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The Effect of Elastic Deformation of Bearing Surface on The Steady State Performance of Offset Halves Bearing

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Abstract

In this research a theoretical study was presented to find the performance of the offset halves bearing which used in applications of high velocities ,like centrifugal machines, dentist tools and high precession machines.

In studying the elastic deformation, three types of alloys have been used for coating the bearing surface. These alloys Cu-alloy, Al-alloy and Mg-alloy. And these show that the bearing material has a considerable effect on the essential bearing characteristics. The bearing made of soft material has lower performance than that made of harder material.

In this work the bearing performance was investigated under the following conditions :
1-Elastohydrodynamic lubrication at a rotational velocity of 2000rpm and L/D ratios Of(0.5.)and (1)
2-Elastohydrodynamic lubrication at a rotational velocity of 15000rpm and L/D ratios of 0.5and (1).
The results of analysis show that the offset bearing deformation (elastic deformation) has a considerable influence on the performance at a high eccentricity values (n>0.9) i.e. under high loa

Notations

Symbol	Definition	Unit
B-	constant	-
D-	diameter of bearing	m
e-	eccentricity	m
E-	modulus of elasticity	N/m
h-	film thickness of oil	m
L-	length of the bearing	m
P-	nodal pressure	N/m
m	no of grids circumferentially	
k	no of grids axially	
R-	bearing radius	m
r-	journal radius	m
s-	offset value	m
t-	thickness of bearing	m
Δt-	elastic deformation	m
θ-	angle of any point on the bearing surface that counted at mid plain of the journal	degree degree
ϕ -	attitude angle of any point on the bearing surface that counted	
ϵ -	strain	-
σ-	stress	N/m
μ-	oil dynamic viscosity	N.sec/m

Abbreviations

EHD- elasto hydrodynamic contact
EHL- elasto hydrodynamic lubrication
HDL- hydrodynamic lubrication
Subscripts
x- circumferential direction
z- axial direction
i,j location of any node in bearing surface

Introduction

The first step of the theoretical approach is the solution of Reynolds equation numerically using finite difference method. The hydrodynamic pressure will be predicted for each eccentricity ratio. This hydrodynamic pressure; is used to find the elastic deformation of the inner bearing surface. The hydrodynamic pressure means stress in physical meaning and according to the Young's modulus,

$$E = \frac{\sigma}{\varepsilon} \qquad (1)$$

Where Young's modulus is constant for each material and the strain is

$$\varepsilon = \frac{\Delta t}{t} \qquad (2)$$

The offset halves bearing (shown on (fig.1) is one of the non-circular type .This type is suitable for the high speed requirements .These requirements are high stiffness, damping and stability.The bearing thickness is available, so that the change in bearing thickness can be obtained by substituting equation(2) in (1) ,this yields;

$$\Delta t = \frac{p.t}{E} \quad (3)$$

Which was the deformation of the inner bearing surface. By adding this value to the oil film thickness relationship, a new elastohydrodynamic pressure will be predicted as a result of the change of the film thickness. Recalculating procedure of the elastic deformation should be repeated many times until the solution is obtained. In this work , three different materials are taken for comparison with each other .These materials are Copper, Aluminum ,and Magnesium alloys. Four cases were studied , two of them are for the normal speeds and L/D ratio equals (0.5)and(1) and the other two cases are for high speeds and L/D ratio equals (0.5)and (1). These four cases carried out under two different conditions .These different conditions are HDL and EHL.

