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CONSIDERATION OF CAVITATION EFFECT IN FINITE JOURNAL BEARING USING NUMERICAL TECHNIQUE

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ABSTRACT

Journal bearings are widely applied in different rotating machineries. These bearings allow for transmission of large loads at mean speed of rotation. In machinery the parameters characterizing operation such as, power losses, vibration amplitude and frequency are dependent on the type of bearings used, specific loads, bearing clearance, and load orientation. Hydrodynamic Journal bearing based on hydrodynamic lubrication, hydrodynamic lubrication means that the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent metal-to-metal contact, and that the stability thus obtained can be explained by the laws of fluid mechanics. In journal bearing that operate with stationary load under steady state condition, cavitation takes place at the sub-atmospheric pressure commonly encountered in divergent section of the oil film. Lubricating oil contains roughly 10% by volume of dissolved gas when saturated with air. If oil pressure falls below the usual atmospheric saturation pressure, this dissolved air tends to come out of solution as cavity bubbles. Reynolds equation derived from Navier Stock's equation, it is highly nonlinear partial differential equation and very complex to solve analytically. Hence the Reynolds equation solves using numerical technique with help of computer program. Finite difference method is suitable for handle the differential equation and reduced differential equation is solved using a successive over relaxation (SOR) technique. The main aim is to find out hydrodynamic journal-bearing performance characteristics, such as pressure distribution, attitude angle and maximum pressure, using the Swift-Stieber Boundary Condition. Also this boundary condition helps to encounter the cavitation effect and its location.

Keywords: journal bearing, reynolds equation, cavitation, successive over relaxation.

INTRODUCTION

Hydrodynamic Journal bearing based on hydrodynamic lubrication, hydrodynamic lubrication means that the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent metal-to-metal contact as shown in Figure-1, and that the stability thus obtained can be explained by the laws of fluid mechanics.

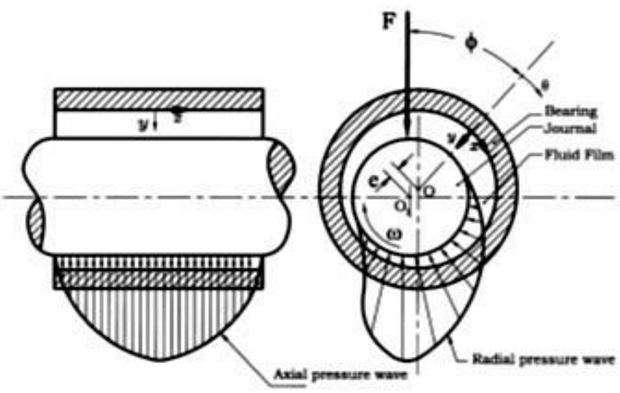

Figure-1. Hydrodynamic journal bearing [8]

Reynolds derived the governing equations for lubricating films in simplifying the Navier-Stokes equations [1] considering thin-film effect [2]. Hence the Reynolds equation solves using numerical technique [3] with help of computer program.

Swift-Stieber Boundary Condition is most practical boundary condition [4] to find the static characteristics of bearing [5]. Lubricating oil contains roughly 10% by volume of dissolved gas when saturated with air. If oil pressure falls below the usual atmospheric saturation pressure, this dissolved air tends to come out of solution as cavity bubbles. Swift-Stieber Boundary Condition explain the phenomenon of liquid cavitation in steadily loaded fluid film bearings and notes the most adequate boundary conditions at the inception as shown in Figure-2 and reformation boundaries of the cavitation zone [6].

