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Effect of number and size of recess on the performance of hybrid (hydrostatic/hydrodynamic) journal bearing

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Abstract

This paper describes a theoretical study concerning static performance of four pocket rectangular recess hybrid journal bearing. Effect of recess length and width variation, number of recess variation on the load bearing capacity and oil flow parameter for rectangular recess has been carried out. The Reynolds equation for non rotating recessed hybrid bearing was solved on a high speed computer satisfying appropriate boundary conditions and using finite difference method. Various results for different recess axial length to bearing axial length, different recess circumferential length to circumferential length of bearing, various L/D ratios and number of recesses are presented.

Keywords: recessed hybrid bearing, recess, oil flow capacity, load bearing capacity

Nomenclature

A_e : bearing effective area (N×a×b)
N : Number of recess in the bearing
a : Recess axial length
b : recess circumferential length
h : height of fluid film
D : Diameter of bearing
e : eccentricity
ε : eccentricity ratio
γ : Sommerfeld variable
Φ : attitude angle
L : length of journal bearing
X : displacement in x direction
y : displacement in y-direction
β : pressure ratio (P_r/P_s)
W : load carrying capacity
P_s : Supply pressure
r : radius of the shaft
c : clearance
P_r : recess pressure
W : load on the bearing
μ : coefficient of viscosity
D : diameter of journal bearing

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1. Introduction

Hydrostatic bearing is defined as a system of lubrication in which the load supporting fluid film, separating the two surfaces, is created by an external source, like pump, supplying sufficient fluid under the pressure. Since the lubricant is supply under pressure, this type of bearing called externally pressurized bearing. In this type bearing as the pump start high pressure fluid is admitted in the clearance space, forcing the surface of bearing and journal to separate out. Hydrostatic bearings are of prime importance to the engineers as the machine parts supported on these bearings operate with incomparable smoothness. High load carrying capacity, increased minimum fluid film thickness, long life and increased support damping make them attractive of various precision applications such as turbo machinery, machine tool spindles and precision grinder spindles etc. The development of externally pressurized bearings has progressed over approximately 150 years since Girard (1863) first obtained a patent. Since the 1950s externally pressurized journal bearings have increased in use in engineering applications such as machine tools, test equipment, medical equipment, measuring instruments, and radio telescopes and in the aerospace industry.

Raimondi and Boyd [1] is a pioneer who presented a complete theoretical analysis for hydrostatic journal bearing based on one dimensional model. In the year of 1963 Rippel [2] presented detailed analysis, design procedure and design data for various configurations of multi -recess hydrostatic bearing. Cowley and Kher [3] presented both experimental and theoretical results for a multi-recess hydrostatic bearing. Davies [4] presented a theoretical analysis considering the effect of shaft rotation. O'Donoghue and Rowe [5, 6] developed a design method for externally pressurized bearing based on the pad load and flow coefficients which depend only on the geometrical shape and proportions of the bearing and are independent of the control device used. O'Donoghue and Rowe [7] also gave the exact computer solution of Reynolds equation for a finite multi-recess hydrostatic bearing using finite difference method. Decker and Shaprio[8] reported on the computer aided design procedure for hydrostatic bearings. Ghose [9,10] developed a computer solution of Reynolds equation for bearings with large circumferential sill and showed that an optimum sill angle existed at which load capacity of the bearing was a maximum. Ghose and Majumdar [11] presented computer generated design data in terms of load capacity and oil flow for multi-recess hydrostatic journal bearing.

To the best of author knowledge result available for bearing with large circumferential and axial sills are not enough and the effect of number of recess, axial recess length, circumferential recess length etc are not depicted clearly in the literature. This paper presents the theoretical relation between load carrying capacity, oil flow, number of recess, recess length and recess width.

2. Analysis

A hydrostatic journal bearing system is shown in Fig.1. The oil is supplied at a constant pressure source at a pressure p_s and it enters the bearing through the restrictor. The recess pressure is adjusted in such a manner so that it carries the net load on the bearing.

