Effect of the location of axial groove on the steady state and dynamic characteristics of oil journal bearings

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Abstract: The steady state and dynamic characteristics including whirl instability of journal bearings with an axial groove, one located at the top and another at the bottom from which oil is supplied at a constant pressure are obtained theoretically. The Reynolds equation is solved numerically by finite difference method satisfying the appropriate boundary conditions. The dynamic behaviour in terms of stiffness and damping coefficients of film and stability are found using a first-order perturbation method. It has been shown that both load capacity and stability improve for smaller groove angle and smaller groove length are used for the bearing feeding from the top.

Keywords : axial groove, journal bearings, stability, steady state.

NOTATION

C	radial clearance (m)
D	diameter of the journal (m)
$D_{rr}, D_{\phi\phi}, D_{r\phi}, D_{\phi r}$	damping coefficients (N s /m)
$\bar{D}_{rr}, \bar{D}_{\phi\phi}, \bar{D}_{r\phi}, \bar{D}_{\phi r}$	non-dimensional damping coefficients, ($\bar{D}_{ij} = D_{ij}C\omega/(LDp_s)$)
e	eccentricity (m)
$h, \bar{h}$	local film thickness (m), $\bar{h} = h/C$
$K_{rr}, K_{\phi\phi}, K_{r\phi}, K_{\phi r}$	stiffness coefficients (N/m)
$\bar{K}_{rr}, \bar{K}_{\phi\phi}, \bar{K}_{r\phi}, \bar{K}_{\phi r}$	non-dimensional stiffness coefficients, ($\bar{K}_{ij} = K_{ij}C/(LDp_s)$)
L	length of the bearing (m)
$M, \bar{M}$	rotor mass per bearing (kg), $\bar{M} = MC\omega^2/LDp_s$
$p, \bar{p}$	film pressure (Pa), $\bar{p} = p/p_s$
p_s	supply pressure (Pa)
η	coefficient of absolute viscosity of the lubricant (Pa-s)
$Q, \bar{Q}$	end flow of oil (m^3 / s), $\bar{Q} = 2Q\eta L/(C^3 Dp_s)$
R	journal radius (m)
t	time (s)
$W, \bar{W}$	load capacity (N), $\bar{W} = W/(LDp_s)$
α	groove angle (deg)
ε	eccentricity ratio, $\varepsilon =$ e/C
$\theta, \bar{z}$	non-dimensional coordinates, $\theta = x/R, \bar{z} = z/(L/2)$

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λ	whirl ratio = ω_p/ω
Λ	bearing number, $6\eta\omega/[p_s(C/R)^2]$
$\mu, \bar{\mu}$	coefficient of friction, $\bar{\mu} = \mu(R/C)$
τ	non-dimensional time, $\tau = \omega_p t$
ϕ	attitude angle (rad)
ψ	assumed attitude angle (rad)
ω	journal rotational speed (rad/s)
ω_p	frequency of journal vibration (rad/s)
$(\ )_0$	steady state value

1 INTRODUCTION

The quantity of oil flow in a journal bearing plays an important role in maintaining an uninterrupted oil film and removing most of the frictional heat to cool the bearing. The oil flow rate depends on several factors, such the viscosity of the lubricant, the geometry (length, diameter and radial clearance) of the bearing, operating eccentricity, the inlet oil pressure and the arrangement of feeding sources.

One of the simplest ways of feeding oil is a single hole through the bearing which is usually a stationary member at the unloaded region .The external load on such a bearing should be unidirectional and constant in magnitude.

In an internal combustion engine bearing both the magnitude and direction of load change. A novel method to cope up this situation is the use of a submerged oil bearing system proposed by Floberg [1] . However, an elaborate arrangement of the feeding system is to be designed.

Oil is also fed by providing axial grooves. Therefore, when a bearing is to be designed with an axial groove, its location is important from the operational point of view. Two such arrangements either feeding from the top or from the bottom may be proposed (Fig.1). In both the cases the radial load is applied from the top.

Several analyses on the grooved journal bearings are available [2-7].These do not consider the stability characteristics. When these bearing are used in high speed machinery, the stability is an important consideration.In the present study two such feeding arrangements have been taken up to find the steady-state and dynamic characteristics including stability.

