International Journal of Advance Research and Innovation
Vol. 4(3), Jul-Sept 2016, pp. 116-122
Doi: 10.51976/ijari.5417018
www.gla.ac.in/journals/ijari

Article Info

Received: 25 Jan 2016 | Revised Submission: 20 Feb 2016 | Accepted: 28 Feb 2016 | Available online: 15 Jun 2016

CFD Investigation of Clearance on Pressure Distribution and Fluid Film Thickness in Hydrodynamic Journal Bearing

Amit Mahajan, RK Awasthi**, Sarabjeet S Sidhu***, Sandeep Devgan**** and Harpuneet Singh******

ABSTRACT

Clearance plays a vital role in fluid film journal bearing and thus it's important to consider its effect on performance prediction. In order to ascertain the extent to which relative clearance affects the performance of journal bearing system, this paper investigates the pressure distribution and the fluid film thickness in axial and circumferential directions of bearing at relative clearances. Under the steady state condition, the two-dimensional Reynolds equation is solved using finite difference based CFD method. The pressure of the fluid film is computed by using ADI technique. The code developed in FORTRAN 90 is debugged and validated for its accuracy. The results indicate that the fluid film pressure rises and the minimum fluid film thickness decrease with the reduction in relative clearance, which requires the optimal value of clearance ratio for better performance.

Keywords: *Relative Clearance; Length to Diameter Ratio; Minimum Oil Film Thickness; Reynolds Equation.*

1.0 Introduction

In hydrodynamic bearing, the fluid film separates the bearing and journal and thus permits a relative motion between the contacts surfaces with minimal friction. This fluid film reduces the wear of machine elements and also carries away the generated heat (Tiwari and Kumar, (2012)). Clearance $[c=R-r]$ between the journal and bearing shells is the most important factors when it comes to the health and durability of assembly because that open area is filled with oil which provides a cushion between the journal itself and the bearing (Bhandari,(2010)). A few micron variations in the clearance may cause serious changes in different performance parameters of the bearing (Chu and Kay,(1974)). When everything in the assembly of journal bearing is accurate, the oil keeps the journals and the bearings separated, and they can't come in contact with each other. But a problem arises when the bearing clearances aren't correct. Thus, the control of this parameter is necessary for achieving high performance. This important parameter can be managed at three stages of bearing life cycle such as designing, manufacturing and usage (Sharma et al. (2009)). Fig.1 shows the geometry of fluid film journal bearing.

(Ocvfik and D'ubois(1958)) scrutinized the effect of bearing-clearance on the friction power

loss, the film thickness, and the peak pressure in the oil film. They reported that the load capacity passes through a maximum value at a small bearing clearance and then decreases with increasing bearing clearance. (Mitsui (1986)) concluded that with the decrease of clearance ratio, there is an increment in the maximum bearing temperature, speed and lubricant viscosity. (Prasad (1988)) presented the effects of viscosity and clearance on the performance of hydrodynamic Journal Bearings. The higher clearance leads to an increase in the axial flow rate particularly under high applied loads and high rotational speeds (Pierre and Fillon (2000)). (El-Kersh et al. (2001)) analyzed the effect of thermal expansion on three polymer composite journal bearings at different rotational speeds and clearance

**Correspoinding Author: Department of Mechanical Engineering, Khalsa College of Engineering & Technology, Amritsar, Punjab (E-mail: amitmahajan291@gmail.com)*
***Department of Mechanical Engineering, Beant College of Engineering &Technology, Gurdaspur, Punjab, India*
****Department of Mechanical Engineering, Beant College of Engineering &Technology, Gurdaspur, Punjab, India*
*****Department of Mechanical Engineering, Khalsa College of Engineering & Technology, Amritsar, Punjab*
******Department of Mechanical Engineering, Khalsa College of Engineering & Technology, Amritsar, Punjab*

ratio. They found that, the increase of bearing clearance ratio leads to increase the friction coefficient and minimum oil film thickness and at the same time reduces the maximum bearing temperature and load carrying capacity. Similar observations were made by (Abass et al. (2007)). They scrutinized the counter rotating floating ring journal under different working conditions. Their investigation revealed that by decreasing the radii ratio (R2/R1) of the ring and clearance ratio (c1/c2), the coefficient of friction decreases at the same time, with this the load carrying capacity of the bearing increases. (Gangrade and phalli (2016)) found that the performance of hydrodynamic journal bearing is mainly affected by clearance ratio, aspect ratio and speed.

