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The aim of the present chapter is to study the problem of axial current induced pinch on the behaviour of a non-Newtonian squeeze film between circular plates with a concentric circular pocket accounting for both temporal as well convective lubricant inertia. Both the methods - momentum integral method as well as energy integral method have been used.

5.1 INTRODUCTION

Archibald (1956) analysed the behaviour of Newtonian fluid lubricated squeeze films between parallel bearing surfaces of various configurations. However, the effect of inertia was neglected by him. Na (1966) obtained an inertialess solution for non-Newtonian squeeze films. Ramanaiah (1967) analysed the problem of squeeze film with power law fluid considered as lubricant and the inclusion of inertia effects, using the method of average inertia. However, he did not include the local inertia in the equation of motion. Elkough (1976) accounted for all the inertia terms in the equation of motion for a laminar non-Newtonian squeeze film. He assumed fluid to be incompressible, exhibiting viscometric properties in agreement with a power law fluid and by using the momentum and energy integral approach,
obtained expressions for the pressure and the load capacity. He discussed non-Newtonian and inertia effects and compared with the available expressions. In all these analyses the lubricant was considered to be electrically non-conducting and the plates were considered without pocket.

Considerable attention has been paid to the development of the MHD bearings particularly for the ones utilized under the high temperature conditions such as in Rankine cycle for space vehicles or coolant circulating pumps for nuclear power plants. Hence, for high temperature conditions use of liquid metals which has high thermal conductivity has become imperative. But these lubricants have low viscosities.

With current trends towards higher operating speeds, the inertia effects are likely to be of increasing importance. Ramanaiah (1966b) studied the problem of squeeze film between circular plates with axial current induced pinch effect using power law fluid as lubricant, neglecting lubricant inertia in his analysis.

In many practical situations, involving mechanical elements, pockets are also encountered in the form of dents and cavities. These pockets are mostly circular and concentric in nature and result from the wearing out of the
material due to rotating motion. Hence the study of lubrication involving such machine elements is also important from practical point of view.

Here we analyse the problem of axial current induced pinch on the behaviour of a non-Newtonian squeeze film between circular plates with concentric circular pocket including the lubricant inertia in Navier Stockes equations using the energy integral and the momentum integral approach.

5.2. ANALYSIS:

The stress-rate strain relation for a most common fluid as a non-Newtonian lubricant called power law fluid may be represented by:

$$\tau = -[m|\frac{1}{2}(\Delta: \Delta)^{n-1}|] \Delta$$

where, \(\tau\) is the stress, \(\Delta\) is the rate of deformation tensor, \(m, n\) are experimental viscometric constants. When \(n=1\), the fluid is called Newtonian, for \(n < 1\), the fluid is called pseudoplastic fluid and for \(n > 1\), the lubricant is called dilatant fluid. The lubricant is assumed to be electrically conducting and is contained between two circular plates, the upper plate approaching the lower one with a normal velocity. The radii of the plates are 'a'. The concentric pocket is located in the lower plate along \(b < r < d\)
Where 'b' is the inside radius of the pocket and 'd' is the outside radius of the pocket. The two circular plates are taken to be ideal conductors. A potential is applied between the plates, hence an axial current density $J_z$ will exist between the plates. This current density gives rise to an azimuthal magnetic induction and interacts with it to result in the pinch effect. Making the usual assumptions of hydromagnetic lubrication applicable to thin films and retaining the inertia terms, one obtains the following equation of motion in the r-direction for the flow system shown in fig 5.1. [Ramanaiyah(1966b)].

