Dynamically Loaded Analysis of The Adjustable Hydrodynamic Pads Bearing

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Received on: 6/6/2007
Accepted on: 3/2/2008

Abstract

This paper presents the instantaneous journal center velocity under dynamic load for adjustable hydrodynamic four pads bearing. The velocity is calculated by mobility method in different values of length - diameter ratios (0.25, 0.5, 0.75 and 0.95). Finite difference method was used to solve the Reynolds equation in two dimensions. This equation includes the total oil film thickness (oil film thickness and elastic deformation pad material due to generated pressure).The effects of many parameters were studied such as dynamic coefficients (stiffness and damping coefficients) and power loss on the bearing performance. The results proved that the journal center velocity and mobility components is affected by the sign of the eccentricity ratio components, also the maximum value of the journal center velocity is determined by the values of the length - diameter ratios. The vertical stiffness coefficient (K_{yy}) and damping coefficient (C_{yy}) increase as the eccentricity ratio increased. The power loss were found increase when length diameter ratio increase.

Keywords: Eccentricity Velocity, Mobility Method, Dynamic Bearing Coefficients, Adjustable Pads Bearing

التحليل الديناميكي لمسند تحميل هيدروديناميك ذات وسائد قابلة للتعيير

الخلاصـــة

يعرض البحث السرعة اللحظية لمركز محور يعمل تحت حمل ديناميكي لمسند هيدروديناميكي ذو أربعة وسائد قابلة للتعيير وذلك باستخدام طريقة قابلية الحركة و بأخذ قيم مختلفة لنسـبة طـول المسند - لقطره وهي (0.95,0.75,0.5,0.25) .تم استخدام طريقة الفروقات المحـددة لحـل معادلـة رينولد للبعدين تحتوي هذه المعادلة على السمك الكلي لطبقة الزيت حيث تتضمن سمك طبقـة الزيـت يضاف إليها التشوه المرن الحاصل في معدن الوسادة نتيجة الضغط المتولد. تمت دراسة تأثير كل من معاملات المرونة و التخميد وكذلك مقدار الطاقة الضائعة على أداء المسند. أثبتت النتـائج أن سـرعة مركز المحور و مركبات قابلية الحركة تتأثر بإشارة (اتجاه) مركبات نسـبة الانحـراف اللامركـزي وكذلك أقصى قيمة لسرعة مركز المحور تتحدد بقيمة نسبة طول المسند لقطره. أن قيم كل من معامـل المرونة العمودي (Kyy) و معامل التخميد العمودي (Cyy) تزداد مـع زيـادة نسـبة الانحـراف اللامركزي . إما مقدار ضياع الطاقة تتناسب طرديا مع نسبه طول المسند لقطره.

https://doi.org/10.30684/ etj.26.8.11
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Nomenclature

B_c	Bearing center	–
C_{yx}, C_{yy} C_{xx}, C_{xy}	Elements of bearing damping	N.sec./ m
Dc	Deformation Coefficient	–
d_r	Radial displacement deformation	mm
E	Modulus of elasticity	N / m^2
e	Eccentricity; e_x, e_y eccentricity components	mm
f	Oil film load; $\mathbf{f}_{\square}$, $\mathbf{f}_{\varepsilon}$ film load components	*N*
h	Oil film thickness	mm
h_t	Total oil film thickness	mm
J_c	Journal center	–
J_{cs}	Steady-state equilibrium position of journal center	–
K_{xx}, K_{xy} K_{yx}, K_{yy}	Elements of Bearing stiffness	N / m
L	Axial length of bearing	mm
M	Mobility; M_x, M_y mobility components	–
N	Journal speed	r.p.m
P	Hydrodynamic pressure in the bearing	N / m^2
R	Radius of bearing	mm
r	Radius of journal	mm
t	Time	sec.
u	Surface velocity of sliding surface	m /sec.
v	Radial velocity of journal surface	m /sec.
W	External load	N

γ_o	Coefficient depending on Poisson's ratio of the bearing material	–
ε	Eccentricity ratio	–
ε_x, ε_y	Eccentricity ratio components	–
θ	Angular coordinate taken from centers lines	degree
μ	Viscosity of oil lubricant	N.sec/m^2
□	Attitude angle	degree
ω	Angular journal speed	rad/sec.
ω_{av}	Average angular speed of journal and bearing	rad/sec.

Introduction

The conventional journal bearing surface which is a continuous surface and it could be represented by a circle whereas the adjustable hydrodynamic pads bearing has multi- pads arranged circumferentially around the journal,[1]. The analysis of dynamically loaded of noncircular bearings is necessary for proper design of bearings in many industrial applications, mobility method is an extremely clever method for solution of dynamically loaded journal bearing problem.

