# Effect of Frequency and Load on Minimum Film Thickness in Gas Lubricated Journal Bearings

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## ABSTRACT

*Gas lubricated journal bearing are an attractive means of support for rotating members in micro fabricated machines. Their low drags and near zero wear makes them particularly important to high speed machines. The principal of an air or gas bearing is simple, when two surfaces form a wedge, and one surface moves relative to the other surface, pressure is generated between the surfaces because of the hydrodynamic action of the fluid which carries load. In this paper, the analysis of one dimensional dynamic loading of gas lubricated journal bearing has been developed. The effect of load on minimum film thickness with respect to time has been studied in this work. The present analysis has been carried out under the assumption of isothermal condition for the sake of simplicity. The Reynolds' equation is discretized using finite difference method and a computational algorithm based upon Newton-Raphson technique has been developed. The variation of gas bearing characteristics in terms of minimum film thickness, eccentricity ratio, attitude angle and friction coefficient has been studies with respect to time.*

**Key Words: -** *Journal Bearing, Minimum film thickness, Gas lubricated Journal Bearing.*

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## 1. INTRODUCTION

The self-acting gas-lubricated journal bearing as illustrated in the Fig. 1.1, consists of two non concentric cylinders, where c is the radial clearance, e the eccentricity, L the bearing length, Ob is the centre of the bearing, Oj is the centre of the journal, R is the journal radius, W the applied load, WH in the direction of the centers, WV normal to the direction of the line of centers, θ, z the circumferential coordinate and axial coordinate, respectively, β the attitude angle, ω angular velocity of the journal. The principle of operation is same for both hydrodynamic or a hydrostatic bearing using gas as a lubricant and oil lubricated bearing. The difference is that the gases have very low coefficient of absolute viscosity as compare to oils. Hence the viscous resistance is very much less. The film thickness in case of gas lubrication is lesser then oil lubricated bearing. The minimum film thickness may be of the same order as the surface roughness of the journal and the bearing if the surface finish is not very good. Also a slight waviness of the surfaces will cause the fluid to alternatively expand and compress which distorts the pressure profile and the pattern of flow [1-12].

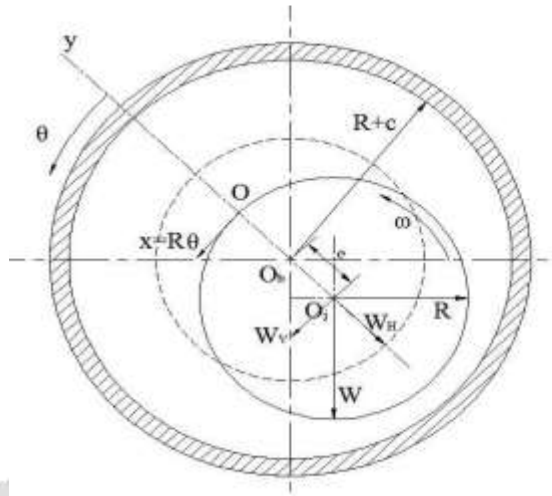


**Fig. 1.1 Schematic diagram of self-acting gas-lubricated journal bearings.**

**1.1 Reynolds Equation for Newtonian fluid in non-dimensional form**

As derived in the previous chapter Reynolds equation for Newtonian fluid in dimensional form is

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)=\frac{u}{2}\frac{\partial h}{\partial x} \quad (1.1)$$

The oil film thickness can be written as a function of x:

$$h = h_o + s_h\left(1-\frac{x}{l}\right) \quad (1.2)$$

In dimensionless form equation 1.1 & 1.2 become

$$\frac{\partial}{\partial X}\left(H^3\frac{\partial P}{\partial X}\right)=A_k\frac{\partial H}{\partial X} \quad (1.3)$$

$$H = H_o + 1 - X \quad (1.4)$$

Where, $P=\frac{p}{p_o}$ $p_o=\frac{w}{l}$

$H=\frac{h}{s_h}$ $H_o=\frac{h_o}{s_h}$

$X=\frac{x}{l}$ and $A_k=\frac{6\eta u l^2}{w s_h^2}$

Now, from equation 1.4

$$\frac{\partial H}{\partial X}=-1$$

Then equation 1.3 becomes

$$\frac{\partial}{\partial X}\left(H^3\frac{\partial P}{\partial X}\right)+A_k=0 \quad (1.5)$$

Now this equation 1.5 is used to develop a FORTRAN program to calculate pressure distribution in journal bearing along x axis for Newtonian lubrication behavior.

