The restrictive effects of capillary compensation on the stability of the Jeffcott rotor-hybrid bearing system


a Department of Mechanical Engineering, Nanya Institute of Technology, Taiwan.
b Department of Mechanical Engineering, Chung Yuan Christian University, Chung-Li 320, Taiwan.

Received 24 April 2002; received in revised form 17 June 2002; accepted 26 June 2002

Abstract

The influence of restriction parameters, recess depth and land-width ratios on the load capacity and stability of a Jeffcott rotor supported by single-row, six-recessed hybrid bearings with capillary compensation is studied. The finite difference method is used to solve Reynolds equation, whilst the determination of stability threshold uses the Routh-Hurwitz method. The load capacity, stability threshold, and the critical whirl ratio, versus the changing restriction parameters, are each simulated for both shallow-recessed and the deep-recessed bearings with various land-width ratios. Simulated results indicate that small land sizes are necessary for shallow-recessed bearings in order to yield good performance, and these bearings are superior to deep-recessed bearings. Furthermore, both load capacity and stability threshold become correspondingly greater with a decrease in the restriction parameters. Nevertheless, the appropriate design of both restrictor and land size in deep-recessed bearings might well induce both load capacity and stability threshold greater than in shallow-recessed bearings.

© 2002 Elsevier Science Ltd. All rights reserved.

Keywords: Hybrid bearing; Jeffcott rotor; Capillary compensation; Restriction effect; Stability

1. Introduction

One particular type of bearing which is of major importance for use in precision machinery is the hybrid bearing, since this combines the performance merits of both hydrostatic and hydrodynamic bearings. However, since hybrid bearings are characterized by their versatile geometric configurations and diversified fluid-feed flow control devices, it becomes necessary to adopt very complex and arduous procedures and strategies for their design and selection.

The Studies of both the static and dynamic characteristics of many types of hybrid bearings have been undertaken using a combination of experiments and numerical solutions. Raimondi and Boyd [1] proposed analytical solutions and tests to evaluate the static characteristics of hydrostatic journal bearings for various restriction parameters of capillary and orifice compensations. Ghosh [2], Ghosh, et al. [3], Ghosh and Viswanath [4], Ghosh, et al. [5] and Guha, et al. [6] each presented theoretical analyses of the static and dynamic characteristics of hybrid journal bearings with either orifice or capillary restrictors. Within each of these studies, the generalized Reynolds equation for a multi-recessed finite bearing was solved by use of the perturbation method, with the results providing load capacity, attitude angle, flow, stiffness and damping coefficients of certain types of hybrid bearings.

A review of the literature in this area suggests that the influences of restriction parameters on the stability and dynamic characteristics of most types of hydrostatic and hybrid bearings have not yet been sufficiently investigated. Restriction parameters are, nevertheless, extremely important in the design of a restrictor, since they determine the diameter and tube length of capillary (or orifice diameter) along with other specifications over which the designer may be able to exercise some control. The pressure flow, for example, within each recess of a multi-recessed bearing is determined by the restriction parameters of capillary and orifice compensations. Ghosh [2], Ghosh, et al. [3], Ghosh and Viswanath [4], Ghosh, et al. [5] and Guha, et al. [6] each presented theoretical analyses of the static and dynamic characteristics of hybrid journal bearings with either orifice or capillary restrictors. Within each of these studies, the generalized Reynolds equation for a multi-recessed finite bearing was solved by use of the perturbation method, with the results providing load capacity, attitude angle, flow, stiffness and damping coefficients of certain types of hybrid bearings.

A review of the literature in this area suggests that the influences of restriction parameters on the stability and dynamic characteristics of most types of hydrostatic and hybrid bearings have not yet been sufficiently investigated. Restriction parameters are, nevertheless, extremely important in the design of a restrictor, since they determine the diameter and tube length of capillary (or orifice diameter) along with other specifications over which the designer may be able to exercise some control. The pressure flow, for example, within each recess of a multi-recessed bearing is determined by the restriction
Nomenclature

