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"Design Analysis of a Hydrostatic Thrust Pad using CFD Approach"

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ABSTRACT: The distribution of pressure in the film of liquid between hydrostatic pad and the surface on which it slides is not uniform. In analytical approach of designing, the pressure distribution is assumed constant in pocket and varies linearly in the land. Practically there is logarithmic pressure drop from inner edge of the land to the outer edge of the land. And other limitation is that, many coefficients are required for evaluating rate of flow of liquid. As such, a new numerical method using CFD approach is developed in the present dissertation for computing all the characteristics of a hydrostatic thrust pad. Using this CFD approach, these coefficients are not required for analyzing the rate of flow of liquid. The object of this project is to analyze the flow of fluid by modelling the fluid region between hydrostatic pad and the surface on which it slides. CFD module "FLOTRON" of ANSYS analysis package has been used for carrying out the analysis.

Keywords: Hydrostatic, Vibration, CFD, FLOTRON.

I. INTRODUCTION

This paper presents a study concerning the Plane bearing(slide way bearing) are restricted to application where load on the pad is always directed towards the slide way, their principal use to date has been for supporting heavy loads moving at no more than moderate speeds. However, a application involving comparatively high speeds are to be found in planing, grinding machines with quick return motion but, in the main inertia forces set lower limits for translatory speeds than for rotary speeds. A large component such as the rotary table of a vertical boring mill or the reflector of a large astronomical telescope may be mounted on a ring of hydrostatic pads on a circular track as a number of separate hydrostatic pads on sliding on a plane slide way.

There are three Basic Types of Guide way Systems used in Machine Tools

The oldest, the plain sliding way, was built in both a dovetail and a square edge style, with the square edge or "box" way being predominant. The advantage of box ways was simple design and high rigidity, but the disadvantage was high friction, stick-slip, wear, and limited speed capability. In the late '60's and early 70's, Teflon-impregnated sheet-type materials such as Turcite and Rulon became available, and were applied as way liners, which reduced the friction and the stick-slip of plain box ways. However, speed is still limited and wear is inevitable. But, feed rates were improved over plain ways and positioning and profiling accuracy was improved due to reduction of stick-slip.

To attain even higher speeds and eliminate wear, rolling element way systems were developed, in which hardened ball or cylindrical roller bearings carried the load and greatly reduced friction and eliminated stick slip. The introduction of antifriction rolling element way systems, or "linear ways", utilizing either recirculating ball or roller bearings in a "truck" assembly, traveling on a linear guiderail, made machine construction easier as they were sold as pre-assembled units which could easily be bolted on to machine structures. The disadvantage of linear ways, however, is that, due to the relatively small load-bearing contact area, particularly with ball-type rollers, they unfortunately have very low dynamic stiffness. To improve it, most systems utilize opposed sets of bearings which are preloaded against each other. This helps a little, but still does not match that of box ways. It is dynamic stiffness that resists shock, chatter and vibration, and improves surface finish and tool life.

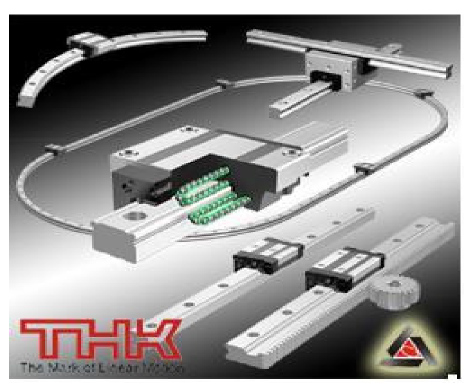

Fig. 1. The Linear Motion (LM) Guide mechanism.

