Study of Gas Lubricated Slider Bearing used in Hard Drive with Different Profiling

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Abstract - *The behavior of load carrying capacity of the gas lubricated bearing depends upon the boundary condition as well as geometry of the bearing. Here, a numerical analysis of a gas lubricated bearing with different type of surface having conditions with different type of geometry is consider for same set of known values and finally correlated all the geometry and explaining which is more efficient in terms of load carrying capacity and flow rate of air etc.*

Obtain new general Reynolds equation for gas lubricated bearing consider nonlinearity and Numerical simulation for pressure distribution of Gas slider bearing and calculating load carrying capacity as varying bearing number.

Keywords: Gas lubricated bearing, Reynolds equation, simulation, hydrodynamics and boundary condition.

1. Introduction

A bearing is a device to permit fixed direction motion between two parts, typically rotation or linear movement. Bearings may be classified broadly according to the motions they allow and according to their principle of operation. Lubrication is fundamental to the operation of all engineering machines. It is required to minimize friction, wear and also provides a cooling function and a surface protection function.

1.1 Study of Gas Lubricated Bearing

Lubrication is fundamental to the operation of all engineering machines. It is required to minimize friction, wear and also provides a cooling function and a surface protection function. Gas lubricated bearings have numerous advantages over liquid and solid lubricated bearings for a wide range of applications. A gas bearing is virtually frictionless, silent, and vibration free bearing. Gas bearings can be used for extremely large surface velocities. A gas bearing can eliminate the risk of contaminating a process with lubricant. A gas / air bearing can be hydrodynamic or hydrostatic. In hydrodynamic bearings the gas is introduced into the bearing surfaces by the action of the bearing. In hydrostatic bearing the gas is introduced under pressure from an external source. Fluid bearings are bearings which support the bearing's loads solely on a thin layer of gas.

They can be broadly classified according to their principle operation:

a. Hydrostatic bearings are externally pressurized fluid bearings, where the fluid is air, and the pressurization is done by a pump.
b. Hydrodynamic bearings rely on the high speed of the journal self-pressurizing the gas in a wedge between the faces.

1.2 Applications of Gas Lubricated Bearing

Because gas bearings have such characteristics as low fiction, high precision and low pollution, they have been successfully used in many commercial applications, such as navigation systems, computer disk drives, high-precision instruments and sensors, dental drills, machine tools, and turbo- compressors.

1.3 Computer disk drives

Computer disk drive is a drive which uses magnetic storage with help of slider which record the magnetic signal from magnetic disk and transmit to the computer for future work. Hence slider bearing play a vital role in this system.

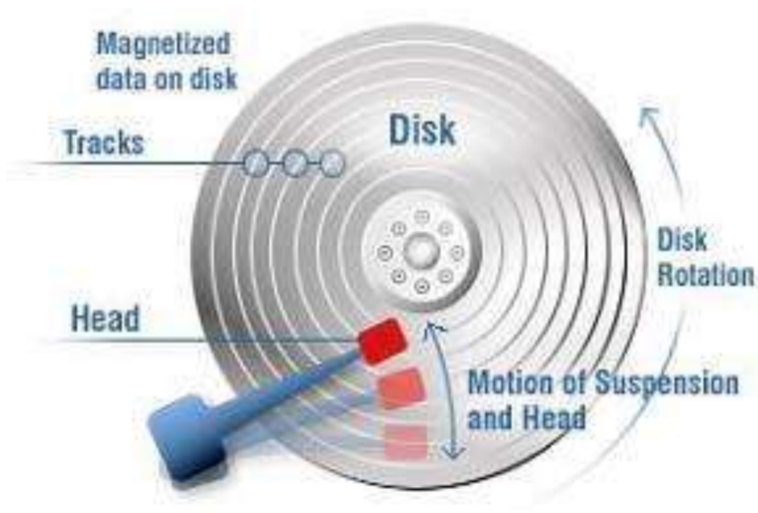

Fig -1: Disc drive using slider [2]

2. Mathematical Modeling

2.1 Basic Equation

The generalized Reynolds equation, a differential equation in pressure, which is used frequently in hydrodynamics theory of lubrication, can be deduced from the Navier-Stokes equation along with the continuity equation under certain assumptions.