\( A \): effective recess area
\( a \): axial flow land width
\( B_{xx}, B_{yy} \): direct and cross damping coefficients in the \( x-y \) coordinate
\( B_{xy}, B_{yx} \): dimensionless direct and cross damping coefficients in \( x-y \) coordinate, \( \tilde{B}_{ij} = B_{ij}c\omega/P_s LD, (i,j = x,y) \)
\( B_{\bar{xx}}, B_{\bar{yy}} \): dimensionless direct and cross damping coefficients in \( x-y \) coordinate, \( \tilde{B}_{ij} = B_{ij}c\omega/P_s LD, (i,j = x,y) \)
\( B_{\bar{xy}}, B_{\bar{yx}} \): radial clearance of bearing
\( c_\beta \): decaying exponent for a damping whirl
\( D \): journal diameter
\( d_c \): diameter of the capillary restrictor
\( e \): eccentricity
\( F_x, F_y \): dimension and dimensionless fluid-film force components in the \( x \) and \( y \) directions, \( (\tilde{F}_x,\tilde{F}_y) = (F_x/F_y)/P_s LD \)
\( F_{\bar{x}}, F_{\bar{y}} \)
\( h_p \): recess depth
\( h, \bar{h} \): dimension and dimensionless film thickness at the center of \( r \)-th recess \( h_r = \bar{h}_r/c \)
\( i \): characteristic frequency ratio of journal motion
\( K_{eq}, \tilde{K}_{eq} \): dimension and dimensionless equivalent stiffness of film at critical instability, \( \tilde{K}_{eq} = K_{eq}c/P_s LD, \; K_{eq} = m\omega_c^2/2 \)
\( K_{xx}, K_{yy} \): direct and cross stiffness coefficients in \( x-y \) coordinate
\( \tilde{K}_{xx}, \tilde{K}_{yy} \): dimensionless direct and cross stiffness coefficients in \( x-y \) coordinate, \( \tilde{K}_{ij} = K_{ij}c/P_s LD, (i,j = x,y) \)
\( K_{\bar{xx}}, K_{\bar{yy}} \): dimensionless direct and cross stiffness coefficients in \( x-y \) coordinate, \( \tilde{K}_{ij} = K_{ij}c/P_s LD, (i,j = x,y) \)
\( K_{\bar{xy}}, K_{\bar{yx}} \)
\( L \): axial length of bearing
\( l_c \): length of the capillary restrictor
\( M \): dimensionless mass parameter, \( M = m\omega^2/P_s LD \)
\( M_c \): stability threshold
\( m \): rotor mass
\( O_b, O_j \): bearing center and journal center
\( P_r, P_{\bar{r}} \): dimension and dimensionless recess pressure at the \( r \)-th recess, \( P_r = P_{\bar{r}}/P_s \)
\( P_s \): fluid supply pressure
\( Q_r, Q_{\bar{r}} \): dimension and dimensionless flow rate at the \( r \)-th recess, \( \bar{Q}_r = 12\mu Q_r/P_s c^3 \)
\( R \): journal radius
\( s \): characteristic frequency ratio of journal motion
\( \tau \): time
\( V_0, \bar{V}_0 \): dimension and dimensionless recess volume, \( V_0 = Ah_p, \; \bar{V}_0 = V_0/cA \)
\( W \): unidirectional constant load
\( W \): load capacity, \( W = W/P_s LD \)
\( x, y \): Cartesian coordinates, \( (\bar{x},\bar{y}) = (x,y)/c \)
\( \tilde{x}, \tilde{y} \)

Greek symbols

\( \beta \): inverse of fluid bulk modulus
\( \delta_c \): capillary restriction parameter, \( \delta_c = 3\pi d_c^4/32l_c^3 \)
\( \varepsilon \): eccentricity ratio, \( \varepsilon = e/c \)
\( \phi_0 \): steady state attitude angle
\( \gamma \): fluid compressibility parameter, \( \gamma = V_0\beta P_s \)
\( \Lambda \): speed parameter, \( \Lambda = 6\mu\omega/P_s(c/R)^2 \)
\( \lambda \): whirl ratio, \( \lambda = \omega_p/\omega \)
\( \lambda_c \): critical whirl ratio, \( \lambda_c = \omega_c/\omega \)
\( \mu : \) absolute viscosity of lubricant  
\( \theta : \) angular coordinate  
\( \rho : \) density of lubricant  
\( \sigma : \) squeeze number, \( \sigma = 12\mu\omega_0/P_{r}(c/R)^2 \)  
\( \tau : \) non-dimensional time, \( \tau = \omega t \)  
\( \omega : \) journal spin speed  
\( \omega_c : \) critical whirl frequency of oil-film  
\( \omega_p : \) whirl frequency of the journal center about the equilibrium point  
\( \psi : \) recess frequency parameter, \( \psi = \sigma A/R^2 \)

parameters; indeed, both the static and dynamic characteristics of a rotor-bearing system are influenced by these restriction parameters.

