Research Article

Effects of Texture Bottom Profile on Static and Dynamic Characteristics of Journal Bearings

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The purpose of this paper is to numerically study the effect of texture bottom profile on static, dynamic, and stability performance parameters of hydrodynamic journal bearings. The different performance parameters of square textured journal bearings with different bottom profiles are numerically investigated and compared with those of smooth journal bearing. There are five bottom profiles of this square texture: flat, curved, isosceles triangle (T1), oblique triangle (T2), and oblique triangle (T3). The static and dynamic coefficients are calculated by solving the steady-state Reynolds equation and the perturbation equations with FDM numerical technique. The performance characteristics under different texture distribution, depth, and bottom profiles are studied, and the current numerical results show that the selection of texture parameters is crucial to improve the static, dynamic, and stability performances of hydrodynamic journal bearing. Meanwhile, it is also found that the square texture with a flat bottom profile has a higher improvement in the values of static performance parameters in comparison to those other bottom profiles. Moreover, the simulation results indicate that the dynamic and stability performances improvement of textured journal bearing is also significant, especially when the eccentricity ratio is smaller.

1. Introduction

The surface texture method originated from surface roughness but is different from the surface roughness. Importantly, as a microbearing, the surface texture can obtain local hydrodynamic pressure and increase the load-carrying capacity due to its microwedge effect [1–3]. Furthermore, the introduction of textures of reasonable size and density can also reduce the friction and increase the service life of the mechanical seal [4–7]. Moreover, the method of introducing surface texture into friction pair has been proved to be an effective mean to improve the tribological performance characteristics.

In the past few decades, some theoretical and experimental studies on textured journal bearings have been published by numerous researchers. Ausas et al. [8] solved the Elrod and Adams [9] model with the mass conservation algorithm and found that the minimum film thickness and friction torque were increased slightly for fully rectangular textured journal bearings. Lu and Khonsari [10] carried out experimental investigation to analyze the performances of journal bearings with different dimples and found that fully textured journal bearing with reasonable texture parameters can minimize the friction coefficient under mixed lubrication condition. Another, Yamada et al. [11] carried out numerical and experimental investigation to study textured journal bearings with square dimples and reported that larger eccentricity ratio and smaller attitude angle were obtained compared with the smooth journal bearing. In addition, Tala-Ighil et al. [12, 13] numerically compared the performance of spherical and cylindrical textured journal bearings with different distributions and found that partial texture region has a positive effect on the performance of journal bearings while full texture region has a negative effect. Brizmer and Kligerman [14] demonstrated that partial texture distribution can obtain greater load-carrying capacity than full texture distribution for spherical textured journal bearing. Also, Zhang et al. [15] optimized the
spherical texture parameters to achieve greater load-carrying capacity of journal bearings. Kango et al. [16] found that microgrooved journal bearing can reduce the friction coefficient and average temperature better than spherical textured journal bearing, and the performance gain is more obvious at lower eccentricity ratio. Researchers [17–19] used a multiobjective optimization method to optimize the texture parameters of the microgrooved journal bearing to obtain greater load-carrying capacity and smaller friction coefficient. Lately, Filgueira Filho et al. [20] experimentally analyzed the performances of textured journal bearings with chevron, saw tooth, oblong dimple, and aligned dimple and found that the chevron textured journal bearing exhibited smaller eccentricity under reasonable texture geometrical parameters. Besides, some researchers [21–23] also analyzed the influence of protruded and dimple textures on the load-carrying capacity and friction of journal bearings and compared the results with that of smooth journal bearing.

Besides the above research studies on the static performance characteristics of textured journal bearings, some researchers have found that surface texture also affects dynamic and stability performance parameters in recent years. Importantly, stiffness and damping coefficients have significant effects on the dynamic performance of rotating systems as important criteria for designers to evaluate performance gains [24]. Jiang et al. [25] investigated the influence of spherical convex and concave textures on dynamic performance of journal bearings and reported that stability critical speed of journal bearing could be obtained with appropriate texture geometric parameters. Further, Yamada et al. [26, 27] numerically and experimentally analyzed the influence of full square texture on the dynamic performance and linear stability-threshold shaft speed of journal bearing; the results show that lower cross-coupled stiffness coefficients and higher stability-threshold speed of journal bearing were obtained. Besides, Matele and Pandey [28] studied the dynamic characteristics of journal bearings with circular, square, and densely distributed square textures and reported that the dynamic performances of journal bearing with properly texture geometric parameters have been significantly improved. Similarly, the effects of triangular-shaped and chevron-shaped textures on the static, dynamic, and stability performances of journal bearings have been studied, and the optimal performance parameters have been found by researchers [29–32]. In addition, Meng et al. [33] carried out a calculative investigation to study the effects of simple rectangular texture and compound texture on the dynamic performance of journal bearings and found that locally distributed compound texture can increase the direct stiffness coefficients, damping coefficients, and critical speed of journal bearing. Recently, Singh and Awasti [34, 35] numerically investigated the effects of texture parameters (texture depth and its distribution) on the static, dynamic, and stability performances of journal bearings with cylindrical, spherical, triangular, and kite textures; the results show that the performance parameters were significantly improved when the texture distribution is in the pressure rising region of bearings.

In numerous above works, the microtextures of several geometric shapes introduced on the bearing surface have been studied such as triangular, chevron, and spherical. Further, it is worth noting that the texture internal structure also plays an important role for tribological performances. Nanbu et al. [36] numerically investigated the effects of texture bottom profiles on lubrication performance and found that the microwedge effect of the texture bottom profile was an important parameter. In addition, Shen and Khonsari [37] carried out numerical and experimental investigation to demonstrate that dimple with flat bottom profile produces higher load-carrying capacity than those triangular profiles. Similarly, Rahmani et al. [38, 39] numerically studied the tribological performances of flat bottom textured journal bearing and found the maximum load-carrying capacity and minimum friction coefficient than those triangular profiles. Qiu et al. [40] conducted parametric optimization analysis on the textures of six different texture shapes and found that the ellipsoidal texture with curved bottom profiles could obtain the minimum friction coefficient and the maximum stiffness by comparison. Some researchers [41–43] used optimization algorithms to optimize the bottom profile of the texture, which improved the tribological performance. However, it should be noted that the selection principle of the texture optimal design parameters is not universal because the improvement of tribological performance is closely related to the operating conditions [44].

Although researchers have done numerous studies on the influence of texture parameters on performance characteristics of hydrodynamic journal bearing, under specific texture parameters, texture bottom profile affects static performance characteristics of journal bearing, such as load-carrying capacity, and friction force. And each bottom profile has a difference in static performance gain of journal bearing. However, to the knowledge of the authors, a comparative study of the effects of texture bottom profiles (flat, curved, isosceles triangular T1, oblique triangular T2, and oblique triangular T3) on the dynamic and stability performance characteristics of journal bearings by changing texture parameters (texture distribution and depth) and eccentricity ratios has not been reported. Therefore, the purpose of this paper is to systematically study the static, dynamic, and stability performance characteristics of square textured journal bearing with different bottom profiles. Thus, the texture distribution and depth are considered, and then the performance parameters of textured journal bearings are compared with those of smooth journal bearing to obtain the optimal texture parameters under specific conditions.

2. Theoretical Analysis

2.1. Reynolds Equation. In the study, the schematic diagram of textured journal bearing with the coordinate system and the expanded surface of textured journal bearing is shown in Figures 1(a) and 1(b). Besides, the square texture with different bottom profiles such as flat, curved, isosceles
triangular T1, oblique triangular T2, and oblique triangular T3 is introduced on the bearing surface in Figure 1(c).