Stout-KJ; Tawfik-M; Pink-EG [*1*]**:** In the year 1978 has prepared useful design charts and procedure to assist in designing both journal and thrust bearing. In addition to that general guidelines are given for material selection, bearing geometry and system design. Padate-JC; Samanth-LN; Samasundram-S [*2*]: In the year 1976 presented a study of hydrostatic and aerostatic slide way bearing by numerical methods. Jayachandra-Prabhu-T; Ganeshan-N [*10*]**:** In the year Dec 1984 described the finite element analysis and experimental study of circular recess hydrostatic thrust bearing. Braun,MJ; Dzodzo-M [*14*]**:** In the year 1995 describes on a study of the pocket depth on the flow patterns and pressure magnitudes in a hydrostatic bearing pocket, the development of the pressure is analyzed both along the transversal and in the depth of the pocket and restrictor. Galerkin finite element method using linear triangular element has been developed to solve the Reynolds equation including the rotational lubricant inertia term in polar coordinate system. The method has been applied to study plane hydrostatic bearing with four circular recesses.

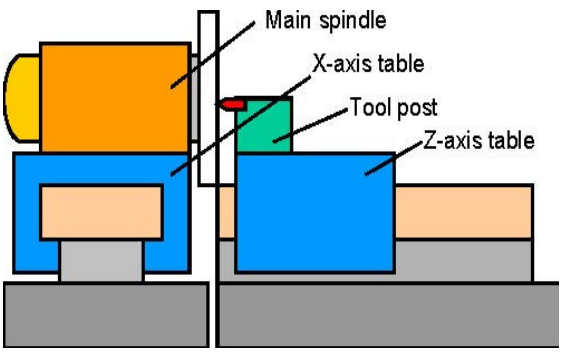

Fig. 2 Example of an ultra precision lathe.

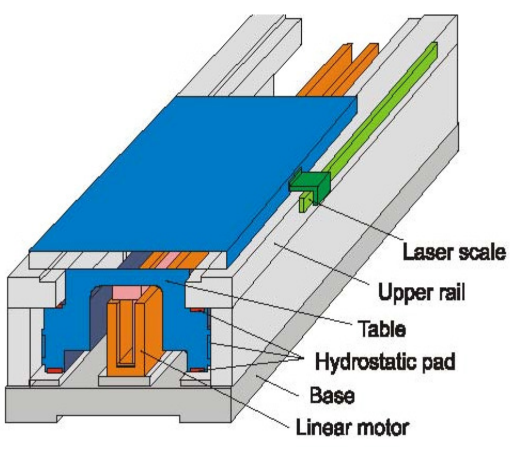

Fig. 3. The structure of the hydrostatic guide way was designed.

The development of the Linear Motion (LM) Guide mechanism. Today, the LM Guide is an indispensable component of mechanical and electronic systems in a wide variety of industries whether it is machining centers and grinding machines used by the Machine tool industry or an X-Ray machine or MRI used in the Medical field. The THK LM Guide is used everywhere from huge skyscrapers for earthquake protection to positioning in optical instruments

The "ultimate" way system is the hydrostatic way. It combines the simple way design and high rigidity of box ways with a sophisticated system of a controlled high pressure film of oil on which the slide moves. Hydrostatic ways are not a new invention, as they have been in common use for many years in many types of machine tools, including milling machines, turning machines, and grinders. They are also used in other precision structures such as large telescopes.

II. HYDROSTATIC WAY SYSTEM

Hydrostatic ways in their large milling and turning machines since the 1960's, so it is a proven design. It is only with the introduction of the new Sigma Series, however, that hydrostatic ways have been applied to their precision machining centers. In Mitsubishi's hydrostatic way system, the machine is constructed with large cross section box ways for each slide way. A controlled flow rate of pressurized oil is introduced into pockets which have been machined into the mating slide surface that rides on top of the way. When the system is turned on, the oil lifts the slide a very small distance above the slide way, creating a small gap, approximately 0.05mm thick, between the way and the slide surface. Because the oil is incompressible, any force which attempts to reduce the thickness of the gap is automatically counteracted by an immediate increase in the pressure of the oil pockets, self-restoring the thickness of the gap.

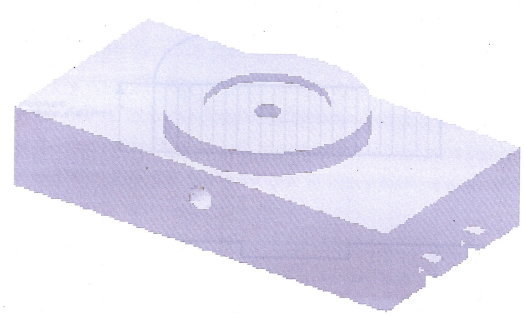

Fig. 4. A typical hydrostatic circular pad.