2 THEORY

The governing equation is the Reynolds equation in two dimensions for an incompressible fluid (Fig. 1). It can be written in dimensionless form as

$$\frac{\partial}{\partial\theta}\left(\bar{h}^3\frac{\partial\bar{p}}{\partial\theta}\right)+(D/L)^2\,\bar{h}^3\frac{\partial^2\bar{p}}{\partial\bar{z}^2}=\Lambda\frac{\partial\bar{h}}{\partial\theta}+2\Lambda\lambda\frac{\partial\bar{h}}{\partial\tau} \tag{1}$$

2.1 Steady state characteristics

Under steady state condition equation (1) can be reduced to

$$\bar{h}_0^3\frac{\partial^2\bar{p}_0}{\partial\theta^2}+3\bar{h}_0^2\frac{\partial\bar{h}_0}{\partial\theta}\frac{\partial\bar{p}_0}{\partial\theta}+(D/L)^2\,\bar{h}_0^3\frac{\partial^2\bar{p}_0}{\partial\bar{z}^2}-\Lambda\frac{\partial\bar{h}_0}{\partial\theta}=0 \tag{2}$$

For axially grooved journal bearings, the boundary conditions are

$\bar{p}_0 = 1$ in the groove, $\bar{p}_0 = 0$ at the bearing ends and the pressure is set equal to 0 when the pressure falls below zero. (3)

In a conventional cylindrical bearing the coordinate θ in the circumferential direction is taken from the position of maximum film thickness. Here in the grooved bearing this position needs to be found beforehand. This is done by assuming an arbitrary value of attitude angle ψ and the coordinate Θ is measured from the vertical position, as shown in Fig.1.Using this ψ the film thickness equation can be written as $\bar{h}_0 = 1+\varepsilon_0\cos(\Theta-\psi)$.The equation (2) is solved numerically by finite difference method using the boundary conditions of the equation (3). The load components along the line of centres and its perpendicular direction are found from

$$\bar{W}_r\left(=\frac{W_r}{LDp_s}\right) = -\frac{1}{2}\int_0^1\int_0^{2\pi} \bar{p}_0\cos(\Theta-\psi)\,d\theta\, d\bar{z} \tag{4a}$$

$$\bar{W}_t\left(=\frac{W_t}{LDp_s}\right) = \frac{1}{2}\int_0^1\int_0^{2\pi} \bar{p}_0\sin(\Theta-\psi)\,d\theta\, d\bar{z} \tag{4b}$$

The load capacity and attitude angle are given by

$$\bar{W} = [\bar{W}_r^{\,2} + \bar{W}_t^{\,2}]^{1/2} \tag{5}$$

$$\phi_0 = \tan^{-1}\left(\frac{\bar{W}_t}{\bar{W}_r}\right) \tag{6}$$

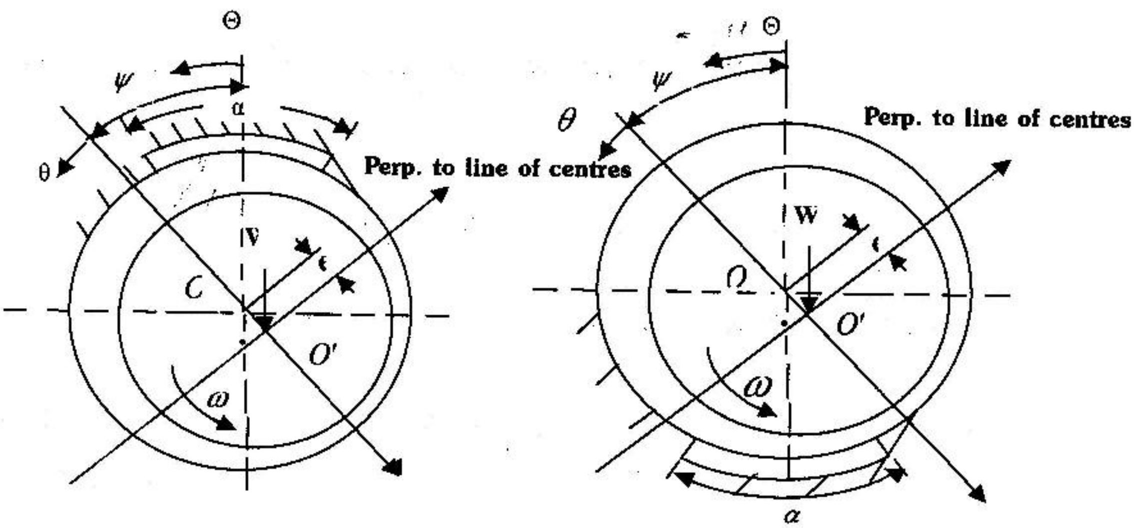

Fig. 1 Axial grooved journal bearings with co-ordinate system

The attitude angle calculated from equation (6) is compared with the assumed value of attitude angle (ψ). The value of ψ is modified with a small increment of ψ and the Reynolds equation is solved using the modified value until ψ is equal to ϕ_0. The volume rate of flow and the coefficient of friction are calculated from the pressure distribution.