In the fluid lubrication state, the Reynolds equation based on the assumptions of isothermal, laminar, and incompressible Newtonian fluids in Cartesian coordinate can be written as

\[
\frac{\partial}{\partial x} \left( h^3 \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial P}{\partial z} \right) = U \frac{\partial h}{\partial t},
\]

where \( P \) represents the fluid film pressure, \( \mu \) represents the fluid film dynamic viscosity, \( U \) denotes the journal speed in the \( x \) direction, and \( t \) denotes the time.

Besides, in order to unify the full fluid film region and cavitated region, the film rupture and the reformation boundary can be automatically determined in the numerical solution process, and the dimensionless dependent variable \( \phi \) and the switch function \( F \) are introduced as [45].

\[
F \phi = \frac{P - P_c}{P_r},
\]

\[
F = \begin{cases} 
1, & \phi \geq 0, \\
0, & \phi < 0,
\end{cases}
\]

where \( P_r \) represents the reference pressure and \( P_c \) represents the cavitation pressure.

Moreover, the fluid density can be showed as follows [46]:

\[
\frac{\rho}{\rho_c} = 1 + (1 - F)\phi,
\]

where \( \rho_c \) represents the local fluid density at the cavitation region.

The dimensionless parameters used can be expressed as

\[
\phi = \frac{x}{R},
\]

\[
Z = \frac{z}{L/2},
\]

\[
\bar{h} = \frac{h}{c},
\]

\[
\bar{t} = \frac{t}{P_r},
\]

\[
P_r = \frac{2 \mu \phi_0 h^2}{c^2}.
\]
where \( R \) denotes the radius of bearing, \( L \) represents length of bearing, \( c \) denotes radial clearance, \( \omega \) denotes angular velocity by the rotation of the journal, and \( \phi \) and \( Z \) represent the circumferential and axial coordinates of journal bearing.

\[
\frac{\partial}{\partial \phi} \left( \frac{\partial (F \phi)}{\partial \phi} \right) + \frac{2R}{L} \frac{\partial}{\partial Z} \left( \frac{\partial (F \phi)}{\partial Z} \right) = 3 \frac{\partial}{\partial \phi} \left[ \frac{H[1 + (1 - F)\phi]}{\partial \phi} \right] + 6 \frac{\partial}{\partial \phi} \left[ \frac{H[1 + (1 - F)\phi]}{\partial \phi} \right].
\]

(6)

And then the dimensionless fluid film pressure can be calculated by

\[
\bar{P} = \bar{P}_c + \phi.
\]

(7)

2.2. Fluid Film Thickness Equations. The dimensionless fluid film thickness for textured journal bearing can be described as

\[
\bar{h} = 1 + \varepsilon \cos \phi + \bar{h}_{\text{tex}},
\]

where \( \varepsilon \) represents the eccentricity ratio of journal bearing, \( \phi \) represents the circumferential axial coordinates of journal bearing, and \( \bar{h}_{\text{tex}} \) represents the dimensionless fluid film thickness contributed by surface texture. Besides, the expression for a square texture with different bottom profiles such as flat, curved, isosceles triangular T1, oblique triangular T2, and oblique triangular T3 can be calculated as follows [36].

For smooth surface,

\[
\bar{h}_{\text{tex}} = 0.
\]

(9a)

For flat bottom profile,

\[
\bar{h}_{\text{tex}} = \begin{cases} 
\bar{h}_p, & \text{if } |x - x_i| \leq \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| \leq \bar{r}_z, \\
0, & \text{if } |x - x_i| > \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| > \bar{r}_z.
\end{cases}
\]

(9b)

For curved bottom profile,

\[
\bar{h}_{\text{tex}} = \begin{cases} 
\bar{h}_p \left( 1 - \left( \frac{x - x_i}{\bar{r}_x} \right)^2 \right), & \text{if } |x - x_i| \leq \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| \leq \bar{r}_z, \\
0, & \text{if } |x - x_i| > \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| > \bar{r}_z.
\end{cases}
\]

(9c)

For isosceles triangular T1 bottom profile,

\[
\bar{h}_{\text{tex}} = \begin{cases} 
\bar{h}_p, & \text{if } |x - x_i| \leq \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| \leq \bar{r}_z, \\
0, & \text{if } |x - x_i| > \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| > \bar{r}_z.
\end{cases}
\]

(9d)

For oblique triangular T2 bottom profile,

\[
\bar{h}_{\text{tex}} = \begin{cases} 
\bar{h}_p |x - x_i| + \bar{h}_p, & \text{if } |x - x_i| \leq \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| \leq \bar{r}_z, \\
0, & \text{if } |x - x_i| > \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| > \bar{r}_z.
\end{cases}
\]

(9e)

For oblique triangular T3 bottom profile,

\[
\bar{h}_{\text{tex}} = \begin{cases} 
\bar{h}_p |x - x_i|, & \text{if } |x - x_i| \leq \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| \leq \bar{r}_z, \\
0, & \text{if } |x - x_i| > \bar{r}_x \text{ and } |\bar{z} - \bar{z}_i| > \bar{r}_z.
\end{cases}
\]

(9f)

2.3. Fluid Film Perturbation Pressure Equations. For the calculation of the dimensionless fluid film perturbation pressure, the dimensionless fluid film pressure is obtained from the steady-state Reynolds equation (10a) without considering the time term and equation (7), and then combined with the fluid film thickness equations (8) and ((9a) to (9f)), the dimensionless form of steady-state Reynolds equation and fluid film perturbation pressure equations in \((\varepsilon, \theta)\) coordinate systems can be obtained as follows:

(1) The dimensionless form of the steady-state Reynolds equation is as follows:

\[
\frac{\partial}{\partial \phi} \left( \bar{h} \frac{\partial (F \phi)}{\partial \phi} \right) + \left( \frac{2R}{L} \right)^2 \frac{\partial}{\partial Z} \left( \bar{h} \frac{\partial (F \phi)}{\partial Z} \right) = 3 \frac{\partial}{\partial \phi} \left[ \frac{\bar{h}[1 + (1 - F)\phi]}{\partial \phi} \right].
\]

(10a)

(2) The dimensionless form of fluid film perturbation pressure equation of \( \varepsilon \) is as follows:
\[
\frac{\partial}{\partial \phi} \left( \frac{\partial (\bar{P}_\phi)}{\partial \phi} \right) + \left( \frac{2R}{L} \right)^2 \frac{\partial}{\partial Z} \left( \frac{\partial (\bar{P}_Z)}{\partial Z} \right) = 3\bar{h} \left[ \frac{\partial \bar{P}}{\partial \phi} \cos \phi + \bar{h} \sin \phi \right] \frac{\partial (F_\phi \bar{P}_\phi)}{\partial \phi} + \left( \frac{2R}{L} \right)^2 \cos \phi \frac{\partial \bar{P}}{\partial Z} \frac{\partial (F_\phi \bar{P}_\phi)}{\partial Z} \right] + \left( -3 \sin \phi - \frac{9}{\bar{h}} \cos \phi \frac{\partial \bar{P}}{\partial \phi} \right) F_\phi. \] (10b)

(3) The dimensionless form of fluid film perturbation pressure equation of \( \theta \) is as follows:

\[
\frac{\partial}{\partial \phi} \left( \frac{\partial (\bar{P}_\theta)}{\partial \phi} \right) + \left( \frac{2R}{L} \right)^2 \frac{\partial}{\partial Z} \left( \frac{\partial (\bar{P}_Z)}{\partial Z} \right) = 3\bar{h} \left[ \frac{\partial \bar{P}}{\partial \phi} \sin \phi - \bar{h} \cos \phi \right] \frac{\partial (F_\phi \bar{P}_\phi)}{\partial \phi} + \left( \frac{2R}{L} \right)^2 \sin \phi \frac{\partial \bar{P}}{\partial Z} \frac{\partial (F_\phi \bar{P}_\phi)}{\partial Z} \right] + \left( +3 \cos \phi - \frac{9}{\bar{h}} \sin \phi \frac{\partial \bar{P}}{\partial \phi} \right) F_\phi. \] (10c)

Here, \( \bar{P}_\ell(\xi = \epsilon, \theta, \dot{\epsilon}, \dot{\theta}) \) represents the dimensionless fluid film perturbation pressure.