Literature Review

Oh and Geonka, 1985 [1] studied the EHL solution of the journal bearing .Cavitations boundary conditions were used . And the method was applied to the lubrication of an automotive connecting rod bearing .
Houpert and Hamrock, 1986 [2] presented a fast approach for calculating film thickness and pressure in EHL contacts at high loaded bearing.
Geoneka and Oh 1986 [3] presented an optimum short bearing theory for the EHD solution of journal bearings. An approximate .
method for solving the EHL problem has been developed.
Hou, Zhu and Wen,1987 [4] studied an inverse solution to the point contact EHL problem under heavy loads .
Chang, 1989 [5] presented a new method of an efficient and accurate formulation of the surface deformation matrix in EHD point contacts. Durany,Garcia and Vazquez,1996[6] studied the numerical computation of free boundary problems in EHL. Hoglund,1999 [7] studied the influence of lubricant properties on EHL. The influence of pressure and temperature on viscosity. Goodyer,Fairlie,Berzins and Scales ,2001 [8] studied an adaptive mesh method for EHL. The solution of EHL problems is both computationally intensive and industrially relevant. Udofia and Jin ,2003 [9] analysised the EHL for a typical metal-on –metal hip-resurfacing prosthesis under a simple steady state rotation. Both numerically analyzed using the finite difference method. Krupka,Hartl and Liska,2006 [10] studied a thin lubricating film behavior at very high contact pressure .Thin film colorimetric interferometer has been used to examine the behavior of the EHD lubricant films.

Numerical Solution

A numerical solution was used to analyze the problem and as follows;

Boundary Conditions

The appropriate boundary conditions for the offset halves bearing working under steady state , incompressible fluid are :-
a-The pressure at the edges is assumed to be zero (ambient),i ,e.

p=0 at z=±L/2
b- p=0 at θ= θc

$$\left(\frac{\partial p}{\partial \theta}\right) = 0 \quad \text{at } \theta = \theta c$$

Where **θc** is the rupture angle of film pressure
Reynolds equation of two dimensions form was used then numerically solved using afinite difference technique. The finite difference technique that used here is five nodes scheme. Reynolds equation of two dimensions form was used and solved numerically using a finite difference technique.The finite difference technique that used hereis the type of five nodes scheme. Reynolds equation of two dimensions form is.

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right) = 6u\frac{\partial h}{\partial x} \quad (4)$$

This equation is expanded and then the oil film thickness equation is substituted. The oil film thickness equations are:-

$$h=\left\{R^2-\left[\left(S+\frac{e\sin\theta}{\sin(\phi+\theta)}\right).\sin(\phi+\theta)\right]^2\right\}^{1/2}+\left[S+\frac{e\sin\theta}{\sin(\phi+\theta)}\right].\cos(\phi+\theta)+e\cos\theta-\frac{e\sin\theta}{\tan(\phi+\theta)}-r \quad .... \quad (5)$$

This equation is applicable for

$$-\phi\le\theta\le(\pi-\phi)$$

$$h=\left\{R^2-\left[\left(-S+\frac{e\sin\theta}{\sin(\phi+\theta)}\right).\sin(\phi+\theta)\right]^2\right\}^{1/2}+\left[-S+\frac{e\sin\theta}{\sin(\phi+\theta)}\right].\cos(\phi+\theta)+e\cos\theta-\frac{e\sin\theta}{\tan(\phi+\theta)}-r \quad .... \quad (6)$$

This equation is applicable for

$$(\pi-\phi)\le\theta\le(2\pi-\phi)$$

Then by dividing the bearing surface into small elements using circumferential and axial grids, this is to allow the finite difference technique to be used and the pressure gradients to be calculated as follow:

$$\left(\frac{\partial p}{\partial x}\right)_{i,j}=\frac{p_{i+1,j}-p_{i-1,j}}{\left(2\frac{\pi R}{m-1}\right)} \quad (7)$$

$$\left(\frac{\partial^2 p}{\partial x^2}\right)_{i,j}=\frac{p_{i+1,j}-2p_{i,j}+p_{i-1,j}}{\left(\frac{\pi R}{m-1}\right)^2} \quad (8)$$

$$\left(\frac{\partial p}{\partial z}\right)_{i,j}=\frac{p_{i,j+1}-p_{i,j-1}}{\left(2\frac{L}{k-1}\right)} \quad (9)$$

$$\left(\frac{\partial^2 p}{\partial z^2}\right)_{i,j}=\frac{p_{i,j+1}-2p_{i,j}+p_{i,j-1}}{\left(\frac{L}{k-1}\right)^2} \quad (10)$$

Then using the parameters B1……B5 in order to eliminate the long coefficients of the pressure at the five neighbors nodes, then Reynolds equation becomes as follow:

$$p_{i,j}+B_1p_{i+1,j}+B_2p_{i-1,j}+B_3p_{i,j+1}+B_4p_{i,j-1}=B_5 \quad (11)$$

And this equation is solved numerically by applying the pressure boundaries mentioned previously in this research. And the results is obtained.

