Influence of Bearing Geometry on the Frictional Characteristic of Multirecess Hybrid Journal Bearing

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Abstract: *The fluid film bearing is a machine element used to support and guide the heavy turbo machinery and is used industry machine tool for required highly precious surface finishing. This study examines, the influence of offset factor on the performance of a membrane and capillary compensated two-lobe four recess hybrid journal-bearing system has been presented. To model the behavior of the oil flow in the journal bearing clearance gap, the generalized fluid flow governing the Reynolds equation presents the unknown pressure field of journal bearing clearance space, which is determined using Galerkin's method. The results have been presented for membrane restrictors of the two-lobe four recess hybrid journal bearing system that offers excellent operational parameter characteristics. The analytical results suggest that an appropriate selection of a type of compensated bearing for the influence lobe factor and the value of* δ *being got lower side has the better frictional characteristic as compared with the four pockets regular circular bearing. However, it can also consume less power due to the offset factor* $\delta < 1$

Keywords: Hybrid journal bearing, Offset factor (δ)***, Membrane, and Capillary Restrictor, Frictional losses.***

1. Introduction

The current emerging requirement of a fluid film journal bearing is used and the selection of bearing based on static and dynamic performance characteristics of bearing shell in high load carrying capacity of fluid film and fluid film stabilized in the higher load and temperature area. It is a focused research activity on the necessity of reduction of power losses and the fluid film is more stable in dynamic load conditions. The multi-lobe multi-pocket hybrid journal bearings shell or inner profile available studies present circular and noncircular bush, which is a different form of multi pockets deep and shallow type journal bearing, with a 2-lobe and more lobes journal bearing system.

Noncircular journal bearing is used in high-precision machines to achieve high profile accuracy and is also used in high-speed machine tools and turbo machines for use in power generation gas turbines. The two-lobe multirecess circular journal bearing presents excellent characteristics such as good dynamic stability about the bearing center position. For the case of hybrid journal-bearing studies restrictors are attached to the bearing shell, like capillary and membrane flow control devices. Non-dimensional parameter compliance the bearing profile as known offset factor (δ). It's defined as the modification in a circular journal bearing profile.

In the noncircular hybrid mode of operation and configuration with compensated flow control device attached the bearing bush and entry of lubricant in the pocket or recess is used as an oil reservoir. This pocket is limited to use in 2-lobe hybrid journal bearing due to the limit of depth like deep and shallow type pockets. the studied present the pressure development of fluid film lubrication in the bearing flow regime [1-2]

Several investigators [3-8] studied the friction that affected the characteristic parameters of hydrodynamic journal bearings. The dry and wet frictional losses affect the defined parameter of the recess journal bearing. Tareq et al. [3] found that non-Newtonian lubricant properties will change when the rise in the speed of the journal and temperature of lubricant is reinstated the oil properties becoming phase modified in Newtonian fluid type. Regarding the automotive industry, the power losses are projected to cost and reliability of the system. this research presented that phase of lubricant selection is supported to be reduced the frictional loss is 55% in the lubricated bearing system and the viscous frictional losses are reduced due to the relevant lubricant used. [4-5]

There are many studies available on the external factor influencing power loss of journal bearing systems, such as oil flow rate, bearing clearance, supply temperature, and so on. Gregory [6] reported an experimental investigation to study the influence factor to the reduction of friction loss in the thrust bearing, they found that oil supply flow rate and lubricant properties more affected the stability of rotor operated at high-temperature operation side. Fluid film stability is a major contribution to the reduction the friction losses and minimum required continuity of lubricant flow in the journal bearing system its adjustable to enhance the performance [7-8]

Several investigators are focused on the area of noncircular hybrid journal bearings. Cusano et al. [9] investigated the performance Influence of different variables of multi-recess hydrostatic journal bearing, it focused on the ratio pocket area I to the total effective land area of the bearing is more suitable is 0.6, and the bearing frictional performance improved for the reinforcing with the orifice and capillary restrictor and also consider changing the number of bearing recess. [10]. Phalle et al. [11-13] present the theoretical analysis of bearing subjected to quick start-ups and stops. Hence bearing inner surface is worn out. Identified the variables to influence the efficiency of the fluid bearing system that is types of compensating devices is control the supply pressure and flow pattern of lubricant in the clearance area, the fluid film thickness is directly affected the rotor dynamic stability.

Sing and Garner et al. studied the dynamic execution of quantitative dependent under the influencing the geometry profile of bearing shell, therefore oil film stiffness, and damping continue influence under oil viscosity, speed of rotor and supply oil flow directly affects the journal stability and bore shell profile is governed vibration characteristic of the rotor bearing arrangement i.e. the dynamic coefficient of a fluid film which influences the rotor system [14-16]. Dynamic coefficient varies under the influence of lubricant temperature i.e. film thickness of lubricant will be decreased due to temperature rise and rotor goes to the unstable side they rise power losses in the journal bearing system and viscous frictional losses increase [17].

The preceding available literature studied focused on the enhancing performance of the circular bearing and noncircular journal bearing due to the influence of direct variables is supply lubricant pressure, viscosity is the important role of the reduction the losses, it supports the stable film in the high load side and also effects the viscous resistance plays an important role for the reduction of power loss in the noncircular fluid film. This study presents to reduce power loss under the influence of the bearing profile and consider the best suitable compensating device required in the 2-lobe multi-pocket hybrid bearing system. Therefore, the current research is focused on bearing geometric factors varying from 0.8 to 1.2, projected performance variable characteristics such as lubricant film thickness discharge of lubricant, and analytical modeling presents the frictional power loss. The analytical study is expected to be very useful to bearing designers for selecting the best suitable industry machinery to fully fill the requirement and academic group.