2.5. Performance Characteristics of Journal Bearing. The expressions of the static, dynamic, and stability performance parameters for journal bearing are presented in this section. Besides, it should be noted that the \((e, \theta)\) coordinate system is used in the calculation of the dimensionless performance parameters; once the equilibrium position of journal bearing is determined, the \((e, \theta)\) coordinate system needs to be reduced by an angle of \( \theta_c \) and then converted to the Cartesian coordinate system.

2.5.1. Static Performance Characteristics. In order to evaluate the performance characteristics of journal bearings for steady-state condition, when determining the journal center equilibrium position for a given eccentricity ratio, the dimensionless static performance characteristics can be calculated. Among them, the dimensionless static performance parameters include the load-carrying capacity, attitude angle, friction force, friction coefficient, and end leakage flow rate.

(1) The Load-Carrying Capacity. The load-carrying capacity can be determined by the method of force balance. The components of load-carrying capacity, in the circumferential and axial directions, can be obtained by integrating the hydrodynamic pressure generated in the clearance gap of journal bearing, whose dimensionless expressions are given as follows [47]:

\[
\bar{W}_t = -\int_{-1}^{1} \int_{0}^{2\pi} \bar{P} \sin \phi \, d\phi \, dZ, \] (13a)
\[
\bar{W}_r = -\int_{-1}^{1} \int_{0}^{2\pi} \bar{P} \cos \phi \, d\phi \, dZ. \] (13b)
The load-carrying capacity is defined as the total force that the journal bearing system can support, whose dimensionless expressions are expressed as follows:

\[ W = \sqrt{W_x^2 + W_z^2}. \] (13c)

The components of dimensionless load-carrying capacity in the Cartesian coordinate can be obtained as follows:

\[
\begin{bmatrix} W_x & W_y \\ W_x & W_z \end{bmatrix} = \begin{bmatrix} \cos \theta_e \sin \theta_e \\ \sin \theta_e \cos \theta_e \end{bmatrix}. \] (14)

(2) The Attitude Angle. The attitude angle of the journal bearing is expressed as follows:

\[ \theta = \begin{cases} \pi - \arcsin \left( \frac{W_y}{W_z} \right), & W_z \geq 0, \\ \arcsin \left( \frac{W_y}{W_z} \right), & W_z < 0. \end{cases} \] (15)

(3) The Friction Force and the Friction Coefficient. The dimensionless form of viscosity shear stress due to lubricant shearing behavior can be obtained as follows:

\[ \tau = \left( \frac{\tau}{\pi \epsilon} \right) + 1. \] (16a)

The frictional force of lubricant on the journal surface can be calculated by integrating the shear stress, whose dimensionless expression is given as follows:

\[ F_f = -\int_{-1}^{1} \int_{0}^{2\pi} \tau \, d\phi \, dZ. \] (16b)

The dimensionless friction coefficient of lubricant on the journal surface can be calculated as follows:

\[ \left( \frac{F_f}{W} \right) = \frac{\tau}{\tilde{h}}. \] (17)

(4) The End Leakage Flow Rate. The dimensionless end leakage flow rate has two components such as \( \mathcal{Q}_1 \) from the left-hand bearing end and \( \mathcal{Q}_2 \) from the right-hand bearing end which are given as follows:

\[ \mathcal{Q}_1 = -\int_{0}^{2\pi} \frac{\tilde{h}^3}{12} \left( \frac{\partial \bar{P}}{\partial Z} \right)_{Z=0} \, d\phi, \] (18a)

\[ \mathcal{Q}_2 = +\int_{0}^{2\pi} \frac{\tilde{h}^3}{12} \left( \frac{\partial \bar{P}}{\partial Z} \right)_{Z=1} \, d\phi. \] (18b)

The total dimensionless end leakage flow rate can be calculated as follows:

\[ \mathcal{Q}_e = \mathcal{Q}_1 + \mathcal{Q}_2. \] (18c)

2.5.2. Dynamic Performance Characteristics. When journal center is disturbed from the static equilibrium position, the dimensionless dynamic performance characteristics can be calculated. Among them, the dimensionless dynamic performance parameters include the fluid film stiffness coefficients and fluid film damping coefficients. Importantly, the stiffness and damping coefficients are the spring and dashpot systems, which constrain the movement of the journal during operation [24].

(1) The Fluid Film Stiffness Coefficients. After calculating the fluid film perturbation pressure, the fluid film stiffness coefficients can be calculated by integrating the fluid film perturbation pressure, whose dimensionless expressions in the \((e, \theta)\) coordinate systems are expressed as follows [47]:

\[ \begin{align*}
K_{ee} &= -\int_{-1}^{1} \int_{0}^{2\pi} F_f \bar{P} \left\{ \cos \phi \sin \phi \right\} \, d\phi \, dZ, \\
K_{e\theta} &= -\int_{-1}^{1} \int_{0}^{2\pi} F_f \bar{P} \left\{ \cos \phi \sin \phi \right\} \, d\phi \, dZ.
\end{align*} \] (19a, 19b)

The dimensionless fluid film stiffness coefficients in the Cartesian coordinate system are expressed as follows:

\[ \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} = \begin{bmatrix} \sin \theta_e & \cos \theta_e \\ \cos \theta_e & -\sin \theta_e \end{bmatrix} \begin{bmatrix} K_{ee} & K_{e\theta} \\ K_{e\theta} & K_{\theta\theta} \end{bmatrix} \begin{bmatrix} \sin \theta_e & \cos \theta_e \\ \cos \theta_e & -\sin \theta_e \end{bmatrix}. \] (20)

The direct stiffness coefficients \( K_{xx} \) and \( K_{yy} \) reflect the ability to resist external loads in the \( x \) and \( y \) directions, respectively. The larger the values of the direct stiffness coefficients are, the better the stability of the journal bearing system is. The cross-coupled stiffness coefficients \( K_{xy} \) and \( K_{yx} \) reflect the unstable energy input of the bearing. The smaller the absolute values of the cross-coupled stiffness coefficients are, the better the stability of the journal bearing system is. Therefore, the stiffness coefficients have significant influence on the overall dynamic characteristics of the journal bearing system [24].

(2) The Fluid Film Damping Coefficients. After calculating the fluid film perturbation pressure, the fluid film damping coefficients can be calculated by integrating the fluid film perturbation pressure, whose dimensionless expressions in the \((e, \theta)\) coordinate systems are expressed as follows [47]:

\[ \begin{align*}
\bar{C}_{ee} &= -\int_{-1}^{1} \int_{0}^{2\pi} F_f \bar{P} \left\{ \cos \phi \sin \phi \right\} \, d\phi \, dZ, \\
\bar{C}_{e\theta} &= -\int_{-1}^{1} \int_{0}^{2\pi} F_f \bar{P} \left\{ \cos \phi \sin \phi \right\} \, d\phi \, dZ.
\end{align*} \] (21a, 21b)

The dimensionless fluid film damping coefficients in the Cartesian coordinate system can be obtained as follows:
\[
\begin{bmatrix}
C_{xx} & C_{xy} \\
C_{yx} & C_{yy}
\end{bmatrix} = \begin{bmatrix}
\sin \theta_c & \cos \theta_c \\
\cos \theta_c & -\sin \theta_c
\end{bmatrix} \begin{bmatrix}
C_{te} & C_{to} \\
C_{ot} & C_{go}
\end{bmatrix} \begin{bmatrix}
\sin \theta_c & \cos \theta_c \\
\cos \theta_c & -\sin \theta_c
\end{bmatrix}
\]  

(22)

The damping coefficients reflecting the speed of the journal are restored to stability after being impacted by external loads. The larger the values of the damping coefficients are, the better the stability of the journal bearing system is. Similarly, the damping coefficients have significant influence on the overall dynamic characteristics of the journal bearing system [24].

