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Experimental Investigation of Effect of Water Emulsion on Load Carrying Capacity of Hydrodynamic Journal Bearing

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ABSTRACT

Industrial machinery with high horsepower and high loads, such as steam turbines, centrifugal compressors, pumps and motors, utilize journal bearings as rotor supports. In this paper Load Carrying capacity of Journal Bearing is studied by considering Sommerfeld and Reynolds boundary conditions. Solution of two dimensional Reynolds equations is done using Finite difference method and graph is obtained with help of MATLAB. Experiment work is carried out for investigation of effect of water emulsion on Load Carrying Capacity of journal bearing by considering different water ratio on Hydrodynamic Journal Bearing Apparatus. Experimental results are compared for effect of water emulsion on Load Carrying Capacity for same speed and load condition.

Keywords: Journal Bearing, Load Carrying Capacity, Reynold Equation, Sommerfeld & Reynolds Boundry Condition.

I. INTRODUCTION

In hydrodynamic lubricated bearings, there is a thick film of lubricant between the journal and the bearing. When the bearing is supplied with sufficient lubricant, a pressure is build up in the clearance space when the journal is rotating about an axis that is eccentric with the bearing axis. The load can be supported by this fluid pressure without any actual contact between the journal and bearing. The load carrying ability of a hydrodynamic bearing arises simply because a viscous fluid resists being pushed around. Under the proper conditions, the resistance to motion will develop a pressure distribution in the lubricant film that can support a useful load. The load supporting pressure in hydrodynamic bearing arises from either : The flow of a viscous fluid in a converging channel or :The resistances of a viscous fluid to being squeezed out from between approaching surfaces. So many authors has tried to study the pressure distribution of journal bearing, some of the cases are discussed here. J. A. Cole and C. J. Hughest [1] have presented Pressure distribution of Journal bearing for different Boundary conditions. Pressure plotting has been used by a number of investigators to determine the extent of the load-carrying film in complete bearings. A. Cameron and Mrs. W. L. Wood [2] have compared Pressure distribution for various boundary conditions with experimental result done by Nucker. D. M. Nuruzzaman, M. K. Khalil, M. A. Chowdhury, M. L. Rahaman [3] have presented variation of pressure with angular position by analytical and FEM method. The results showed that the hydrodynamic pressure profile increased steadily from zero and it changed very rapidly in the area of the smallest film thickness and reached to a maximum. In this region the film was convergent, the pressure then gradually dropped to zero. D. W. Garside and S. Hother-Lushingtonj-[4] presented the main differences between water and oil which affect its use in plain bearings are:The lack of boundary lubrication properties and The viscosity of water is about 1/30th of that of oil. This implies that only 1/30th of the load can be carried when using water if the same minimum film thickness is to be maintained with the same size bearing.Y. I. Kvitnitsky, N. F. Kirkatch and Y. D. Poltavsky [5] has studied the solution of Reynolds equation for hydrodynamic pressures of the load-carrying lubricating layer in journal bearings is discussed. B.P. Williamson, K. Walters, and T.W. Bates [6]: In this paper authors has address a question that as to whether the viscoelastic properties of multi grade oils can have a measurable effect on lubrication characteristics. Nabil Motosh [8] has studied The effect of the dependence of oil viscosity on the temperature and pressure in the oil layer is examined for the case of cylindrical plain bearings under static loading. M. B. W. Nabhan, G. A. Ibrahim and M. Z. Anabhawi [9] have presented effect of water emulsion on load carrying capacity of Journal Bearing. Here viscosity ratio is viscosity of upper layer divided by viscosity of lower layer. D Ashman [10] has given details of a combined theoretical and experimental investigation of a plain journal bearing under heavily loaded conditions together with a metrological study of the bearing geometry.

II. METHODS AND MATERIAL

A. Theoretical Analysis

The Reynolds equation is given by

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

The next requirement is to obtain the value of the load resulting from these pressures, or stated more formally, to find the value of the external load, in magnitude and direction, which will equilibrate the pressures generated in the oil film.

