PERFORMANCE OF HYDROSTATIC BEARINGS

I- DESIGN OF EXTERNALLY PRESSURIZED CIRCULAR PAD INCOMPRESSIBLE LUBRICANT BEARINGS

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ABSTRACT

This paper is concerned with a new design procedure of externally pressurized incompressible circular bearings. New charts have been developed, according to the method of bearing compensation, for evaluating of the performance characteristics of the bearing which are, the load carrying capacity, the volume flow rate and the static stiffness of fluid film. This procedure will give a good result for designer.

INTRODUCTION

A literature survey of investigations of externally pressurized incomoressible circular bearings revealed that the performance of this type of bearings can be calculated from certain equations according to some assumptions. However, no complete picture shows the relation between the bearing geometry, pressures, and films and its performance characteristics.
It's really too hard to calculate the performance characteristics of the bearing for a wide range of the above parameters. Most of previous work (1, 2, 3) gave formulas of the load carrying capacity, the volume flow rate and the static stiffness or many complicated charts to verify this situation (4).
Since the change of the lubricant viscosity or bearing applied load changes fluid film thickness, inlet recess pressure and consequently the bearing load, volume and static stiffness. So, these performance characteristics must be calculated for each case.
In this paper the change of oil viscosity through different compensating elements, fluid film, and applied load will be considered during the design procedure, in order to overcome any weakness in the bearing design, for capillary tube, sharp-edged orifice and flow control valve as restrictors.

ANALYSIS

Figure (1) shows an externally pressurized incompressible circular bearing. To predict the performance characteristics, of externally pressurized

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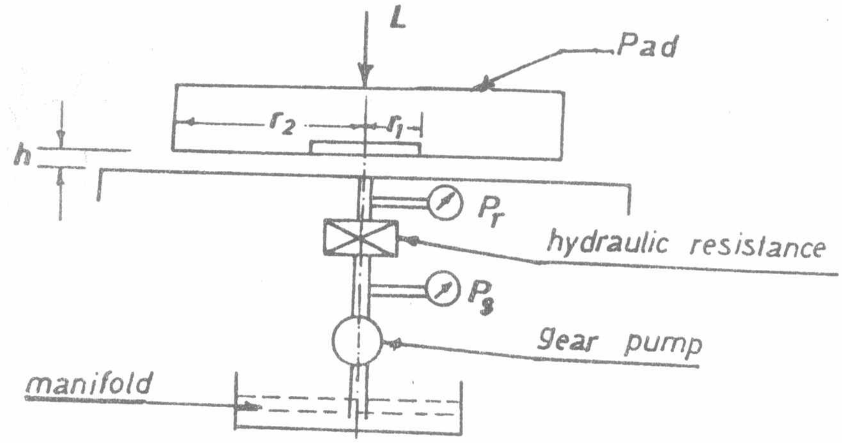

Fig.(1) Single Pad Incompressible Bearing

incompressible circular bearing, the general lubrication assumptions as well as the continuity equation are made on the Navier-Stoke's equations. The load carrying capacity will be

$$L = P_1 \pi r_2^2 G. \tag{1}$$

The volume flow rate is given by

$$Q = \frac{P_1 \pi H^3}{6 \mu \ell_n (r_2/r_1)} \tag{2}$$

The static stiffness depends mainly on the method of bearing compensation. Consequently, for capillary compensating element the static stiffness is given by :

$$\lambda = \frac{3 L}{H} \left(1 - \frac{P_1}{P_s}\right) \tag{3}$$

, for sharp-edged orific compensating element, the static stiffness is :

$$\lambda = \frac{3 L}{H} \left[\frac{2 (1 - P_1/P_s)}{2 - P_1/P_s}\right] \tag{4}$$