Parkins (1998) [2], invented an adjusting member for pads bearing as a threaded screw type in contact with the adjustment member. The pads bearing may be adjusted in a radial direction to improve the bearing performance during the operation or maintenance. Krodkiewski, Cen, and Sun (2002) [3], were presented the modeling and analysis of a rotor - bearing system with a new type of active oil bearing, the active bearing is supplied with an adjustable sleeve .The author clears up the deformation that can be reformed during operation of the rotor by mechanism of the adjusting member. Dimofte and Hendricks (2002) [4], studied the dynamic behavior of the wave journal bearing (noncircular bearings) with an eccentrically mounted shaft. The following conclusions were drawn; a stable running can be obtained for a dynamically loaded three-wave journal bearing and the subsynchronous whirl motion influences the bearing housing center orbits, if the bearing speed is in the region where the bearing itself is susceptible to subsynchronous whirl instability. Biao, and Sawicki (2000), [5] compared the mobility method with the rigorous method in a study of dynamically loaded journal bearings. This comparison led to the following conclusions; mobility method is a very efficient tool to obtain a quick solution in a process of dynamically loaded journal bearing design and for the constant feed pressure bearings.

In this paper the topic has been divided into two parts, the first is confined the evaluation of the total of the oil film thickness, while the second using the mobility method to predict the eccentricity location under dynamic load. The finite difference method was used to solve the Reynolds equation in two dimensions and finally study the adjustable hydrodynamic pads bearing performance after using tilting angle such as dynamic stiffness coefficients, dynamic damping coefficients and the effects of the eccentricity ratio on the power loss.

The Adjustable Hydrodynamic Pads Bearing Characteristics

The below data lists the dimensions of this bearing, standard materials of construction of pad body are high strength steel with high tin content Babbitt face (white metal). The white metal metallurgical bonded to the body and the thickness of white metal is 5 mm,[5].

Adjustable Pads Bearing	Value
Diameter of bearing, D	50.150 mm
Diameter of journal, d	50 mm
Radial clearance, Cr	0.075 mm
Pad thickness, tp	12.7 mm
Pad central angle, PCA	80°
Pad tilt angle, δ	0.1°
Number of pads, Npads	4
Angular dimension of oil groove(10°) for each pad	

The oil film thickness equation of adjustable hydrodynamic pads bearing, [1].

$$h= [R^2-(L1 \sin(\theta+\zeta))^2]^{1/2} - L1 \cos(\theta+\zeta) - r \quad .....(1)$$

The radial displacement (d_r) at any point (i,j) on the fluid-film –pad-liner interface can be expressed in terms of the pressure (P) at any point by the relation, [6].

$$\bar{d}_r(i,j) = \gamma_o D_c \bar{p}(i,j) \quad .....(2)$$

$$D_c= \frac{\mu_o \omega_j}{E} \frac{t_p}{R} \left(\frac{R}{C_r}\right)^3 \quad ..... (3)$$

The radial displacement of the deformation that is demonstrated in equation (2) can be added to equation (1) (if the deformation becomes existed) to get the total film thickness which can be expressed as follows:

$$h_t= h + C_r \bar{d}_r \quad ..... (4)$$

Reynolds Equation (2D) for Dynamically Loaded Bearings

The Reynolds equation is the governing equation for the distribution of pressure in dynamically loaded bearings, [7].

$$\frac{1}{6\mu}\left[\frac{\partial}{\partial x}\left(h^3\frac{\partial P}{\partial x}\right)\right]+\frac{1}{6\mu}\left[\frac{\partial}{\partial z}\left(h^3\frac{\partial P}{\partial z}\right)\right]=u\frac{\partial h}{\partial x}+2v \quad .. (5)$$

The right-hand side of the equation (5) can be stated by the boundary conditions of the velocities. Under dynamic conditions, the journal center does not remain constant but it is in motion in radial and tangential directions.

$$u = \omega r + \frac{de}{dt}\sin\theta - e\frac{d\varphi}{dt}\cos\theta \quad \text{..... (6)}$$

$$v = \omega r\frac{\partial h}{\partial x} + \frac{de}{dt}\cos\theta + e\frac{d\varphi}{dt} \quad \text{..... (7)}$$

Where:

$\frac{de}{dt}$ = the radial component of the instantaneous journal center velocity

$\frac{d\varphi}{dt}$ = the tangential component of the instantaneous journal center velocity

The motion of journal center in radial direction is more than that the motion in tangential direction. Therefore, most solutions of the Reynolds equation were neglected the tangential component of the velocity of the journal center, [8].