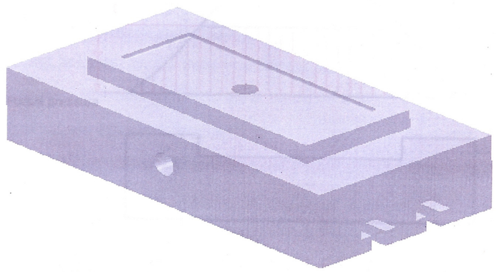

Fig. 5. A typical hydrostatic rectangular pad.

Hydrostatic pads may be of any shape in plane rectangular or circular as shown in Figure 4 and Figure 5 The lowest values of lubricant flow and pumping power for an individual pad are achieved with circular pads.

If certain assumption are made it is not difficult to calculate, with fair degree of precision, the operating characteristics of liquid –lubricated hydrostatic pad working at low or moderate speeds when all particulars of the design, including the physical properties and supply of pressure of the liquid, are decided.

The assumption are a) The liquid may be considered completely incompressible . b) The faces of the land and slide way are strictly parallel. c) The height of the inevitable surface irregularities is negligible compared with normal working clearance. d) If capillary inflow restrictor is used the flow in it is purely laminar. e) the flow in the gap between the land and sideways is purely laminar. f) the viscosity of the liquid does not change significantly between entry to the inflow restrictor and exit from the working gap. g) The relative sliding speed is too low. h) The depth of pocket is large compared with normal working clearance say not less than 20 times clearance. i) The bearing operates on the constant supply system. j) End effects at entry to and exit form the inflow restrictor and the out can be neglected.

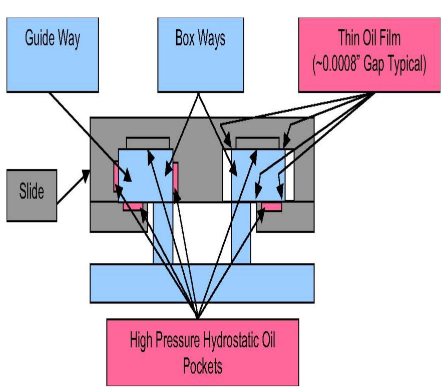

Fig. 6. The slide is supported and guided by these thin films of pressurized oil.

III. ANALYSIS RESULTS AND DISCUSSION

Virtual bearing area (A)

Figures shows the some important parameters of a hydrostatic circular thrust pad. Here 'D' is the outside diameter and'd' inside diameter of the land region P2 is the pressure distribution in the thrust pad. It can be seen from the fig. that pressure is constant in the pocket and it decrease to zero at the end. The oil enters into the pocket to that small central orifice and it flows out radially across the pocket and the land.

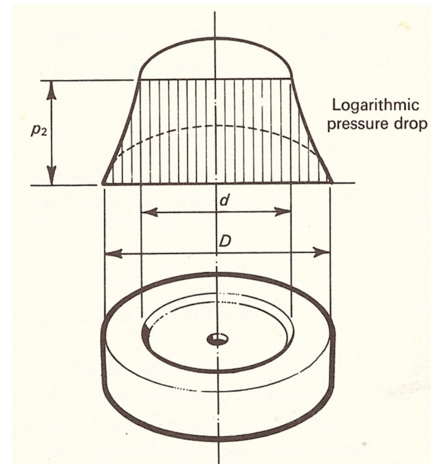

Fig. 7. Actual pressure distribution.

The ANSYS FLOTRAN elements, FLUID141 and FLUIDI42, used to solve for 2-D and 3-D flow, pressure, and temperature distributions in a single phase viscous fluid. For these elements, the ANSYS program calculates velocity components, pressure, and temperature from the conservation of three properties: mass, momentum, and energy.

Table 1: FLOTRAN Elements.