,and for flow control valve, the static stiffness of the bearing is :

$$\lambda = 3 L / H \tag{5}$$

NOMOGRAMS

Figures (2 - 4) present the nomograms that are used to design. In these nomograms, the performance characteristics are shown as functions of bearing outer radius (r_2), shape factor (G), bearing inlet recess pressure, (P_1), lubricant film thickness (H), and lubricant viscosity (μ) in vertical lines, while the load carrying capacity, volume flow rate and static

stiffness are plotted for externally pressurized incompressible circular bearing for different conpensating elements.
Auxillary lines are used to construct the nomogram in a simple manner. Line (q) represents the relation (H^3 / 6 μ), line (N) presents the relation ($H^3/6\mu$)(1 / $\ell n(r_2/r_1)$), and line (k) presents (AG). While line (W) presents ($1/H$ ($1 - P_1/P_s$)^2) in Fig. (2) and represents the relation ($1/H$)x $\left[2(1 - (p_1/p_s))/(2 - P_1/P_s)\right]$ in Fig. (3) .

DESIGN PROCEDURE

The procedure to be followed in the design of externally pressurized circular bearing under incompressible flow condition can be explained by the following example.
It is required to design a circular thrust incompressible bearing that has the following specifications :-
Load carrying capacity (L), fluid film thickness (H) and outer bearing radius (r_2).
Since the static stiffness of the bearing is depending mainly on the method of compensation of lubricant, it is very important to choose the restrictor type which will be used in the bearing compensation.
For the capillary tube and the sharp-edged orifice compensating elements, the static stiffness is usually function of pressure ratio (P_1/P_s), while for the flow control valve the stiffness is independant of the pressure ratio (P_1/P_s).
For using cipillary tube or sharp-edged orifice as a restictor, the following procedure is used :

1- Assume initial value of recess radius (r_1) and calculate the value of the shape factor. Connect the value of the shape factor with that of outer bearing radius by a line number (1), as shown in the Figs.(2 & 3).
2- From the point of intersection of line (1) and vertical line (k), connect this point with the line (L) at the given value of the load carrying capacity, Extend line (2) from the point of intersection to the left hand side until line (P_1) which presents the value of recess pressure. The intersection between the line (P_1) and line (2) gives the required value of recess pressure.
3- For this bearing, assume that the available lubricant viscosity is 4 x 10^{-7} kg.sec./cm^2. Connect this value with the value of film thickness by a line (3) which intersects the vertical line (9) in a point.
4- From the point of intersecting lines (3) and (q), connect this point with the value of $\ell n(r_2/r_1)$ by line (4) which intersecting the vertical line (N) in a point.
5- From the point of intersecting lines (4) and (N), connect this point with the value of the resulting recess pressure, which is obtained from step No. (2), by a line (5). The point of intersection of line (5) and line (Q) gives the required value of volume flow rate and consequently the pump capacity.
6- For the case of using the capillary tube compensation the maximum stiffness is occurred at P_1/P_s = ½, while the maximum stiffness of bearing compensated by sharp-edged orifice occurred at P_1/P_s = 0.61, for constant supply pressure condition. In Fig. (2) assume that P_1/P_s = 0.5 and connect this value with the value of a given film thickness by a line (6).
7- From the point of intersection of line (6) and the vertical line (w), connect this point with the value of the load carrying capacity, which is given by line (7). The value of static stiffness is then obtained from the point of intersection of the lines (7) and (λ).

When using flow control valve as a restrictor, steps number (1) to (5) will be repeated again, while step No. (6) will be changed as follows

6- Connect the value of the load carrying capacity with the given film thickness, this line will intersect the vertical line (λ) in a point which will present the value of bearing static stiffness for this operating case.

Assume that the bearing load increased by (ΔL) this causes the decrease of fluid film thickness by (ΔH) and vice versa. The increase of the load increases the inlet recess pressures volume flow rate and static stiffness changes due to any changes of the applied load, film thickness or inlet recess pressure. To define the new operating condition it is required to find the load and the final film thickness from step (8) for capillary compensating element and sharp-edged orifice, or from step (7) for flow control valve.

8/7- For the obtained static stiffness of the fluid film and knowing the change of load or film thickness, find the final load and film thickness. It is required to repeat the previous procedure again to obtain the recess pressure, load carrying capacity, volume flow rate and static stiffness of the new case.