$$
\frac{\partial}{\partial t} \left( \rho v_r \right) + v_r \frac{\partial v_r}{\partial r} + v_z \frac{\partial v_r}{\partial z} = -\frac{\partial p}{\partial r} + \frac{\partial}{\partial z} \left[ m | \frac{\partial v_r}{\partial z} |^{n-1} \frac{\partial v_r}{\partial z} \right] - c r. 
$$

where $\rho$ is the density of the lubricant, $v_r$ and $v_z$ are the radial and axial components of lubricant velocity, $p$ is the lubricant pressure, $c = \frac{\mu_e I^2 a^4}{2\pi}$ is the current parameter characterizing the effect of axial current induced pinch. $I = J_z \pi a^2$ is the total current, and $\mu_e$ is the magnetic permeability of free space. The equation of continuity is:
Figure 5-1. Configuration of bearing.
\[
\frac{1}{r} \frac{\partial}{\partial r} (r v_r) + \frac{\partial}{\partial z} (v_z) = 0 \tag{5.2}
\]

The boundary conditions are

\[
v_r (r, o, t) = v_r (r, h, t) = v_z (r, o, t) = 0
\tag{5.3}
\]

\[
v_z (r, h, t) = \dot{h} \text{ and } \int_0^h v_r \, dz = -\frac{1}{2} \dot{h} r
\]

where \( h \) is the instantaneous film thickness and \( \dot{h} \) is the velocity of upper moving plate. The inertialess solution, obtained by neglecting the inertia terms in the equation of motion (5.1) is obtained as [Elkough (1976)]:

\[
v_r = - (2)^{1/n} \left( \frac{2n+1}{n+1} \right) r \left( \frac{\dot{h}}{h} \right) \left[ \left( \frac{1}{2} \right)^{\frac{n+1}{n}} \right] \left( \frac{1}{2} - \frac{\dot{z}}{h} \right)
\tag{5.4}
\]

where \( \dot{z} = \frac{z}{\dot{h}} \) and

\[
p(r, t) = -2m \left( \frac{2n+1}{n+1} \right)^n \frac{1}{n+1} \frac{\dot{h} |\dot{h}|}{h^n} \left[ \left( \frac{a}{h} \right)^{n+1} - \left( \frac{r}{h} \right)^{n+1} \right] + \frac{c}{2} (a^2 - r^2)
\tag{5.5}
\]

5.3 ENERGY INTEGRAL METHOD

To obtain the effect of the inertia terms on the flow,
equation (5.1) is multiplied by \( v_r \) and integrated across the film to give the energy integral equation:

\[
3 \left[ \int_{0}^{h} \left( v_r \frac{\partial v_r}{\partial t} + v_r^2 \frac{\partial v_r}{\partial t} + \frac{v_r^2}{2r} \frac{\partial}{\partial r} (r \cdot v_z) \right) \, dz \right] = - \int_{0}^{h} v_r \frac{\partial p}{\partial r} \, dz + (m)(n) \int_{0}^{h} v_r \frac{\partial^2 v_r}{\partial z^2} \left| \frac{\partial v_r}{\partial z} \right|^{n-1} \, dz - c \cdot r \int_{0}^{h} v_r \, dz
\]

(5.6)

where equation (5.2) and the boundary condition (5.3) have been used.

Substituting \( v_r \) from equation (5.4) into equation (5.6) one gets:

\[
\frac{\partial p}{\partial r} = 2m \left( \frac{2n+1}{n} \right)^n \frac{\dot{r}}{h} \left| \frac{\dot{r}}{h} \right|^{n-1} \frac{r^n}{h^{2n+1}} + 3r \left\{ \left( \frac{\dot{h}}{h} \right)^2 \left( \frac{2n+1}{2(2n+1)^2(3n+2)} \right) \left( \frac{n(n+1)}{(4n+3)} \right) \right\} - c \cdot r
\]

(5.7)

Integrating equation (5.7) with respect to \( r \) with the boundary condition that \( p(a,t) = 0 \), we get:
The dimensionless pressure is

\[ p^* = \frac{\rho g h^{2n}}{m^2 |h|^{2n-2}} \text{ and } A, B, E \text{ are given by} \]

\[ A = \frac{2}{n+1} \left( \frac{2n+1}{n} \right)^n \]

\[ (5.9) \]
B = \frac{2n+1}{2(3n+2)}

E = \frac{(2n+1)}{4(n+1)^2(3n+2)} \left\{ \frac{1}{4} \left( \frac{4n+3}{3n+5} \right) + \frac{n(n+1)}{(4n+3)} \right\}