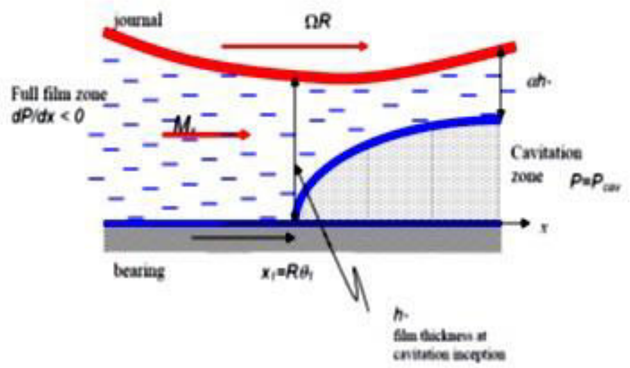

Figure-2. Cavitation inception in a thin film. [6]

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MATHEMATICAL MODELING OF JOURNAL BEARING

Generalized Reynolds equation for hydrodynamic lubrication is given by:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right)=6\left(U\frac{dh}{dx}\right) \quad (1)$$

Where z = bearing length, R= journal radius, θ = motion angle, U = journal speed, μ = viscosity

For the sake of convenience, the above equation is first non-dimensionalized by the introduction of the following non-dimensional quantities

$$x = R\theta \qquad z = \bar{z}L$$

$$h = C\bar{H} \qquad \lambda = \frac{L}{D}$$

$$P = \frac{\mu N_s}{1}\left(\frac{R^2}{C^2}\right)\bar{P}$$

Then, Reynolds equation, Equation (1) can be non-dimensionalized in the following form

$$3\bar{H}^2\frac{\partial \bar{H}}{\partial\theta}\frac{\partial\bar{P}}{\partial\theta}+\bar{H}^3\frac{\partial^2\bar{P}}{\partial\theta^2}+\frac{1}{4\lambda^2}\frac{\partial^2 P}{\partial z^2}=12\pi\frac{\partial\bar{H}}{\partial\theta} \quad (2)$$

Apply finite difference method to reduce above equation in to algebraic equations.

Finite difference method

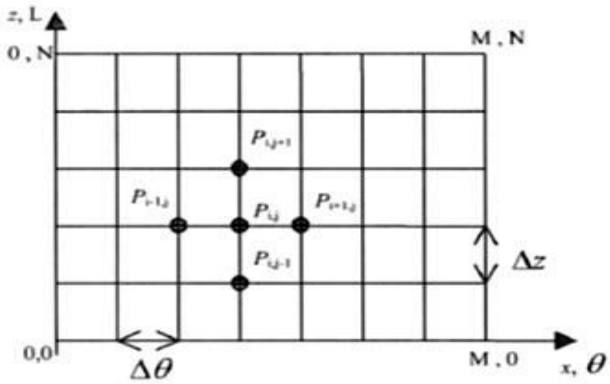

Figure-3. Finite difference grid for the oil film mesh.

From Figure-3, the appropriate finite difference operators for the partial derivatives are:

$$\frac{\partial\hat{P}}{\partial\theta}=\frac{\hat{P}_{i+1,j}-\hat{P}_{i-1,j}}{2\Delta\theta}$$

$$\frac{\partial\hat{P}}{\partial\hat{Z}}=\frac{\hat{P}_{i,j+1}-\hat{P}_{i,j-1}}{2\Delta\hat{Z}}$$

$$\frac{\partial^2\hat{P}}{\partial\theta^2}=\frac{\hat{P}_{i+1,j}-2\hat{P}_{i,j}+\hat{P}_{i-1,j}}{(\Delta\theta)^2}$$

$$\frac{\partial^2\hat{P}}{\partial\hat{Z}^2}=\frac{\hat{P}_{i,j+1}-2\hat{P}_{i,j}+\hat{P}_{i,j-1}}{(\Delta\hat{Z})^2}$$

By writing difference equations of the form of Equations 4 at each point within the lubricant film where Reynolds equation applies, obtains a set of algebraic equations for the discrete unknown pressure. We obtain P (i, j) in the following form

$$P_{i,j}=AP_{i+1,j}+BP_{i-1,j}+CP_{i,j+1}+DP_{i,j-1}-E \quad (4)$$

Where A, B, C, D & E are constant.

If the number of nodal points at which the pressure is to be calculated is M in circumferential direction and N in the length direction, then we have M x N equations of the form of Equation (4). The boundary conditions are given as the pressure at the points on the boundary. By solving these simultaneously, the pressures at respective nodal points are obtained. For locations at the edges of the lubricant film, one simply sets the pressure equal to the known boundary pressure. This set of equations can be solved by several methods for the discretized pressures defined at each i, j grid point, some of which are described in linear algebra and numerical analysis texts.

