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The Dynamic Coefficients and Elastic Deformation with Thermal Effect For Cylindrical Pivot Tilting 5-Pad Bearing

Dr. Muhannad Z. Khelifa*

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Abstract

The paper describes the results of theoretical study of the tilting pad bearing consist of five pads each pad can be titling 0.2^o about a cylindrical pivot at high speed of rotating journal. The total oil film thickness includes elastic deformation with thermal effect had been estimated. By using finite element method to solve the Reynolds equation for dynamic load, from this solution generated pressure, components of oil film forces and moments are expressed as functions of journal co-ordinates, dynamic coefficients (stiffness and damping coefficients) these coefficients of oil film can be determined by differentiation of the oil film force finally elastic deformation of pad surface was obtained. This paper also includes the study of many parameters such as pivot offset, preload factor, direction of load (load on pad and load between load) and dynamic coefficients were presented as functions of Sommerfeld number.

Results show that good bearing performance when the pad preload factor is increase and pivot offset decreases.

Keywords: Tilting pads bearing, Cylindrical pivot, Dynamic coefficients, Elastic deformation, Pivot offset

المعاملات الديناميكية و التشوه المرن بالتاثير الحراري لمسند ذو خمسه وسائد قابلة للإمَالَة حول مرتكز اسطواني

الخلاصة

يَصِفُ البحث نَتائِجَ الدراسةِ النظريةِ لمسند ذو وسائد قابلة للإمالة مكون من خمس وسائد، كُــلّ وسادة يُمْكِنُها إن تميل بمقدار 0.2^o حول مرتكز أسطواني عند سرعةِ دورانية عاليـــة للمحـــور. إن سمك طبقة التزييت الكلية تتضمن التشوه المرن مع التأثير الحراري قد تم حسابه. باستخدام طريقـــه العناصر المحددة لحل معادلةِ Reynolds للحمل الديناميكي، مِنْ هذا الحَلِّ وجد الـــضغط المتولـــد ، مركبات القوة ألناشئه في طبقه الزيت وكذلك مركبات العزوم وتكون قيمهم كدالة لإحداثيات وضـــع المحور, معاملات الديناميكية (معاملات المرونة و التخميد) هذه معاملات طبقة الزيت تم احتـــسابها من تفاضل القوة ألناشئه في هذه الطبقة و أخيرا التشوه المرن لسطح الوسادة تم الحـــصول علـــيهم. تضمن هذا البحث كذلك دراسة العديد من المتغيرات منها موضع المرتكز بنسبة ســـطح الوســـادة الخارجي , معامل التحميل اتجاه تحميل الحمل على الوسادة (الحمل على سطح الوسادة و الحمـــل مابين وسادتين) وكذلك معاملات الديناميكية كدالة لتغيير عدد Sommerfeld. تظهر النَتـــائِجَ ان أداءِ المسند يكون جيد عندما preload factor يزداد وكذلك عند نقصان pivot offset.

الكلمات المرشدة: مسند قابل للإمالة, مرتكز اسطواني, المعاملات الديناميكية, التشوه المرن,انحراف المرتكز.

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2412-0758/University of Technology-Iraq, Baghdad, Iraq

Notation

C_p	Radial pad clearance $C_P = (R_P - r)$	m
C_r	Radial bearing clearance $C_r = (R - r)$	m
D	Diameter of bearing	m
d_{L-P}	Distance between the leading edge of the pad and the pivot	m
e	Journal eccentricity	m
f_p	Fractional angular position of the pad pivot	degree
H	Heat transfer coefficient	$Wm^{-2}C^{-1}$
h	Oil film thickness is measured at any angle (θ)	m
h_{min}	Minimum inlet oil film thickness	m
h_{in}	Inlet oil film thickness for current pad	m
$h_{r.d}$	Oil film thickness due to radial displacement	m
O_j	Center of journal	_
O_b	Center of bearing	_
O_p	Point pre- tilt position of the pad center of curvature	_
O_p^*	Point post - tilt position of the pad center of curvature	_
PF	Preload factor	
L	Axial pad length	m
NPE	Number of nodes per element	_
R	Bearing radius	m
r Δp	Pad arc length	m
r	Journal radius	m
R_p	Pad radius of curvature	m
S	Sommerfeld number	_
u_j	Journal surface speed	m/s
t_p	The distance between the pivot point and the surface of the pad.	m
W	load	kN
Δθ	Increment in pad angle	degree
Δp	Total angular length of the pad	_
m_0	Lubricant supply viscosity	N/m^2. sec.
q_c	The angle to the line-of-centers which connects O_b to O_j	degree
q	Circumferential angle	degree

q_s	Start angle of the pad	degree
q_e	End angle of the pad and	degree
δ	Tilting angle	degree
ω	Angular speed	rad/sec