$$\left[\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial z}\right)\right] = 6\left[\omega r + \frac{de}{dt}\sin\theta\right]\frac{\partial h}{\partial x} + 12\left[\omega r + \frac{de}{dt}\cos\theta\right]\frac{\partial h}{\partial x} \quad \text{..... (8)}$$

The Journal Center Velocity Determination by Mobility Method

During the operation the center of journal move at an arbitrary position inside the eccentricity circle from the initial position and let time pass until the journal center position is steady. Mobility method is used to study the effect of this motion on journal bearing behavior by mobility map that is the loci of familiar attitude-eccentricity curves, [5].

$$\frac{de}{dt} = \frac{W \, . \, M \, . \left(\frac{C_r}{R}\right)^3}{2 \, . \, \mu \, . \, L} + \omega_{av} \, . \, e \quad \text{... (9)}$$

If the mobility vector (M), initial position vector (e) and the load (W) are known, it is easy to obtain (e) in current time step from equation (9). Also, the resultant of the mobility vector and eccentricity vector can be set as follow:

$$M = \sqrt{M_x^2 + M_y^2}$$

$$e = \sqrt{e_x^2 + e_y^2} \quad \text{..... (10)}$$

Hence, equation (9) can be written in matrix form as follows:

$$\frac{d}{dt}\begin{bmatrix} e_x \\ e_y \end{bmatrix} = \frac{W \, C_r^3}{2\,\mu\mu L^3}\begin{bmatrix} M_x \\ M_y \end{bmatrix} + \omega_{av}\begin{bmatrix} 0 & -1 \\ 1 & 0 \end{bmatrix}\begin{bmatrix} e_x \\ e_y \end{bmatrix} \quad \text{...... (11)}$$

Mobility components were determined numerically to gives [5]:

$$M_x = \frac{(1-\varepsilon_x)^{2.5}}{\pi^2 (L/D)^2}\left(1+0.5\frac{(L/D)^2}{1-\varepsilon_x}\right)\left(\sqrt{3\pi} - 0.24(L/D)^2 e^{-(\varepsilon_x+\varepsilon_y)^2} + \frac{\varepsilon_y^2}{1-\varepsilon_x}\left(1-0.4\sqrt{1-\varepsilon_x}\right) + \varepsilon_x^2 (L/D)\left(\frac{4}{3}-\frac{L}{D}\right)\right) \quad \text{.....(12)}$$

$$M_y = -\frac{\varepsilon_x}{1-\varepsilon_x}\frac{(1-\varepsilon_x)^{2.5}}{\pi (L/D)^2}\left(1+0.5\frac{(L/D)^2}{1-\varepsilon_x}\right)\left(\frac{5}{4}+\frac{\varepsilon_x L}{7D}+\frac{\varepsilon_x^2}{8}(1+\varepsilon_x) - 0.3\varepsilon_x^2\left(1+\frac{1}{2\sqrt{3}}\varepsilon_x^3\right)+\frac{2}{15}\frac{\varepsilon_y^2}{1-\varepsilon_x}+\left(1-\frac{0.034}{(L/D)^2}\right)\left(\frac{0.016}{1.03-\varepsilon_x}\right)\right) \quad \text{.....(13)}$$

Dynamic Coefficients of The Bearing Performance

The linearized bearing dynamic coefficients are calculated based on steady state analysis. Assume that the journal center start from an arbitrary position (under dynamic load) and let time pass until the journal center position (and everything else) is steady. Clearly, the displacement of the journal from the center of the bearing is a function of the fluid film force and the journal speed. Thus, the bearing oil film stiffness coefficients (rate change of force with displacement), K_{xx}, K_{xy}, K_{yx} and K_{yy}, are functions of the journal's steady-state equilibrium position. The bearing's damping coefficients (rate change of force with velocity); C_{xx}, C_{xy}, C_{yx} and C_{yy}, generated by bearing are similarly functions of the journal's steady-state equilibrium position.

The general existing definitions of the lineraized stiffness and damping coefficients in polar coordinates are derived from Taylor series expansions of the radial components ($\mathbf{f}_\varepsilon$) and tangential components ($\mathbf{f}_\varphi$) of the fluid force (**f**) as shown in Figure (2). In this figure , (B_c^*) is the center of the adjustable pads bearing after using tilting angle (□) is equal 0.1^o; (J_{cs}) is the steady-state equilibrium position of the journal center; (J_c) is the dynamic position of the journal center. Referring to Figure (2), the fluid force (**f**) is,[9].