2. Analysis

The mathematical representation of a 2-lobe four-pocket hybrid journal bearing arrangement is shown in fig. (1). The lubricant flow behavior in the journal bearing clearance space is represented by the governing generalized Reynolds equation. The lubricant flow follows the continuous or laminar stream and lubricant properties same in all directions of the same i.e. lubrication is incompressible is represented in the non-dimensional form [2]

$$\frac{\partial}{\partial \alpha}\left(\bar{h}^3 \bar{F}_2 \frac{\partial \bar{p}}{\partial \alpha}\right)+\frac{\partial}{\partial \beta}\left(\bar{h}^3 \bar{F}_2 \frac{\partial \bar{p}}{\partial \beta}\right)=\Omega \frac{\partial}{\partial \alpha}\left\{\left(1-\frac{\bar{F}_1}{\bar{F}_o}\right) \bar{h}\right\}+\frac{\partial \bar{h}}{\partial \bar{t}} \tag{1}$$

Where F_o, F_1, and F_2 are constant of cross film viscosity integral

$$\bar{F}_o = \int_0^1 \frac{1}{\bar{\mu}}\, d\bar{z}, \qquad \bar{F}_1 = \int_0^1 \frac{\bar{z}}{\bar{\mu}}\, d\bar{z}, \qquad F_2 = \int_0^1 \frac{\bar{z}}{\bar{\mu}}\left(\bar{z}-\frac{\bar{F}_1}{\bar{F}_o}\right) d\bar{z}.$$

2.1. Fluid film Thickness

It represents the analytical formulation of lubricant film thickness in the form of non-dimensional for the multi-pocket journal bearing as shown in Fig. 1 [11]

$$\bar{h}_0 = \frac{1}{\delta} - \left(\bar{X}_J - \bar{X}_L^i\right) cos\,\alpha - \left(\bar{Z}_J - \bar{Z}_L^i\right) \sin\alpha \quad (2)$$

where $\bar{X}_J$ and $\bar{Z}_J$ is presented equilibrium coordinates of the journal position about the center of bearing, lob center coordinate is $\bar{X}_L^i$ and $\bar{Z}_L^i$ for the i[th] lobe.

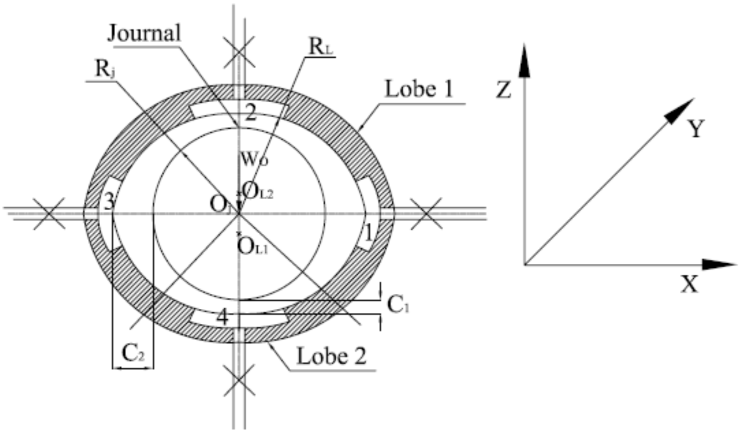


Figure 1. 2- lobe four pocket hybrid journal bearing system

2.2. Restrictor flow equation

For the continuity of flow maintained in the gap between journal and bearing through for the compensated device in multi-pocket hydrostatic/hybrid bearing system. The presented flow of rate of the equation in the lubricated analytical domain. Lubricant flow through the membrane and capillary restrictor is defined in the non-dimensional form equation (3) and (4).

Membrane restrictor [18]

$$\left.\begin{aligned}\bar{Q}_{Ri} &= \bar{C}_{s2}\left[1 + C_m\left(\bar{P}_{ci} - \bar{P}_{cj}\right)\right]^3 \left(1 - \bar{P}_{ci}\right)\\ \bar{Q}_{Rj} &= \bar{C}_{s2}\left[1 + C_m\left(\bar{P}_{ci} - \bar{P}_{cj}\right)\right]^3 \left(1 - \bar{P}_{cj}\right)\end{aligned}\right\} \quad (3)$$

Where, $\bar{Q}_{Ri}$ and $\bar{Q}_{Rj}$ flow of restrictor

Capillary restrictor [11]

$$\bar{Q}_R = \bar{C}_{S2}\left(1 - \bar{P}_C\right)^{1/2} \quad (4)$$

Where $\bar{C}_{S2}$ = Design factor of capillary restrictor

2.3. Finite element formulation

The lubricant flow domain is discretized in a finite number of elements and the shape quadrilateral type nodes are presented in the global coordinate system. the four pockets hydrostatic/hybrid journal bearing present the effective number of elements is observed that 148 elements are more accurate for the solution convergence. the mesh of bearing fluid geometry is shown in Fig. 2.

$$\bar{p} = \sum_{j=1}^{4} N_j \bar{P}_j \quad (5)$$

The discretized flow domain shows that the isoparametric element is subjected to the Lagrangian interpolation function is presents in the element is bilinearly distributed is shown in equation (6)

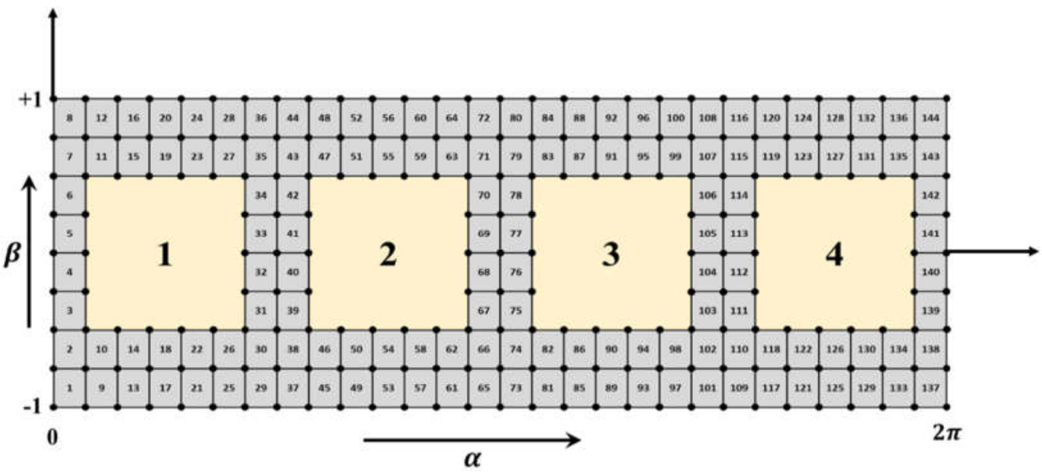


Figure 2. Mesh plot of bearing fluid working domain

$$N_i = \frac{1}{4}(1+\xi\xi_i)(1+\eta\eta_i) \text{ where } i = 1,2,3,4 \tag{6}$$

using the weight residual approximate Solution of $\bar{p}$, Eq. (1) can be expressed as

$$\frac{\partial}{\partial\alpha}\left(\bar{h}^3\bar{F}_2\frac{\partial\bar{p}}{\partial\alpha}\right)+\frac{\partial}{\partial\beta}\left(\bar{h}^3\bar{F}_2\frac{\partial\bar{p}}{\partial\beta}\right)-\Omega\frac{\partial}{\partial\alpha}\left\{\left(1-\frac{\bar{F}_1}{\bar{F}_o}\right)\bar{h}\right\}-\frac{\partial\bar{h}}{\partial\bar{t}}=R^e \tag{7}$$