R_e is the Reynolds number signifying the effect of convective inertia and N is the acceleration squeeze number characterising the effect of acceleration of the moving plate:

\[ R_e = - g h \left| h \right|^{1-n} \frac{h^n}{m} \]

\[ N = \frac{\dddot{h}}{h^{2/3}}. \]

The load carrying capacity is given by

\[ W = \int_b^a 2\pi r p(r,t) \, dr + \int_a^b 2\pi t p(r,t) \, dr. \]

In dimensionless form, it is obtained using equation (5.9) as:

\[ \overline{W} = \frac{W g h^{2n-2}}{m^2 \left| h \right|^{2n-2}} \]

\[ = \pi \left[ {\left( R_e \right)^2 \left( b^{4} - (R_e)^3 \left( b \right)^{4} \right)} \right] \]
\[
(R_e)^2 \left( \frac{c^*}{2}, \frac{b^*}{2} \right) \} + \pi [R_e A a^* \left\{ \left(1 - \frac{d^2}{a^2}\right) - \frac{2}{3} \left(1 - \frac{d^{n+3}}{a^{n+3}}\right) \right\} \\
- (R_e)^2(NB-E) \frac{a^*}{2} \left(1 - \frac{d^2}{a^2}\right)^2 + (R_e)^2 \frac{c^*}{2} \frac{a^*}{2} \left(1 - \frac{d^2}{a^2}\right)^2] \\
\]

(5.10)

where \( F = A \left(\frac{n+1}{n+3}\right) \)

\[ b^* = \frac{b}{h} \]

For centrally located pocket \( b^* = 0 \) and

\[
\bar{W}_c = \pi [A(R_e)a^* \left\{ 1 - \frac{d^2}{a^2}\right) - \frac{2}{3} \left(1 - \frac{d^{n+3}}{a^{n+3}}\right) \} \\
- (R_e)^2(NB-E) \frac{a^*}{2} \left(1 - \frac{d^2}{a^2}\right)^2 + (R_e)^2 \frac{c^*}{2} \frac{a^*}{2} \left(1 - \frac{d^2}{a^2}\right)^2 ] \]

(5.11)

Further, when there is no pocket \( d^* = 0 \), then the load carrying capacity \( \bar{W}_o \) for circular plate is:

\[
\bar{W}_o = \pi [R_e F a^* \left(\frac{n+3}{2}\right) - (R_e)^2(NB-E) \frac{a^*}{2} + (R_e)^2 \frac{c^*}{2} \frac{a^*}{2} ] \\
\]

(5.12)
If there is no axial current we put $c^* = 0$ in equations (5.10), (5.11) and (5.12) to get corresponding load carrying capacities. These results are valid for small values of Reynolds number $R_e$ and the acceleration squeeze number $N$. The effect of axial current induced pinch on the performance characteristics of the bearing comes through the current parameter $c^*$. The bearing will support a non-zero load even when there is no squeezing motion. The effect of axial current induced pinch on the load carrying capacity is independent of the type of fluid as characterised by $m$, $n$ and also the inertia effect. For $n=1$, the present results reduces to those corresponding to the case of Newtonian fluid. For $b^* = 0$ and $d^* = 0$ with $c^* = 0$, the results reduce to those obtained by Elkough (1976). The increase in the pocket radius resulting from wear or design consideration reduces the load carrying capacity of the bearing.

