Research Article

Vibration Characteristics of Hydrodynamic Fluid Film Pocket Journal Bearings

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Theoretical analyses of hydrodynamic fluid film bearings with different bearing profiles rely on solutions of the Reynolds equation. This paper presents an approach used for analysing the so-called pocket bearings formed from a combination of offset circular bearing profiles. The results show that the variation of the dynamic bearing characteristics with different load inclinations for the pocket bearings is less than that for the elliptic bearing counterpart. It is shown that the natural frequencies as well as the critical speeds, and hence the vibrational behaviour, can also be significantly different for an industrial rotor supported by the different bearings.

1. Introduction

In order to increase productivity and reduce machine downtime, hydrodynamic bearings are frequently used to support high-speed rotating machinery where reliability, long running life, and minimum vibration levels are of primary concern. Simple circular bore journal bearings sometimes cause instability, which may result in catastrophic failure of the machinery. To improve resistance to such failure, different bearing types with different bearing clearance profiles have been developed and used in practice. Typical examples are elliptic bearings and tilting pad bearings, the former providing stabilizing preload and the latter minimizing the troublesome cross-coupled bearing forces. Another bearing type, formed by a combination of offset circular bearing profiles and referred to as a pocket bearing, is also sometimes used in machinery such as turbogenerators.

Theoretical analyses of hydrodynamic fluid film bearings with different bearing profiles rely on solutions of the Reynolds equation. This paper presents an approach used for analysing the so-called pocket bearings formed from a combination of offset circular bearing profiles. The results show that the variation of the dynamic bearing characteristics with different load inclinations for the pocket bearings is less than that for the elliptic bearing counterpart. It is shown that the natural frequencies as well as the critical speeds, and hence the vibrational behaviour, can also be significantly different for an industrial rotor supported by the different bearings.

2. Theory

2.1. General Theory. The schematic of a simple circular hydrodynamic fluid film bearing is shown in Figure 1. Following the usual assumptions of hydrodynamic lubrication theory, the Reynolds equation can be written as [1]

\[
\frac{\partial}{\partial X} \left( h^3 \frac{\partial p}{\partial X} \right) + \frac{\partial}{\partial Z} \left( h^3 \frac{\partial p}{\partial Z} \right) = -6\mu U \frac{\partial h}{\partial X} + 12\mu V,
\]
where \[ h = C - z \sin \psi - y \cos \psi, \] (2)
\[ \psi = \theta + \phi + \frac{\pi}{2}, \] (3)

Upon nondimensionalisation, (1) becomes
\[
\frac{\partial}{\partial \psi} \left( \frac{\partial \bar{P}}{\partial \psi} \right) + \left( \frac{D}{L} \right)^2 \frac{\partial}{\partial \bar{Z}} \left( \frac{\partial \bar{P}}{\partial \bar{Z}} \right) = -\left[ (2\bar{\psi} - \bar{\psi}) \sin \psi + (2\bar{\psi} + \bar{\psi}) \cos \psi \right],
\] (4)
in which
\[ \bar{Z} = \frac{Z}{L/2}, \]
\[ \bar{P} = \frac{P}{\mu \omega (R/C)^2}, \]
\[ \bar{h} = \frac{h}{C} = 1 - \bar{Z} \sin \psi - \bar{\psi} \cos \psi, \quad \bar{\psi} = \frac{y}{C}, \] and so forth,
\[ \bar{h} = -\bar{Z} \sin \psi - \bar{\psi} \cos \psi. \] (5)

The pressure boundary conditions are that \( \bar{P} = 0 \) at \( \bar{Z} = \pm 1 \) (bearing edges) and at \( \psi_1 \) and \( \psi_2 \) (the boundary coordinates at the onset and end of the fluid film, resp.). The Reynolds cavitation boundary condition is used to define the cavitation region [1].