1. The height of film h is very small compared to the bearing length, *l*
2. Inertia and body force term are negligible compared with the pressure and viscous terms.
3. There is no variation of pressure across the fluid film, means $\partial p / \partial y = 0$.
4. There is no slip in the fluid solid boundaries.
5. No external force act on the film.
6. The flow is viscous and laminar.
7. Due to geometry of fluid film the derivatives of *u* and *w* with respect to y are much larger than other derivatives of velocity components.

The generalized Reynolds equation will be:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial y}\right)=\frac{\partial}{\partial x}\left(\frac{\rho(u_a+u_{b)h}}{2}\right)+\frac{\partial}{\partial z}\left(\frac{\rho(w_a+w_{b)h}}{2}\right)+$$

$$\rho(v_a-v_b)\quad \rho u_a\frac{\partial h}{\partial x}-\rho w_a\frac{\partial h}{\partial x}+\frac{\partial(\rho h)}{\partial t}\qquad \ldots\ldots\ldots\ldots 3.1.1$$

The two terms in the left hand side describe the net flow rates due to pressure gradients, the first Terms of the right side describe the flow rates due to surface velocities and these are known as Poiseuille and Couette terms, respectively. The last four terms describe the net flow rates due to Squeeze motion and local compression. In practice all the velocity component are not present. If the boundaries velocities confined to the Following values

$$u_a = u_a \qquad v_a = 0 \qquad w_a = 0$$

$$u_b = u_b \qquad v_b \sim u_b \frac{\partial h}{\partial x} \qquad w_b = 0$$

Hence equation 3.1.1 can be written as

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial y}\right) = U\frac{\partial}{\partial x}\left(\frac{\rho h}{2}\right)$$

………………..3.1.2

Equation 3.1.2 is known as standard **Reynolds equation**.

Gas-lubricated film obeys a polytrophic relation:

$$\frac{p}{\rho^n} = \text{constant}$$

…………………3.1.3

Where n is the polytropic gas-expansion exponent, whose value lies between 1 and when flow is adiabatic, i.e., there is no transferred heat and the change in internal Energy equals the compression work, (2) with n =γ =C_p/C_v follows directly from the Equation of state and the energy equation. When flow is isothermal, with n = 1 derives From the equation of state is the specific heat per unit weight for constant pressure and is the specific heat per unit weight for constant volume.

Hence using equation 3.1.3 in equation 3.1.2, and taking isothermal condition, we get

$$\frac{\partial}{\partial X}\left(h^3 p\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial Y}\left(h^3 p\frac{\partial p}{\partial y}\right) = 6\eta\, u_o\frac{\partial(ph)}{\partial x} \qquad \text{……….3.1.4}$$

Non–dimensional form of equation 3.1.4 is

$$\frac{\partial}{\partial X}\left(H^3 P\frac{\partial P}{\partial X}\right) + \frac{\partial}{\partial Y}\left(H^3 P\frac{\partial P}{\partial Y}\right) = \mathrm{C}\frac{\partial(PH)}{\partial X} \qquad \text{……..3.1.5}$$

Where,

$$\mathrm{Y}=\frac{y}{B},\ \mathrm{H}=\frac{h}{h_m},\ \mathrm{C}=\frac{6\eta\,\mathrm{U}_o B}{h_m^2 p_a},\quad \mathrm{X}=\frac{x}{B}$$

By solving equation 3.1.5 we get

$$\left[\left[H^3\left(\frac{\partial P}{\partial x}\right)^2 + \frac{3PH^2}{1}\frac{\partial H}{\partial x}\frac{\partial P}{\partial x} + PH^3\frac{\partial^2 P}{\partial x^2}\right] - C\left[\frac{\partial P}{\partial x}H + \frac{\partial H}{\partial x}P\right] + \left(\frac{B}{L}\right)^2\left\{H^3\left(\frac{\partial P}{\partial Y}\right)^2 + 3PH^2\frac{\partial H}{\partial Y}\frac{\partial P}{\partial Y} + PH^3\frac{\partial^2 P}{\partial Y^2}\right\}\right] = 0$$