The generalized Reynolds equation for a non rotating journal using basic assumptions can be written as

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3\frac{\partial p}{\partial y}\right)=0 \tag{1}$$

The non-dimensional lubrication equation that governs the pressure distribution of a laminar incompressible Newtonian film of a non rotating journal bearing is:-

$$\frac{\partial}{\partial \theta}\left(H^3\frac{\partial P}{\partial \theta}\right)+\left(\frac{D}{L}\right)^2 H^3\frac{\partial^2 P}{\partial Y^2}=0 \tag{2}$$

Using the following substitutions

$P=p/p_s$, $H=h/c$, $\theta=x/R$, $Y=y/(L/2)$

Where H is non dimensional film thickness, which is function of θ only:-

$H = 1+ \varepsilon \cos(\theta)$, where $\varepsilon=e/c$

Equation (2) is written in a finite difference form and solved by iteration using a high speed digital computer satisfying the following appropriate boundary condition:

1. The film pressure at the end of bearing, equals to the atmospheric pressure , P=0 at $Y=\pm 1$
2. The pressure field is symmetric about the bearing mid plane, $\partial P/\partial \theta = 0 \; at \; Y = 0$

Equation (2) in finite difference form is written as

$$p_{i,j}=\left\{\left(\frac{D}{L}\right)^2 H_{i,j}{}^3\frac{p_{i+1,j}+p_{i-1,j}}{(\Delta Y)^2}+\frac{H_{i,j+\frac{1}{2}}{}^3 p_{i,j+1}+H_{i,j+\frac{1}{2}}{}^3 p_{i,j-1}}{(\Delta\theta)^2}\right\}\left\{2\left(\frac{D}{L}\right)^2\frac{H_{i,j}{}^3}{(\Delta Y)^2}+\frac{H_{i,j+2}{}^3+H_{i,j-2}{}^3}{(\Delta\theta)^2}\right\}^{-1} \tag{3}$$

Solution scheme:-

1.) The pressure ratio P at one of the stations (say at station 1 where $\theta = 0$) is taken as an arbitrary value and the pressures at the other stations are kept at ambient pressure.

2.) The initial pressures at the mesh points in the film are taken on the basis of a linear pressure drop. Equation (3) is then solved by iteration satisfying the boundary conditions for pressures.

3.) After obtaining the pressure distribution the load, frictional torque and flow components were computed and expressed in dimensionless forms.

4.) This process was repeated by pressurizing each station by the arbitrary pressure while maintaining ambient pressure in the other stations, and bearing factors were computed.

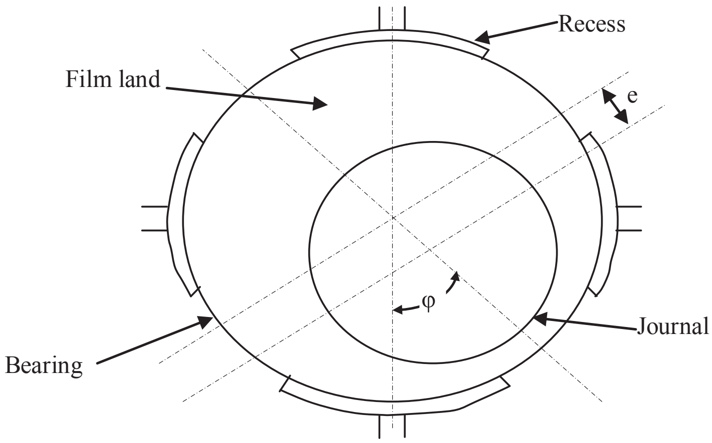

Fig.1: Geometry of recessed hybrid journal bearing

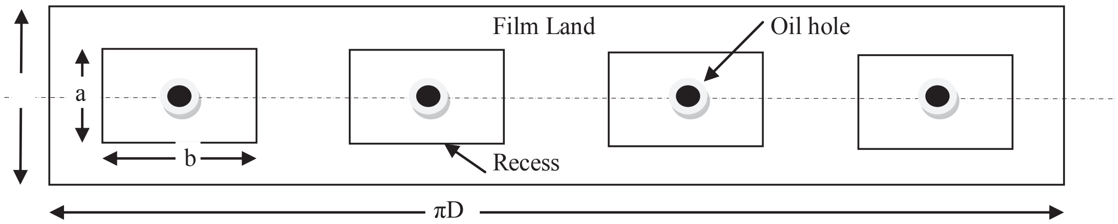

Fig.2: Unwrapped recessed hybrid journal bearing surface

3. Bearing Performance Parameters

3.1. Load Factor

The radial and tangential load components acting on the bearing are determined by integrating the film pressure over the bearing area. For the nth station

Radial load parameter

$$W_{nr} = -\frac{1}{2}\int_0^1\int_0^{2\pi} P\cos\theta\,\partial\theta\,\partial Y$$