The flow rate in the dimensionless form can be written as

$$\bar{Q}\left(=\frac{2Q\eta L}{C^3 p_s D}\right) = -\frac{1}{3}\int_0^{2\pi} \bar{h}_0^{\,3}\, \frac{\partial \bar{p}_0}{\partial \bar{z}}\Big|_{\bar{z}=1} d\theta \tag{7}$$

The friction variable is given by $\bar{\mu} = (\frac{R}{C})\mu = \frac{\bar{F}}{\bar{W}}$

where $\bar{F}(= \frac{F}{2LCp_s}) = \int_0^1 \int_0^{2\pi} \left(\frac{1}{4}\bar{h}_0 \frac{\partial \bar{p}_0}{\partial \theta} + \frac{\Lambda}{12}\frac{1}{\bar{h}_0} \right) d\theta d\bar{z}$

Since no cavitation was observed, the integrations were carried out from 0 to 2π .

2.2 Dynamic Characteristics

The Reynolds equation under dynamic condition is the equation (1). The pressure and film thickness can be expressed for small amplitude of vibration as:

$$\bar{p} = \bar{p}_0 + \varepsilon_1 e^{i\tau}\bar{p}_1 + \varepsilon_0 \phi_1 e^{i\tau}\bar{p}_2 \qquad (8)$$

$$\bar{h} = \bar{h}_0 + \varepsilon_1 e^{i\tau}\cos\theta + \varepsilon_0 \phi_1 e^{i\tau}\sin\theta \qquad (9)$$

Substitution of equations (8) and (9) into the equation (1) and retaining the first linear terms,gives the three differential equations in $\bar{p}_0, \bar{p}_1$ and $\bar{p}_2$. The equations for $\bar{p}_1$ and $\bar{p}_2$ are solved satisfying the modified boundary conditions of equation (3) and known values of $\bar{p}_0$.

Following an approach given by given by Majumdar, Brewe and Khonsari [8], stiffness and damping coefficients are found and used in the equations of motion to obtain the following equations,

$$\bar{M} = \frac{1}{\lambda^2\left(\bar{D}_{\varphi\varphi} + \bar{D}_{rr}\right)}\left[\left(\bar{K}_{rr}\bar{D}_{\varphi\varphi} + \bar{D}_{rr}\bar{K}_{\varphi\varphi}\right) - \left(\bar{K}_{\varphi r}\bar{D}_{r\varphi} + \bar{D}_{\varphi r}\bar{K}_{r\varphi}\right) - \frac{\bar{W}}{\varepsilon_0}\left(\bar{D}_{\varphi r}\bar{W}\sin\varphi_0 - \bar{D}_{rr}\bar{W}\cos\phi_0\right) \qquad (10)$$

$$\bar{M}^2\lambda^4 - \lambda^2\left[\bar{M}\left(\frac{\bar{W}_0\cos\phi_0}{\varepsilon_0} + \bar{K}_{\phi\phi} + \bar{K}_{rr}\right) + \left(\bar{D}_{rr}\bar{D}_{\phi\phi} - \bar{D}_{\phi r}\bar{D}_{r\phi}\right)\right]$$
$$+\left(\bar{K}_{rr}\bar{K}_{\phi\phi} - \bar{K}_{\phi r}\bar{K}_{r\phi}\right) + \frac{\bar{W}_0}{\varepsilon_0}\left(\bar{K}_{rr}\cos\phi_0 - \bar{K}_{\phi r}\sin\phi_0\right) = 0 \qquad (11)$$

Equations (10) and (11) are linear algebraic equations in $\bar{M}$ and λ and solution of these will give $\bar{M}$ and λ. The speed of the journal calculated from this value of is the threshold speed, above which the bearing system will be unstable.