…….3.1.6

2.2 Boundary conditions

The boundary condition for any gas slider bearing is just depends upon slip function. A first order slip boundary condition is derived using the solution of the equation for the velocity profile & the new slip model includes an additional term due to the pressure gradient along the flow direction.

$$P(X,0)=1, \quad P(0,X)=1, \quad P(1,Y)=1, \quad \frac{dp}{dx}(0,Y)=0 \qquad [\text{Where, } 0 < X < 1,\ 0 < Y < 1]$$

The above Reynolds equation is used directly for simulating the results but in reference paper some approximation is used.

According to reference paper they simply put $u=P^2 H^2$ in equation 3.1.5 then equation become,

$$-H(X,Y)\left(\frac{\partial^2 u}{\partial^2 X}+\frac{\partial^2 u}{\partial^2 Y}\right)+\frac{\partial H}{\partial X}\frac{\partial u}{\partial X}+\frac{\partial H}{\partial Y}\frac{\partial u}{\partial Y}+2\left(\frac{\partial^2 H}{\partial^2 X}+\frac{\partial^2 H}{\partial^2 Y}\right)u+\frac{C}{\sqrt{u}}\frac{\partial u}{\partial Y}=0 \quad \text{..........3.1.7}$$

The boundary condition will be

$$u(X,0)-H^2(X,0),$$
$$u(X,1)-H^2(X,1),$$
$$u(L/2,Y)-H^2(L/2,Y),$$
$$\frac{\partial u}{\partial X}(0,Y)=0$$

Using finite difference method some approximation is taken in reference paper is

$$\frac{C}{\sqrt{u}}\frac{\partial u}{\partial Y} \approx \frac{C}{\sqrt{u(i,j)}}\frac{\Delta u(i,j)}{\Delta Y} \quad \text{…………3.1.8}$$

3. Numerical Solution

In this project simulation is doing on MATLAB. The area is converted into numbers of grids and centre difference method is applied for calculating the pressure in every node of the grid.

According to Centre Difference method:

$$\frac{\partial p}{\partial x}=\frac{P(i+1,j)-P(i-1,j)}{2\Delta X}$$

$$\frac{\partial p}{\partial Y}=\frac{P(i,j+1)-P(i,j-1)}{2\Delta Y}$$

$$\frac{\partial^2 P}{\partial^2 Y}=\frac{P(i,j+1)-2P(i,j)+(P(i,j-1)}{\Delta^2 Y}$$

$$\frac{\partial^2 P}{\partial^2 X}=\frac{P(i+1,j)-2P(i,j)+(P(i-1,j)}{\Delta^2 X}$$

In gas slider bearing Reynolds equation is non-linear hence for approximation trial and error method “Newton Raphson” is applicable, according to Newton Raphson

$$P_{new} = P_{old} - \frac{P(i,j)}{P'(i,j)}$$

Where $\frac{P(i,j)}{P'(i,j)} = \delta$,δ =relaxtation factor

4. Results and Discussion

As a non-linear partial differential equation, Reynolds equation for hydrodynamic gas lubricated Bearings are difficult to obtain exact analytical solution. It is simple and Convenient to transform Reynolds equation into linear partial differential one through Proper approximation and attain numerical solution by the means of FDM.

4.1 Gas Lubricated Slider Bearing

A numerical solution was obtained for this bearing with the following base values: L= 1.0, Ls = 0.75, W = 0.75, Hi = 1.25, U (bearing number) = 50. The computed load support is F = 1.291.