Upon integration of the film pressure, the fluid film force components in y and z directions are
\[
\begin{pmatrix} F_y \\ F_z \end{pmatrix} = -\int_{\psi_1}^{\psi_2} \left[ P \left( \cos \psi \right) R dZ d\psi \right] = \begin{pmatrix} 0 \\ W \end{pmatrix}. \] (6)

The corresponding nondimensional force components are [3]
\[
\begin{pmatrix} f_y \\ f_z \end{pmatrix} = 3 \int_{\psi_1}^{\psi_2} \left[ \bar{P} \left( \cos \psi \right) \sin \psi \right] d\bar{Z} d\psi = \begin{pmatrix} 0 \\ 1 \end{pmatrix}. \] (7)

For small perturbations \( \Delta \bar{\psi}, \Delta \bar{Z}, \Delta \bar{\psi}, \Delta \bar{\psi} \) in the y and z directions about the equilibrium position, the dynamic bearing coefficients are defined as
\[
\begin{pmatrix} \Delta f_y \\ \Delta f_z \end{pmatrix} = \begin{pmatrix} K_{zz} & K_{zy} \\ K_{yz} & K_{yy} \end{pmatrix} \begin{pmatrix} \Delta \bar{\psi} \\ \Delta \bar{\psi} \end{pmatrix} - \begin{pmatrix} C_{zz} & C_{zy} \\ C_{yz} & C_{yy} \end{pmatrix} \begin{pmatrix} \Delta \bar{Z} \\ \Delta \bar{Z} \end{pmatrix}. \] (8)

Ignoring higher-order terms, the Taylor series expansion about the equilibrium position \( (\bar{\psi}, \bar{Z}, \bar{\psi}, \bar{Z} = 0) \) yields \( K_{zz} = -\partial f_z / \partial \bar{\psi}, C_{zy} = -\partial f_z / \partial \bar{\psi}, \) and so forth.

Note that for more than one pad or clearance profile, it is the resultant of the forces on all the pads or clearance profiles in the y and z directions which equals zero and W, respectively, in (6); it is the perturbation in these resultant force components which is used to evaluate the dynamic bearing coefficients.

The above is the summary of the general theory for determining the static and dynamic bearing characteristics (i.e., Sommerfeld number, attitude angle, stiffness coefficients, and damping coefficients). Different bearing configurations impose different geometric relationships for the position of the journal in the bearing.

2.2. Pocket Bearing Configurations. Two types of pocket bearings are to be considered. The first type, formed in a similar way to an elliptic bearing (Figure 2), is schematically shown in Figure 3 and is referred to as type 1. The two halves of a bearing block are held together, separated by shims corresponding to the ellipticity \( 2d_1 \). A circular hole of diameter \( D \) is cut, and the shims are then removed to form a normal elliptic bearing with the top pad centre a distance \( d_1 \) below, and the bottom pad centre a distance \( d_1 \) above the bearing centre. Concentric arc portions (the pockets) of
a specified angular extent are then machined as shown in Figure 3. Such a pocket bearing has three curvature centres: one above one below the bearing centre for the side arcs just like the elliptic bearings and one coinciding with the bearing centre for the pockets.

The second type of pocket bearing, referred to as type 2, is shown schematically in Figure 4. It can be conveniently machined by NC machines. It also has three arc centres, but the two for the side arcs are located at a distance \( d_2 \) to the left and right of the bearing centre, the left centre for the left arcs on both the top and bottom pads, and the right one for the right arcs, again on both pads. The angular extent of the side arcs is determined by the arc radii and the centre offset \( d_2 \).

The two types of the pocket bearings are expected to have similar vibration characteristics.