Where R^e = residue of nondimensional lubricant flow field equation

$$\iint_{\Omega^e} N_i\left[\frac{\partial}{\partial\alpha}\left(\bar{h}^3\bar{F}_2\frac{\partial\bar{p}}{\partial\alpha}\right)+\frac{\partial}{\partial\beta}\left(\bar{h}^3\bar{F}_2\frac{\partial\bar{p}}{\partial\beta}\right)-\Omega\frac{\partial}{\partial\alpha}\left\{\left(1-\frac{\bar{F}_1}{\bar{F}_o}\right)\bar{h}\right\}-\frac{\partial\bar{h}}{\partial\bar{t}}\right]d\Omega^e=0 \tag{8}$$

Applying Galerkin's technique in flow field elemental equation (8) to reduce residue and the approximate solution presents the close agreement of the exact solution obtained in the form matrix

$$\left[\bar{F}\right]^e\{\bar{p}\}^e=\left\{\bar{Q}\right\}^e+\Omega\left\{\bar{R}_H\right\}^e+\bar{\dot{X}}_J\left\{\bar{R}_{X_J}\right\}^e+\bar{\dot{Z}}_J\left\{\bar{R}_{Z_J}\right\}^e \tag{9}$$

For the e^th element presented in the matrix and column vector form

$$\bar{F}_{ij}^e=\iint_{\Omega^e}\bar{h}^3\left(\bar{F}_2\frac{\partial N_i}{\partial\alpha}\frac{\partial N_j}{\partial\alpha}+\bar{F}_2\frac{\partial N_i}{\partial\beta}\frac{\partial N_j}{\partial\beta}\right)d\Omega^e \tag{10(a)}$$

$$\begin{aligned}\bar{Q}_i^e&=\int_{\Gamma^e}\left(\bar{F}_2\bar{h}^3\frac{\partial\bar{p}}{\partial\alpha}l_\alpha+\bar{F}_2\bar{h}^3\frac{\partial\bar{p}}{\partial\beta}l_\beta\right)N_i\,d\Gamma^e\\&\quad-\Omega\int_{\Gamma^e}\bar{h}\left(1-\frac{\bar{F}_1}{\bar{F}_o}\right)l_\alpha N_i\,d\Gamma^e\end{aligned} \tag{10(b)}$$

$$\bar{R}_H^e=\iint_{\Omega^e}\bar{h}\left(1-\frac{\bar{F}_1}{\bar{F}_o}\right)\frac{\partial N_i}{\partial\alpha}d\Omega^e \tag{10(c)}$$

$$\bar{R}_{xj}{}^e=\iint_{\Omega^e}N_i\cos\alpha\,d\Omega^e \tag{10(d)}$$

$$\bar{R}_{zj}{}^e=\iint_{\Omega^e}N_i\sin\alpha\,d\Omega^e \tag{10(e)}$$

where l_α and l_β are cosines angle direction vector, and i, j = 1, 2,..., n^e is presented local number of nodes per element. $\Omega^e$is e^{th} fluid element domain area and Γ^e is an e^{th} element of boundary. Expressed the resulting global matrix system in equation (10)

$$\left[\bar{F}\right]\{\bar{p}\}=\left\{\bar{Q}\right\}+\Omega\left\{\bar{R}_H\right\}+\bar{\dot{X}}_J\left\{\bar{R}_{X_J}\right\}+\bar{\dot{Z}}_J\left\{\bar{R}_{Z_J}\right\} \tag{11}$$

The fluid flow domain elements are expressed the discretized two-dimensional is written in an expanded form as

$$\begin{bmatrix} \bar{F}_{11} & \bar{F}_{12} & \cdots & \bar{F}_{1j} & \cdots & \bar{F}_{1n} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{i1} & \bar{F}_{i2} & \cdots & \bar{F}_{ij} & \cdots & \bar{F}_{in} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{j1} & \bar{F}_{j2} & \cdots & \bar{F}_{jj} & \cdots & \bar{F}_{jn} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{n1} & \bar{F}_{n2} & \cdots & \bar{F}_{nj} & \cdots & \bar{F}_{nn} \end{bmatrix} \begin{Bmatrix} \bar{P}_1 \\ \vdots \\ \bar{P}_i \\ \vdots \\ \bar{P}_j \\ \vdots \\ \bar{P}_n \end{Bmatrix} = \begin{Bmatrix} \bar{Q}_1 \\ \vdots \\ \bar{Q}_i \\ \vdots \\ \bar{Q}_j \\ \vdots \\ \bar{Q}_n \end{Bmatrix} + \Omega \begin{Bmatrix} \bar{R}_{H1} \\ \vdots \\ \bar{R}_{Hi} \\ \vdots \\ \bar{R}_{Hj} \\ \vdots \\ \bar{R}_{Hn} \end{Bmatrix} + \bar{\dot{X}}_J \begin{Bmatrix} \bar{R}_{X_J 1} \\ \vdots \\ \bar{R}_{X_J i} \\ \vdots \\ \bar{R}_{X_J j} \\ \vdots \\ \bar{R}_{X_J n} \end{Bmatrix} + \bar{\dot{Z}}_J \begin{Bmatrix} \bar{R}_{Z_J 1} \\ \vdots \\ \bar{R}_{Z_J i} \\ \vdots \\ \bar{R}_{Z_J j} \\ \vdots \\ \bar{R}_{Z_J n} \end{Bmatrix} \quad (12)$$

Where $[\bar{F}]$ is present the value of matrix and $\{\bar{Q}\}$, $\{\bar{R}_H\}$, $\{\bar{R}_{X_J}\}$ and $\{\bar{R}_{Z_J}\}$ is fluid elements of column vector are calculated using Eqs (10) (a) - (10) (e), respectively

The flowing film pressure are general function of journal displacement and velocity, total pressure generated in j^{th} point of the lubricated area is expressed as a function of

$$\bar{P}_j = \bar{P}_j\left(\bar{X}_j, \bar{Z}_j, \bar{\dot{X}}, \bar{\dot{Z}}\right) \quad (13)$$

For the steady-state condition of the journal position considered in the present problem, the flooded film pressure demonstrated is

$$\bar{p} = \bar{p}(\bar{X}_j, \bar{Z}_j) = \bar{p}_o \quad (14)$$

where $\bar{p}_o$ = steady-state nodal pressure

In the steady-state condition, Eq. (13) is demonstrated as $\bar{p}_j = \bar{p}_{oj}$. The element nodal pressure column vector is set out as $\{\bar{p}\} = \{\bar{p}_o\}$

