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Marine Bearings

A. ROSE, B.Sc., C.Eng., A.M.R.I.N.A. (Associate Member)

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A. ROSE, B.Sc., C.Eng., A.M.R.I.N.A. (Associate Member)*

The paper emphasizes that reliability is the all-important factor in the design of marine bearings and, because of this, design methods tend to be conservative. Well-established methods for calculating film thickness and power losses of fluid-film bearings are given. Some notes are given on elasto-hydrodynamics, but the design and selection of roller bearings are based on a limited fatigue life. Emphasis is given on the need for design data based on experimental work on realistic shafts of about six inches diameter and the difficulties in using data from small scale experiments are pointed out.

Various types of bearing are described, also the effect that the positioning of bearings and the seating arrangements can have on loads and vibration. Typical bearing failures are described and a chart given showing potentially dangerous material and oil combinations for turbine bearings. Future trends are examined and the conclusion reached that plastics will have the greatest single effect in bearing design.

INTRODUCTION

The reliable operation of rotating and reciprocating machinery is very largely dependent upon the correct functioning of the bearings which carry load between components in relative motion to each other.

In the design of marine bearings, as opposed to other machinery applications, two points must be considered:

1) the possible tragic effects of machinery breakdown at sea;
2) that all machinery is mounted on a continually flexing platform and that actual service loads may be in excess of those calculated assuming a static platform; this is particularly true of, say, propeller shaft bearings.

Although economy takes second place to reliability in marine machinery, every effort has to be made to ensure economical use of machinery and, as far as the bearings are concerned, this means that first cost and power loss must be as low as possible, consistent with reliability. It is also established in marine engineering that designs based on limited life are not acceptable, as for instance in aeronautical engineering. Thus, for example, the automatic replacing of bearings after a certain number of hours would not be countenanced.

That bearings have to be reliable and generally last the life of the ship means that marine bearings are usually conservatively rated and that the design methods used are well proven. Modifications can only be made after extensive experience has been gained in less vulnerable applications.

One of the difficulties facing the designer of marine bearings which, on the whole, tend to be large, is the difficulty in obtaining reliable experimental data.

For reasons of cost, most experimental work is carried out on relatively small diameter bearings of one or two inches diameter. It is accepted that the laws of similarity probably hold good only for a twofold increase in scale and, since the majority of marine bearings are in excess of six inches diameter, there exists a considerable gap in our knowledge.

The British Ship Research Association is exceptional in having excellent facilities for testing 6-in turbine journal bearings and turbine thrusts up to about 10 000 rev/min as well as being able to run full load tests on main thrust blocks.

* Manager, Research and Development Department, Michell Bearings, Ltd., Newcastle upon Tyne.

Pametrada and B.S.R.A. have also been responsible for measurements of thrusts and torque on propeller shafts and measurements of turbine thrust and bearing temperature in vessels at sea.

Information is also available of measurements on the film thickness, temperatures and thrust loads of USS *Barry*(9).

No information seems to be available on the measured performance of heavy-oil engine bearings and, in view of continuing developments in heavy-oil engine propulsion, a definite need exists for experimental work.

In spite of efforts to ensure trouble-free running, bearings fail and, as another aspect of reliability bearings should be designed so that in the event of failure, the consequential damage to the machinery would be as slight as possible. An example of this consequential damage would be damage to turbine blading if journal-bearing failure allowed blade-rub.

Faults in machinery, not necessarily due to the bearings, often manifest themselves at the bearings either as failure, or merely as abnormal running such as excess vibration or overheating. This type of fault may be unbalance, resonant vibration, abnormal loads due to cargo distribution etc.

The methods of design put forward in this paper are well proven by experience at sea. They tend to be conservative in that film thickness and power losses are on the pessimistic side for sliding bearings and in that roller bearings are rated well within their design capacity.

The paper has been divided into two main sections. The first describes design methods for evaluating the actual bearing performance of sliding and rolling bearings, the second discusses particular applications.

BASIC DESIGN FORMULAE FOR SLIDING BEARINGS

Sliding Bearings

The majority of the prime mover and propulsion system bearings are sliding bearings.

Sliding bearings can take radial load (journal bearings) or axial load (thrust bearings) and may be divided into fixed or tilting-element bearings.

Thus there are plain journal bearings and tilting-pad journal bearings, both of which find widespread use at sea, and fixed and tilting-pad thrust bearings. The fixed-pad thrust bearing is little, if at all, used in British marine practice

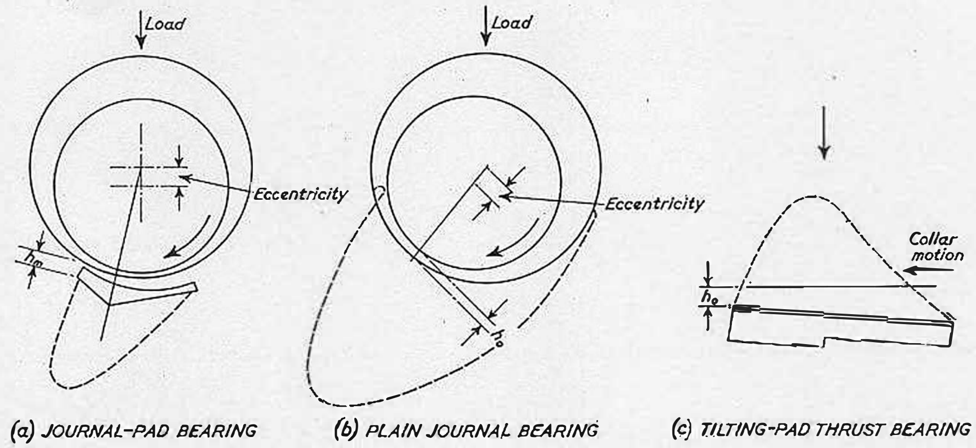

FIG. 1—*Film thickness and pressure distribution in hydrodynamically-lubricated bearings*