5.3.1 Conclusions

Numerical results for load carrying capacity $\bar{W}$ are tabulated in Tables 5.1 and 5.2 for $R_e = 1$, $N = -1$, $n = 2$ and $a^* = 100$. $\bar{W}$ is computed for different values of $c^*$ and $b^*$ for $d^* = 40$, and shown in Table 5.1. It is easily noticed that while the increasing axial pinch, increases the load carrying capacity, the reduction in pocket size increases it. In Table
5.2, \( \bar{W} \) is presented for different values of \( b^* \) and \( d^* \) and a fixed value of \( c^* = 0.5 \). This table shows the effect of pocket size as well as its location on the load carrying capacity. For a fixed \( b^* \), the increase in \( d^* \) results in the reduction of load capacity and an opposite effect is seen for a fixed \( d^* \) with increasing \( b^* \). In conclusion, we observe that a bearing with axial current induced pinch effect performs better than an identical conventional bearing. Also, if pockets are to be included then a bearing with outside pocket radius fixed will perform better if inside radius is kept at a maximum. Similarly if inside radius of the pocket is kept fixed then for a better performance of the bearing outside radius should be minimum.

5.4 MOMENTUM INTEGRAL METHOD:

To obtain the effect of the inertia terms on the flow, equation (5.1) is integrated across the film to give the momentum integral equation

\[
g \left[ \int_0^h \left( \frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + \frac{v_r}{r} \frac{\partial}{\partial r} \left( rv_r \right) \right) \, dz \right] = \\
- \int_0^h \frac{\partial p}{\partial r} \, dz + (m)(n) \int_0^h \frac{\partial^2 v_r}{\partial z^2} \left| \frac{\partial v_r}{\partial z} \right|^{n-1} \, dz - cr \int_0^h \, dz \tag{5.13}
\]

where equation (5.2) and the boundary conditions (5.3) have
been used.

Substituting $v_r$ from equation (5.4) into equation (5.13)

$$\frac{\partial p}{\partial r} = 2m \left( \frac{2n+1}{n} \right) \left( \frac{h|h|}{h^{2n+1}} \right) r^n + g \left[ \frac{h}{2h} - \frac{h}{n} \frac{3(2n+1)}{2(3n+1)} \right] r - cr$$

(5.14)

Integrating equation (5.14) with respect to $r$ with the boundary condition $p(a,t) = 0$

$$p = -2m \left( \frac{2n+1}{n} \right) n \left( \frac{h|h|}{h^{2n+1}} \right) \left( \frac{1}{n+1} \right) (a^{n+1} - r^{n+1})$$

$$- g \left[ \frac{h}{2h} - \left( \frac{h}{h} \right)^2 \frac{3(2n+1)}{2(3n+2)} \right] \frac{1}{2} \left( a^2 - r^2 \right) + \frac{c}{2} \left( a^2 - r^2 \right)$$

(5.15)

The dimensionless pressure is:

$$p^* = Re A (a^{*n+1} - r^{*n+1}) - (Re)^2 \left[ \frac{N}{4} - \frac{3B}{2} \right] (a^{*2} - r^{*2})$$

$$+ (Re)^2 \frac{c^*}{2} (a^{*2} - r^{*2})$$

(5.16)

where $A$ and $B$ are given by

$$A = \left( \frac{2}{n+1} \right) \left( \frac{2n+1}{n} \right)^n$$
\[ B = \frac{2n + 1}{2(3n+2)} \]

\[ N = \frac{\mathcal{h}^2}{\mathcal{h}^2} \text{ is the acceleration squeeze number characterising the effect of acceleration of the moving plate.} \]

\[ R_e = - g \cdot \frac{\mathcal{h}^2}{|\mathcal{h}|^{1-n}} \cdot \frac{\mathcal{h}^n}{m} \]

\[ p^* = \frac{3\rho \mathcal{h}^{2n}}{m^2 |\mathcal{h}|^{2n-2}} \]

\[ r^* = \frac{r}{h}, \text{ is the dimensionless radial coordinate,} \]

\[ a^* = \frac{a}{h}, \text{ is the dimensionless radius of circular plate, and} \]

\[ c^* = \frac{c}{h} \left( \frac{h}{\mathcal{h}} \right)^2 \text{ is the dimensionless current parameter.} \]

The load carrying capacity is given by:
\[ \mathcal{W} = \frac{b}{2\pi} \int_0^a r \mathcal{p}(r,t) \, dr + \int_0^a 2\pi r \mathcal{p}(r,t) \, dr \]