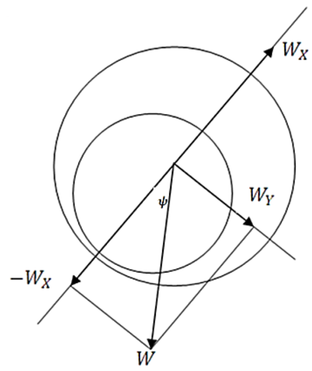

Figure 1. Pressure Curve and Attitude Angle

The forces on this small element can be integrated over the whole pressure field, so if W_X is the total integrated force acting in the X direction and W_Y is the force in the normal, Y, direction then

$W_X = L\int_0^{2\pi} pR\,d\theta\cos\theta$ and

$$W_Y = L\int_0^{2\pi} pR\,d\theta\sin\theta$$

$$W = \sqrt{(W_Y^2 + W_X^2)}$$

FINITE DIFFERENCE SOLUTION FOR 2-D REYNOLDS EQUATIONS (FDM)

The Reynolds full equation in 2D is,

$$\partial/\partial x\,(h^3\,\partial p|\,\partial x) + \partial/\partial y\,(h^3\,\partial p|\,\partial y) = 6\,U\eta\,dh/dx$$

Substituting $h = c(1 + e\cos\theta)$,

$x = r\theta$, so $dx = rd\theta$,

$Y = Ly$, so $dy = Ldy$,

$$\partial/r\,\partial\theta\,[c^3(1 + e\cos\theta)^3\,\partial p/r\,\partial\theta] + \partial/L\,\partial y\,[c^3(1 + e\cos\theta)^3\,\partial p/L\,\partial y] = 6\,U\eta\,\partial/r\,\partial\theta\,(1 + \varepsilon\cos\theta)\,c$$

Again, $p = 6U\eta rP/c^2$

$$\partial/r\,\partial\theta\,[c^3(1 + \varepsilon\cos\theta)^3\,\partial/r\,\partial\theta(6U\eta rP/c^2)] + \partial/L\,\partial y\,[c^3(1 + \varepsilon\cos\theta)^3\,\partial/L\,\partial y(6U\eta rP/c^2)] = 6\,U\eta\,\partial/r\,\partial\theta\,(1 + \varepsilon\cos\theta)\,c$$

Here, $\partial/r\,\partial\theta\,(6U\eta rP/c^2) = (6U\eta r/c^2)\,\partial P/r\,\partial\theta$ &
Canceling $(6U\eta r/c^2)$ from both sides,

$$\partial/r\,\partial\theta\,[(1 + \varepsilon\cos\theta)^3\,\partial P/r\,\partial\theta] + (r\,/\,L^2)\,\partial/\partial y\,[(1 + \varepsilon\cos\theta)^3\,\partial P/\partial y] = \partial/r\,\partial\theta\,(1 + \varepsilon\cos\theta)$$

Taking $1/r$ as common from both sides,

$$\partial/\partial\theta\,[(1 + \varepsilon\cos\theta)^3\,\partial P/r\,\partial\theta] + (r^2\,/\,L^2)\,\partial/\partial y\,[(1 + \varepsilon\cos\theta)^3\,\partial P/\partial y] = \partial/\partial\theta\,(1 + \varepsilon\cos\theta)$$

Now $\partial f/\partial x\,(u * v) = u\,\partial v/\partial x + v\,\partial u/\partial x$,
Similarly,

$$[3(1 + \varepsilon\cos\theta)^2\,\partial P/\partial\theta(-\varepsilon\sin\theta)] + (1 + \varepsilon\cos\theta)^3\,\partial^2P/\partial^2\theta + r^2\,/\,L^2[(1 + \varepsilon\cos\theta)^3\,\partial^2P/\partial^2 y] = -\varepsilon\sin\theta$$

Taking,

$$(1 + \varepsilon\cos\theta) = A\ \&\ r/L = a$$

$$(\ -3A^2\varepsilon\sin\theta\ )\ \partial P/\partial\theta + A^3\,\partial^2P/\partial^2\theta + A^3a^2\,\partial^2P/\partial^2 y = -\varepsilon\sin\theta$$

Rearranging the terms,

$$A^3\,\partial^2P/\partial^2\theta - 3A^2\varepsilon\sin\theta\,\partial P/\partial\theta + A^3a^2\,\partial^2P/\partial^2y + \varepsilon\sin\theta = 0$$

This is non-dimensional formulation of 2-D Reynolds equation.

