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Viscosity Effect on Stiffness of Non-conventional (Five Tilted Pads) Journal Bearing

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A B S T R A C T

In this tribological study, we highlight the effect of lubricating oil viscosity in the Multi-pads hydrodynamic journal bearings generate important improvement in characteristics of stiffness and stability in the high speed turbomachines. Depending on viscosity of oil film (three values) variation for five tilted pads bearing, each pad is pivoted and is facilitated to be tilted with small angles, by using Matlab program, we calculate the oil film thickness for convergence layer. We applied Reynold's equation and solved it's numerically by using finite difference method with 5 nodes technique to find the pressure distributed on each node in the mesh of tilted pad, then calculate stiffness coefficients. Results show that there is clear effect on stiffens with viscosity change. The increase in value of *Krr* (for *n* = 0.3) between viscosity (0.04 Pas. s) and viscosity (0.058 Pas. s) is14.33 MN/m, while the increase in *Krr* value between viscosity (0.058 Pas. s) and viscosity (0.087 Pas. s) is 11.37 MN/m. the increase in value the of *Kss* (for *n* = 0.3) between viscosity (0.04 Pas. s) and viscosity (0.058 Pas. s) is5.921 MN/m, while increase in Kss value between viscosity (0.058 Pas. s) and viscosity (0.087 Pas. s) is9.55 MN/m respectively. the increase in value of *Ksr* (for n = 0.3) between viscosity (0.04 Pas. s) and viscosity (0.058 Pas. s) is 8.95 MN/m, while the increase in *Ksr* value between viscosity (0.058 Pas. s) and viscosity (0.087 Pas. s) is 14.41 MN/m respectively. the increase in value of *Krs* (for *n* = 0.3) between viscosity (0.04 Pas. s) and viscosity (0.058 Pas. s) are 5.08 MN/m, while the increase in *Krs* value between viscosity (0.058 Pas. s) and viscosity (0.087 Pas. s) is8.19 MN/m respectively. The values of the dominate principal coefficients *Krr* is greater than that of *Ksr*, also The values of the principal coefficients *Kss* is greater than that of cross coupling *Krs* for all values of viscosity that studied. From this result, we can conclude the side effect of cross coupling coefficients (*Ksr* ,*Krs*) can be overcome by great values for principal coefficient (Krr, Kss) respectively, so we can get good improvement instability for this bearing by variation the viscosity. After that, we regarded to use high viscosity lubricant in multi-pad journal bearing to improve the performance and stability by controlling the stiffness coefficients.

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تأثير لزوجة الزيت على معامل نابضية المساند الغير تقليدية (ذو خمس وسادات قابلة للإمالة)

الخلاصة

تم في هذه الدراسة الترايبولوجية البحث في تأثير لزوجة المزيت المستخدم في المساند الهيدروديناميكية ذات الخمس وسادات القابلة للأمالة والذي سبب تحسن مهم في خصائص نابضيه واستقراريه مساند محركات السرع العالية. بالاعتماد على تغير لزوجة شريحة الزيت (تم أخذ ثلاث قيم) للمسند ذو الخمس وسادات، وحيث أن كل وسادة متمحورة ومثبتة لتكون قابلة للأملة بزاوية صغيرة، وباستخدام برنامج الماتلاب، تم حساب سمك شريحة الزيت المحصورة. وبتطبيق معادلة رينولدز وحلها عددياً باستخدام طريقة الفروقات ذات الخمس عقد تم أيجاد الضغط الموزع في كل عقدة على شبكة سطح الوسادة، ليتم بعد ذلك حساب معاملات نابضية المسند. أظهرت النتائج تأثير واضح في معاملات النابضية عند تغير لزوجة المزيت. فعند نسبة لامركزية 0.3 للزوجة بين (0.04 Pas. s) و (0.058 Pas. s) كانت قيمة الزيادة في *Krr* هي 14.33 MN/m، بينما عند نسبة لامركزية 0.3 للزوجة بين (0.058 Pas. s) و (0.087 Pas. s) كانت قيمة الزيادة في *Krr* هي 11.37 MN/m. بينما عند نسبة لامركزية 0.3 للزوجة بين (0.04 Pas. s) و (0.058 Pas. s) كانت قيمة الزيادة في *Kss* هي 5.921 MN/m ، بينما عند نسبة لامركزية 0.3 للزوجة بين (0.058 Pas. s) و (0.087 Pas. s) كانت قيمة الزيادة في *Kss* هي 9.55MN/m على التوالي. في حالة نسبة لامركزية 0.3 للزوجة بين (0.04 Pas. s) و (0.058 Pas. s) كانت قيمة الزيادة في *Ksr* هي 8.95 MN/m ، بينما عند نسبة لامركزية 0.3 للزوجة بين (0.058 Pas. s) و (0.087 Pas. s) كانت قيمة الزيادة في *Kss* هي 8.19 MN/m على التوالي. قيم المعاملات المسيطرة الرئيسية لـ *Krr* كانت أكبر من *ksr*، كذلك فأن قيم المعاملات المسيطرة الرئيسية لـ *Kss* كانت أكبر من *Krs* لكل قيم اللزوجة التي تم دراستها. نستنتج من هذا أن التأثير السلبي لمعاملات الأزدواج المتقاطع لـ (*Ksr* ,*Krs*) قد تنتج من القيم العالية للمعاملات الرئيسية لـ (*Krr, Kss*) على التوالي وبهذا فأنه بالأمكان تحسين قيمة الأستقرارية لهذا المسند بتغيير لزوجة المزيت. وبهذا نوصي بأستخدام مزيتات ذات لزوجة عالية في المساند ذات الوسادات المتعددة لتحسين الأداء والأستقرارية من خلال التحكم بمعاملات النابضية للمسند.

