



# **A Study Analysis of Static Load Characteristics on Hydrostatic Bearing with Variable Pressure and Temperature**

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**ABSTRACT:** Hydrostatic bearings have an excellent static and dynamic behavior and are used for different kinds of application. Application of hydrostatic bearings is limited by friction and therewith by velocity. Typical characteristics of the hydrostatic system (load, stiffness, flow) are calculated without a velocity dependency. The geometry of the hydrostatic bearing pockets and their restrictors are optimized by using time continuous pressure distribution at the bearing pocket, laminar flow behavior as well as constant velocity of the bearing. The dynamic effects of the flow at high velocities are not considered. The paper reflects the common design and calculation methods and shows their limitations in regard to the calculation of hydrostatic bearings at high velocities. It analyzes the results of complex dynamic flow simulations of hydrostatic bearings and presents a new design and optimization concept of hydrostatic bearings. This concept analyses the oil flow at high bearing velocities and it optimizes the bearing geometry, the restrictor geometry as well as the geometry of the main mechanical components.

The parameters optimization design of a square hydrostatic bearing by using the HTGA/Gray method. We can find the performance simulation results of the proposed design are better than general hydrostatic bearing. Therefore, the proposed design have higher load capacity, higher stiffness and damping coefficients, lower flow rate and uniform pressure distribution by using the HTGA/Gray method. A Hydrostatic bearings test bench has been designed, built and set-up. The test bench has been monitored with pressure, flow-rate, temperature, displacement and force sensors.

Expressions are obtained for the temperature distribution in an externally pressurized thrust bearing for 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 the 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. This results in an increase in the pumping losses and a decrease in the load capacity of the bearing. A further loss in the performance of the bearing is found when the radial inertia term in addition to the rotational inertia term is included in the analysis.

## **I. INTRODUCTION**

The load capacity and direct stiffness of hydrostatic bearings do not depend on fluid pressure, thus making them ideal rotor support elements in process fluid pumps. Current applications intend to replace oil lubricated bearing with hybrid bearings to improve efficiency with shorten rotor spans and less mechanical complexity. Current cryogenic liquid turbopumps implement hydrostatic bearings enabling an all fluid film bearing technology with very low number of parts and no DN limit operation. Details on the bulk-flow analysis of turbulent flow hydrostatic bearings are given along with the discussion of performance characteristics, static and dynamic, for hydrostatic bearings supporting a water pump. Angled liquid injection produces a hydrostatic bearing with unsurpassed dynamic force and stability characteristics.

In a hydrostatic bearing an external source of pressurized fluid forces lubricant between two surfaces, thus enabling non-contacting operation and the ability to support a load. Hydrostatic bearings can support large loads without journal rotation and provide large (accurate and controllable) direct stiffness as well as damping (energy dissipation)

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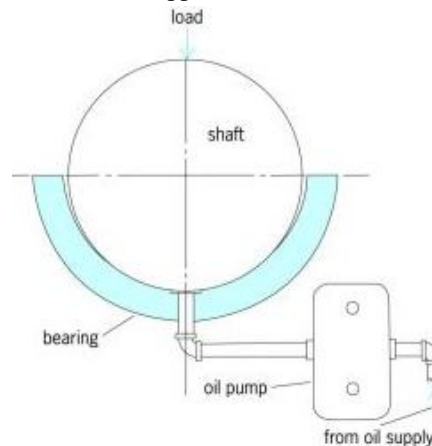
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coefficients. Hydrostatic bearings rely on external fluid pressurization to generate load support and a large centering stiffness, even in the absence of journal rotation.

Hydrostatic bearings derive their load capacity not from shear flow driven effects (hydrodynamic wedge and surface sliding) but rather from the combination of temperature versus flow resistance effects through a feed restrictor and in the film lands. Figure 1 depicts thrust and radial hydrostatic bearing configurations for process fluid lubrication turbopumps. Table 1 presents the major advantages and disadvantages of hydrostatic bearings.