Taking Summerfield boundary condition,

P = 0 @ θ = 0, & y = 0

P = 0 @ θ = pi, & y = 1

$$\partial^2P/\partial^2\theta = P_{i-1,j} - 2P_{i,j} + P_{i+1,j}/dh^2$$
$$\&\ \partial P/\partial\theta = P_{i+1,j} - P_{i-1,j}/2dh$$
$$\partial^2P/\partial^2y = P_{i,j-1} - 2P_{i,j} + P_{i,j+1}/dk^2$$

Putting this all condition in equation Non dimensional form of Reynolds equation,

$$A^3\left[P_{i-1,j} - 2P_{i,j} + P_{i+1,j}/dh^2\right] - 3A^2\varepsilon\sin\theta\left[P_{i+1,j} - P_{i-1,j}/2dh\right] + A^3a^2\left[P_{i,j-1} - 2P_{i,j} + P_{i,j+1}/dk^2\right] + \varepsilon\sin\theta = 0$$

Taking $(2dh^2dk^2)$ as LCM,

$$2dk^2A^3P_{i-1,j} - 4dk^2A^3P_{i,j} + 2dk^2A^3P_{i+1,j} - 3A^2\varepsilon\sin\theta\,P_{i+1,j} + 3A^2\varepsilon\sin\theta\,P_{i-1,j} + 2dh^2A^3a^2P_{i,j-1} - 4dh^2A^3a^2P_{i,j} + 2dh^2A^3a^2P_{i,j+1} = -2\,\varepsilon\sin\theta\,dh^2dk^2$$

$$4A^3[dk^2 + dh^2a^2]P_{i,j} = A^3dk^2[2 - 3\,\varepsilon\sin\theta\,dh/A]P_{i-1,j} + A^3dk^2[2 + 3\,\varepsilon\sin\theta\,dh/A]P_{i+1,j} + 2dh^2A^3a^2\left[P_{i,j-1} + P_{i,j+1}\right] + 2\,\varepsilon\sin\theta\,dh^2dk^2$$

$$[dk^2 + dh^2a^2]P_{i,j} = dk^2/4\,[2 - 3\,\varepsilon\sin\theta\,dh/A]P_{i-1,j} + dk^2/4\,[2 + 3\,\varepsilon\sin\theta\,dh/A]P_{i+1,j} + dh^2a^2/2\left[P_{i,j-1} + P_{i,j+1}\right] + \varepsilon\sin\theta\,dh^2dk^2/2A^3$$

$$[dk^2 + dh^2a^2]P_{i,j} = [dk^2/2 - 3\,\varepsilon\sin\theta\,dh\,dk^2/4A]P_{i-1,j} + [dk^2/2 + 3\,\varepsilon\sin\theta\,dh\,dk^2/4A]P_{i+1,j} + dh^2a^2/2\left[P_{i,j-1} + P_{i,j+1}\right] + \varepsilon\sin\theta\,dh^2dk^2/2A^3$$

Now using this FDM (2-D) equation, we made the MATLAB program to study pressure distribution of journal bearing by putting different boundary condition like Half Sommerfeld, Full Sommerfeld and Reynolds condition.

B. Experimental Investigation

Apparatus: The 'DYNAMIC' apparatus consists of a journal with brass bush pressed over outer diameter. Bearing caps are provided on both sides of bearing to which loading attachment is fixed. The journal is rotated by variablel speed D.C. motor. A torque arm with scale is fixed to bearing. This is used along with sliding weight to determine the friction torque. An oil corrector tray is provided for measurement of oil flow from bearing.

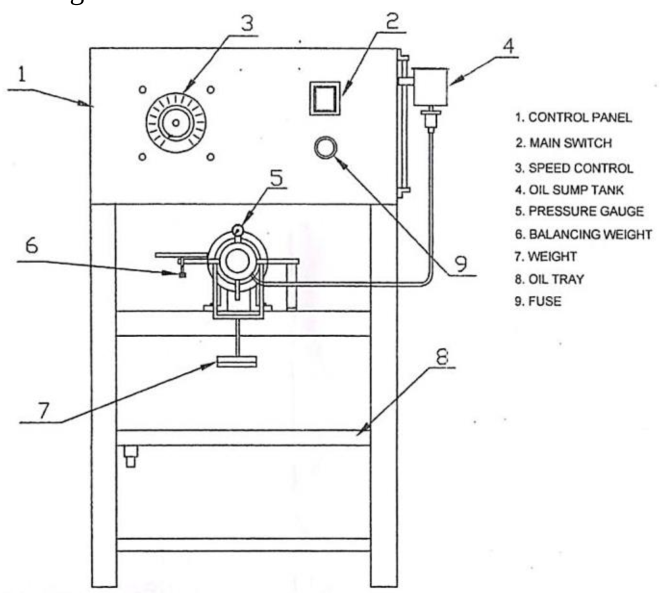

Figure 2. Line Diagram of Journal Bearing Apparatus

Experimental Procedure:

Fill up sufficient oil in the oil supply tank and open the bottom cock so that oil is let to the bearing. Adjust the pressure gauge at 0°. Adjust the pointer on torque arm to match with the zero on the scale fitted on the frame. Put 'On' the supply and start the motor at required speed. Pressure will start to develop. Put the required weight in the weight hanger. Put small weight in balancing hook & adjust the distance so that the pointer should again coincide with zero on the scale. Note down weight & its distance. Wait for some time for pressure to build up. When pressure remains steady, note down pressure. Insert measuring flask at flowing oil and measure the time required for 10ml. also, hold thermometer in dropping oil and note down oil temperature. Repeat the procedure for different speeds and loads, and complete the observation table.

Now ,keeping one speed constant and varying different load, pressure are measured for pure oil, then 10% water in oil emulsion then similarly for 20% and 30%. Now speed is changed and similar procedure will give us reading for pressure.

Calculation Procedure:

Load carrying capacity of the bearing:

$$W = \frac{U\eta L^3}{C^2} \times \frac{\pi}{4} \times \frac{\varepsilon}{(1-\varepsilon^2)^2} \times (0.62\,\varepsilon^2 - 1)^{0.5}$$

Where

W = Total load, N

(Initial weight of the loading arrangement over the bearing is 4.39 kg)

U = Surface speed of shaft, m/s

$$= \frac{\pi D N}{60}$$

Where

D = Shaft dia. = 0.0498 m

L = Length of bearing = 0.06m

N = Shaft Speed in rpm

η = Viscosity of oil, N-s/m2 = Centipoise / 100.

From known (applied) load, determine ε, by some trial & error. We have made a MATLAB program to find value of eccentricity ratio ε.

Measurement of Kinematic Viscosity:

Figure 3. Red Wood Viscometer

Figure 4. Digital Tachometer

Experimental Setup

The Redwood viscometer No.1 consists essentially of a cylindrical brass cup provided with an agate jet in its base. The oil cup is surrounded by a loose cylinder provided with vanes for stirring the liquid bath.

Figure 5. Digital Thermometer

The latter which is cylindrical, is provided either with a side tube for heating with an electric heating element applied to the walls but not to the underside of the bath. The gate jet may be closed by means, of a metal ball attached to the stiff wire. This can be removed and suspended from the thermometer support during the run. A wire attached internally near the top of the side of the cup indicate, the level to which the sample mist be adjusted at the beginning of the experiment. A Variac may be used to control heating process by varying the voltage applied to the heater element.

Method of calculation

The kinematic viscosity of oil is experimentally determined by using the empirical equation,

$\gamma = At - B/t$, Where,

γ = Kinematic viscosity in centistokes of the oil at the test temperature.

t = Time of flow of oil in seconds through the orifice of the Redwood Viscometer of fixed volume of the oil in the apparatus at the test temperature.

A & B are experimental constants determined by the design of the instrument sometimes referred to as the coefficient of kinetic energy value of A. & B can be obtained by calibration of the apparatus.

The values are A = 0.26 & B = 171

For the Redwood Viscometer No. 1 available in laboratory. The equation can be written as:

Kinematic Viscosity = (0.26 t – 171/t) centistokes.
Absolute of dynamic viscosity can be obtained by noting down the density of oil at different temperatures and multiplying by the kinematic viscosity at that temperature.