2.5.3. Stability Performance Characteristics

(1) The Critical Mass. For the journal bearing system, the dimensionless form of linearized motion equation is expressed with the known eight dynamic coefficients as follows [47, 48]:

\[
\begin{bmatrix}
\mathcal{M}_f \\
\mathcal{M}_j
\end{bmatrix} \begin{bmatrix}
\ddot{X}_f \\
\ddot{Y}_f
\end{bmatrix} + \begin{bmatrix}
C_{xx} & C_{xy} \\
C_{yx} & C_{yy}
\end{bmatrix} \begin{bmatrix}
\ddot{X}_f \\
\ddot{Y}_f
\end{bmatrix} + \begin{bmatrix}
K_{xx} & K_{xy} \\
K_{yx} & K_{yy}
\end{bmatrix} \begin{bmatrix}
X_f \\
Y_f
\end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix},
\]

(23a)

where \(X_f\) and \(Y_f\) are the displacement, \(X_f' = \varepsilon \sin \phi\), \(Y_f' = \varepsilon \cos \phi\), \(\dot{X}_f\) and \(\dot{Y}_f\) are velocity, and \(\ddot{X}_f\) and \(\ddot{Y}_f\) are the accelerations, for the journal in \(x\) and \(y\) the directions. Importantly, it should be noted here that the system is considered stable when the journal mass is smaller than the critical mass \((\bar{M}_f < \bar{M}_j)\). The dimensionless critical mass is written as follows [47, 48]:

\[
\bar{M}_c = -\frac{\mathcal{M}_f}{\mathcal{M}_2 - \mathcal{M}_3},
\]

(23b)

Here,

\[
\mathcal{M}_1 = (C_{xx} C_{yy} - C_{xy} C_{yx}),
\]

\[
\mathcal{M}_2 = \frac{(C_{xx} + C_{yy})(K_{xx} K_{yy} - K_{xy} K_{yx})}{(C_{xx} K_{yy} + C_{yy} K_{xx} - C_{xy} K_{yx} - C_{yx} K_{xy})}
\]

\[
\mathcal{M}_3 = \frac{(K_{xx} C_{xx} + K_{yy} C_{yy} + K_{xy} C_{yx} + K_{yx} C_{xy})}{(C_{xx} + C_{yy})}
\]

(24)

(2) The Threshold Speed of Stability. The threshold speed of stability is an important parameter to evaluate the stability of the journal bearing system. Importantly, the value of threshold speed of stability is higher than that of the journal speed, indicating that the journal bearing system will become more stable. After calculating the dimensionless value of the critical mass, the dimensionless value of threshold speed of stability is written as follows [47, 48]:

\[
\bar{\omega}_th = \frac{\bar{M}_c}{W_0}
\]

(25)

Here, \(W_0\) is the reaction force of fluid film \((\partial \hat{h} / \partial \hat{e} = 0)\).

3. Solution Procedure and Numerical Methods

In the present study, in order to calculate the influence of texture parameters on the static, dynamic, and stability performance characteristic parameters of journal bearing, the global numerical calculation process is divided into two parts: the steady-state parameters numeration (step 2 to step 8) and dynamic parameters numeration (step 9 to step 11). Moreover, the computation process is operated in MATLAB R2014a. Figure 2 illustrates the flowchart of the global solution procedure, and the details of numerical procedure and methods are shown in the Appendix.

4. Results and Discussion

In this section, the static, dynamic, and stability performance characteristic parameters of journal bearings are numerically calculated. Besides, five different bottom profiles of a square texture, flat, curved, isosceles triangle (T1), oblique triangle (T2), and oblique triangle (T3) have been introduced on the bearing surface, and the values of circumferential texture region, texture depth, and eccentricity ratio are considered. Furthermore, performance characteristic parameters of textured journal bearings are numerically compared with those of smooth journal bearing. Here, the operating parameters and texture parameters in the current study are listed in Table 1. It should be emphasized that, unless otherwise specified, the values of these parameters remain unchanged in the following theoretical analysis.

4.1. Validation. In order to verify the accuracy of current model numerical solution, model validation has been shown in the two types. In the first type, for the verification study, the static performance characteristics of smooth, partially textured, and fully textured journal bearings are calculated. Table 2 presents the results of comparison of the current model compared with Tala-Ighil et al. [12]. Besides, in order to clearly show the validation results, the dimensionless fluid film thicknesses and fluid film pressures of smooth, partially textured, and fully textured journal bearings are shown in Figure 3. In the second type, for the verification study the static and dynamic performance characteristics of partially textured journal bearing with texture region located at rising pressure region, Figure 4 presents the results of comparison of the current model compared with Singh and Awasthi [34]. It is important to emphasize that because of the external
force acting in the $y$ direction, the direct stiffness coefficient ($K_{yy}$) is the key factors to consider for dynamic performance; the direct stiffness coefficient ($K_{yy}$) and the load-carrying capacity ($W$) can be improved in rising pressure texture region. In particular, the performance gains are superior at lower eccentricity ratios. Moreover, the geometric parameter values of journal bearing in the validation are the same as those in the reference studies, and the numerical differences obtained are all within the acceptable range.

4.2. Influence of the Circumferential Texture Region. In this subsection, the influences of the circumferential texture region on the static, dynamic, and stability performance parameters of smooth/square textured journal bearings with flat, curved, triangle T1, triangle T2, and triangle T3 bottom profiles are observed in Figures 5 and 6. Here, five circumferential texture regions are taken into consideration, rising pressure texture region ($0°$–$128.5°$), first half texture region ($0°$–$180°$), middle texture region ($128.5°$–$257°$), second half texture region ($180°$–$360°$), and full texture region

**Figure 2:** Flowchart of the numerical calculation process for the steady state and dynamic performance parameters of journal bearing.

