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Design and analysis of new compound journal bearing

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Abstract
Journal bearing is one of the most important mechanical parts which can vastly be found in any application of mechanical camcorders such as engine, roterdynamic, aerodynamic and many more. One of the main performance related problems that this mechanical part may encounter during its service life is due to friction and wear over a short period of time, especially at high speeds, a one of the main challenges was to keep the shaft running efficiently for another hours. Many designs and modifications are attempted by researchers to improve certain specifications of the journal. In this construction a new design of the magazine bearing system was developed. It consists of three floors; the second is the bearing (internal and external) and the third is the thrust bearing. All three bearing units are combined into a single case as a magazine assembly. In this configuration, two hydrodynamic pressure profiles can be generated around the inner and outer fluid films in the circumferential direction and one in the axial direction. To predict the journal parameters, a mathematical model for the new magazine bearing design was developed using Reynolds' hydrodynamic equations. Lubrication in two dimensions. The governing partial differential equation was solved using Finite Difference method via MATLAB software. The main bearing parameters such as pressure, load capacity, friction force and so on are evaluated and their effect on journal geometry and operating conditions are investigated. A sample journal model was fabricated using CNC machine and tested experimentally with specially made testing rig, Vibration characteristics and stability of the journal are analyzed and studied also. The stress analysis of the journal housing are performed by using Finite Element method and excited by ANSYS 14. The effect of using Nano additive to the lubricant oil is studied both theoretically and experimentally. The preliminary results of this work indicated that the new suggested journal can significantly increase the ability of withstanding higher pressure and carrying load by about 23.60% and 12.90% respectively more than the traditional journal. Furthermore it exhibits less friction and higher critical operating speed. Using oil with Nano-particles can reduce friction by 16% with higher temperature operation. The optimum design conditions are occurred at 0.92 eccentricity ratio and 0.5 compound size ratio.

Keywords: Hydrodynamic journal bearing; Nano lubricants; Viscosity; Pressure distribution; Load carrying capacity.

1. Introduction
Traditional bearings consist of two cylinders that rotate with each other. As shown in Figure 1 [1], the outer cylinder (bearing) is stationary and the inner cylinder (shaft) rotates at an angular velocity of called the logarithm. The main purpose of plain bearings is to support a rotating machine by providing sufficient

lubrication for the s separate moving parts and minimizing friction and wear of the due to rotation [2]. Due to the rotation of the magazine, the high pressure liquid film in the gap between the magazine and the bearing provides the lubrication and bearing capacity of the hydrodynamic film. The displacement of the center of the shaft with respect to the center of the bearing is called eccentricity [3]. Almost all high performance industrial turbines use some sort of liquid diaphragm bearing to support the weight of the shaft and control the movement caused by the unbalanced force. The two main advantages of liquid film bearings over rolling bearings are excellent energy absorption to reduce vibration and durability due to the absence of rolling contact stress [4]. Damping is important in many types of rotating machines. Here, fluid diaphragm bearings are often the main source of energy needed to control vibration. Liquid membrane plain bearings also have a significant impact on the dynamic stability of the rotor and should be carefully selected and used as an important step in system development. Excellent rotor bearings [5]. Plain bearings have a surprisingly long service life if the lubricant is clean. Therefore, dynamic analysis of fluid bearings is important due to the imbalance force of the machine, the aerodynamic force, and the force applied to the shaft by external excitation from the seal and coupling [6, 7].

2. Literature review

D. M. Nuruzzaman et al. [1] studied and compared pressure distribution and load carrying capacity using analytical method and finite element method. The effects of variations in operating variables such as eccentricity ratio and shaft speed on the load capacity of the bearing were calculated using isothermal conditions. Results reveal good agreement between analytical method and finite element method to evaluate pressure distribution and load carrying capacity.

H. Hirani et al. [2] described simple and rapid method to evaluate important design parameters such as maximum pressure, load capacity, maximum temperature, and flow and power loss. Proposed analytical method agrees with finite element method result for a wide range of design parameters. The maximum temperature calculated by present method compared with THD analysis and experimental results, which matches quite well. Values of other parameters also lie within range of different numerical methods, so from this point of view, proposed design method is efficient and simple.

