A multiobjective optimization of journal bearing with double parabolic profiles and groove textures under steady operating conditions

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Abstract. The double parabolic profiles can help journal bearing to reduce bushing edge wear, but it also reduces load carrying capacity and increases friction loss. To overcome these drawbacks, in this study, a multiobjective optimization of journal bearing with double parabolic profiles and groove textures is researched under steady operating conditions using Taguchi and grey relational analysis methods. Firstly, a lubrication model with journal misalignment, elastic deformation, asperity contact, thermal effect is established and formation cause of drawbacks is illustrated. Then, an orthogonal test with considering six factors, i.e., groove number, groove depth, groove length, axial width of double parabolic profiles, radial height of double parabolic profiles and groove location is conducted, meanwhile the effects and significances of each factor on response variables are revealed. Finally, an optimal parameters combination of six factors is determined by grey relational analysis, which gives maximum load carrying capacity and minimum friction loss. Overall, this study may give guidance on journal bearing design to enhance its tribological performance.

Keywords: Double parabolic profiles / Groove textures / Load carrying capacity / Friction loss / Multiobjective optimization

1 Introduction

Journal bearing is a critical component in practical engineering. As the journal misalignment, deformation, machining and installation errors are unavoidable, bushing edge wear is found in some applications, as illustrated in Figure 1. Earlier study \cite{1} had shown the double parabolic profiles can reduce bushing edge wear, but it also reduces load carrying capacity and increases friction loss, which still need to be well solved.

For mechanical components, surface texturing has been widely used in the past decades to improve their contact performance \cite{2}. Specially, effects of surface textures on performance of journal bearing also attracted wide attentions to scholars. Ji et al. \cite{3} employed sinusoidal waves to characterize rough surface, which showed the greater roughness ratio can suppress the hydrodynamic effect of textures significantly. Hence, it is necessary to minimize roughness of textured surface. Gu et al. \cite{4} presented a mixed lubrication model to analyze the performance of groove textured journal bearing with non-Newtonian fluid operating from mixed to hydrodynamic lubrications. Their results showed the surface texturing can increase the asperity contacts, but the contact behaviors mainly arise in first cycle of start-up process. When journal bearing works under normal operating conditions, the wear due to asperity contacts will be small. Literatures \cite{5–10} showed the partial textures can positively affect bearing performance, but the optimal locations are some different depending on geometrical parameters and working conditions \cite{5}. Yu \cite{6} and Lin \cite{7} showed the textures located at rising phase of pressure field increases load carrying capacity, while the textures located at falling phase of pressure field reduces load carrying capacity. However, Tala-Ighil et al. \cite{8,9} showed the textures located in declining part of pressure field can generate extra hydrodynamic lift. Shinde and Pawar \cite{10} pointed out among three partial grooving configurations (90–180\(^\circ\), 90–270\(^\circ\), 90–360\(^\circ\)), the first configuration gives the maximum pressure increase while the last configuration gives the minimum frictional power loss.

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indicate the optimal location also depends on the optimization target.

Studies mentioned above have shown the texture performance is affected by many factors, which makes it very complicated to obtain the optimal design. To address this issue, some optimization methods are adopted by researchers, including the genetic algorithm [11–13], neural network [14] and sequential quadratic programming [15], while these methods are somewhat difficult in mathematics. By contrast, Taguchi method may provide a handy way for this issue. Taguchi assumes that introducing quality concepts at design stage may be more valuable than inspection after manufacturing. Hence, Taguchi method aims to optimize process to minimize quality loss with objective functions such as “the-nominal-the-best”, “the-larger-the-better”, or “the-smaller-the-better” depending on experimental objectives [16]. In practical engineering, this method uses the concept of orthogonal array to reduce the numbers of experiments, which facilitates to research multi-parameters concurrently and evaluate the effects of each parameter. Some studies [17–19] have already adopted Taguchi method to optimize surface textures for journal bearing to maximize load carrying capacity, minimize side leakage and friction loss.

Despite of remarkable progress on research of surface textures, few studies have been researched the journal bearing with double parabolic profiles and groove textures. The novelty of this study is to adopt the Taguchi and grey relational analysis methods to conduct a multiobjective optimization of journal bearing with double parabolic profiles and groove textures, i.e., double parabolic profiles are applied to eliminate bushing edge wear, while groove textures are applied to overcome the negative effects of the former, such as reduced load carrying capacity and increased friction loss. The results show this study may help journal bearing to enhance its tribological performance.

2 Lubrication model

2.1 Geometric model

Figure 2 illustrates the layout of double parabolic profiles and groove textures investigated in this study. In Figure 2, $B$ and $d_b$ are the bushing width and thickness; $L_y$ and $L_z$ are the axial width and radial height of double parabolic profiles, whose equation can be described as $d_z = \left(\frac{L_z}{L_y}\right)^{\frac{1}{2}} y^2$; $(\theta_s - \theta_e)$, $w_g$, $d_g$, $l_g$ and $w_e$, are the groove location, width, depth, length and gap, and their specific values will be introduced in Section 4.
2.2 Film thickness

Figure 3 illustrates a misaligned journal bearing under external moment $M_e$, whose lubricating oil is supplied through the axial oil feeding groove. For simplicity, only the misalignment in vertical plane $yoz$ is considered.