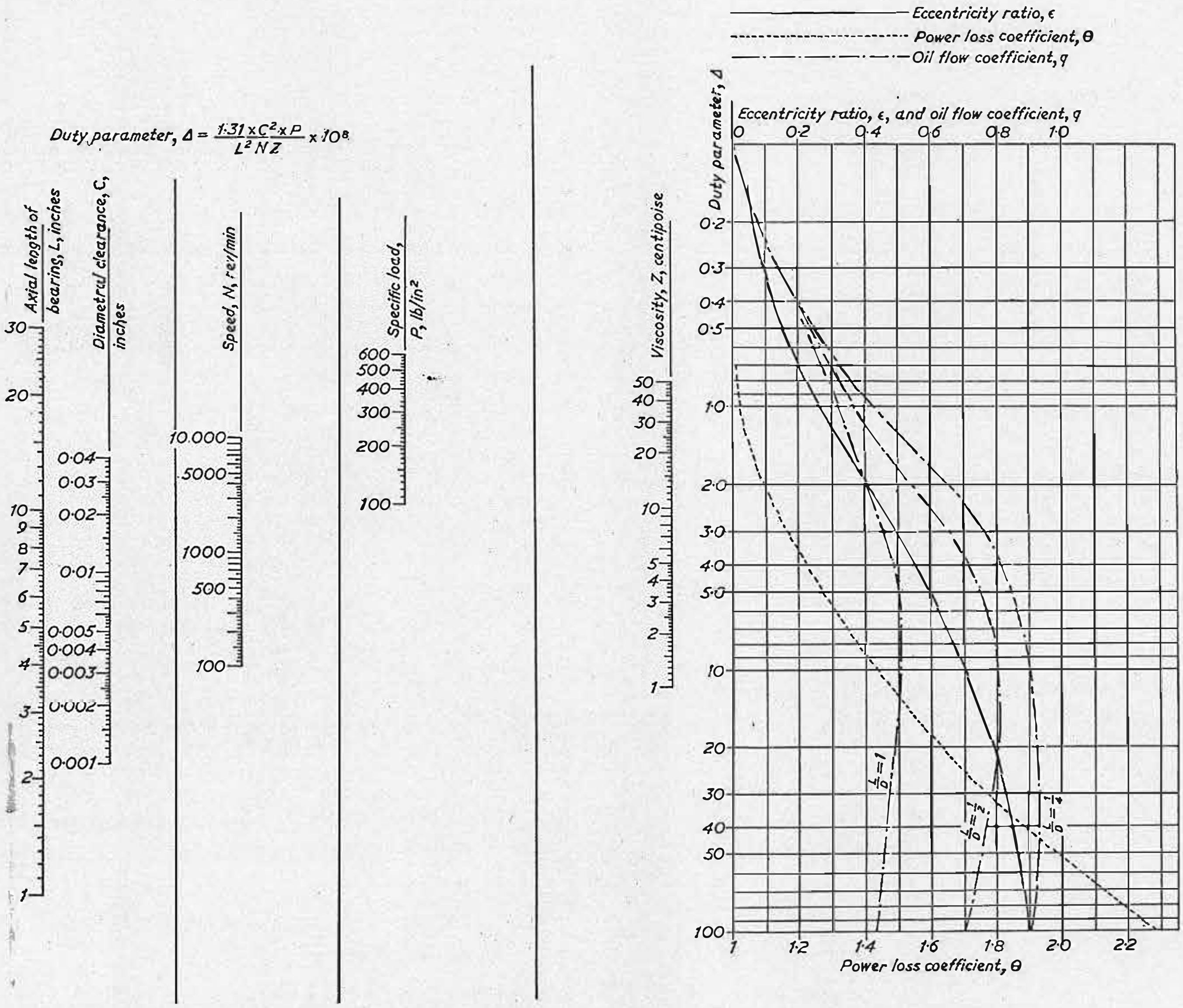

FIG. 2—*Nomogram for eccentricity ratio, power loss coefficient and oil flow coefficient*

although it is used in some turbines built in the United States.

Since the oil film separating the moving parts has to sup-port load, pressure must be generated in the film. That pressure is generated in an oil film was first demonstrated by Beauchamp Towers.

Shortly afterwards Osborne Reynolds arrived at the rela-tionship between speed, viscosity and pressure generated within the oil film, this relationship being:

$$\frac{\partial}{\partial x}\frac{h^3}{\mu}\frac{\partial p}{\partial x} + \frac{\partial}{\partial z}\frac{h^3}{\mu}\frac{\partial p}{\partial z} = 6U\frac{\partial h}{\partial x} \qquad (1)^*$$

*For list of symbols see Appendix I.

All design methods and oil-film calculations depend upon the solution of this equation, but because of the assumptions involved in solving it, even the best design methods can only be approximate. The assumptions usually made for the solution of the Reynolds equation are:

i) the lubricant is Newtonian in its behaviour, i.e., viscosity is independent of rate of shear;
ii) the lubricant is incompressible;
iii) the viscosity of the lubricant is constant throughout the film;
iv) inertia effects on the lubricant film are negligible;
v) the velocity boundaries are given by $U=0$ at the stationary part of the bearing and $U=U$. at the mov-ing part of the bearings;
vi) lubricant flow is laminar; in general this may not be true, but for all marine bearings probably is;
vii) oil films are so thin that, for journal bearings, the curvature of the bearing surfaces may be neglected and that pressure is constant across the film from the shaft to bearing.

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