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The Effects of Bearing Dimensions and Adjustable Mechanism Position for Six Pads Bearing under Dynamic Load

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Abstract

The hydrodynamic bearing consists of six pads these pads have the ability to tilt about the clamping edge. Reynolds equation (2D) for dynamically loaded was used to find the generated pressure value throughout the mobility method and the finite difference method. The effects of many parameters of bearing were studied in this paper such as length to diameter ratios, power loss, oil flow rate, Sommerfeld number, load number, friction coefficient and by using the "Ansys program" for stress and strain analysis over the pad surface (white metal), to select the best position location for adjustable pads mechanism from leading edge angle to trailing edge angle of pad central angle (PCA=55°) by taking nine different positions. The best angle for the adjusting member was found about (41.25^o) after leading angle of the pad, the adjusting member in the maximum pressure region gave minimum radial displacement (elastic deformation) values.

Keywords: Non-circular bearings, adjustable mechanism, Hydrodynamic bearing parameters.

تأثيرات أبعاد المسند و موقع آليةِ التعيير لمسند ذو ست وسائد تحت الحملِ الديناميكي

الخلاصة

مسند هيدروديناميكي ذو ستّ وسائد هذه الوسائد لَها القدرةُ للإمَالَة حول حافةِ التعليق. معادلة رينولد (لبعدين) لحمل ديناميكي نستخدمها لإيجاد قيمةِ الضغطِ المتولد بالاعتماد على طريقةِ قابلية الحركة وطريقة الفروقات المحدودةِ. تأثيرات العديد مِنْ متغيرات المسند دُرستْ بهذا البحث مثل نسب طول المسند لقطره ، الطاقة الضائعة ، معدل تدفق زيت التزييت، عدد Sommerfeld، عدد حمل المسندِ، معامل احتكاك وباستخدام "برنامج Ansys" لتحليلِ الإجهادَ والانفعال على معدن سطح الوسادة (المعدن الأبيض)، لاختيار أفضل موقع لآليةِ تعيير (إمالة) الوسادة والتي تبدأ من زاوية حافة الدخول (التحميل) للوسادة إلى زاوية نهاية (ذيل) الوسادة التي تبلغ 55^o وتم ذلك بأخْذ تسع مواقع مختلفة. أفضل زاوية لموقع إلية التعديل وُجِدتْ حوالي (41.25^o) بعد زاوية الدخول (التحميل) للوسادة، إلية التعديل تقابل مجال الضغطِ الأقصى لذلك تجعل الإزاحةِ المحورية (التشوه المرن) في أدنى قيم .

Notation

C_r	Radial bearing clearance	m
D	Bearing and journal diameter respectively	m

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E	Modulus of elasticity	N / m^2
e	Eccentricity	m
F	Oil film load	N
f	Friction coefficient	–
h	Oil film thickness	m
L	Axial length of bearing	m
Ln	Load number	–
P	Generated pressure in oil film thickness	N / m^2
P_L	Power loss	W
Q	Oil flow rate and oil flow rate components	m^3/s
$\overline{Q}$	Dimensionless Oil flow	–
R, r	Bearing and journal radius respectively	m
S	Sommerfeld number	–
t_p	Pad thickness	m
W	External load, radial W_r and tangential W_t components	N
ϕ	Calculated attitude angle	–
δ	Pad tilt angle	deg.
ε	Eccentricity ratio	–
μ	Viscosity of lubricant	Pa.s
ρ	Density of lubricant	kg / m^3
ω	Angular journal speed	rad/s
α	Groove angle	deg.
λ	Whirl ratio	–
ψ	Assumed attituted angle	deg.
Λ	Bearing number	–