Tangential load parameter

$$W_{nt} = -\frac{1}{2}\int_0^1\int_0^{2\pi} P\sin\theta\,\partial\theta\,\partial Y$$

Load parameter

(i.) The radial load parameter is

$$W_r = \sum_1^n W_{rn} P_n$$

(ii.) The tangential load parameter is

$$W_t = \sum_1^n W_{tn} P_n$$

Here W_r and W_t are the radial and tangential load factor of the pressure field that is due to station pressure P_n, Hence the load parameter of the bearing is

$$W = (W_r^2 + W_t^2)^{1/2}$$

Load factor in non dimensional form can be written as

$$\overline{W} = W/(LDPs)$$

Where $W = p_r \times A_e$

or $\overline{W} = p_r \times \frac{Ae}{LD\,p_s}$

where $A_e = Nab$

$$\overline{W} = \frac{p_r Nab}{LD\,p_s}$$

or $\overline{W} = \pi\, \beta . \bar{a} . \bar{b}$ (4)

where $\beta = p_r/p_s$, $\bar{a} = a/L$ and $\bar{b} = Nb/\pi D$

3.2. Flow Parameter

The total bearing flow q is determined by taking the sum of flow through the orifice restrictors of all bearing station and is expressed as

$$Q = \sum_{n=1}^{n} Qn\ \ Pn$$

Flow parameter can be calculated by the formula given by Rowe [12]

$$Q = \frac{p_s h^3 \beta \bar{B}}{\mu} \tag{5}$$

Where $\bar{B}$=bearing flow factor, it is a ratio of bearing area(πDL) to land area $(\pi DL - Nab)$

or put h = c(1+ε cosθ) and $\bar{B} = \frac{\pi DL}{\pi DL - Nab}$ in above Eq. (5)

$$\bar{Q} = \frac{\mu Q}{c^3 p_s} = (1 + \varepsilon \cos\theta)^3 \beta \left(\frac{\pi DL}{\pi DL - Nab}\right)$$

or

$$\bar{Q} = (1 + \varepsilon \cos\theta)^3 \beta \left(\frac{1}{1 - \bar{a}\bar{b}}\right) \tag{6}$$

4. Result

4.1. Effect of L/D Ratio

Dimensionless load capacity $\overline{W}$ vs L/D ratio is shown in Fig 3. I t is observed that the dimensionless load capacity of the bearing decreases with increases with L/D ratio.

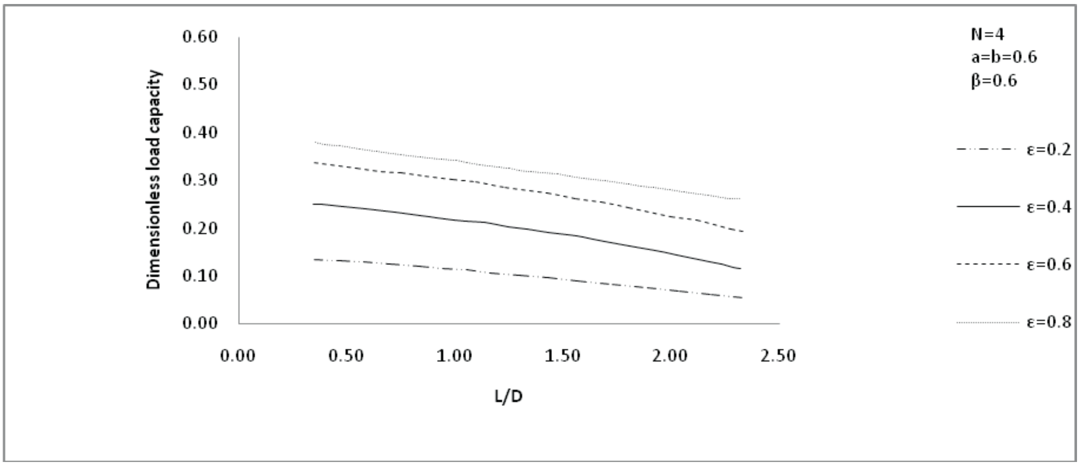

Fig.3 Load capacity vs L/D ratio for different eccentricity ratio

4.2. Effect of Concentric Pressure Ratio

Variation of dimensionless oil flow $\bar{Q}$ vs concentric pressure ratio β is shown in Fig 4. Oil flow is found to increase with increase in β. Since large amount of oil flow required more power to pump the lubricant into the bearing. So, it would be recommended to operate at a value of β=0.6.

