ANALYSIS OF STATIC AND DYNAMIC LOAD ON HYDROSTATIC BEARING WITH VARIABLE VISCOSITY AFFECTED BY THE ENVIRONMENTAL TEMPERATURE

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ABSTRACT
Hydrostatic bearing find wide application in machine tools with various technology because of their high stiffness and damping characteristic. With the advantage of new cutting materials, modern tool design is to increased speeds. For high speed applications it is necessary to have design data including the effect of rotational lubricant inertia. This research work contain the design data including the rotational effect have been obtained for annular recess conical bearing, multi recess pad and multi pad bearing with varying viscosity and pressure differences and the relationship with the environmental temperature. Expressions are obtained for the viscosity variation in an externally pressurized thrust bearing the condition when one bearing surface is rotated. The influence of centripetal acceleration and the combined effect of rotational and radial inertia terms are included in this analysis. Rotation of the bearing causes the lubricant to have a velocity component in an axial direction towards the rotating surface as it spirals radially outwards between the bearing surfaces.

INTRODUCTION
It is customary in the theory of lubrication to assume that the influence of the inertia terms in the equations of motion are negligible compared with the effect of the viscous term. When the lubricant gap and the flow rate are small, the pressure falls logarithmically in a radial direction towards the edge of the bearing, as indicated by the exact solution of the Navier Stokes equations. The author B.C. Majumdar ignoring the inertia terms and assuming negligible transverse velocity components. When the lubricant gap and flow rate are large, the viscosity variation is governed mainly by the inertia terms in the equations of motion and positive pressure gradients are encountered at the smaller radii and negative pressure gradients further out towards the edge of the bearing. Solutions for the condition of large lubricant gap and flow rate for stationary parallel discs are well known. T. Jayachandra Prabhu, investigation shown that the radial inertia of the lubricant can create a significant change in the viscosity variation, particularly when air is used as the lubricant, resulting in a loss in the load capacity of the bearing. In the practical applications of externally pressurized thrust bearings, it is usual for one of the bearing surfaces to rotate. As the speed of rotation increases, the centripetal inertia term can become dominant relative to the viscous term reducing the overall viscosity variation. T.A. Osman, Osterle and Dowson have investigated this particular case, deriving expressions for the viscosity variation for plane and stepped bearings. The same was revealed by Ashraf S. that the importance of both the radial and the rotational inertia terms in the equations of motion has recently increased due to extreme bearing
operating conditions. This paper presents an expression for the viscosity variation including both these effects.

Hydrostatic thrust bearings have been used in many industrial heavy applications due to their favorable performance characteristics. The author L. San Andrés includes the high load-carrying capacity, zero wear of bearing surfaces, low friction at low or zero speeds, large fluid film stiffness and damping, as shown in Fig.1 reduced vibrations and good positional accuracy. The following reports are important investigations concerning hydrostatic thrust bearings this research. Different bearing configurations were considered and it was reported that the addition of the stem enabled this bearing to support radial loads. Statish C. Sharma analytically studied the performance of circular thrust pad hydrostatic bearing of various recess shapes, i.e., triangular, square, rectangular etc. They compared results with the solutions obtained by an electrical analog technique.

Owing to rapid technological advancements in manufacturing techniques, super heavy constant flow hydrostatic thrust bearing having sector recess can be used widely in the many industrial heavy applications. In order to provide reasonable data for design, lubrication, thermal deformation and force deformation computation for hydrostatic thrust bearing in the heavy equipment, the author of previous study T Jayachadraprabhu reveals that the important to compute viscosity variation, temperature distribution and oil flow rate of the super heavy constant flow hydrostatic thrust bearing having sector recess.

Flat-land thrust bearings are the simplest and least expensive to make. They handle light loads for simple positioning of rotors in electric motors, appliances, crankshafts and other machinery. Flat-land bearings carry 10 to 20% the load of other thrust-bearing types. Antoon van Beek studies gives flat parallel surfaces do not directly build oil-film pressure through pumping action as shown in Fig.2. They depend instead on thermal expansion of both the oil film and bearing surface to generate an oil-supporting wedge.

Self-acting hydro dynamically lubricated slider bearings have, however certain important disadvantages:

1. If the design speed is low, it may not be possible to generate sufficient hydrodynamic pressure.
2. Fluid film lubrication may break down during starting, direction changing and stopping.
3. In a journal bearing the shaft runs eccentrically and the bearing location varies with load, thus implying low stiffness.