Normally, the bearing clearance \( C \) is used to nondimensionalise the film thickness and the relative journal to bearing displacements. Due to the bearing profile, each pocket bearing has two clearances: \( C_p \) for the pocket portion and \( C_e \) for the side arcs. Using different clearances will produce different nondimensional bearing coefficients, but for a given bearing, the dimensional coefficients should not be affected. Following the approach for elliptic bearings [5], \( C_e \) could be chosen as the nondimensionalising parameter. However, the type-2 pocket bearing does not have such a corresponding clearance, though it also has two clearances: \( C_p \) for the pocket portion and \( C_s \) for the side arcs. Hence, in order to have comparable results, the pocket clearance \( C_p \), common to both types of pocket bearings, is used as the nondimensionalising parameter.

2.3. Type-One Pocket Bearing. From (2), the nondimensional film thickness for the side arcs becomes

\[
\bar{h}_e = \frac{C_e}{C_p} - \bar{y}_j \sin \psi - \bar{y}_1 \cos \psi
\]

and for the pocket arcs

\[
\bar{h}_p = 1 - \bar{y}_j \sin \psi - \bar{y}_1 \cos \psi.
\]

Note that in these equations, \( \bar{y} \) and \( \bar{y}_1 \) are the nondimensional journal centre coordinates from the corresponding arc centres.

For the pocket portion:

\[
\bar{y}_p = \varepsilon \sin (\phi - \beta),
\]

\[
\bar{y}_1 = -\varepsilon \cos (\phi - \beta),
\]

and for the bottom side arcs,

\[
\bar{y}_b = \varepsilon_b \sin (\phi_b - \beta),
\]

\[
\bar{y}_b = -\varepsilon_b \cos (\phi_b - \beta),
\]
where, from Figure 5,
\[ \varepsilon_b = \sqrt{\varepsilon^2 + \delta^2 + 2\varepsilon\delta \cos \phi}, \]
\[ \sin \phi_b = \frac{\varepsilon}{\varepsilon_b} \sin (180^\circ - \phi). \] (13)

For the top side arcs,
\[ \varepsilon_t = \sqrt{\varepsilon^2 + \delta^2 - 2\varepsilon\delta \cos \phi}, \]
\[ \sin(180^\circ - \phi_t) = \frac{\varepsilon}{\varepsilon_t} \sin \phi. \] (15)

The clearances \( C_e \) and \( C_p \) are related by the pocket arc extent \( \alpha \) and the ellipticity of the side arcs \( d_1 \). As shown in Figure 6, at the intersection between the side and pocket arcs, one has
\[ R_e^2 = R_p^2 + d_1^2 - 2R_p d_1 \cos \left(180^\circ - \frac{\alpha}{2}\right). \] (16)

Solving for \( R_p \) and omitting higher-order terms
\[ R_p = -d_1 \cos \left(\frac{\alpha}{2}\right) + \sqrt{R_e^2 - d_1^2 \sin^2 \left(\frac{\alpha}{2}\right)} \approx R_e - d_1 \cos \left(\frac{\alpha}{2}\right). \] (17)

The journal radius can be expressed as
\[ r = R_e - C_e = R_p - C_p. \] (18)

Substitution of (17) into (18) gives
\[ C_e - C_p = R_e - R_p \approx d_1 \cos \left(\frac{\alpha}{2}\right), \] (19)
or
\[ \frac{C_e}{C_p} \approx 1 + \frac{d_1 \cos(\alpha/2)}{C_p} = 1 + d_1 \cos \left(\frac{\alpha}{2}\right) \] (20)
which is always greater than 1.

2.4. Type-Two Pocket Bearing. Again, following an approach similar to that for the type 1 bearings, one has for the side arcs
\[ \overline{h}_s = \frac{C_e}{C_p} - \overline{\psi} \sin \psi - \overline{\psi} \cos \psi \] (21)
and for the pocket arcs
\[ \overline{h}_p = 1 - \overline{\psi} \sin \psi - \overline{\psi} \cos \psi. \] (22)
The clearances \( C_s \) and \( C_p \) are related by the pocket arc extent \( \alpha \) and the offset of the side arc centers \( d_2 \). As shown in Figure 7, at the intersection between the side and pocket arcs, one has

\[
R_c^2 = R_p^2 + d_2^2 - 2R_p d_2 \cos \left(90^\circ - \frac{\alpha}{2}\right).
\]