Elements	Dimension	Shape or Characteristic	DOFS
IFLUID 141	2-D	Quadrilateral, four nodes or triangle, three nodes	Fluid velocity, pressure, temperature, turbulent kinetic energy, turbulent energy dissipation, multiple species mass fractions for up to six fluids
FLUID 142	3-D	Hexahedral, eight nodes or tetrahedral, four nodes or wedge, six nodes or pyramid, five nodes; tetrahedral and hexahedral elements can be combined by pyramids	Fluid velocity, pressure, temperature, turbulent kinetic energy, turbulent energy dissipation, multiple species mass fraction for up to six fluids

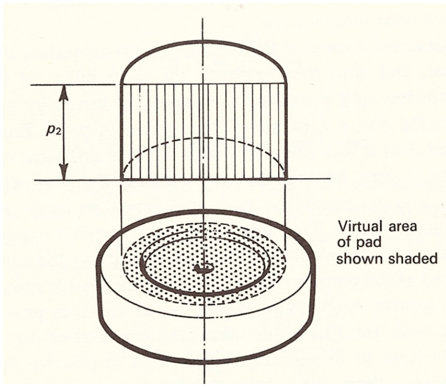

Fig. 8. Assumed pressure distribution.

The annular gap between the land and the slide way constitutes a divergent passage the cross sectional area of which increases steadily from 'лdh' at inlet to 'лDh' at outlet.

Since the fluid is virtually incompressible and therefore the volumetric flow 'Q' is the same at all sections along the passage i.e. at all radii-the flow velocity must decrease steadily from inlet at the inner edge of the land to outlet at its outlet edge.

The flow velocity at any radius is dependent on the pressure gradient at this radius and it can be shown that the radial distribution of pressure required to produce the steadily decreasing flow velocity inherent in the constant volumetric flow is a logarithmic relation between fluid pressure and radius across the face of the land. The resulting thrust between the land and the slide way is given by.

$$T = \left[\quad P_2 \frac{\pi}{4}\frac{(D^2-d^2))}{2\log e D/d} - d^2\right] \ldots\ldots\ldots (1)$$

V. CASE 1: STUDY OF THRUST FORCE AND FLOW RATE WHEN SUPPLY PRESSURE IS VARIED

GAP in the land, h= 0.05mm, Pocket radius, Ro=29.34mm, R1=58.68mm, Orifice diameter= 0.72mm, length of the resistor= 87.15mm, Dynamic viscosity of oil= 8.7x 10-7 NS/mm^2, density of oil = 8x 10-7 kg/mm^3

For Pressure 1.8 N/mm^2:

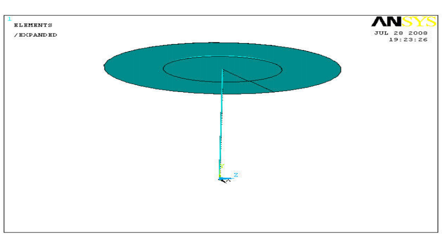

Fig. 9. Geometric model of the region.

Figure 5 shows the geometric model of the region i.e. model since the geometry as well as loads and boundary conditions are axisymmetric, it is sufficient to model axisymmetric portion.

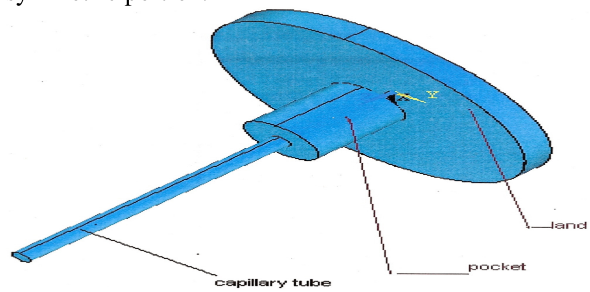

Fig. 10. 3-D Geometric model of the region.

Fig. 11. Finite Element Method (FEM) model of the thrust pad.