The present work aims to evaluate the effects of restriction parameter on the stability characteristics of a Jeffcott rotor supported by hybrid oil-film journal bearings with capillary compensation. It is intended that on the basis of the numerical results drawn from this study, appropriate restriction parameters for stable operation can be determined for use in the bearing design process.

### 2. The Jeffcott rotor-hybrid bearing model

For a rigid Jeffcott rotor, supported horizontally on two identical hybrid journal bearings (as illustrated in Fig. 1), the perturbation equations of journal motion about the equilibrium point in the fixed reference coordinates \((O_b, x, y)\) are described as:

\[
\frac{\partial^2 F_x}{\partial \tau^2} = -2F_x - 2(K_{xx} \dot{x} + K_{xy} \dot{y} + \lambda B_{xx} \ddot{x})
\]

\[
+ \lambda B_{xy} \ddot{y})
\]

\[
\frac{\partial^2 F_y}{\partial \tau^2} = -2F_y - 2(K_{xy} \dot{x} + K_{yy} \dot{y} + \lambda B_{yx} \ddot{x})
\]

\[
+ \lambda B_{yy} \ddot{y})
\]

Components of bearing force, along with the stiffness and damping coefficients, are determined by integration of the fluid-film pressure distribution. For an incompressible flow situated in the viscous thin-fluid film bearing, linearization of the generalized Reynolds equation governing the static and dynamic characteristics of lubricant film is achieved by the use of the perturbation method. This is solved numerically for pressure distribution by the finite difference method with a successive over-relaxation scheme to the satisfaction of the boundary conditions and the flow continuity equation of capillary restrictor. In this study, the formulation and programming of the solving process are carried out in accordance with Ghosh, et al. [5].

The recess flow continuity equation for a capillary compensated bearing is described by:

\[
\frac{\pi d^2}{128l_{\mu}} (P_r - P_i) = \dot{Q}_r + \frac{\partial}{\partial \tau} (Ah_i) + \frac{1}{\rho \tau} (\rho \dot{V}_w)
\]

where the term on the left of the equal sign refers to the mass flow into the recess through a capillary; the first term on the right of the equal sign is the mass flow out of the recess through the bearing clearance, and the second and third terms refer to the time rate of the changing of the recess content mass, due to squeeze effect and the fluid compressibility effect.

Using non-dimensional parameters: \( \bar{h}_i = h_i/c, \bar{P}_r = P_r/P_c, \bar{Q}_r = 12\mu Q_r/P_c c^3, \bar{V} = V_0/Ac, \delta_c = 3\pi d^2/32l_{\mu} c^3, \tau = \omega t, \psi = \sigma A/R^2, \sigma = 12\mu\omega_0/P_{r}(c/R)^2, \gamma = V_0\beta P_c, \)

the dimensionless flow continuity equation is given by:

\[
\delta_r (1 - \bar{P}_r) = \bar{Q}_r + \frac{\partial}{\partial \tau} (\bar{h}_i) + \psi \frac{\partial}{\partial \tau} (\bar{P}_r)
\]

Since the static and dynamic recess pressures are determined by the Reynolds equation associated with the recess flow continuity equation, the restriction parameter will significantly affect the characteristics of the bearing.

### 3. Determination of stability threshold

When the bearing is operating under steady-state condition, if the film pressure is disturbed, the rotor will attempt to whirl at a frequency \( \omega_p \) in order to maintain

![Fig. 1. Jeffcott rotor model.](image)
the flow balance; depending on the dynamic characteristics of the rotor bearing system, this whirl will either become more pronounced, or die out. Based on the equilibrium position of the journal, for a small disturbance, the respective displacement functions of journal center can be expressed as:

$$\ddot{x} = Xe^{\lambda}\dot{x}$$

(5)

and

$$\ddot{y} = Ye^{\lambda}\dot{y}$$

where $X$ and $Y$ are constant, depending on initial conditions, and $s$ represents the characteristic frequency of the journal motion. Substituting Eq. (5) into Eqs. (1) and (2) gives coupled equations of $X$ and $Y$ as follows:

$$\{2\lambda^{2}M\omega^{2} + 2\lambda B_{xx}\ddot{X} + 2K_{xx}\dddot{X} + 2\lambda B_{xy}\ddot{Y} + 2K_{xy}\dddot{Y}\}$$

