Analysis of a non-constant gap externally pressurized conical bearing with temperature and pressure dependent viscosity

P Sinha1*, P Chandra1 and S Bhartiya2
1Department of Mathematics, Indian Institute of Technology, Kanpur, India
2Department of Mathematics, Indian Institute of Technology, Bombay, India

Abstract: The present paper analyses an externally pressurized non-constant gap conical bearing rotating with a uniform angular velocity. The lubricant is assumed to be incompressible and its viscosity varies with both pressure and temperature. Although the inertia effect due to lubricant flow has been neglected, rotational inertia is taken into account. The governing system of coupled momentum and energy equations, in conical coordinates, is solved numerically using the finite difference method, to determine various bearing characteristics. The effect of the viscosity–pressure exponent, \( \gamma \), on various characteristics of the bearing has been studied. It is seen that variation of \( \gamma \) does not produce any significant change in the load capacity for convergent and constant gap conical bearings. For the divergent gap also, no significant difference is seen when the gap is slightly divergent. However, for highly divergent gaps, the load capacity increases with an increase in \( \gamma \). It is also seen that variation in \( \gamma \) does not produce any significant change in torque of the bearing.

Keywords: conical bearings, externally pressurized, temperature–pressure-dependent viscosity

NOTATION

- \( a \): accuracy parameter
- \( c \): specific heat of the fluid
- \( D \): dissipation parameter
- \( E \): a type of Eckert number
- \( h \): lubricant film thickness
- \( h_0 \): lubricant film thickness at the outlet
- \( k \): thermal conductivity
- \( K \): geometrical parameter accounting for divergence or convergence of the bearing gap
- \( L \): load capacity of the bearing
- \( M \): torque on the bearing surface
- \( p \): gauge pressure
- \( Pr \): Prandtl number
- \( Q \): flowrate
- \( R \): rotational parameter
- \( Re^* \): modified Reynolds number
- \( T \): temperature of the lubricant
- \( T_1, T_o \): temperature of the pad and the slider respectively
- \( \psi \): semi-angle between the cone angles of the slider and the pad
- \( \omega \): angular velocity of the slider
- \( \alpha \): angle between the stationary pad surface and a plane perpendicular to the bearing axis
- \( \beta \): viscosity–temperature exponent
- \( \gamma \): viscosity–pressure exponent
- \( \eta \): viscosity of the lubricant
- \( \eta_0 \): viscosity of the lubricant at pad temperature
- \( \nu_o \): kinematic viscosity of the lubricant
- \( \rho \): density of the lubricant
- \( \phi \): angular velocity of the slider
- \( \xi \): angular velocity of the slider
- \( \psi \): semi-angle between the cone angles of the slider and the pad
- \( \omega \): angular velocity of the slider

1 INTRODUCTION

Most of the research work on the conical bearing is based on the assumption that the gap between the two
conical surfaces (slider and pad) is constant (i.e. the two surfaces are optically parallel). This may be a rather idealistic situation and very difficult to achieve physically due to misalignment during the manufacturing process. Also, sometimes it may be deliberate, in order to enhance the load-carrying capacity due to hydrodynamic action in converging gaps. In such situations, the cone angles of the slider and the pad may differ by a very small angle (<1°). Saxena et al. [1] made the thermohydrodynamic analysis of an externally pressurized conical bearing with a non-constant gap by assuming the viscosity of the lubricant to be a function of temperature alone. However, many common lubricants, including petroleum-based lubricants, show considerable change in viscosity when subjected to high pressures. It has been observed that viscosity of the lubricants increases with an increase in pressure to the extent that some oils actually behave as plastic solids at pressures exceeding 1.4 x 10^8 N/m² (see references [2] to [5]). Thus, it is reasonable to consider viscosity as a function of pressure as well as temperature. Only a few researchers have considered this aspect of viscosity variation because of the complexity introduced in the solution of governing equations. Besides, an exact empirical relation has not yet been established. Some of the endeavours made in this direction are by Simon [6, 7] and Dong and Shi-Zhu [8]. While studying elasto-hydrodynamic problems in various types of bearings, they considered the mathematical relationship given by

\[ \eta = \eta_0 \exp[\gamma p - \beta(T - T_l)] \]  

(1)

where \( \eta_0 \) is the viscosity at the pad temperature \( T_l \), \( \gamma \) is the viscosity–pressure exponent and \( \beta \) is the viscosity–temperature exponent.