V. K. Dwivedi et al. [3] investigated the effect on pressure distribution of hydrodynamic journal bearing by eccentricity ratio of short bearing approximation. Reynolds equation is solved by MATLAB which is compared with literature available and found to be in quite agreement. The results are summarized as; high value of eccentricity increases the maximum pressure as well as the negative pressure and increases the steepness of the curve.

Ketan Tamboli et al. [4] experimentally investigated water-lubricated hydrodynamic bearing used in nuclear power plants wherein water is used as process fluid. Bearings are designed considering different techniques like ESDU chart, Reason and Narang technique, Cameron method and Raymondi & Boyd method. After calculating all parameters by above method, comparison of minimum thickness, clearance and power loss, etc. has been made. Result reveals that (a) Design procedure of water-lubricated bearing follows same process as oil-lubricated bearing. (b) Material should be non-corrosive, adequate tolerances should be provided and should be properly surface finished. (c) With pressure feed condition load carrying capacity is increased (d) temperature rise and power losses are under control (e) by properly feeding, side leakage can be minimized.

Chasalevris et al. [5] evaluated finite journal bearing characteristics using exact analytical solution of the Reynolds equation and expression for pressure distribution. The evaluation of bearing parameters and characteristics made for all four different L/d ratios of 0.25, 0.5, 1 and 4 with very good agreement. The position of minimum film thickness and maximum pressure evaluated analytically and numerically using FDM approach for all above aforesaid cases which were found in quite agreement.

Salmiah Kasolanga et al. [6] conducted experimental work to determine pressure distribution around journal bearing and fluid friction force due to shearing of lubricant. Journal bearing test rig with 12 pressure sensors is used in this work. A journal with length to diameter ratio of 0.5 and 100 mm length is taken and pressure results at different radial load and 600 rpm were obtained. Experiment results were then compared with predict values from Raimondi and Boyd chart. Maximum pressure value from Experiments have shown that it is higher than the theoretical maximum pressure value. The position of the maximum value was found to be fairly consistent in both the theoretical and experimental cases. The coefficient of friction of the oil lubricant in this experiment decreases with increasing load. This shows similar trends to the Raimondi and Boyd charts. It was also observed that the experimental coefficient of friction was significantly higher than expected.

Conclusion Remark
Research are to improve pressure profile and other bearings parameters by making modification on the journal geometry. However, in this work the suggested design imply the following:

- Double pressure profile
- Increase bearing contact area and reduce wearing
- The ability to resist radial and axial loading

3. Theory
Described simple and rapid method to evaluate important design parameters such as maximum pressure, load capacity, maximum temperature, and flow and power loss [8]. Proposed analytical method agrees with finite element method result for a wide range of design parameters. The maximum temperature calculated by present method compared with THD analysis and experimental results, which matches quite well. Values of other parameters also lie within range of different numerical methods, so from this point of view, proposed design method is efficient and simple. Investigated the effect on pressure distribution of hydrodynamic journal bearing by eccentricity ratio of short bearing approximation [9, 10]. Reynolds equation is solved by MATLAB which is compared with literature available and found to be in quite agreement Figure 1. The results are summarized as; high value of eccentricity increases the maximum pressure as well as the negative pressure and increases the steepness of the curve as shown in Figure 2.

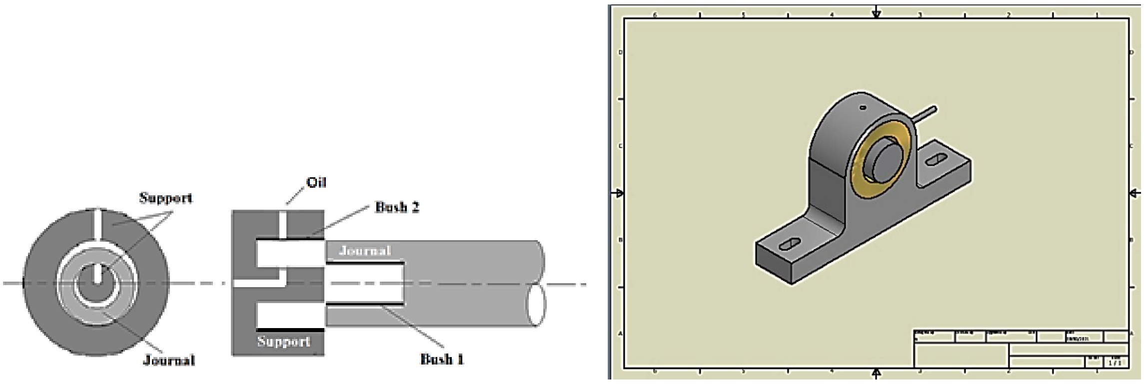

Fig.1. Solid work drawing of compound journal bearing.