The graph is between load carrying capacity and red plot show reference paper [21] in which some approximation method is used and it make nonlinear equation into linear and present work is on without any approximation Reynolds equation which shows nonlinearity

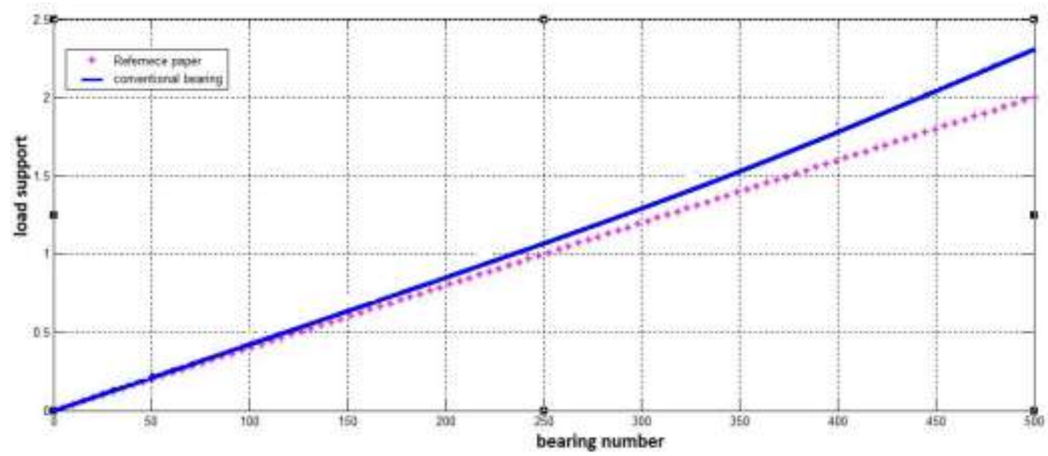

Chart -1: Load support vs. bearing number for simple slider.

Flow rate is also shown in below chart -2 it's also vary non linearly but some deviation is their because equation 3.1.5 is not containing any approximation hence its shows nonlinearity its shown by blue line and its comparative with reference work is shown below

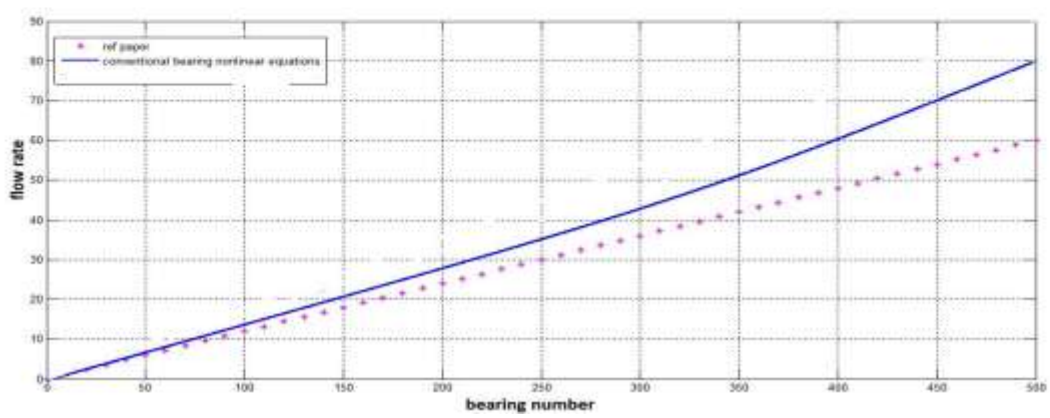

Chart -2: Flow rate of air with respect to bearing number

As shown in pattern with specification "a" is the distance between centre of slider and centre of recess. And the variation of load support with this distance is as shown below. The below graph (chart -3) show that