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Nomenclature

a	element area on the pad, (m^2)
Ap	total area of pad, (m^2)
Cr	radial clearance, (m)
Ftr	radial component of w, (N)
Fts	tangential component of w, (N)
h	oil film thickness, (m)
Krr	principal stiffness coefficient in radial direction, (N/m)
Krs	cross coupling stiffness coefficient of radial force, (N/m)
Ksr	cross coupling stiffness coefficient of tangential force, (N/m)
Kss	principal stiffness coefficient in tangential direction, (N/m)
L	longitudinal length of pad, (m)
m, n	maximum number of nodes in x and z direction, respectively
n	eccentricity ratio
p	pressure in oil film, (Pa)
U	tangential velocity, (m/s)
W	hydrodynamic force, (N)
x	circumferential axis, (m)
z	axis normal on the paper, (m)

Greek symbols

δ	tilted angle, (deg.)
θ	circumferential angle, (deg.)
θp	pad angle, (deg.)
ϕ	attitude angle, (deg.)

1. INTRODUCTION

Hydrodynamic instability at high speed rotating operation with tighter clearances machines like compressors, turbines, pumps, electric motors and electric generators which are commonly supported in fluid film of simple design incorporated plane cylindrical bearings, represent one of serious and important challenges for industrial's manufacturer. With need to increasing the power output, the operating life and efficiency of typical turbomachines, tilting pads bearing devised to reduce potential amplitude of subsynchronous vibrations due to oil whirl or whip and offer an unsurpassed advantage to increasing rotor bearing critical speed cause each pad is able to rotate about a pivot thus attaining its own equilibrium position, with tighter convergence oil film layer for each load pad [1].

The viscosity of lubricant oil film offers a damping action to any external disturbance forces. This damping action is described by four coefficients or more. These coefficients (stiffness and damping coefficients) are commonly named dynamic coefficients or dynamic characteristics of the bearing which control the bearing stability. High values for the cross coupling stiffness coefficients are responsible for disturbing the bearing stability. This is because its displacement is perpendicular to the force direction and so it causes Journal disturbance, journal whirl and other consequences, which resonance of the bearing system early at low journal speed [2]. The bearing stiffness basically is described by four coefficients. Two of them are essential (principal) stiffness coefficients and the other two are secondary (cross coupling) stiffness coefficients [3]. The non-conventional bearing is used to overcome the instability bearing that appears at high speed in conventional by means variation of stiffness coefficients. From earliest searches that studied stiffness in non-conventional bearing which studied stiffness of three lubes journal bearing in 1978 [4]. While the dynamic characteristic (stiffness and damping) on adjustable multi-lubes was studied at 1980 [5]. Stiffness of elliptical journal bearing was studied at 2015 [6]. Tilting pad bearing gets an importance because of their stiffness coefficients that added good improvements to overcome vibration problems in high speed machine, this type of bearing researches interested on it's at 1997 [7]. Above searches deal with geometric shape effect on stiffness of journal bearing but it can be effected by operational parameters like viscosity and speed, from those searches that deals with viscosity and micropolar fluids effects on journal bearing stiffness that done at 1989 [8].