In dimensionless form, it is obtained using equation (5.16) as:
\[ \bar{W} = \frac{\mathcal{W} \mathcal{h}}{m^2 |\mathcal{h}|^{2n-2}} \]
\[
= \pi \frac{[Re \cdot D \cdot b^*]^{n+3}}{2} - (Re)^2 \left( \frac{N}{4} - \frac{3B}{2} \right) \frac{b^*}{2} + (Re)^2 c^* \frac{b^*}{4}
\]

\[
+ \pi \left[ A(Re) a^* \left\{ \left( 1 - \frac{d}{a} \right)^2 - \left( \frac{2}{n+3} \right) \left( 1 - \frac{d}{a} \right)^2 \right\} \right]
\]

\[
- (Re)^2 \frac{a^*}{2} \left( \frac{N}{4} - \frac{3B}{2} \right) \left( 1 - \frac{d^2}{a^2} \right) + (Re)^2 \frac{a^*}{4} c^* \left( 1 - \frac{d^2}{a^2} \right)^2
\]

(5.17)

where \(D = A \left( \frac{n+1}{n+3} \right)\)

\(b^* = \frac{b}{n}\) is the dimensionless inside radius of the pocket.

For centrally located pocket \(b^* = 0\) and in this case

\[
\bar{W}_c = \pi \left[ A \ Re \ a^* \left\{ \left( 1 - \frac{d}{a} \right)^2 - \left( \frac{2}{n+3} \right) \left( 1 - \frac{d}{a} \right)^2 \right\} \right]
\]

\[
- (Re)^2 \frac{a^*}{2} \left( \frac{N}{4} - \frac{3B}{2} \right) \left( 1 - \frac{d^2}{a^2} \right) + (Re)^2 \frac{a^*}{4} c^* \left( 1 - \frac{d^2}{a^2} \right)^2 \]

(5.18)

Further, when there is no pocket \(d^* = 0\), then the load carrying capacity for circular plate is:
\[ \bar{\nu}_o = \pi \left[ R_e \cdot D \cdot a^* \cdot n + 3 \left( R_e \right)^2 \left( \frac{N}{4} - \frac{3B}{2} \right) \frac{a^*}{2} \right] + \left( R_e \right)^2 c^* \frac{a^*}{4} \]  \tag{5.19}

The results are valid for the range of values of the Reynolds number \( R_e \) for which the motion of the lubricant remains laminar. The effect of axial current induced pinch on the performance characteristics of the bearing comes through the parameter \( c^* \). It is easily seen from equations (5.17)–(5.19) that the bearing will support a nonzero load even when there is no squeezing motion. The effect of axial current induced pinch on the load carrying capacity is independent of the type of lubricant as characterised by \( m, n \) and also the inertia effect. For \( n=1 \), the present results reduce to those corresponding to the case of Newtonian fluid. For \( b^*=0 \) and \( d^*=0 \) with \( c^*=0 \), the result reduce to those obtained by Elkough (1976). The increase in the pocket radius resulting from wear or design considerations reduces the load carrying capacity of the bearing.

5.4.1. CONCLUSIONS:

Numerical results are tabulated in Tables 5.3 and 5.4 for \( R_e=1, N=-1, n=2 \) and \( a^*=100 \). \( \bar{\nu} \) is computed for different
values of $c^*$ and $b^*$ for $d^* = 40$, and shown in Table 5.3. It is easily noticed that while the increasing axial pinch increases the load carrying capacity, the reduction in pocket size increases it. In Table 5.4, $\bar{W}$ is presented for different values of $b^*$ and $d^*$ and for a fixed value of $c^*=0.5$. This table shows the effect of pocket size as well as its location on the load carrying capacity for a fixed $b^*$, the increase in $d^*$ results in the reduction of load carrying capacity and an opposite effect is seen for a fixed $d^*$ with increasing $b^*$. Numerical results for a circular bearing without pocket are tabulated in Tables 5.5 and 5.6 for $R_e = 1$, $N = -1$, $n=2$ and $a^* = 100$ for different values of $C^*$. In Table 5.5 the lubricant pressure distribution $P^*$ given for different values of $r^*$ and $c^*$. It is observed that the increasing values of $c^*$ increase both the, lubricant pressure as well as load carrying capacity of the bearing.