Introduction

Dynamics of high speed rotating machinery depend strongly on journal bearing types. Tilting pad journal bearings (TPJB) are dominant as shaft support in such machinery, tilting pad bearings can be classified into the following categories based on the type of pivots they employ [1]: (i) Point contact (Sphere in cylinder). (ii) Line contact (Rocker back or axial pin type of pivot) (iii) Surface contact (Ball in Sphere). It may be noted that they are arranged in the increasing order of load carrying capacity based on pivot strength; two distinct factors limit the load carrying capacity of a tilting pad bearing, viz the maximum pad metal/oil film temperature and the pivot stresses. Tilting pad journal bearings can either be of a load on pad (LOP) or of a load between pads (LBP) configuration [2]. Waldemar (2005) [3], analyzes the variations of the stiffness and damping characteristics for the tilting pad journal bearings with the frequency of excitation and describes the analytical and experimental techniques used to evaluate these properties. The dynamic characteristics of tilting 12-pads journal bearing were also studied by Olszewsski and etal (2003) [4], this study led to the following conclusions: the number of tilting-pads can be basically 3 to 5 depending on the required operating parameters of rotating machine. Large number of pads allowed good cooling of bearing and the load capacity increases with the increase of relative clearance of pad. Stanislaw and Zygmunt (2003) [5], the paper introduces the results of calculation of dynamic coefficients of the tilting pads journal bearing characterized by asymmetric support of pads at different temperature of supplied oil as well as at different thermal conditions between the pads bearing.

This paper analyzes the variations of the load carrying capacity, dynamic coefficients (stiffness and damping coefficients) and elastic deformation with thermal effect for the tilting 5-pad journal bearings having cylindrical pivot, also study the effects of the preload factor, pivot offset and the load directions on the pads.

Tilting Pad Bearing Geometry

Figure (1) show the configurations for tilting pad bearing geometry with of five shoes, configuration (a) show load between pad (LBP) and (b) show load on pad (LOP), Each pad is able to rotate about a pivot thus attaining its own equilibrium position, with a strongly converging film region for each loaded pad. Tilting pad bearing have no cross-coupled stiffness for certain load conditions and offer an operation free of sub synchronous whirl [2]. Table (1) shows the tilting pads bearing details using in this paper.

Total Oil Film Thickness with Cylindrical Pivot

A single pad of a TPJB with a cylindrical pivot can be illustrated as shown in Figure (2), pad is depicted before (dotted line) and after (solid line) tilting it through an angle (δ).

$$h_{total} = h_{oil} + h_{r.d} \qquad .....(1)$$

Where: $h_{r.d}$ is the oil film thickness due to radial displacement.

This pad is located at an angular position (θp) relative to a stationary coordinate system with origin (O_b),(q_c) represents the angle to the line-of-centers which connects (O_b) to (O_j). In the TPJB case defines (θ) with respect to (O_b)andh_{oil} a stationary coordinate system, with the counter-clockwise direction being positive. the tilting pad journal bearing (TP JB) film thickness expression,[6].

$$h_{oil} = O_b N - O_b J \qquad ..... (2)$$

$$O_b N = R_p - (R_p - R)\cos(q - q_p) - d\,(R_p + t_p)\sin(q - q_p) \qquad ..... (3)$$

$$O_b J = r - e.\cos(q_c - q) \qquad ..... (4)$$

$$h_1 = C_p - C_p.PF\,.\cos(q - q_p)$$

$$h_2 = (r + t_p).d.\sin(q - q_p) + e.\cos(q_c - q)$$

$$h_{oil} = h_1 + h_2 \qquad .....(5)$$

Equilibrium Journal Position

The pressure distribution must first be obtained by numerically solving the Reynolds' equation (dynamically loaded equation). This pressure must then be used to balance the forces on the journal and the moments on each pad in the TPJB so that the equilibrium position of the journal can be determined.