One of the methods to solve these simultaneous equations is Successive over Relaxation (SOR). Pressure at all the nodal points can thereby be found in a finite number of operations. Further, if we sweep the lubricating domain two dimensionally with the calculation of equation 5.4 at each nodal point in consecutive order starting from a suitable nodal point, and if this is repeated a sufficient number of times, then it is expected that the pressure obtained at each nodal point gradually approaches the true value of the pressure. This is called the iterative method (successive approximation method).

These simultaneous equations solved with the help of computer program. MATLAB program developed to solve the M x N simultaneous equation using successive over relaxation method. In this case, replace the previous iteration pressure value into the latest iteration pressure value and the calculation will be repeated until the following relation is satisfied:

$$\sum_{i,j}^{M,N}\frac{\left|P_{i,j}^{\ k}-P_{i,j}^{\ k-1}\right|}{P_{i,j}^{\ k}}<\in$$

Where, $P_{i,j}^{\ k}$ is pressures obtained in the kth calculation, $P_{i,j}^{\ k-1}$ are those in the previous calculation, and $\in$ is a sufficiently small allowable error. The pressure $P_{i,j}^{\ k}$ obtained in the kth calculation will be a solution of pressure distribution $P_{i,j}$. For a high accuracy of calculation, a put the value of $\in$ is 10^{-6} to 10^{-12}.

Successive over relaxation

This method reduces all equations in the system to equations in only one unknown only which can be solved immediately. All other unknowns in each equation are considered to be known and equal to some previously computed value. Therefore, an initial distribution is guessed and then successively improved by solving the system of equations in one unknown. Make repetition in the solution procedure number of time unless the error between current pressure value and previous pressure is negligible.

The SOR method is similar to the Jacobi and Gauss-Seidel methods, but it uses a scaling factor, ω, to more rapidly reduce the approximation error hence minimum number of iteration required.

$$tan\phi = \frac{W_y}{W_x}$$

When 1 < ω, the procedures are called over-relaxation methods, which are used to accelerate the convergence for systems that are convergent by the Gauss-Seidel technique. These methods are abbreviated SOR for Successive Over-Relaxation and are used for solving the linear systems that occur in the numerical solution of certain partial-differential equations.

Reynolds boundary condition

In the Reynolds boundary condition the oil film is assumed to terminate at a certain position $(\theta_{cav} = \pi + \delta)$ at which both the pressure and pressure gradient are zero, simultaneously.

$$\bar{P} = 0 \quad \& \quad \frac{\partial \bar{P}}{\partial \theta} = 0 \qquad at \qquad \theta = \theta_{cav}$$

Load capacity

The load capacity is solved by integration of the pressure wave around the bearing. In the case of a finite bearing, the pressure is a function of z and θ. The following are the two equations for the integration for the load capacity components in the directions of W_x and W_y of the bearing center line and the normal to it:

$$W_x = -\int_0^{2\pi}\int_0^{L} P cos\theta R d\theta dz$$

$$W_y = \int_0^{2\pi}\int_0^{L} P sin\theta R d\theta dz$$

The attitude angle, ∅, is determined from the two load components:

$$tan\phi = \frac{W_y}{W_x}$$

RESULTS AND DISCUSSIONS

Reynolds equation has been solved using finite difference method for the pressure distribution considering cavitation effect. Pressure distribution of various eccentricity ratios that is, ε = 0.2, 0.4, 0.6 and 0.8 over the 0 - 360^0 angle with cavitation phenomenon is shown in Figure-4.