**1.2 Load Equilibrium Equation**

In a quasi steady state problem the net radial force resulting from the fluid film pressure should always be equal to applied load. And the same quasi steady state problem has been assumed in this paper. The load carrying capacity of the oil film per unit length is:

$$\int_{x_i}^{x_a} p\,dx = W \tag{1.6}$$

Here, $W$ is the load imposed

The above equation is expressed in non-dimensional form as follows

$$\int_{x_i}^{x_a} P\,dX = 1 \tag{1.7}$$

The above integral is calculated using Simpson's rule and it can be written in the following form:

$$\Delta W = \sum_{j=2}^{N} C_j P_j - 1 = 0 \tag{1.8}$$

Where, $C_j = \begin{cases} \Delta X/3 & j=1 \\ 4\Delta X/3 & j=2,4,6... \\ 2\Delta X/3 & j=3,5,7... \end{cases}$

## 1.3 Newton-Raphson Formulation

The simultaneous system of N equations represented by discretized Reynolds equations (1.5) and discretized load equilibrium equation are solved using the Newton-Raphson technique. The N system unknowns are P2, P3, P4,…., PN-1 and $H_0$. The matrix equation of this system is

[J][Δ]=[F], i.e.,

$$\begin{bmatrix} \frac{\partial f_2}{\partial P_2} & \frac{\partial f_2}{\partial P_3} & \cdots & \frac{\partial f_2}{\partial P_N} & \frac{\partial f_2}{\partial \bar{\varepsilon}} \\ \frac{\partial f_3}{\partial P_2} & \frac{\partial f_3}{\partial P_3} & \cdots & \frac{\partial f_3}{\partial P_N} & \frac{\partial f_3}{\partial \bar{\varepsilon}} \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ \frac{\partial f_N}{\partial P_2} & \frac{\partial f_N}{\partial P_3} & \cdots & \frac{\partial f_N}{\partial P_N} & \frac{\partial f_N}{\partial \bar{\varepsilon}} \\ C_2 & C_3 & \cdots & C_N & 0 \end{bmatrix} \begin{bmatrix} \Delta P_2 \\ \Delta P_3 \\ \cdots \\ \cdots \\ \Delta P_N \\ \Delta \bar{\varepsilon} \end{bmatrix} = - \begin{bmatrix} f_2 \\ f_3 \\ \cdots \\ \cdots \\ f_N \\ \Delta W_0 \end{bmatrix}$$

# 2. Solution Procedure

### 2.1 Input Data

For the calculation of minimum film thickness ($H_{min}$) with respect to time some input values of different parameters like load, speed, frequency and amplitude are used. These values of load, speed etc. are used in the program to see their effect on $H_{min}$ with respect to time. The input values of these parameters are given in the following table.

**Table 2.1** Input Values of Different Variables.

| Sr.No. | Parameters | Minimum Value | Maximum Value |
|---|---|---|---|
| 1 | Load | 4000 N/m | 5000 N/m |
| 2 | Speed | 7.5 m/sec | 18.75 m/sec |
| 3 | Frequency | 100 Hz | 500 Hz |
| 4 | Amplitude | 0.1 | 0.2 |

### 2.2 Overall solution procedure

The steps involved in the overall solution scheme are given below:

1. The pressure distribution [P], minimum film thickness $H_{min}$ and outlet boundary co-ordinate $X_o$ were initialized to some reference values. Take $X_{in} = 0$ and $X_o = 1$.

2. Evaluated the fluid film thickness, H, at every node by using film thickness equation.

3. The residual vector [f] was calculated at each node.

4. The residual vector ΔW was calculated from the discretized load equilibrium equation.

5. The residual vectors calculated in the steps 3 and 4 were assembled in a single vector [F] to facilitate execution of Newton-Raphson scheme.