Table 1 Value of Kinetic Viscosity for Different Water in Oil Emulsion Ratio

Pure Oil / water Emulsion	Time Required for 50 ml of the oil to collect in Sec.	Kinematic Viscosity (Centistokes)
Pure Oil	667	173.16
10% water	604	156.75
15% water	566	146.86
20% water	536	139.04
25% water	498	129.13
30% water	470	121.80

III. RESULTS AND DISCUSSION

Results for Different Speed, Load and Pressure:

Table 2 Value of Eccentricity Ratio and Pressure at 300 rpm

Sr. No.	Load (N)	Eccent. Ratio	P0 Kg/cm2	P10 Kg/cm2	P20 Kg/cm2	P30 Kg/cm2
1	141.1	0.0693	1.35	1.23	1.83	1.65
2	239.2	0.1174	2.34	2.15	2.72	2.45
3	337.3	0.1656	3.12	2.80	1.1	1.48
4	435.4	0.2138	1.87	2.19	2.43	0.98
5	543	0.2667	1.28	1.65	1.92	2.19

Table 3 Value of Eccentricity Ratio and Pressure at 500 rpm

Sr. No.	Load (N)	Eccent. Ratio	P0 Kg/cm2	P10 Kg/cm2	P20 Kg/cm2	P30 Kg/cm2
1	141.1	0.0415	1.48	1.35	1.19	1.05
2	239.2	0.0705	2.14	2.02	1.73	1.53
3	337.3	0.0994	2.67	2.42	2.16	1.89
4	435.4	0.1283	2.96	2.66	2.39	2.08
5	543	0.1600	3.43	3.09	2.78	2.51

Table 4 Value of Eccentricity Ratio and Pressure at 700 rpm

Sr. No.	Load (N)	Eccent. Ratio	P0 Kg/cm2	P10 Kg/cm2	P20 Kg/cm2	P30 Kg/cm2
1	141.1	0.0297	1.63	1.47	2.36	2.16
2	239.2	0.0500	2.83	2.57	3.23	2.92
3	337.3	0.0710	2.83	2.57	3.23	2.92
4	435.4	0.0916	3.64	3.29	1.94	2.31
5	543	0.1143	2.62	2.31	2.59	2.94

Table 5 Value of Eccentricity Ratio and Pressure at 900 rpm

Sr. No.	Load (N)	Eccent. Ratio	P0 Kg/cm2	P10 Kg/cm2	P20 Kg/cm2	P30 Kg/cm2
1	141.1	0.0231	1.86	1.73	2.79	2.54
2	239.2	0.3910	2.98	2.69	3.46	3.14
3	337.3	0.0552	3.82	3.46	1.49	2.29
4	435.4	0.0712	2.41	2.62	3.05	1.32
5	543	0.0889	1.98	2.14	2.42	2.68