As the elastic module of journal is much higher than that of bushing, only elastic deformation of bushing surface is considered. Thus, the nominal film thickness $h$ is

$$ h = h_g + \delta_e $$

where $h_g$ is the nominal film thickness without elastic deformation, which is

$$ h_g = c + (e + y \tan \gamma) \cos(\theta - \varphi) + \delta_s + \delta_{tex} $$

where $c$ is the radial clearance, $e$ the eccentricity of the midplane, $\varphi$ the attitude angle of the midplane, $y$ the axial coordinate, $\gamma$ the misalignment angle, $\delta_s$ the clearance added by double parabolic profiles, $\delta_{tex}$ the clearance added by groove textures. Obviously, for journal bearing with plain profile, $\delta_s = \delta_{tex} = 0$, and for journal bearing with only double parabolic profiles, $\delta_{tex} = 0$.

In this study, the elastic deformation $\delta_e$ is obtained by Winkler/Column model [20], which gives a simpler way than finite element method [21] to estimate elastic deformation and has been used in literatures [22–24]. The model assumes the local elastic deformation is only dependent on local film pressure, as expressed in equation (3)

$$ \delta_e = \frac{(1 + v)(1 - 2v)}{(1 - v)} \frac{d}{E} p $$

where $v$ is the Poisson’s ratio of alloy layer, $E$ the elastic modulus of alloy layer, $p$ the film pressure, $d$ the thickness of alloy layer.

2.3 Reynolds equation

The Reynolds equation based on average flow model is utilized to determine the roughness effects on performance of journal bearing [25,26], as expressed in equation (4)

$$ \frac{\alpha}{\alpha x} \left( \phi_x \frac{h^3}{12 \mu} \frac{\partial h}{\partial x} \right) + \frac{\alpha}{\alpha y} \left( \phi_y \frac{h^3}{12 \mu} \frac{\partial h}{\partial y} \right) = \frac{U_1 + U_2}{2} \frac{\partial h_T}{\partial x} + \frac{U_1 - U_2}{2} \sigma \frac{\partial \phi_s}{\partial x} + \frac{\partial h_T}{\partial t} $$

where $\mu$ is the viscosity of lubricating oil, $p$ the film pressure, $U_1$ and $U_2$ the velocities of two surfaces, $\sigma$ the standard deviation of combined roughness, $\phi_x, \phi_y$ the pressure flow factors, $\phi_s$ the shear flow factor, $h_T$ the local film thickness.

For journal bearing under steady operating conditions, equation (4) can be expressed as followed by the variable transformation $x = R\theta$

$$ \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( \phi_x \frac{h^3}{\mu} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial y} \left( \phi_y \frac{h^3}{12 \mu} \frac{\partial p}{\partial y} \right) = 6\omega \frac{\partial h_T}{\partial \theta} + 6\omega \frac{\partial \phi_s}{\partial \theta} $$

where $\omega$ is angular velocity of journal.

2.4 Asperity contact pressure

The asperity contact model proposed by Greenwood and Tripp [27] is utilized here to estimate interaction effects of asperities, which is widely used in the analysis of rough surfaces contact of journal bearing. The asperity contact pressure $P_{asp}$ is given by

$$ P_{asp} = \frac{16 \sqrt{2} \pi}{15} (\eta \beta \sigma)^2 \sqrt{\frac{\sigma}{\beta}} E_c F_{2,5}(h/\sigma) $$

where $\eta$ is the number of asperities per unit area, $\beta$ the mean radius of curvature of the asperities, $\sigma$ the standard derivation, $E_c$ the composite elastic modulus, $F_{2,5}(h/\sigma)$ the Gaussian distribution function. Note the surface pattern parameter $\gamma$ is assumed as 1, which means the roughness structures are isotropic.

2.5 Friction loss

It is assumed when journal bearing operates in mixed lubrication, the total friction force consists of hydrodynamic friction force arising from the shearing of lubricating oil and asperity contact friction force [28]. Hence, the total friction force $f$ is

$$ f = \int_0^B \int_0^{2\pi} \left( \frac{\mu U}{h} \left( \phi_x + \phi_{fs} \right) + \phi_{fp} \frac{h}{2R} \frac{\partial p}{\partial \theta} + \mu_{asp} P_{asp} \right) Rd\theta dy $$

where $U = \omega R$, $\phi_p, \phi_{fs}, \phi_{fp}$ are shear stress factors, $\mu_{asp}$ boundary friction coefficient, here $\mu_{asp} = 0.02$. The friction loss $P_f$ can be calculated by

$$ P_f = fU. $$
2.6 Leakage flowrate

The leakage flowrate $Q_l$ from front-end plane of bearing and $Q_f$ from rear end plane of bearings are [21]

$$Q_1 = -\int_0^{2\pi} \phi y \frac{h^3}{12\mu} \frac{\partial p}{\partial y} \bigg|_{y=0} Rd\theta$$

$$Q_2 = -\int_0^{2\pi} \phi y \frac{h^3}{12\mu} \frac{\partial p}{\partial y} \bigg|_{y=B} Rd\theta$$

The total leakage flowrate $Q$ is

$$Q = |Q_1| + |Q_2|.$$

2.7 Thermal effect

As well known, the temperature of lubricating oil will increase and its viscosity will decrease during operations, so it is more accurate to adopt a variable viscosity model in calculation. In this study, an effective temperature is obtained based on the adiabatic flow hypothesis of lubricating oil, as shown below

$$T_e = T_i + \frac{k}{Q\rho c_i}$$

where $T_e$ is the effective temperature of lubricating oil, $T_i$ the inlet oil temperature, $P_f$ the friction loss, $Q$ the total leakage flowrate, $\rho$ the density of lubricating oil, $c_i$ the specific heat of lubricating oil, $k$ the correction factor and $k = 0.9$ [29]. This simple method avoids the complex computation of thermohydrodynamic lubrication and has been used in the literatures [24].