6. This was followed by computation of Jacobian coefficients.

7. The corrections to the system variables were computed by inverting the Jacobian matrix using Gauss elimination.

8. The corrections, calculated in step 7, were added to the corresponding system variables to get the new values of the pressure distribution [P] and minimum film thickness Ho.

9. The outlet boundary co-ordinate Xo was corrected by using an appropriate scheme.

10. The termination of the iterative loops required the fulfillment of the predefined convergence criteria to arrive at an accurate solution. In order to check the convergence of the pressure distribution, the sum of the nodal pressures corresponding to the current iteration (say $n^{th}$) was calculated. If the fractional difference between this value and that corresponding to the previous iteration was less than the prescribed tolerance TOL, the pressure distribution was assumed to have converged. Thus,

$$\frac{\left|\left[\sum_{i=1}^{N} P_i\right]_n - \left[\sum_{i=1}^{N} P_i\right]_{n-1}\right|}{\left|\left[\sum_{i=1}^{N} P_i\right]_{n-1}\right|} \le TOL$$

11. The minimum film thickness was assumed to converge if the fractional change in its value became less than the prescribed tolerance in successive iterations

$$\frac{\left|\left[H_o\right]_n - \left[H_o\right]_{n-1}\right|}{\left|\left[H_o\right]_{n-1}\right|} \le TOL$$

    The value of TOL adopted in the analysis was $1 \times 10^{-4}$ as it has been found that a lower value does not contribute to improve the accuracy of the solution. The iterative loop terminates and the current values were considered as the final solution only if all the relevant convergence criteria were satisfied simultaneously.

12. If any one or more of the relevant criteria were not satisfied, the next iteration began and the control was shifted back to the step 2.

Finally the values of friction coefficient (μ) and minimum film thickness ($H_{min}$) were calculated using suitable formulae.

## 3. RESULT AND DISCUSSION

### 3.1 Effect of Load

The table 3.1.1 and 3.1.2 shows the percentage change in Hmin for percentage increase in load at constant frequency and speed respectively. And the same is represented in the figures 3.1.1 and 3.1.2 respectively.

**Table 3.1.1** Percentage Change in $H_{min}$ for varying Load and constant Frequency

| Sr. No. | Value of Frequency (Hz) | Percentage Increase in Load | Decrease of $H_{min}$ at Maximum Loading | Decrease of $H_{min}$ at Minimum Loading |
|---|---|---|---|---|
| 1 | 100 | 25% | 11.95% | 9.56% |
| 2 | 500 | 25% | 11.29% | 10.23% |

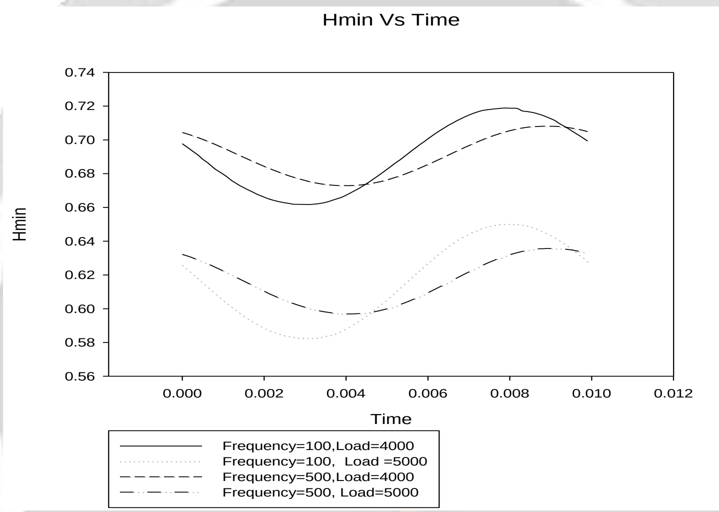


**Figure 3.1.1** Minimum Film Thickness Vs Time (Frequency and Load)

**Table 3.1.2** Percentage change in $H_{min}$ for varying load and constant Amplitude

| Sr. No. | Value of Amplitude(Hz) | Percentage Increase in Load | Decrease of $H_{min}$ at Maximum Loading | Decrease of $H_{min}$ at Minimum Loading |
|---|---|---|---|---|
| 1 | 0.1 | 25 | 11.97% | 9.56% |
| 2 | 0.2 | 25 | 13.15% | 6.30% |

