Test Setup for Magnetohydrodynamic Journal Bearing

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ABSTRACT

Journal bearings are well known for their lowest wear rate and high damping coefficients under fully developed hydrodynamic lubrication mechanism. However, metal-metal contact at start/stop and low speeds, causing bearing wear and excessive power loss, are major disadvantages of hydrodynamic bearing. Therefore, there is a need to augment the hydrodynamic bearing for satisfactory operation at start/stop and low speed operations. In this paper, a concept of hydrodynamic-permanent-magnetic hybrid journal bearing has been introduced. Concepts of repulsive type passive magnetic levitation, which has an advantage of having reducing starting torque at low speed, and hydrodynamic lubrication mechanism, which has an advantage of low friction at medium and high speed, are used in single bearing arrangement. A parametric study is carried out using MAXWELL software to determine the dimensions of permanent magnets. In addition, current paper presents a design of an experimental setup to investigate the performance characteristics of proposed hybrid type bearing.

1 INTRODUCTION

Life of well-designed machine components, in relative motion, is decided by wear rate occurred in running conditions. On comparing the data related to wear coefficients collected under different mechanical contact conditions, as shown in Table-1 [1], one finds lowest wear rate in the hydrodynamic mechanism. The lowest value of friction coefficient under fully developed hydrodynamic conditions, as shown in Table-2 [2], is an additional advantage of hydrodynamic journal bearing. Rolling contact bearing, a competitor of hydrodynamic bearings, has a finite life for a given load and speed. Further low damping, noisy operations, and high contact stresses make rolling element bearings inferior to hydrodynamic journal bearing. High values of load and speed cause significant reduction in rolling-bearing life. However, the major disadvantage of hydrodynamic bearing is its inability to make hydrodynamic-lubricant-film at start, stop and low speed operations, which brings journal bearing under boundary lubrication conditions. Table-2 indicates a reduction in the coefficient of friction from 0.25 (during start/stop operation) to 0.001 (under fully developed hydrodynamic lubrication). In other words, if a system of rotor bearing system requires X horse power for normal operation, then it needs 250X horsepower during start-up operation. This requirement of huge horsepower just for starting purpose is objectionable in the present competitive world, where minimization of power loss, weight and cost are main objective functions. Therefore, there is a need to augment the hydrodynamic bearing for satisfactory operation at low speeds.

As an alternative, hydrostatic bearing is the best choice for its minimum start-up friction (0.0001), high stiffness and high load carrying capacity at low speed. Hydrostatic bearings, however, have the disadvantage of requiring complex lubrication system consisting of high pressure subsystem, filtering subsystem, feedback control system, and complex geometry. Such a system requires high initial and running costs, and occupies space. Another alternative is an active magnetic bearing that provides lubricant free, adjustable damping and stiffness characteristics in all directions, and allows high-speed relative motion. However, this magnetic bearing incurs high cost, has complicated structure, and relatively low (5-15 %) capacity compared to hydrodynamic bearing.

Table-1: Wear Rate [1]

Type	Hydrodynamic	EHD	Boundary lubricated	Unlubricated
Range of wear coefficient	$<10^{-13}$	10^{-9} -10^{-13}	10^{-6} -10^{-10}	10^{-2} -10^{-7}

Table-2: Different Friction Coefficients [2]

Type	Hydrodynamic Bearing	Dry-Bearing	Rolling Element Bearing
Start-up friction Coefficient	0.25	0.15	0.05
Running friction Coefficient	0.001	0.1	0.05-0.001

One can think of hybrid bearings such as: hydrodynamic-hydrostatic, hydrodynamic-active-magnetic or hydrodynamic-permanent-magnetic bearings. Hydrodynamic-hydrostatic bearing is an old concept and has been used in almost every journal-bearing configuration. Generally hydrodynamic bearing is made with a lubricant supply groove, and the lubricant is fed at a pressure above the atmospheric pressure. Level of supply pressure decides the hydrostatic action. However, in many cases, supply pressure pump is linked with shaft (such as in internal combustion engines), supported on the bearing. Therefore, supply pressure cannot be achieved at the start/stop of shaft-rotation. So, first two hybrid concepts (hydrodynamic-hydrostatic and hydrodynamic-active-magnetic), which are complicated in structure, and having expensive hydraulic systems, sensors, power amplifiers and controllers, cannot be justified for ordinary industrial applications. Therefore, the best solution is the use of repulsive type passive magnetic levitation in the hydrodynamic bearing to extract the advantages of magnetic bearing at low speeds for reducing the start-up friction and the advantages of hydrodynamic bearing at high speeds which will be attractive in commercial applications for their low cost, simplicity in structure and no metal-to-metal contact. The main aim of this work is to design a hydrodynamic-permanent-magnetic hybrid journal bearing and an experimental setup to investigate the performance characteristics of this hybrid type bearing.