Pressure Distribution

Reynolds' equation (6) can be solved over the domain of each pad surface in the TPJB to give the pressure distribution. A typical meshed pad surface with essential pressure boundary conditions.

The finite element method will be chosen for simplicity. In this method, basis functions are used to approximate the exact solution to a differential equation, [7].

$$\frac{\partial}{\partial x}\left(\frac{h^3}{m}\frac{\partial P}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{m}\frac{\partial P}{\partial z}\right) = 6u_j\frac{\partial h}{\partial x} + 12\frac{\partial h}{\partial t}$$

….. (6)

$$\iint_{element} W\left[\frac{\partial}{\partial x}\left(\frac{h^3}{m}\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{m}\frac{\partial P}{\partial z}\right)-6u_j\frac{\partial h}{\partial x}-12\frac{\partial h}{\partial t}\right]dA=0$$

….. (7)

Transient part of the last term in equation (6) can now be expressed as:

$$\frac{\partial h}{\partial t}=-\left[\dot{X}_J\ Cosq+\dot{Y}_J\ Sinq+\dot{d}\,(R_p+t_p)\ Sin(q-q_p)\right]$$

….. (8)

Boundary conditions of Reynolds equation,

- at the leading edge, P=*0*
- at the trailing edge, P=*0*
- When $Z=\pm\frac{L}{2}$ at the sides of the bearing, P=*0*

Forces and Moments

The pressure distribution can be integrated over an infinitesimal pad area to give a force which can be translated into a moment. For static equilibrium to be achieved, these moments must be equal to zero,[8].

$$Moment_i=\int_{-L/2}^{L/2}\int_{X_s}^{X_e}(r+t_p)(q-q_p)PCos(q-q_p)\,dxdz=0,$$
$$i=1,2,\ldots,N_{pads}$$

….(9)

Where: $X_s=r.q_s$ and $X_e=r.q_e$

$$\sum F_X=\sum_{i=1}^{N_{pads}}\left[\int_{-\frac{L}{2}}^{\frac{L}{2}}\int_{X_s}^{X_e}(P.\cos q)\,dx\,dz\right]=0$$

……(10a)

$$\sum F_Y=\sum_{i=1}^{N_{pads}}\left[\int_{-\frac{L}{2}}^{\frac{L}{2}}\int_{X_s}^{X_e}(P.\sin q)\,dx\,dz\right]-W_b=0$$

……(10b)

Parameters Controlling the Elastic Deformation and Dynamic Coefficients

(i). Pivot Offset (a)

$$a=\frac{q_p}{b}$$ ….. (11)

Where:

q_p : is the pivot angle measured from the pad leading edge.

b : is the pad extent angle.

When the pivot offset (a) is equal 0.5 the pads are centrally pivoted. Offset pivot bearings with (a) large than 0.5 are used in rotating machinery which will not run in reverse. The offset pivot ensures a converging wedge foe all conditions,[7].

(ii) Bearing Preload Factor (PF)

Preload is often used to adjust bearing coefficients in order to obtain specific rotor response characteristics. The preload can vary from 0 to 1, with each pad having different radii and hence different preload [7].

$$PF = \frac{C_p - C_r}{C_p} \quad ..... (12)$$

(iii) External Loading Condition

The direction of loading also affects the static and dynamic force characteristics of a tilting pad bearing. The load can either be directed towards on a pad (LOP), or directed between the pads (LBP). The (LOP) configuration is chosen for lightly loaded bearings operating at high speed cases, whereas the (LBP) configuration is chosen for heavily loaded bearings [8].

(vi). Pivot Flexibility

The contact between a bearing pad and the supporting pivot occurs through a small contact area which can be easily susceptible to wear. Most tilting pad pivots show non-linear stiffness with a strong dependency on the applied load and must be evaluated by Hertzian contact models, [2].

$$d_{pivot} = \frac{F_{pivot}}{K_{pivot}} \quad ..... (13)$$

Where: K_{pivot} = pivot stiffness,
d_{pivot} =pivot deflection
F_{pivot} =component of the applied load passing through the pivot

Thermal Distribution over Pad

$$T_{(i+1)} = \left[1 - \Delta q\left(\frac{2H}{r_1 C_1 wh}\right)\right] T_{(i)} + \Delta q\left[\frac{2HT_{amb}}{r_1 C_1 wh} + \frac{2wmr^2}{r_1 C_1 h^2}\right]_{(i)} \quad ..... (14)$$

Equation (4) can now be solved by specifying an inlet temperature at the leading edge of each tilted pad. The inlet temperature (T_{in}) at one of the pads is taken to be the sum of the lubricant supply temperature and the estimated average temperature rise (ΔT_{av}) [9].