In hydrostatic also called externally pressurized lubricated bearings surfaces are separated by a fluid film maintained by a pressure source outside the bearing. Hydrostatic bearings avoid disadvantages 1 and 2 and reduce the variation of bearing location with load mentioned in disadvantage 3. The characteristics of hydrostatically lubricated bearings are:

1. Extremely low friction

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**Fig. 1**: Pad thickness (load and bearing)

**Fig. 2**: Frequency with thickness of pad

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![Diagram](image.png)
2. Extremely high load-carrying capacity at low speeds
3. High positional accuracy in high-speed, light-load applications
4. A lubrication system more complicated than that for self-acting bearings.

Therefore, hydrostatically lubricated bearings are used when the requirements are extreme as in large telescopes and radar tracking units, where extremely heavy loads and extremely low speeds are used, or in machine tools and gyroscopes, where extremely high speeds, light loads, and gas lubricants are used.

Fluid bearings use a thin layer of liquid or gas fluid between the bearing faces, typically sealed around or under the rotating shaft. There are two principal ways of getting the fluid into the bearing:
1. In fluid static, hydrostatic and many gas or air bearings, the fluid is pumped in through an orifice or through a porous material.
2. In fluid-dynamic bearings, the bearing rotation sucks the fluid on to the inner surface of the bearing, forming a lubricating wedge under or around the shaft.
3. Hydrostatic bearings on an external pump. The power required by that pump contributes to system energy loss just as bearing friction otherwise would. Better seals can reduce leak rates and pumping power, but may increase friction.

Hydrodynamic bearings rely on bearing motion to suck fluid into the bearing and may have high friction and short life at speeds lower than design or during starts and stops. An external pump or secondary bearing may be used for startup and shutdown to prevent damage to the hydrodynamic bearing. A secondary bearing may have high friction and short operating life, but good overall service life if bearing starts and stops are infrequent.

The thin films can be thought to have pressure and viscous forces acting on them. Because there is a difference in velocity there will be a difference in the surface traction vectors. Because of mass conservation we can also assume an increase in pressure, making the body forces different.

Fluid bearings can be relatively cheap compared to other bearings with a similar load rating. The bearing can be as simple as two smooth surfaces with seals to keep in the working fluid. In contrast, a conventional rolling-element bearing may require many high-precision rollers with complicated shapes. Hydrostatic and many gas bearings do have the complication and expense of external pumps.

**AIMS AND OBJECTIVES**

To analyse the viscosity variation of the lubricant used for the hydrostatic thrust bearing.
To analyse the load carrying capacity of the lubricant used for the hydrostatic thrust bearing.
To analyse the static and dynamic characteristic of various film thickness.
To analyse the temperature variation of the lubricant used for the hydrostatic thrust bearing.

**MATERIAL AND METHODS**

**Research analysis modeling**

The research deals with the designing of a complex hydrostatic bearing of self-acting type satisfying the requiring design. The pressure pads in hydrostatic bearings carry the load capacity. Most hydrostatic systems (thrust or journal bearings) use several evenly spaced pads so nonsymmetrical load distributions can be handled. To estimate performance, pads can be treated separately. Pressure is maintained in the recess by fluid pumped through a flow restrictor. The pressure lifts the rotor until flow out of the recess and over the land equals flow in.

A constant gap is maintained for a given recess pressure and bearing load. The gap establishes the volume of fluid pumped through the bearing. An alternative design is to connect a fluid displacement pump (gear or vane type) directly to each pocket without flow restriction. Pressurizing pump power can be estimated from the product of pad pressure and total flow through the pads. For an estimate of lubricant flow rate, a gap (or film thickness) is assumed typical values are 0.001 to 0.002 in.

Design of flow restrictors influences bearing stiffness, pumping power, supply pressure and lubricant flow. A flow restrictor is necessary to provide a pressure drop between the supply manifold and the pad recesses to ensure pressure requirements in any given pad never exceed supply pressure. A bearing with restrictors is called a
compensated bearing. Action of the restrictors is such that if the thrust load is centered, pressure in all the pads is equal and lower than pump pressure. If the load is off center, the gap decreases somewhat on the loaded side and flow from the pad also decreases. This results in an increase in loaded-pad pressure and a decrease in pad pressure on the opposite side. The runner automatically seeks a nearly level attitude.