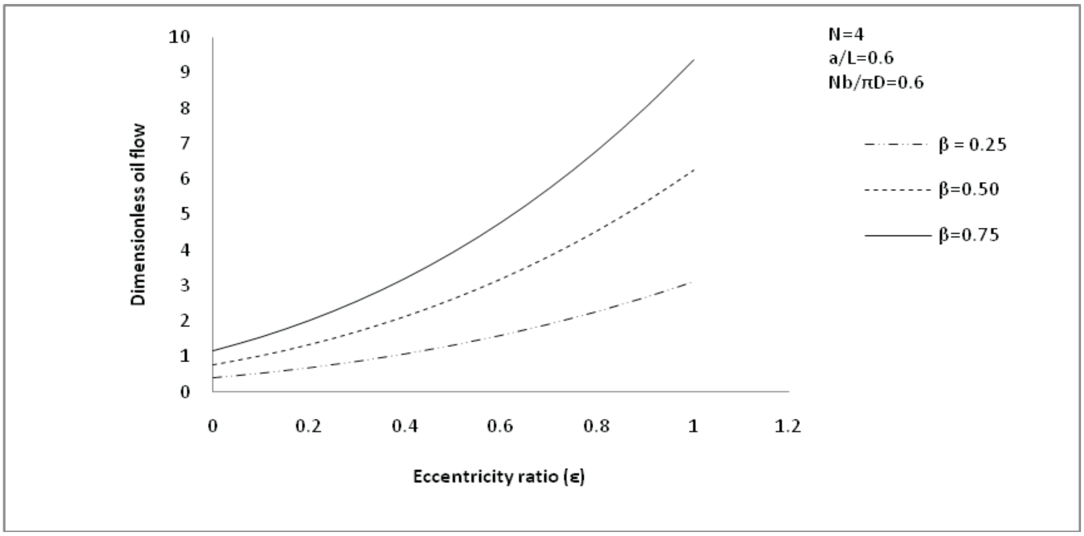

Fig.4 Oil flow of bearing vs eccentricity ratio (ε) for different pressure ratio (β)

4.3 Effect of non dimensional axial recess length ($\bar{a}$) and non dimensional circumferential recess length($\bar{b}$)

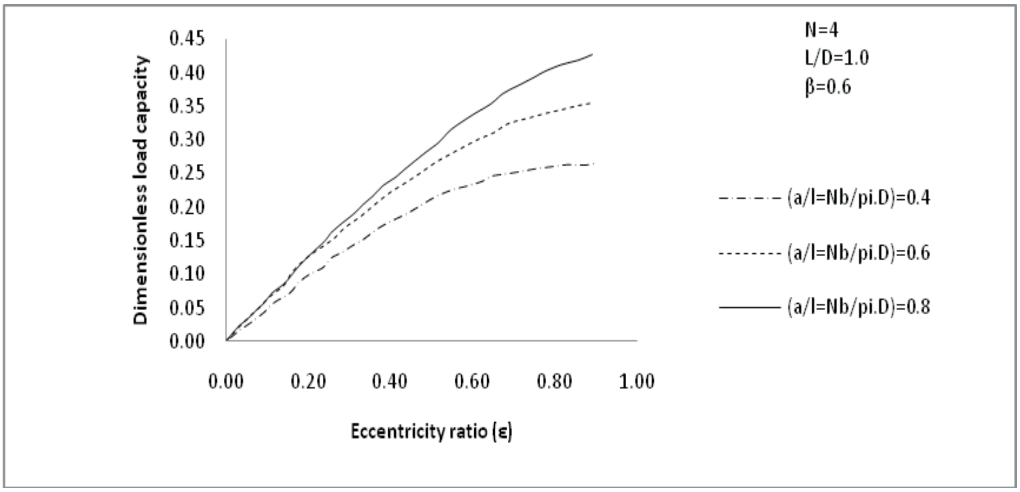

Fig.5 Oil flow of bearing vs eccentricity ratio (ε) for different recess to bearing area ($\bar{a}, \bar{b}$)

Variation of dimensionless load capacity $\overline{W}$ vs eccentricity ratio (ε) ratio is shown in Fig 5 for different $\bar{a}, \bar{b}$. It is seen that dimensionless load capacity $\overline{W}$ is increases with increases in $\bar{a}, \bar{b}$. Variation of dimensionless oil flow $\bar{Q}$ vs eccentricity ratio (ε) ratio is shown in Fig 6 and Fig 7 for different $\bar{a}$, and $\bar{b}$. It is seen that dimensionless oil flow $\bar{Q}$ is increases with increases in $\bar{a}$ as well as $\bar{b}$.