CD40 lubricating oil is used here and its viscosity-temperature equation can be expressed as

$$\ln \left( \frac{1}{\alpha} \ln(1000\mu) \right) = bT_e^2 + cT_e$$

where the unit of $\mu$ is Pa-s, and $a = 6.163, b = 8.721 \times 10^{-5}, c = -0.0455$, respectively. Once the effective temperature is obtained, the effective viscosity can be calculated by the equation (12).

2.8 Load equilibrium

In this study, the external load is assumed as a pure moment whose direction is parallel to $x$ axis, which only leads to a journal misalignment in vertical plane $yoz$. The static equilibrium of journal center can be described as

$$M_x + M_z = 0$$

where $M_x$ is the external moment, $M_z$ the resultant moment of hydrodynamic moment $M_{oil}$ and asperity contact moment $M_{asp}$, namely $M_z = M_{oil} + M_{asp}$.

The load equilibrium equations along $x$ and $z$ axis are

$$\begin{cases}
M_{ex} + M_{ez} = 0 \\
M_{ez} + M_{ex} = 0
\end{cases}$$

where $M_{ex}$ and $M_{ez}$ are the external moment along $x$ and $z$ axis, $M_{ez}$ and $M_{ex}$ the resultant moment along $x$ and $z$ axis, which can be expressed as follows

$$\begin{cases}
M_{ex} = M_{oilx} + M_{aspx} \\
M_{ez} = M_{oilz} + M_{aspz}
\end{cases}$$

where $M_{oilx}$ and $M_{oilz}$ are the hydrodynamic moment along $x$ and $z$ axis, $M_{aspx}$ and $M_{aspz}$ the asperity contact moment along $x$ and $z$ axis, which can be calculated by

$$\begin{cases}
M_{aspx} = \int_0^B \int_0^{2\pi} \frac{yp_{aspx} R \sin \theta d\theta dy}{C_{Wi,j}} \\
M_{aspz} = \int_0^B \int_0^{2\pi} \frac{yp_{aspx} R \cos \theta d\theta dy}{C_{Wi,j}}
\end{cases}$$

3 Numerical procedure and verification

Apply the finite difference method to discretize equation (5), then solve the difference equations by overrelaxation iterative method. Reynolds boundary conditions are used to determine the rupture zone of oil film, and the pressures in oil feeding groove and both bearing ends are assumed as zero. The discretized pressure can be calculated by

$$p_{i,j}^{(k_p+1)} = p_{i,j}^{(k_p)} - \omega \times \left[ p_{i,j}^{(k_p)} - \left( DD_{i,j} - CS_{i,j} p_{i+1,j}^{(k_p)} - CN_{i,j} p_{i-1,j}^{(k_p+1)} \right) / CC_{i,j} \right]$$

where $p_{i,j}^{(k_p+1)}$ is the film pressure for node $(i, j)$ at the $(k_p+1)$th iteration, $p_{i,j}^{(k_p)}$ the film pressure for node $(i, j)$ at the $k_p$th iteration, $\omega$ the overrelaxation factor, here $\omega = 1.5$. $DD_{i,j}$, $CS_{i,j}$, $CN_{i,j}$, $CE_{i,j}$, $CW_{i,j}$, $CC_{i,j}$ are the difference coefficients during the pressure solution.
The film pressure convergence criteria at the \( k_{th} \) iteration is given by

\[
\frac{\sum_{j=1}^{n_p} \sum_{i=1}^{n_p} |P_{ij}^{(k+1)} - P_{ij}^{(k)}|}{\sum_{j=1}^{n_p} \sum_{i=1}^{n_p} P_{ij}^{(k+1)}} \leq \varepsilon_p \tag{19}
\]

where \( \varepsilon_p \) is the allowable precision of pressure solution, \( n_p \) and \( n_g \) are the nodes numbers along circumferential and axial direction.

Based on the equation (11), the effective temperature convergence criteria at the \( k_{th} \) iteration is given by

\[
\frac{|T^{(k+1)} - T^{(k)}|}{T^{(k)}} \leq \varepsilon_t \tag{20}
\]

where \( \varepsilon_t \) is the allowable precision of effective temperature solution, here \( \varepsilon_t = 10^{-4} \).

When journal bearing operates in steady operations, it can be assumed the sum of hydrodynamic and asperity contact moments is approximately equal to the external moment, specifically, the motion of journal can be obtained by correcting eccentricity ratio \( \varepsilon (\varepsilon = e/c) \), attitude angle \( \phi \) and misalignment angle \( \gamma \). The correction strategies can be expressed as follows

\[
\varphi = \phi + \omega_\varphi \arctan \left( \frac{M_{tx}}{M_{tz}} \right) \tag{21}
\]

\[
\begin{align*}
\varepsilon &= \varepsilon - \omega_\varepsilon \left( \frac{M_t}{M_c} - 1 \right) \left( \frac{M_t}{M_c} \geq 1 \right) \\
\varepsilon &= \varepsilon + \omega_\varepsilon \left( 1 - \frac{M_t}{M_c} \right) \left( \frac{M_t}{M_c} < 1 \right) \tag{22}
\end{align*}
\]

