\frac{dp}{Rd\theta} = \frac{\mu u[(1+\varepsilon)(1-\emptyset_0)]^2}{D}\left\{\frac{1}{[1+\varepsilon cos\theta - (1+\varepsilon)(1-\emptyset_0)]^2} + \frac{\alpha}{c[1+\varepsilon cos\theta - (1+\varepsilon)(1-\emptyset_0)]^3}\right\} \quad (9)$$

The integration of equation (9) yields:

$$\bar{p} = \left\{\frac{2[\varepsilon(-1+\emptyset_0)+\emptyset_0]\lambda}{(1+\varepsilon)^{3/2}{\emptyset_0}^{3/2}[\varepsilon(-2+\emptyset_0)+\emptyset_0]^{3/2}} - \gamma\right\} + \frac{\alpha}{c}\left(\frac{\left(4\varepsilon(-1+\emptyset_0)\emptyset_0+2{\emptyset_0}^2+\varepsilon^2(3-4\emptyset_0+2{\emptyset_0}^2)\right)\lambda}{(1+\varepsilon)^{5/2}{\emptyset_0}^{5/2}[\varepsilon(-2+\emptyset_0)+\emptyset_0]^{5/2}} - \frac{\gamma}{2} - \frac{3\gamma[\varepsilon(-1+\emptyset_0)+\emptyset_0]}{2(1+\varepsilon)\emptyset_0[\varepsilon(-2+\emptyset_0)+\emptyset_0]}\right) + \beta \quad (10)$$

Where $\lambda = \text{ArcTan}\left[\frac{\sqrt{\varepsilon(-2+\emptyset_0)+\emptyset_0}\,\text{Tan}\left[\frac{\theta}{2}\right]}{\sqrt{1+\varepsilon}\sqrt{\emptyset_0}}\right]$

$$\gamma = \frac{\varepsilon\text{Sin}[\theta]}{(1+\varepsilon)\emptyset_0[\varepsilon(-2+\emptyset_0)+\emptyset_0][\varepsilon(-1+\emptyset_0)+\emptyset_0+\varepsilon\text{Cos}[\theta]]}$$

$$\bar{p} = \frac{pD}{\mu uR[(1+\varepsilon)(1-\emptyset_0)]^2}$$

and β is an arbitrary constant.

To solve for pressure, the boundary conditions are required, the pressure at the highest porous layer thickness and lowest porous layer thickness is zero:

- $\bar{p} = 0$ when $\theta = 0$

$$(0+0) + \frac{\alpha}{c}(0+0-0) + \beta = 0 \quad (11)$$

Hence, $\beta = 0$.

- $\bar{p} = 0$ when $\theta = \pi$ (Half-Sommerfeld)

$$\frac{\alpha}{c}\left[\frac{[4\varepsilon(-1+\emptyset_0)\emptyset_0+2{\emptyset_0}^2+\varepsilon^2(3-4\emptyset_0+2{\emptyset_0}^2)]\lambda_\pi}{(1+\varepsilon)^{5/2}{\emptyset_0}^{5/2}[\varepsilon(-2+\emptyset_0)+\emptyset_0]^{5/2}}\right] + \frac{2[\varepsilon(-1+\emptyset_0)+\emptyset_0]\lambda_\pi}{(1+\varepsilon)^{3/2}{\emptyset_0}^{3/2}[\varepsilon(-2+\emptyset_0)+\emptyset_0]^{3/2}} = 0 \quad (12)$$

Where $\lambda_\pi = \text{ArcTan}\left[\frac{\sqrt{\varepsilon(-2+\emptyset_0)+\emptyset_0}\,\text{Tan}\left[\frac{\pi}{2}\right]}{\sqrt{1+\varepsilon}\sqrt{\emptyset_0}}\right]$

$$\frac{\alpha}{c} = -\frac{2(1+\varepsilon)\emptyset_0[\varepsilon(-2+\emptyset_0)+\emptyset_0][\varepsilon(-1+\emptyset_0)+\emptyset_0]}{4\varepsilon(-1+\emptyset_0)\emptyset_0+2{\emptyset_0}^2+\varepsilon^2(3-4\emptyset_0+2{\emptyset_0}^2)} \quad (13)$$

Substitute back to (10), the pressure distribution formula is now completed. Equation (10) now represents the pressure estimation model for a 1-D journal compressible porous bearing. The pressure is then plotted accordingly in ***Fig. 3, 4*** and ***5.***

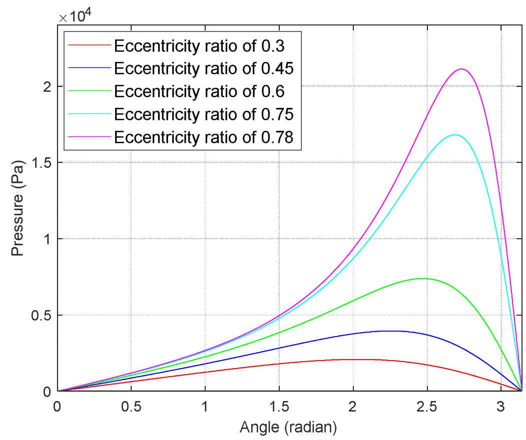

Fig. 3. Pressure distribution for journal compressible porous bearing at different eccentricity ratios

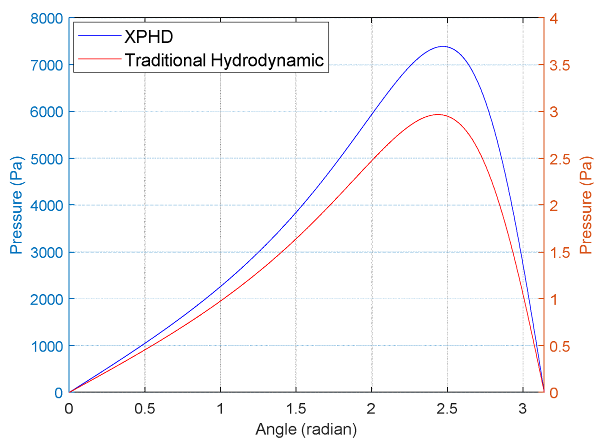

Fig. 4. Pressure distribution comparison with traditional hydrodynamic journal bearing.