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Table 1: Geometrical and texturing parameters for the studied textured journal bearing.

| Parameters                        | Notation | Values     |
|-----------------------------------|----------|------------|
| Journal radius                    | $R$      | 20 mm      |
| Length to diameter ratio          | $L/D$    | 1          |
| Radial clearance                  | $c$      | 50 $\mu$m  |
| Eccentricity ratio                | $\varepsilon$ | 0.3      |
| Journal speed                     | $n$      | 3000 rpm   |
| Viscosity                         | $\mu$    | 0.08 Pa $\cdot$ s |
| Environment pressure              | $P_0$    | 100000 Pa   |
| Cavitation pressure               | $P_c$    | 27860.21 Pa |
| Texture length                    | $r_x$    | 3 mm       |
| Texture width                     | $r_z$    | 3 mm       |
| Texture depth                     | $r_p$    | 10 $\mu$m  |
| Texture density                   | $S_p$    | 0.5        |
| Texture number along              | $N_x$    | 5          |
| Circumferential direction texture number along | $N_z$ | 4      |
| Axial direction circumferential texture region | $\frac{(z_2 - z_1)}{L}$ | $0^\circ$ – 128.5$^\circ$ |
| Axial texture region              | $\frac{(\phi_2 - \phi_1)}{2\pi}$ | 0.8 |

Table 2: Model validation of static performance characteristics of smooth/textured journal bearing ($L/D$, $c = 30 \mu$m, $\mu = 0.0035$ Pa $\cdot$ s, and $\omega = 625.4$ rad/s).

| Performance parameters | Results of Tala-Ighilet al. [12] | Present study |
|------------------------|----------------------------------|---------------|
|                        | Smooth (0.601)                  | Partially textured (0.595) | Fully textured (0.709) |
| Load-carrying capacity, $\bar{W}$ | 2.624 | 2.624 | 2.624 | 2.618 | 2.616 | 2.629 |
| Minimum film thickness, $h_{\min}$ | 0.399 | 0.405 | 0.291 | 0.399 | 0.405 | 0.291 |
| Maximum pressure, $P_{\max}$ | 1.594 | 1.568 | 1.708 | 1.590 | 1.561 | 1.701 |
| Leakage flow rate, $Q_{Z\max}$ | 0.467 | 0.464 | 0.381 | 0.466 | 0.460 | 0.378 |
| Attitude angle, $\theta$ (°) | 50.5 | 49.3 | 46.1 | 50.396 | 49.108 | 45.998 |

Figure 3: Continued.
Figure 3: The dimensionless fluid film thickness $\overline{h}$ and fluid film pressure $\overline{p}$: (a-b) smooth journal bearing; (c-d) partially textured journal bearing; (e-f) fully textured journal bearing.

Figure 4: Continued.
Figure 4: Model validation of static and dynamic performance characteristics of partially textured journal bearing with texture region located at rising pressure region between present results and literature: (a) the load-carrying capacity ($W$); (b) the friction coefficient ($f$); (c) the direct stiffness coefficient ($K_{xx}$); (d) the direct stiffness coefficient ($K_{yy}$); (e) the direct damping coefficient ($C_{xx}$); (f) the direct damping coefficient ($C_{yy}$).

Figure 5: Continued.
(0°–360°), and the corresponding number of textures along the circumferential direction is 5, 7, 5, 7, and 14, respectively.

As seen from Figures 5(a) to 5(f), the influence of the circumferential texture region on the static performance parameters of textured journal bearings is shown, such as the load-carrying capacity ($W$), the maximum fluid film pressure ($P_{\text{max}}$), the friction coefficient ($\tau$), the end leakage flow rate ($Q_z$), the attitude angle ($\theta$), and the friction force ($F_f$).

From Figures 5(a) and 5(b), it is observed that the load-carrying capacity ($W$) and the maximum fluid film pressure ($P_{\text{max}}$) of textured journal bearings are higher than those of smooth journal bearing in rising pressure texture region and first half texture region, while other texture distributions have negative effects. Furthermore, it is found that square textured journal bearing with flat bottom profile has the highest values of $W$ and $P_{\text{max}}$ amongst other textured journal bearings with different bottom profiles in rising pressure texture region. In Figure 5(c), the variation values of the friction coefficient ($\tau$) are opposite to those of the load-carrying capacity ($W$) of textured journal bearings in each texture region. Moreover, it is found that square textured journal bearing with flat bottom profile has the lowest value of $\tau$ amongst other textured journal bearings with different bottom profiles in rising pressure texture region. In Figure 5(d), the end leakage flow rate ($Q_z$) of textured journal bearings is higher than that of smooth journal bearing in rising pressure texture region, first half texture region, and full texture region, and then values are lower.
Figure 6: Effect of the circumferential texture region on dynamic and stability performance characteristics for smooth/textured journal bearings: (a) the direct stiffness coefficient ($K_{xx}$); (b) the direct stiffness coefficient ($K_{yy}$); (c) the direct damping coefficient ($C_{xx}$); (d) the direct damping coefficient ($C_{yy}$); (e) the critical mass ($\bar{M}_c$); (f) the threshold speed of stability ($\bar{\omega}_{th}$).
than those of smooth bearing in other texture distributions. Furthermore, it is found that square textured journal bearing with flat bottom profile has the highest value of the end leakage flow rate ($Q_{2e}$) amongst other textured journal bearings with different bottom profiles in rising pressure texture region. In Figure 5(e), the attitude angle ($\theta$) of textured journal bearings is lower than that of smooth bearing in rising pressure texture region, first half texture region, and full texture region, and then values are higher than those of smooth bearing in other texture regions. Moreover, it is found that square textured journal bearing with flat bottom profile has the lowest value of $\theta$ amongst other textured journal bearings with different bottom profiles in first half texture region. In Figure 5(f), the friction force ($F_f$) of textured journal bearings is lower than that of smooth bearing in each texture region. Besides, it is found that square textured journal bearing with flat bottom profile has the lowest value of $F_f$ amongst other textured journal bearings with different bottom profiles in full texture region.

As seen from Figures 6(a) to 6(f), the influence of the circumferential texture region on the dynamic and stability performance parameters of textured journal bearings is shown, such as the direct stiffness coefficients ($K_{xx}$), the direct damping coefficients ($F_f$), the critical mass ($M_c$), and the threshold speed of stability ($\omega_{th}$). In Figure 6(a), the direct stiffness coefficients ($K_{xx}$) of textured journal bearings are lower than those of smooth bearing in each texture regions except in first half texture region. In addition, it is found that square textured journal bearing with flat bottom profile has the highest value of $K_{xx}$ amongst other textured journal bearings with different bottom profiles in first half texture region. In Figure 6(b), the direct stiffness coefficients ($K_{yy}$) of textured journal bearings are higher than those of smooth bearings in rising pressure texture region and second half texture region; nevertheless, the direct stiffness coefficients ($K_{yy}$) of textured journal bearings are lower than those of smooth bearings in other texture regions. Moreover, it is found that square textured journal bearing with flat bottom profile has the highest value of $K_{nn}$ amongst other textured journal bearings with different bottom profiles in second half texture region. From Figures 6(c) and 6(d), it is observed that the direct damping coefficients ($F_f$) of textured journal bearings are lower than those of smooth bearing in each texture region. Furthermore, it is found that smooth journal bearing has the highest values of $C_{xx}$ and $C_{yy}$ amongst textured journal bearings with different bottom profiles in each texture region. In Figure 6(e), the critical mass ($M_c$) of textured journal bearings is higher than that of smooth bearings in rising pressure texture region and first half texture region; nevertheless, the critical mass ($M_c$) of textured journal bearings is lower than that of smooth bearings in other texture regions. Besides, it is found that square textured journal bearing with flat bottom profile has the highest values of $M_c$ amongst other textured journal bearings with different bottom profiles in first half texture region. In Figure 6(f), it is observed that the threshold speed of stability ($\omega_{th}$) of textured journal bearings is lower than that of smooth bearing in rising pressure texture region, and then the threshold speed of stability ($\omega_{th}$) of textured journal bearings is higher than that of smooth bearing in other texture regions. Furthermore, it is found that square textured journal bearing with flat bottom profile has the lowest values of $\omega_{th}$ amongst other textured journal bearings with different bottom profiles in rising pressure texture region.