\[
\begin{align*}
\gamma &= \gamma - \omega_\gamma \left( \frac{M_t}{M_c} - 1 \right) \left( \frac{M_t}{M_c} \geq 1 \right) \\
\gamma &= \gamma + \omega_\gamma \left( 1 - \frac{M_t}{M_c} \right) \left( \frac{M_t}{M_c} < 1 \right) \tag{23}
\end{align*}
\]

where \( M_c \) is the amplitude of \( M_{tx} \), \( M_t \) is the amplitude of \( M_{tx} \), \( \omega_\varphi \), \( \omega_\varepsilon \), \( \omega_\gamma \) are the correction factors of \( \varphi \), \( \varepsilon \), \( \gamma \), and \( \omega_\varphi = 0.9 \), \( \omega_\varepsilon = 10^{-2} \), \( \omega_\gamma = 10^{-5} \), respectively.

As mentioned above, the external load is a pure moment whose direction is parallel to the \( x \) axis. Accordingly, the equilibrium convergence criteria can be given by

\[
\frac{|M_{tx}|}{M_{tz}} \leq err_{xz} \tag{24}
\]

\[
\frac{|M_t - M_c|}{M_c} \leq err_M \tag{25}
\]

where \( err_{xz} \) and \( err_M \) are allowable precisions for the equilibrium calculation, here \( err_{xz} = 10^{-3} \), \( err_M = 10^{-3} \).

The whole computational process is shown in Figure 4, and the programing platform is Intel Core i7-4790 at 3.6GHz with 32GB RAM.

Before the following analysis, it is necessary to validate the model. The calculated pressures are compared with Ferron’s experimental results [30], as illustrated in Figure 5. The comparisons show the calculated results agree well with the experimental ones, which confirms the validity of this model.

4 Results and discussions

The detailed parameters of the journal bearing investigated in this study are listed in Table 1.

Mesh refinement analysis is conducted based on the journal bearing with plain profile. Various mesh schemes and corresponding minimum film thickness (\( h_{min} \)) are listed in Table 2. It can be seen that \( h_{min} \) is converged when the mesh is 1441 \times 181. Considering the solving time and accuracy, 1441 \times 181 mesh is adopted.

Here one typical case is conducted to prove the bushing edge wear can be reduced by double parabolic profiles. For journal bearing with double parabolic profiles, the chosen values of axial width \( L_a \) and radial height \( L_r \) are 10 mm and 6 \( \mu \)m, and the relative variation of each performance parameter is defined as

\[
\delta_{dpp,X} = \frac{X_{dpp} - X_{pp}}{X_{pp}} \times 100\% \tag{26}
\]

where \( \delta_{dpp,X} \) is the relative variation of parameter \( X \), pp means plain profile and dpp means double parabolic profiles. Note in this study, the maximum film pressure \( (P_{max}) \) is used to indirectly reflect the variations of load carrying capacity, smaller \( P_{max} \) means better load carrying capacity and vice versa.

The comparison results are listed in Table 3. It can be observed that although the \( \varepsilon \) and \( \gamma \) both increase to some extent, the \( h_{min} \) has sharply increased and the \( L_{aspmax} \) has reduced to zero in dpp case, which confirms its validity in term of reducing edge wear. Note \( \varphi \) has reduced a little, this variation trend indicates the bearing stability is also improved as the dpp reduces effective bearing width, which agrees with the common sense that using short bearing is favorable for stability improvement. However, the load carrying capacity is reduced as the dpp leads to a less capacity area. Hence, greater \( P_{max} \) is generated and hydrodynamic friction force also increases which causes greater \( P_c \). It is obvious that \( Q \) is increased as the dpp increases the film pressure gradient at bushing edge, so the comprehensive effects of greater \( P_c \) and \( Q \) lead to a tiny change of \( T_c \) based on equation (11).

Figure 6 illustrates the variations of film thickness and pressure between pp and dpp case, in Figure 6a, \( \Delta h = h_{dpp} - h_{pp} \) and in Figure 6b, \( \Delta P = P_{dpp} - P_{pp} \) It can be seen that, compared with pp case, the dpp case presents thicker oil film at bushing edge, which avoids the asperity contacts between journal and bushing surfaces, so the asperity contact pressure reduces to zero. However, in the region away from bushing edge, the oil film in dpp case is thinner.
than that in pp case, which causes greater film pressure here, i.e., the dpp can move the film pressure peak to inside region.

As mentioned above, although dpp can reduce bushing edge wear, it also reduces load carrying capacity and increases friction loss. The main motivation of this study is to overcome these drawbacks by groove textures, as illustrated in Figure 2. Considering the journal bearing with double parabolic profiles and groove textures (dppgt), six factors are involved in the analysis: groove number \( n_g \), depth \( d_g \), length \( l_g \), axial width of dpp \( L_y \), radial height of dpp \( L_z \), and location \( (\theta_a - \theta_e) \). Three levels for each factor are listed in Table 4. Note the total area of groove textures, \( S_g = w_g \times l_g \times n_g \), is constant, while the groove width \( w_g \), gap \( w_c \), and number \( n_g \) are different. The values of \( w_g \) and \( w_c \) for corresponding \( n_g \) are listed in Table 5.

Clearly the study of six factors at three optional levels needs 3 \( \times 3 \) simulation tests if full factorial designs are implemented. To reduce the computing cost, Taguchi method with orthogonal array \( L_{18} \) is adopted here, as shown in Table 6. \( L_{18} \) is commonly used and it is more concise than \( L_{27} \) array.