In the present study, the effect of clearance on pressure distribution and fluid film thickness of fluid film journal bearing has been investigated. The behavior of these performance parameters at different relative clearance in circumferential and axial direction is scrutinized.

2.0 Analysis

Using the conservation of global mass in a fluid-film and incorporating Elrod algorithm, an expression for conservation of mass in journal bearings is given by (Vaidyanathan and Keith Jr, (1989))

$$\frac{\partial}{\partial t}(h\rho)+\frac{\partial}{\partial x}\left(\dot{m}_x\right)+\frac{\partial}{\partial z}\left(\dot{m}_z\right)=0 \quad (1)$$

Equation [1] represents the mass conservation equation: the first term on the left is the squeeze term, while the second and the third term are the net mass flow rate in the *x* –direction (i.e., the flow direction) and in the *z* -direction (i.e., the axial direction) respectively. Where the mass flux $(\dot{m}_x)$ is composed of a shear flow term (coquette flow) and pressure induced term (Poiseulle flow) and $(\dot{m}_z)$ composed of only pressure induced term, On rearranging the equation (1) will be

$$\frac{\partial}{\partial x}\left(\frac{h^3\rho}{12\mu}\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3\rho}{12\mu}\frac{\partial P}{\partial z}\right)=\frac{\partial \rho h}{\partial t}+\frac{U}{2}\frac{\partial(\rho h)}{\partial x} \quad (2)$$

Incorporating elrod algorithm, a single expression for conservation of mass in journal bearings is given by (Vaidyanathan and Keith Jr, (1989)): Incorporating elrod algorithm, a single expression for conservation of mass in journal bearings is given by (Vaidyanathan and Keith Jr, (1989)):

$$\frac{\partial}{\partial x}\left(\frac{\rho_c g\beta h^3}{12\mu}\frac{\partial \alpha}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho_c g\beta h^3}{12\mu}\frac{\partial \alpha}{\partial z}\right)=\frac{U}{2}\frac{\partial(\rho_c \alpha h)}{\partial x} \quad (3)$$

After applying finite difference method and utilizing alternating direction implicit (ADI) technique, the resulting expression be:

$$-F_{i,j}\alpha_{i+1,j}^{n+1}+D_{i,j}\alpha_{i,j}^{n+1}-E_{i,j}\alpha_{i1,j}^{n+1}=A_{i,j}\alpha_{i,j+1}^{n+1/}+B_{i,j}\alpha_{i,j-1}^{n+1/2}+C_{i,j} \quad (4)$$

Where

$$A_{i,j}=a\left[(\bar{h}_{i,j+1}+\bar{h}_{i,j})^3 g_{i,j+1}\right]$$

$$B_{i,j}=b\left[\bar{h}_{i,j-1}\left(1-g_{i,j-1}\right)\right]+a\left[(\bar{h}_{i,j}+\bar{h}_{i,j-1})^3 g_{i,j-1}\right]$$

$$F_{i,j}=f\left(\bar{h}_{i,j}^{3}\, g_{i-1,j}\right)$$

$$D_{i,j}=f\left[\left(\bar{h}_{i,j}^{3}+\bar{h}_{i,j}^{3}\right)g_{i,j}\right]+b\left[\bar{h}_{i,j}\left(1-g_{i,j}\right)\right]+a\left[(\bar{h}_{i,j}+\bar{h}_{i,j-1})^3 g_{i,j}\,(\bar{h}_{i,j}+\bar{h}_{i,j+1})^3 g_{i,j}\right]$$

3.0 Validation of Code

The mathematical model using FDM is generated for the present study and the solution for this model is developed by using Fortran 90. In order to validate the accuracy of the developed code, the simulated results are compared with the published results.

The computational grid selected for this purpose is 72 x 10 (with 73 nodes in circumferential direction and 11 nodes in axial direction). This computational grid is selected as a compromise between accuracy and computational time. For validation of code, results for Sommerfeld number versus eccentricity ratio are compared with already published results of Vaidyanathan, K. and Keith Jr, T.G. (1989).