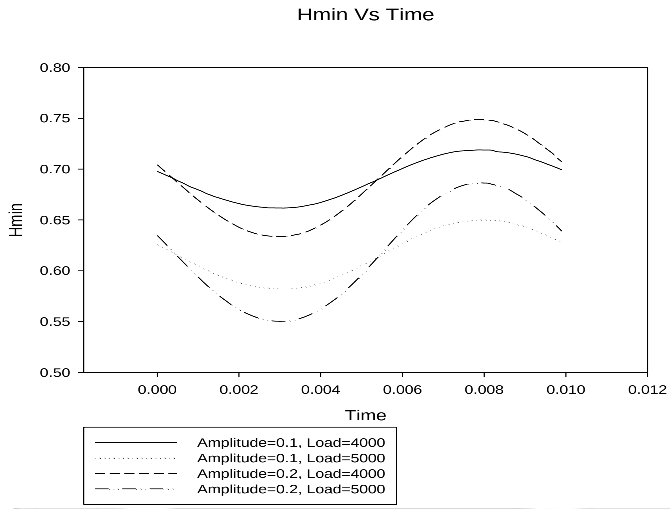


**Figure 3.1.2** Minimum Film Thickness Vs Time (Amplitude and Load)

## 3.2 Effect of Frequency

The table 3.2.1 and 3.2.2 shows the percent change in Hmin for percentage increase in frequency at constant load and speed respectively. And the same is represented in the figures 3.2.1 and 3.2.2 respectively.

**Table 3.2.1** Percentage change in $H_{min}$ for varying frequency and constant Load

| Sr. No. | Value of Load (N/m) | Percentage Increase in Frequency | Decrease of $H_{min}$ at Maximum Loading | Decrease of $H_{min}$ at Minimum Loading |
|---|---|---|---|---|
| 1 | 4000 | 400% | 1.96% | 1.95% |
| 2 | 5000 | 400% | 2.56% | 3.01% |

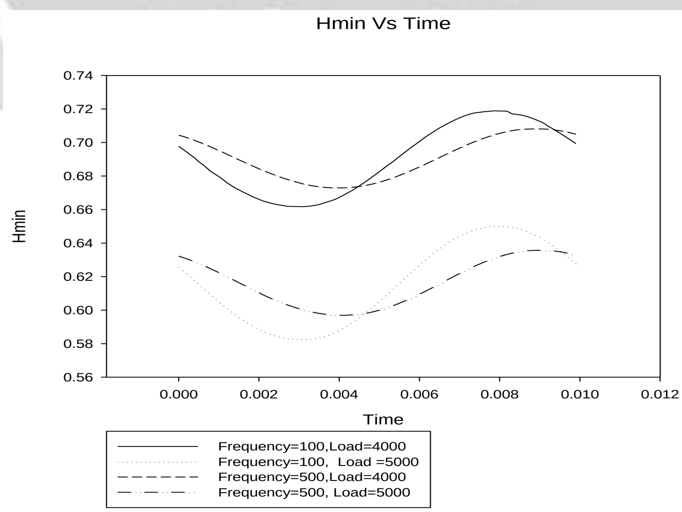


**Figure 3.2.1** Minimum Film Thickness Vs Time (Frequency and Load)

**Table 3.2.2** Percentage change in $H_{min}$ for varying frequency and constant Speed

| Sr. No. | Value of Speed(m/sec) | Percentage Increase in Frequency | Decrease of $H_{min}$ at Maximum Loading | Decrease of $H_{min}$ at Minimum Loading |
|---|---|---|---|---|
| 1 | 7.5 | 400% | 6.75% | 5.74% |
| 2 | 18.75 | 400% | 1.81% | 2.09% |

