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EFFECT OF SURFACE ROUGHNESS ON JOURNAL BEARING PERFORMANCE

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ON

JOURNAL BEARING PERFORMANCE

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SUMMARY

The effect of surface roughness on the thermoelastohydrodynamic performance of journal bearing is investigated theoretically and experimentally. The performance of infinitely long journal bearings is predicted with considerations of surface roughness of the bounding solids, frictional heating effects on both viscosity and density of lubricant and the thermoelastic distortion of the bearing components .

The thermal analysis is made through the solution of the energy equation with allowance of heat dissipation by conduction, convection and radiation. The heat flow through the bounding solids is approximated according to an existing theory and empirical formulae .

The resulting equations are solved by iterations through a special computer program .

The experimental work includes the design of a suitable journal bearing test set-up, and the design, production and calibration of the special bearing dynamometer .

Series of a planned experimental tests were carried out to verify the theoretical work. Experimental results showed close agreement with the theoretically derived equations .

Results are presented for bearings having an average surface roughness from 0.0 up to 4.847 μm (R_a), and radial clearances from 0.075 up to 0.125 mm, operating at different rotational speeds (1200, 1800 and 2400 r.p.m) .
Performance charts are presented and comparisons between the theoretical and experimental results are made whenever applicable .

The following points are concluded : -

1. The maximum oil temperature tends to increase with increasing surface roughness. The maximum temperature is generally about 1.5 - 2 times the inlet temperature. Furthermore, the effect of engineering surface roughness (up to 1 μm, R_a) is to increase the maximum temperature by 10 - 15% of the predicted values for smooth bearing surface.
2. The effect of surface roughness on velocity distribution is to increase the maximum negative velocity deviation of the oil at the entry point, whilst increasing the maximum positive velocity deviation at the point of minimum film thickness. These phenomena tend to increase the length of the cavitation zone of the bearing .
3. Furthermore, the effect of engineering surface roughness is to reduce the maximum bearing pressure, and hence the load carrying capacity of

the bearing. The reduction percent is not linear, though an approximate average value (detected under test conditions) may be taken as 10 - 15% per μm of the average roughness (R_a) of the bearing surface.

4. The load carrying capacity of each of the tested bearings is reduced at an average rate of 5700 N/μm when the surface roughness is increased up to 0.3 μm. Further increase of surface roughness reduces the reduction rate to a some steady value of 500 N/μm .

5. The predicted reduction in the load carrying capacity (W) of rough bearing surfaces readily affects the Sommerfeld number of the bearing ($\frac{\mu N}{P}$) . Thus the Sommerfeld number of the bearing is underestimated by about 25% per μm of the R_a when calculated based on the assumption of smooth bearing surface .

6. The coefficient of friction increases with increasing surface roughness, though an asymptotic value is always reached at surface roughness of about 5 μm average roughness (R_a) .

NOMENCLATURE

The following symbols are used in the theoretical analysis .

A	Area of the outer surface of the bush .
C	A constant in the viscosity-temperature relationship.
D	Diameter of the bush .
E	Modulus of elasticity .
F	Frictional force .
G	Modulus of rigidity .
J	Mechanical equivalent of heat .
K	Overall heat transfer coefficient .
L	Bush length .
R	Radius of the shaft .
R_{mean}	Mean bush radius .
T	Temperature .
U	Surface velocity of the journal .
W	Total load .
c	Radial clearance between bearing surfaces .
c_p	Specific heat of lubricant at constant pressure.
e	Eccentricity between journal & bearing centres .
f	Coefficient of friction .
h	Fluid film thickness between bearing surfaces .
k	Thermal conductivity .
ε	Eccentricity ratio, (e/c).
p	Pressure .
u	Linear velocity component in the x-direction.
v	Linear velocity component in the y-direction.
w	Linear velocity component in the z-direction.
x,y,z	Cartesian co-ordinates .
α	Coefficient of thermal expansion of the solid .
γ	Power exponent in viscosity-pressure relationship.

ς	Temperature coefficient of density .
θ	Angle to a point on bearing surface, measured from load line .
λ	Power exponent in viscosity-temperature relationship .
μ	Absolute viscosity of lubricant .
ρ	Density of lubricant .
σ	Poisson's ratio of the bearing material
τ	Viscous shear stress .
ϕ	Attitude angle between line of centres and load line .

Subscripts : -

amb	Ambiant .
f	Lubricant film .
in	Inner .
m	Mean .
max	Maximum .
min	Minimum .
o	Indicate the value of the parameter at condition of entry to the oil film .
out	Outer
s	Solid

Supersubscripts : -

–	Dimensionless .

INTRODUCTION

Lubrication is an essential feature of all modern machinery and has been throughout the ages. The first authenticated use of lubricants is found in the relics of ancient Egypt. It is probable that the massive stones of the pyramids were floated into place on thin semi-fluid layers of mortar .