The non-conventional bearings have many Advantages like as: maximum possible stability of rotating parts, low sensitivity to load direction, oil flow can be minimized - it reduces losses caused by friction, spare parts (for these bearings are the pads). In this bearings, Preload describes relationship between the pad diameter, shaft diameter and bearing diameter. Bearings which have a positive preload, means the pad radius is larger than bearing radius. Standard pads are supplied with positive value preload ratio in the range of (0.3 – 0.55). The load capacity of this Journal Bearings depends on different factors such as oil viscosity, shaft speed, direction of load in reference to the pad position, inlet oil temperature etc. There are two possibilities of direction of load in reference to the pad position - load on pad and load between pads [9].

So, the viscosity effect on the stiffness coefficients for this type of non-conventional bearing did not studied in past, that cause us to search on this problem.

2. THEORETICAL ANALYSIS

2.1. Mathematical model and stiffness coefficients expression

In the mathematical model, the journal center moves in the two dimensional (X, Y) plane, the tilted pad journal bearing consists of multi pads fixed circumferentially on the bearing, each pad is pivoted in its leading edge and is adjust to be tilted around this pivot with small certain angles, in this search, these pads arranged about the shaft with pad angle (θp) of (36°) where the space between two preceding pads is (36°) , where only one pad is drawn in Fig (1) from five pads that distributed around journal.

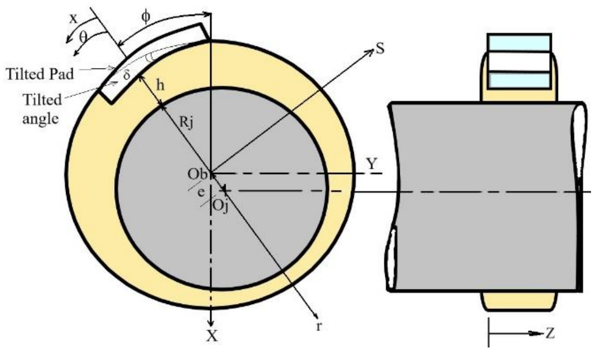

Fig. 1: The unconventional tilted pad bearing.

The 5 tilted pads journal bearing operating under load is required to maintain an appropriate minimum fluid-film thickness to minimize the metal to metal contact. The fluid film thickness is influenced by hydrodynamic forces which shown schematically in Fig. 1 for one of five pads in a tilted journal bearing system, the governing oil-film thickness expression is characterized by [4]:

$$h = C_r\,(1 + n \times \cos\,\theta\,) - x\,\tan\delta \qquad (1)$$

To estimate the resistance of disturbing forces it is necessary to know the relative stiffness between the shaft and the supporting structure along the line of pivot, produced by the lubricated oil film. The mean of stiffness in rigid material can be applied on lubricated oil found between rotating journal shaft and bearing, so, the stiffness and damping of this oil can be presented in many springs and dampers which connected with rotating shaft and bearing [10].

The generated forces on the tilted pads journal bearing could be analyzed in either access, (X, Y) or (s, r), therefore the stiffness coefficients can be expressed in either axis's [11]. In this mathematical model, these Stiffness coefficients are derived on (s, r) axis using the expression of Krr, Kss, Krs and Ksr, which can be obtained directly from the values of (hydrodynamic force, attitude angle (ϕ) represented in Eq. (2), the pivot eccentricity (e) and ratio of eccentricity (n) and the change in this values and by considering the range of eccentricity ratio (n) from (0.1) to (0.5), we can existing the magnitudes of coefficients of stiffness as write from Eqs. (3) to (6).

$$\varphi = \tan^{-1}\frac{F_{ts}}{F_{tr}} \tag{2}$$

$$k_{rr} = \frac{\partial w}{\partial e}\cos\varphi - w\frac{\partial \varphi}{\partial e}\sin\varphi \tag{3}$$

$$k_{ss} = \frac{w}{e}\cos\varphi \tag{4}$$

$$k_{rs} = \frac{-w}{e}\sin\varphi \tag{5}$$

$$k_{sr} = \frac{\partial w}{\partial e}\sin\varphi + w\frac{\partial \varphi}{\partial e}\cos\varphi \tag{6}$$

The bearing dimensions and journal speed as shown in Table 1.

Table 1
The characteristic properties of the journal.