As we are mainly concerned with load carrying capacity and friction loss, only the results of \( P_{max} \) and \( P_f \) are given in Table 7. Note reference group are the results of dpp case (\( L_y = 5, 10, 15 \) mm, \( L_z = 3, 6, 9 \) \( \mu \)m, totally 9 cases, each case will be repeated twice).

Fig. 4. Flow chart of the computational process.
As can be seen from Table 7, compared with reference cases, the groove textures show positive effects only in No. 3, 4, 8, 11, 13 and 18 tests (shown in bold). Take the No. 8 test to explain this phenomenon: As illustrated in Figures 7 and 8, the groove textures can strengthen the hydrodynamic effect of lubricating oil and generate extra local pressure at the downstream of film pressure filed, which increases the load carrying capacity. Meanwhile, the friction force arising from the shearing of oil reduces with thicker oil film, which causes less friction loss. Note the groove locations of these six tests are all 200°–350°, which indicates the proper groove location is beneficial to performance enhancement.

While for the remaining tests, the groove textures show negative effects on either load carrying capacity or friction loss compared with reference cases. Take No. 5 and 15 tests.

| Table 1. Detailed parameters of the journal bearing. |
|------------------------------------------------------|
| Parameters                                 | Values     |
| Lubricating oil                      | CD40       |
| Oil feeding temperature (°C)         | 70         |
| Bearing diameter $D$ (mm)           | 230        |
| Bearing width $B$ (mm)              | 90         |
| Bearing bushing thickness $d_b$ (mm) | 5          |
| Radial clearance $c$ (mm)           | 0.14       |
| Oil feeding groove (°)              | 18         |
| Standard deviations of the roughness of the bearing surface $\sigma_b$ (µm) | 0.8 |
| Standard deviations of the roughness of the journal surface $\sigma_j$ (µm) | 0.4 |
| Elastic modulus of copper-lead-tin alloy layer $E_a$ (GPa) | 97 |
| Poisson’s ratio of copper-lead-tin alloy layer $\nu_a$ | 0.3 |
| Thickness of alloy layer $d$ (mm)   | 1          |
| External moment $M_e$ (Nm)          | 2000       |
| Rotary speed (rpm)                  | 1080       |

| Table 2. Mesh schemes for mesh refinement analysis. |
|-----------------------------------------------------|
| Scheme                                           |   |   |   |   |   |
| $n_u \times n_y$                  | 721×121 | 1081×151 | 1441×181 | 1441×211 | 1801×211 |
| $h_{\text{min}}$ (µm)              | 1.9417  | 1.9347  | 1.9095  | 1.9103  | 1.8974  |

| Table 3. Comparisons between journal bearing with plain profile (pp) and double parabolic profiles (dpp). |
|------------------------------------------------------------------------------------------------|
| Parameters                           | pp         | dpp        | $\delta_{\text{dpp}}$ (%) |
| Eccentricity ratio of midplane $\varepsilon$ | 0.9167     | 0.9263     | +1.05                |
| Attitude angle of midplane $\varphi$ (°)   | 189.7093   | 189.5775   | +0.07                |
| Misalignment angle $\gamma$ (10^{-2})     | 1.2416     | 1.2968     | +4.45                |
| Min. film thickness $h_{\text{min}}$ (µm)  | 1.9095     | 3.5935     | +88.20               |
| Max. asperity contact pressure $P_{\text{esm}}$ (MPa) | 0.6644  | 0          | -100.00              |
| Max. film pressure $P_{\text{max}}$ (MPa)   | 60.0312    | 68.1382    | +13.51               |
| Friction loss $P_f$ (W)                  | 4130.5236  | 4273.4494  | +3.46                |
| Leakage flowrate $Q$ (10^{-4} m³/s)      | 1.2720     | 1.3007     | +2.26                |
| Effective temperature $T_e$ (°C)         | 86.6712    | 86.8673    | +0.23                |

As can be seen from Table 7, compared with reference cases, the groove textures show positive effects only in No. 3, 4, 8, 11, 13 and 18 tests (shown in bold). Take the No. 8 test to explain this phenomenon: As illustrated in Figures 7 and 8, the groove textures can strengthen the hydrodynamic effect of lubricating oil and generate extra local pressure at the downstream of film pressure filed, which increases the load carrying capacity. Meanwhile, the friction force arising from the shearing of oil reduces with thicker oil film, which causes less friction loss. Note the groove locations of these six tests are all 200°–350°, which indicates the proper groove location is beneficial to performance enhancement.

While for the remaining tests, the groove textures show negative effects on either load carrying capacity or friction loss compared with reference cases. Take No. 5 and 15 tests.
Table 4. Six control factors and their optional levels.

| Factors  | Level 1 | Level 2 | Level 3 |
|----------|---------|---------|---------|
| \( n_g \) | 120     | 60      | 30      |
| \( d_g (\mu m) \) | 10      | 15      | 20      |
| \( i_g (mm) \) | 40      | 50      | 60      |
| \( L_g (mm) \) | 5       | 10      | 15      |
| \( L_z (\mu m) \) | 3       | 6       | 9       |
| \( \theta_c - \theta_e (\degree) \) | 180–330 | 190–340 | 200–350 |

Table 5. Groove sizes for their optional levels.

| Groove numbers \( n_g \) | 120 | 60 | 30 |
| Groove widths \( w_g (mm) \) | 1.5 | 3.0 | 6.0 |
| Groove gaps \( w_e (mm) \) | 1.0 | 2.0 | 4.0 |