(6)

$$= 0$$

$$2\lambda B_{yx}\ddot{X} + 2\lambda B_{yy}\dddot{X} + 2\lambda K_{yy}\dddot{Y} + 2K_{xy}\dddot{Y} = 0$$

The non-trivial solution of $X$ and $Y$ provides the coefficient determinant of Eq. (6), being equal to zero. Expanding this determinant equation provides the following characteristic equation of eigenvalue $s$ of journal motion:

$$C_{0}s^{4} + C_{1}s^{3} + C_{2}s^{2} + C_{3}s + C_{4} = 0$$

(7)

where constant coefficients are $C_{0} = 2\lambda^{2}M^{2}$, $C_{1} = 2\lambda^{2}M(B_{xx} + B_{xy})$, $C_{2} = 2\lambda^{2}M(K_{xx} + K_{yy}) + 4\lambda^{2}(B_{xx}B_{yy} - B_{xy}B_{yx})$, $C_{3} = 4\lambda(K_{xx}B_{yy} + K_{yy}B_{xx} - B_{xy}B_{yx})$, $C_{4} = 4\lambda(K_{xx}B_{yy} - K_{xx}B_{xy})$.

The eigenvalues of Eq. (7) have the form $s = c_{b}/\omega_{0}\pm i$. When all $c_{b}$ is 0, the journal motion is at the critical boundary between stability and instability. If one of $c_{b}$ is positive, the rotor motion demonstrates a self-excited vibration arising from the fluid film. For the Jeffcott rotor to be stable, all roots of Eq. (7) must have a negative real part.

According to Routh–Hurwitz theory, sufficient and necessary conditions are provided for all roots of Eq. (7) without positive real part. Normally, for a hybrid bearing the both direct stiffness and direct damping coefficients may far greater than the both cross coefficients. Thus, all the necessary conditions are satisfied by $C_{i} > 0$, $i = 0 \sim 4$. The sufficient condition is obtained by estimating the fourth Hurwitz determinant and which can be described by:

$$C_{1}C_{2}C_{3}C_{4} + C_{1}C_{3}C_{4}$$

(8)

where equality gives the stability threshold of the system parameter.

Therefore, the existence of stability threshold must be theoretically satisfied by all necessary conditions, and the $M_{c}$, is obtained by solving equality in Eq. (8):

$$M_{c} = \frac{2A_{1}A_{4}A_{5}}{A_{2}^{2} - A_{2}A_{4}A_{5}}$$

(9)

where $A_{1} = K_{xx}B_{yy} + K_{yy}B_{xx} - K_{xy}B_{yx} - K_{yx}B_{xy}$, $A_{2} = K_{xx}K_{yy} - K_{xy}K_{yx}$, $A_{3} = B_{xx}B_{yy} - B_{xy}B_{yx}$, $A_{4} = K_{xx} + K_{yy}$, $A_{5} = B_{xx} + B_{yy}$.

Thus, self-excited vibration occurs when value $M$ of the Jeffcott rotor hybrid bearing system is larger than $M_{c}$. Conversely, for $M < M_{c}$, the vibration resulting from initial perturbation of the rotor dies out exponentially with time. However, $M = M_{c}$ relates to the critical condition at which the rotor whirls to a limit cycle through transient vibration.

4. Determination of whirl frequency

Within the ideal flow of a non-viscous lubricant, the half-frequency whirl ($\lambda = 0.5$) can be caused by a flow balance; however, the whirl frequency within a real rotor-bearing system may be influenced by both inertia and damping effects, the details of which can be obtained by determining the characteristic equation for the critical condition. When a journal is at a critical whirling condition, there will exist at least one set of purely imaginary roots as described by:

$$s = \pm i\omega_{n}$$

(10)

At this critical condition, $c_{b}$ is zero and $\lambda_{c} = \omega_{n}/\omega_{0}$.