..... (15)

$$H = 25.5\, u_j^{0.7}\, m_o^{-0.2} (r \Delta_p)^{-0.4} \quad .....(16)$$

Elastic and Thermal Deformation of Pads

The deformation of elastic structures is described by means of the constitutive relation and equilibrium equation. The constitutive equation for a linearly elastic material is given as, [10].

$$\underline{\underline{s}} = \underline{\underline{C}}\,\underline{\underline{e}} + \underline{\underline{s}}_0 \quad ..... (17)$$

The equilibrium equation for a stationary elastic media is given by the following:

$$\underline{\nabla}.\underline{\underline{s}} + \underline{b} = \underline{f} \quad ..... (18)$$

Where:
$\underline{\underline{s}}$, $\underline{\underline{e}}$ the stress tensor and strain tensor respectively.
$\underline{\underline{s}}_0$ The residual stress tensor (typically due to thermal effects).
b, f the body force vector and load vector respectively.

The pad deformation is composed as the linear superposition of elastic deflection due to the hydrodynamic pressure and thermal expansions due to the temperature gradients. The thermal distribution from solution to pad conduction equation is applied to obtain thermal expansion, while the oil pressure is integrated to obtain forces; these forces are resolved into components in the X and Y directions and applied on the pad surface to obtain mean deformations. Sommerfeld number is widely used as a characteristic number for journal – bearing equation (19) which relates the operating variables, viscosity, speed and bearing clearance , [8].

Evaluation of the Dynamic Coefficients

Tilting pad bearing stiffness and damping coefficients are obtained after bearing static equilibrium is achieved. The pivot and pad tilt degrees of freedom should be included is determining the dynamic force coefficients. The stiffness coefficients are given by perturbing the journal position (X and Y), pad tilt angle (δ) and pivot deformation equation (14). The changes in forces/ moments are obtained for these perturbations. The elastic deformation is represented as a change in the pad radius (R_p). The relationship for the stiffness coefficients is, [10].

$$K_{ij}^{(eq)} = \frac{K_{ij}\, K_z\,(K_{ij}+K_z) - w^2\, K_z\, C_{ij}^2}{(K_{ij}+K_z)^2 + w^2\, C_{ij}^2} \quad \text{.....(19)}$$

$$C_{ij}^{(eq)} = \frac{K_z^2\, C_{ij}}{(K_{ij}+K_z)^2 + w^2\, C_{ij}^2} \quad \text{..... (20)}$$

$$K_z = m_p w^2 + K_{pivot}$$

Results and Discussions

The results of this paper are illustrated below. Figure (3) shows how does the oil film thickness change with rotating angle (θ)for each pads surface, these variables values of oil film thickness depend on the position of the pad during the rotating operation, pad No.4 has maximum oil film thickness (hmax. =3.17 mm) and pad No.2 has minimum oil film thickness (hmin.=0.62 mm) these values are in the allowable range (by comparison these profile curves in Figure (3) with [12] for tilting 3-pads journal bearing, it is evident that both gave the same profile for oil film thickness but at different values).

Figure (4) shows the variations of generated pressure in the oil film thickness with rotating angle for each pad at load between pad and preload factor equal zero. From this figure it can be noted that, when there is an increases in the pivot offset (α) there is an increase in generated pressure. The position of maximum generated pressure over pad surface depends on the pivot offset (α) value.

Figures (5) and (6) shows the relationship between nondimensional Sommerfeld number and nondimensional dynamic bearing coefficients when the preload factor change from 0 to 0.5 values respectively for two types configurations load on pad and load between pads, when preload factor equal zero its more reasonable comparing with preload factor equal 0.5, because (Kxx) and (Cxx) are responsible about the stability of tilting pad bearing at high speed. The cross-coupled coefficients Kxy, Kyx, Cxy and Cyx are usually close to zero and negligible compared with the other linear coefficients, [7].

