PERFORMANCE ANALYSIS OF LONG GROOVED JOURNAL BEARING WITH SLIP, NO-SLIP, AND SLIP/NO-SLIP TEXTURED SURFACE CONFIGURATIONS

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ABSTRACT

The efficiency of the hydrodynamic journal bearing is one of the most important factors in operating high load machinery, thus it requires better performance and reliability. Applying boundary slip and surface texturing onto the bearing surface are some of the most common approaches in increasing bearing performance. In this paper, the comparative analysis is done by involving boundary slip, no-slip and the combination of slip and no-slip on the textured surface. The load capacity acts as a benchmark for the bearing’s performance based on several parameters. Based on the results obtained, increasing texture length will produce a positive improvement over the load capacity. Besides, various surface configurations act differently in different eccentricity ratio, thus providing better configuration selection on different operating conditions.

Keywords: Texture, slip region, load capacity, hydrodynamic journal bearings.

INTRODUCTION

Surface texturing and boundary slip are widely used in hydrodynamic lubrication to increase the performance of the bearing. The surface texture and slip configurations can be used to improve load capacity and the lifetime of the bearing itself.

Textured surface in lubricated contacts gives a big impact towards the load capacity increment and the reduction of motion friction. Tala-Ighil et al. [1] studied the influence of the texture distribution of the bearing at static load. He explored that full texturing gives a negative impact towards load capacity due to cavitation. Meanwhile, Kango et al. [2] showed that the negative texturing in forms of micro-cavities at different locations on the journal bearing does decrease the friction force. On the other hand, Sinanoglu et al. [3] applied the surface texturing on the journal instead of the bearing surface. From the theoretical and experimental investigation, the trapezoidal shaft tends to give better load capacity compared to the saw shaft. Wang et al. [4] investigated the triangular based surface texture towards the lubrication contact performance, which consisted of optimal dimple depth, direction, ratio, and size. In addition, Brizmer and Kligerman [5] also analyzed the laser surface texture (LST) of the journal bearing towards the load capacity and attitude angle performance.

Boundary slip is another set of approach in which has also been proven to improve the hydrodynamic lubrication contacts. Aurelian et al. [6] used the finite element method to analyze the wall slip influence towards static and dynamic load bearings. Chen et al. [7] in a similar method applied anisotropic boundary slip in the load capacity and friction coefficient performance. Ma et al. [8] studied the wall slip characteristics towards two-dimensional journal bearing performance and optimizations. A similar approach has been done by Wu [9] by using finite element method and programming algorithm method.

In other cases, the implementation of the slip and no-slip in one configuration has also been done to determine the significant improvement in a journal bearing performance. Lin et al. [10] in their research showed in their research that the location of the configuration affected the cavitation and load capacity. Wu et al. [11] studied the lubrication behavior towards mixed slip surface optimization in slider bearing performance. Salant and Fortier [12] meanwhile used the numerical analysis on the slip and no-slip surface in load capacity and friction force performance.

The objective of this paper is to perform comparative analysis in terms of load capacity of hydrodynamic journal bearing which includes the combination of texture and slip surface. In this paper, three sets of groove journal bearing configurations are selected and computed; full slip, no-slip, and slip-no-slip surface texture to determine the performance difference between the load carrying capacity. Therefore, the basic classical Reynolds equation is used and modified. Dimensionless load capacity is determined based on the boundary conditions applied in the modified equations. The overall results are collected and compared together to determine the variation of load capacity in different eccentricity ratios, in which varies in different sets of parameters such as a number of regions, texture length, slip ratio magnitude, and also groove depth.

METHOD OF ANALYSIS

Journal bearing is analyzed by applying sets of groove and no groove surfaces as shown in Figure. 1, in which the surface will be configured in three sets of configurations; full slip, no-slip and the combination of slip and no-slip. The computations from the sets of
expressions obtained are done by using MATLAB software to obtain the load capacity. The groove’s angular length is defined as $\theta_{1,2} - \theta_{1,1} = \theta_{2,2} - \theta_{2,1} = \cdots = \theta_{n,2} - \theta_{n,1}$, and the no groove angular length is known as $\theta_{1,3} - \theta_{1,2} = \theta_{2,3} - \theta_{2,2} = \cdots = \theta_{n,3} - \theta_{n,2}$. The textured length is defined as $\theta_t$. The Reynolds boundary conditions are applied to the modified Reynolds equation in this set up. The groove at the end of partially textured surface is noted as $\theta_g - \theta_{1,2}$.