$$\mathbf{f} = \mathbf{f}_\varepsilon \mathbf{e}_\varepsilon + \mathbf{f}_\varphi \mathbf{e}_\varphi = \begin{bmatrix} \mathbf{e}_\varepsilon & \mathbf{e}_\varphi \end{bmatrix} \begin{bmatrix} \mathbf{f}_\varepsilon \\ \mathbf{f}_\varphi \end{bmatrix} \quad \ldots..(14)$$

Taylor series expansions of the radial component and tangential component of the fluid force (**f**) separately as follows:

$$\frac{d\mathbf{f}}{dt} = \begin{bmatrix} \mathbf{e}_\varepsilon & \mathbf{e}_\varphi \end{bmatrix} \begin{bmatrix} \dfrac{\partial \mathbf{f}_\varepsilon}{\partial \varepsilon}\dfrac{d\varepsilon}{dt} + \left(\dfrac{\partial \mathbf{f}_\varepsilon}{\partial \varphi} - \mathbf{f}_\varphi\right)\dfrac{d\varphi}{dt} + \dfrac{\partial \mathbf{f}_\varepsilon}{\partial \dot\varepsilon}\dfrac{d\dot\varepsilon}{dt} + \dfrac{\partial \mathbf{f}_\varepsilon}{\partial \dot\varphi}\dfrac{d\dot\varphi}{dt} \\ \dfrac{\partial \mathbf{f}_\varphi}{\partial \varepsilon}\dfrac{d\varepsilon}{dt} + \left(\dfrac{\partial \mathbf{f}_\varphi}{\partial \varphi} - \mathbf{f}_\varepsilon\right)\dfrac{d\varphi}{dt} + \dfrac{\partial \mathbf{f}_\varphi}{\partial \dot\varepsilon}\dfrac{d\dot\varepsilon}{dt} + \dfrac{\partial \mathbf{f}_\varphi}{\partial \dot\varphi}\dfrac{d\dot\varphi}{dt} \end{bmatrix} \quad \ldots..(15)$$

For a very short-time interval (Δt) when a small perturbation is applied to the journal center during the operation that was in the steady-equilibrium position (ε_s, φ_s) equation (15) can be approximated as follows through multiplying the two sides by time interval (Δt).

Now, the stiffness and damping coefficients can be defined as follows: state

$$\begin{bmatrix} \mathbf{f}_\varepsilon - (\mathbf{f}_\varepsilon)_s \\ \mathbf{f}_\varphi - (\mathbf{f}_\varphi)_s \end{bmatrix} = -\begin{bmatrix} K_{\varepsilon\varepsilon} & K_{\varepsilon\varphi} \\ K_{\varphi\varepsilon} & K_{\varphi\varphi} \end{bmatrix}\begin{pmatrix} C_r\Delta\varepsilon \\ C_r\varepsilon\Delta\varphi \end{pmatrix} - \begin{bmatrix} C_{\varepsilon\varepsilon} & C_{\varepsilon\varphi} \\ C_{\varphi\varepsilon} & C_{\varphi\varphi} \end{bmatrix}\begin{pmatrix} C_r\Delta\dot\varepsilon \\ C_r\varepsilon\Delta\dot\varphi \end{pmatrix} \quad \ldots..(16)$$

$$\begin{bmatrix} K_{\varepsilon\varepsilon} & K_{\varepsilon\varphi} \\ K_{\varphi\varepsilon} & K_{\varphi\varphi} \end{bmatrix} = -\begin{bmatrix} \dfrac{\partial \mathbf{f}_\varepsilon}{C_r \partial \varepsilon} & \dfrac{\partial \mathbf{f}_\varepsilon}{\varepsilon C_r \partial \varphi} - \dfrac{\mathbf{f}_\varphi}{C_r \varepsilon} \\ \dfrac{\partial \mathbf{f}_\varphi}{C_r \partial \varepsilon} & \dfrac{\partial \mathbf{f}_\varphi}{\varepsilon C_r \partial \varphi} + \dfrac{\mathbf{f}_\varepsilon}{C_r \varepsilon} \end{bmatrix}$$

$$\begin{bmatrix} C_{\varepsilon\varepsilon} & C_{\varepsilon\varphi} \\ C_{\varphi\varepsilon} & C_{\varphi\varphi} \end{bmatrix} = -\begin{bmatrix} \dfrac{\partial \mathbf{f}_\varepsilon}{C_r \partial \dot\varepsilon} & \dfrac{\partial \mathbf{f}_\varepsilon}{C_r \varepsilon \partial \dot\varphi} \\ \dfrac{\partial \mathbf{f}_\varphi}{C_r \partial \dot\varepsilon} & \dfrac{\partial \mathbf{f}_\varphi}{C_r \varepsilon \partial \dot\varphi} \end{bmatrix} \quad (17)$$