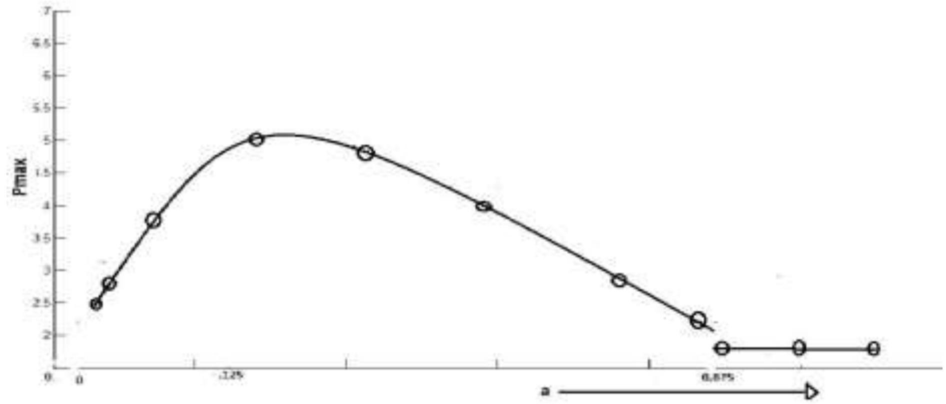

Chart -3: changing the max. value of pressure with changing "a"

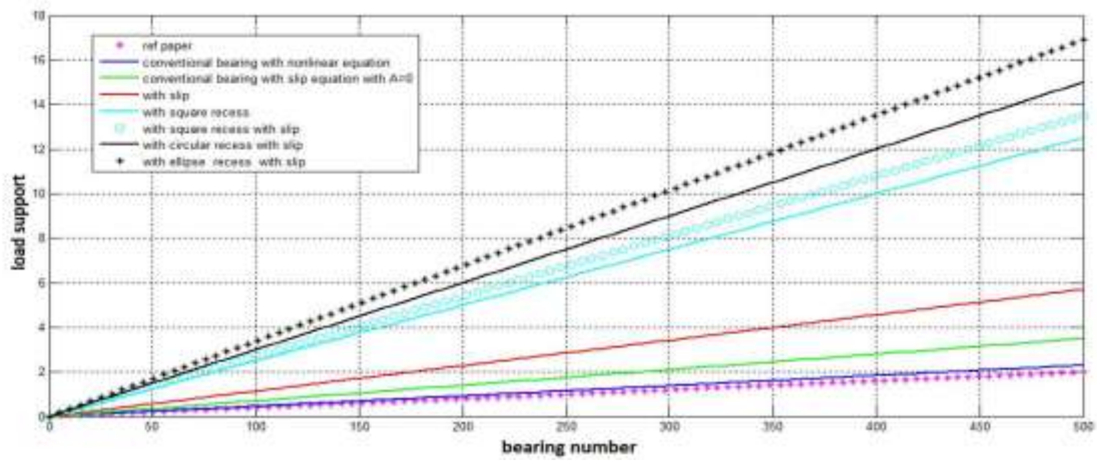

Chart -4: Correlate Load support for different values of bearing number for all patterns in slider.

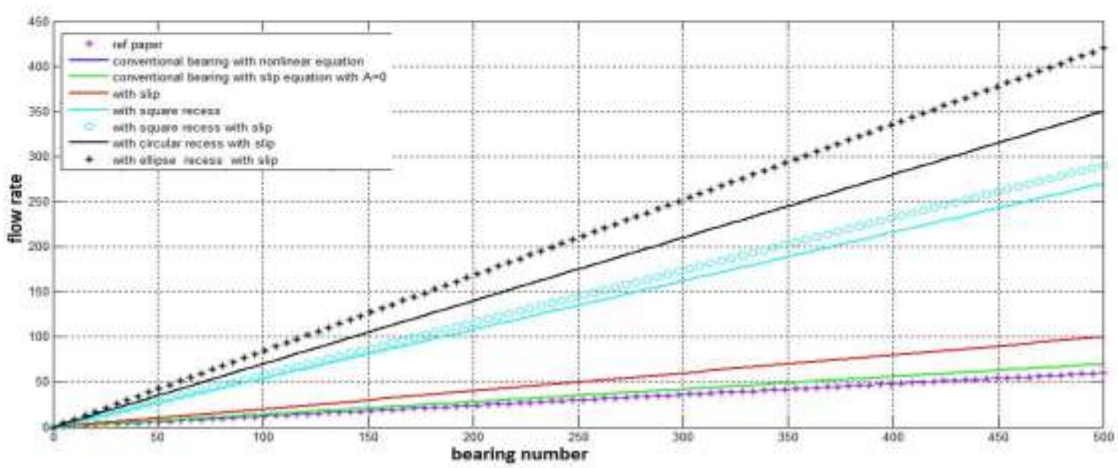

Chart -5: Correlate of flow rate for different values of bearing number for all patterns in slider.