The Figures (7) and (8) illustrated the elastic deformation with thermal effect over surface of pad shell No.1 for load on pad configuration; at pivot offset are equal 0.6 and 0.5 respectively. It was found that the maximum elastic deformation in the pad surface occurs in the center of pivot for each figures are about 0.009125 mm and 0.008429 mm respectively. When comparison between these figures, figure (7) shows graded bedding of colors, that means the effect of elastic deformation is continuous for more area of pad surface.

Conclusions

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

1- The maximum and minimum of the oil film thickness value depends on the type of bearing pivot and the angular position of pad.

2-When the pad preload factor is increase, the dynamic stiffness and damping coefficients will increase also; this result gives the best performance for bearing.

3-The bearing load carrying capacity increases when the pivot offset (α) increase, because when the pivot offset equal 0.5 this means the load carrying distributed uniformly over the pad surface.

4-When pivot offset increases that mean the elastic deformation also increases (because the elastic deformation depends on the value of generated pressure).

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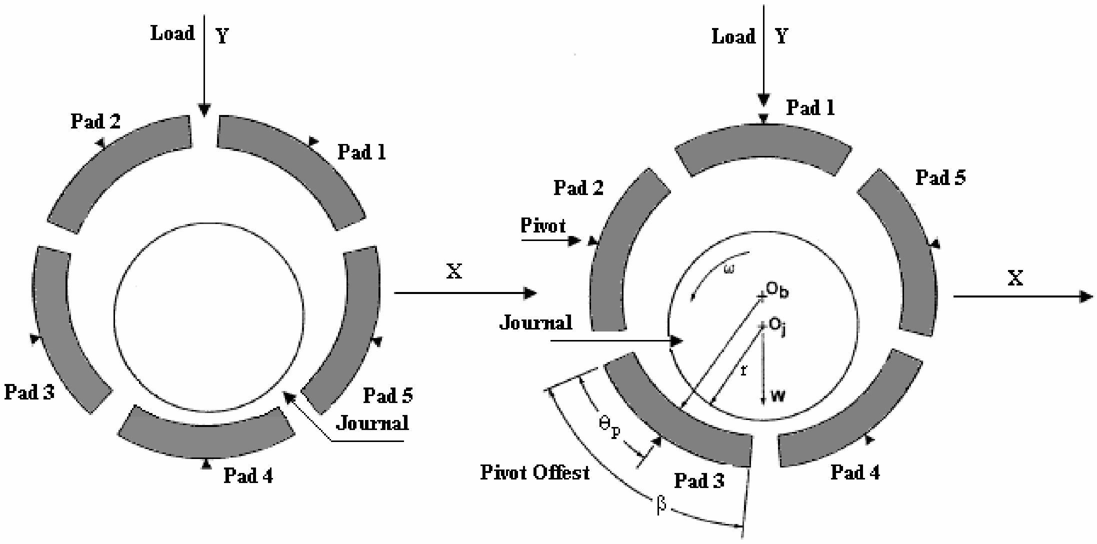
Table (1) Tilting Pads Bearing Details

Parameter	Value
Number of pads	5
Radius of bearing	5.4035 cm
Radius of journal	5.35 cm
Pad thickness	1.21 cm
Angular dimension of pad	62^o
Leading edge of each pads (LOP)	$59^o,131^o,203^o,275^o,347^o$
Leading edge of each pads (LBP)	$23^o,95^o,167^o,239^o,311^o$
Pad tilt angle	0.2^o
Fractional angular position of pivot	0.5
Preload factor	0.45
Bearing load	4 kN to 4.5 kN
Lubricant viscosity	0.04 N/m^2.sec.
Pad Configuration	LOP and LBP
Length diameter ratio	0.55
Journal speed	9000 r.p.m



(a) (b)

Figure (1) Configurations for Tilting 5-Pad Bearings
(a) Load Between Pad (LBP) (b) Load on Pad (LOP)