Recently Q. Tan, et al [3] developed a magnetic-hydrodynamic hybrid thrust bearing. The force of the bearing comes from the both hydrodynamic film and magnetic field. They have studied a model of the bearing force and develop an experimental rig to investigate the thrust bearing performance.

In the same year (2002), Q. Tan, et al [4] developed a permanent magnetic-hydrodynamic hybrid journal bearing. In their setup, permanent magnetic provides rotor-support during starting and stopping a machine.

In setups developed by Q. Tan et al [3-4], side-by-side magnetic and journal bearings were used, which occupy more space and make the system complicated in structure. Further, during every start and stop, the outer permanent magnetic ring needs to be mechanically shifted to create the magnetic levitation force. Such an adjustment might results misalignment due to mismatch in the radial shift of magnet on both sides of the shaft. Moreover, cylindrical shaped permanent magnet, used by Q. Tan et al [3-4] produces a lower radial force and demands unnecessary larger bearing dimensions.

In view of shortcomings in the available literature, a design concept of hybrid bearing which is simpler in structure, better in the performance and economic in cost is proposed. Repulsion between permanent magnets and hydrodynamic action between tribo-pairs are clubbed together in single bearing geometry.

In order to design the permanent magnet bearing, first step is to select the bearing configuration. In the present paper, a repulsive type axial permanent magnet configuration has been selected. Finite element analysis has been carried out using MAXWELL software to determine the proper dimensions of permanent magnet.

2 MODELING OF PERMANENT MAGNET AND SIMULATION

To determine the optimum dimensions of permanent magnet for a given force, a parametric study has been carried out for better understanding. Required force is modeled in the geometric and magnetic parameters.

2.1 Simulation

It is seen from the literature [4] that the levitation force can vary with various parameters, such as magnet thickness, air-gap (eccentricity), arc and length. For this reason the simulations have been carried out for different magnet lengths, magnet thickness, different arc lengths and different air gaps. The lower part of the stator has been kept at 180^0 and upper part of the stator (arc) has been varied to get optimum arc length.

Figure 1 shows that, for a particular thickness (say 6mm) and air gap (25 micron), radial force increases with increase in length of the magnet. Magnetic force decreases with increase in arc length from zero to 180 deg. From the Fig-2, it is seen that, for a particular length (say 15mm) and air gap (25 micron), radial force increases with increase in thickness of the magnet.

The stiffness increases with increase in eccentricity, i.e. with decreasing the air gap for a particular length (15 mm) and thickness (6 mm) as shown in Fig 3. Again, stiffness increases with increase in arc length. This increment of stiffness with arc length is predominant from zero deg. Arc to 90 deg. Arc. Increment of stiffness after that is not so prominent.

From the results of Fig-1 and Fig-2, it is difficult to fix the length and thickness for getting particular load requirement of 100N. Therefore, first the length has been fixed at15mm. Now both the Fig-1 and Fig-2 encourage to use no arc. Because, without arc it shows the maximum radial force, means lower cost and simpler in structure. But in magnetic bearing, stiffness is a vital parameter from stability point of view and one can not neglect that. Fig-3 shows that from no arc to 90 deg., arc stiffness increases predominantly and after that the increment is negligible. Again, Fig-1 and Fig-3 show that radial force decreases with increases in arc length. So there is a trade-off between radial force and stiffness. Therefore, we have taken a 45-degree arc to get the average effect of stiffness (50% of peak value at 90 degree) and keep the radial force at higher value. From the Fig-2, it is seen that at 6mm thickness, magnetic force is coming around 110 N at 25-micron air gap. Dimensions of the magnetic rings have been shown in Table-3

3 DESCRIPTION OF SET UP

The set up as shown in Fig-4 is made up from following different assembly and subassembly.