It is clear from Fig.2 and Table-1 that the data of published work and present work are nearly equal, which shows the accuracy of the code. Once the code is validated, the simulated results we obtain from computer code could be used for understanding the fluid film bearing system. The bearing system considered in the present work is rigid. The code developed does not account for any deformation of the bearing occurs owing to the hydrodynamic pressure of fluid film. Also, the code does not account for the effect of viscosity variation which may occur due to viscous friction resulting the heat generation.

$$C_{i,j} = b\Big\{\Big[\bar{h}_{i,j-1}^{3}\, g_{i,j-1} + \frac{g_{i,j}\, g_{i,j-1}}{2}\big(\bar{h}_{i,j} - \bar{h}_{i,j-1}\big)\Big] - \Big[\bar{h}_{i,j}\, g_{i,j} + \frac{g_{i,j}\, g_{i,j+1}}{2}\big(\bar{h}_{i,j+1} - \bar{h}_{i,j}\big)\Big]\Big\} + a\Big[\big(\bar{h}_{i,j} + \bar{h}_{i,j}^{3}\big)\big(g_{i,j} - g_{i,j-1}\big) + \big(\bar{h}_{i,j} + \bar{h}_{i,j+1}\big)^{3}\big(g_{i,j} - g_{i,j+1}\big)\Big] - f\,\bar{h}_{i,j}^{3}\big(g_{i+1,j} - 2\,g_{i,j} + g_{i-1,j}\big)$$

The minimum film thickness $(\bar{h})$ for equilibrium conditions (i.e. journal centre equilibrium position) can be obtained by Equation [7] (D'ubois and Ocvfik (1953)),

$$h = c - X_j\cos(\alpha) - Z_j\sin(\alpha) \quad [5]$$

$$\frac{h}{c_o} = \frac{c}{c_o} - \frac{X_j}{c_o}\cos(\alpha) - \frac{Z_j}{c_o}\sin(\alpha) \quad [6]$$

$$\bar{h} = c_{rel} - \bar{X}_j\cos(\alpha) - \bar{Z}_j\sin(\alpha) \quad [7]$$

Where $\alpha = \dfrac{\rho}{\rho_c}$

Fig: 1-Geometry of Journal Bearing

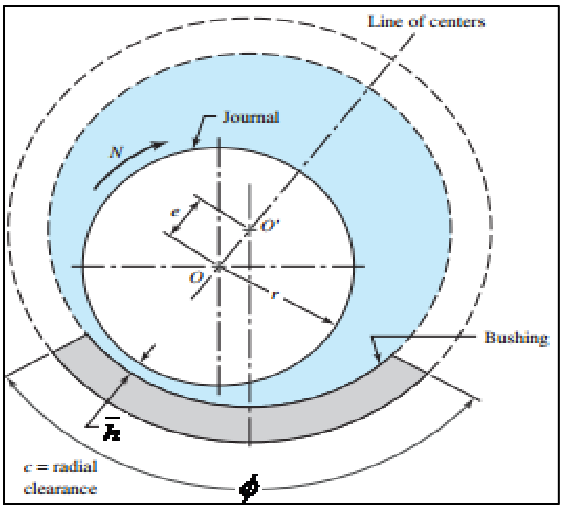


Fig: 2- Comparison of Results (Horizontal Scale is Logarithmic)