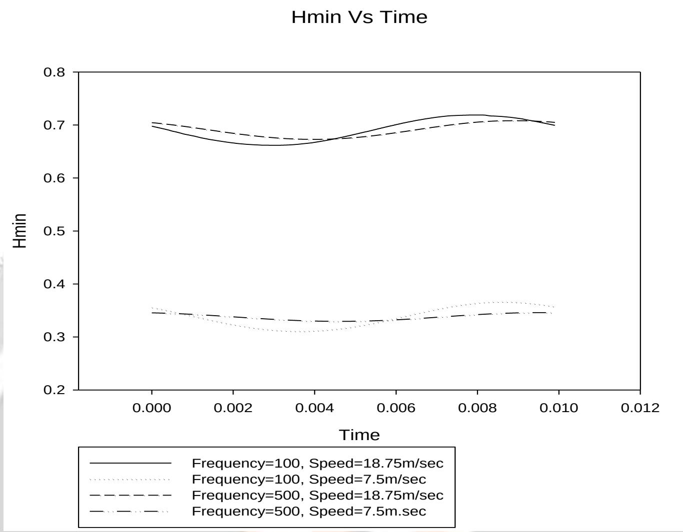


**Figure 3.2.2** Minimum Film Thickness Vs Time (Frequency and Speed)

## 4. CONCLUSION AND SCOPE FOR FUTURE WORK

### 4.1 Concluding remarks

On the basis of the results presented in last section, the following conclusion is drawn:

1 The value of minimum film thickness ($H_{min}$) is minimum at maximum loading conditions and is maximum at minimum loading conditions. An increase in the value of load results in a decrease in the value of minimum film thickness.
2 The value of $H_{min}$ for the first half cycle is higher for greater value of frequency and smaller for lesser value of frequency for constant load.
3 The value of $H_{min}$ for the first half cycle is higher for greater value of frequency and smaller for lesser value of frequency for constant speed.

### 4.2 Scope for future work

In future, the following points can be investigated for gas bearings:

1. The solution of Reynolds Equation is limited to isothermal conditions (i.e. uniform temperature over the entire surface of the bearing). Thermal effects can be investigated for future work.
2. The same analysis can be carried out for high bearing numbers too.

## REFERENCES

1. Chandra M., Malik M. and Sinhasan R. , "Design Data for Centrally Loaded Partial arc Journal Bearings - Part II," Department of Mechanical and Industrial Engineering, University of Roorkee (India), January 7, 1983.
2. Gross W. A., "Gas Bearing- A Survey," Research Laboratory, Ampex Corporation, Redwood City, Calif. (U.S.A.), 1963, pp. 423-443.
3. Heshmat H., Walowit J.A., Pinkus O., "Analysis of Gas Lubricated Foil Journal Bearings," ASME, ASLE Lubrication Conference, Washington, Oct. 1982.
4. Florin Dimofte, "Fast Methods to Numerically Integrate the Reynolds Equation for Gas Fluid Films," STLE-ASME Joint Tribology Conference St. Louis, Missouri, October 13-16, 1991.
5. Faria Marco Tulio C. , Andres Luis San, "On the Numerical Modeling of High-Speed Hydrodynamic Gas Bearings," Trans. ASME, 2000, Vol. 122, pp. 124-130.
6. Heshmat Crystal A., David S. Xu, Heshmat Hooshang, "Analysis of Gas Lubricated Foil Thrust Bearings Using Coupled Finite Element and Finite Difference Methods," Trans. ASME, 2000, Vol. 122, pp. 199-204.
7. Peng J.P., Carpino M., "Calculation of Stiffness and Damping Coefficients for Elastically Supported Gas Foil Bearings," Trans. ASME, 1993, Vol. 115, pp. 20-27.
8. Roger Ku C.P., Heshmat H.,"Compliant Foil Bearing Structural Stiffness analysis-Part-2:Experimental Investigation," Trans. ASME, 1993, Vol. 115, pp. 364-369.
9. Giri L.Agrawal, "Foil Air/Gas Bearing Technology ~ An Overview," International Gas Turbine & Aeroengine Congress & Exhibition Orlando, Florida, June 2-June 5,1997.
10. Stachowiak Gwidon W., Batchelor Andrew W., "Engineering Tribology" Second Edition, Butterworth Heinemann, 2001.
11. Piekos E. S., Breuer K. S., "Manufacturing Effects in Micro fabricated Gas Bearings: Axially Varying Clearance," Trans. ASME, 2002, Vol. 124, pp. 815-821.
12. Peng Zhengchun, "Thermodynamics Analysis of Compressible Gas Flow in Compliant Bearing," A Thesis of Master of Science in Mechanical Engineering, Beijing Institute of Technology, China, 2003.