Substituting Eq. (10) into Eq. (6) gives

$$\{K_{eq} - \omega_{0}^{2}i\lambda_{c}B_{xx}\ddot{X} - (K_{xx} + i\lambda_{c}B_{xy})\dddot{Y} = 0$$

(11)

$$-\{K_{yx} + i\lambda_{c}B_{yx}\ddot{X} + (K_{eq} - \omega_{0}^{2}i\lambda_{c}B_{yy})\dddot{Y} = 0$$

(12)

where $K_{eq}$ represents an equivalent oil-film stiffness acting as two isotropic springs, without damping, to support a rotor at the critical condition.

The non-trivial conditions for $X$ and $Y$ of Eqs. (11) and (12) are derived as follows:

$$K_{eq}(B_{xx} + B_{yy}) - (K_{xx}B_{yy} + K_{yy}B_{xx}) + K_{xy}B_{yx}$$

(13)

$$+ K_{yx}B_{xy} = 0$$

$$K_{eq}(B_{xx} - B_{yy}) - (K_{xx}B_{yy} - K_{yy}B_{xx})$$

(14)

$$- K_{xy}K_{yx} = 0$$

$K_{eq}$ is determined from Eq. (13) and substituted into Eq. (14) to give the critical whirl ratio, which is expressed as:

$$\lambda_{c} = \left[\frac{(K_{eq} - K_{xx})(K_{eq} - K_{yy}) - K_{xy}K_{yx}}{B_{xx}B_{yy} - B_{xy}B_{yx}}\right]^{1/2}$$

(15)

5. Results and discussion

Using the procedures and programs outlined in Ghosh, et al. [5], and this study has evaluated the dynamic
characteristics of the Jeffcott rotor supported by hybrid bearings with capillary compensation. The single-row six-recessed hybrid bearing is illustrated in Fig. 2, with the lubricant being supplied from a constant pressure source flowing into each recess through a capillary restrictor to the sill, and thus generating a pressure drop. For shallow recesses, the pressure within the recesses is no longer constant and thus hydrodynamic pressure is generated.

The load capacity \( \overline{W} \), stability threshold \( \overline{M_c} \) and critical whirl ratio \( \lambda_c \) have each been determined for various values of land-width ratio \( a/L \) and different recess depth ratios \( h_p/c \). The results of the analysis are compared for the same eccentricity \( e \) equal to 0.5 and the same speed parameter \( \Lambda \) equal to 6.0.

For those cases where \( a/L = 0.1, 0.25 \) and 0.4, the variation in load capacity \( W \) with respect to restriction parameter \( \delta_c \) are shown in Fig. 3(a) for a shallow-recessed bearing \( (h_p/c = 1.0) \), and in Fig. 3(b) for a deep-recessed bearing \( (h_p/c = 5.0) \).

For shallow-recessed bearings, load capacity decreases as \( \delta_c \) increases, with the exception of small land-width ratio \( (a/L = 0.1) \). With an increase in recess area, there is a corresponding weakening of the influence of restriction parameter \( \delta_c \) on load capacity \( \overline{W} \), since the restriction effect also exits within the shallow recess. In terms of land area, for large land-width ratio \( (a/L = 0.4) \), \( \overline{W} \) decreases as \( \delta_c \) increases, whereas, for small land-width ratio \( (a/L = 0.1) \), \( \overline{W} \) is almost constant. The load capacities for the three land-width ratios approach a mutually closed value when the restriction parameter is greater than seven.

For deep-recessed bearings, maximum load capacity occurs at a critical restriction parameter, with the higher maximum load capacity being obtainable at smaller parameters of restrictor when the land-width ratio is greater. Furthermore, when \( \delta_c < 4.0 \), deep-recessed bearings with a large land-width ratio have a higher \( \overline{W} \) than shallow-recessed bearings with a moderate land-width ratio \( (a/L = 0.25) \). Thus, in the design of a bearing with deep recesses, the adoption of large land-width ratio and small restriction parameters is necessary in order to yield a fairly large load capacity. It is also of some importance in the design of deep-recessed bearings to select an appropriate restriction parameter between two and five, and to adopt a moderate land-width ratio.

For deep-recessed bearings, hydrodynamic pressure is responsible for inducing the greatest amount of static force upon the land area; thus, a larger land-width ratio results in a larger load capacity, whereas, for shallow-recessed bearings, hydrodynamic pressure is generated within the recesses. Therefore, the load capacity of a shallow-recessed bearing is higher than that of a deep-recessed bearing with the same land-width ratio.