Table 6. Orthogonal array L18.

| Test No. | \( n_g \) | \( d_g (\mu m) \) | \( i_g (mm) \) | \( L_g (mm) \) | \( L_z (\mu m) \) | \( \theta_c - \theta_e (\degree) \) |
|----------|---------|-------|----------|--------|--------|--------|
| 1        | 120     | 10    | 40       | 5      | 3      | 180–330 |
| 2        | 120     | 15    | 50       | 10     | 6      | 190–340 |
| 3        | 120     | 20    | 60       | 15     | 9      | 200–350 |
| 4        | 60      | 10    | 40       | 10     | 6      | 200–350 |
| 5        | 60      | 15    | 50       | 15     | 9      | 180–330 |
| 6        | 60      | 20    | 60       | 5      | 3      | 190–340 |
| 7        | 30      | 10    | 50       | 5      | 9      | 190–340 |
| 8        | 30      | 15    | 60       | 10     | 3      | 200–350 |
| 9        | 30      | 20    | 40       | 15     | 6      | 180–330 |
| 10       | 120     | 10    | 60       | 15     | 6      | 190–340 |
| 11       | 120     | 15    | 40       | 5      | 9      | 200–350 |
| 12       | 120     | 20    | 50       | 10     | 3      | 180–330 |
| 13       | 60      | 10    | 50       | 15     | 3      | 200–350 |
| 14       | 60      | 15    | 60       | 5      | 6      | 180–330 |
| 15       | 60      | 20    | 40       | 10     | 9      | 190–340 |
| 16       | 30      | 10    | 60       | 10     | 9      | 180–330 |
| 17       | 30      | 15    | 40       | 15     | 3      | 190–340 |
| 18       | 30      | 20    | 50       | 5      | 6      | 200–350 |

Fig. 6. (a) Differences \( \Delta h \), (b) Differences \( \Delta P \) between pp and dpp cases.
to explain this phenomenon: As illustrated in Figures 9–12, the groove textures located at 180°–330° (No. 5 test) and 190°–340° (No. 15 test) increase film thickness at main loading region, so the continuous film pressure generation is destroyed and greater pressure is generated in multi-peaks at untextured land (region between two adjacent grooves). This phenomenon is more evident when the groove textures locate at 180°–330°. Meanwhile, the hydrodynamic friction loss also reduces as explained previously. Moreover, the friction loss in No. 1, 9, 12, and 16 orthogonal tests are greater than those in reference cases, coincidentally the groove locations of these four tests are all 180°–330°, which indicates the improper groove location may bring harmful effects on texture performance.

Based on the orthogonal tests, the main effect analysis and analysis of variance (ANOVA) are performed to show the effects and significance of each factor [31]. The effects of six factors on load carrying capacity ($P_{\text{max}}$) are illustrated in Figure 13. It is observed the optimal parameters combination is at groove number 120, depth 10 μm, length 50 mm, axial width of dpp 10 mm, radial height of dpp 3 μm, and location 200°–350°, which may give a best load

Table 7. Results of orthogonal tests and reference group.

| Test No. | Orthogonal tests | Reference group |
|---|---|---|
| | $P_{\text{max}}$ (MPa) | $P_f$ (W) | $P_{\text{max}}$ (MPa) | $P_f$ (W) |
| 1 | 69.4134 | 4244.7385 | 63.2395 | 4192.0197 |
| 2 | 73.3438 | 4173.0918 | 68.1382 | 4273.4494 |
| 3 | **73.6209** | **4149.3208** | **73.6402** | **4353.0359** |
| 4 | **67.9519** | **4217.0836** | **68.1382** | **4273.4494** |
| 5 | 157.6918 | 4239.6102 | 73.6213 | 4353.0359 |
| 6 | 115.3453 | 3643.9945 | 63.2395 | 4192.0197 |
| 7 | 68.4509 | 4238.9667 | 66.8922 | 4257.0855 |
| 8 | **62.8481** | **4113.0399** | **64.0726** | **4207.6524** |
| 9 | 123.7151 | 4366.6101 | 68.7591 | 4283.2996 |
| 10 | 77.3296 | 4250.6529 | 68.7591 | 4283.2996 |
| 11 | **66.8692** | **4155.7142** | **66.8922** | **4257.0855** |
| 12 | 72.1152 | 4214.6034 | 64.0726 | 4207.6524 |
| 13 | **63.5710** | **4124.5873** | **63.9085** | **4205.7407** |
| 14 | 147.5468 | 4004.2271 | 65.4839 | 4233.6927 |
| 15 | 147.6417 | 3877.2633 | 71.4743 | 4318.6635 |
| 16 | 69.5520 | 4625.5756 | 71.4743 | 4318.6635 |
| 17 | 64.3493 | 4202.2618 | 63.9085 | 4205.7407 |
| 18 | **64.5257** | **4167.0737** | **65.4839** | **4233.6927** |

Fig. 7. (a) Film thickness of No. 8 test, (b) Film thickness of reference case 8.
Fig. 8. (a) Film pressure of No. 8 test, (b) Film pressure of reference case 8.

Fig. 9. (a) Film thickness of No. 5 test, (b) Film thickness of reference case 5.

Fig. 10. (a) Film pressure of No. 5 test, (b) Film pressure of reference case 5.
Fig. 11. (a) Film thickness of No. 15 test, (b) Film thickness of reference case 15.