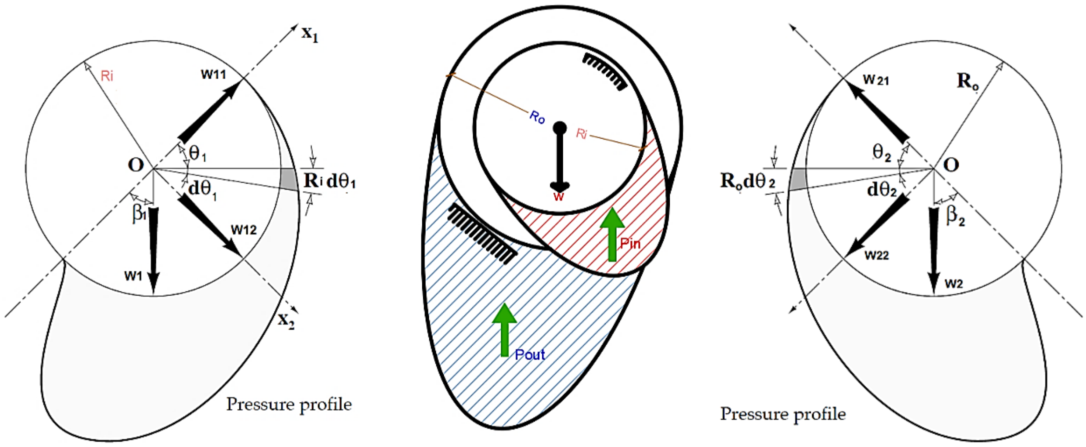

Fig.2. pressure profiles of compound journal bearing.

4. Numerical analysis
Research are to improve pressure profile and other bearings parameters by making modification on the journal geometry. However, in this work the suggested design imply the following.
Double pressure profile:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3\frac{\partial p}{\partial y}\right)=6U\eta\frac{dh}{dx} \qquad (1)$$

Pressure profile and other bearings parameters by making modification on the journal geometry. However, in this work the suggested design imply the following.
Double pressure profile [11]

$$\frac{\partial}{\partial x^*}\left(h^{*3}\frac{\partial p*}{\partial x^*}\right)+\left(\frac{R}{L}\right)^2\frac{\partial}{\partial y^*}\left(h^{*3}\frac{\partial p*}{\partial y^*}\right)=\frac{\partial h^*}{\partial x^*} \quad (2)$$

Where,

$$h^*=\frac{h}{c}.x^*=\frac{x}{R}\ ,\ y^*=\frac{y}{L},\ p^*=\frac{pc2}{6U\eta R}$$

$$MV=p^*h^{*\,1.5} \quad (3)$$

Substituting into eq.2 one gets,

$$\frac{\partial^2 M_v}{\partial\chi^2}+\left(\frac{R}{L}\right)^2\frac{\partial^2 M_v}{\partial\gamma^2}=FM_v+G \quad (4)$$

Where,

$$F=\frac{0.75\left[\left(\frac{\partial\lambda}{\partial\chi}\right)^2+\left(\frac{R}{L}\right)^2\left(\frac{\partial\zeta}{\partial\gamma}\right)^2\right]}{\zeta^2}+\frac{1.5\left[\left(\frac{\partial\lambda}{\partial\chi}\right)+\left(\frac{R}{L}\right)^2\left(\frac{\partial\zeta}{\partial\gamma}\right)\right]}{\zeta} \quad (5)$$

$$G=\frac{\left(\frac{\partial\lambda}{\partial\chi}\right)}{\zeta^{1.5}} \quad (6)$$

Proposed design method is efficient and simple. Investigated the effect on pressure distribution of hydrodynamic journal bearing by eccentricity ratio of short bearing approximation. Reynolds equation is solved by MATLAB which is compared with literature available and found to be in quite agreement. The results are summarized as; high value of eccentricity increases the maximum pressure as well as the negative pressure and increases the steepness of the curve [12].