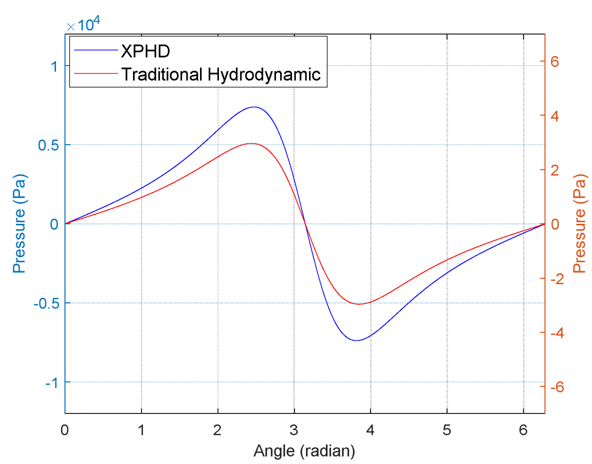

Fig. 5. Pressure distribution with negative pressure

3. DISCUSSION

The expression of pressure distribution is quite extensive, but it bears resemblance to the classic hydrodynamic one. The general shape of the distribution curve appears strikingly familiar, exhibiting peaks at 2-3 radians and levelling off at 0 and 3.14 radians, as per ***Fig. 4.*** The graph for the traditional regime is provided by derivations from Cameron [12]. The XPHD model is represented by the blue line curve in ***Fig. 4*** and ***Fig. 5***, aligned with left-hand y axis, while HDL is the red line, associated to right-hand y axis. For these plots, the eccentricity ratio of $\varepsilon = 0.6$ was used and the radial clearance were calculated accordingly using equation (4).

A representative set of data was used, with the sliding velocity of $u = 0.06$ m/s, shaft radius of $R = 0.05\ m$. The porous layer's uncompressed thickness is 3.66 mm, has an original porosity of 0.976 and complexity constant $D = 3.8 * 10^{-13}$.

It's very interesting when comparing the pressure generated in two different regimes. The XPHD model displayed a clear superiority with peak pressure estimated to be 4 orders of magnitude larger than that of the hydrodynamic (HD) one (for the representative operating condition). The finding is in close alignment with the results obtained for inclined and parallel step pads [3].

During the derivation process, the authors made some intriguing observations. First, the mathematical structure of the permeability-porosity equation helped simplify the pressure formula greatly. The Carman-Kozeny equation was selected ingeniously in Pascovici's paper [3] which introduced the Reynolds equation and presented derivation for 1-D sliding pads. This led to a considerable reduction in the complexity of the finalized theoretical models, facilitating the integration. The formulas presented hereby benefits from this neat selection as it enables the pressure to be calculated directly, without relying on numerical estimations. While alternative permeability equations were considered in the models, none approached the same level of simplification. The Half-Sommerfeld boundary condition was employed to solve for the exact solution of the integration. If the fluid-solid film does not rupture, sustaining the negative pressure then the Sommerfeld boundary condition is applied. The pressure curve would be inverted after $\theta = \pi$ and returns to zero at $\theta = 2\pi$ as presented in ***Fig. 5***.

The load carried by the fluid phase can be calculated from the following formulas, which were used for fluid film bearing [4]:

$$\frac{W_x}{B} = \int_0^{\pi} pRcos\theta d\theta$$

$$\frac{W_y}{B} = \int_0^{\pi} pRsin\theta d\theta$$

Where W_x and W_y are load carried in x and y direction respectively, B is the thickness of the bearing in z direction.

For the representative operating condition, the total load carried for XPHD bearing was found to be 463.33 N/m compared to only 0.19 N/m of the HD regime.

The significant difference and the total dominance in favour of the new mechanism aligned with theoretical reviews for various sliding geometries [3, 4].

4. CONCLUSION

An analytical model of a soft, porous journal bearing is introduced, enabling pressure distribution and load capacity to be calculated. The obtained results show a load-carrying capacity three orders of magnitude higher than that of conventional fluid film mechanism. These findings align consistently with established models across various bearing configurations [2, 3].

Additionally, the study emphasizes the prevailing load advantage of the later regime, highlighting the potential for an innovative bearing design.

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Lagar de alunecare cu lubrificatie cu strat poros elastic

Abstract: Folosite intens in utilaje industrial si vehicule rutiere, lagarele cu alunecare joaca un rol crucial in functionarea si longevitatea acestor sisteme. Aceste lagare hidrodinamice cilindrice se bazeaza pe presiunea indusa in filmul de lubrifiant ca sa suporte sarcini radiale. Este cunoscut, pe de alta parte ca aceste lagare au o efectivitate redusa la viteze reduce ale arborelui. Lubrificatia cu un mediu poros elastic poate adresa aceast dezavantaj deoarece s-a demonstrat ca este capabila sa genereze o presune in film mult mai ridicata chiar la viteze mai reduse. Idea fundamentala a acestui mecanism de lubrifiere implica inlocuirea filmului fluid de lubrifiant care separa suprafetele de alunecare cu un strat poros cu module de elsaticitate redus si imbibat cu fluid. Acest studiu introduce un model analitic a presiunii lagarului cilindric, adaptat pentru lubrificatia cu strat poros compresibil.

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