Fig. 12. (a) Film pressure of No. 15 test, (b) Film pressure of reference case 15.

Fig. 13. Main effect plots of $P_{\text{max}}$ (MPa).
carrying capacity. However, this combination does not exist in orthogonal table and another computation is conducted, denoted as No. 19 test. The results of this test are $P_{max} = 64.0170$ MPa and $P_f = 4105.1540$ W, respectively, and $P_{max}$ of all 19 tests are illustrated in Figure 14. It can be seen the No. 8 test, i.e., groove number 30, depth 15 μm, length 60 mm, axial width of dpp 10 mm, radial height of dpp 3 μm, and location 200°–350°, gives a maximum load carrying capacity, with only 4.69% increases of $P_{max}$ than that in pp case.

Table 8 lists the results of ANOVA for load carrying capacity ($P_{max}$). The columns represent the sources, degrees of freedom (DF), sum of squares (SS), mean of squares (MS), F-values, and $F_{0.05}$ (2, 5). Table 8 shows the groove number $n_g$, location $\theta_s-\theta_e$ and depth $d_g$ are the significant factors at 95% confidence level as their F-values are greater than $F_{0.05}$ (2, 5). The percentage contributions (PCR) of all factors are also given in Table 8, which shows the groove number $n_g$ is the most important factor whose percentage contribution is 38.66%, followed by location $\theta_s-\theta_e$ and depth $d_g$ whose percentage contribution are 25.78% and 16.85%.

![Fig. 14. $P_{max}$ of all 19 tests.](image)

The effects of six factors on the friction loss ($P_f$) are illustrated in Figure 15. It is clearly that Figure 15 suggests the optimal parameters combination is in the No. 6 test, i.e., groove number 60, depth 20 μm, length 60 mm, axial width of dpp 5 mm, radial height of dpp 3 μm, and location 190°–340°, which gives a minimum friction loss. The $P_f$ of all 19 tests are illustrated in Figure 16, which also confirms the No. 6 test gives a minimum friction loss, 3643.9945 W, with 11.78% decrease than that in pp case.

Table 9 lists the results of ANOVA for friction loss ($P_f$), which shows the groove number $n_g$, depth $d_g$, axial width of dpp $L_m$, radial height of dpp $L_u$, and location $\theta_s-\theta_e$ are the significant factors at 95% confidence level. The percentage contributions of all factors are also given in Table 9, which shows the groove number $n_g$ is the most important factor whose percentage contribution is 33.45%, followed by the location $\theta_s-\theta_e$ depth $d_g$, axial width of dpp $L_m$ and radial height of dpp $L_u$ whose percentage contributions are 21.57%, 21.00%, 11.39% and 8.04% respectively.

Note the interactions between factors are not considered in. In fact, the interactions can be neglected if the orthogonal table is reasonably designed. The ANOVA shows that, compared with the six factors, the PCR of errors are small, 6.62% of $P_{max}$ and 2.70% of $P_f$, which indicates the interactions hidden in error are very limited. It can be found the interactions are also neglected in literatures [17–19].

From the above analysis, it can be observed the No. 8 test gives a maximum load carrying capacity while the No. 6 test gives a minimum friction loss. To find an optimal configuration, the grey relational analysis (GRA) method is used for this multiobjective optimization [19]. The main steps of GRA are listed as follows:

Step 1: Solution for normalized sequence.

As the optimization objectives are increasing load carrying capacity (smaller $P_{max}$) and reducing friction loss (smaller $P_f$), the “the-smaller-the-better” criterion is used to normalize the orthogonal test results between 0 and 1, as shown below

$$X_i^*(k) = \frac{\max(x_i(k)) - x_i(k)}{\max(x_i(k)) - \min(x_i(k))}$$

where the $X_i^*(k)$ is the normalized sequence, $x_i(k)$ the sequence of $P_{max}$ or $P_f$ for $k = 1, 2$ (1 for $P_{max}$ and 2 for $P_f$), and $i = 1, 2, ..., 19$ (test No.).
Step 2: Solution for deviation sequence.

The deviation sequence, denoted as $\Delta_0(k)$, is the absolute difference between the reference and normalized sequences, as shown below

$$\Delta_0(k) = |X_0^*(k) - X_1^*(k)|$$ \hspace{1cm} (28)

where $X_0^*(k)$ is the reference sequence.

Step 3: Solution for grey relational coefficient (GRC).

The GRC is calculated depending on the deviation sequence to describe the correlation between the reference and normalized sequences, as shown below

$$\xi_i(k) = \frac{\Delta_{\text{min}} + \zeta \Delta_{\text{max}}}{\Delta_0(k) + \zeta \Delta_{\text{max}}}$$ \hspace{1cm} (29)

where $\Delta_{\text{min}}$ and $\Delta_{\text{max}}$ are the minimum and maximum values of $\Delta_0(k)$, $\zeta$ the distinguishing factor between 0 and 1, here $\zeta = 0.5$, which means the increasing load carrying capacity and reducing friction loss are equally important.
Journal bearing can be also improved. However, it also has visible drawbacks, such as the reduced load carrying capacity (with 13.51% increase of $P_{\text{max}}$) and increased friction loss (with 3.46% increase of $P_f$).

- Compared with double parabolic profiles, double parabolic profiles with proper groove textures can increase the load carrying capacity due to extra local pressure. Meanwhile, the friction loss also reduces with thicker oil film. The improper groove textures can deteriorate the performance as they will destroy continuous film pressure generation and increase friction loss.