3.0 RESULTS AND DISCUSSION

3.1 Analysis of results for the bearing feeding from the top :

When the bearing operates at a small speed, the hydrodynamic effect is not predominant. The hydrodynamic pressure is insufficient to balance the applied load when fed from top. Thus it is difficult to run the bearing at low speeds. Therefore, there is a speed below which the bearing cannot be operated. In this present analysis, it has been found that the limiting value of non-dimensional speed parameter is Λ = 8.2.To be on the safe side, we have considered the speed parameter Λ is above 10.

Before discussing on the results, the flow rates are compared with the approximate formula(Table1) given by Martin and Lee [4].The flow rate calculated from the present method of solution gives higher value because the flow is due to feed pressure and hydrodynamic pressure.

Table 1 : Comparison of non dimensional flow rate with reference [4] for 36 ° and 18 ° groove angle, L/D=1.0, Λ=10.0 and groove length=1/2 of total length of the bearing

L/ D ratio	ε_0	$\bar{Q}$	$\bar{Q}$ (Martin and Lee) [4]
1	0.2	1.8156 (1.5660)	1.2813 (1.1546)
	0.4	3.2527 (2.9266)	1.8175 (1.6108)
	0.6	5.2971 (4.7910)	2.7000 (2.4417)
	0.8	7.8526 (7.1234)	4.3750 (4.0958)

* Terms in the bracket indicates the corresponding values for 18 ° grooved angle bearing

Table 2 Comparison of 36 ° and 18 ° groove angle bearing having L/D =1.0, Λ=10.0 groove length= ½ of total length of the bearing

ε_0	Groove angle (deg)	$\bar{W}$	ϕ_0	$\bar{Q}$	$\bar{\mu}$	$\bar{M}$	λ
0.2	36	0.0285	84.2595	1.8156	195.3831	4.4219	0.5670
	18	0.0769	83.1530	1.5660	73.0160	3.6415	0.5189
0.4	36	0.5845	70.2401	3.2527	9.7634	7.1472	0.5840
	18	0.6831	68.7400	2.9266	8.8515	6.8051	0.5740
0.6	36	1.7538	54.6000	5.2971	4.0541	15.3080	0.5964
	18	1.8416	53.8861	4.7910	3.8739	15.2697	0.5932
0.8	36	5.5699	37.3000	7.8526	1.8116	39.1319	0.6684
	18	5.6459	37.1456	7.1234	1.7952	39.0850	0.6613

From Table 2 it is seen that both load capacity and stability improve for 18° angle grooved bearing, whereas friction variable is less .The improved load may be due to higher pressure development in the larger land area. However, it can be seen that the friction forces (coefficient of friction multiplied by load) are higher for smaller grooves.This is expected from physical point of view. Therefore, a bearing with a smaller groove angle and groove length (as shown in Table 3) gives better load and stability at the cost of higher power loss due to viscous friction .The load carrying capacity and friction variable of a bearing having 36 ° groove angle are shown in Fig. 2 for various bearing numbers and eccentricity ratios.

Load capacity increases with an increase in eccentricity, as expected. The friction variable decreases as the eccentricity ratio increases. The load capacity increases with bearing number, which is a function of journal speed. This increase is sharp at higher eccentricity ratio.The coefficient of friction decreases with an increase in journal speed .The rise in friction is particularly high at lower eccentricity ratios, as shown in Fig. 2.

The mass parameter $\bar{M}$ and whirl ratio λ are used as a measure of stability. These are plotted in Fig.3 .The upper portion of the curve is unstable and the lower one is stable .The bearing should be operated in the stable region. The stability increases sharply at a very high eccentricity ratio.

Table 3 Comparison of results of 18 °groove angle bearing having L/D =1.0, Λ =10.0 and groove length= ½ and 1/ 4 of total length of the bearing

ε_0	Groove length	$\bar{W}$	ϕ_0	$\bar{Q}$	$\bar{\mu}$	$\bar{M}$	λ
0.2	1/2	0.0769	83.1530	1.5660	73.0160	3.6415	0.5189
	1/4	0.1865	81.8925	1.2346	30.1540	2.9270	0.5543
0.4	1/2	0.6831	68.7400	2.9266	8.8515	6.8051	0.5740
	1/4	0.7897	67.6619	2.4645	7.6900	6.5319	0.5826
0.6	1/2	1.8416	53.8861	4.7910	3.8739	15.2697	0.5932
	1/4	1.9423	53.3650	4.0430	3.6882	15.2376	0.5955
0.8	1/2	5.6459	37.1456	7.1234	1.7952	39.0850	0.6613
	1/4	5.7390	36.9600	5.9094	1.7707	39.1162	0.6628