$$\left(\frac{\partial^2\mu_v}{\partial\chi^{*2}}\right)_i=\frac{\mu_{v,i+1}+\mu_{v,i-1}-2\mu_{v,i}}{(\Delta\chi)^2} \quad (7)$$

Substitute into eq.4 to get,

$$\mu_{v,i,j}=\frac{\left(\frac{1}{(\Delta\chi)^2}\right)(\mu_{v,i+1,j}+\mu_{v,i-1,j})+\left(\frac{R}{L}\right)^2\left(\frac{1}{(\Delta\gamma)^2}\right)(\mu_{v,i+1,j}+\mu_{v,i-1,j})-G_{i,j}}{2\frac{1}{(\Delta\chi)^2}+2\frac{1}{(\Delta\gamma)^2}+F_{i,j}} \quad (8)$$

5. Bearing parameters

The main parameter to be investigated are as the follows,

i. Pressure Distribution

Investigated the effect on pressure distribution of hydrodynamic journal bearing by eccentricity ratio of short bearing approximation. Reynolds equation is solved by MATLAB which is compared with literature available and found to be in quite agreement. The results are summarized as; high value of eccentricity increases the maximum pressure as well as the negative pressure and increases the steepness of the curve [13].

$$p=p^*(6U\eta R)/C^2 \quad (9)$$

ii. Load Capacity

Pressure profile and other bearings parameters by making modification on the journal geometry. However, in this work the suggested design imply the following.
Double pressure profile:

iii. Friction Force

$$F=\int_0^L\int_0^{2\pi R}\tau\,dxdy \quad (11)$$

Where the shear stress 'τ' is given by,

$$\tau=\frac{\eta U}{h}+\frac{h}{2}\frac{dp}{dh} \quad (12)$$

$$\tau=\frac{1}{h*}+3h^*\frac{dp*}{dx*} \quad (13)$$

And equation (11) can be re-written in the following form,

$$F=F^*\left(\frac{RL\eta U}{c}\right) \quad (14)$$

iv. Coefficient of Friction

$$\mu = \frac{F}{W} = \frac{F_* \left(\frac{RL\eta U}{c}\right)}{W_* \left(\frac{6R^2LU\eta}{c^2}\right)} = \left(\frac{C}{6R}\right)\left(\frac{F_*}{W_*}\right) \tag{15}$$

v. Bearing Geometry

Profile and other bearings parameters by making modification on the journal geometry. However, in this work the suggested design imply the following.
Double pressure profile [14]

$$h = c\,(1 + \cos\theta) \tag{16}$$

6. Experimental work

Experimentally investigated water-lubricated hydrodynamic bearing used in nuclear power plants wherein water is used as process fluid. Bearings are designed considering different techniques like ESDU chart Fig3. Reason and Narang technique, Cameron method and Raymondi & Boyd method. After calculating all parameters by above method, comparison of minimum thickness, clearance and power loss, etc. has been made. Result reveals that (a) Design procedure of water-lubricated bearing follows same process as oil-lubricated bearing test rig Fig 4. (b) Material should be non-corrosive, adequate tolerances should be provided and should be properly surface finished Fig 5. (c) With pressure feed condition load carrying capacity is increased Fig 6. (d) temperature rise and power losses are under control Fig 7

Fig 3 CNC turning machine

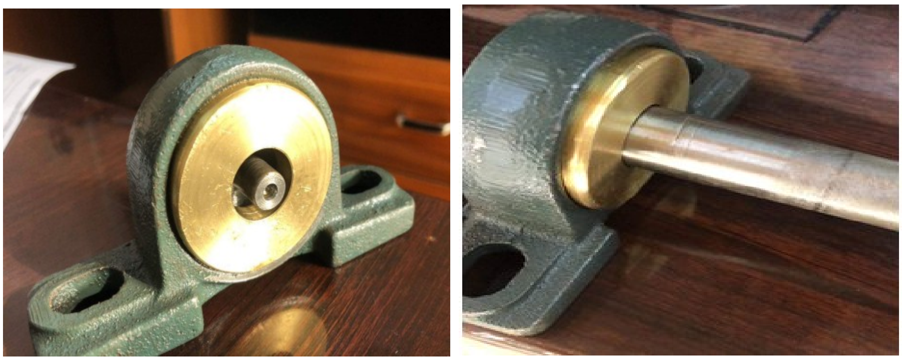

Fig.4. manufacturing process of new design prototype of compound journal bearing using high accuracy cutting tools machine