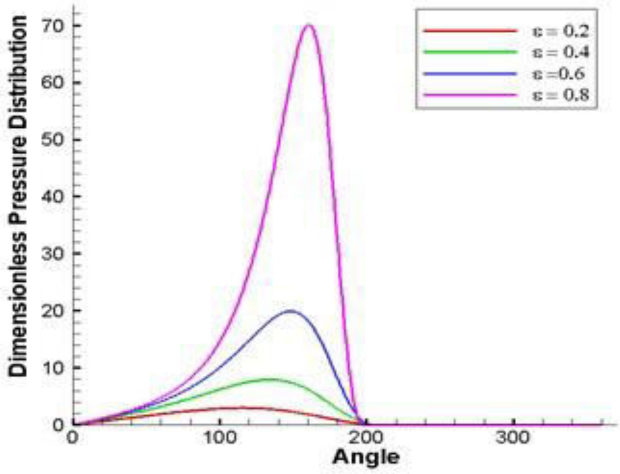

Figure-4. Pressure distribution of various eccentricity ratios.

In the Figure-5 maximum dimensionless pressure obtained through the numerical method compared with Khonsari's *et.al.* results. As the eccentricity ratio increase maximum pressure increase and the comparison between the maximum pressures values of FDM and Khonsari *et. al.* showed that the results was satisfactory.

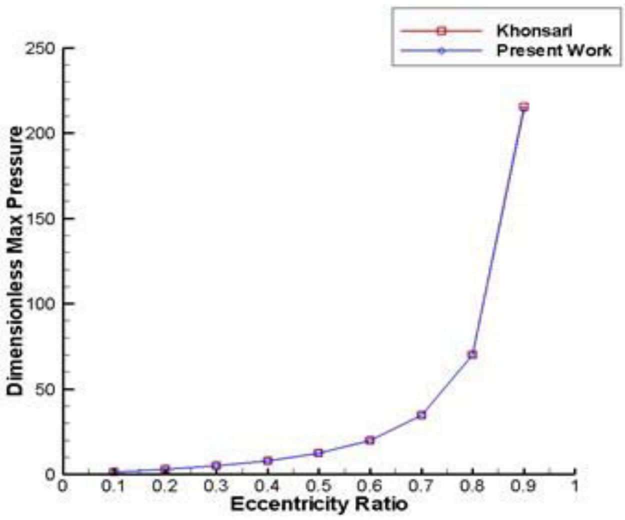

Figure-5. Maximum pressure with eccentricity ratios.

As the eccentricity ratio increases cavitation angle decreases that indicates as the gap between the journal and bearing decreases cavitation angle reduces and it starts earlier. Comparison of cavitation angle obtained from finite difference method with the Khonsari *et. al.*[7] shown in the Figure-6.

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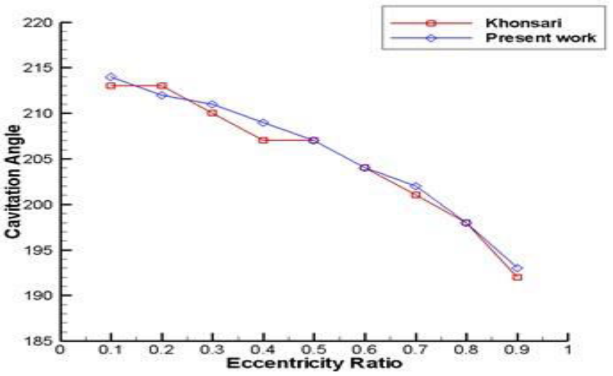

Figure-6. Cavitation angle with eccentricity ratios.

Figure-7 shows the polar plot of eccentricity ratio verses attitude angel. Attitude angle decreases gradually with eccentricity ratio increases. Comparison of attitude angle calculated using finite difference method made with Khonsari's *et.al.* and results showed very good agreement.

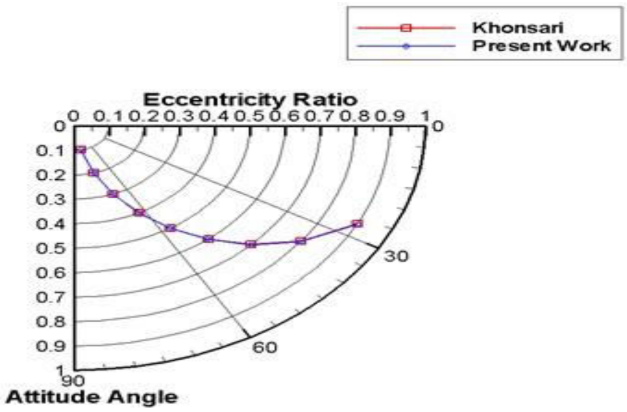

Figure-7. Attitude angle with eccentricity ratio.