The hydrostatic stiffness is of unique importance for the centering of high-precision milling machines, gyroscopes, large arena movable seating areas, telescope bearings, and even cryogenic fluid turbo pumps for rocket engines. Note that hydrostatic bearings require an external pressurized supply system and some type of flow restrictor. Also, under dynamic motions, hydrostatic bearings may display a pneumatic hammer effect due to fluid compressibility. However, and most importantly, the load and static stiffness of a hydrostatic bearing are independent of fluid pressure; thus making this bearing type very attractive for application with non-viscous fluids, including gases and cryogenes.



substantially low friction, high stiffness, high accuracy, and long service life etc. However, the performance of this type of bearing is greatly affected by the recess shape during the design process and the type of restrictor. A through scan of the literature concerning the hydrostatic bearing indicates that the majority of the studies are a rectangular shape of the recess. This may be because of ease of manufacturing. However, owing to rapid technological advancements in manufacturing techniques, the other recess shapes can now be easily manufactured.

## II. MODELING OF A HYDORSTATIC BEARING

The principle of a hydrostatic bearing is shown in Fig. 1. The hydrostatic bearing working principle is that liquid at a constant supply pressure  $S P$  is pumped into the bearing [1-3]. The lubricating oil is compulsively injected into the recess forms bearing capacity of hydrostatic bearing through throttling action of the gap between bearing pad and worktable. Thus, the pressure inside the recess of the bearing pad is constant and declines to the pressure along the bearing land. The resulting pressure distribution and the external load stay in equilibrium. Hydrostatic bearings test bench has been designed and set up to study the grinding machines wheelhead and to validate the theoretical models. The demonstrator, figure 1, has three hydrostatic bearings: two radials (A and B) and one axial (C).

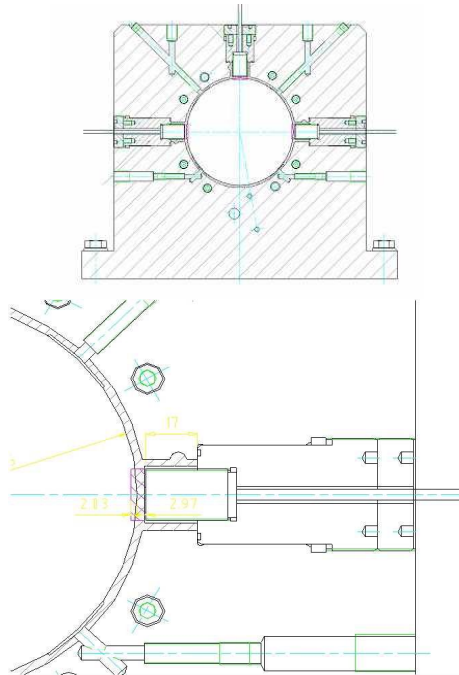
### Mathematical modeling

Some basic assumptions are applied in this study before deriving the equation of the hydrostatic bearing. Consider a steady, we assume that the fluid properties are isotropic, incompressible and isoviscous [4-6]. The governing equations for this problem can be written as follows: where  $p$  is the pressure distribution of the bearing,  $\mu$  is the pressure of lubricating,  $u$ ,  $v$  and  $w$  are three component of the velocity vector respectively along  $x$ ,  $y$  and  $z$  axes. Eqs. (1) may be integrated twice and evaluated to determine the velocity distribution. The boundary conditions employed are  $u = 0$  and  $v = 0$  at  $z = 0$  or  $h$ . The velocity distribution is given by

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## Hydrostatic bearings' theoretical study

Bearings lubrication theory is based in the formulation published by O. Reynolds in 1886 [4]. This equation result applying movement quantity conservation and continuity equation to the volume control shown in the figure 9. The difference in pressure between the upper and lower pads of the bearing is:

$$\Delta P = P_u - P_l = P_s \left( \frac{R_u}{R + R_u} - \frac{R_l}{R + R_l} \right)$$