Symbol	Value	Unit
Rb	0.025	m
Cr	0.00004375	m
Rp	0.025	m
L	0.05	m
N	10000	rpm
η	0.04, 0.058, 0.087	Pa. s
δ	0.01	deg.
$p\theta$	36	deg.

2.2. Solving Reynolds Equation

The pressure variation in the oil film is represented by two dimension Reynolds equation as shown in Eq. (7).

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right) = 6U\frac{\partial h}{\partial x} \tag{7}$$

Reynolds equation can have simplified as following equation [5]:

$$p_{i,j} = C_1 p_{i+1,j} + C_2 p_{i-1,j} + C_3 p_{i,j+1} + C_4 p_{i,j-1} - C_5 \tag{8}$$

We solved this equation numerically by finite difference method with 5 nodes in two dimensional mesh (in X and Z directions), to find the pressure distributed on each pad. The total area of pad has expressed by following equation:

$$A_p = (R_p \times \theta p) \times L \tag{9}$$

While the area of each element in the mesh is presented by:

$$a = \frac{A_p}{(m-1)(k-1)} \tag{10}$$

We find the forces distributed in each node by following equation

$$(F)_{ij} = a \times p_{i,j} \tag{11}$$

$$(F_r)_{ij} = a \times p_{i,j} \times \cos\theta i \tag{12}$$

$$(F_s)_{ij} = a \times p_{i,j} \times \sin\theta i \tag{13}$$

The summation of forces on each pad surface can be represented by:

$$F_r = \sum_{i=1}^{m}\sum_{j=1}^{k} a\, p_{ij} \cos\theta_i \tag{14}$$

$$F_s = \sum_{i=1}^{m}\sum_{j=1}^{k} a\, p_{ij} \sin\theta_i \tag{15}$$

The total forces in all tilted pads journal bearing submitted by two directions (parallel and perpendicular) force components as follow:

$$F_{tr} = \sum_{pad\,1}^{pad\,n} F_r \tag{16}$$

$$F_{ts} = \sum_{pad\,1}^{pad\,n} F_s \tag{17}$$

The hydrodynamic force that generated in bearing is calculated by:

$$W = \sqrt{F_{tr}^{\,2} + F_{ts}^{\,2}} \tag{18}$$

Assumptions:
In this study we use many assumptions as:
- The pressure is constant in the axial direction.
- Neglect all body forces.
- Surface curvature of bearing is very large compared with oil film thickness.
- No slipping in the oil layers in boundaries of shaft and bearing.
- Lubricated Oil is Newtonian and the inertia forces neglect compared with viscosity forces.
- Neglect thermal variation influence which cause thermal deformations.

3. RESULTS AND DISCUSSION

We programing the equations of Mathematical model and stiffness coefficients by Matlab program to solved them numerically, the results show effect of viscosity variation in range (0.04, 0.058, 0.087) on the response of stiffness coefficients of lubricated oil film in the bearing under study as shown in Figs. 2-5.

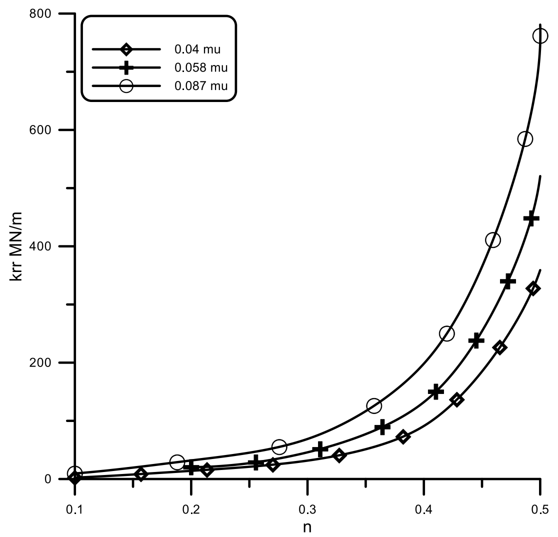

Fig. 2: Variation of Krr with the eccentiricity ratio (n) for different values of viscosity.