Results:

Effect of film thickness behavior in EHL

Figures (2, 3 and 4) show a comparison between the film thickness behavior at the mid plain of the EHL analysis at eccentricity ratio (0.9) for L/D ratios range (o.5-1) .It can be seen that the effect of elastic deformation of the bearing surface is clear on the minimum film thickness (i ,e. increases minimum film thickness). This effect increases

With the ratio of L/D from (0.5) to (1). The maximum increase in film thickness occurs for magnesium alloy at angle 108 degree, as shown in figure (4). The behavior above is due to the high pressure generation in the film thickness at high velocities which causes high elastic deformation in the bearing surface and high increases in the film thickness for the same eccentricity ratio. That means for soft material.

(low value of modulus of elasticity) the film thickness reaches to maximum value more than the harder material(high value of modulus of elasticity) because of the ability of deformation in the softer one .

Effect of pressure distribution on EHL for Cu,AL and Mg alloys

The circumferential pressure distribution at the mid plain of the offset bearing of the HDL and EHL analysis under the same working conditions of the film thickness is plotted in figures (5 and 6), with each velocity and L/D ratio for Cu , AL and Mg .These figures show that the elastic deformation of the bearing surfaces has a sudden effects on the pressure generation, especially at high value of velocities and L/D ratio ,for softer material (low modulus of elasticity)(i ,e.Mg alloy), and accompanied of this effect, maximum decrease in EHL against the HDL in pressure distribution is occurs. In HDL case the pressure increases too much more than the pressure in EHL case; because the effect of deformation is negligible. In EHL case there is a pressure decrease , because the elastic deformation which causes an increases in film thickness. And the later causes a pressure decreases and a low load carrying capacity.

Effect of (EHL) on load carrying capacity, oil flow rate and friction force ratio

Figs.(7,8 and 9) shows that (EHL) hardly effect the load carrying capacity, oil flow rate and friction force ratio. This is because the change in oil film thickness due to elastic deformation is very small. Thus it is not worth to consider (EHL) effect in low load application(i.e. small size bearing)

Conclusions

The following conclusions can be withdrawn:

1-As the bearing is softer the elastic deformation is larger .And as a result the maximum achievable pressure is lower .This justifies the use of hard alloy for coating a highly loaded bearing such as Cu alloy.

2- EHL phenomenon has a bad influence on the bearing characteristics ,e.g .load carrying capacity

3- It is not worth, that to consider the (EHL) effect in the case of low load application (i.e. small bearing).

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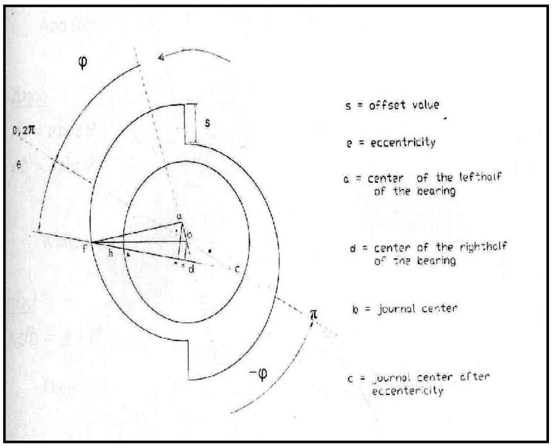