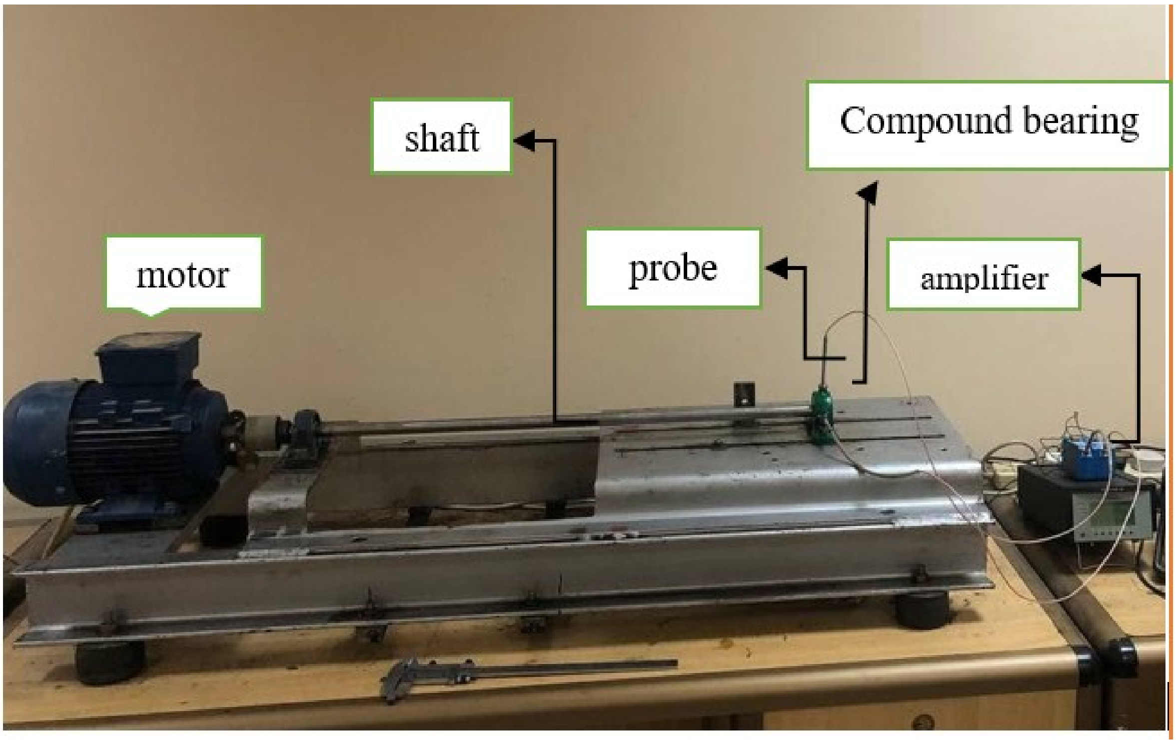

Fig.5. Test Rig

Fig. 6. Eddy sansor .

Fig7. Test instrumentation

Described simple and rapid method to evaluate important design parameters such as maximum pressure, load capacity, maximum temperature, and flow and power loss. Proposed analytical method agrees with finite element method result for a wide range of design parameters. The maximum temperature calculated by present method compared with THD analysis and experimental results which matches quite well. Values of other parameters also lie within range of different numerical methods, so from this point of view, proposed design method is efficient and simple shown in Fig 8 and Fig 9.[15]. The following steps are made for testing and measuring the journal bearing parameters,

1. Selected the speed by adjusting the speed control.
2. Check the alignment of system to avoid vibration and noise.
3. Read the output of the proximity sensor probe.
4. Real-time data transmission to oscilloscope and computer for signal analysis.
5. Repeat the process for multiple points and different rotational speeds.
6. Using the shape of the outer magazine bearing the eccentric ratio, the film thickness and attitude angle can be found from the gap displacement measurement sx and sy.