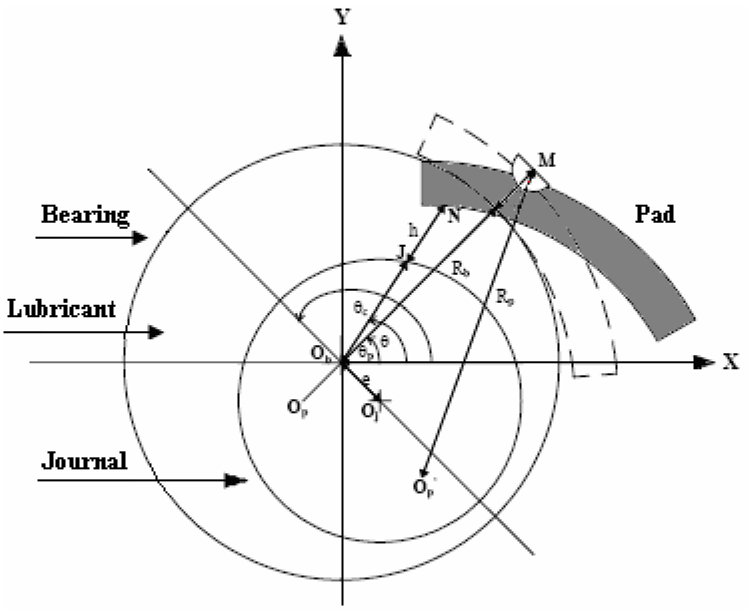


Figure (2) Tilting Pad Bearing with Cylindrical Pivot, [7].