The lubricant flow through membrane restrictor is expressed in Eq. (4) is given for the rotor steady state position is follows

$$\bar{Q}_{Ri/j} = \bar{C}_{s2}\left[1 + C_m\left(\bar{P}_{ci} - \bar{P}_{cj}\right)\right]^3\left(1 - \bar{P}_{ci/j}\right) \quad (15)$$

The restrictor required for the maintaining continuous flow in bearing clearance. Let j^{th} node is lying on the entry of lubricant, then the $\bar{Q}_j$ on the right side of Eq. (12) is becomes equal to $\bar{Q}_R$. i.e. $\bar{Q}_j = \bar{Q}_R$

$$\begin{bmatrix} \bar{F}_{11} & \bar{F}_{12} & \cdots & \bar{F}_{1j} & \cdots & \bar{F}_{1n} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{i1} & \bar{F}_{i2} & \cdots & \bar{F}_{ij} & \cdots & \bar{F}_{in} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{j1} & \bar{F}_{j2} & \cdots & \bar{F}_{jj} & \cdots & \bar{F}_{jn} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{n1} & \bar{F}_{n2} & \cdots & \bar{F}_{nj} & \cdots & \bar{F}_{nn} \end{bmatrix} \begin{Bmatrix} \bar{P}_{o1} \\ \vdots \\ \bar{P}_{oi} \\ \vdots \\ \bar{P}_{oj} \\ \vdots \\ \bar{P}_{on} \end{Bmatrix} = \begin{Bmatrix} \bar{Q}_1 \\ \vdots \\ \bar{Q}_i \\ \vdots \\ \bar{C}_{s2}[1 + C_m(\bar{P}_{ci} - \bar{P}_{cj})]^3\left(1 - \bar{P}_{ci/j}\right) \\ \vdots \\ \bar{Q}_n \end{Bmatrix} + \Omega \begin{Bmatrix} \bar{R}_{H1} \\ \vdots \\ \bar{R}_{Hi} \\ \vdots \\ \bar{R}_{Hj} \\ \vdots \\ \bar{R}_{Hn} \end{Bmatrix} \quad (16)$$

The solution domain equation (15) becomes nonlinear for the fluid film bearing compensated with membrane and capillary restrictor. This nonlinear fluid domain equation solves with the help of Newton Raphson method as given below

2.4. Newton–Raphson Method

fluid film rotor bearing system Eq. (16) for the bearing compensated with membrane and capillary restrictor is written in general form

$$\left.\begin{aligned}
\bar{F}_1 &= \bar{F}_{11}\,\bar{P}_{o1} + \cdots + \bar{F}_{1i}\,\bar{P}_{oi} + \cdots + \bar{F}_{1j}\,\bar{P}_{oj} + \cdots + \bar{F}_{1n}\,\bar{P}_{on} - \bar{Q}_1 - \Omega\bar{R}_{H1} = 0\\
&\vdots\\
\bar{F}_i &= \bar{F}_{i1}\,\bar{P}_{o1} + \cdots + \bar{F}_{ii}\,\bar{P}_{oi} + \cdots + \bar{F}_{ij}\,\bar{P}_{oj} + \cdots + \bar{F}_{in}\,\bar{P}_{on} - \bar{Q}_i - \Omega\bar{R}_{Hi} = 0\\
&\vdots\\
\bar{F}_j &= \bar{F}_{j1}\,\bar{P}_{o1} + \cdots + \bar{F}_{ji}\,\bar{P}_{oi} + \cdots + \left(\bar{F}_{jj}\,\bar{P}_{oj} - \bar{Q}_R\right) + \cdots + \bar{F}_{jn}\,\bar{P}_{on} - \Omega\bar{R}_{Hj} = 0\\
&\vdots\\
\bar{F}_n &= \bar{F}_{n1}\,\bar{P}_{o1} + \cdots + \bar{F}_{ni}\,\bar{P}_{oi} + \cdots + \bar{F}_{nj}\,\bar{P}_{oj} + \cdots + \bar{F}_{nn}\,\bar{P}_{on} - \bar{Q}_n - \Omega\bar{R}_{Hn} = 0
\end{aligned}\right\} \quad (17)$$

Its noticed that the fluid element of vector $\bar{Q}$ occurs at only the nodes situated on rectangle recess, for rest of the fluid elements nodes term $\bar{Q}$ is become equal zero. Therefore, the expression a fluid element node on the pockets and jj^{th} term in Eq. (18)

$$\begin{aligned}
\Delta\bar{F}_j &= \bar{F}_{jj}\,\bar{P}_{oj} - \bar{Q}_R\\
\left.\frac{\partial\Delta\bar{F}_j}{\partial\bar{P}_{oj}}\right|_0 &= \bar{F}_{jj} - \left.\frac{\partial\bar{Q}_R}{\partial\bar{P}_{oj}}\right|_o
\end{aligned} \quad (18)$$

If $\bar{P}_{oj}\big|_0$ $(j = 1,2\cdots\cdots n)$ is the prime estimated for the fluid element nodal pressure, $\bar{P}_{oj}$ is rotor is set to stable of the system. then the Eq. (17) is becoming of linearized using Taylor's series approximation of the function $\bar{F}_j(j = 1,2\cdots\cdots n)$

$$\left.\begin{aligned}
\bar{F}_1|_0 + \left.\frac{\partial\bar{F}_1}{\partial\bar{P}_{o1}}\right|_0 \Delta\bar{P}_{o1} + \cdots + \left.\frac{\partial\bar{F}_1}{\partial\bar{P}_{oi}}\right|_0 \Delta\bar{P}_{oi} + \cdots + \left.\frac{\partial\bar{F}_1}{\partial\bar{P}_{oj}}\right|_0 \Delta\bar{P}_{oj} + \cdots + \left.\frac{\partial\bar{F}_1}{\partial\bar{P}_{on}}\right|_0 \Delta\bar{P}_{on} &= 0\\
\vdots\\
\bar{F}_i|_0 + \left.\frac{\partial\bar{F}_i}{\partial\bar{P}_{o1}}\right|_0 \Delta\bar{P}_{o1} + \cdots + \left.\frac{\partial\bar{F}_i}{\partial\bar{P}_{oi}}\right|_0 \Delta\bar{P}_{oi} + \cdots + \left.\frac{\partial\bar{F}_i}{\partial\bar{P}_{oj}}\right|_0 \Delta\bar{P}_{oj} + \cdots + \left.\frac{\partial\bar{F}_i}{\partial\bar{P}_{on}}\right|_0 \Delta\bar{P}_{on} &= 0\\
\vdots\\
\bar{F}_j\big|_0 + \left.\frac{\partial\bar{F}_i}{\partial\bar{P}_{o1}}\right|_0 \Delta\bar{P}_{o1} + \cdots + \left.\frac{\partial\bar{F}_j}{\partial\bar{P}_{oi}}\right|_0 \Delta\bar{P}_{oi} + \cdots + \left.\frac{\partial\bar{F}_j}{\partial\bar{P}_{oj}}\right|_0 \Delta\bar{P}_{oj} + \cdots + \left.\frac{\partial\bar{F}_j}{\partial\bar{P}_{on}}\right|_0 \Delta\bar{P}_{on} &= 0\\
\vdots\\
\bar{F}_n|_0 + \left.\frac{\partial\bar{F}_n}{\partial\bar{P}_{o1}}\right|_0 \Delta\bar{P}_{o1} + \cdots + \left.\frac{\partial\bar{F}_n}{\partial\bar{P}_{oi}}\right|_0 \Delta\bar{P}_{oi} + \cdots + \left.\frac{\partial\bar{F}_n}{\partial\bar{P}_{oj}}\right|_0 \Delta\bar{P}_{oj} + \cdots + \left.\frac{\partial\bar{F}_n}{\partial\bar{P}_{on}}\right|_0 \Delta\bar{P}_{on} &= 0
\end{aligned}\right\} \quad (19)$$