The final expressions for the stiffness and damping coefficients described by equation (17),[10].

$$\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} = \begin{bmatrix} \cos\varphi_s & -\sin\varphi_s \\ \sin\varphi_s & \cos\varphi_s \end{bmatrix}\begin{bmatrix} K_{\varepsilon\varepsilon} & K_{\varepsilon\varphi} \\ K_{\varphi\varepsilon} & K_{\varphi\varphi} \end{bmatrix}\begin{bmatrix} \cos\varphi_s & \sin\varphi_s \\ -\sin\varphi_s & \cos\varphi_s \end{bmatrix} \quad \ldots.(18)$$

$$\begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} = \begin{bmatrix} \cos\varphi_s & -\sin\varphi_s \\ \sin\varphi_s & \cos\varphi_s \end{bmatrix} \begin{bmatrix} C_{\varepsilon\varepsilon} & C_{\varepsilon\varphi} \\ C_{\varphi\varepsilon} & C_{\varphi\varphi} \end{bmatrix} \begin{bmatrix} \cos\varphi_s & \sin\varphi_s \\ -\sin\varphi_s & \cos\varphi_s \end{bmatrix}$$

..... (19)

The Power Loss

The power consumption of a journal bearing results from the shearing of the oil in the bearing oil film. The total power consumption for this bearing is given by,[11].

$$P_L = \sum_{i=1}^{N_{pad}} 2.\pi.\omega.\left(L.D.P\frac{e.\sin\phi}{2} + \int_0^{80} \frac{\pi.\mu.\omega.D.B_i.R^2}{h_t} d\theta \right)$$

......(20)

In Equation (20), the second term in the brackets represents the torque as computed from the average velocity shear rate a cross the oil film. Because of the superposition of a parabolic pressure induced velocity profile on the linear velocity profile caused by shaft rotation, this second term is the torque at mid – film, which is midway between the torque on the shaft and the torque on the bearing. While the first term represents half the torque correction due to shaft eccentricity for the difference between bearing torque and shaft torque.

Results and Discussions

Table (1) shows the input of different values of length diameter ratios ($\frac{L}{D}$) are equal 0.25,0.5,0.75 and 0.95 with eccentricity ratio components ($□_x$,$□_y$) to obtain the output values of mobility (M) and journal center velocity ($\frac{de}{dt}$) which depends on equation (9). This table shows that when length-diameter ratio ($\frac{L}{D}$) decreased the mobility parameter (M) and journal center velocity will increased.

To determine the journal center velocity ($\frac{de}{dt}$); the mobility map for different length-diameter ratio ($\frac{L}{D}$) can be drawn as shown in Figures (3), (4), and (5) which represent the clearance circle map of mobility of journal center under dynamic load, these figures help to know the orbit of journal center during the steady state. The center of each map represents the center of the bearing, the radius from this center is equal to (□). These maps are as a function of eccentricity ratio (□) and length diameter ratios ($\frac{L}{D}$), when the eccentricity ratio component ($□_x$) in X-direction change from (0) to (1); the mobility parameter (M) will decrease when the eccentricity ratio component ($□_x$) in X- direction change from (0) to (-1); the mobility parameter (M) will increase if length-diameter ratio ($\frac{L}{D}$) is constant.

Figure (6) shows the dynamic stiffness coefficients (K_{ij}) as a function of eccentricity ratio (□), the magnitudes of the cross-coupled stiffness coefficients and direct stiffness are increased with the increase in the values of eccentricity ratio. It was found that, the vertical stiffness (Kyy) value is much more than the horizontal stiffness (Kxx) value at the maximum values of eccentricity ratio, of particular importance for stability considerations are the cross-coupled terms, (Kyx) and (Kxy) [this fact concluded from [4]].The term (Kyx) is actually negative (inverted for plotting) and has a very large magnitude across the entire eccentricity ratio range.

Figures (7) shows the dynamic damping coefficients (C_{ij}) as a function of eccentricity ratio (□) , the magnitudes of the cross-coupled damping coefficients and direct damping are increased with the increase of the values of eccentricity ratio. It was found that the vertical damping (Cyy) is much than the horizontal damping (Cxx) at the maximum values of eccentricity ratio. The coupled damping coefficients (Cxy) and (Cyx) have the same values (negative values) in the range of the studied load capacities.

Figure (8) illustrats the relationship between the eccentricity ratio (□) and the power loss (P_L) versus length –diameter ratio ($\frac{L}{D}$). The power loss proportional with the eccentricity ratio (□) at constant length –diameter ratio ($\frac{L}{D}$). Also, the eccentricity ratio proportional with the power loss at constant length –diameter ratio ($\frac{L}{D}$). The power loss for this bearing were found to increase when length –diameter ratio ($\frac{L}{D}$) increase, the power loss (large load, low speed, and ligth lubricant viscosity) determine large operating eccentricities, □→1.0).