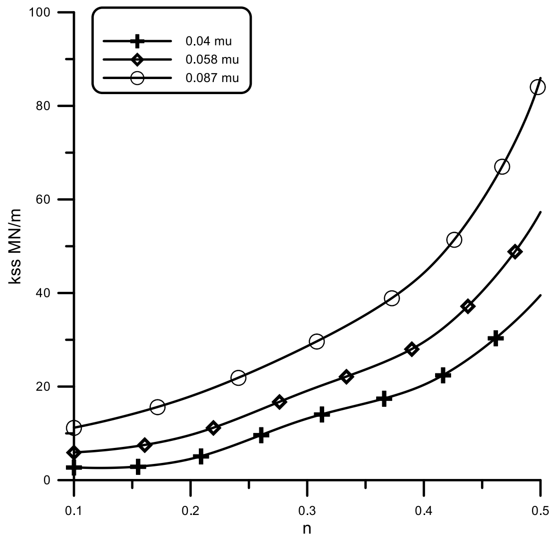

Fig. 3: Variation of Kss with the eccentiricity ratio (n) for different values of viscosity.

Referring to Fig. 2, there is clear increase in krr coefficient with viscosity increase for the range of eccentricity ratio in the interval (0.1-0.5), this increase of principal coefficient enhances the stability of this type of bearing, that is coincide with investigated in viscosity stiffness characteristic performance of lubricated two lobe journal bearing [12].

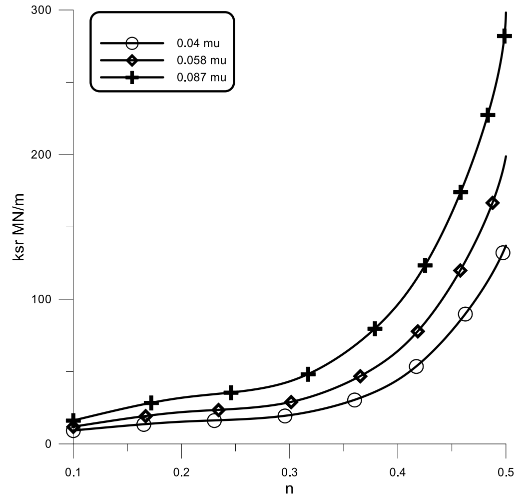

Fig. 4: Variation of Ksr with the eccentiricity ratio (n) for different values of viscosity.

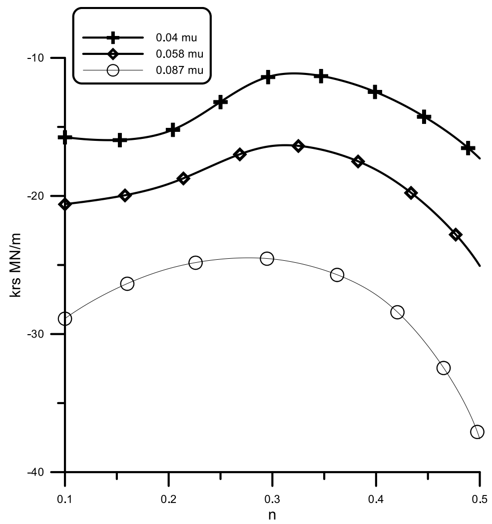

Fig. 5: Variation of Krs with the eccentiricity ratio (n) for different values of viscosity.

In Fig. 3 we find that the change in eccentricity with increase in viscosity cause increasing in values of Kss clearly, which is indicate positive meaning for improvement the stability, that is identical with stiffness improvement results in behavior of increase viscosity for non-Newtonian (micropolar) fluid film [13].

The cross coupling stiffness Ksr stiffness coefficient in Fig. 4 which show increases with viscosity by the value which not exceed the principal coefficient with Krr when viscosity varied in the studied range, this behavior represents good indication for dynamic characteristic in high speed roto-dynamic systems, which identical with Performance characteristics of tilted three-lobe journal bearing configurations [5].

Fig 5 illustrated that increasing the magnitude of viscosity leads to low significant effect in the value of cross coupling stiffness Krs, that cause Kss dynamic characteristic stiffness will be controlling coefficient in that direction, that is coincide with stability analysis of a compliant lemon bore journal bearing and results of dynamic Coefficients and elastic deformation for cylindrical pivot tilting 5-pad bearing [14,15].

This increase in stiffness coefficients was a result of increase in pressure value that corresponding the increase in viscosity.

4. CONCLUSIONS

From this search appears that conclusion which can be summarized as flows:

- There is good increase in principal stiffness Krr and Kss of the bearing with viscosity under studding.
- There is some increasing in cross coupling stiffness with viscosity but not greater than dominate principal stiffness.
- The increasing in the dominate principal stiffness Krr above the cross coupling in the same direction Ksr and other direction Ksr will increase the stability of this bearing.

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