Eq. (19) is presented in matrix form as

Table 1. Bearing operating and geometric parameters

Sr. No.	Geometric Parameters	
1.	Bearing aspect ratio (λ)	1.0
2.	Land width ratio($\bar{a}_b$)	0.14
3.	Clearance ratio,c/R_J	0.001
4.	Restrictor design parameter ($\bar{C}_{S2}$)	0.935
5.	Type of compensating element	Membrane, Capillary Restrictor
6.	Membrane compliance ($\bar{C}_m$)	0.25, 0.35
7.	Offset factor(δ) for Hybrid mode of operation	0.80, 1.00, 1.20
8.	Speed parameter(Ω)	0.5
9.	Number of pockets in each bearing	4
10.	Pocket shape	Rectangular

$$\begin{bmatrix} \bar{F}_{11} & \bar{F}_{12} & \cdots & \bar{F}_{1j} & \cdots & \bar{F}_{1n} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{i1} & \bar{F}_{i2} & \cdots & \bar{F}_{ij} & \cdots & \bar{F}_{in} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{j1} & \bar{F}_{j2} & \cdots & (\bar{F}_{jj} + D_1) & \cdots & \bar{F}_{jn} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{n1} & \bar{F}_{n2} & \cdots & \bar{F}_{nj} & \cdots & \bar{F}_{nn} \end{bmatrix} \begin{Bmatrix} \bar{P}_{o1} \\ \vdots \\ \bar{P}_{oi} \\ \vdots \\ \bar{P}_{oj} \\ \vdots \\ \bar{P}_{on} \end{Bmatrix} = \begin{Bmatrix} \bar{Q}_1 \\ \vdots \\ \bar{Q}_i \\ \vdots \\ \bar{Q}_{R0} \\ \vdots \\ \bar{Q}_n \end{Bmatrix} + \Omega \begin{Bmatrix} \bar{R}_{H1} \\ \vdots \\ \bar{R}_{Hi} \\ \vdots \\ \bar{R}_{Hj} \\ \vdots \\ \bar{R}_{Hn} \end{Bmatrix} - \begin{bmatrix} \bar{F}_{11} & \bar{F}_{12} & \cdots & \bar{F}_{1j} & \cdots & \bar{F}_{1n} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{i1} & \bar{F}_{i2} & \cdots & \bar{F}_{ij} & \cdots & \bar{F}_{in} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{j1} & \bar{F}_{j2} & \cdots & \bar{F}_{jj} & \cdots & \bar{F}_{jn} \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{F}_{n1} & \bar{F}_{n2} & \cdots & \bar{F}_{nj} & \cdots & \bar{F}_{nn} \end{bmatrix} \begin{Bmatrix} \bar{P}_{o1} \\ \vdots \\ \bar{P}_{oi} \\ \vdots \\ \bar{P}_{oj} \\ \vdots \\ \bar{P}_{on} \end{Bmatrix} \quad (20)$$

The expression for D_1 is obtained by differentiating Eq. (15) with respect to $\bar{P}_{oi/j}$

$$D_1 = -{}^{\partial \bar{Q}_R}\!\big/_{\partial \bar{P}_{oi/j}} \quad (21)$$

2.5. Load Carrying Capacity

The fluid film reaction due to external load on the rotor of hydrostatic / hybrid journal bearing. oil film reaction components are direction of X and Z respectively []

$$\bar{F}_X = -\int_{-\lambda}^{\lambda}\int_{0}^{2\pi} \bar{p}\ \cos\alpha\, d\alpha\, d\beta \quad (22)(a)$$

and

$$\bar{F}_Z = -\int_{-\lambda}^{\lambda}\int_{0}^{2\pi} \bar{p}\ \sin\alpha\, d\alpha\, d\beta \quad (22)(b)$$

Then, the respective force component resultant reaction is expresssed as

$$\bar{F}\ = [\bar{F}_X + \bar{F}_Z]^{1/2} \quad (22)(c)$$

2.6. Boundary Condition

The oil flow field analysed in the bearing fluid domain boundary limitation is given [19]

1. Relative pressure is equal to zero for noncircular bearing having node situated on the open environment pressure $\bar{p}|_0 = \pm 1.0 = 0.0$.
2. Fluid passing through the restrictor hole or pocket is equivalent to nodal pressure.
3. The nodal flow is nonzero, its situated on a hole or recess having same pressure.
4. Lubricant flow area having trailing edge in positive region is subjected to governing flow equation limitation i.e.

$$\bar{p} = \frac{\partial \bar{p}}{\partial \beta} = 0.0$$

3. Performance Characteristics

The 2-lobe journal bearing performance parameters in the terms of lubricant minimum film thickness ($\bar{h}_{min}$), lubricant flow requirement ($\bar{Q}$), frictional torque ($\bar{T}_{fric}$) have been computed. In the lubricant flow in the bearing is generate heat due to internal viscous resistance of fluid layer. The friction torque acting on the journal surface can be find by integration of viscous resistance around the journal surface.