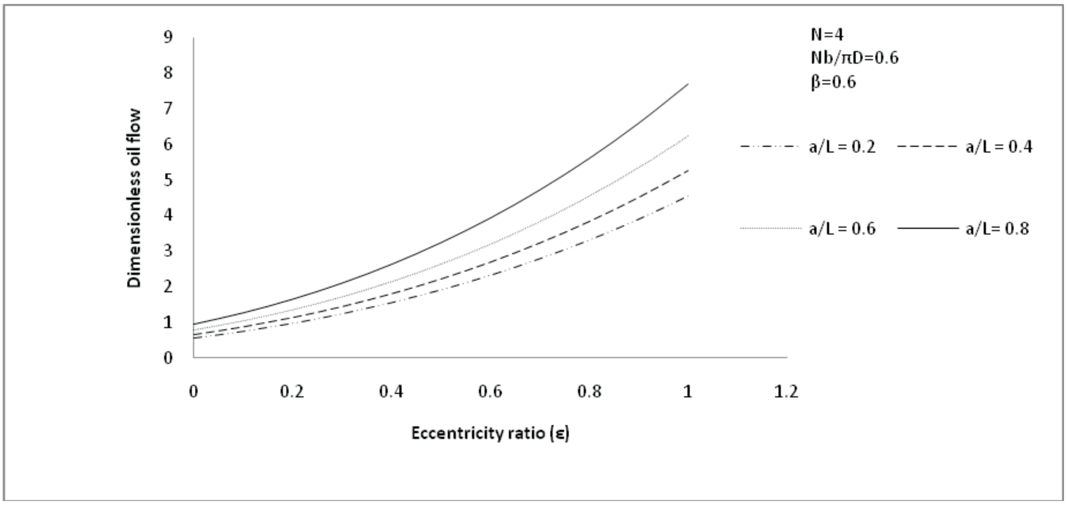

Fig.6 Oil flow of bearing vs eccentricity ratio (ε) for different axial recess length to bearing length ($\bar{a}$)

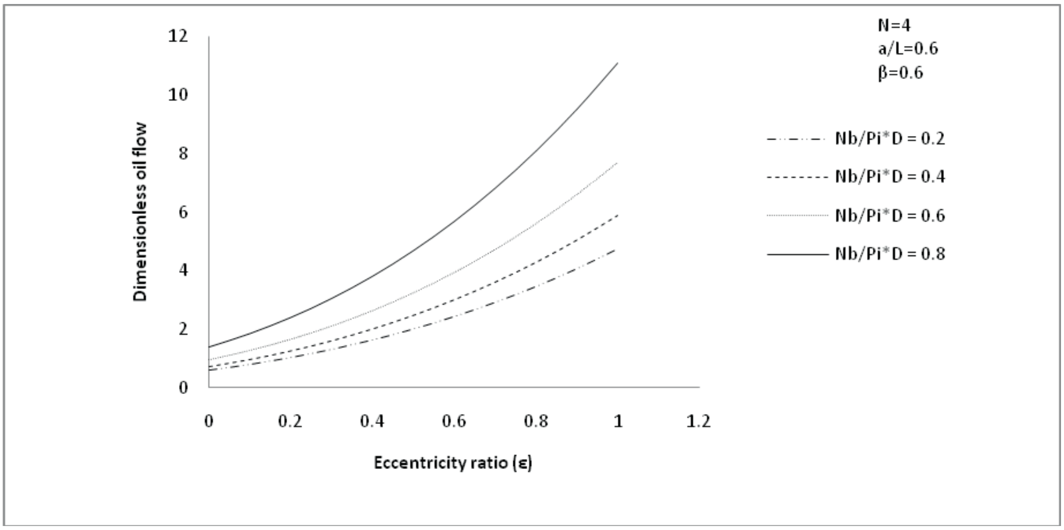

Fig.7 Oil flow of bearing vs eccentricity ratio (ε) for different dimensionless circumferential recess length ($\bar{b}$)

4.4 Effect of Number of Recesses

The variation of dimensionless oil flow $\bar{Q}$ and variation of dimensionless load capacity $\overline{W}$ vs eccentricity ratio (ε) ratio is shown in Fig 8 and Fig 9 for different number of recesses. It is very clear from the figure that the number of recesses has very little effect on either load capacity or oil flow of bearing. However, a symmetrical number of recess (either 4 or 6) is usually selected in the design.