Introduction

The hydrodynamic bearing having the ability to adjustable pads angle manually controlled from outside of bearing when it is operating under dynamic load.The adjusting process is occurring when the radial dimension (oil film thickness) is extended due to the wear or from other external causes, this process generates a small displacement (very small tilt angle) in the pad. Subbiah (1999) [1], invented an adjustable pad bearing for the rotor of a steam turbine power plant. Moreover, an adjustable pad with other two tilting pads was designed to provide stability in the rotor system by eliminating bearing cross-coupling properties. The adjustment member starts to adjust in position the pad to verify minimum pad deflection in all directions. Parkins (1999) [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. The adjustable sleeve is also a part of a hydraulic damper whose parameters can be controlled during operation. Yansong and Chao [4] were presented the steady state mixed-TEHD (thermo-elasto-hydrodynamic) model for journal bearings has been developed. The model considers the fluid flow in the gap formed by surface thermoelastic deformations and a temperature–pressure–viscosity relationship for the lubricant, as well as an angular misalignment between the journal and the bearing. Numerical simulations of the operation of a typical journal bearing are conducted and the importance of several contributing factors in mixed lubrication is discussed. In this work the topic has been divided into two parts, the first is confined the effects of the length to diameter ratios (L/D) on the generated pressure and many bearing parameters for six pads after using tilting pad angle (δ), and secondly by using the "Ansys program" to select the best position location for adjustable pads mechanism from leading edge angle (αL) to trailing edge angle (αT) of the pad.

Description of the Pad and Adjusting Member

The hydrodynamic bearing consists of six pads as shown in figure (1),the bearing pad include two layers of white metal and high strength steel, also the pivot and an adjusting member are mechanically fastened with bearing (the adjusting member is mechanically fastened with high strength steel layer and extends outside to couple with the bearing case). In this type of bearing, thickness of the case is more than 2-times of pad thickness (the white metal its about 5 mm and lower layer of high strength steel its about 6.5 mm). Adjusting member is drawn one pad as shown in Figure (2). The adjusting mechanism presented as two square threaded screw, the square thread of power screw is mechanically fastened with high strength steel layer and extends outside to couple with the bearing case (square threaded has maximum efficiency and minimum radial or bursting pressure on the net). During the adjusting process, rotation of the screw translates to the pad to move in the radial direction by the angle (δ). The considerations for the power screw selection is based on the maximum shear stress that can occur in the screw which must be equal or less than the shear yield strength of the screw material. Table (1) lists the dimensions of the bearing.

Theoretical Analysis

The pressure distribution for dynamically loaded journal bearing can be obtained by solving the Reynolds equation (1), by using finite difference method. Under dynamic action, using mobility method to determination the

eccentricity velocity for journal center and selected the oil film thickness equation for non-circular bearing after using tilting angle (δ),[5].

$$\frac{\partial}{\partial q}\left(\bar{h}^{3}\frac{\partial \bar{p}}{\partial q}\right)+\left(\frac{D}{L}\right)^{2}\bar{h}^{3}\frac{\partial^{2}\bar{P}}{\partial \bar{z}^{2}}=\Lambda\frac{\bar{h}}{\partial q}+2\Lambda l\frac{\partial \bar{h}}{\partial t} \quad ......(1)$$

(i).Steady State Force Analysis

Under steady state condition equation (1) can be reduced to

$$\bar{h}^{3}\frac{\partial^{2}\bar{p}}{\partial q^{2}}+3\bar{h}^{2}\frac{\partial \bar{h}}{\partial q}\frac{\partial \bar{p}}{\partial q}+\left(\frac{D}{L}\right)^{2}\bar{h}^{3}\frac{\partial^{2}\bar{p}}{\partial \bar{z}^{2}}+\Lambda\frac{\partial \bar{h}}{\partial q}=0 \quad ..... (2)$$

The load components along the line of centers and its perpendicular direction are found from [6].

$$\bar{W}_{r}=\frac{W_{r}}{L.D.P}=\sum_{pad\ No\ 1}^{pad\ No\ 6}\frac{1}{2}\int_{0}^{L}\int_{a_L}^{a_T}\bar{P}.\cos(q-\Psi)dq.d\,z \quad ...... (3)$$

$$\bar{W}_{t}=\frac{W_{t}}{L.D.P}=\sum_{pad\ No\ 1}^{pad\ No\ 6}\frac{1}{2}\int_{0}^{L}\int_{a_L}^{a_T}\bar{P}.\sin(q-\Psi)\,dq.d\,z \quad ...... (4)$$

$$\bar{W}=[\bar{W}_{r}^{2}+\bar{W}_{t}^{2}]^{2} \quad ..... (5)$$

$$\bar{F}=\frac{F}{2.L.C_{r}.P}=\int_{0}^{L}\int_{a_L}^{a_T}\left(\frac{1}{4}.\bar{h}.\frac{\partial \bar{P}}{\partial q}+\frac{\Lambda}{12}.\frac{1}{\bar{h}}\right)dq\,d\,z \quad ..... (6)$$