The simulation results of stability threshold \( \overline{M_c} \) are shown in Fig. 4(a) for a shallow-recessed bearing \( (h_p/c = 1.0) \), and in Fig. 4(b) for a deep-recessed bearing \( (h_p/c = 5.0) \); \( \overline{M_c} \) decreases as \( \delta_c \) increases for both a shallow-recessed bearing and deep-recessed bearing. The variation in \( \overline{M_c} \) is alleviated with an increase in \( \delta_c \),
since the hydrodynamic effect begins to dominate when $\delta_c$ is sufficiently large, whilst the superiority of the hydrostatic effect is dissipated.

In contrast to the effect of $\delta_c$ on $\bar{W}$, a smaller land-width ratio is associated with a larger $\bar{M}_c$. For small land-width ratio ($a/L = 0.1$) the $\bar{M}_c$ of a deep-recessed bearing is closed to that of a shallow-recessed bearing. When $\delta_c < 4.0$, the $\bar{M}_c$ of a shallow-recessed bearing is larger than that of a deep-recessed bearing; conversely, when $\delta_c < 4.0$, the $\bar{M}_c$ of a deep-recessed bearings is larger than that of a shallow-recessed bearing. The reason for this is that as a result of the large value of $\delta_c$, when sufficient external pressurized oil is fed into the system, a deep and large recess ($a/L = 0.1$) provides an additional damping effect, suppressing self-excited oil-film whirl.

For moderate land-width ratio ($a/L = 0.25$) and large land-width ratio ($a/L = 0.4$), the $\bar{M}_c$ of a shallow-recessed bearing is larger than that of a deep-recessed bearing. For a shallow-recessed bearing, the $\bar{M}_c$ is close for both ($a/L = 0.25$) and ($a/L = 0.4$); for a deep-recessed bearing, a large $\bar{M}_c$ can be obtained by the use of moderate land-width ratio ($a/L = 0.25$) and small restriction parameters ($\delta_c < 1$); however, when large land-width ratio is used ($a/L = 0.4$), $\bar{M}_c$ cannot exceed eleven, even when the restriction parameters are very small ($\delta_c < 1$).

The simulation results of the critical whirl ratio ($\lambda_c$) are provided in Fig. 5(a) for a shallow-recessed bearing, and in Fig. 5(b) for a deep-recessed bearing. In all cases, $\lambda_c$ increases along with an increase in $\delta_c$ since a higher instability threshold is induced in the lower value of critical whirl ratio. For both moderate and large land-width ratios, the $\lambda_c$ of a deep-recessed bearing is higher than that of a shallow-recessed bearing. For a shallow-recessed bearing, the $\lambda_c$ is larger, with a correspondingly greater land-width ratio. The smallest $\lambda_c$ of a deep-recessed bearing occurs in the case of a large recess.
(a/L = 0.1) for the same reason that a deep, large recess provides the additional damping effect.

6. Conclusions

The general belief amongst bearing designers in the past has always been that shallow-recessed bearings provide a superior performance to deep-recessed bearings; nevertheless, the influence of restriction parameters is clearly significant. However, as this study reveals, with the appropriate selection of restriction parameters and land-width ratio, superior load capacity and stability threshold may be obtainable for a deep-recessed bearing. Although the performance of a shallow-recessed bearing is, by and large, superior to that of a deep-recessed bearing, it nevertheless hinges upon the geometric design of both restrictors and bearings.

In all cases, load capacity is greater with a larger land-width ratio. There is, however, contrary behavior in the stability threshold for bearings with the same recess depth ratio and the same restriction parameters. This is due to the load capacity being determined by hydrodynamic pressure, whereas the stability threshold is determined by hydrostatic pressure when the shaft is rotating.

For small land-width ratio, a deep-recessed bearing provides the minimum load capacity in all cases, whilst the stability threshold is very close to that of a shallow-recessed bearing which, with small land-width ratio, provides the maximum stability threshold. For large land-width ratio a shallow-recessed bearing provides a large load capacity, but with less stability than that provided by a small land-width ratio.

From a design perspective, in order to ensure the stable operation of a heavy rotor, a small restriction parameter is necessary for any hybrid bearing with shallow recesses. The construction of a practical restrictor is, however, constrained by the limitations of micro-manufacturing processes, which are currently incapable of obtaining such a necessarily small capillary diameter. Thus, in the design process, it becomes necessary to select a corresponding restriction parameter, for the maximum stability threshold, based upon obtainable capillary dimensions.

References