Conclusions

From the results of this work the following conclusions can be obtained:

1-When the eccentricity ratio (□) components has a minus value, this results give high values of journal center velocity also mobility (M) components.

2-When the length diameter ratio ($\frac{L}{D}$) is increased, the maximum value of the journal center velocity (de/dt) is decreased.

3- The vertical stiffness coefficient (Kyy) rapidly increase at eccentricity ratio equal 0.58 and over that. This means that the bearing becomes more stable at large value of eccentricity ratio.

4- The vertical damping coefficient (Cyy) depends upon the eccentricity ratio .

5- The power loss is small for range of the eccentricity ratio when length diameter ratio is small .

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Table (1) The Change of Mobility Components with Length –Diameter Ratios ($\frac{L}{D}$) and Eccentricity Ratio ( ) Components

L/D	*x*	*y*	M_x	M_y	*M*	$\frac{de}{dt}$ **10³*
0.25	-0.7	-0.5	20.215	7.138	21.438	1154.179
0.25	0.7	0.5	0.339	-0.668	0.750	40.372
0.25	0.7	-0.5	0.339	0.668	0.750	40.372
0.25	0.9	0.0	0.021	0.000	0.021	1.136
0.25	0.9	-0.2	0.024	0.061	0.065	3.499
0.25	0.0	0.0	5.107	0.000	5.107	274.950
0.50	-0.7	-0.5	5.328	1.760	5.612	151.071
0.50	0.7	0.5	0.110	-0.206	0.233	6.272
0.50	0.7	-0.5	0.110	0.206	0.233	6.272
0.50	0.9	0.0	0.009	0.000	0.009	0.246
0.50	0.9	-0.2	0.010	0.024	0.026	0.691
0.50	0.0	0.0	1.372	0.0	1.372	36.933
0.75	-0.7	-0.5	2.486	0.769	2.602	46.542
0.75	0.7	0.5	0.066	-0.117	0.135	2.421
0.75	0.7	-0.5	0.066	0.117	0.135	2.421
0.75	0.9	0.0	0.007	0.0	0.007	0.130
0.75	0.9	-0.2	0.008	0.017	0.019	0.341
0.75	0.0	0.0	0.677	0.000	0.677	12.112
0.95	-0.7	-0.5	1.446	0.414	1.504	21.240
0.95	0.7	0.5	0.050	-0.082	0.097	1.372
0.95	0.7	-0.5	0.050	0.082	0.097	1.372
0.95	0.9	0.0	0.006	0.000	0.006	0.084
0.95	0.9	-0.2	0.006	0.013	0.015	0.211
0.95	0.0	0.0	0.430	0.000	0.430	6.073

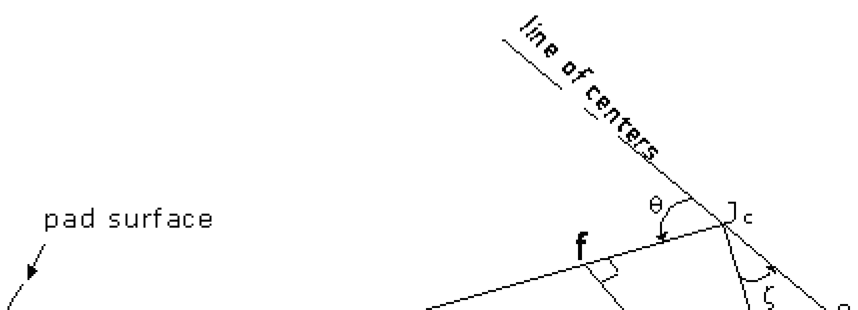

Figure (1) Point on Pad Surface During The Adjusting Process, [1]