(ii) The Bearing Parameters

The dimensionless number "S " is known as Sommerfeld number and is widely used as a characteristic number for journal – bearing, [7].

$$S=\left(\frac{R}{Cr}\right)^{2}\mu\frac{\omega}{P_{T}} \quad ..... (7)$$

Where,

$$P_{T}=\frac{1}{LD}\sum_{pad\ No\ 1}^{pad\ No\ 6}W \quad ..... (8)$$

The dimensionless friction coefficient (f), defined as the total drag force on the journal surface in the bearing divided by the total load, is given by, [7]:

$$f=\frac{C_{r}}{2.r}\left[\frac{4.r^{3}.L.w.m}{W.C_{r}^{2}.\sqrt{(1+S)^{2}-e^{2}}}+\frac{e}{2}.\sin b+\frac{48.w.m.L^{3}.r.(1-2.a^{2})S^{2}.e^{2}}{W.C_{r}^{2}}\sum_{pad\ No\ 1}^{pad\ No\ 6}\frac{1}{(2.i-1)^{4}}.\int_{0}^{55}\frac{\sin f}{(1+S+e.\cos f)^{2}}df\right] \quad ..... (9)$$

The load number depends on the parameters in Sommerfeld number and can be determined by

$$Ln = \frac{F(C_r/R)^2}{\omega_{av}\mu LD} \quad \text{..... (10)}$$

The total power consumption for the new bearing is given by, [8].

$$PL = \sum_{i=1}^{N_{pad}} 2pw\left(L.D.P\frac{e\sin f}{2} + \int_0^{55}\frac{pmwDB_1R^2}{h}\,dq\right)_{pad} \quad \text{..... (11)}$$

$$\overline{PL} = \frac{PL.C_r}{m.w^2.L.D^2.r} \quad \text{.....(12)}$$

Where, $\overline{P}_L$ = Dimensionless Power loss

In Equation (13), the second term in the brackets represents the torque as computed from the average velocity shear rate a cross the oil film.

Oil Flow Rates

The bearing has a gap in the pad through which oil could leave the bearing. Therefore, the oil flow in direction, z and x must be estimated. Total oil flow rate for the positive pressure regions from both the z and x directions is then given by, [9].

$$Q = Q_q + Q_z \quad \text{..... (13)}$$

$$\overline{Q} = \frac{Q}{L.C_r.D.w} \quad \text{.....(14)}$$

(i).Oil Flow Rate in the q-Direction

The oil flow rate in the θ-direction can be calculated using the following formula:

$$Q_q = \sum_{padNo1}^{padNo6}\left[\frac{u.h.L}{2}\int_0^L\left(\frac{h^3.L}{12.m}.\frac{\Delta P}{PCA.D.(m-1)}\right)dz\right]_{pad} \quad \text{..... (15)}$$

(ii).Oil Flow Rate in the Z-Direction

The oil flow rate in the Z-direction, can be calculated using the same principles:

$$Q_z = \sum_{pad\ No\ 1}^{pad\ No\ 6}\frac{r}{0.5L}\sum_{i=1}^{i=m}\sum_{j=1}^{j=n}\int_{a_L}^{a_T}\left[\frac{\Delta P}{m}.h^3.dq\right]_{pad} \quad \text{.....(16)}$$

Selection a Suitable Position for the Adjusting Member

Selection a suitable position for the adjusting member is very important to reduce the elastic deformation in the pad surface. **Ansys** program was used for stress and strain analysis to select a suitable position for the adjusting mechanism, where nine cases taken for this purpose starts from the pad leading edge (nine positions taken for an adjustable mechanism), i.e. (θ_{adj}) is the angular position of the adjusting mechanism measured from the pad leading edge angle.

Discussions of Results

The results of this paper are illustrated below. Figure (3) represented the effect of changing the values of length diameter ratios (L/D) on the pressure distribution in the circumferential direction for the pads. From this Figure, when (L/D) ratio increases the pressure distribution decreases if eccentricity ratio remain constant.