In view of this and the observation that in the case of the divergent gap conical bearing the pressure at the inlet is extremely high [1], it is of interest to include the variation in viscosity due to pressure also. The present paper is thus an extension of the authors’ previous work and is devoted to the analysis of a non-constant gap externally pressurized conical bearing where viscosity of the lubricant varies according to expression (1). The inertia effect due to lubricant flow has been neglected.

2 GOVERNING EQUATIONS

The geometry of the problem considered in this paper is shown in Fig. 1. Steady, axially symmetric, laminar flow of an incompressible lubricant is considered through the gap of an externally pressurized conical bearing. The slider rotates with constant angular velocity \( \omega \). The pad and the slider are maintained at constant but arbitrary temperatures, \( T_l \) and \( T_u \) respectively. The surfaces of the slider and pad are conical with a non-constant gap between them and \( \psi \), the semi-angle of the difference between the cone angles of the slider and the pad, is assumed to be small (<1°). It must also be noted that \( h \ll x_{in} \) (the figure is not to scale).

Using the conical coordinates and relation (1), the transformed Navier-Stokes equations in non-dimensional form with convective inertia neglected are [9]

\[
\frac{\partial \hat{u}}{\partial \hat{x}} + \frac{\partial \hat{v}}{\partial \hat{y}} = 0
\]  

(2)

\[
\frac{\partial^2 \hat{u}}{\partial \hat{y}^2} - \frac{\beta}{\partial \hat{y}} \frac{\partial \hat{T}}{\partial \hat{y}} \frac{\partial \hat{u}}{\partial \hat{y}} = \exp[-\gamma p + \beta \hat{T} - 1] \left( \frac{dp}{dx} - \frac{R \hat{u}^2}{x} \right)
\]  

(3)

and the energy equation is

\[
\frac{\partial^2 \hat{T}}{\partial \hat{y}^2} + D \exp[\gamma p - \beta \hat{T} - 1] \left( \frac{\partial \hat{w}}{\partial \hat{y}} \right)^2 = 0
\]  

(4)

where

\[
\hat{u} = u \hat{X}, \quad \hat{v} = v \hat{Y}, \quad \hat{T} = \frac{T - T_l}{T_u - T_l}
\]

\[
\hat{w} = \frac{w}{h}, \quad R = \frac{R}{h^2}, \quad D = \frac{D}{h^4}
\]
where

\[
\bar{x} = \frac{x}{x_0}, \quad \bar{y} = \frac{y}{h_0}, \quad \bar{h} = \frac{h}{h_0}
\]

\[
\bar{u} = \frac{u}{U}, \quad \bar{v} = \frac{v}{V}, \quad \bar{w} = \frac{w}{W}
\]

\[
U = \frac{Q}{2\pi x_0 h_0 \cos \alpha}, \quad V = U \frac{h_0}{x_0}
\]

\[
W = \omega x_0 \cos \alpha, \quad \bar{T} = \frac{T}{T_1}
\]

\[
\bar{\beta} = \beta T_1, \quad \bar{\gamma} = \frac{\eta_0 x_0 U}{h_0^2} \gamma
\]

\[
\bar{\eta} = \frac{\bar{\eta}}{\eta_0} = \exp[\gamma \bar{p} - \bar{\beta}(T - 1)]
\]

\[
\bar{p} = \frac{\rho h_0^2}{\eta_0 x_0 U}, \quad \bar{D} = \frac{Pr ER}{Re^*}, \quad \bar{R} = \frac{2\pi \omega^2 x_0^2 h_0^3 \cos^3 \alpha}{\nu_0 Q}
\]

(6)

where \( h_0 \) is the film thickness at the outlet, which remains constant, and \( U \) is the mean radial velocity at the outlet.

The boundary conditions in dimensionless form associated with these equations are

\[
\bar{u}(\bar{x}, 0) = \bar{v}(\bar{x}, 0) = \bar{w}(\bar{x}, 0) = \bar{u}(\bar{x}, \bar{h}) = \bar{v}(\bar{x}, \bar{h}) = 0
\]

\[
\bar{w}(\bar{x}, \bar{h}) = \bar{x}, \quad \bar{T}(\bar{x}, 0) = 1, \quad \bar{T}(\bar{x}, \bar{h}) = T_0 = T_u / T_1
\]

\[
\bar{p}(1) = 0
\]

(7)

and the non-dimensional film thickness \( \bar{h}(\bar{x}) \) is defined as

\[
\bar{h} = 1 + K(\bar{x} - 1)
\]

(8)

where the parameter \( K \) determining the bearing geometry is given by

\[
K = \frac{x_0 \tan \psi}{h_0}
\]

(9)

where \( \psi > 0 \) and \( \psi < 0 \) represent divergent and convergent gap bearing cases respectively.