- The main effects analysis based on orthogonal test results shows that, for load carrying capacity, the No. 8 test gives a minimum $P_{\text{max}}$, 62.8481 MPa, with only 4.69% increase than that in the case of plain profile. While for friction loss, the No. 6 test gives a minimum $P_f$, 3643.9945 W, with 11.78% decrease than that in the case of plain profile.

- The ANOVA shows that, for load carrying capacity, the groove number $n_\text{g}$ and groove location $\theta_\text{u} - \theta_\text{g}$ are the significant factors at 95% confidence level, whose percentage contributions are 38.66%, 25.78%, 16.85% respectively. While for friction loss, the groove number $n_\text{g}$, groove location $\theta_\text{u} - \theta_\text{g}$, groove depth $d_\text{g}$, axial width of double parabolic profiles $L_y$, and radial height of double parabolic profiles $h_z$ are the significant factors at 95% confidence level, whose percentage contributions are 33.45%, 21.57%, 21.00%, 11.39%, and 8.04% respectively.

- The GRA shows that, the parameters combination of No. 8 test, i.e., groove number $30$, groove depth $15\mu m$, groove length $60\mm$, axial width of double parabolic profiles $10\mm$, radial height of double parabolic profiles $3\mu m$, and groove location $200^\circ - 350^\circ$, is the optimal solution: $P_{\text{max}}$ is 62.8141 MPa and $P_f$ is 4113.0399 W, with only 4.69% increase while 0.42% decrease than those in the case of plain profile, meanwhile the bushing edge wear is totally eliminated.

The cavitation area of oil film determined by Reynolds boundary conditions may be less accurate than mass-conservative treatment, which is a limitation of this study. In future work, cavitation effects will be treated in mass-conservative way, and other optimization methods will also be tried for journal bearing to find the optimal texture design.

5 Conclusions

In this study, based on Taguchi and GRA methods, a multiobjective optimization of journal bearing with double parabolic profiles and groove textures is researched under steady operating conditions. Following conclusions can be drawn from numerical results:

- Compared with plain profile, double parabolic profiles can eliminate bushing edge wear, and the stability of

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\[ \text{Fig. 17. GRGs of all 19 tests.} \]
δ_z Variation clearance caused by double parabolic profiles
δ_{tex} Variation clearance caused by groove textures
δ_c Elastic deformation of bushing surface
E Elastic modulus of the bushing
ν Poisson’s ratio of the bushing
μ Viscosity of lubricating oil
p Film pressure
U_1, U_2 Velocities of the two surfaces
σ Standard deviation of combined roughness
φ_x, φ_y Pressure flow factors
φ_f Shear flow factor
h_T Local film thickness
ω Angular velocity of journal
P_{asp} Asperity contact pressure
η The number of asperities per unit area
β The mean radius of curvature of the asperities
E Composite elastic modulus
F_{2,5}(h/σ) Gaussian distribution function
φ_f, φ_{fp}, φ_{fp} Shear stress factors
μ_{asp} Boundary friction coefficient
f Friction force
P_f Friction loss
Q_{f1} Leakage flowrate from the front end plane
Q_{f2} Leakage flowrate from the rear end plane
Q Total leakage flowrate
T_c Effective temperature of lubricating oil
T_i Inlet oil temperature
ρ Density of lubricating oil
c Specific heat of lubricating oil
M_e External moment
M_l Resultant moment
M_{oil} Hydrodynamic moment
M_{asp} Asperity contact moment
M_{asz}, M_{az} External applied moment along the x and z axes
M_{lxz}, M_{lz} Resultant moment along the x and z axes
M_{asz}, M_{az} External applied moment along the x and z axes
M_{asz}, M_{az} Hydrodynamic moment along the x and z axes
M_{asz}, M_{az} Asperity contact moment along the x and z axes
ω_s Overrelaxation factor
ω_p Allowable precision for the solution of film pressure
ω_l Allowable precision for the solution of the effective temperature
ε Eccentricity ratio of the midplane
M_e Amplitude of external moment
M_l Amplitude of resultant moment
ω_φ Correction factor of φ
ω_ε Correction factor of ε
ω_γ Correction factor of γ
er_{2z}, er_{M} Allowable precision for the calculation of load equilibrium
σ_y Standard deviations of the roughness of the bearing surface
σ_f Standard deviations of the roughness of the journal surface
n_y, n_g Numbers of nodes along the circumferential and axial direction
h_{min} Minimum film thickness
P_{max} Maximum film pressure
P_{asp max} Maximum asperity contact pressure
w_g Groove width
w_c Groove gap
l_g Groove length
n_g Groove numbers
n_{g1} × n_{g2} Mesh of single groove
d_g Groove depth
S_g Total area of groove textures
DF Degrees of freedom
SS Sum of squares
MS Mean of squares
X_y(k) Normalized sequence of P_{max} or P_f
x(k) Sequence of P_{max} or P_f
X_0(k) Reference sequence
Δ_{min}, Δ_{max} Minimum and maximum values of Δ_{01}(k)
ξ Weight of P_{max} or P_f
γ_i Grey relational coefficient
γ_g Grey relational grade

Abbreviations

pp Plain profile
dpp Double parabolic profiles
dppgt Double parabolic profiles with groove textures
ANOVA Analysis of variance
GRA Grey relational analysis
GRC Grey relational coefficient
GRG Grey relational grade

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