The no groove and groove film thickness surfaces are expressed as $H + H_r$ and $H' = H + H_g$, respectively as shown in Figure-1, in which

$$H = (1 + e \cos \theta)$$

(1)

![Figure-1. Geometry of textured fragment through groove/no groove configuration of the grooved journal bearing.](image)

Three sets of configurations are carried out; full slip, no-slip, and slip/no-slip configuration. The parameters of slip “A” in the equation changed, per sets of calculation;

- Full Slip: $A=1$
- No-Slip: $A=0$
- Slip/No-Slip: $A=0$ at groove region, and $A=1$ at no-groove region.

The section for each groove-no groove region involve respective boundary conditions stated as;

$$P|_{\theta=0} = 0, P|_{\theta_{1,2}} = P_{1,2} \text{ and } P|_{\theta_{1,3}} = P_{1,3}$$

(2)

Dimensionless Reynolds equation for full texture is modified according to the configuration as;

$$\frac{d}{d\theta} \left[ \frac{H^2(H+2A) dP}{(H+A)d\theta} \right] = 6 \frac{d}{d\theta} \left[ \frac{H(H+2A)}{(H+A)} \right]$$

(3)

Dimensionless pressure profile is then obtained by integrating Equation. 3;

$$\frac{dP}{d\theta} (0 \leq \theta \leq \theta_{1,2}) = \frac{6(H+2A)}{H^3(H+4A)} \frac{d}{d\theta} \left[ \frac{H(H+2A)}{(H+A)} \right]$$

(4)

$$\frac{dP}{d\theta} (\theta_{1,2} \leq \theta \leq \theta_{1,3}) = \frac{6(H+4A)}{H^2(H+4A)} \frac{d}{d\theta} \left[ \frac{H(H+2A)}{(H+A)} \right]$$

(5)

Dimensionless pressure profile is obtained from integration from Equation. 4 and 5 with applied boundary conditions from Equation. 2;

$$P_{1,2}(0 \leq \theta \leq \theta_{1,2}) = P|_{\theta=0} + \left[ \frac{\theta_{1,2}}{6} \frac{(H+2A) d\theta}{H^2(H+4A)} - \frac{\theta_{1,2}}{12} \frac{(H+4A) d\theta}{H^2(H+4A)} \right]$$

(6)

$$P_{1,3}(\theta_{1,2} \leq \theta \leq \theta_{1,3}) = P|_{\theta=0} + \left[ \frac{\theta_{1,2}}{6} \frac{(H+2A) d\theta}{H^2(H+4A)} - \frac{\theta_{1,3}}{12} \frac{(H+4A) d\theta}{H^2(H+4A)} \right]$$

(7)

The dimensionless flow volume, $Q$ is obtained by simplifying dimensionless pressure yields from Reynolds equation and applied boundary conditions;

$$Q = \frac{\theta_{1,2}}{6} \frac{(H+2A) d\theta}{H^2(H+4A)} + \frac{\theta_{1,3}}{12} \frac{(H+4A) d\theta}{H^2(H+4A)}$$

(8)

The dimensionless pressure is then derived, gives the dimensionless radial and tangential load capacity as in Equation. 9 and 10;

$$W_e = - \int_{\theta_{1,2}}^{\theta_{1,3}} P_1 \cos \theta \ d\theta - \int_{\theta_{1,2}}^{\theta_{1,3}} P_2 \cos \theta \ d\theta - \int_{\theta_{1,2}}^{\theta_{1,3}} P_3 \cos \theta \ d\theta - \cdots - \int_{\theta_{1,2}}^{\theta_{1,3}} P_T \cos \theta \ d\theta$$

(9)

$$W_r = \int_{\theta_{1,2}}^{\theta_{1,3}} P_1 \sin \theta \ d\theta + \int_{\theta_{1,2}}^{\theta_{1,3}} P_2 \sin \theta \ d\theta + \int_{\theta_{1,2}}^{\theta_{1,3}} P_3 \sin \theta \ d\theta + \cdots + \int_{\theta_{1,2}}^{\theta_{1,3}} P_T \sin \theta \ d\theta$$

(10)

The load capacity in dimensionless form is then known as;

$$W = \sqrt{W^2_e + W^2_r}$$

(11)
RESULTS AND DISCUSSION

To determine the load capacity performance in a journal bearing with groove and no groove surfaces, specific parameters are required to be mentioned. \(A\) is the coefficient of slip with its magnitude of zero at the no-slip regions and one at the slip region; eccentricity ratio \(\varepsilon\) of 0.2 and 0.8; the groove depth \(H_g\) of 1, 2, 3 and 4; the slip length \((\theta_t)\) which defined as 40°, 80°, 120° and 160°; angular length \((\theta_g)\) of 180°; slip to no-slip ratio \(\gamma\) which is set to 0.2, 0.4, 0.6 and 0.8; and the number of slip regions \(n\) which is fixed to 2, 4, 6 and 8.