3.1 Elements of the Setup

This setup consists of bearing assembly, shaft, rotor, loading assembly, lubrication system, proximity sensors and motor. The functions of the each part and its justifications have been given in following paragraphs.

a) Bearing Assembly: Here magnets of NdFeB material have been used as a bushing for journal bearing for its highest energy product and high coercive force among all existing permanent magnets.Cylindrical magnets of same material have been mounted on the shaft. Journal bushing has been made with these magnets and aluminum ring. Details have been shown in Fig-a. Diagram of housing has been shown in Fig-5. Dimensions of the magnets have been tabulated on the Table-3.

b) Shaft: The shaft material is of Stainless Steel (SS201) which is non-magnetic. Dimensions are shown in Fig-8. Length and diameter of the shaft have been chosen on the basis of strength (379 N/mm^2) of the shaft material, required torque (15 N-m) and applied load around 200N. One sleeve, as shown in Fig-6a of another material (AISI4140), has been mounted on the shaft as a target material for the proximity sensors.

c) Loading Assembly: A lever arrangement as shown in Fig-7 is provided with a lever ratio of 1: 2 to get a higher output load on the rotor than input load at loading pan during running condition when hydrodynamic force is generated. At one end of the lever a loading pan as shown in Fig-7a is attached on which the weights can be placed from out side. To the other end a ball bearing is mounted through which load is applied on the rotor as a point load.

d) Lubrication system: The oil is supplied to the bearing clearance gap by gravity feeding through the oil hole located at 40^0 from load line as shown in Fig-a by means of a flexible pipe from the oil tank.

Here position of the oil inlet has been chosen on the basis of applied load (200N), speed **(6000 rpm),** clearance (25 micron) to keep the power loss and temperature rise to minimum value. The outlet **of the oil** is shown in Fig-b.

e) Proximity Sensors: Non-contact type eddy current proximity sensors have been used to measure the film thickness of the bearing. Two sensors have been mounted on the shaft by a rectangular block at 90^0 apart corresponding to X and Y coordinate axis as shown in Fig-4.

g) Motor: The system is driven by the motor. Motor is of 1HP 3-phase 3000 rpm standard motor suitable for supply through variable frequency drive.

3.2 Principle of operation

Instead of metal bushing in journal bearing, the powerful permanent magnets NdFeB have been put on the journal bearing housing and also a cylindrical magnet of same material has been placed on the shaft in the position of the ring magnets as shown in Fig-6a. All magnets have been magnetized in the axial direction. These like-pole permanent magnets generate repulsive force in the radial direction and also produce some axial unbalance force.

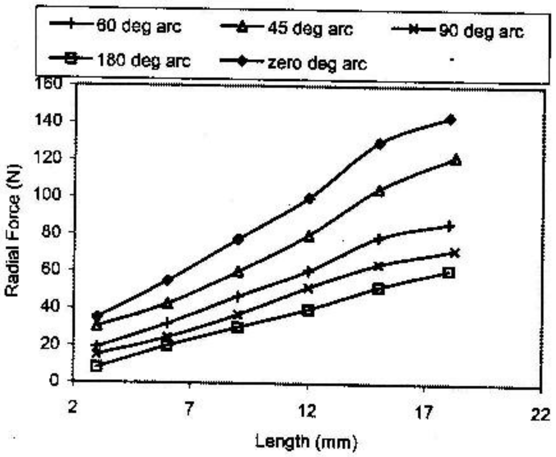


Fig-1 Variation of radial force with magnet length for different arc length and for 6mm thickness and air gap 25 micron

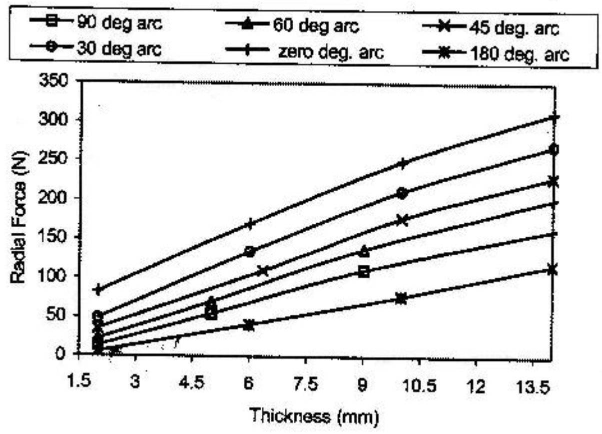


Fig-2 Variation of radial force with thickness for different arc length keeping 15 mm andair gap 25 micron fixed

Unless this axial force is supported by mechanical or servo control electromagnetic force, the system will not be stable, which is mathematically proved by Earnshaw [5]. In this model thrust bearings have been used for axial support as shown in Fig-4.