Figure (4) represents the relationship between the eccentricity ratio (ε) and the Sommerfeld number (S). When (L/D) ratio constant, the Sommerfeld number (S) decreases when the eccentricity ratio increases. If (L/D) ratio decreases the Sommerfeld number increases for the same eccentricity ratio.

Figure (5) illustrates the relationship between the coefficient of friction (f) and eccentricity ratio versus (L/D) ratios. When eccentricity ratio increases the coefficient of friction decreases at constant (L/D) ratio. In addition, when (L/D) ratio increases the coefficient of friction decreases.

Figure (6) represents the relationship between the eccentricity ratio (ε) and the dimensionless oil flow ($\overline{Q}$). When (L/D) ratio constant, the dimensionless oil flow increases when the eccentricity ratio increases. If (L/D) ratio decreases the dimensionless oil flow increases at the same eccentricity ratio. Table (2) illustrates the values of the load number and power loss for different values at eccentricity (ε) and (L/D) ratio.

The Figure (7) illustrates the displacement in the pad shell No.3 at e=0.02 mm and N=11000 rpm, figure (A) when L/D=0.5 and (B) when L/D=0.25. It was found that the maximum displacement in the pad occurs in the tangent with the maximum pressure that arise in oil film thickness as shows in figure (3) and the minimum displacement obtained when the adjusting member is under maximum pressure region. Table (3) gives more information for the values of the maximum displacements in the nine cases of the locations of the adjusting members of the pads. Finally, It was found that the maximum pressure occur in the more-or-less about 41.25^{o} from the pad leading edge, therefore the adjusting member must be put in this region from the pad.

Conclusions

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

1- The maximum pressure values was somewhere near to the midpoint in the second half of the length of the pad and it has maximum value in midpoint from the width of the pad.
2- The pressure distribution on each pad was found to be dependent on the eccentricity ratio and length diameter ratio.
3- Sommerfeld number increases at low values of eccentricity ratio.
4- As the flow rate shows an increase in magnitude with eccentricity ratio but when the axial length increase to diameter the oil flow rate decrease, because the area of pad surface is increase.
5- When the length diameter ratio increase, the load number and dimensionless power loss increase also, but these increment values

depend on the values of eccentricity ratio.

6- When the length diameter ratio increase, the elastic deformation decreases over pad surface, because the friction coefficient decreases.

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Table (1) Basic Dimensions and Operation Conditions for The Bearing

Bearing Parameter	Value
Diameter of bearing, D	65.7 mm
Diameter of journal , d	65.553 mm
Radial clearance, C_r	0.0985 mm
Pad thickness, t_p	11.5 mm
Pad central angle, PCA	55^o
Pad tilt angle, δ	0.15^o
Number of pads, N_{pads}	6
Angular journal speed	11000 r.p.m
Angular dimension of each oil groove	5^o

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Table (2) The Change of Load Number and Dimensionless Power Loss with Length diameter Ratios and Eccentricity Ratio

Eccentricity Ratio (e)	Length Diameter Ratio(L/D)	Load Number (Ln)	Dimensionless Power Loss (PL)
0.1	0.25	0.017	9.101
0.2	0.25	0.035	9.933
0.3	0.25	0.061	13.771
0.4	0.25	0.097	15.321
0.5	0.25	0.168	26.098
0.6	0.25	0.291	36.871
0.7	0.25	0.544	48.144
0.8	0.25	1.231	51.099
0.1	0.5	0.080	17.162
0.2	0.5	0.099	20.302
0.3	0.5	0.141	22.908
0.4	0.5	0.301	29.001
0.5	0.5	0.651	42.730
0.6	0.5	0.911	73.220
0.7	0.5	1.698	89.531
0.8	0.5	4.001	99.019
0.1	0.9	0.167	36.091
0.2	0.9	0.420	38.009
0.3	0.9	0.901	44.032
0.4	0.9	1.201	49.542
0.5	0.9	1.933	98.415
0.6	0.9	3.212	140.641
0.7	0.9	6.944	159.431
0.8	0.9	15.331	179.001

Table (3) The Results of Ansys Program of Nine Cases for Pad No.3.
8
(L/D=0.5, e=0.02 mm, d=0.15^{o} **)**