Fig (1) Geometry of offset halves bearing

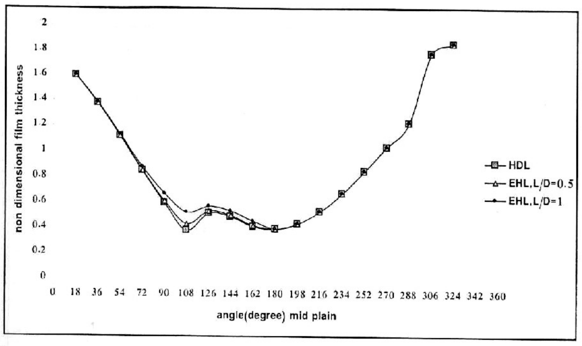

FIG. (2) CIRCUMFRENTIAL FILM THICKNESS DISTRIBUTION AT (VELOCITY=15000, L/D=0.5-1, n=0.9) FOR CU

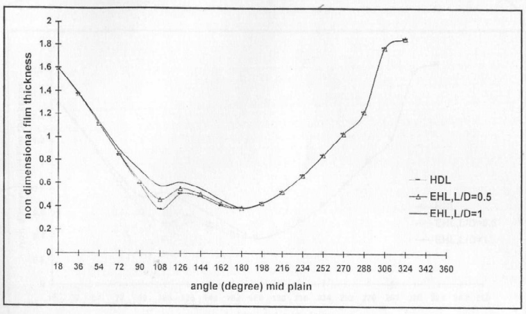

FIG. (3) CIRCUMFRENTIAL FILM THICKNESS DISTRIBUTION AT (VELOCITY=15000, L/D=0.5-1, n=0.9) FOR AL

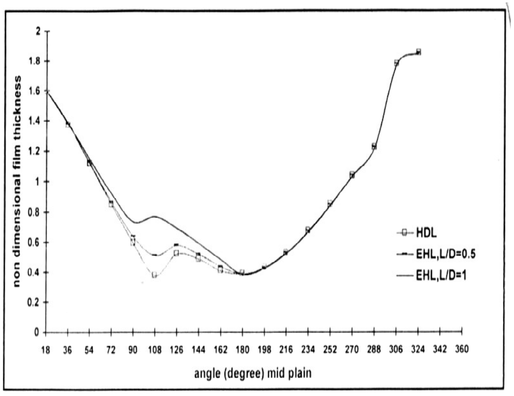

FIG. (4) CIRCUMFRENTIAL FILM THICKNESS DISTRIBUTION AT (VELOCITY=15000, L/D=0.5-1, n=0.9) FOR Mg

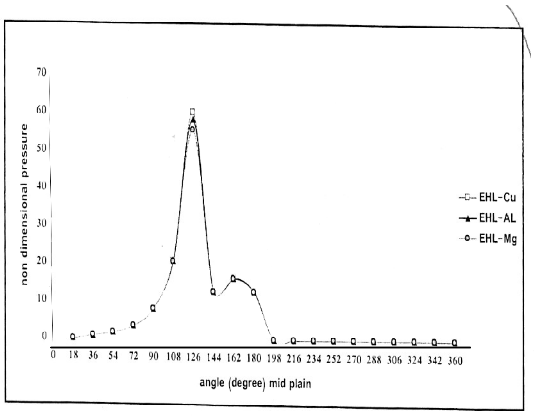

FIG (5) CIRCUMFRENTIAL PRESSURE DISTRIBUTION AT (VELOCITY=2000, L/D=0.5, n=0.9) FOR CU, AL, Mg

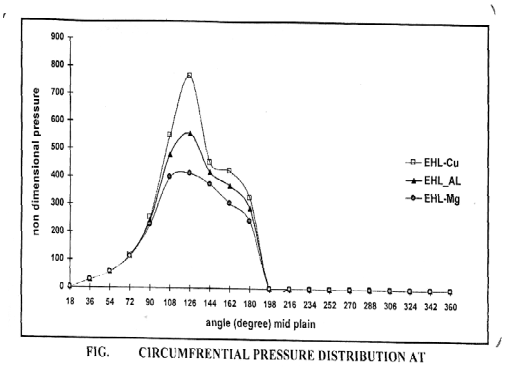

FIG. CIRCUMFRENTIAL PRESSURE DISTRIBUTION AT
(VELOCITY=15000, L/D=1, n=0.9) FOR CU, AL, Mg

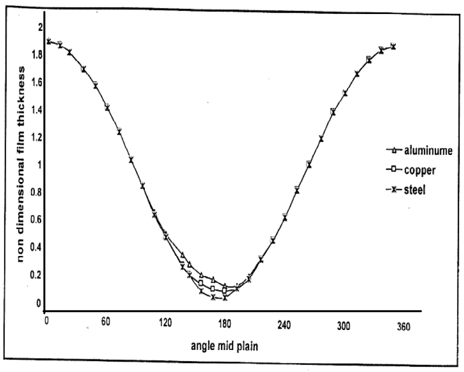

FIG. LOAD CAPACITY V_S ECCENTRICITY RATIO,

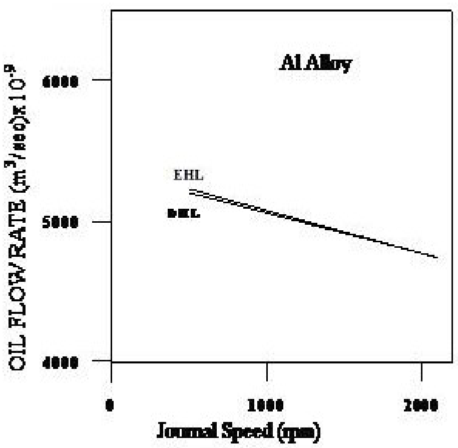

FIG.(8) OIL FLOW RATE Vs JOURNAL SPEED

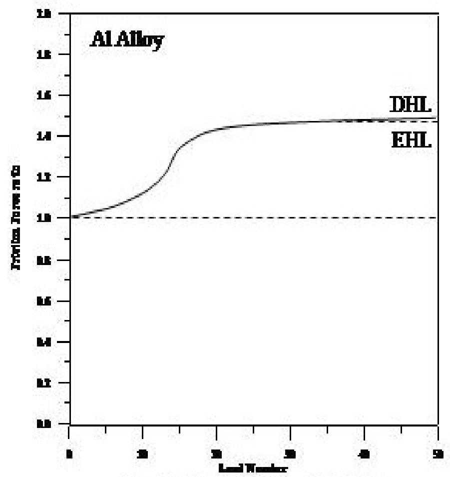

Fig (9) Friction Force Ratio Vs Load Number