- For a nominal gap  $h$  and small excursions  $\delta$  of the structure:

$$R_u = \frac{\gamma}{(h - \delta)^3} \quad R_l = \frac{\gamma}{(h + \delta)^3}$$

$$\Delta P = P_s \gamma \left[ \frac{1}{R(h - \delta)^3 + \gamma} - \frac{1}{R(h + \delta)^3 + \gamma} \right]$$

The difference in pressure across the bearing is: If the inlet flow resistance  $R$  was zero, the bearing could support no load [7-9]. Effective is the effective bearing area. At maximum load capacity, the bearing stiffness is high. In a hydrostatic bearing an external source of pressurized fluid forces lubricant between two surfaces, thus enabling non-contacting operation and the ability to support a load. Hydrostatic bearings can support large loads without journal rotation and provide large (accurate and controllable) direct stiffness as well as damping (energy dissipation) coefficients [10-12].

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## 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

$$\rho \frac{Du}{Dt} = \rho X - \frac{\partial p}{\partial x} + \frac{2}{3} \frac{\partial}{\partial x} \left\{ \eta \left( \frac{\partial u}{\partial x} - \frac{\partial v}{\partial y} \right) \right\} + \frac{2}{3} \frac{\partial}{\partial x} \left\{ \eta \left( \frac{\partial u}{\partial x} - \frac{\partial w}{\partial z} \right) \right\} + \frac{\partial}{\partial y} \left\{ \eta \left( \frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \right) \right\} + \frac{\partial}{\partial z} \left\{ \eta \left( \frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right) \right\} \quad \text{---(5)}$$

$$\rho \frac{Dv}{Dt} = \rho Y - \frac{\partial p}{\partial y} + \frac{2}{3} \frac{\partial}{\partial y} \left\{ \eta \left( \frac{\partial v}{\partial y} - \frac{\partial u}{\partial x} \right) \right\} + \frac{2}{3} \frac{\partial}{\partial y} \left\{ \eta \left( \frac{\partial v}{\partial y} - \frac{\partial w}{\partial z} \right) \right\} + \frac{\partial}{\partial z} \left\{ \eta \left( \frac{\partial w}{\partial y} + \frac{\partial v}{\partial z} \right) \right\} + \frac{\partial}{\partial x} \left\{ \eta \left( \frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \right) \right\} \quad \text{---(6)}$$

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

## III. SIMULATION RESULTS

In this section, the optimization approach for HTGA/Gray method is used to determine suitable parameters. The simulation results under constant parameters setting with 25 bar of supply pressure and  $25 \mu\text{m}$  of film thickness as shown in Table 2 and Fig. 6. From Fig. 6, we can find the uniform pressure distribution of a square hydrostatic bearing via HTGA/Gray method [13-15].

The main performance characteristics of interest in the present study are load capacity, stiffness and damping coefficients and flow rate. The performance characteristics of different supply pressures and film thicknesses as shown in Fig. 7. The following sections compare the performance characteristics of the proposed design with general hydrostatic bearing having recesses of different shapes [7]. Fig. 8 shows the recess shape of a general hydrostatic bearing. Comparisons were made on the basis of the same supply pressure, film thickness and bearing size.

Therefore, we can obtain the better performance of a bearing by using the HTGA/Gray method. Fig. 7 (a) The load capacities of the different supply pressures and film thicknesses. (b) The stiffness coefficient of the different supply pressures and film thicknesses. (c) The flow rate of the different supply pressure and film thicknesses. (d) The damping coefficient of the different supply pressures and film thicknesses. A pressure distribution can observe into bearing's small clearance surface, while into the recesses (pockets of the bearings) the pressure keeps constant. Analysing the pressure drop in lateral clearance, a linear distribution of pressure can be observed.