CONCLUSION

The design procedure, using new charts for different compensating elements, has been developed. The effect of changing the lubricant viscosity on the characteristics of bearing has been taken in consideration in a simple manner. The design procedure has been applied for the case of sharp-edged orifice, capillary tube and flow control valve that were used as restroctprs. The charts will help in the externally pressurized circular thrust incompressible bearing design, manufacturing and operation.

REFERENCES

1- Rippel, H.C., "Basic Concepts and Pad Design, Machine Design, August 1, (1963).

2- Rippel, H.C, , "Influence of Restrictors on Performance ," Machine Design, August 29, (1963).

3- Salem, E.A. , "Analysis and applications of Externally pressurized Bearings", The Victoria University of Manchester Faculty of Technology, Ph.D. thesis January (1966).

4- Stout, K.I., and Tawfik, M. , "Design of Hydrostatic Thrust Bearings", Engineering Journal, Nov. (1978).

NOMENCLATURE

G : Shape factor = $0.5 \left[1 - (R_1/R_2)^2 / \log_e(R_2/R_1)\right]$

H : fluid film thickness.

Q : volume flow rate.

L : load carrying capacity.

r_2 : outer bearing raduis.

r_1 : recess radius.

P_1 : inlet recess pressuse.

P_s : supply pressure.

ΔH : the change of fluid film thickness.

ΔL : the change of the bearing load.

μ : lubricant viscsity

λ : static stiffness.