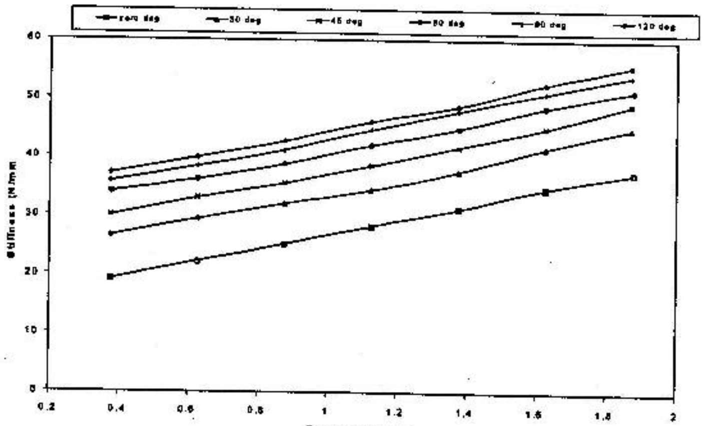
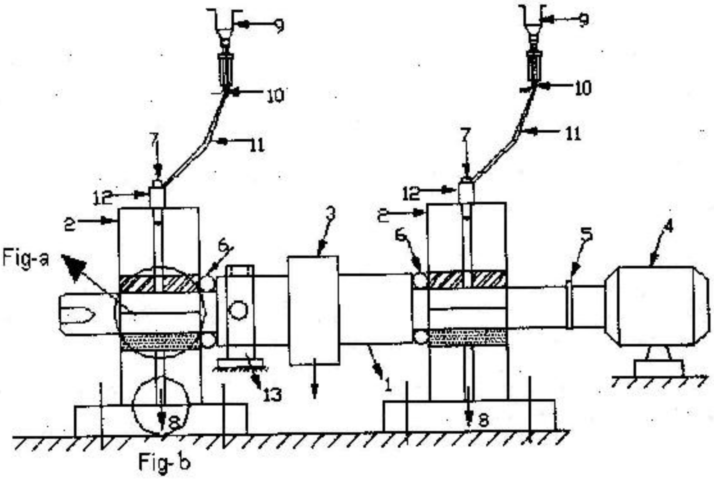


Fig-3 Variation of stiffness with eccentricity for different arc length keeping magnet length15mm and thickness 6 mm fixed.

When a hydrodynamic film is not formed, during starting and stopping a machine, the journal bearing relies on the magnetic force which prevents the start-up friction and reduces the huge power loss



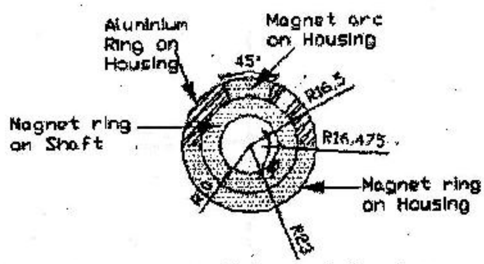


Fig-a: Arrangement of Magnets in Bearing

Fig-4: SCHEMATIC DIAGRAM OF EXPERIMENTAL SETUP OF HYBRID BEARING.

1. Shaft, 2.Housing, 3.Rotor, 4.Motor, 5.Flexible Coupling (Love & Jaw), 6. Thrust Bearing, 7.Inlet of lubricant, 8.Outlet of lubricant, 9.Oil Tank with Filter at Bottom, 10.Brass Ball Valve, 11.Flexible Pipe, 12.Nozzle, and 13.Stand for sensors