There is variation in the cavitation angle accordingly change in the L/D ratios. At lower eccentricity ratios much variation in cavitation angle, at higher eccentricity ratios there is the less variation in cavitation angle shown in the Figure-8.

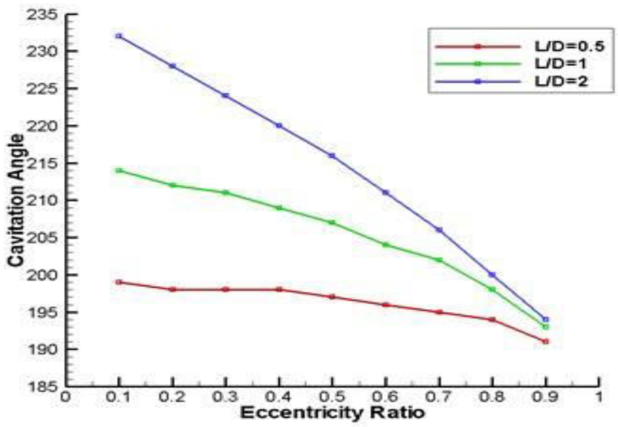

Figure-8. Variation in cavitation angle with eccentricity ratio.

Figure-9 shows the variation of dimensionless maximum pressure with eccentric ratio for different values of L/D ratio. As said earlier at for long length bearing there is the high maximum pressure value as compared to short length bearing and finite bearing maximum pressure value in between them shown in Figure 9. As pressure increases the load capacity is also increases, that mean long bearing having maximum load capacity as compared with short and finite length bearing.

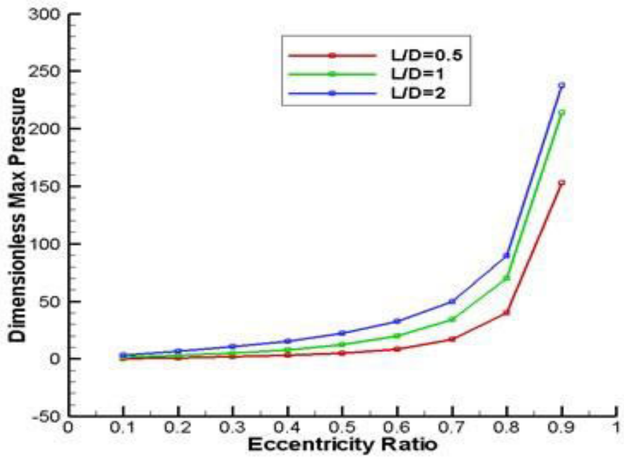

Figure-9. Variation of maximum pressure at L/D =0.5, 1 & 2 with eccentricity ratio.

Figure-10 shows the comparison of dimensionless load capacity of three bearings with various eccentricity ratios for different L/D ratios.

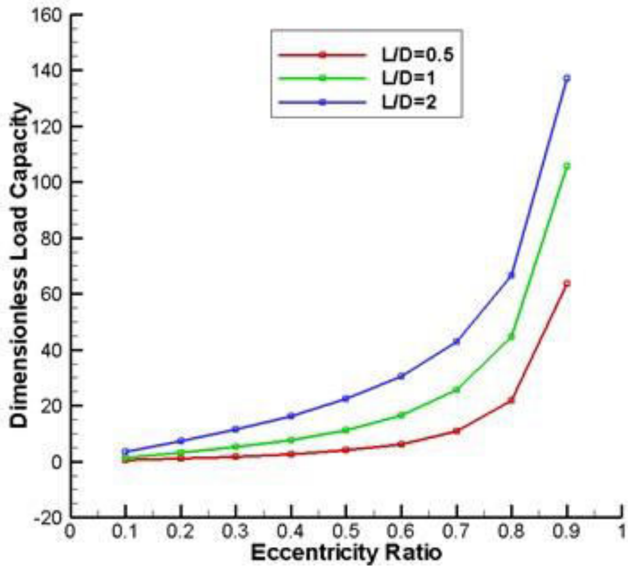

Figure-10. Variation of load capacity with eccentricity ratio.