The boundary conditions for slip flow at the surface of a gas bearing can be written as

\[ u_{slip} = \sigma \left( \frac{2 - f}{f} \right) \lambda \left( \frac{\partial u}{\partial z} \right)_{wall} \]  

(1)

Where \( f \) is the reflection coefficient, \( \lambda \) is the mean free path and \( \sigma \) is a numerical constant. Because \( \sigma \) and are close to unity it can be assumed that is unity. As the molecular mean free path depends upon fluid viscosity, pressure and temperature it can be approximated by the relation.

\[ \lambda = \frac{16}{5(2\pi)^{1/2}} \frac{\eta}{p} (RT)^{1/2} \]  

(2)

Where \( R \) is the gas constant, \( T \) the temperature of the gas, \( \eta \) the viscosity of the gas and \( p \) its pressure.

The effect of slip is also important on the flow behaviour of liquids especially.

When the bearing surface is very smooth and is operating at higher surface temperatures as shown in Fig. 3 where the viscosity of the base oil decreases near the surface. This effect has been studied for liquids. The slip velocity at the wall can be written as

\[ u_{slip} = \frac{1}{\beta} \left( \eta \frac{\partial u}{\partial z} \right)_{wall} \]  

(3)

where \( \beta \) is the coefficient of sliding friction at the wall and \( \eta \) is the liquid viscosity.

Reynolds equation

The physical configuration of fluid flow between two curved surfaces is shown in Fig. 1. The basic equations of motion and the equation of continuity for a Newtonian fluid considering the variation of fluid properties both across and along the film thickness can be written as

\[ \rho \frac{Du}{Dt} - \rho x \frac{\partial p}{\partial x} + 2 \frac{\partial}{\partial y} \left( \eta \frac{\partial u}{\partial y} \right) + \frac{2}{3} \frac{\partial}{\partial z} \left( \eta \frac{\partial w}{\partial z} \right) + \frac{\partial}{\partial x} \left( \eta \frac{\partial u}{\partial x} \right) \]  

(4)

\[ \rho \frac{Dw}{Dt} = \rho y \frac{\partial p}{\partial y} + 2 \frac{\partial}{\partial x} \left( \frac{\partial u}{\partial x} \right) \]  

(5)

\[ \rho \frac{Dw}{Dt} = \rho Z \frac{\partial p}{\partial z} + 2 \frac{\partial}{\partial z} \left( \frac{\partial u}{\partial z} \right) \]  

(6)

With the usual assumptions of lubrication theory equations, can be simplified to

\[ \frac{\partial p}{\partial x} = \frac{\partial}{\partial z} \left( \eta \frac{\partial u}{\partial z} \right) \]  

(7)

\[ \frac{\partial p}{\partial y} = \frac{\partial}{\partial z} \left( \eta \frac{\partial u}{\partial z} \right) \]  

(8)

where \( p = p(x, y) \) is the film pressure.

Equation represents a generalized form of Reynolds equation for compressible fluid film lubrication considering slip velocities at the bearing surfaces. The two sets of functions \( F \) and \( G \) depend upon the variation of fluid properties both along and across the film and on the slip conditions at the surfaces which is shown in Fig. 4.
Viscosity variation across the film

The viscosity of the lubricant can vary across the film thickness which is shown in Fig. 5 and may be different near the bearing surfaces owing to the reaction of additives and surfactants with the surfaces. The most general form of Reynolds equation to study such a situation is given by equation.

Considering a reasonable case where the density and viscosity of the lubricant near the bearing surfaces may be different from that of the central region gives the viscosity changes.

The pressure boundary condition necessary for the solution of eqn. in general are not satisfactorily known and hence a one dimensional analysis is carried out in section 2.2 to study qualitatively the effect of pseudo plasticity on piston ring lubrication.

**Research on lubrication performance of super heavy constant flow hydrostatic thrust bearing**

In order to increase rotational speed and bearing capacity of a constant flow hydrostatic thrust bearing, a theoretical study concerning lubrication
performance of a super heavy constant flow hydrostatic thrust bearing having sector recess is described. The computational fluid dynamics and the finite volume method have been used to compute the lubrication characteristics of a super heavy constant flow hydrostatic thrust bearing, such as recess pressure, recess temperature and oil flow rate. This study theoretically analyzes the influence of workbench rotational speed on the bearing lubrication performance according to computational fluid dynamics and lubricating theory. It has revealed its viscosity variation law, temperature distribution law and oil flow rate. Liquid hydrostatic thrust bearing working principle is that lubricating oil which is compulsively injected into oil cavity forms bearing capacity of hydrostatic bearing through throttling action of the gap between resistive oil edges and the rotary table, lifts bearing spindle, and bears external loads. The working principle of hydrostatic bearing with quantitative oil supply is shown as Fig. 1. Lubricating oil enters into oil cavity from pump along inlet and flows out along the radial shallow recess and resistive oil edges of external ring as shown in Fig. 1. Flow mobility of the fluid between HIP round rail and rotating worktable must meet mass conservation, momentum conservation and energy conservation.