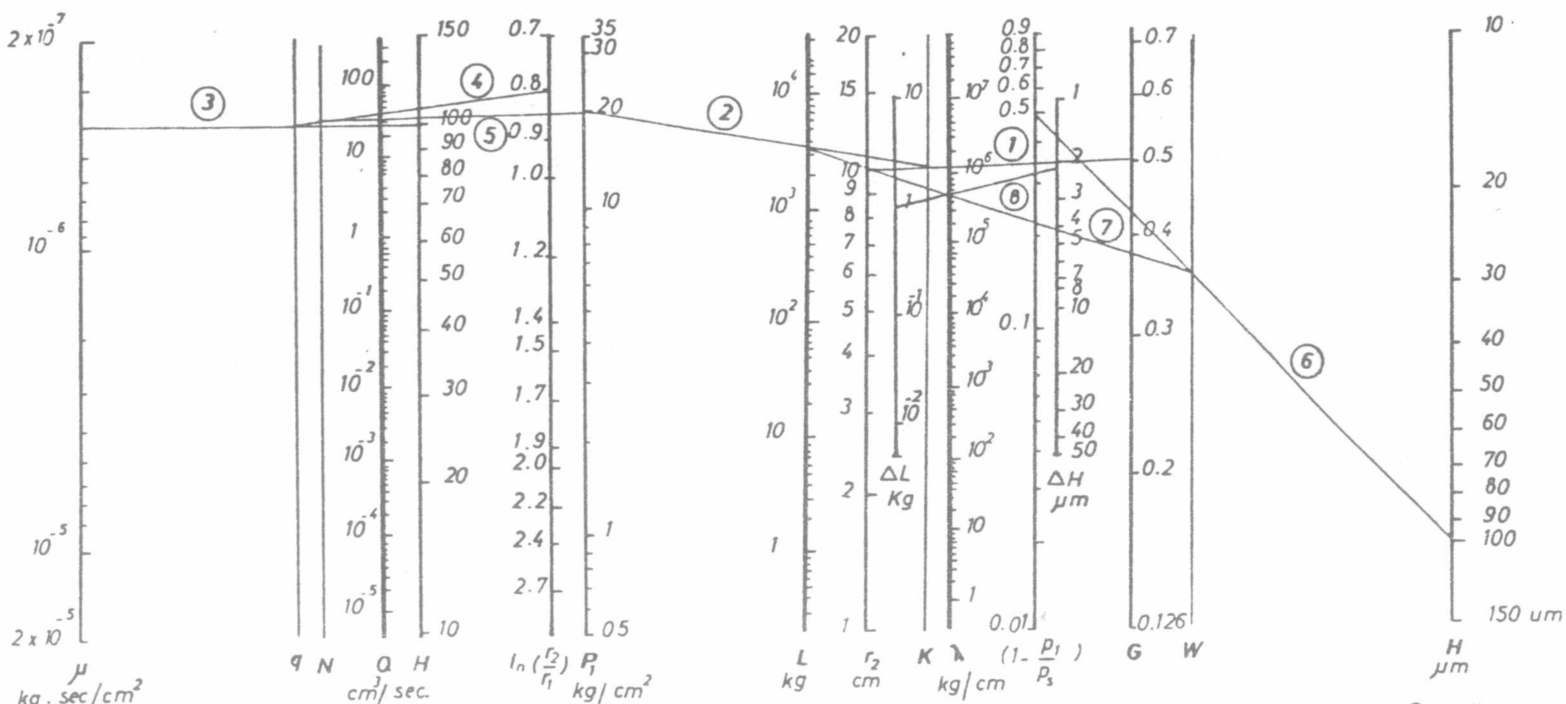

Fig. (2) Design Chart of Circular Pad Incompressible Lubricant Compensated By Capillary Tube.