## IV. RESULTS DISCUSSION

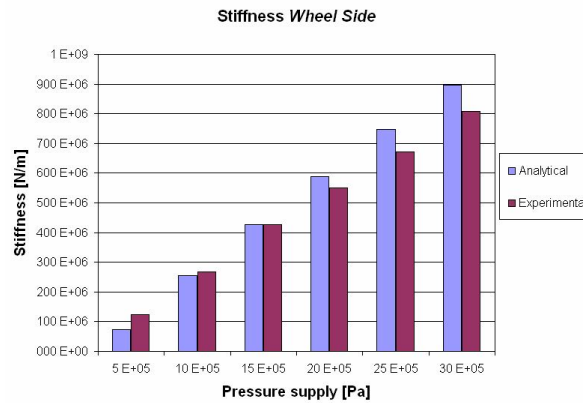
Hydrostatic bearings' test bench has been experimented, and its results are verified with analytical model. On the other hand, these results are compared with numerical model to validate the capability of this tool to design new lubricated bearings. Table 1 shows the lubricant temperature increase when the bearing is working at 3 MPa. These results have been measured once the bearing temperature has become stabilized [16-18]. This fact happens approximately 20 minutes

after the bearing starts to run. The differences between experimental and numeric results are due to fluid temperature changes, and therefore pressure variations

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In figure 13 can be observed the force applied in the bearing and the displacement of the shaft. The relation of both is defined as experimental hydrostatic stiffness. These tests are done for different supply pressures [19-22]. The stiffness is compared with analytical results, and all these results are shown in figure 14.

## V. CONCLUSION

The conclusions of this study are as follows:

- (1) In the mathematic model, we were simulated by applying finite difference method to analysis the pressure distribution of a square hydrostatic bearing, then from pressure distribution to calculate the performance characteristic of the proposed design such as load capacity, stiffness and damping coefficients and flow rate.
- (2) We proposed a new optimization approach, which integrated different method included HTGA and RGA; it is called the HTGA/Gray method. This method can be more robust, stability, and quickly convergent. Furthermore, it can consider multiple quality characteristic in optimization problems.
- (3) The parameters optimization design of a square hydrostatic bearing by using the HTGA/Gray method. We can found the performance simulation results of the proposed design are better than general hydrostatic bearing. Therefore, the proposed design have higher load capacity, higher stiffness and damping coefficients, lower flow rate and uniform pressure distribution by using the HTGA/Gray method.

A Hydrostatic bearings test bench has been designed, built and set-up. The test bench has been monitored with pressure, flow-rate, temperature, displacement and force sensors. Based on analytical models of bibliography (Bassani, 1992), it has been developed a software to calculate the performance of the bearing. This program is validated with experimental data of test bench. Numerical model of lubricated journal bearing has been developed. The results of this model are quite according to analytical and experimental results, but the difficulty to fit the pressure and clearance to the working conditions, limits the use of this tool to design new, and more sophisticated hydrostatic bearings.

Once has been analyzed the lubrication of conventional hydrostatic bearings, the next step in this work is to the develop an all-new active hydrostatic bearing. The main objective of this bearing is to increase de work range of actual bearings, such the maximum rotating speed as the load capacity, stiffness and dampness. After observe the most influent variables of the bearing, the pressure become as fundamental factor because it limits the maximum speed (friction power) and gives stiffness and dampness. This way, the control of active bearing through ferrofluids is a very promising field of development.

## REFERENCES

1. O'Donoghue JP, Rowe WB and Hooke CJ. Design of Hydrostatic Using An Operating Parameter. Wear, Vol. 14, pp. 355-362, 1969.
2. Lydia Caroline, M., Kandasamy, A., Mohan, R., Vasudevan, S., "Growth and characterization of dichlorobis 1-proline Zn(II): A semiorganic nonlinear optical single crystal", Journal of Crystal Growth, ISSN : 0022-0248, 311(4) (2009) pp. 1161-1165.
3. Rowe WB, O'Donoghue JP and Cameron A. Optimization of Externally Pressurized Bearings for Minimum Power and Low Temperature Rise. Tribology, Vol. 3, No. 3, pp. 153-157, 1970.