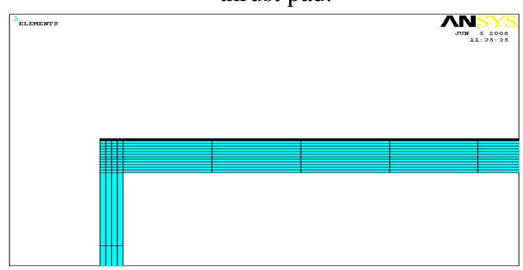

Fig.12. The zoomed up portion of the mesh in the pocket region.

VI. THE BOUNDARY CONDITION APPLIED IN THE MODEL

Application of loads and boundary conditions is the very important aspect also a difficult task for a designer. For this a through understanding of the problem is necessary. The boundary conditions are defined for outer surface Vx=0, Vy= 0 and normal velocity component Vx=0 to its axis of symmetry and supply pressure (Ps) is defined at the inlet cylinder surface is 2.25 N/mm^2 and Outlet pressure as 0,figures 9 & 10 shows the Boundary Condition Applied in the Model.

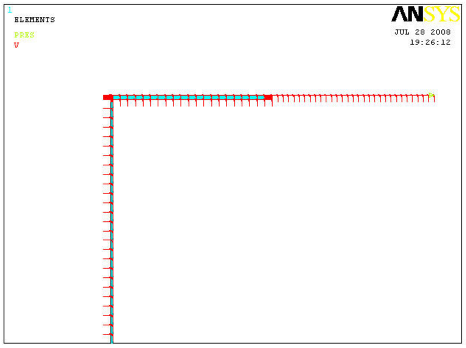

Fig.13. Boundary Condition Applied in the Model

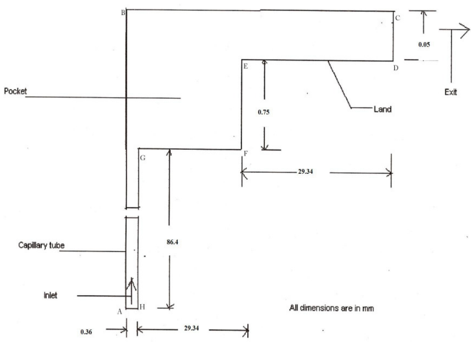

Fig.14. Boundary Condition Applied in the Model.

Velocity, Vx = 0 on the line AB.
Velocity, Vy =0 on the line AC.
Velocity, Vx =0, Vy =0, along the lines CD, DE, EF, FG, GH.
Pressure at the outlet, P = 0 (exposed to atmospheric pressure).

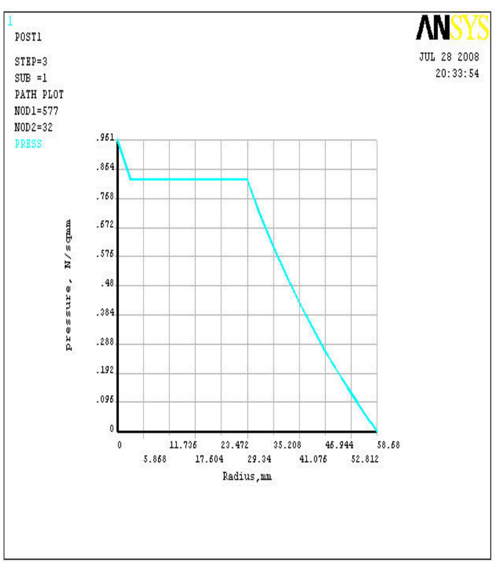

Fig. 15. Pressure acting on the slide Vs pocket radius.

Table 1 shows the Pressure acting on the slide Vs Pocket Radius. The table also shows the thrust load acting on the slide and cumulative thrust due to this pad.

Radius, mm	Pressure, N/sqmm	Force, N
0	0.961	
		24.231
2.934	0.831	
		2224.878
29.34	0.831	
		438.720
32.274	0.714	
		411.771
35.208	0.610	
		380.305
38.142	0.515	
		343.552
41.076	0.426	
		301.945
44.010	0.344	
		256.119
46.944	0.267	
		205.709
49.878	0.194	
		151.446
52.812	0.126	
		93.708
55.746	0.0613	
		32.326
58.68	0	
Cumulative Force		4864.71

The procedure for computing the thrust force is given below.