Based on the above analysis of performance parameters of journal bearing, the results indicate that different texture regions have different effects on the performances of textured journal bearings with different bottom profiles. It is found that the textured journal bearing can maximize the load-carrying capacity ($W$), the maximum fluid film pressure ($P_{max}$), and lubrication flow ratio ($Q_f$) and minimize the friction coefficient ($F_f$), compared with smooth journal bearing. It is important to emphasize that because of the external force acting in the $y$ direction, the direct stiffness coefficient ($K_{yy}$) and the critical mass ($M_c$) are the key factors to consider for dynamic performance. The direct stiffness coefficient ($K_{xx}$) and the critical mass ($M_c$) can also be improved simultaneously in rising pressure texture region. Moreover, the performance of textured journal bearing with flat bottom profile is better than that of textured journal bearings with other bottom profiles. Therefore, the performance of square textured journal bearings in rising pressure texture region will be analyzed in the following sections.

4.3. Influence of the Texture Depth. In this subsection, it should be emphasized that the selected texture region is the rising pressure texture region of journal bearing. The influences of the texture depth ($h_t$) on the static, dynamic, and stability performance parameters of smooth/square textured journal bearings with flat, curved, triangle $T_1$, triangle $T_2$, and triangle $T_3$ bottom profiles are shown and discussed in Figures 7 and 8, and the texture depth ($h_t$) ranges from 0 to 1.2 in the study.

As seen from Figures 7(a) to 7(f), the influence of the texture depth ($h_t$) on the static performance parameters of textured journal bearings is shown, such as the load-carrying capacity ($W$), the maximum film pressure ($P_{max}$), the friction coefficient ($F_f$), the end leakage flow rate ($Q_{2e}$), the attitude angle ($\theta$), and the friction force ($F_f$). From Figures 7(a) and 7(b), the results show that the load-carrying capacity ($W$) and the maximum film pressure ($P_{max}$) of textured journal bearings are higher than those of smooth bearing in the texture depth ($h_t$) ranges. Moreover, with the increase in the texture depth ($h_t$), the load-carrying capacity ($W$) and the maximum film pressure ($P_{max}$) of textured journal bearings firstly increase and then decrease, respectively. It can also be found that there is an optimum value of texture depth for the load-carrying capacity ($W$) and the maximum film pressure ($P_{max}$), and that depends on the operating conditions of the bearing. Furthermore, the optimal texture depth of textured journal bearings with different bottom profiles is different, and square textured journal bearing with flat bottom profile has the highest values of $W$ and $P_{max}$ amongst other textured journal bearings with different bottom profiles. From Figures 7(c) and 7(d), the results show that the friction coefficient ($F_f$) and the
Figure 7: Continued.
Figure 7: Effect of the texture depth on static performance characteristics for smooth/textured journal bearings: (a) the load-carrying capacity ($W$); (b) the maximum fluid film pressure ($P_{\text{max}}$); (c) the friction coefficient ($f$); (d) the end leakage flow rate ($Q_z$); (e) the attitude angle ($\theta$); (f) the friction force ($F_f$). Bearings are lower than that of smooth bearing in the texture depth ($h_y$) ranges. However, the end leakage flow rate ($Q_z$) keeps increasing and the friction force ($F_f$) keeps decreasing, respectively.

Figure 8: Continued.
attitude angle ($\theta$) of textured journal bearings are lower than those of smooth bearing in the texture depth ($h_p$) ranges. Contrary to the changing trend of the load-carrying capacity ($W$), with the increase in the texture depth ($h_p$), the friction coefficient ($f$) and the attitude angle ($\theta$) of textured journal bearings firstly decrease and then increase, respectively. Furthermore, it is also found that square textured journal bearing with flat bottom profile has the lowest values of $f$ and $\theta$ amongst other textured journal bearings with different bottom profiles. From Figures 7(d) and 7(f), the results show that the end leakage flow rate ($Q_Z$) of textured journal bearings is higher than that of smooth bearing and the friction force ($F_f$) of textured journal.

As seen from Figures 8(a) to 8(f), the influence of the texture depth ($h_p$) on the dynamic and stability performance parameters of textured journal bearings is shown, such as the direct stiffness coefficients ($K_{xx}$, $K_{yy}$), the direct damping coefficients ($C_{xx}$, $C_{yy}$), the critical mass ($M_c$), and the threshold speed of stability ($\omega_{th}$). And from the results, the direct stiffness coefficients ($K_{xx}$, $K_{yy}$), the direct damping coefficients ($C_{xx}$, $C_{yy}$), and the threshold speed of stability ($\omega_{th}$) of textured journal bearings are lower than those of smooth bearing in the texture depth ($h_p$) ranges; moreover, with the
increase in the texture depth ($\text{R}_{t}$), the direct stiffness coefficients ($K_{xx}$), the direct damping coefficients ($C_{xx}$, $C_{yy}$), and the threshold speed of stability ($\omega_{th}$) of textured journal bearings are always decreased in Figures 8(a), 8(c), 8(d), and 8(f), respectively. Furthermore, it is found that square textured journal bearing with flat bottom profile has the lowest values of $K_{xx}$, $C_{xx}$, $C_{yy}$, and $\omega_{th}$ amongst other textured journal bearings with different bottom profiles. From Figures 8(b) and 8(e), the direct stiffness coefficient ($K_{yy}$) and the critical mass ($M_{c}$) firstly increase and then decrease, respectively. It can also be found that there is an optimum value of texture depth for the direct stiffness coefficient ($K_{yy}$) and the critical mass ($M_{c}$), and that depends on the operating conditions of the bearing. Furthermore, the optimal texture depth of textured journal bearings with different bottom profiles is different, and square textured journal bearing with flat bottom profile has the highest values of $K_{yy}$ and $M_{c}$ amongst other textured journal bearings with different bottom profiles.

Based on the above analysis of performance parameters of journal bearing, the results indicate that different texture depth ($\text{R}_{t}$) has different effects on the performances of textured journal bearings with different bottom profiles, and the optimal texture depth of each bottom profiles is also different. There is an optimum value of texture depth that maximize the load-carrying capacity ($\overline{W}$), maximum fluid film pressure ($P_{max}$), lubrication flow ratio ($Q_{z}$), direct stiffness coefficient ($K_{yy}$), and critical mass ($M_{c}$) and minimize the friction coefficient ($F_{f}$), respectively, compared with smooth journal bearing. Moreover, for the texture depth ($\text{R}_{t}$) within the selected range, the performance of textured journal bearing with flat bottom profile is better than that of square textured journal bearings with other bottom profiles.

4.4. Influence of the Eccentricity Ratio. In this subsection, it should be emphasized that the selected texture region is the rising pressure texture region of journal bearing. The influences of the eccentricity ratio ($\varepsilon$) on the static, dynamic, and stability performance parameters of smooth/square textured journal bearings with flat, curved, triangle T1, triangle T2, and triangle T3 bottom profiles are presented and discussed in Figures 9 and 10, and the eccentricity ratio ($\varepsilon$) ranges from 0.2 to 0.8 in the study.

As seen from Figures 9(a) to 9(f), the influence of the eccentricity ratio ($\varepsilon$) on the relative difference on static performance parameters of textured journal bearings relative to smooth bearing is presented, such as the load-carrying capacity ($\overline{W}$), the maximum film pressure ($P_{max}$), the friction coefficient ($F_{f}$), the end leakage flow rate ($Q_{z}$), the attitude angle ($\theta$), and the friction force ($F_{f}$). And the results show that, with the increase in the eccentricity ratio ($\varepsilon$), the load-carrying capacity ($\overline{W}$), the maximum film pressure ($P_{max}$), the end leakage flow rate ($Q_{z}$), and the friction force ($F_{f}$) of smooth journal bearing increase, while the friction coefficient ($F_{f}$) and the attitude angle ($\theta$) of smooth journal bearing decrease accordingly. From Figures 9(a) to 9(c), the load-carrying capacity ($\overline{W}$) and the maximum film pressure ($P_{max}$) of textured journal bearings are higher than those of smooth bearing, and the friction coefficient ($F_{f}$) of textured journal bearings is lower than that of smooth bearing for the eccentricity ratio ($\varepsilon$) ranges from 0.2 to 0.6; however, the trend of change of these values is the opposite for the eccentricity ratio ($\varepsilon$) exceeding 0.6. Furthermore, for the value of the eccentricity ratio 0.3 ($\varepsilon$), it is also found that square textured journal bearing with flat bottom profile has the highest values of $\overline{W}$, $P_{max}$, and the lowest value of $F_{f}$ amongst other textured journal bearings with different bottom profiles. In Figure 9(d), regardless of the change of eccentricity ratio ($\varepsilon$), the end leakage flow rate ($Q_{z}$) of textured journal bearings is always higher than that of smooth bearing. Furthermore, for the value of the eccentricity ratio 0.3 ($\varepsilon$), it is also found that square textured journal bearing with flat bottom profile has the lowest values of $\overline{W}$, $P_{max}$, and the lowest value of $F_{f}$ amongst other textured journal bearings with different bottom profiles.