3 NUMERICAL METHOD

Equation (2) along with the boundary conditions (7) give the matching condition

\[
\int_0^{\bar{h}} \bar{u}(\bar{x}, \bar{y}) \, d\bar{y} = 1
\]

(10)

The finite difference method has been used to solve equations (3) to (5). The difference scheme, according to the grid details shown in Fig. 2, is

\[
\bar{u}(i, j) = \frac{1}{2} [\bar{u}(i, j + 1) + \bar{u}(i, j - 1)]
\]

\[
- \exp\left\{\gamma \bar{p}(i) + \bar{\beta} \bar{T}(i, j - 1)\right\}
\]

\[
\times \left\{\frac{[\Delta y(i)]^2}{2} \left(\frac{d \bar{p}}{d \bar{y}}(i) - \frac{R [\bar{w}(i, j)]^2}{\bar{x}(i)}\right)\right\}
\]

\[
- \frac{\bar{\beta}}{8} [\bar{u}(i, j + 1) - \bar{u}(i, j - 1)]
\]

\[
\times [\bar{T}(i, j + 1) - \bar{T}(i, j - 1)]
\]

(11)

\[
\bar{w}(i, j) = \frac{1}{2} [\bar{w}(i, j + 1) + \bar{w}(i, j - 1)]
\]

\[
- \frac{\bar{\beta}}{8} [\bar{w}(i, j + 1) - \bar{w}(i, j - 1)]
\]

\[
\times [\bar{T}(i, j + 1) - \bar{T}(i, j - 1)]
\]

(12)

\[
\bar{T}(i, j) = \frac{1}{2} [\bar{T}(i, j + 1) + \bar{T}(i, j - 1)]
\]

\[
+ \frac{D}{8} \exp\left\{\gamma \bar{p}(i) - \bar{\beta} \bar{T}(i, j - 1)\right\}
\]

\[
\times [\bar{w}(i, j + 1) - \bar{w}(i, j - 1)]^2
\]

(13)

Fig. 2 Grid details
where $\Delta x(j)$ and $\Delta y(i)$ are mesh widths in the $x$ and $y$ directions respectively and are given by

$$
\Delta x(j) = \sqrt{\frac{(j-1)[K(\bar{x}_m - 1)]}{(my-1)2+(1-\bar{x}_m)^2}},
$$

$$
j = 1, \ldots, my
\tag{14}
$$

and

$$
\Delta y(i) = \frac{h(i)}{my-1}, \quad i = 1, \ldots, mx
\tag{15}
$$

The overall process of solving the difference equations (11) to (13) is given in the flow chart of Fig. 3. Pressure $\bar{p}$ at the inlet is prescribed initially and the pressure gradient, $d\bar{p}/dx$, is assumed at the inlet ($i = 1$). The difference equations (12) and (13) are solved to determine $\bar{w}(1, j)$ and $\bar{T}(1, j)$ respectively, by the Gauss–Siedel iteration method. Then $u(1, j)$ is determined using the difference equation (11). The convergence criterion adopted for the relative error is

$$
\left| S_{\text{new}}(i, j) - S(i, j) \right| \leq a
\tag{16}
$$

where $S$ represents $\bar{u}$, $\bar{w}$, $\bar{T}$ and $a$ is the accuracy parameter. Matching condition (10) is then used to check the validity of the assumed pressure gradient at $i = 1$. If the matching condition (10) is not satisfied, the pressure gradient is modified using the half-interval method and the process of determining $\bar{w}(1, j)$, $\bar{T}(1, j)$ and $\bar{u}(1, j)$ is repeated using the modified value of the pressure gradient until condition (10) is satisfied. In this manner $d\bar{p}/dx$ (1), $\bar{u}(1, j)$, $\bar{w}(1, j)$ and $\bar{T}(1, j)$ are determined. Then the process moves on to the next grid point $i = 2$ and the pressure gradient at $i = 2$ is assumed. Using the initially prescribed value of $\bar{p}(1)$ and the corrected value of the pressure gradient at $i = 2$, the pressure at $i = 2$ is calculated by the forward difference formula, given as

$$
\bar{p}(i) = \bar{p}(i - 1) + \Delta x(i) \frac{d\bar{p}}{dx}(i - 1)
\tag{17}
$$