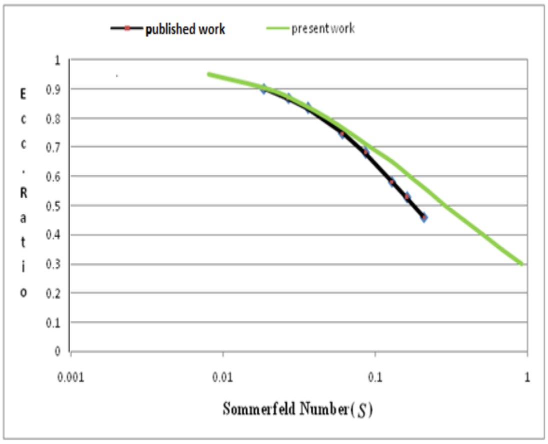


Fig:3- Pressure Distributions in Circumferential Direction

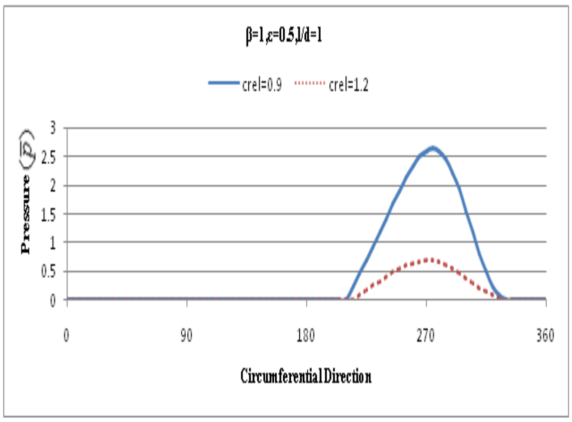


Fig 4: Pressure Distributions in Axial Direction

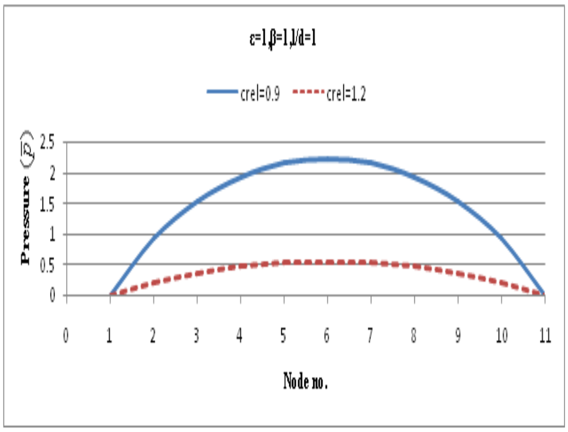


Fig 5: Minimum Film Thickness in Circumferential Direction

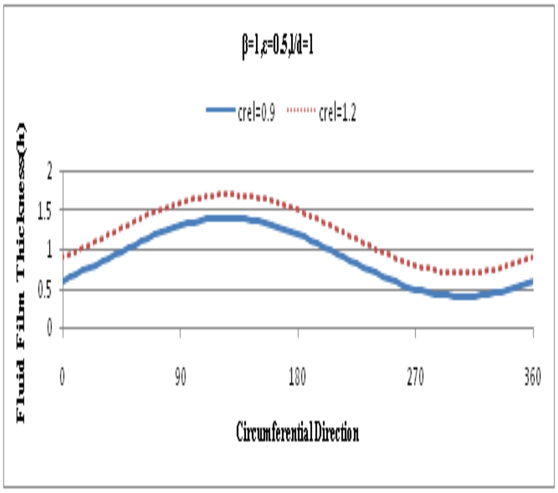
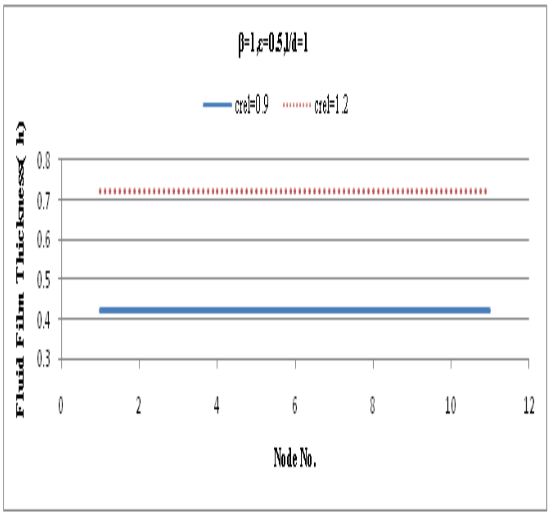


Fig 6: Minimum Film Thickness in axial direction



4.0 Results and Discussion

The input parameters used in the current study is described in Table 2.

The range of the parameters selected are most common, and as per the published work.

The influence of relative clearance at fixed Bulk Modulus, aspect ratio and eccentricity ratio on the pressure distribution and minimum film thickness of fluid-film journal bearing in terms of dimensionless values have been described.