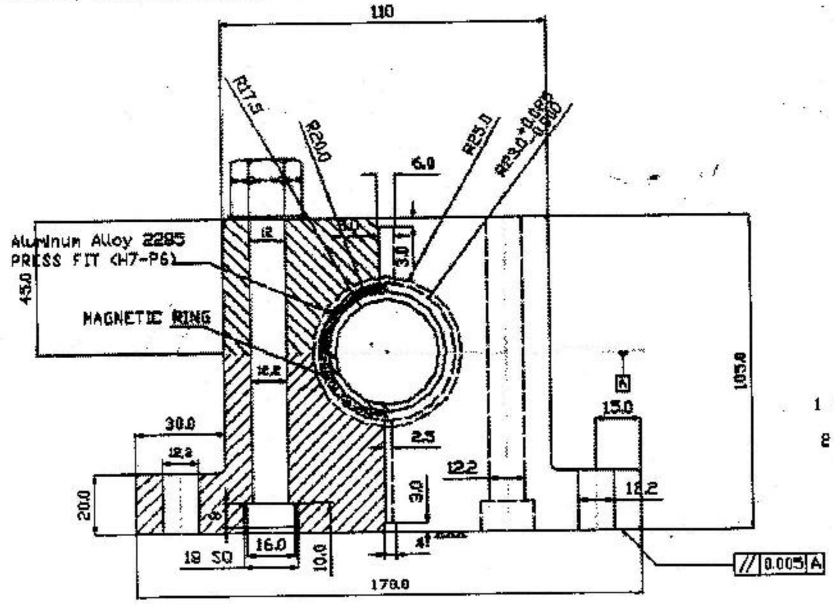


Fig-5 Housing (Font View); Matl.- EN3A

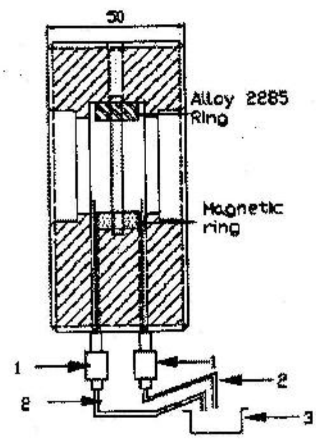
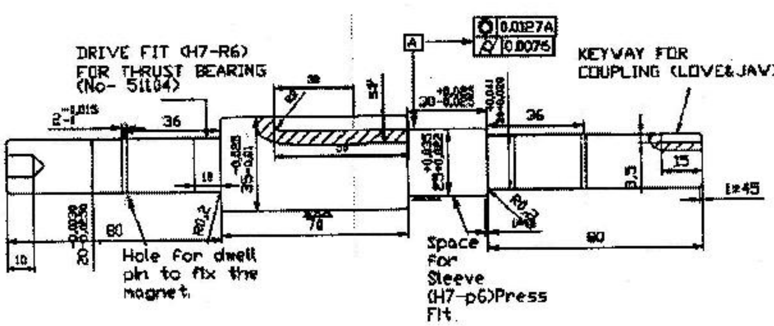


Fig-b Outlet of Oil (1. Nozzle, 2. Flexible Pipe, 3. Tank)



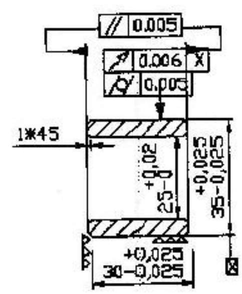


Fig-6 Shaft; Matl.- Annealed Standard Austenitic Stainless Steel AISI Type SS201

Fig-6a Sleeve Matl.- AISI 4140

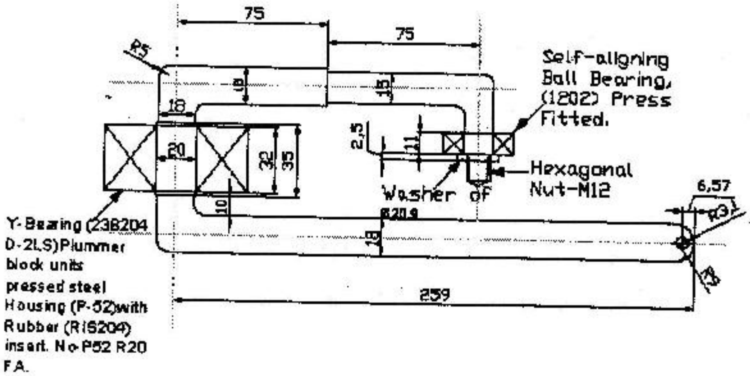


Fig-7 Lever for load application

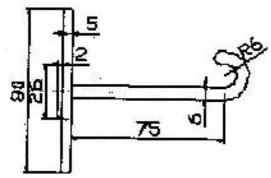


Fig-7a Loading pan

Table-3: Properties of the Permanent magnetic material and dimensions of the magnetic rings.

Residual induction, B_r (T)	1.18	Inner radius of inner ring, (mm)	10
Coercive force, H_c (kA/m)	915	Outer radius of inner ring, (mm)	16.475
Energy Product, $(BH)_{max}$,(kJ/m)	280	Inner radius of outer ring, (mm)	16.5
Curie Temp., T_c, (°C)	310	Outer radius of outer ring, (mm)	23
Working Temperature, T_w,(°C)	100	Width of the ring, (mm)	15

CONCLUSION

The setup is made for studying the characteristics of hybrid bearing. Bearing can be checked at different speeds and different load conditions. The setup is also suited for checking the journal bearings. The system has been ready for experiment. The data from this test facility can provide valuable input for bearing development and application.

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