$$\bar{T}_{fric} = \sum_{e=1}^{n} \int_{-1}^{1} \int_{0}^{2\pi} \left(\frac{\bar{h}}{2} \frac{\partial p}{\partial \alpha} + \frac{\Omega}{\zeta * \bar{h}} \right) d\alpha \, d\beta \tag{23}$$

where

$$\zeta = 1 - \frac{2N}{l_m \bar{h}} \tanh(\frac{Nhl_m}{2})$$

4. Solution Procedure

The analysis of the membrane and capillary restrictor with multi pocket 2-lobe journal bearing system used in laminar regime requires an iterative solution for lubricant flow field Eq. (16), lubricant viscosity is constant and the rotor is stable position ($\bar{\dot{X}}_j, \bar{\dot{Z}}_j = 0$). The fluid element node values of fluid film pressures are computed for a particular value of journal center coordinates $\bar{X}_J, \bar{Z}_J$ and the iterative bend procedure is continued until the journal center establish the static equilibrium of rotor center position defined by

$$\bar{F}_X = 0 \; and \; \bar{F}_Z - \bar{W}_o = 0 \tag{24}$$

In journal bearing system growing small amplitude motion i.e. dynamic displacements are much smaller than the bearing clearance. The small amplitude motion about an equilibrium position expressing the fluid film reaction forces $\bar{F}_X$ and $\bar{F}_Z$ in Eq. (25) as Taylor series expansion around the static journal position and the increments $\left(\Delta\bar{X}_j^i, \Delta\bar{Z}_j^i\right)$ on the rotor coordinates are obtained as given below [18]

$$\Delta\bar{X}_j^i = -\frac{1}{D_j} \left[\left.\frac{\partial \bar{F}_Z}{\partial Z_j}\right|_i - \left.\frac{\partial \bar{F}_X}{\partial Z_j}\right|_i \right] \begin{Bmatrix} \bar{F}_X^i \\ \bar{F}_Z^i - \bar{W}_o \end{Bmatrix} \tag{25(a)}$$

$$\Delta\bar{Z}_j^i = -\frac{1}{D_j} \left[\left.\frac{\partial \bar{F}_Z}{\partial X_j}\right|_i - \left.\frac{\partial \bar{F}_X}{\partial X_j}\right|_i \right] \begin{Bmatrix} \bar{F}_X^i \\ \bar{F}_Z^i - \bar{W}_o \end{Bmatrix} \tag{25(b)}$$

The new rotor center position co-ordinates $\left(\bar{X}_j^{i+1}, \bar{Z}_j^{i+1}\right)$ are given as

$$\left.\begin{aligned} \bar{X}_j^{i+1} &= \bar{X}_j^i + \Delta\bar{X}_j^i \\ \bar{Z}_j^{i+1} &= \bar{Z}_j^i + \Delta\bar{Z}_j^i \end{aligned}\right\} \tag{26}$$

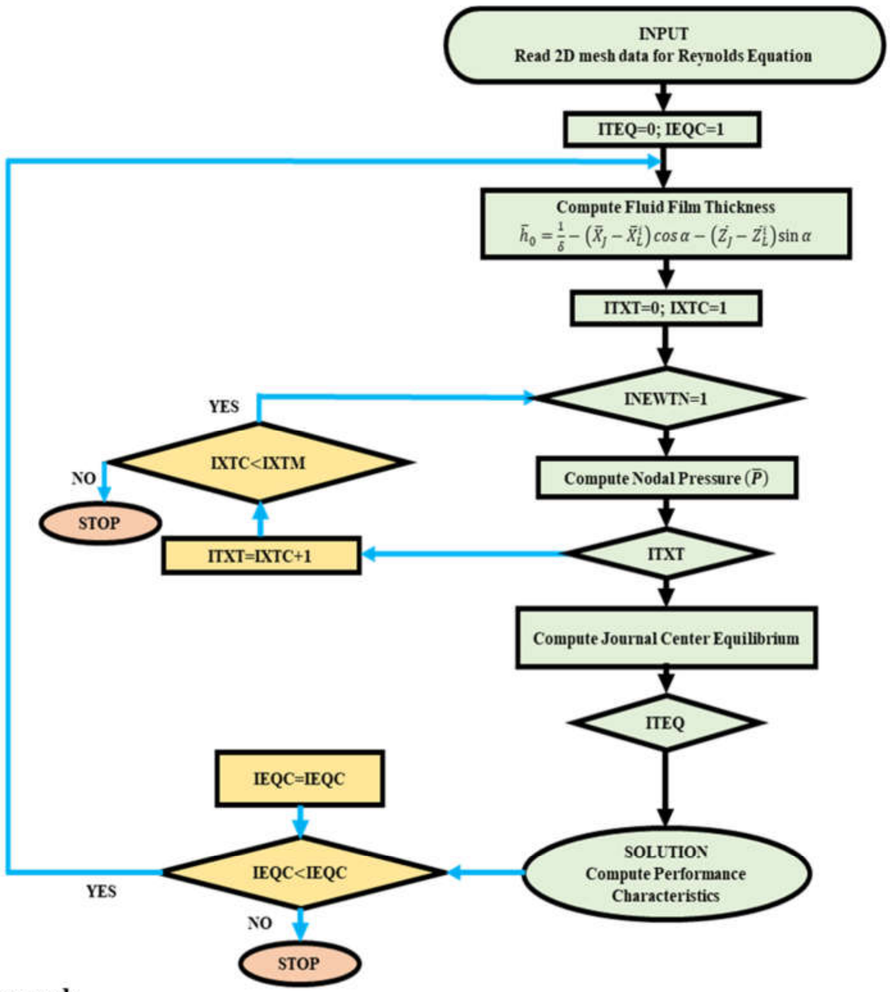
where are $\left(\bar{X}_j^i, \bar{Z}_j^i\right)$the coordinates of the i^{th} bearing rotor center position. Iterations are continued until the following convergence criterion is not satisfied:

$$\left[\frac{\left(\left(\overline{\Delta X}_j^i\right)^2 + \left(\overline{\Delta Z}_j^i\right)^2 \right)^{1/2}}{\left(\left(\bar{X}_j^i\right)^2 + \left(\bar{Z}_j^i\right)^2 \right)^{1/2}} \right] \times 100 \; < 10^{-3} \tag{27}$$

5. Results and Discussion

The analytical results of multi pocket hybrid journal bearing system have been computational simulation using the Finite element analysis (FEM) as describe in earlier section and the developed solution algorithm of FEM model for bearing rotor stable condition $\left(\bar{\dot{X}}_j, \bar{\dot{Z}}_j = 0\right)$ i.e. journal rotate about set positions in the bearing shell, as there are result available of in the previous published study examine the influence of offset factor (δ) for the case of four pocket 2-lobe hybrid journal bearing system compare study of membrane and capillary restrictor is show in Fig. 1.