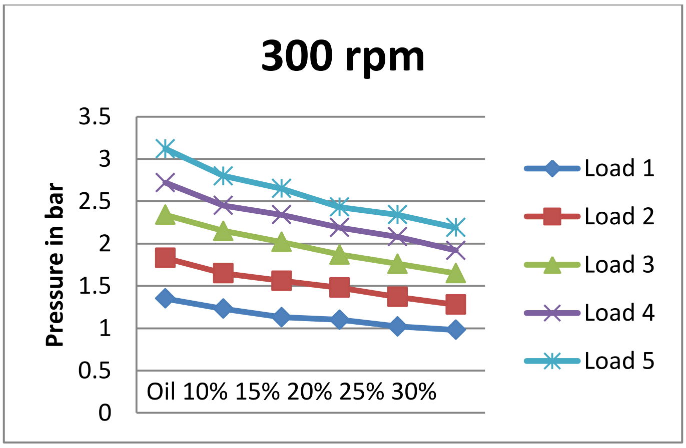

Figure 6. Pressure for Different Water in Oil Emulsion Ratio at 300 rpm

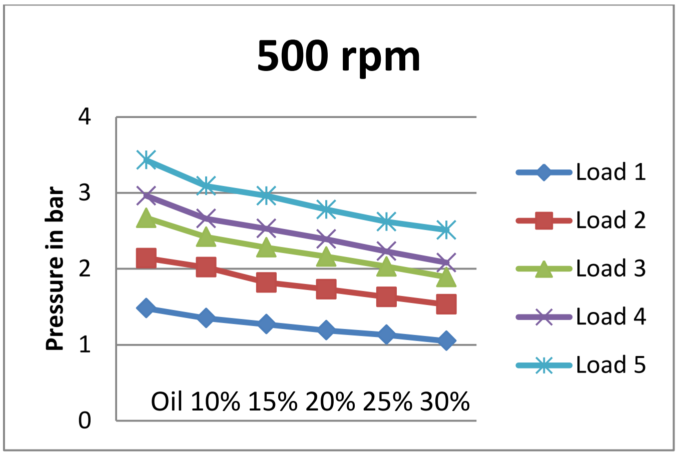

Figure 7. Pressure for Different Water in Oil Emulsion Ratio at 500 rpm

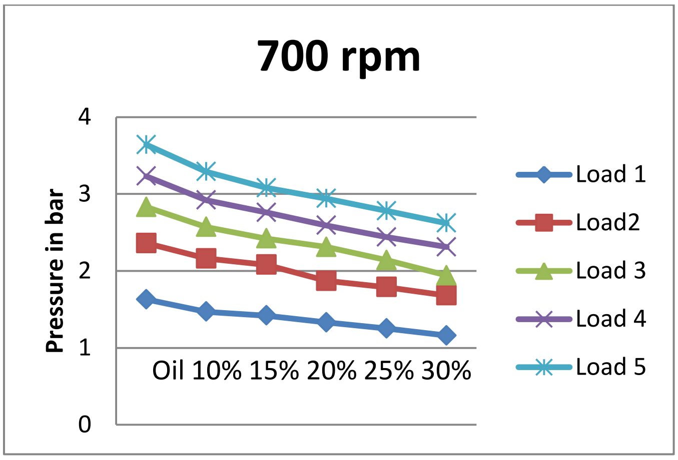

Figure 8. Pressure for Different Water in Oil Emulsion Ratio at 700 rpm

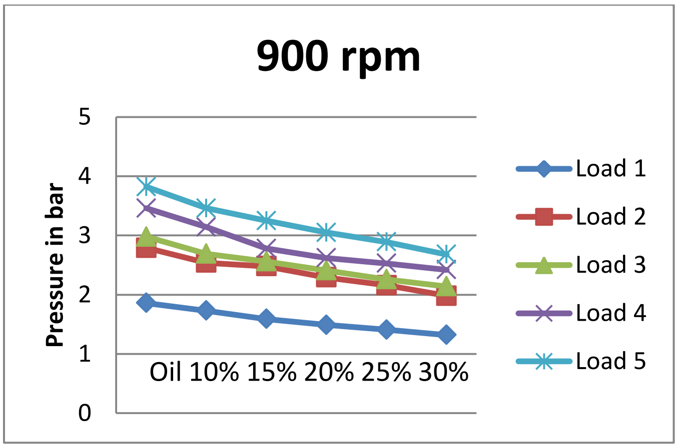

Figure 9. Pressure for Different Water in Oil Emulsion Ratio at 900 rpm

From Above four Graphs we can conclude that for same load with increase in water in oil emulsion % there is decrease in Pressure and that ultimately reduces Load Carrying Capacity of Journal Bearing.

Table 6 Theoretical and Experimental Reduction in Load Capacity

Sr. No.	Different Water in Oil Emulsion Ratios	Theoretical Reduction In Load Capacity	Experimental Reduction In Load Capacity
1	Pure Oil	-	-
2	10% water	9.67 %	9.09 %
4	20% water	19.33 %	19.66 %
6	30% water	29 %	29.06 %

IV. CONCLUSION

The one dimensional solution method is extended to get the two dimensional solution for the pressure distribution in the Journal Bearing. Peak pressure obtained in two dimensional cases is less than the one dimensional case, which is true because we neglect the side leakages in one dimensional analysis.

Experimental work is carried out on Journal Bearing Apparatus. Different combination of water in oil emulsion is taken and pressure is measured for operating parameters for different Speed & load combination. Calculation related to eccentricity ratio is doneWhen the bearing is supplied with sufficient lubricant, a pressure is build up that support the load. curve are presented. Experimental work is done on Journal Bearing Apparatus. Different combination of oil with water is taken and pressure is measured for same load and same speed for different water and oil combination. Observation table and related Graph are presented which clearly shows reduction in Pressure for same load and same speed with increase in water percentage which ultimately reduces load carrying capacity of Journal bearing. In the literature review Full Sommerfeld, Half Sommerfeld and Reynold condition with graph represented by different author is studied. Then effect of viscosity on pressure and load capacity is studied and finally objectives are defined.

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