Solving for \( R_p \) and omitting higher-order terms,

\[
R_p = d_2 \sin \left(\frac{\alpha}{2}\right) + \sqrt{R_c^2 - d_2^2 \cos^2 \left(\frac{\alpha}{2}\right)} \approx R_c + d_2 \sin \left(\frac{\alpha}{2}\right).
\]

(24)

The journal radius can be expressed as

\[
r = R_s - C_s = R_p - C_p.
\]

(25)

From (24) and (25), this gives

\[
C_p - C_s \approx d_2 \sin \left(\frac{\alpha}{2}\right),
\]

(26)

or

\[
\frac{C_s}{C_p} \approx 1 - \frac{d_2 \sin(\alpha/2)}{C_p} = 1 - \delta_2 \sin \left(\frac{\alpha}{2}\right).
\]

(27)

To relate the offset \( d_2 \) to the ellipticity \( d_1 \), it is assumed that, apart from the same pocket arc extent \( \alpha \), the horizontal widths of the two bearings are the same, that is,

\[
R_c + d_2 = \sqrt{R_c^2 - d_1^2} \approx R_c.
\]

(28)

Substitution of (24) and (17) into (28) gives

\[
R_p - d_2 \sin \left(\frac{\alpha}{2}\right) + d_2 = R_p + d_1 \cos \left(\frac{\alpha}{2}\right),
\]

(29)

or

\[
d_2 = d_1 \frac{\cos(\alpha/2)}{1 - \sin(\alpha/2)}.
\]

(30)

The relationships between \( \psi, \zeta \) and \( \varepsilon, \phi \) are the same as before. However, the calculations for \( \varepsilon_l, \varepsilon_r, \phi_l \), and \( \phi_r \) need different expressions. From Figure 8,

\[
\varepsilon_l = \sqrt{\varepsilon^2 + \delta_l^2 + 2\varepsilon \delta_l \sin \phi},
\]

\[
\sin(90^\circ - \phi_l) = \frac{\varepsilon}{\varepsilon_l} \sin(90^\circ + \phi),
\]

\[
\varepsilon_r = \sqrt{\varepsilon^2 + \delta_r^2 - 2\varepsilon \delta_r \sin \phi},
\]

\[
\sin(90^\circ + \phi_r) = \frac{\varepsilon}{\varepsilon_r} \sin(90^\circ - \phi).
\]

(31)

### 3. Solution Procedures

To calculate the static and dynamic characteristics of the pocket bearings, Gauss-Seidel iteration with successive over relaxation is used to solve the finite difference formulation of the Reynolds equation [5], thereby ensuring that the Reynolds condition is used as the cavitation boundary. All results here assume same zero gauge inlet, outlet, and cavitation pressures.

![Figure 9](image-url) Comparison of the nondimensional bearing characteristics for the two types of pocket bearings ((1) and (2) correspond to type 1 and 2, resp.)
Figure 10: Nondimensional bearing characteristics for elliptic and type-1 pocket bearings at $\varepsilon = 0.2$. 
Figure 11: Nondimensional bearing characteristics for elliptic and type-1 pocket bearings at $\varepsilon = 0.4$. 
4. Results

Figure 9 compares the nondimensional static and dynamic characteristics of the two types of pocket bearings as a function of load inclination beta ($\beta$), using the same base clearance $C_p$ and the the same pocket extent of $\alpha = 60^\circ$. Apart from some difference in the attitude angle, the nondimensional characteristics are virtually identical.

Figures 10 and 11 compare the bearing characteristics of the type-1 pocket bearing to a similar elliptic bearing as functions of the load inclination angle $\beta$ for given eccentricities $\varepsilon = 0.2$ and 0.4, respectively. It can be seen that in general there is less variation in the dynamic bearing characteristics of the pocket bearing due to the change of $\beta$.