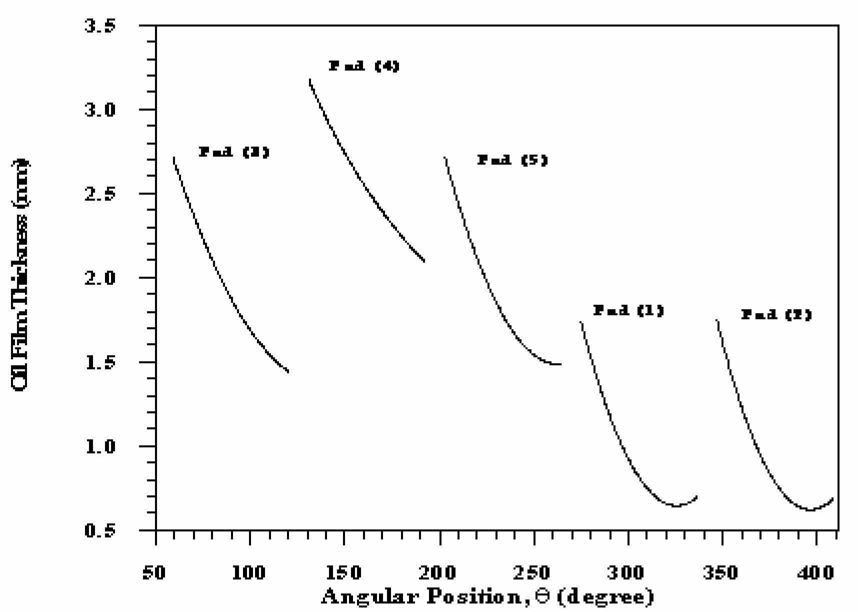


Figure (3) Oil Film Thickness Variation with Angular Position over 5-Pad

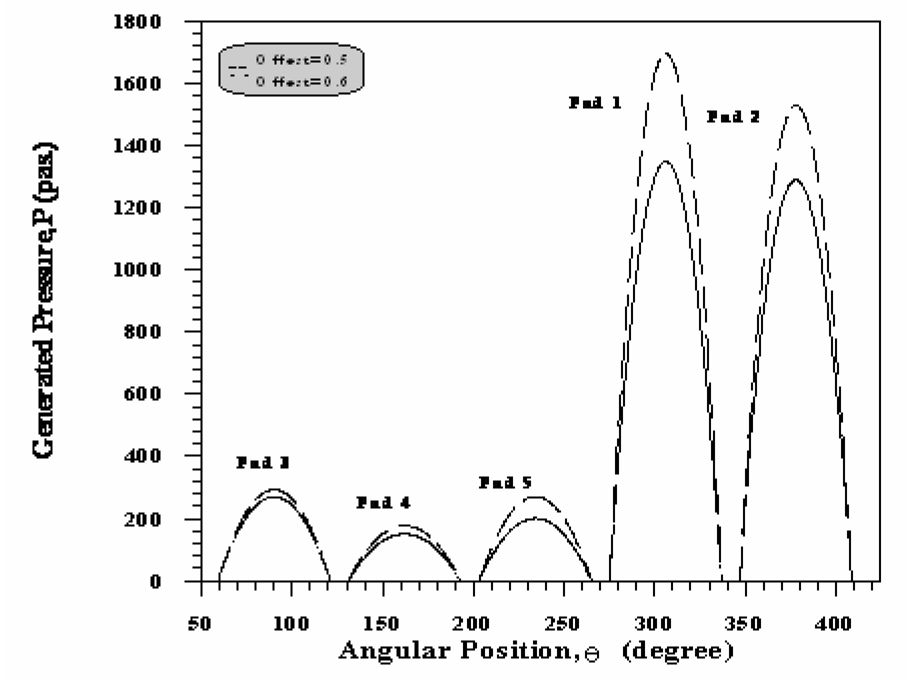


Figure (4) Pivot Offset Effect on the Distribution Pressure at Load Between Pad (LBP) and PF =0