Case No.	Adjustable Member Position, θ_{adj}	Max. Displacement
1	0^{o}	0.008243 mm
2	6.875^{o}	0.007901 mm
3	13.75^{o}	0.007001 mm
4	20.625^{o}	0.006231 mm
5	27.5^{o}	0.005432mm
6	34.375^{o}	0.003212 mm
7	41.25^{o}	**0.001984 mm**
8	48.125^{o}	0.004321 mm
9	55^{o}	0.005898 mm

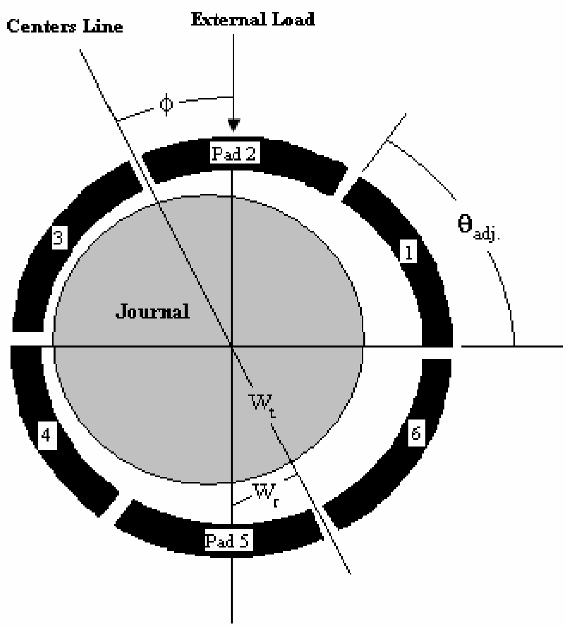


Figure (1) Schematic Diagram of Adjustable 6-Pads Bearing

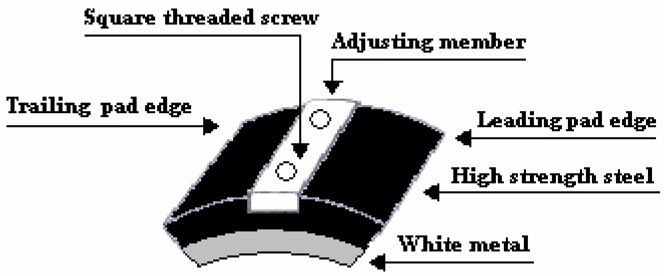