The result have been computed for various valued of external applied load $\overline{W}_o$.As there are no experimental as well theoretical results available in the published literature considering the influence of power losses for various offset factors of 2-lobe journal bearing, on the performance of bearings for the case of a four pocket hybrid journal bearing system attached with capillary and membrane restrictors compare performance of characteristics parameters of hydrostatic/hybrid fluid film bearing. Therefore, in order to validate the selected methodology, the numerically model stimulated results from the present study, and using an optimum working domain grid, for the finite bearing length, data were generated for the two lobe axial groove hydrodynamic journal bearing. The dynamic coefficient is compute corresponding to an Speed and offset factor (δ) respectively.



Legend:

IEQC-Iteration counter for journal center equilibrium

IXTC-Iteration counter for extent of fluid film

IEQM-Maximum number of iteration for journal center equilibrium

IXTM-Maximum number of iteration for extent of fluid film

IR- Code for type of restrictor

Figure 3 Flow Chart of iterative solution Procedure

The evaluate results have been compared with the published results of Lund et al. [17], as shown in table 2. The results bearing characteristics parameters show quite a good agreement. and their percentage change of two lobe four recess hybrid journal bearing with the respect of non-circular bearing due to offset factor has been present in Fig 4 to 6, the geometric parameter has been used in present study as shown in table 1.The characteristics parameters has been enumerating for the different values of lobe factors (δ).

Table 2. Comparisons of Bearing Characteristics Parameters of a 2- Lobe Hydrodynamic Journal Bearing

L/D = 1, Ω = 1, δ = 0.5, $\overline{W}_o$ = 5.0		
Bearing Characteristics Parameters	Present Work	Ref. [17] Lund et al.
ε	0.3426	0.340
ϕ	80.997	82.100
$\bar{S}_{xx}/\bar{W}_0$	1.203	1.110
$\bar{S}_{xz}/\bar{W}_0$	1.812	1.700
$\bar{S}_{zx}/\bar{W}_0$	-3.5004	-3.610
$\bar{S}_{zz}/\bar{W}_0$	5.1544	5.130
$\bar{C}_{xx}/\bar{W}_0$	2.801	2.620
$-(\bar{C}_{xz}/\bar{W}_0) = (\bar{C}_{zx}/\bar{W}_0)$	0.2568	0.238
$\bar{C}_{zz}/\bar{W}_0$	9.3348	9.430

5.1. Influence of minimum film thickness $(\bar{h}_{min})$ versus external load $(\bar{W}_o)$

Fig. (4) present the changes of lubricant fluid film thickness under the influence of compensated device for various lobe factor of bearing. its noticed that in Fig. (4), a capillary compensated bearing shows a reduction in the value of $\bar{h}_{min}$, as compare with membrane compensated bearing that is quantify the value of $\bar{h}_{min}$ with respect of offset factor of bearing for various load condition that is capillary restrictor is higher film thickness for δ=0.8 as compare the offset factor 1.0 and 1.2 of $\bar{h}_{min}$. the effect of lobe factor δ=0.8 on the value of film thickness significantly higher as compared to bearing having the offset factor δ=1.2. for the increases membrane compliance $(\bar{C}_m)$ in the membrane compensated bearing flow regain this loss and oil film thickness$(\bar{h}_{min})$ value is increase 6.63%. it may be shows that rotor is more dynamically stable about an equilibrium position that is presented characteristics of bearing performance are more suitable for reduce the power losses in the hybrid journal bearing system in the value of lobe factor δ=0.8 and denoted the relative performance of 2-lobe four pocket hybrid journal bearing lubricant film thickness its seen that bearing compensated with membrane restrictor shows better fulfilment than that of capillary compensated bearing shell.

5.2. Influence of lubricant flow $(\bar{Q})$ versus external load $(\bar{W}_o)$

The lubricant flow fulfilment of a 2-lobe four recess hybrid journal bearing decrease as the value of lobe factor (δ) increases. For the instance of hybrid bearing, the fluid film supported the applied external load $(\bar{W}_o)$ its varies the rotor position the about the bearing center i.e. change the lubricant flow $(\bar{Q})$ in the bearing clearance space its change the rotor loaded condition and bearing is compensated with membrane and capillary restrictor. For the using of capillary restrictor flow of lubricant needed for the varies bearing lobe factor. The percentage rise in the value of $\bar{Q}$ due to influence of offset factor δ = 0.8 in hybrid bearing it found to be of the order of 10.87%, 11.43% and 12.79% for capillary and membrane $(\bar{C}_m = 0.25, 0.35)$ compensated bearings is shows in Fig. (5), for the compare with a base circular bearing. However, it may observed that the offset factor (δ = 0.8) significantly effect of lubricant flow in the 2-lobe hybrid journal bearing with capillary restrictor as compared to four pocket circular bearing for the value of the offset factor (δ = 1.0). Therefore, the capillary restrictor is more suitable for the requirement of stabilised flow in bearing system, i.e. flow maintaining continuous in the flow domain of hybrid bearing system, loss of lubricant flow shown that higher viscous power loss or may be bearing system is less stable as compare the case of more lubricant flow.