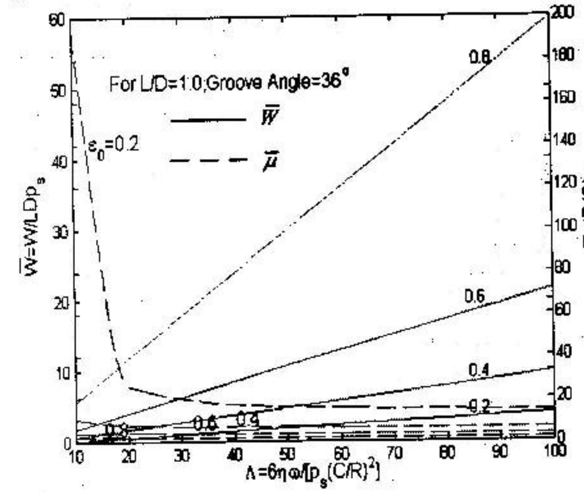

Fig. 2 Variation of load capacity and friction variable with Λ for various ε_0

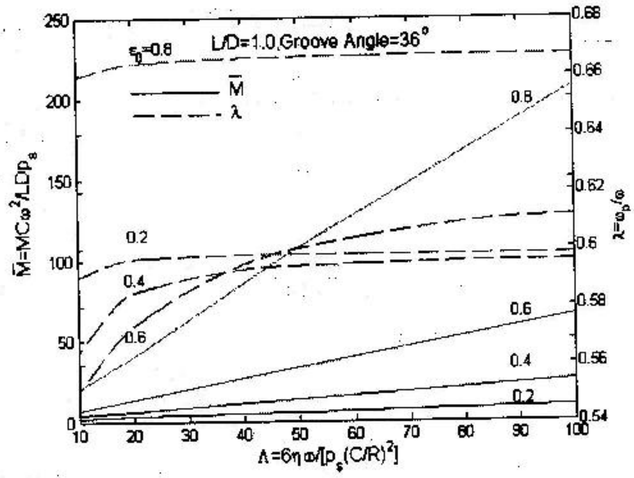

Fig. 3 Variation of stability and whirl with Λ ratio for various ε_0

3.2 Analysis of results for the bearing feeding from the bottom:

Although there is no problem as indicated in section 3.1 for a bearing feeding from the top , in the present feeding arrangement(feeding from the bottom) ,the results here are also presented for speed parameter above 10 for the sake of uniformity.

From Table 4 it is seen that both load capacity and stability improves for 18°angled grooved bearing, where as friction variable is less but the magnitude of the load carrying capacity is much lower as compared to the bearing that is fed from the top. However it is seen that frictional forces (coefficient of friction multiplied by load) are higher for smaller grooves.

The load capacity and friction variable of a bearing having 36 ° groove angle are as shown in Fig. 4 for various bearing numbers and eccentricity ratios.Load capacity increases with an increase in the ecentricity ratio. The load capacity increases with bearing number, which is a function of journal speed.The increase is higher at higher eccentricity ratios. The coefficient of friction increases with an increase in journal speed

The mass parameter, $\bar{M}$ and the whirl ratio, λ are given if Fig.5.

Table 4 Comparison of results of 36 ° and 18 ° groove angle bearing
L/D =1.0 , Λ = 10.0, groove length= ½ of the total length of the bearing

ε_0	Groove angle (deg)	$\bar{W}$	ϕ_0	$\bar{Q}$	$\bar{\mu}$	$\bar{M}$	λ
0.2	36	0.6001	77.7619	1.4784	9.4168	2.5110	0.6686
	18	0.5496	76.9173	1.3765	9.9928	0.7310	0.6567
0.4	36	0.7198	68.1579	1.9128	8.4529	4.7000	0.5690
	18	0.7266	67.0096	1.9110	8.0680	3.0623	0.6190
0.6	36	0.9442	57.4952	2.4377	7.4354	7.8000	0.5450
	18	1.0642	51.7007	2.5565	6.2588	6.1901	0.5517
0.8	36	1.6723	43.7636	3.2123	5.6445	15.0500	0.5640
	18	2.2307	41.4725	3.4641	4.0230	16.2700	0.5199