From Figures 10(a) to 10(f), the influence of the eccentricity ratio ($\varepsilon$) on the relative difference in dynamic and stability performance parameters of textured journal bearings relative to smooth bearing is shown, such as the direct stiffness coefficients ($K_{xx}$,$K_{yy}$), the direct damping coefficients ($C_{xx}$, $C_{yy}$), the critical mass ($M_{c}$), and the threshold speed of stability ($\omega_{th}$). And from the results, with the increase in the eccentricity ratio ($\varepsilon$), the direct stiffness coefficients ($K_{xx}$,$K_{yy}$), the direct damping coefficients ($C_{xx}$, $C_{yy}$), the critical mass ($M_{c}$), and the threshold speed of stability ($\omega_{th}$) of smooth journal bearing also increase accordingly. In Figures 10(a), 10(c), 10(d), and 10(f), regardless of the change of eccentricity ratio ($\varepsilon$), the direct stiffness coefficients ($K_{xx}$), the direct damping coefficients ($C_{xx}$, $C_{yy}$, $C_{zz}$), and the threshold speed of stability ($\omega_{th}$) of textured journal bearings are always lower than those of smooth bearing. Furthermore, for the value of the eccentricity ratio 0.3 ($\varepsilon$), it is also found that square textured journal bearing with flat bottom profile has the lowest values of $K_{xx}$, $C_{xx}$, $C_{yy}$, and $\omega_{th}$ amongst other textured journal bearings with different bottom profiles. From Figures 10(b) and 10(e), the critical mass ($M_{c}$) and the direct stiffness coefficient ($K_{yy}$) of textured journal bearings are higher than those of smooth bearing for the eccentricity ratio ($\varepsilon$) ranges from 0.2 to 0.45, and then it is the opposite for the eccentricity ratio greater than 0.45. Furthermore, for the value of the eccentricity ratio 0.3 ($\varepsilon$), it is also found that square textured journal bearing with flat bottom profile has the highest values of $M_{c}$ and $K_{yy}$ amongst other textured journal bearings with different bottom profiles. It is important to emphasize that because of the external force acting in the $y$ direction, the direct stiffness coefficient ($K_{yy}$) is the key factor to consider for dynamic performance.
Figure 9: Continued.
Figure 9: Effect of the eccentricity ratio on static performance characteristics for smooth/textured journal bearings: (a) the load-carrying capacity ($W$); (b) the maximum fluid film pressure ($P_{\text{max}}$); (c) the friction coefficient ($\mathcal{T}$); (d) the end leakage flow rate ($Q_z$); (e) the attitude angle ($\theta$); (f) the friction force ($F_f$).

Figure 10: Continued.
Based on the above analysis of performance parameters of journal bearing, the results indicate that different eccentricity ratios ($\varepsilon$) have different effects on the performances of textured journal bearings with different bottom profiles. It can be found that the larger eccentricity ratio ($\varepsilon$), minimum film thickness will be smaller, and the hydrodynamic pressure effect will be more obvious. And the eccentricity ratio ($\varepsilon$) is smaller, the relative difference of performance parameters of journal bearings provided by surface texture is more obvious. However, with the increase in eccentricity ratio ($\varepsilon$), the performances improvement become gradually weakened. Moreover, for the eccentricity ratio ($\varepsilon$) within the selected range, the results show that square textured journal bearing with flat bottom profile can obtain better performance improvement.

5. Conclusions

In this paper, the effects of a square texture with flat, curved, isosceles triangle ($T_1$), oblique triangle ($T_2$), and oblique triangle ($T_3$) bottom profiles on the static, dynamic, and stability performance parameters of journal bearings are numerically studied at different texture distribution and depth. Firstly, the film pressure is obtained through solving the steady-state Reynolds equation, and then the perturbation pressures are also obtained through solving the perturbation equations. Next, the static, dynamic, and stability performance parameters of journal bearings are calculated. Finally, the performance characteristic parameters of the textured journal bearings with difference bottom profiles are compared.

![Graphs showing the effects of eccentricity ratio on dynamic and stability performance characteristics for smooth/textured journal bearings.](image-url)
with those of the smooth journal bearing. The relevant conclusions are drawn as follows:

1. Reasonable geometrical parameters of surface texture can not only improve the static performance but also improve the dynamic performances and further enhance the stability of the journal bearing system, compared with the smooth journal bearing.

2. For the circumferential texture region, it is found that textured journal bearing can maximize the load-carrying capacity ($W$), maximum fluid film pressure ($P_{\text{max}}$), and lubrication flow ratio ($Q_2$) and minimize the friction coefficient ($f$), compared with smooth journal bearing in rising pressure texture region. It is important to emphasize that because of the external force acting in the $y$ direction, the direct stiffness coefficient ($K_{yy}$) and the critical mass ($M_c$) are the key factors to consider for dynamic performance; the direct stiffness coefficient ($K_{yy}$) and the critical mass ($M_c$) can also be improved simultaneously in rising pressure texture region.

3. For the texture depth ($h_t$), there is an optimum value of texture depth that maximize the load-carrying capacity ($W$), maximum fluid film pressure ($P_{\text{max}}$), lubrication flow ratio ($Q_2$), direct stiffness coefficient ($K_{yy}$), and the critical mass ($M_c$) and minimize the friction coefficient ($f$), respectively, compared with smooth journal bearing. Moreover, the optimal texture depth of each bottom profiles is also different.

4. For the texture bottom profile, the results indicate that different bottom profiles have different effects on the performances of textured journal bearings, such as flat, curved, isosceles triangle (T1), oblique triangle (T2), and oblique triangle (T3) bottom profiles. Moreover, for the five different bottom profiles, the performance of textured journal bearing with flat bottom profile is better than that of textured journal bearings with other bottom profiles.

5. For the eccentricity ratio, the larger eccentricity ratio ($\varepsilon$), minimum film thickness will be smaller, and the hydrodynamic pressure effect will be more obvious. And the eccentricity ratio ($\varepsilon$) is smaller, the relative difference of performance parameters of textured journal bearings provided by surface texture is more obvious, compared with smooth journal bearing. However, with the increase in eccentricity ratio ($\varepsilon$), the performances improvement become gradually weakened.

Based on the study of the performance characteristics of textured journal bearings, while designing, designers should focus on those optimal values of texture region, depth, and bottom profile given, which can maximize the static and dynamic performance and enhance the stability of journal bearings.

Appendix

The details of numerical procedure and methods are shown as follows:

1. Input data values of the eccentric ratio $\varepsilon$, the length $L/D$, and the texture and mesh parameters.

2. Guess the initial values of the dimensionless fluid film thickness $\bar{h}_{i,j}(\bar{\phi},\bar{Z})$, the dependent variable $\phi_{i,j}(\bar{\phi},\bar{Z})$, and the switch function $F_{i,j}(\bar{\phi},\bar{Z})$.