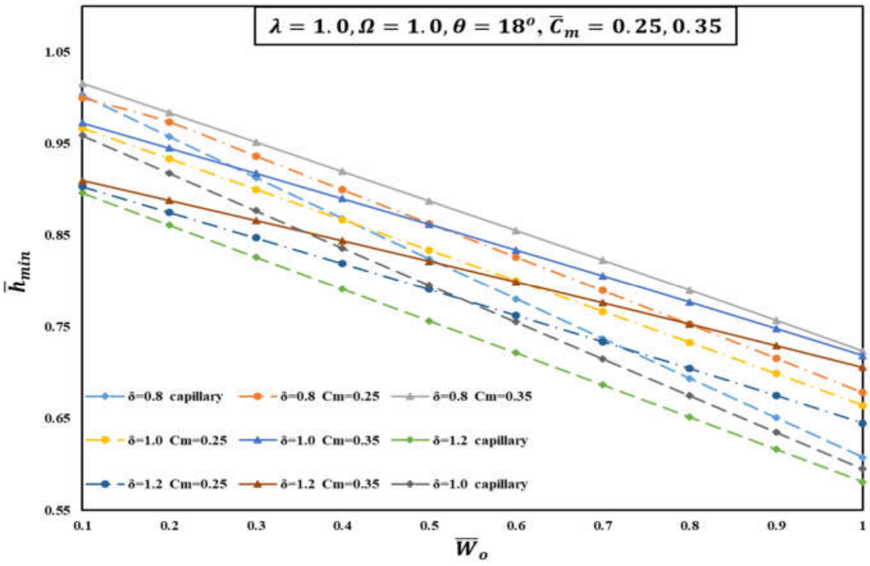


Figure 4. variation of $\bar{h}_{min}$ with $\overline{W}_o$

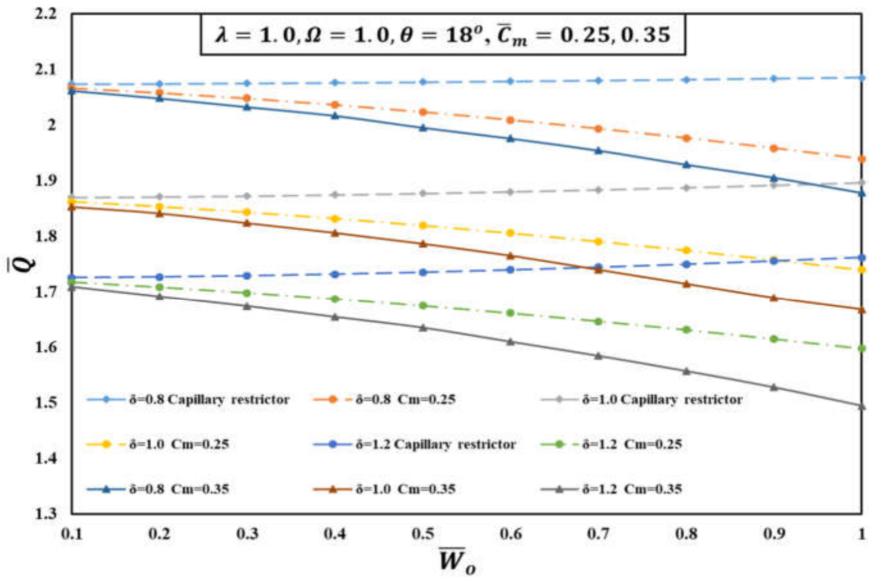


Figure 5. variation of $\bar{Q}$ with $\overline{W}_o$

5.3. Influence of frictional torque $(\overline{T}_{fric})$ versus external load $(\overline{W}_o)$

Fig. (6) shows the variation frictional torque $(\overline{T}_{fric})$ verses external load $(\overline{W}_o)$ for different bearing lobe factor ($\delta = 0.8, 1.0$ *and* 1.2) of bearing with compensated capillary and membrane restrictor is varies the compliance parameter ($\bar{C}_m = 0.25, 0.35$). the value of $(\overline{T}_{fric})$ gets enhanced with the increase the lobe factor of bearing studies. Its notably that frictional torque is depends on fluid film thickness $(\bar{h}_{min})$ and lubricant flow rate $\bar{Q}$. In general, the value of minimum film thickness and lubricant flow increased to get decreased the offset factor of bearing. The percentage decrease in the value of $\overline{T}_{fric}$ due to change of offset factor is δ = 0.8 in hybrid bearing is found to be higher side of 6.95% for membrane compensated ($\bar{C}_m = 0.35$) bearing is compare with a base circular bearing. therefore, the frictional torque is rises 5.33% for the value of δ = 1.2, In a bearing lobe factor is δ = 0.8 and compensated with membrane restrictor is flow compliance ($\bar{C}_m =$

0.35) is more suitable for the must have reduction power losses in the 2-lobe hybrid bearing system.

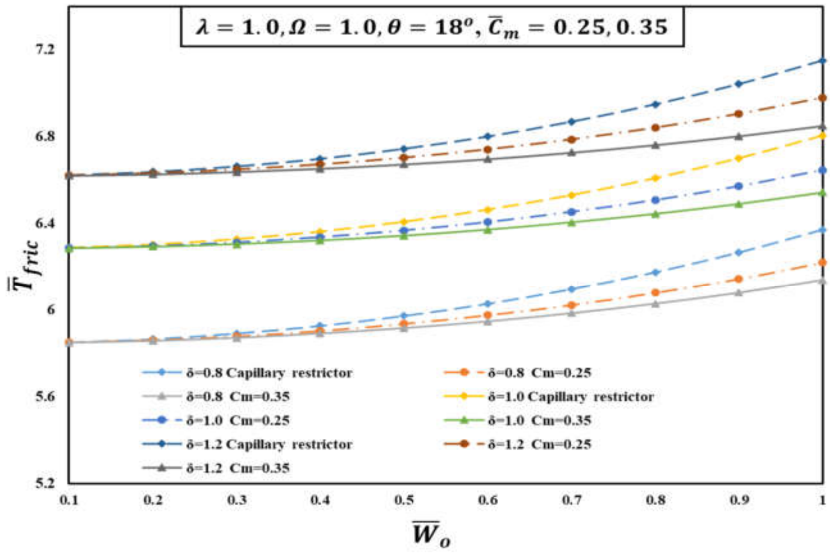


Fig. (6) variation of $\bar{T}_{fri}$ with $\bar{W}_o$

6. Conclusion

The analytical study for the characteristics parameters of 2-lobe hybrid journal bearing shell set up with capillary and membrane restrictor, considering effect of bearing lobe factor (δ) is presented for the variation of external load. Thus, the following conclusions are drawn.

- In general, the conducting of a 2- lobe hybrid journal bearing changes offsetting of the bearing shell parameter for the capillary and membrane compensating device. It is pointed that the 2-lobe four-pocket journal bearing with offset factor δ =0.8 compensated with membrane restrictor may be more structured for the reduction power losses and reliable for using operational cost.
- For the 2-lobe bearing shell irregular profile i.e. δ = 0.8, 1.2 is compare regular shell shape the value of lubricant film thickness reduce and decreases the lubricant flow $(\bar{Q})$ as the value of $\bar{W}_o$ and or δ increase, by selecting appropriate compensating device . It may be noticed that general designer can make use of following
  $\bar{h}_{min|Membrane\ (\bar{C}_m = 0.35)} > \bar{h}_{min|Membrane\ (\bar{C}_m = 0.25)} > \bar{h}_{min|Capillary}$
  $\bar{Q}|_{Capillary} > \bar{Q}|_{Membrane\ (\bar{C}_m = 0.25)} > \bar{Q}|_{Membrane\ (\bar{C}_m = 0.35)}$
- The value of frictional torque $(\bar{T}_{fri})$ may be climb by 5.33% for offset factor is δ = 1.2, however power loss reduction in bearing shell set up with membrane restrictor is 6.95% for δ = 0.8 as compared the regular shape of bearing. the use of 2-lobe hybrid journal bearing improved the value of frictional torque. The following trend is noticed in the value of $\bar{T}_{fri}$
  $\bar{T}_{fri|\text{Capillary}} > \bar{T}_{fri|\text{Membrane}\ (\bar{C}_m = 0.25)} > \bar{T}_{fri|\text{Membrane}\ (\bar{C}_m = 0.35)}$

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