The first quantitative investigations of hydrodynamic behaviour were carried out towards the end of the nineteenth century. As early as 1849, the idea of introducing full fluid film lubrication between a shaft and its bearing was first studied by Von Pauli; and later by Hinn in 1854 (quoted after, Cameron[1]).

Beauchamp Tower, conducted experiments to determine the friction of the journal of railway car wheel, and observed the generation of hydrodynamic pressure in the first partial journal bearing. The pressure generated even in these early experiments of 1885, were sufficiently large to displace plugs or stoppers placed in oil holes in the bearing casing .

About the same time (1883 - 1885), Petroff carried out experiments with bearing lubricated with mineral oil, and showed that the friction of a bearing could be entirely explained by shearing forces in the lubricant film. Petroff

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applied Newton's law of viscosity to a rotating concentric journal, and expressed frictional shear in terms of lubricant viscosity, speed, and physical dimensions. The Petroff's equation is still in application to-day in the case of lightly loaded journal bearings[2].

Later, in 1886, Osborn Reynolds[3], published his classical theory of hydrodynamic lubrication, and deduced his well known pressure equation, which is widely used in the design of modern machinery .

Reynolds's theory for journal bearing was subsequently developed by Sommerfeld in 1904. The Sommerfeld number is widely recognised to-day as the most significant parameter in bearing design[2] .

In general all solutions for the pressure developed in bearings and its power consumption were based on smooth round surfaces. This assumption, however, is not entirely valid due to the existence of surface texture .

Salama[4], in 1950, presented both experimental and theoretical treatment, studying the effect of the surface texture on the performance of the parallel-faces thrust bearing. These treatments have shown that the performance of this type of thrust bearing is sensitive to the changes in the surface macro-roughness profile, and there seems to be sufficient justification for relating the film lubrication performance of this type of bearing to the hydrodynamic effect of the oil pockets shaped by the surface profile .

Though the journal bearing problem is far less understood, specially when both elastic and thermal distortions are involved. Thus it is felt that further study of the effect of the primary surface texture is required .

The aim of the present work, is to analyse theoretically and to investigate experimentally the effect of the variation of primary surface texture on the behaviour of an infinitely long journal bearing as indicated by the variation in pressure distribution, load carrying capacity, friction characteristic, temperature distribution and oil film thickness .

CHAPTER 1

LITERATURE REVIEW

In almost every surface when oil is somewhere present a lubricant film manages to get between the surfaces and carry part of the load . The mechanisms by which it does this are different . In some cases the mating parts are specially shaped to ensure the presence of a complete fluid film , as in thrust and journal bearings. In others, such as gears and rolling bearing elements, the metal parts contribute in the generation of the film. In a third case, such as machine tool slide ways the thermal distortion of the metal produces a wedge. (Cameron[5]).

The performance of journal bearings when operating under light loads and high speeds can be accurately described by the hydrodynamic theory where the important physical properties of the system are its geometrical dimension and the viscosity of the lubricant. At higher loads and lower speeds, however, this simple theory is insufficient. Although a complete theory based on these conditions is lacking, it has been found that many other physical properties of the system affect the behaviour of lubricated bearings. These may be grouped as, those of the lubricant and those of the bearing surfaces .

The first group consists of : -

1. The heat capacity of the oil .
2. The adsorptive ability .
3. The corrosion products with the metal surfaces.

There are a number of well known published papers which deals with these aspects, however, for space limitations will not be mentioned in this section. The second group consists of : -

1. The thermal wedge .
2. The viscosity wedge .
3. The surface deformation .
4. The surface roughness .

1.1 The Thermal Wedge

This factor was first presented by Fogg[6], in 1946, who pointed out that Reynolds, when deriving the continuity equation assumed constant volume flow i.e. constant density and viscosity, whilst in fact the oil is thermally expanding in the direction of motion due to the increase in temperature. This expansion, as given by Fogg is equal to $\alpha (t_2 - t_1)$ where α is the coefficient of thermal expansion of the lubricant and t_2 & t_1 are the outlet and inlet temperature respectively. The oil, therefore, is squeesed in the direction of motion, thus producing the same effect as decreasing the film thickness .

Bower[7] in an attempt to explain Fogg's results produced an analytical solution for Reynolds's equation assuming linear variation for both viscosity and density over the bearing length. The results showed the effects of viscosity and density on the load carrying capacity for bearing of different geometrical shapes .

These results can be summarized in the following .

i. The variation of the density alone can support a small load .

ii. The variation of the viscosity alone is not capable of supporting any load .

Shaw[8], in 1947, also analysed Reynolds's equation with the assumption that the density varies linearly along the bearing length without any variation in the viscosity . His results confirmed the results given by Bower, hence, the thermal(density variation) wedge was not enough to explain the high loads observed by Fogg .

1.2 The Viscosity Wedge

In discussing the work of Shaw[8], Block[9] criticised the assumption of linear temperature rise along the bearing. He emphasised that the temperature rise, and hence, the variation in density and viscosity which are dependent on temperature, should be governed by the so called energy equation .