Figure-2(a). Dimensionless load capacity with groove to no-groove ratio at \(\theta_g=180°, \theta_t=120°, H_g=1, n=4\).

Figure-2(a) shows the performance of load capacity based on the groove to no groove ratio \(\gamma\) with two eccentricity ratios, 0.2 and 0.8. At the eccentricity ratio of 0.2, the slip surface shows greater load capacity despite having slight decrement compared to no-slip and slip/no-slip surface. Meanwhile, the slip/no-slip surface shows unimpressive load capacity as slip and no-slip surface shows greater result at low eccentricity ratio. However, the different pattern shows at high eccentricity ratio of 0.8, in which load capacity decreases along with groove to no groove ratio. At this point, no-slip surface shows greater load capacity, followed by slip/no-slip surface and full slip surface. In overall, high eccentricity ratio resulted high load capacity, but decreases due to higher ratio of groove to no groove magnitude.

At Figure-2(b), it shows that at 0.2 eccentricity ratio, full slip surface has better load capacity, followed by no-slip and slip/no-slip surface. At high eccentricity ratio of 0.8, the no-slip surface has slightly higher output than slip/no-slip surface, meanwhile full slip surface shows less improvement compared to other two. In overall, increasing groove depth will produce a slight decrease in load capacity for both eccentricity ratios. However, at high eccentricity ratio of 0.8, the decreasing pattern is clear from \(H_g=1\) to \(H_g=2\).

Figure-2(c). Dimensionless load capacity with number of slip region at \(\theta_g=180°, \theta_t=120°, H_g=1, \gamma=0.5\).

From the Figure-2(c), increasing the number of slip region shown not much improvement towards load capacity. However, a different configuration is having
high load capacity at different eccentricity ratio. At eccentricity ratio of 0.2, the full slip has higher load capacity, followed by no-slip and the slip/no-slip surface. But at the eccentricity ratio of 0.8, so slip surface has higher load capacity, followed by slip/no-slip surface and last is the full slip surface.

Figure-2(d). Dimensionless load capacity with groove/no groove texture length at $\theta_t = 180^\circ$, $H_0 = 1$, $\gamma = 0.5$, $n = 4$.

Figure-2(d) shows the significant improvement of load capacity based on groove/no groove texture length, $\theta_t$. At 0.2 eccentricity ratio, full slip surface shows positive improvement over $\theta_t$ compared to no-slip and slip/no-slip surface in which the load capacity is decreased. The load carrying capacity is increased for all defined configurations at eccentricity ratio of 0.8, Figure 2(d). In overall, the no-slip surface shows greater load capacity along $\theta_t$, followed by slip/no-slip and full slip surface. Therefore, increasing texture length $\theta_t$ leads to higher load capacity at high eccentricity ratio.

CONCLUSIONS

A recent study examines the comparative analysis of slip and no-slip configuration of the grooved journal bearing in terms of load capacity performance. From the comparisons and analysis made, it can be shown that:

- Full slip surface produces higher load capacity at low eccentricity ratio of 0.2. Meanwhile, slip/no-slip surface shows less load capacity at the same ratio.
- At high eccentricity ratio of 0.8, no-slip surface results in better load capacity for all factors, and full slip surface has lower load capacity performance compared with the other configurations.
- Increasing texture length $\theta_t$ leads to increase of load capacity performance at high eccentricity ratio for all three configurations.

NOMENCLATURE

$A =$ Slip parameter  
$c =$ Radial clearance  
$e =$ Eccentricity  
$\varepsilon =$ Eccentricity ratio ($n = \frac{e}{c}$)  
$h =$ Lubrication film thickness  
$H_g =$ Groove depth ($H_g = \frac{b}{c}$)  
$L =$ Bearing length  
$n =$ Number of slip region  
$P_f =$ Fluid film pressure at grooved region  
$P_n =$ Fluid film pressure at no-grooved region  
$r =$ Journal radius  
$R =$ Bearing radius  
$Q =$ Dimensionless flow volume  
$W =$ Dimensionless load carrying capacity  
$W_r =$ Dimensionless radial load carrying capacity  
$W_\theta =$ Dimensionless tangential load carrying capacity  
$\omega =$ Journal’s angular velocity  
$\theta =$ Angle throughout circumferential direction  
$\theta_e =$ Groove length  
$\theta_t =$ Textured length

ACKNOWLEDGEMENTS

The authors would like to thank Ministry of Higher Education Malaysia (MOHE) for supporting this research project under FRGS-0153AB-K49; Fundamental Research Grant Scheme.

REFERENCES