Table-1 contain the dimensionless values of bearing parameters obtained through finite difference method for finite bearing with Reynold's boundary condition.

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Table-1. Parameters of finite bearing with Reynold's boundary condition.

L/D ratio	Eccentricity ratio	Attitude angle	Cavitation angle	Load capacity	Friction force	Coefficient of friction (R/C)f	Max. pressure	Inlet oil flow	Side leakage
1/2	0.1	81.7526	199	0.46176	39.723	86.0257	0.44167	1.07791	0.18496
	0.2	75.158	198	0.98352	40.4827	41.161	0.9767	1.16825	0.36976
	0.3	68.5342	198	1.63799	41.8419	25.5447	1.72291	1.25806	0.55454
	0.4	61.8285	198	2.54382	43.9714	17.2856	2.87689	1.34738	0.73949
	0.5	55.0394	197	3.92272	47.1931	12.0307	4.83578	1.43623	0.92476
	0.6	48.1453	196	6.25067	52.1416	8.34175	8.53807	1.52463	1.11047
	0.7	40.9159	195	10.7748	60.2204	5.58903	16.6859	1.61258	1.29712
	0.8	33.0278	194	21.7343	75.2745	3.46339	39.7721	1.70005	1.48531
	0.9	23.7211	191	63.5323	113.571	1.78761	152.694	1.78703	1.67608
1	0.1	79.485	214	1.50103	39.8249	26.5317	1.39011	1.05991	0.15743
	0.2	73.9625	212	3.16011	40.8999	12.9426	2.99258	1.12852	0.31328
	0.3	68.3673	211	5.13485	42.8165	8.33842	5.05186	1.19283	0.46777
	0.4	62.6787	209	7.6653	45.7984	5.97478	7.95023	1.25286	0.62096
	0.5	56.8212	207	11.1698	50.2599	4.49962	12.393	1.30861	0.77295
	0.6	50.6813	204	16.4827	56.9993	3.45814	19.9101	1.36006	0.92386
	0.7	43.9962	202	25.606	67.7306	2.64511	34.4964	1.4072	1.07396
	0.8	36.3394	198	44.7053	86.992	1.9459	70.0707	1.45	1.22346
	0.9	26.5553	193	105.784	133.13	1.25851	214.056	1.48847	1.37267
2	0.1	75.3886	232	3.5681	40.0226	11.2168	3.13811	1.02543	0.10602
	0.2	71.2386	228	7.34414	41.6833	5.67573	6.54872	1.05063	0.20784
	0.3	67.0545	224	11.5016	44.5621	3.87442	10.51	1.06331	0.30562
	0.4	62.7237	220	16.3164	48.8751	2.99545	15.455	1.06442	0.39952
	0.5	58.0532	216	22.288	55.0415	2.46956	22.1714	1.0551	0.48991
	0.6	52.8959	211	30.3851	63.8873	2.10258	32.2916	1.0367	0.57701
	0.7	46.9371	206	42.8872	77.2138	1.80039	49.9675	1.01064	0.66115
	0.8	39.5938	200	66.6647	99.7875	1.49686	89.2538	0.97836	0.74274
	0.9	29.2405	194	137.007	150.802	1.10069	237.331	0.94123	0.82239

CONCLUSIONS

Cavitation effect considered in long and finite bearing obtained the parameters compared with literature shown good agreement between them and more than 90% accuracy was achieved.

- The static behavior of a cylindrical journal bearing has been studied. The fluid film pressure distribution was obtained by solving the governing Reynolds equation via finite difference method with successive over relaxation Method.
- The optimum over-relaxation factor was found using trial and error method, which is exact and thus shortens the compilation time of running the program. The optimum over-relaxation factor or scaling factor, ω, and value is 1.85.

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