In conclusion, we observe that a bearing with axial current induced pinch effect performs better than an identical conventional bearing. Also, if pockets are to be included then a bearing with $d^*$ fixed will perform better if the inside radius $b^*$ of the pocket is kept at maximum. Similarly if inside radius of the pocket is kept fixed then for a better performance of the bearing outside radius should be minimum.
Table 5.1 Non-dimensional load carrying capacity for different values of $c^*$ and $b^*$ for $d^* = 40$, $R_e = 1$, $N = -1$, $n = 2$ and $a^* = 100$ \(10^{-8} \bar{w}\)

<table>
<thead>
<tr>
<th>$b^*$</th>
<th>0</th>
<th>5</th>
<th>10</th>
<th>15</th>
<th>20</th>
</tr>
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<tbody>
<tr>
<td>0.00</td>
<td>581.96405</td>
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<td>581.97068</td>
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<tr>
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</tr>
<tr>
<td>0.50</td>
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<td>582.51799</td>
<td>582.51895</td>
<td>582.52498</td>
<td>582.54454</td>
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</tbody>
</table>
Table 5.2 Non-dimensional load carrying capacity for different values of \( d^* \) and \( b^* \) for \( c^* = 0.5, R_e = 1, N = -1, n = 2 \) and \( a^* = 100 \times 10^{-8} \bar{W} \)

<table>
<thead>
<tr>
<th>( b^* )</th>
<th>0</th>
<th>5</th>
<th>10</th>
<th>15</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>( d^* )</td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>25</td>
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Table 5.3 Non-dimensional load carrying capacity for different values of $c^*$ and $b^*$ for $d^* = 40$, $R_e = 1$, $N = -1$, $n = 2$ and $a^* = 100$ 

\[ 10^{-8} \frac{W}{n} \]

<table>
<thead>
<tr>
<th>$c^*$</th>
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<td>582.49183</td>
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Table 5.4 Non-dimensional load carrying capacity for different values of $d^*$ and $b^*$ for $c^* = 0.5$, $R_e = 1$, $N = -1$, $n = 2$ and $a^* = 100 	imes 10^{-8} \bar{W}$

<table>
<thead>
<tr>
<th>$d^*$</th>
<th>0</th>
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<th>10</th>
<th>15</th>
<th>20</th>
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<td>670.04059</td>
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<td>628.65685</td>
<td>628.70866</td>
<td>628.90248</td>
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<td>40</td>
<td>582.09878</td>
<td>582.09903</td>
<td>582.10678</td>
<td>582.15859</td>
<td>582.35241</td>
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</table>
Table 5.5 Variation of lubricant pressure distribution for different values of $r^*$ and $c^*$ for $R_e = 1$, $N = -1$, $n = 2$ and $a^* = 100$

\[
10^{-3} p^*\]

<table>
<thead>
<tr>
<th>$r^*$</th>
<th>$c^*$</th>
<th>0.0</th>
<th>0.5</th>
<th>1.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>00</td>
<td>4173.854</td>
<td>4176.354</td>
<td>4178.854</td>
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<td>4142.633</td>
<td>4145.034</td>
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<td>3908.138</td>
<td>3910.238</td>
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<tr>
<td>80</td>
<td>2035.921</td>
<td>2036.821</td>
<td>2037.721</td>
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<tr>
<td>100</td>
<td>0.0</td>
<td>0.0</td>
<td>0.0</td>
<td></td>
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</tbody>
</table>
TABLE 5.6
Variation of non-dimensional load carrying capacity for different values of $c^*$ for $R_e = 1$, $N = -1$, $n = 2$, $a^* = 100$

<table>
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<th>$c^*$</th>
<th>$10^{-8} \bar{W}_o$</th>
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<tr>
<td>0.00</td>
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<td>1.00</td>
<td>786.9134</td>
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