Fig 7: Fluid Film Thickness $(\bar{h})$ Versus Relative Clearance For Different Bearing Aspect Ratio (l/d)

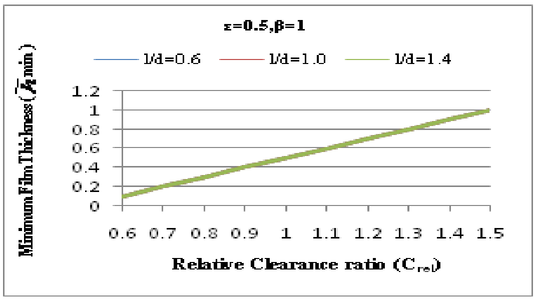


Table 1: Comparison of Results for Validation of Code

Eccentricity ratio	Sommerfeld Number Present work	Sommerfeld Number Published work
0.5	0.28	0.18
0.6	0.16	0.12
0.7	0.09	0.07
0.8	0.04	0.04
0.9	0.02	0.018

Table 2: Input Parameters Used in the Current Study

Geometrical Parameters	Aspect Ratio ($\frac{L}{D}$) Relative Clearance (c_{rel})	0.6,1.0,1.4 0.6-1.5
Operating parameters	Bulk Modulus (β) Eccentricity Ratio (ε) Speed parameter (Ω)	1.0 0.5 1,3,5
Grid size	No. of elements in circumferential direction No. of elements in axial direction	72 10

4.1 Pressure distribution $(\overline{P})$ in circumferential direction

To study the influence of relative clearance over pressure profile of fluid film journal bearings is observed at fixed eccentricity 0.5. Fig.3 shows the pressure profile at different relative clearance resulting in the bearing pressure of fluid film journal bearings which achieved a higher value at small relative clearance. The maximum pressure field occurs around the bottom most portion of the journal bearing for all relative clearances. It is clear that from θ = 0 to 180 degrees oil film pressure is zero. The pressure starts rising near θ = 200 degrees and becomes maximum near to 270 degrees, this region from θ degree to 275 degrees represents the full-film convergent section of fluid film bearing system (which is a requirement for positive pressure generation). After that, the region from θ = 275 to 315 degrees of pressure fall represents the full-film divergent section of the fluid film bearing system and pressure gradient starts falling in this region. A relative clearance decreases yields increase in pressure, which is tending to increase the hydrodynamic action or load carrying capacity.

4.2 Pressure distributions in axial direction

In order to visualise the effect of pressure distribution along the bearing length, the pressure corresponding to lower portion of the bearing is plotted and are shown in Fig.4. It represents that at relative clearance 0.9, the pressure becomes maximum and as relative clearance increases up to 1.2 it lowers down.

4.3 Fluid Film Thickness in circumferential direction

Fig.5 shows that as relative clearance increases, along with that minimum fluid film thickness also increases. It shows that the bearing is unwrapped, the minimum fluid film thickness starts increasing up to 125 degrees and then it starts decreasing up to 310 degrees and then it slightly increases up to 360 degrees.

4.4 Fluid Film Thickness in axial direction

Fig.6 shows that the fluid film thickness remains constant throughout the bearing in axial direction. As it has been explained in Fig 5, as the relative clearance increases, the fluid film thickness also increases; same trend is followed in this section.

4.5 Fluid film thickness .$(\overline{h})$ versus relative clearance for different bearing aspect ratio (l/d)

Fig. 7 represents that with the increase in the relative clearance there is change in the value of minimum fluid film thickness.

The bearing length to diameter ratio (l/d) has no effect on the minimum fluid film thickness. It remains constant at 0.6, 1.0 & 1.4. So it is established that whether the bearing is small or long there is no change in minimum fluid film thickness.

5.0 Conclusions

In hydrodynamic journal bearing, change of relative clearance produces an intensive effect on different performance parameters.

Pressure distribution and fluid film thickness are significant performance parameters in hydrodynamic journal bearing. Effect of relative clearance on these performance parameters in axial and circumferential directions are scrutinized in this study. Furthermore, the influence of variation in aspect ratio on fluid film thickness and relative clearance are also examined.

1 In the circumferential direction, the pressure starts rising near θ = 200 degrees and becomes maximum near to 270 degrees. Relative clearance decreases yielding the pressure profile rise, which is tending to increase hydrodynamic action. Whereas in the case of fluid film thickness, it starts increasing up to 125 degrees and then decreasing up to 310 degrees.
2 In axial direction, the fluid film thickness remains constant throughout the bearing but pressure rises at the centre of the bearing.
3 The bearing length to diameter ratio (l/d) has no effect on the minimum fluid film thickness.

4 It is understood from this study that the performance of bearing parameters is more efficient at small bearing clearances. But this study could not explain a single value of clearance on which the bearing performs efficiently. So this study can be extended to find that single value of clearance at which bearing performance parameters works appropriately.