A sample vibration analysis is also performed for an industrial rotor subjected to gravity loading and supported on either type of pocket bearings or similar elliptic bearings. The rotor is shown schematically in Figure 12. Table 1 lists the corresponding data used to characterise the rotor and the bearings. In house vibration analysis software was used to determine the natural frequencies and mode shapes over a speed range up to 5000 rpm. Table 2 lists these natural frequencies, and Figure 13 shows the Campbell diagrams for the elliptic bearing supports and for the type-1 pocket bearing supports.

It can be seen that the system behaves quite differently with the different types of bearing supports. The rotor with the elliptic bearing has three natural frequencies but two critical speeds in the speed range; while the rotor with the type-1 pocket bearing has four natural frequencies but only one critical speed. The second mode in both cases is a backward whirl and is therefore unlikely to be excited. Using equivalent clearances for the different pocket bearing models, the natural frequencies in Table 2 indicate that the two types
Table 1: Sample rotor bearing data.

<table>
<thead>
<tr>
<th>No.</th>
<th>Length (m)</th>
<th>Diameter (m)</th>
<th>Disk mass (kg)</th>
<th>Disk Ip (kg-m²)</th>
<th>Disk Id (kg-m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
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<td>0</td>
<td>0.0358</td>
<td>0.0436</td>
</tr>
<tr>
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<td>0.18</td>
<td>0</td>
<td>0.1294</td>
<td>0.1329</td>
</tr>
<tr>
<td>3</td>
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<td>0.18</td>
<td>0</td>
<td>0.1193</td>
<td>0.1131</td>
</tr>
<tr>
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<td>0.18</td>
<td>0</td>
<td>0.1193</td>
<td>0.1131</td>
</tr>
<tr>
<td>5</td>
<td>0.115</td>
<td>0.258</td>
<td>8.392</td>
<td>0.5447</td>
<td>0.3336</td>
</tr>
<tr>
<td>6</td>
<td>0.115</td>
<td>0.277</td>
<td>1.185</td>
<td>0.5447</td>
<td>0.3336</td>
</tr>
<tr>
<td>7</td>
<td>0.115</td>
<td>0.277</td>
<td>1.185</td>
<td>0.5447</td>
<td>0.3336</td>
</tr>
<tr>
<td>8</td>
<td>0.106</td>
<td>0.328</td>
<td>11.87</td>
<td>1.291</td>
<td>0.7217</td>
</tr>
<tr>
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<td>0.921</td>
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<td>4.408</td>
</tr>
<tr>
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<td>0.5</td>
<td>47.73</td>
<td>6.559</td>
<td>3.17</td>
</tr>
<tr>
<td>13</td>
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<td>0.556</td>
<td>93.86</td>
<td>20.57</td>
<td>8.875</td>
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<td>0.556</td>
<td>139.5</td>
<td>30.48</td>
<td>14.42</td>
</tr>
<tr>
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<td>0.2</td>
<td>0.556</td>
<td>111</td>
<td>25.38</td>
<td>12.52</td>
</tr>
<tr>
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<td>0.556</td>
<td>128.4</td>
<td>28.57</td>
<td>14</td>
</tr>
<tr>
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<td>0.556</td>
<td>126.6</td>
<td>28.19</td>
<td>13.88</td>
</tr>
<tr>
<td>18</td>
<td>0.216</td>
<td>0.556</td>
<td>122.8</td>
<td>27.57</td>
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<td>0.556</td>
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<td>27.05</td>
<td>13.66</td>
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<tr>
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<td>0.556</td>
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<td>30.62</td>
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<tr>
<td>24</td>
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<td>0.704</td>
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<tr>
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</tr>
<tr>
<td>27</td>
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<td>0.353</td>
<td>1.467</td>
<td>2.056</td>
<td>1.335</td>
</tr>
<tr>
<td>28</td>
<td>0.168</td>
<td>0.353</td>
<td>1.467</td>
<td>2.056</td>
<td>1.335</td>
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<tr>
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<tr>
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<td>36</td>
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<td>0.58</td>
<td>131</td>
<td>22.1</td>
<td>11.16</td>
</tr>
</tbody>
</table>

Bearing: \( \delta = 0.5, (\alpha = 60^\circ \) for pocket bearings.