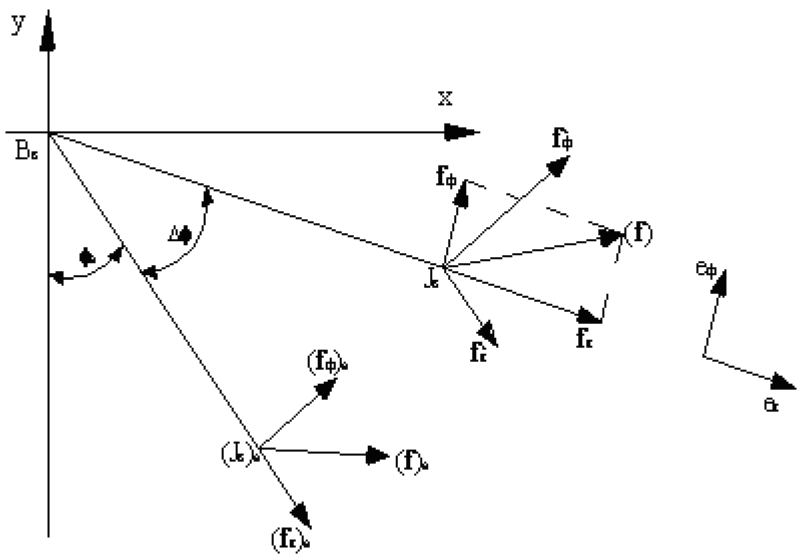

Figure (2) Sketch of The Analysis of The Oil Film Force

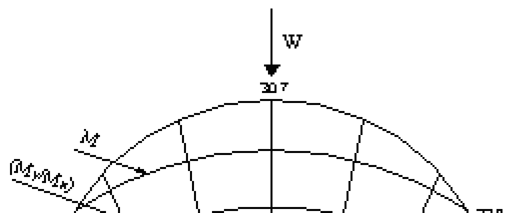

Figure (3) Eccentricity Circle Map of Constant Mobility and Direction of Mobility for L/D=0.25

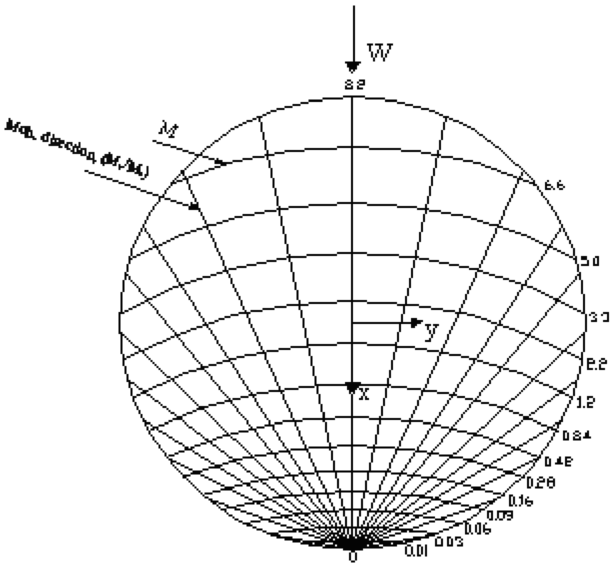

Figure (4) Eccentricity Circle Map of Constant Mobility and Direction of Mobility for L/D=0. 5

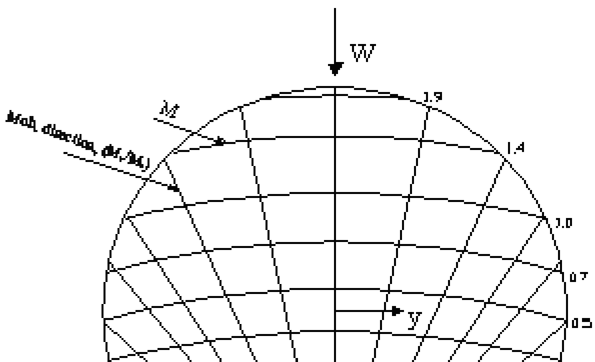

Figure (5) Eccentricity Circle Map of Constant Mobility and Direction of Mobility for L/D=0.95