Force, $F = \pi\,(R_i^2 - R_j^2) * (P_i + P_j)/2$

$F1 = \pi\,(2.934^2 - 0.000^2) * (0.961+0.831)/2 = 24.231N$

$F2 = \pi\,(29.340^2 - 2.934^2) * (0.831) = 2224.878N$

$F3 = \pi\,(32.274^2 - 29.340^2) * (0.714+0.831)/2 = 438.720N$

$F4 = \pi\,(35.208^2 - 32.274^2) * (0.714+0.610)/2 = 411.771N$

$F5 = \pi\,(38.142^2 - 35.208^2) * (0.515+0.610)/2 = 380.305N$

$F6 = \pi\,(41.076^2 - 38.142^2) * (0.515+0.426)/2 = 343.552N$

$F7 = \pi\,(44.010^2 - 41.076^2) * (0.344+0.426)/2 = 301.945N$

$F8 = \pi\,(46.944^2 - 44.010^2) * (0.344+0.267)/2 = 256.119N$

$F9 = \pi\,(49.878^2 - 46.944^2) * (0.194+0.267)/2 = 205.709N$

F10= π $(52.812^2 - 49.878^2)$ * $(0.194+0.126)/2$ = 151.446N

F11= π $(55.746^2 - 52.812^2)$ * $(0.0613+0.126)/2$ = 93.708N

F12= π $(58.68^2 - 55.746^2)$ * $(0.0613+0.0)/2$ = 32.326N

Cumulative Force, F_{total} = 4864.71 N

Cumulative Force, F_{total} = F1+ F2+F3+F4+F5+F6+F7+F8+F9+ F10+F11+F12.

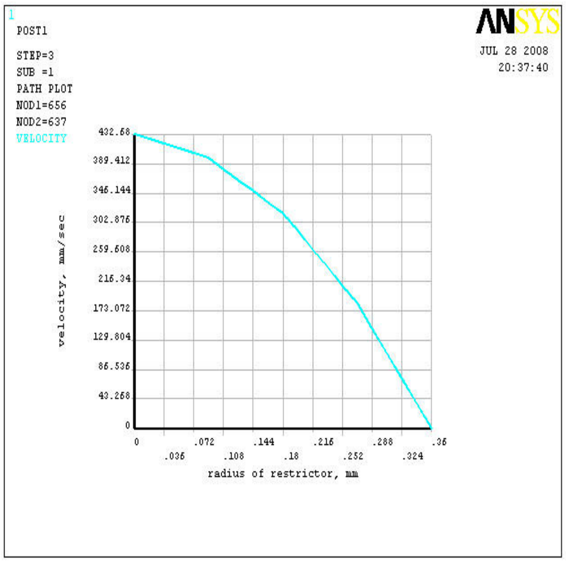

Fig. 16. Velocity profile of the fluid in the restrictor.

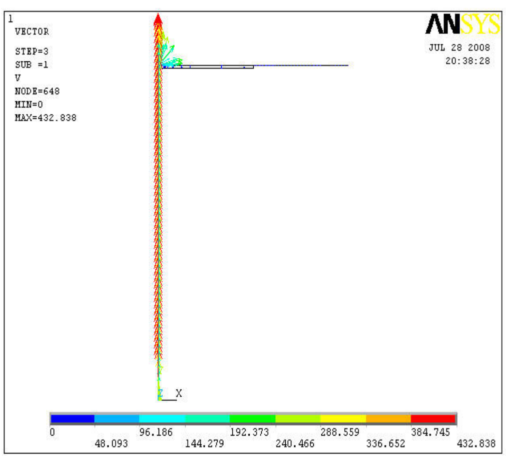

Fig. 17. Vector plot of velocity of the fluid in the restrictor.

Table shows the velocity profile in the restrictor vs Radius of the restrictor. From this Discharge through the thrust pad is computed..

The procedure for computing the discharge is described below.