Figure (2) Pad Materials and the Adjusting member

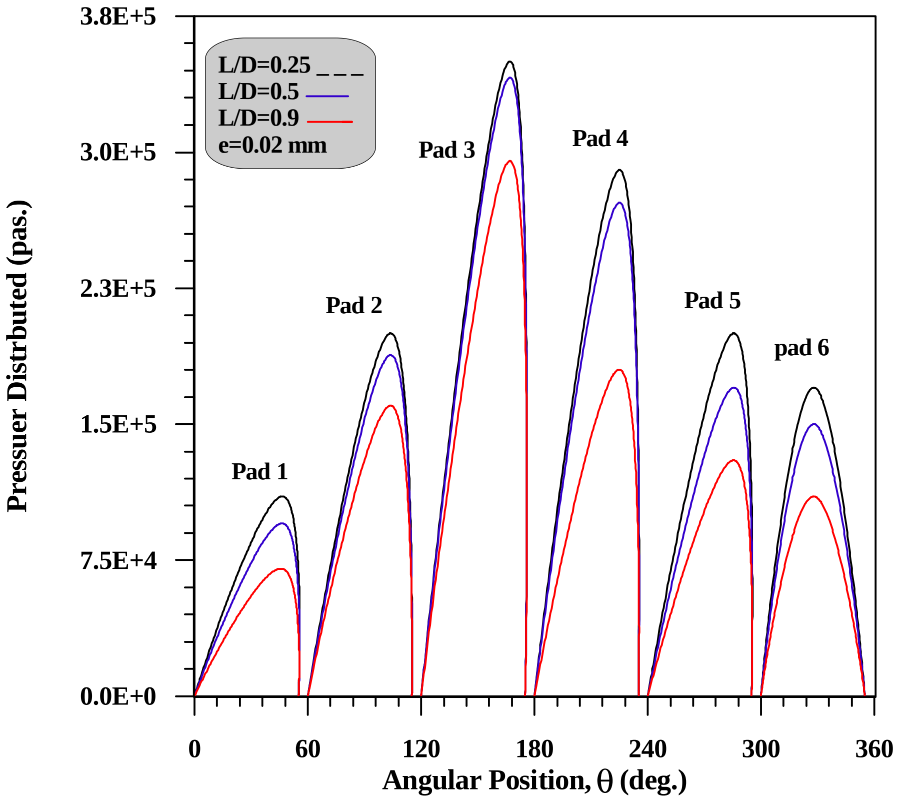


Figure (3) The Effect of Length-Diameter Ratios (L/D) on The Pressure Distribution

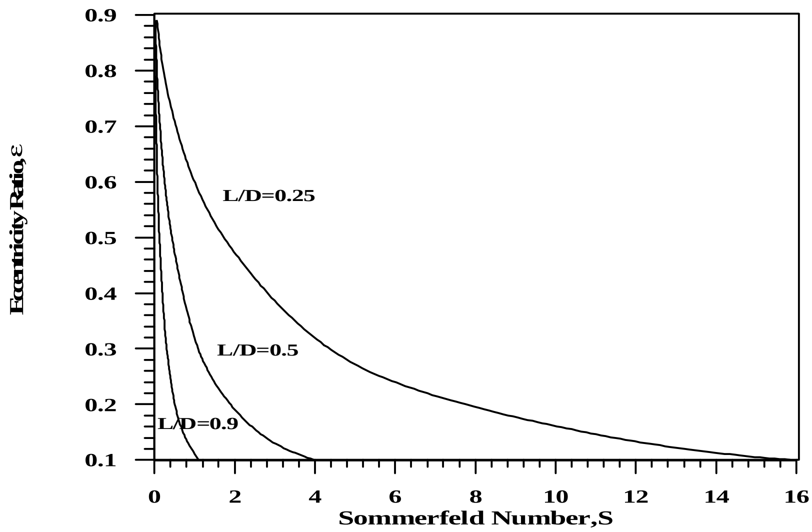


Figure (4) Sommerfeld Number and Eccentricity Ratio For Different (L/D) Ratios

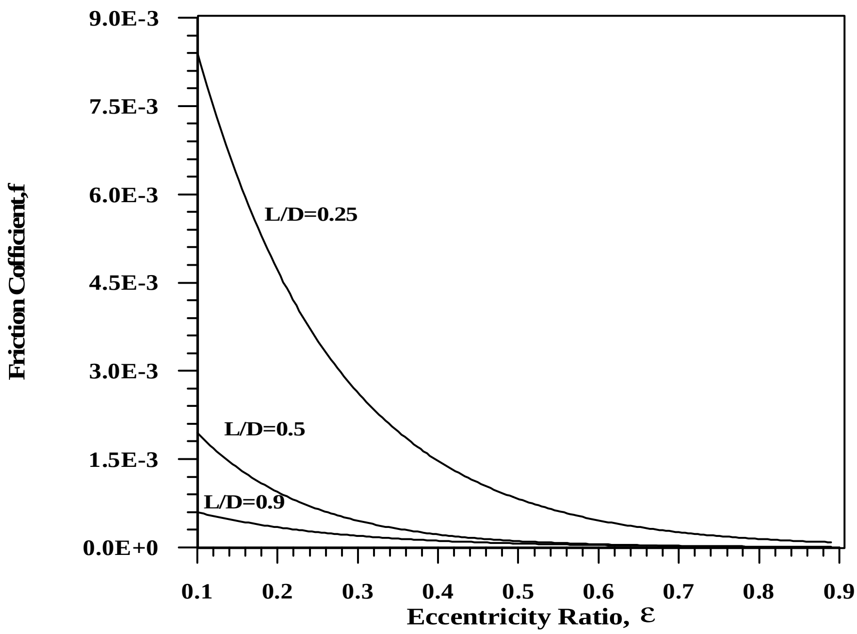


Figure (5) Friction Coefficient, f and Eccentricity Ratio For Different (L/D) Ratios