5. Conclusions

Present works and final results shows that if slider with recess is used than it improve the load support and in place of slider for particular area when we use slip than also load support is increases but the combination of both that is slider having recess and slip on it is use than the load support of the slider is improved means areal density between the slider and stator is improved which helps to improve the performance of the slider.

Among all types of recess if we use elliptical recess which major axis is parallel to the length of the slider the load support is more, after that circle recess is also better compare to rectangular one.

References

[1] Pinkus, O. Analysis of simple slider bearings.-Trans. ASME, 1956, v.78, p.965-973.

[2] Pinkus, O. Solution of Reynolds equation for finite slider bearings.-Trans. ASME, 1958, v.80, p.858-864.

[3] Majumdar, B.C., Brewe, D.E. Stability of a rigid ro-tor supported on gas -film slider bearings under dy-namic load.-National Seminar on Bearings.-Madras, 1987, p.363-368.

[4] Danielyte, J., Gaidys, R. Numerical simulation of tooth mobility using non-linear model of the periodon-tal ligament. -Mechanika. -Kaunas: Technologija, 2008, Nr.3(71), p.20-26.

[5] Wang, J.S., Wang, C.C. Nonlinear dynamic and bifur-cation analysis of short aerodynamic slider bearings. -Tribol. Int., 2004, 38, p.740-748.

[6] Wang, C.C. Nonlinear dynamic and bifurcation analy-sis of rigid rotor supported by a relative short externally pressurized porous gas journal bearing system.-ACTA Mech., 2006, 183(1-2), 41-60.

[7] Wang, C.C., Jang, M.J. Bifurcation and nonlinear dynamic analysis of a flexible supported by relative short gas slider bearing.-Y.L.,Chaos Soliton Frac-tals, 2007, 32(2), p.566-582

[8] Wang, C.C., Jang, M.J. Application of a hybrid nu-merical method to the bifurcation analysis of a rigid ro-tor supported by a spherical gas journal bearing system. -Nonlinear Dynamics, 2007, 45, p.176-183

[9] Pinkus, O., Sternlicht, B. Theory of Hydrodynamic Lubrication. -London: Mcgraw-Hill Company, 1961, p.72-80.

[10] Szeri, A.Z. Tribology. -Washington: Hemisphere Publishing Corporation, 1980, p.172-182.

[11] Majumdar, B.C. Introduction to Tribology of Bearings. -Allhabad: Y P Chopra for A H Wheeler & Com-pany (P) Limited, 1986, p.112-119.

[12] Prasanta Sahoo. Engineering Tribology. -New Delhi Published by Prentice-Hall of India Private Limited, 2005, p.211-223.

[13] Himashu C Patel, Deheri, G.M. Characteristics of lubrication at nano scale on the performance of trans-versely rough slider bearing. -Mechanika. -Kaunas: Technologija, 2009, Nr.6(80), p.64-71.

[14] Disk Storage Technology; IBM Corporation; (1980)

[15] K. Kogure, T. Kita, S. I'ukui; Review of the Electrical Communication Laboratories, Vol 30, No. I, “Mechanical Characteristics of Flying Heads and Multi-Device Disk Storage”; pp. 36-45; (1982).

[16] K.Kogure, S.Fukui, Y.Mistaya, Kaneko; ‘Transactions of the ASME’, Vol. “Design of negative pressure slider for magnetic storage disk”(1992)

[17] I,-Y. Zhu, Bogy; -Tribology and Mechanics of Magnetic Storage Systems; Volume 5.''A Multi-Channel Laser interferometer and Its Use to Study Head-Disk Interface Dynamics in Magnetic Disk Drives"; pp. 168-179; (1994).

[18] H. Lee, R. flempstead, .J. Wciss; IEEE Transition on magnates', Vol. 25, No. 5; "Study of' Head and Disk Interface in Contact Start Stop Test"; pp. 3722-3724; (1989).