Then $\bar{u}(2, j)$, $\bar{w}(2, j)$ and $\bar{T}(2, j)$ are determined in the same manner as described above. This process of assuming the pressure gradient and calculating the pressure $\bar{p}$, by using the modified value of the pressure gradient, and $u$, $w$, $T$ is repeated at all other grid points, i.e. for each $i > 2$.

Having found the pressure gradient and pressure for each $i$, the pressure at the outlet ($i = mx$) is checked. If it does not satisfy the boundary condition, i.e. $\bar{p} = 0$ at $\bar{x} = 1$, the initially assumed inlet pressure, $\bar{p}(1)$, is modified and the whole process is repeated again until the boundary condition at the outlet is satisfied.

### 4 LOAD CAPACITY AND TORQUE

The non-dimensional load capacity on the slider of the conical bearing is defined as

$$
L = \frac{L\bar{h}^2}{\pi x_0^2 \eta_0 U \cos^2 \alpha} = \bar{p}x_0^2 + 2 \int_{x_0}^1 \bar{p}x \, dx \tag{18}
$$

which, on integration by parts, gives

$$
L = -\frac{L\bar{h}^2}{\pi x_0^2 \eta_0 U \cos^2 \alpha} = - \int_{x_0}^1 \left( x^2 \frac{dp}{dx} \right) d\bar{x} \tag{19}
$$

The non-dimensional torque on the bearing is given as

$$
M = \frac{Mh_0}{2\pi x_0^2 \eta_0 U \cos^2 \alpha} = \int_{x_0}^1 \bar{x}^2 \exp[\bar{\gamma}\bar{p} - \bar{\beta}(T - 1)] \frac{\partial \bar{w}}{\partial y} \, d\bar{x} \tag{20}
$$

where $M$ represents the dimensional torque.

Load capacity and torque are calculated by evaluating the integrals (18) and (19) numerically, using Simpson’s rule and previously obtained values of $\bar{p}$, $\bar{w}$ and $T$.

### 5 RESULTS AND DISCUSSION

As seen from equations (18) and (19), bearing characteristics are functions of the inlet point $x_m$, temperature setting of the slider $T_u$, dissipation parameter $D$, rotational parameter $R$, film thickness parameter $K$ and viscosity–temperature exponent $\beta$. Numerical values of these parameters are taken as

$$
\bar{x}_m = 0.1; \quad T_u = 0.8, 1.0, 1.2; \quad D = 1.5
$$

$$
R = 60; \quad -0.6 \leq K \leq 0.6 \; \text{and} \; \bar{\beta} = 1.5
$$

Numerical values of the viscosity–pressure exponent $\bar{\gamma}$ are chosen as 0.0, $1 \times 10^{-4}$, $5 \times 10^{-4}$, $1 \times 10^{-3}$ and $1.5 \times 10^{-3}$; $\bar{\gamma} = 0.0$ represents the case when viscosity is independent of pressure. In this case results are similar to those obtained by Saxena et al. [1]. The results were computed for $\bar{\beta} = 1$ and 5. However, it was observed that for $\bar{\beta} = 1$, the variation of $\bar{\gamma}$ does not produce any significant changes in various characteristics of the bearing. Therefore, the results have been presented and discussed only for $\bar{\beta} = 5$.

Figures 4, 5 and 6 show the velocity profiles $\bar{u}(\bar{x}, \bar{y})$ and $\bar{w}(\bar{x}, \bar{y})$ and the temperature profile $\bar{T}(\bar{x}, \bar{y})$ versus $\bar{y}$ at different cross-sections for different values of $K$ and $T_u$ respectively. It must be noted that only one value of $\bar{\gamma}(0.0)$ has been chosen. This is because variation in $\bar{\gamma}$ does not produce any significant changes in these flow variables.
Fig. 3  Flow chart
5.1 Velocity distribution