Velocity,

V=425.68+418.74+411.78+404.81+394.84+381.58+365.32+349.06+332.80+316.55+290.07+263.59+237.11+210.63+184.15+147.32+110.49+36.831+0

= 5284.381mm/sec

= 5284.381*2 = 10568.762

= 10568.762 + 432.680 = 11001.442mm/sec

V_{total} = 11001.442mm/sec

Vavg = 11001.442 / 41= 268.328 mm/sec

Area = $\pi r^2 = \pi (0.36)^2 = 0.4071$ mm^2

Flow rate, q = Area * Vavg

= 0.4071 * 268.328

q = 109.236 mm^3/sec

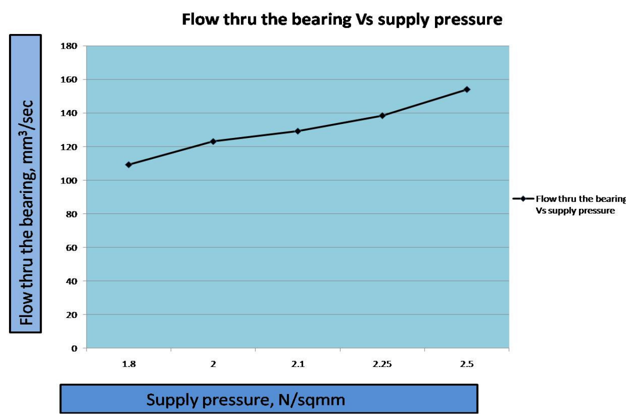

Fig. 18. Flow through the Bearing Vs Supply pressure.

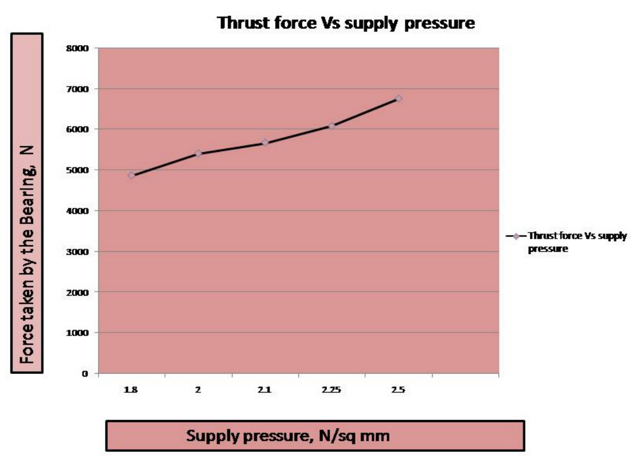

Fig. 19. Thrust force Vs Supply pressure.

CONCLUSIONS

(1) Hydrostatic pads are used in slide ways to take very high loads and with medium to low sliding velocity. FLOTRAN CFD analysis software has been used in the dissertation for analyzing the hydrostatic thrust pad. The working load being resisted by the pad is 6060N.

Since it is known that lubricant flow and consequently, power required is lowest with circular pad, circular pad is being designed now. The pad being axisymmetric in nature, the axisymmetric FEM element has been selected.

(2) In the hydrostatic pad design, there are three important design parameters, the supply pressure, the gap in the land region and the pocket radius. The effects of each of these parameters on the thrust load and flow rate have been studied by varying the values about nominal values.

(3) For the case of variation of supply pressure between 1.8 to 2.5 N/mm^2, it is observed that at lower values of pressure the flow rate is low but thrust load is too low compared to the desired values of 6060N not promote fracture easy-propagation.

(4) For the case of variation of gap in the land between 0.04 to 0.06mm, it is observed that, for a small gap of 0.04mm the thrust load is too high and for the gap of 0.06mm the thrust load is too low i.e. 4372.8N. the desired thrust was achieved for a gap of 0.05mm.

(5) For the case of variation of pocket radius from 26 to 33mm, it is observed that the maximum thrust load is obtained for a pocket radius of 29.34mm. Hence the pocket radius 29.34mm has been finalized for thrust pad.

(6) The pressure distribution in the land is not linearly dropping to zero as approximated in the empirical calculations, but it is varying nonlinearly. This is in accordance with the actual behavior as per the theory of fluid flow where it is stated that the pressure distribution in the land varies logarithmically. As such, the CFD approach is more accurate in computing the thrust load that bearing can take up.

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