Nomenclature

ε : Eccentricity ratio, *(e/c)*
e : Eccentricity
ϕ : Attitude angle, (*radians)*
c : Radial clearance
c_o : Original radial clearance
c_{rel} : Relative radial clearance (c/c_o)
c_r :Clearance ratio
ρ : Density of lubricant, $(\frac{kg}{m^3})$
p : Fluid film pressure, $\frac{N}{m^2}$
ε : Eccentricity ratio, *(e/c)*
e : Eccentricity
ϕ : Attitude angle, (*radians)*
c : Radial clearance
c_o : Original radial clearance
c_{rel} : Relative radial clearance (c/c_o)
c_r :Clearance ratio
ρ : Density of lubricant, $(\frac{kg}{m^3})$
p : Fluid film pressure, $\frac{N}{m^2}$
p_c : Cavitation pressure, $\frac{N}{m^2}$
h : Film thickness, *m*
g : Switch function (cavitation index)
μ : Dynamic viscosity, $\frac{N}{s}m^2$
β : Bulk modulus
U : Mean surface velocity
i : Index for axial direction
j : Index for circumferential direction
x : Circumferential coordinate (direction)
z : Axial coordinate (direction)
$\overline{x_j}$: $\frac{x_j}{c_o}$ $\overline{h}$: $\frac{h}{c_o}$ $\overline{z_j}$: $\frac{z_j}{c_o}$

References

[1] Tiwari, P. and Kumar, V. Analysis of Hydrodynamic Journal Bearing: A Review, Int. J. Eng. Res. Technol., 1, 7, ESRSA Publications, 2012.

[2] Bhandari, B. V. Design Of Machine Elements, 3rd Edition, Tata Mcgraw Hill Education Private Ltd. New Delhi, India, ISBN-13: 978-0-07-068179-8, 2010.

[3] Chu, S. P. and Kay, E. Optimum Clearance Fits for Journal Bearings in Relation to BS 1916 and to Lubrication Theory, Wear, 27, 3, 1974, 329-343.

[4] Sharma, S., Hargreaves, D. and Scott, W. Journal bearing performance and metrology issues. J. Achiev. Mater. Manuf. Eng., 32, 1, 2009, 98-103.

[5] Ocvfik, W. F. and D'ubois B.G. Relation of Journal Bearing Performance to minimum oil film, NACA, TN 4223, 1958.

[6] Mitsui, J., Hori, Y. and Tanaka, M. An Experimental Investigation on the Temperature Distribution in Circular Journal Bearings, ASME J. Trib., 108, 4, 1986,621- 626.

[7] Prashad, H. The Effects of Viscosity and Clearance on the Performance of Hydrodynamic Journal Bearings, Trib. Trans. 31, 2, 1988, 303-309.

[8] Pierre, I. and Fillon, M. Influence of Geometric Parameters and Operating Conditions on the Thermohydrodynamic Behavior of Plain Journal Bearings" Proc. of the Institution of Mechanical Engineers, Part J: J. Eng. Trib., France, Sage publication, 214, 5, 2000, 445-457.

[9.] El-Kersh, A.M. and Ali, W.Y. The effect of thermal expansion and elastic deformation on the performance of polyamide-copper journal bearings, KGK Kautschuk Gummi kunststoffe, 54, (9), 2001,468-473.

[10.] Abass, B. A., Alwan, A. A., and Hunain, M B. An Investigation into the Performance of Counter Rotating Floating Ring Journal Under Different Working Conditions, Al-Khwarizmi Eng. J.,3,1, 2007, 26-39.

[11.] Gangrade, A. K., and Phalle, V. M. Effect of Clearance And Aspect Ratio On The Performance Of Water Lubricated Hydrodynamic Journal Bearing, Int. J. Global Technol. Initiatives, 5, 1, 2016, D59-D64.

[12.] Vaidyanathan, K. and Keith Jr, T.G. Numerical Prediction Of Cavitation in Non Circular Journal Bearings, Trib. Trans, 32, 2, 1989, 215-224.

[13.] Budynas, G. R. and Nisbett, K. J. Shigley's Mechanical Engineering Design, 10th Edition, McGraw-Hill Education, New York, ISBN 978-0-07-339820-4. 2015.

[14.] D'ubois B.G. and Ocvfik, W. F. Analytical Derivation and Experimental Evaluation of Short-Bearing Approximation for Full Journal Bearing, NACA-TR-1157, 1953

.