Figure 4 shows the velocity profile $\bar{u}(\bar{x}, \bar{y})$ versus $\bar{y}$ at different cross-sections for constant ($K = 0.0$), convergent ($K = -0.6$) and divergent ($K = 0.6$) gap bearings. These profiles are shown for different settings of the slider temperature. Near the inlet ($\bar{x} = 0.28$), the velocity profile for $\bar{u}(\bar{x}, \bar{y})$ seems to be symmetric about the centre-line ($\bar{y} = 0.5$). In the mid-region ($\bar{x} = 0.64$), the effect of rotation becomes dominant and the symmetric nature of the flow is disturbed. Values of $\bar{u}(\bar{x}, \bar{y})$ closer to the rotating surface (pad) are higher. This effect is more conspicuous at the outlet. In all the three gaps, there is a continuous decrease in the velocity magnitudes from the mid-region onwards. This happens because of an ever-increasing cross-sectional area of the conical geometry and a fixed flowrate at every cross-section, resulting in smaller values of $\bar{u}(\bar{x}, \bar{y})$. As the lubricant reaches the outlet ($\bar{x} = 1.0$), a reverse flow is observed in all three types of gap. In the case where the slider temperature is less than that of the pad, there is no reversal of flow. As the slider temperature increases, the reverse flow at the outlet becomes prominent. This reversal of flow can be eliminated if the flow rate is increased for a fixed rotational velocity, as already discussed by Sinha et al. [10]. It is also seen that the velocities $\bar{u}(\bar{x}, \bar{y})$ near the inlet are remarkably higher in the case of divergent and lesser in the case of convergent gaps as compared with the constant gap bearing. This may happen because for the divergent gap bearing, the area of cross-section through which the lubricant is to be pumped decreases, whereas in the case of the convergent gap it increases as compared with the constant gap bearing. The effect of $\bar{T}_u$ on $\bar{u}(\bar{x}, \bar{y})$ is the same in all three types of gap; i.e. the
lubricant close to the hot surface moves more rapidly in the \( \bar{x} \) direction as compared with the lubricant closer to the cooler surface. Similar results have been obtained by Sinha et al. [10] for \( K = 0 \).

Figure 5 shows the velocity distribution \( \bar{w}(\bar{x}, \bar{y}) \) versus \( \bar{y} \) at different cross-sections of the bearing for constant \( (K = 0.0) \), convergent \( (K = -0.6) \) and divergent \( (K = 0.6) \) gaps. It is seen that in all three types of gap, the velocities \( \bar{w}(\bar{x}, \bar{y}) \) for the cool slider case are always higher than those for the hot slider. It may thus be interpreted that cooling of the slider leads to smaller velocity gradients in the \( \phi \) direction. This happens because cooling of the slider enhances the viscosity in its proximity and under the no-slip conditions the lubricant with increased viscosity can move faster in the \( \phi \) direction along with the slider, as compared with the other
cases. The variation of \( \bar{w}(\bar{x}, \bar{y}) \) with respect to \( \bar{x} \) can also be visualized from this figure and is seen to increase with an increase in \( \bar{x} \) for a chosen value of \( \bar{y} \). It may finally be concluded that \( \bar{w}(\bar{x}, \bar{y}) \) for convergent or divergent gaps is not very different from that for the constant gap.

Figure 6 shows the temperature \( T(\bar{x}, \bar{y}) \) generated in the lubricant versus \( \bar{y} \) at different cross-sections for constant \( (K = 0.0) \), convergent \( (K = -0.6) \) and divergent \( (K = 0.6) \) gap bearings. It is seen that in all the gaps, \( T(\bar{x}, \bar{y}) \) increases linearly near the inlet region for the hot slider–cool pad condition. As the lubricant moves towards the outlet, a temperature peak is observed at \( \bar{y} = 0.4 \) for \( T_u = 0.8 \), at \( \bar{y} = 0.6 \) for \( T_u = 1.0 \) and at \( \bar{y} = 0.8 \) for \( T_u = 1.2 \). It may be noted that in all three cases of different gaps almost identical temperatures are generated. It is obvious from the coupling of
equations (12) and (13) that temperature depends upon \( w(x, y) \). Since \( w(x, y) \) is also hardly affected by the variation in \( K \) (as seen in Fig. 5), so is \( T(x, y) \).