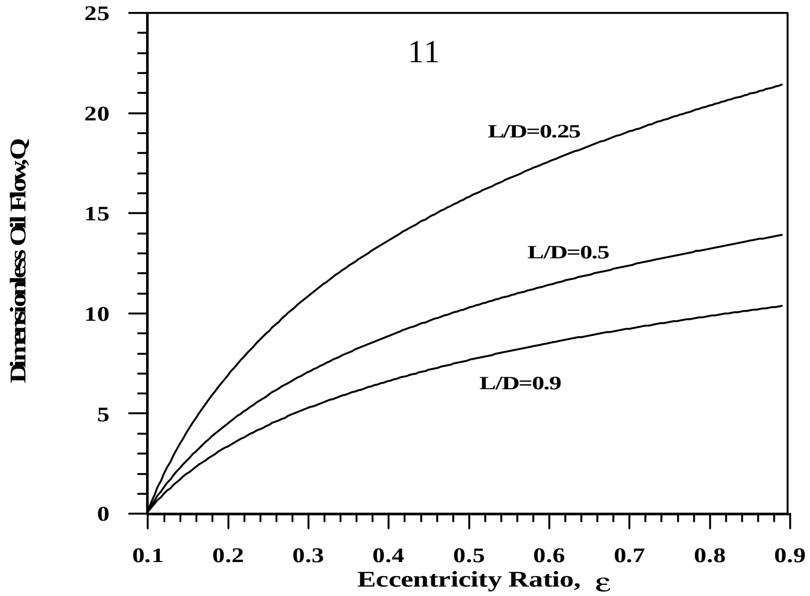
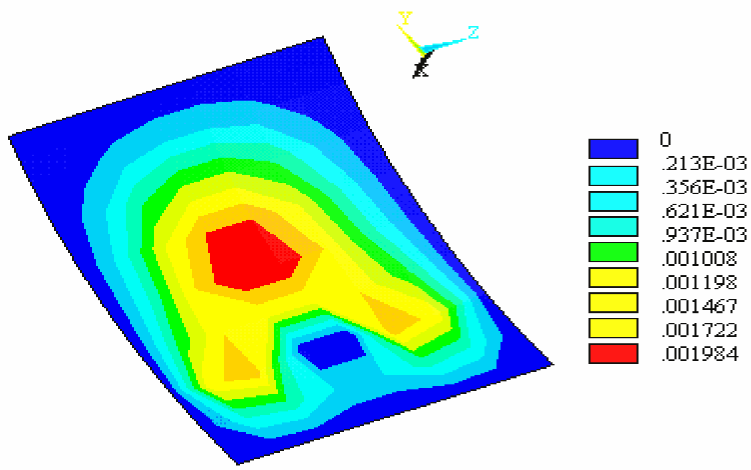
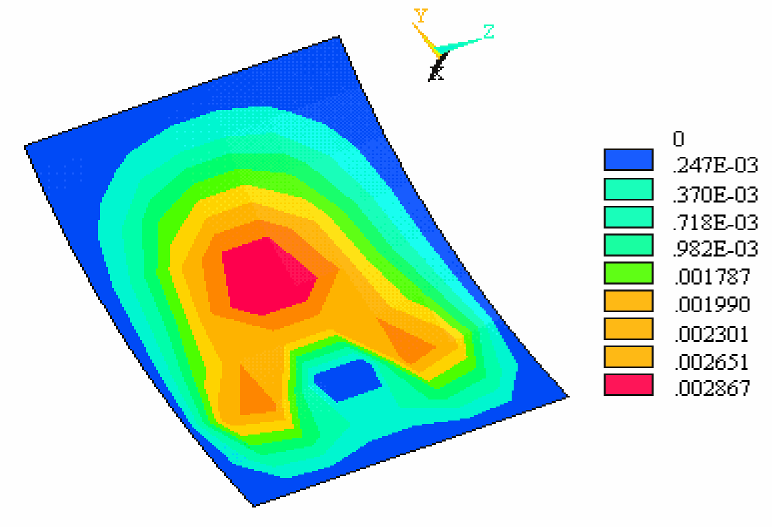


Figure (6) Dimensionless Oil Flow and Eccentricity Ratio For Different (L/D) Ratios



(A)



(B)

Figure (7) The Displacement Distribution in the Loading Direction over Pad No. 3 (A) - L/D=0.5, (B) - L/D=0.25