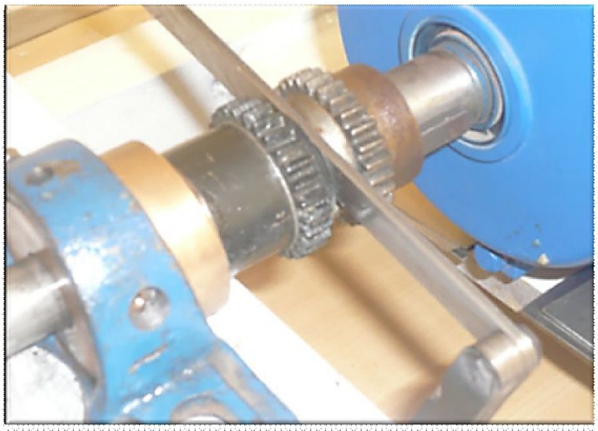

Fig. 8. Face measurement by using failure gauge

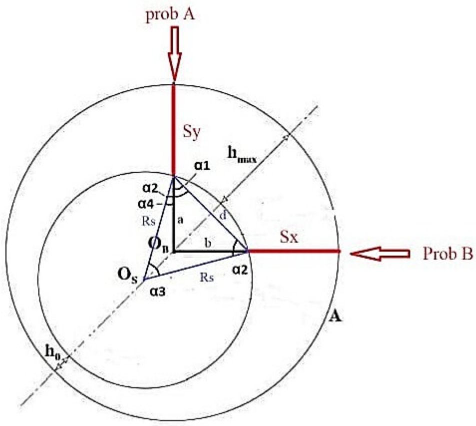

Fig. 9. geometry of outer journal bearing

Referring to the figure, the following expressions can be derived to find attiude angle, eccentricity e. and minimum film thickness Hm;[16],

$$a = Rb - Sy \quad (17)$$
$$b = Rb - Sx \quad (18)$$
$$d = (a^2 + b^2)^{0.5} \quad (19)$$

The attitude angle α 1 can readily found from the following equation,

$$\alpha\,1 = \tan^{-1}\left({}^{b}/_{a}\right) \quad (20)$$

Now by using the sin law one can get,

$$Rs \sin(\alpha 3) = d \sin(\alpha 2) \quad (21)$$
$$\alpha 3 = 180 - 2\,\alpha 2 \quad (22)$$

Solving equations 4 and 6 to find angles α 3 and α 4 as the follows,

$$\alpha 4 = \alpha 2 - \alpha 1 \quad (23)$$

The eccentricity e can be found from using of cosine law,

$$e = (Rs^2 + a^2 - 2 * a * RS * \cos(\alpha 4))^{0.5}\ (\cos \text{law}) \quad (24)$$

Finally the minimum film thickness Hm can be found as,

$$Hm = c\text{-}e \quad (25)$$

As it clear from this analysis one can find attitude angle, eccentricity and minimum film thickness from only measuring the horizontal and vertical gap displacements by using to perpendicular proximate probes.

7. Results and discussions

Simple and rapid method to evaluate important design parameters such as maximum pressure, load capacity, maximum temperature, and flow and power loss. Proposed analytical method agrees with finite element method result for a wide range of design parameters. The maximum temperature calculated by present method compared with THD analysis and experimental results, which matches quite well. Values of other parameters also lie within range of different numerical methods, so from this point of view, proposed design method is efficient and simpleeccentricity ratio from 0.5 to 0.9 with L/D=1 are investigated and the results are shown in Table 1. As is clear from the table, load capacity and pressure the maximum yield increases as the eccentricity ratio increases while both the attitude angle and the coefficient of friction decrease. The effect of the L/D ratio Fig 10 described simple and rapid method to evaluate important design parameters such as maximum pressure, load capacity, maximum temperature, and flow and power loss. Proposed analytical method agrees with finite element method result for a wide range of design parameters. The maximum temperature calculated by present method compared with THD analysis and experimental results, which matches quite well. Values of other parameters also lie within range of different numerical methods, so from this point of view, proposed design method is efficient and simple Fig 11.