### 5.2 Pressure distribution

Figures 7, 8 and 9 show the effect of \( \dot{\gamma} \) on the pressure distribution versus \( x \) for constant \( (K = 0.0) \), convergent \( (K = -0.6) \) and divergent \( (K = 0.6) \) gap bearings respectively, at different settings of the slider temperature. It is seen that variation of \( \dot{\gamma} \) does not affect the pressure distribution significantly for constant and convergent gap bearings (Figs 7 and 8). However, there is a significant increase in pressure in the inlet region with an increase in \( \dot{\gamma} \) for divergent gap conical bearings (Fig. 9). This happens because, for convergent and constant gaps, the inlet pressures are not as high as in the case of divergent gap bearings, as observed by Saxena et al. [1]. Thus, in view of relation (1), even a small increase in \( \dot{\gamma} \) for divergent gap bearings increases the viscosity significantly, resulting in a higher inlet pressure. Further, the effect of \( \dot{\gamma} \) is more pronounced when the slider cools down. This may happen for the same reason stated above, because cooling of the slider increases the viscosity which in turn leads to a higher inlet pressure. It is also observed that as the slider temperature increases, the pressure decreases for all values of \( K \). This observation is similar to that seen by Saxena et al. [1].

### 5.3 Load capacity and torque

Figure 10 shows the effect of \( \dot{\gamma} \) on load capacity versus \( K \) for \( \beta = 5 \) at \( T_u = 0.8, 1.0 \) and 1.2. It is seen that variation of \( \dot{\gamma} \) does not produce any significant change in the load capacity for convergent and constant gap conical bearings (the curves for \( K \leq 0 \) overlap). For
Fig. 7 Effect of $\bar{\tau}$ on pressure distribution for a constant gap bearing at different settings of the slider temperature

Fig. 8 Effect of $\bar{\tau}$ on pressure distribution for a convergent gap bearing at different settings of the slider temperature
Fig. 9 Effect of \( \bar{\gamma} \) on pressure distribution for a divergent gap bearing at different settings of the slider temperature

Fig. 10 Effect of \( \bar{\gamma} \) on load capacity versus \( K \) for different settings of the slider temperature
divergent gaps also, no significant difference is visible up to $K = 0.3$. As the gap diverges further, it is observed that load capacity increases with an increase in $\gamma$. This is a direct consequence of an increase in inlet pressure with an increase in $\gamma$, as seen in Fig. 9.

Figure 11 shows the effect of $\gamma$ on torque on the bearing surface versus $K$ at $\beta = 5$. It is observed that there is no significant effect of $\gamma$ on torque when the gap is convergent. However, for a divergent gap, torque increases with an increase in $\gamma$. The increase in torque can be justified on the basis of high pressure in the inlet, for divergent gaps. From equation (18), it is seen that torque depends not only on the velocity gradient $(\partial w/\partial y)$ but also on the viscosity. It was seen that $w(x, y)$ remains almost unaffected due to variations in $\gamma$; however, the inlet pressure was seen to increase significantly. Thus, the term $\gamma p$ in equation (18) leads to an increase in the viscosity, which in turn increases the torque on the bearing surface.

### 6 CONCLUSIONS

The present paper analyses the non-constant gap externally pressurized conical bearing. Viscosity of the lubricant is assumed to vary both with pressure and temperature. The inertia effect due to lubricant flow has been neglected, but rotational inertia has been taken into account. The effect of the viscosity–pressure exponent, $\gamma$, on various characteristics of the bearing has been studied.

It is observed that variation of $\gamma$ does not affect the pressure distribution significantly for constant and convergent gap bearings, whereas for the divergent gap bearing there is a significant increase in pressure in the inlet region with an increase in $\gamma$. Further, the effect of $\gamma$ is more pronounced when the slider cools down. It is seen that variation of $\gamma$ does not produce any significant change in the load capacity for convergent and constant gap conical bearings. For the divergent gap also, no significant difference is seen when the gap is slightly divergent. However, for highly divergent gaps, it is observed that load capacity increases with an increase in the viscosity–pressure exponent. It is also seen that variation in $\gamma$ does not produce any significant change in torque of the bearing.

Since the pressure–viscosity parameter $\gamma$ influences the load capacity, it would seem interesting to use a more complex form for describing the viscosity pressure–temperature relationship. For instance, a coupling between $\gamma$ and $\beta$ could be considered. This would make the resulting computation much more complex and is under active consideration.

The results obtained in this work confirm the concept that well-designed bearings are remarkably tolerant assemblies.

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**Fig. 11** Effect of $\gamma$ on torque versus $K$ for different settings of the slider temperature

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Proc Instn Mech Engrs Vol 214 Part C
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