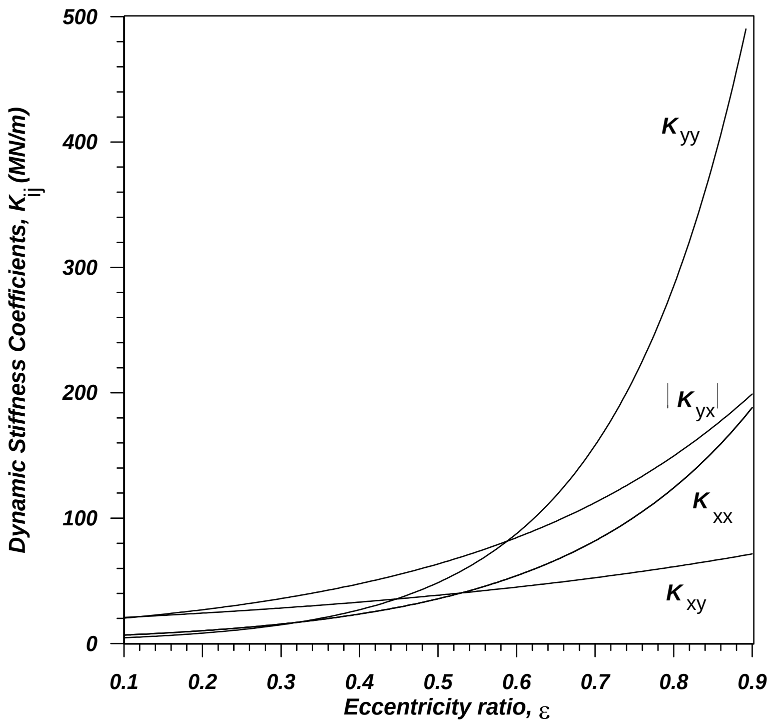

Figure (6) The Eccentricity Ratio Effect on The Dynamic Stiffness Coefficients

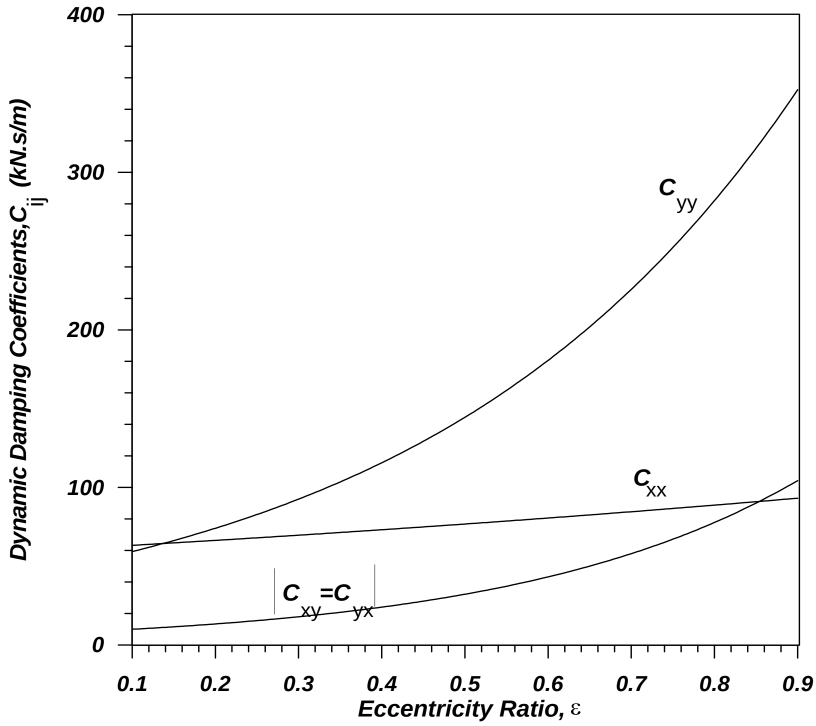

Figure (7) The Eccentricity Ratio Effect on The Dynamic Damping Coefficients

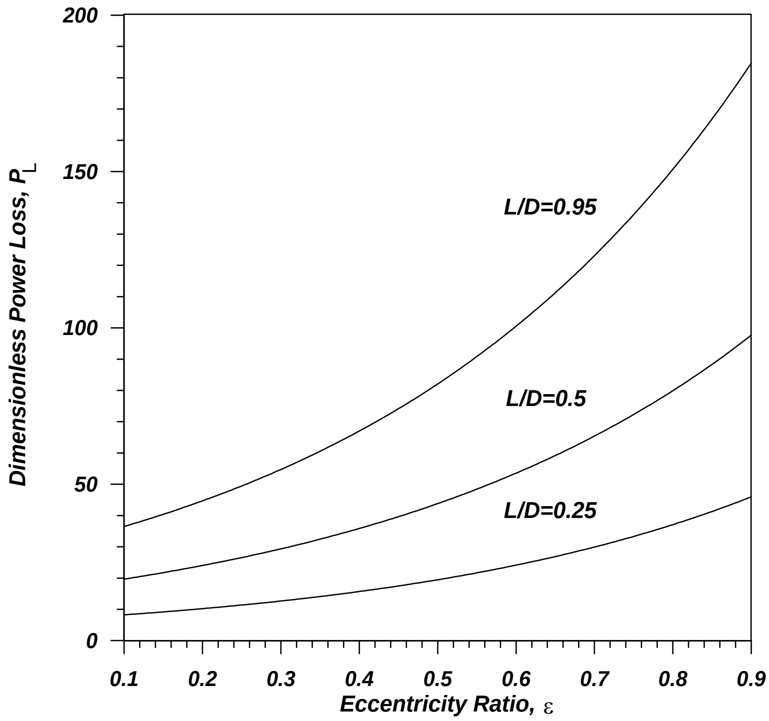

Figure (8) The Dimensionless Power Loss and Eccentricity Ratio for Different (L/D) Ratios