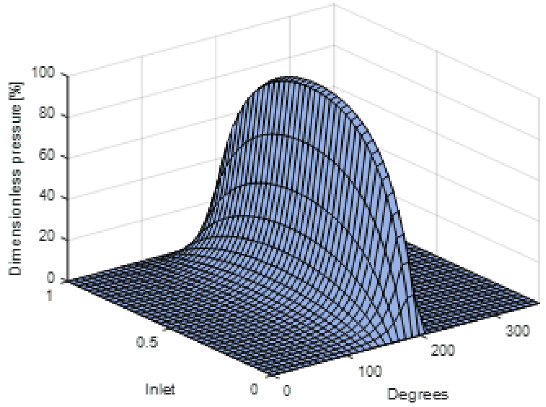

Fig.10 : pressure distribution

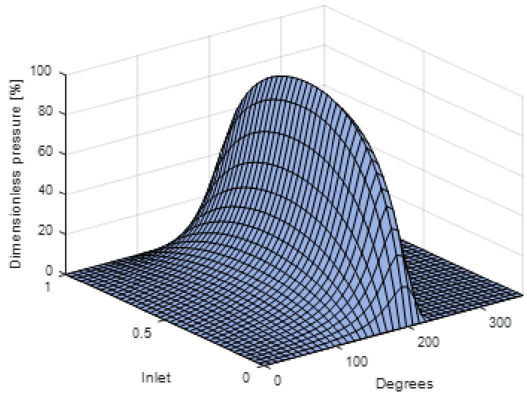

Fig.11 : pressure distribution of compound

Table 1 shows the effect of ε on maximum pressure and load capacity on compound journal bearing with L = 25 mm, Do = 20 mm, Di = 10 mm. it can be seen that increasing with of ε leds to increasing of pressure .Also the load capacity increases, The film thickness decrease as the dimensionless pressure increase, This means that the lighter film thickness led to the higher dimensionless pressure and higher load capacity. At ε = 0.5 pressure is 2.263 , load capacity is 1.6078 .at ε = 0.9 the pressure is 5.438, the load capacity is 2.804, .

Table 1: effect of change of eccentricity ratio

Case	e	W(kg)	P(N/m)	β	μ
1	0.41	0.29162	0.333034	58.3758°	43.4199
2	0.52	0.432644	0.523964	52.0499°	40.433
3	0.64	0.673912	0.942012	44.3856°	27.1375
4	0.72	1.13894	1.83762	36.7414°	11.4317
5	0.82	2.83044	5.43389	26.3888°	5.3947

Table 2 shows the effect of ε on maximum pressure and load capacity on compound journal bearing with L = 50 mm, Do = 30 mm, Di = 20 mm. it can be seen that increasing with of ε led to increasing of pressure .Also the load capacity increases, The film thickness decrease as the dimensionless pressure increase, This means that the lighter film thickness led to the higher dimensionless pressure and higher load capacity. At L/D = 1.61 pressure is 2.3263 , load capacity is 1.6078 .at L/D = 0..63 the pressure is 1.236, the load capacity is 0.7235, the experimental sx and sy shown in Table 3 are values of film thickness and attitude angle β are evaluated

Table 2. Effect of change of L/D ratio.

Case	L/D	W(kg)	P(N/m)	β	μ
1	1.61	1.60783	2.32636	40.5942°	7.02493
2	1.43	1.49202	2.17312	39.4567°	7.44233
3	1.24	1.35272	2.05329	38.1445°	7.94469
4	14	1.18394	1.83762	36.7414	11.3417
5	0.83	0.977357	1.61365	35.3484°	23.8
6	0.63	0.723555	1.2365	33.6644°	69.4237

Table 3. Experimental values of film thickness and attitude angle β are evaluated.

symbols	Sx (mm)	Sy (mm)	h exp. (mm)	β exp. (Deg)	h heo. (mm)	β theo. (Deg)
value	0.111	0.104	0.0103	45	0.008	43.3

8. Conclusions

1- The main operating parameters of compound bearing such as pressure distribution, load capacity, film thickness, friction force and coefficient of friction can be sufficiently evaluated from using of two dimensional Reynolds equation with FDM solution.

2- The suggested compound journal bearing can efficiently withstand both transverse and thrust loading so it can serve for heavily applications, in this regards the compound journal can improve the following:

   a. Introduce hydrodynamic pressure distribution at two oil layer one at inner journal – bush interface and the other at outer journal- bush interface and as a result the maximum pressure will increase about 23.6% as compared with normal journal bearing
   b. Increases the carrying load capacity by 12.90% since the pressure increases
   c. Decrease wear since the contact area increases due to the creating of inner bearing part
   d. Reducing the friction force and coefficient of friction by (23 %) and (16 %) respectively, this also increase wear resistance and hence increase operating life span.
   e. Increase the critical speed of stability by 33%, as the compare between normal and compound journal bearing. So it can be used safely at high speed machinery applications.

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