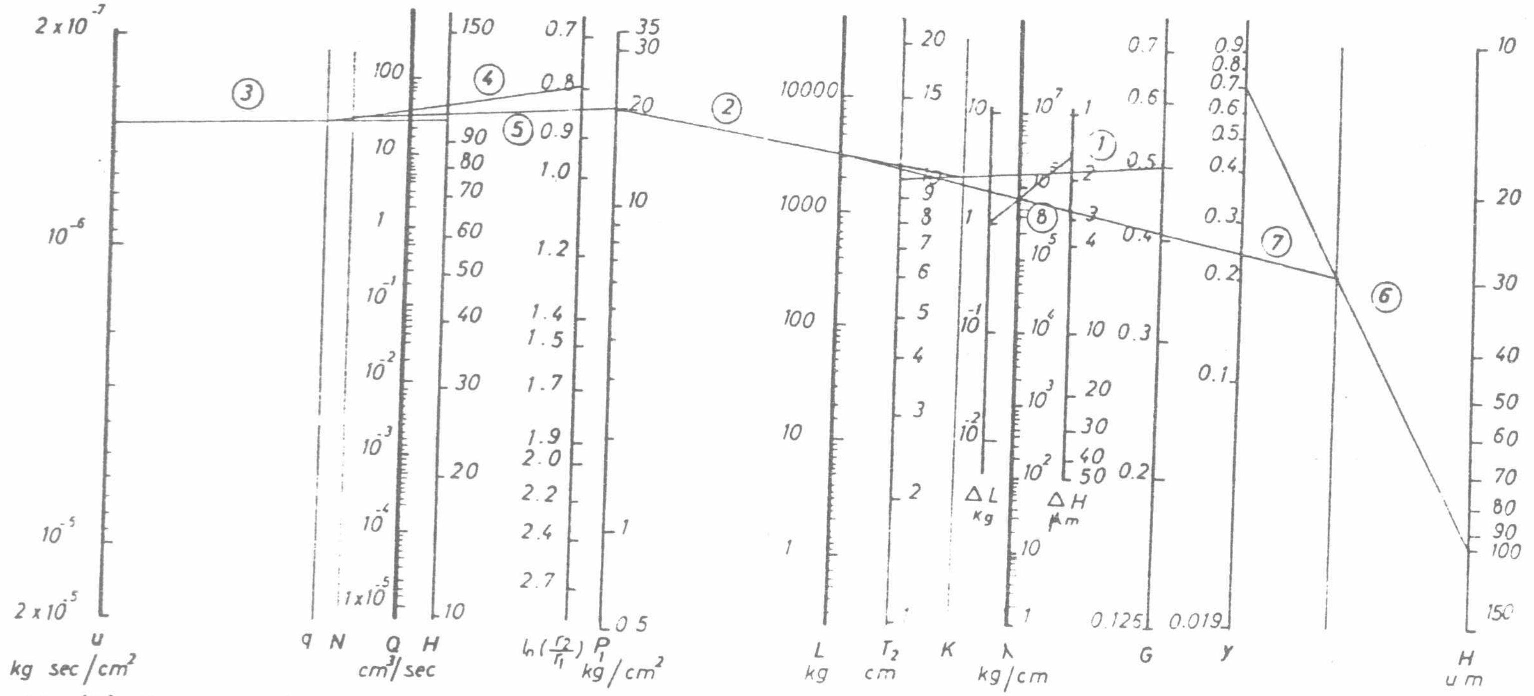

Fig.(3) Design Chart of Circular Pad Incompressible Lubricant Compensated By Sharp Edged Orifice.

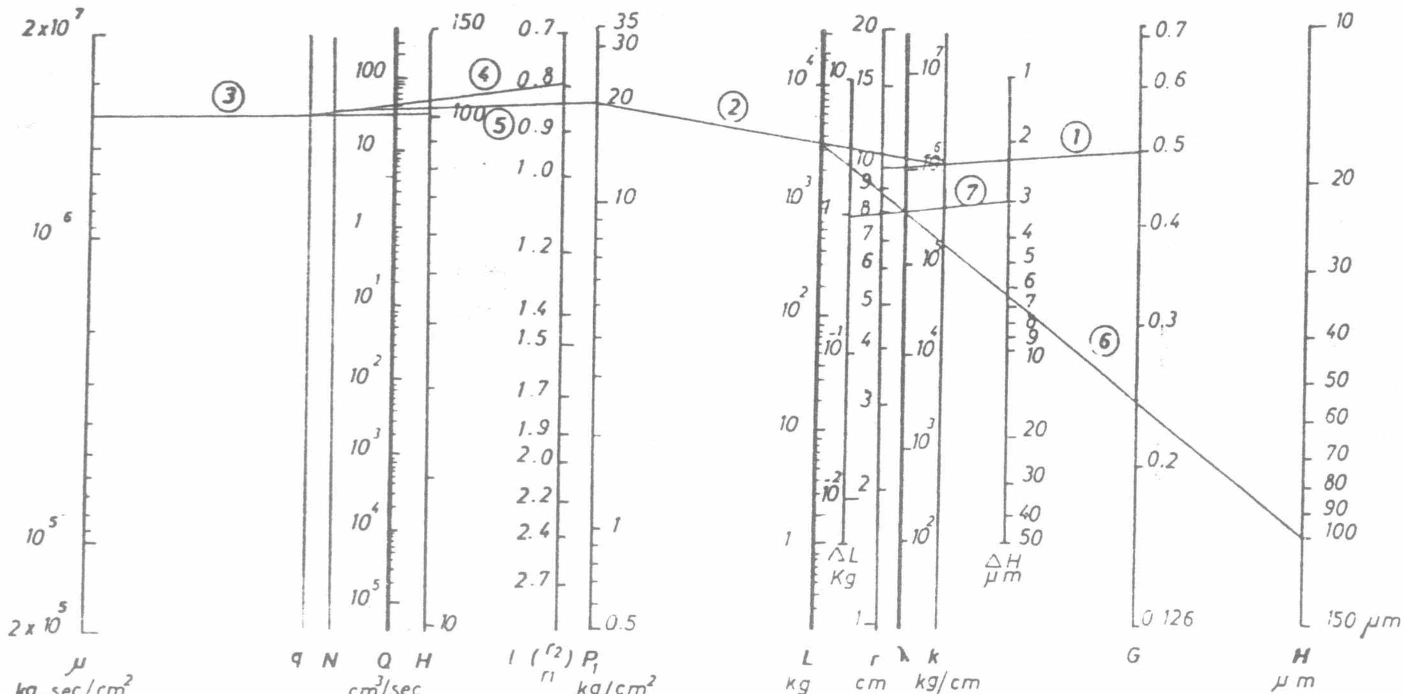

Fig.(4) Design Chart of Circular Pad Incompressible Lubricant Compensated By Flow Control Valve.