تأثير الانفعال المرن لسطح المسند ذوالانصاف الزاحفة على اداءه في حالة الاستقرار

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الخلاصة

وقد تمت دراسة التشوه المرن لثلاث سبائك مختلفة وهذه السبائك هي سبائك النحاس ،سبائك الألمنيوم وسبائك المغنيسيوم ومن ثم مقارنة أداء المسند في الحالات التالية
هذه الدراسة المقارنة لأستخدم السبائك الثلاث قد تمت تحت الظروف ألتاليه لكل سبيكة:
العالية (١٥٠٠٠دورة /دقيقة) مقارنة بالسرع الواطئة(٢٠٠٠دورة بالدقيقة). كما إن لنوع المعدن تأثير على سمات المسند،حيث إن المعدن العالي المتانة يتحمل ظروف عمل أقسى من تلك التي يتحملها المعدن الأقل متانة.
١-دراسة تأثير التشوه المرن في سطح المسند عند سرعة مقدارها ٢٠٠٠دورة /دقيقة ونسبتي طول إلى قطر مقدارهما ٥,٠و ١..
٢- دراسة تأثير التشوه المرن في سطح المسند عند سرعة مقدارها ١٥٠٠٠ دورة/دقيقة ونسبتي طول إلى قطر مقدارهما٥,٠و ١.
لقد أثبتت نتائج هذا البحث بأن التشوه المرن لسطح المسند ذو الانصاف الزاحفة له دور كبير على أداء المسند فقط عند قيم (n>0.9) بتعبير اخر تحت الاحمال الكبيرة.