# International Journal of Innovative Research in Computer and Communication Engineering

(An ISO 3297: 2007 Certified Organization)

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4. Nagarajan, C., Madheswaran, M., "Experimental study and steady state stability analysis of CLL-T series parallel resonant converter with fuzzy controller using state space analysis", Iranian Journal of Electrical and Electronic Engineering, ISSN : 1735-2827, 8(3) (2012) pp.259-267.
5. Ghosh MK. and Majumdar BC. Design of multirecess hydrostatic oil journal bearings, Tribology International. Vol. 13, No. 2, pp. 73-78, 1980.
6. Franchek NM, Childs DW. Experimental test results for four high-speed, high-pressure, orifice-compensated hybrid bearings. ASME J Trib 1994; 116(1):147-53.
7. Sharma SC, Jain SC and Bharuka DK. Influence of Recess Shape on the Performance of a Capillary Compensated Circular Thrust Pad Hydrostatic Bearing. Tribology International, Vol. 35, pp. 347-365, 2002.
8. Parthasarathy R., Ilavarasan R., Karrunakaran C.M., "Antidiabetic activity of Thespesia Populnea bark and leaf extract against streptozotocin induced diabetic rats", International Journal of PharmTech Research, ISSN : 0974-4290, 1(4) (2009) pp. 1069-1072.
9. Sharma SC, Sinhasan R, Jain SC. Performance characteristics of multirecess hydrostatic/hybrid flexible journal bearing with membrane type variable-flow restrictor as compensating elements. Wear, 1992;152:279-300.
10. Singh N, Sharma SC, Jain SC, Reddy SS. Performance of membrane compensated multirecess hydrostatic/hybrid flexible journal bearing system considering various recess shapes. Tribol Int 2004;37:11-24.
11. Abel-Keilhack C., Hesselbach J. Hydrostatic bearings with magnetic fluids. *Production Engineering* 9 (2) pp. 179-184, 2004
12. Bassani R., Piccigallo B. Hydrostatic Lubrication. *Elsevier*, Tribology Series 22, 1992
13. Poongothai S., Ilavarasan R., Karrunakaran C.M., "Simultaneous and accurate determination of vitamins B1, B6, B12 and alpha-lipoic acid in multivitamin capsule by reverse-phase high performance liquid chromatographic method", International Journal of Pharmacy and Pharmaceutical Sciences, ISSN : 0975 - 1491, 2(S4) (2010) pp. 133-139.
14. Chen C.C., Kang Y., Huang Y.N. The Influences of Orifice Restriction and Journal Eccentricity on the Stability of the Rigid Rotor-Hybrid Bearing System. Tribology International 37(2004) pp. 227-234
15. Frene J., Nicolas D., Degueurce B., Berthe D., Godet M. Hydrodynamic Lubrication. *Elsevier*, Tribology Series 33, 1990
16. Kulanthaivel L., Srinivasan P., Shanmugam V., Periyasamy B.M., "Therapeutic efficacy of kaempferol against AFB1 induced experimental hepatocarcinogenesis with reference to lipid peroxidation, antioxidants and biotransformation enzymes", Biomedicine and Preventive Nutrition, ISSN : 2210-5239, 2(4) (2012) pp.252-259.
17. Hernandez J., Uribe-Etxeberria R., de Pedro I. Simulation and experimental analysis of fluid-lubricated bearing monitoring strategies. *36th CIRP International seminar on manufacturing systems*, 03-05 June 2003 Saarbrücken
18. Mohsin M. E., Morsi S. A. The Dynamic Stiffness of Controlled Hydrostatic Bearings. Journal of Lubrication Technology, 91 pp. 597-608, 1969
19. Nikolakopoulos P. G., Papadopoulos C. A. Controllable high speed journal bearings, lubricated with electrorheological fluids. *Tribology International* 31 (5) pp. 225, 1998
20. Rowe W. Hydrostatic and hybrid bearing design. *Butterworth* 1983
21. Santos I. F., Wantanabe F. Y. Feasibility of influence the dynamic fluid film coefficients of a multirecess journal bearing by means of active hybrid lubrication. Journal of Brazilian Society of Mech. Scinc. & Eng. V25 n2, 2003
22. Slocum A. H. Precision Machine Design. Prentice Hall, 1992.