<table>
<thead>
<tr>
<th>No.</th>
<th>Length (m)</th>
<th>Diameter (m)</th>
<th>Clearance (mm)</th>
<th>Viscosity (Ns/m²)</th>
<th>Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.092</td>
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<td>0.1327</td>
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<td>0.2322</td>
<td>0.014</td>
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</table>

of the pocket bearings with similar dimensions could have quite different effects on the system vibration characteristics.

5. Conclusions

An approach to evaluate the static and dynamic bearing characteristics of pocket type bearings, suited for subsequent steady-state vibration analysis of rotating machinery involving such bearings, is presented.

Compared to similar elliptic bearings, the pocket bearings tend to provide less fluctuation in the dynamic bearing coefficients for different load inclinations and may produce significantly different vibration behaviour in a given rotor system.

The two types of pocket bearings investigated here (produced by different machining procedures), apart from some difference in attitude angle, have virtually identical static and dynamic bearing characteristics.
Table 2: Damped natural frequencies of a rotor supported by elliptic or pocket bearings ((1) refers to type 1, etc.).

<table>
<thead>
<tr>
<th>Speed (rpm)</th>
<th>Elliptic</th>
<th>Pocket</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1000</td>
<td>2000</td>
</tr>
<tr>
<td>Mode 1</td>
<td>−19.91, 137.5</td>
<td>−11.17, 143.9</td>
</tr>
<tr>
<td>Mode 2</td>
<td>−4.064, 211.8</td>
<td>−5.900, 207.7</td>
</tr>
<tr>
<td>Mode 3</td>
<td>−129.3, 217.4</td>
<td>−89.73, 261.5</td>
</tr>
</tbody>
</table>

Nomenclature

- **C**: Radial clearance = R − r
- **C_{yy}, · · ·**: Nondimensional bearing damping coefficients
- **D**: Bearing diameter
- **d**: Ellipticity or offset
- **e**: Eccentricity
- **F_y, F_z**: Film force components
- **f_y, f_z**: Nondimensional force components
- **h**: Film thickness
- **K_{yy}, · · ·**: Nondimensional bearing stiffness coefficients
- **L**: Bearing width
- **O**: Centres
- **P**: Pressure (gauge)
- **R**: Bearing radius
- **r**: Journal radius
- **S**: Sommerfeld number = μωRL(R/C)^2/(πW)
- **t**: Time
- **U, V**: Velocities at journal surface in X and Y directions, respectively
- **W**: External load
- **X, Y, Z**: Cartesian coordinates at bearing surface
- **x, y, z**: Cartesian coordinates at curvature centers
- **α**: Pocket extent
- **β**: Load inclination from vertical
- **Δ**: Perturbation; variation in
- **δ**: Nondimensional ellipticity/offset = d/C
- **ε**: Nondimensional eccentricity = e/C
- **μ**: Absolute viscosity
- **φ**: Attitude angle
- **θ**: Angular coordinate from line of centres = X/R
- **τ**: Nondimensional time = ωt
- **ω**: Speed
- **ψ**: Angular coordinate from y axis

Superscripts (If Not Otherwise Defined):

- **−**: Nondimensional
- **′**: Differentiation with respect to nondimensional time τ
- **′′**: Differentiation with respect to dimensional time t

Subscripts (If Not Otherwise Defined):

- **o**: Steady state
- **b, j**: Bearing, journal
- **b, t**: Bottom lobe, top lobe
- **e, p, s**: Elliptic arc, pocket arc, side arc
- **l, r**: Left arc, right arc
- **1, 2**: Pocket type 1, pocket type 2.

References