Table 5 Comparison of results for the bearing feeding from bottom having 36° groove angle, L/D =1.0, Λ =10.0 and groove length= ½ and ¼ of total length of the bearing

ε_0	Groove angle (deg) (=½ and ¼ of total length)	$\overline{W}$	ϕ_0	$\overline{Q}$	$\overline{\mu}$	$\overline{M}$	λ
0.2	½	0.6001	77.7619	1.4784	9.4168	2.5110	0.6680
	¼	0.5421	77.5930	1.2955	0.4274	2.4000	0.6516
0.4	½	0.7198	68.1579	1.9128	8.4529	4.7000	0.5690
	¼	0.7238	67.3900	1.8979	8.4098	4.8400	0.5700
0.6	½	0.9442	57.4952	2.4377	7.4354	7.8000	0.5450
	¼	1.0925	55.8887	2.5890	6.4456	9.0300	0.5528
0.8	½	1.6723	43.7636	3.2123	5.6445	15.0500	0.5640
	¼	2.4111	41.3933	3.4719	3.9735	20.0700	0.5921

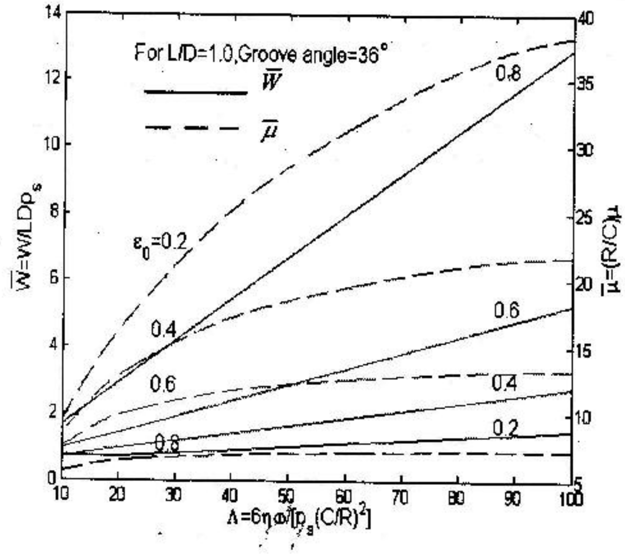

Fig. 4 Variation of load capacity and friction variable with Λ for various ε_0

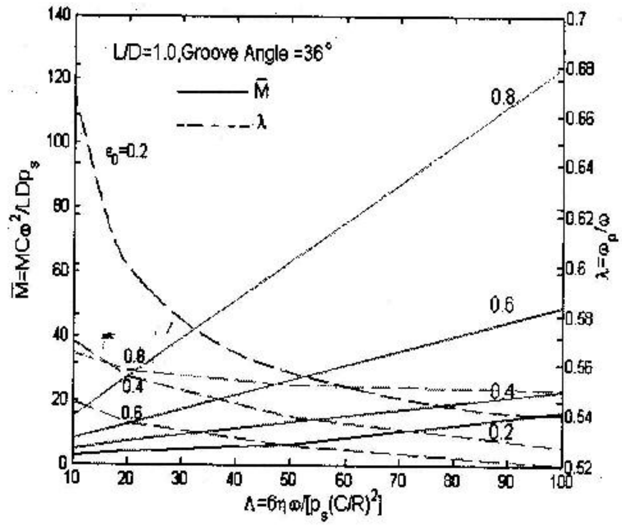

Fig. 5 Variation of stability and whirl with Λ ratio for various ε_0

4. CONCLUSION

From the study the following conclusions are evident:

1. As expected, the load carrying capacity is more when an axial groove is located at the top as compared to the one fed from the bottom.
2. A bearing having smaller groove angle gives higher load capacity. This is due to high pressure in the land region. Flow rate is substantially higher as compared to that of feeding from the bottom.
3. As the flow rate shows an increase in magnitude with eccentricity and speed it would seem to be provide removal of heat sufficiently as compared to the feeding from the bottom.
4. Stability increases for the bearing feeding from the top at higher eccentricity ratio. The stability also improves when smaller groove angles are used.
5. The data obtained from the above analysis can be used conveniently in the design of such bearings, as these are presented in dimensionless form.

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