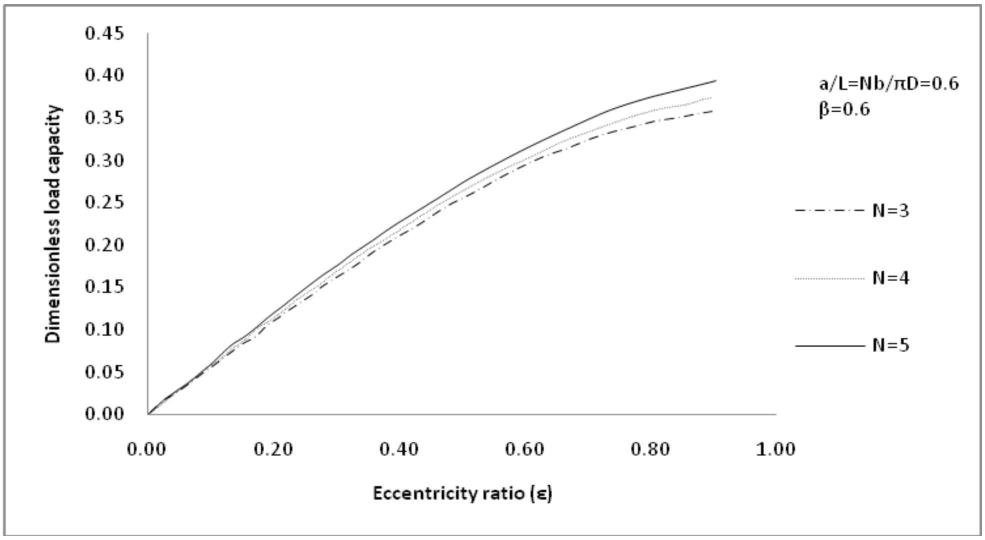

Fig.8 Load capacity of bearing vs eccentricity ratio (ε) for different number of recesses (N)

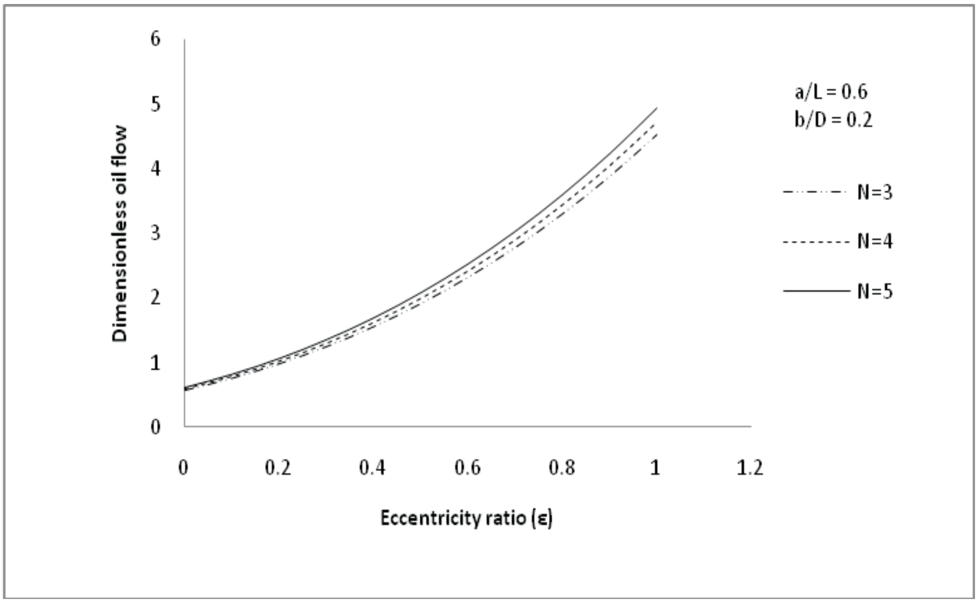

Fig.9 Dimensionless oil flow of bearing vs eccentricity ratio (ε) for different number of recesses (N)

5. Conclusion

Effect of various design parameters e.g. recess pressure to supply pressure ratio (β), dimensionless axial recess length (a/L) ratio, dimensionless circumferential recess length (Nb/πD) ratio, number of recess on the load bearing capacity and oil flow rate for four recess hybrid journal bearings is discussed. The results presented in this paper are very useful for designer for proper selection of dimension of the bearings.

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