3. Calculate the dimensionless fluid film thickness $\bar{h}$ for the smooth and textures different bottom profiles by equations (8) and ((9a)–(9f)).

4. Here, we set the circumferential and axial mesh points to 314 and 100, respectively, to ensure that each texture occupies 15 mesh points in length and width. The finite difference method is used to discretize the steady-state Reynolds equation (10a), and the difference forms are as follows:

\[
\frac{\partial}{\partial \phi} \left( \bar{h} \frac{\partial (F_i \phi_i)}{\partial \phi} \right)_{i,j} = \frac{\bar{h}^3_{i+1/2,j}(F_i \phi_i)_{i+1,j} + \bar{h}^3_{i-1/2,j}(F_i \phi_i)_{i-1,j} - \left( \bar{h}^3_{i+1/2,j} + \bar{h}^3_{i-1/2,j} \right) (F_i \phi_i)_{i,j}}{(\Delta \phi)^2}, \tag{A.1a}
\]

\[
\frac{\partial}{\partial \phi} \left( \bar{h} \frac{\partial (F_i \phi_i)}{\partial Z} \right)_{i,j} = \frac{\bar{h}^3_{i,j+1/2}(F_i \phi_i)_{i,j+1} + \bar{h}^3_{i,j-1/2}(F_i \phi_i)_{i,j-1} - \left( \bar{h}^3_{i,j+1/2} + \bar{h}^3_{i,j-1/2} \right) (F_i \phi_i)_{i,j}}{(\Delta Z)^2}, \tag{A.1b}
\]

\[
\left( \frac{\partial \bar{h} [1 + (1 - F_i) \phi_i]}{\partial \phi} \right)_{i,j} = \frac{\bar{h}_{i+1/2,j}[1 + (1 - F_i) \phi_i]_{i+1/2,j} - \bar{h}_{i-1/2,j}[1 + (1 - F_i) \phi_i]_{i-1/2,j}}{\Delta \phi}. \tag{A.1c}
\]

(5) Further, the dependent variable $\phi_i$ and the switch function $F_i$ are obtained by solving the following difference equations with the over-relaxation iterative method. Here, $A_{i,j}, B_{i,j}, C_{i,j}, D_{i,j},$ and $E_{i,j}$ represent the coefficients of the discretized steady-state Reynolds equation (10a), and the relaxation
factor $\omega_\varphi$ is set to 1.9.

\[
\phi^k_{x(i,j)} = \frac{A_{(i,j)}\phi^k_{x(i+1,j)} + B_{(i,j)}\phi^k_{x(i-1,j)} + C_{(i,j)}\phi^k_{x(i,j-1)} + D_{(i,j)}\phi^k_{x(i,j-1)} - E_{(i,j)}}{A_{(i,j)} + B_{(i,j)} + C_{(i,j)} + D_{(i,j)}},
\]
(A.2a)

\[
\phi^{k+1}_{x(i,j)} = \left(1 - \omega_\varphi\right)\phi^k_{x(i,j)} + \omega_\varphi\phi^{k+1}_{x(i,j)}.
\]
(A.2b)

(6) The expression of the convergence condition of the dependent variable $\phi_x$ and the switch function $F_s$ is judged by the following equation. If not, update $\phi_x$ and $F_s$ and then return step 4. Here, the convergence precisions for $\text{tol}_x$ and $\text{tol}_F$ are set to $1.0 \times 10^{-5}$.

\[
\sum_{i,j} \left| \phi^{k+1}_{x(i,j)} - \phi^k_{x(i,j)} \right| \leq \text{tol}_x,
\]
(A.3a)

\[
\sum_{i,j} \left| F^{k+1}_{s(i,j)} - F^k_{s(i,j)} \right| \leq \text{tol}_F.
\]
(A.3b)

(7) Calculate the components of dimensionless load-carrying capacity $\bar{W}_x$ and $\bar{W}_y$ by equations (13a) and (13b). Further, the initial values $(\bar{\theta}_x, \bar{\theta}_y)$ are given, and calculate the attitude angle $\theta_x$ through the twofold secant method of equation as follows:

\[
E_{x,i,j} = (\Delta \varphi)^2 \left\{ -3 \sin \varphi - \frac{9}{h} \cos \varphi \frac{\partial \bar{h}}{\partial \varphi} \right\} F_s + 3\bar{h} \left\{ \left( \cos \varphi \frac{\partial \bar{h}}{\partial \varphi} + \sin \varphi \frac{\partial \bar{h}}{\partial \varphi} \right) \frac{\partial (F_s \varphi_i)}{\partial \varphi} + \frac{2R}{L} \cos \varphi \frac{\partial \bar{h}}{\partial Z} \frac{\partial (F_s \varphi)}{\partial Z} \right\},
\]
(A.6a)

\[
E_{y,i,j} = (\Delta \varphi)^2 \left\{ -3 \cos \varphi - \frac{9}{h} \sin \varphi \frac{\partial \bar{h}}{\partial \varphi} \right\} F_s + 3\bar{h} \left\{ \left( \sin \varphi \frac{\partial \bar{h}}{\partial \varphi} - \cos \varphi \frac{\partial \bar{h}}{\partial \varphi} \right) \frac{\partial (F_s \varphi_i)}{\partial \varphi} + \frac{2R}{L} \sin \varphi \frac{\partial \bar{h}}{\partial Z} \frac{\partial (F_s \varphi)}{\partial Z} \right\},
\]
(A.6b)

\[
E_{x,i,j} = (\Delta \varphi)^2 F_s \left( 6 \cos \varphi \right),
\]
(A.6c)

\[
E_{y,i,j} = (\Delta \varphi)^2 F_s \left( 6 \sin \varphi \right).
\]
(A.6d)

(10) Next, the fluid film perturbation pressures $P_{\xi}(\xi = \epsilon, \theta, \tilde{\epsilon}, \tilde{\theta})$ are also obtained by solving the following difference equations with the over-relaxation iterative method. Here, the relaxation factor $\omega_\xi$ is set to 1.9.

\[
P_{\xi(i,j)} = \frac{A_{i,j}P_{\xi(i+1,j)} + B_{i,j}P_{\xi(i-1,j)} + C_{i,j}P_{\xi(i,j-1)} + D_{i,j}P_{\xi(i,j-1)} - E_{i,j}}{A_{i,j} + B_{i,j} + C_{i,j} + D_{i,j}},
\]
(A.7a)

\[
P^{k+1}_{\xi(i,j)} = \left(1 - \omega_{\xi}\right)P^k_{\xi(i,j)} + \omega_{\xi}P^{k+1}_{\xi(i,j)}.
\]
(A.7b)
(11) The expression of the convergence condition of the fluid film perturbation pressures $P_t (\xi = \varepsilon, \theta, \dot{\theta})$ is judged by the following equation. If not, update $P_t$ and then repeat step 10. Here, the convergence precision $\text{tol}_t$ is set to $1.0 \times 10^{-5}$.

$$\frac{\sum_{i,j} |P_{t+1}^{(i,j)} - P_t^{(i,j)}|}{\sum_{i,j} |P_t^{(i,j)}|} \leq \text{tol}_t. \quad (A.8)$$

(12) Calculate the static, dynamic, and stability performance parameters for journal bearing by equations \((13a)–(13c)) \) to \((25))\) for journal bearing.

Data Availability

The figure and table data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that there are no conflicts of interest or personal relationships that would affect the work reported in this article.

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