**Mass conservation equation**
The law of mass conservation is the basic law which meets any mobile system. Mass conservation equation is:

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0
\]

Where \( \rho \) is density (kg/m\(^3\)), \( t \) is time(s) and direction component of speed vector.

**Energy conservation equation**
The law of energy conservation is the basic law which meets any mobile system. Energy conservation equation is:

\[
\frac{\partial (\rho T)}{\partial t} + \text{div} (\rho \text{ uT}) = \text{div} \left( \frac{k}{c_p} \text{ g rad} \right) + S_t
\]

Where \( S_t \) is heat source term. Flow capacity is 0.098 kg/s, external pressure is 0.1 Mpa, the worktable rotational speed of 2.5 rpm, 8 rpm, 10 rpm, 12.5 rpm, 16 rpm, 20 rpm, 25 rpm, 31.5 rpm and 40 rpm, the three-dimensional pressure fields, temperature fields and flow fields are shown in Fig. 6. The recess and obviously overlapping in the oil cavities, the flow state is from laminar flow to turbulent flow.

![Temperature variation curve](image)

**Fig. 6**: Temperature variation curve
RESULTS AND DISCUSSION

In this research after using the modified Reynolds equations assuming that there is a relation between the viscosity and film thickness and determine the flow, the load and the pumping power loss for bearing. The simulation model has established that there is a relationship between hydrostatic thrust bearing performance and various film thickness in recess pad. The simulation model has also proved that there is decrease in the viscosity variation due to increasing the viscosity of hydrostatic thrust bearing under the condition when one bearing surface is rotated. The simulation model has also proved that there is decrease in the viscosity variation due to increasing the viscosity of hydrostatic thrust bearing under the condition when both bearing surface is rotated. The simulation model has also proved that there is decrease in the temperature distribution due to increasing the viscosity of hydrostatic thrust bearing under the condition when one bearing surface is rotated. The simulation model has also proved that there is decrease in the temperature distribution due to increasing the viscosity of hydrostatic thrust bearing under the condition when both bearing surface is rotated. The hydrostatic thrust bearings to reduce the numbers of parts and size and to eliminate expensive mineral lubricant storage and pumping, thus further satisfying stringent environmental constraints. Despite the many advantages offered by hydrostatic bearings, rotor dynamic instabilities due to hydrodynamic (shear flow) and fluid compressibility effects are issues of primary concern for high speed operation with large pressure differentials.

Computed analyses are also available in this model for accounting flow turbulence, fluid inertia and compressibility and thermal effects to bearing designers and rotor dynamics engineers. Laboratory measurements of load, leakage, torque, and identification of rotor dynamic force coefficients aided to benchmark are compared with simulation model technique. The agreement also been tested.

CONCLUSION

The lubrication characteristics of a super heavy constant flow hydrostatic thrust bearing, such as recess pressure, recess temperature and oil flow state are computed according to the Computational Fluid Dynamics and the Finite Volume Method. This study theoretically analyzes the influence of workbench rotating velocity on the viscosity variation law, temperature distribution law and oil flow state. The following conclusions are made based on the numerical calculations done by iterative method for the two-dimensional oil film pressure field, temperature field and oil flow state of a super heavy constant flow hydrostatic thrust bearing by the usage of the fluid dynamics, lubricating theory and the Finite Volume Method. The results show that oil cavity pressure is almost invariant by increasing of workbench rotating velocity, oil cavity temperature is increasing gradually by increasing of workbench rotating velocity. Along with the work table rotational speed increasing, the turbulent flow in the oil cavities is more and more obvious. The environmental temperature around a power transmission system is equally important because it establishes the stabilized base temperature for the system.

Each type of bearing has unique geometric and manufactured features that influence the amount of friction that occurs during operation. This, combined with factors discussed previously about how an application can influence heat generation, underscores the fact that a power transmission system is a complex assembly that interacts with its surrounding environment. The more complex the system, the more important it is to assess the application characteristics, to select the proper bearing type and features and to optimize the lubricant type.

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