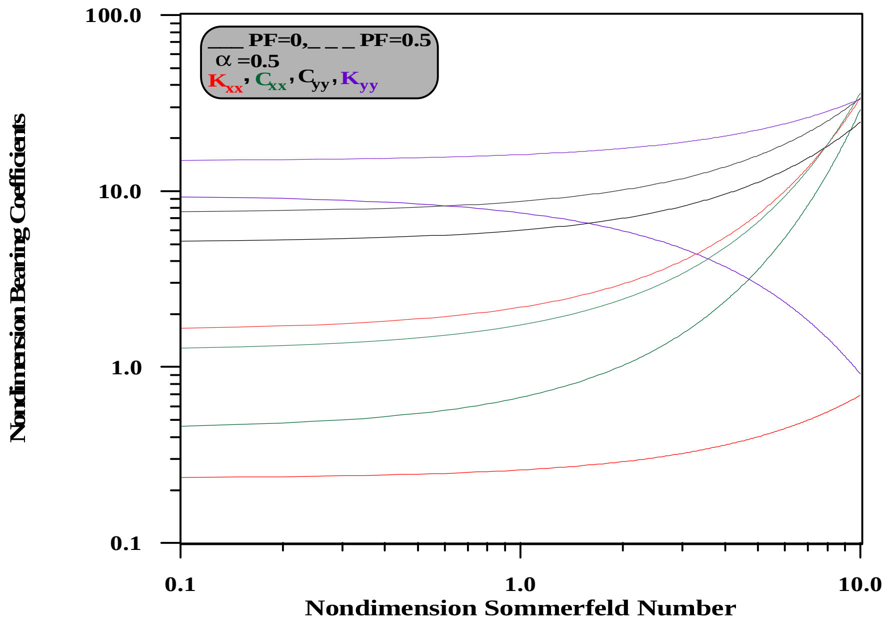


Figure (5) The Preload Factor Effect on The Bearing Dynamic Coefficients

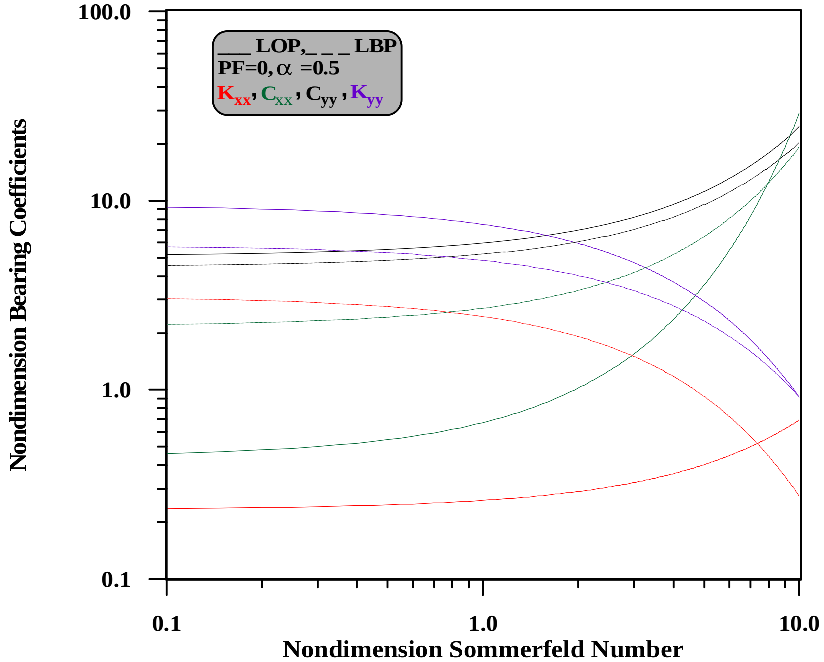


Figure (6) The Pad Loading Position Effect on The Bearing Dynamic Coefficients

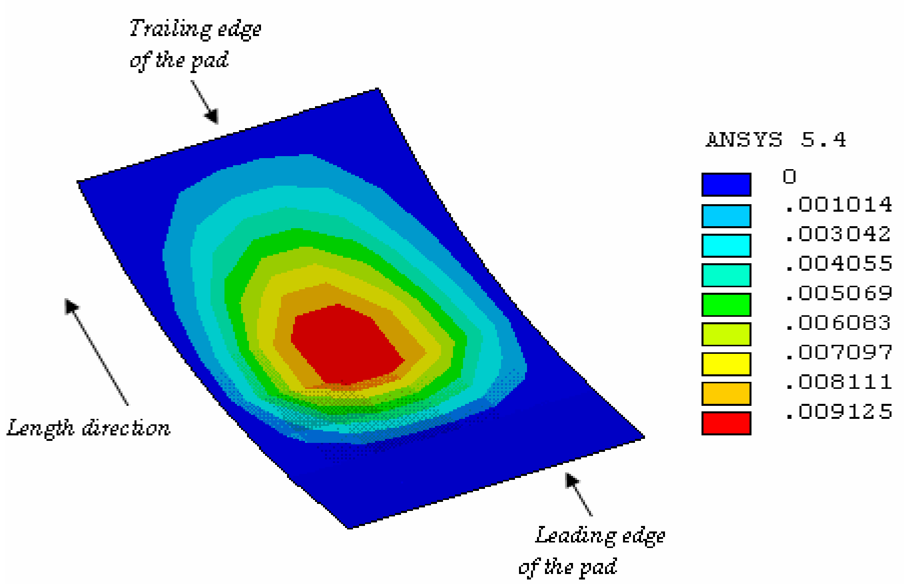


Figure (7) Elastic Deformation (mm) with Thermal Effect Over Pad No.1 (LOP, PF=0 , a=0.6)

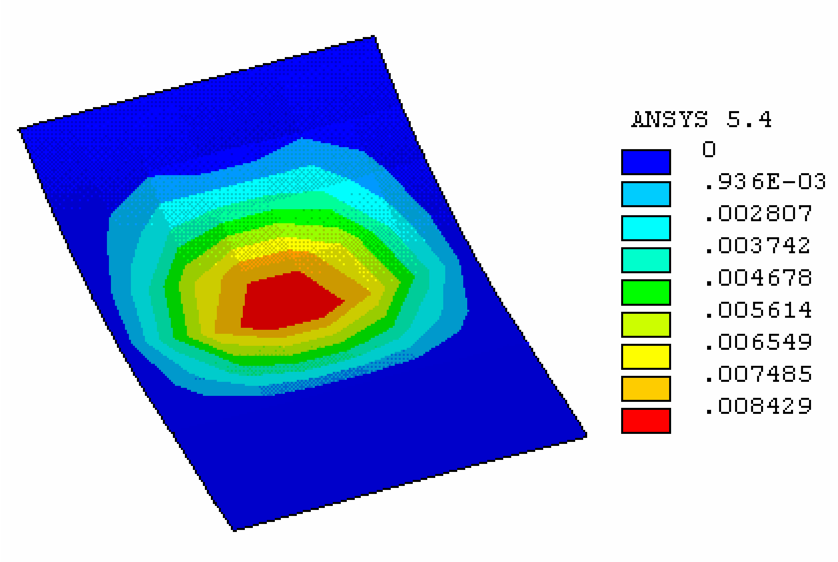


Figure (8) Elastic Deformation (mm) with Thermal Effect Over Pad No.1 (LOP, PF=0 , a=0.05)