THE ANNALS OF "DUNAREA DE JOS" UNIVERSITY OF GALATI
FASCICLE XIV MECHANICHAL ENGINEERING, ISSN 1224-5615
2019

EVOLUTION OF OIL FILM FORMATION IN THE BEARING RADIAL BEARING HOUSINGS

Assistant Daniela PANĂ,
University "Dunărea de Jos", of Galați
Faculty of Engineering and Agronomy of Brăila,
danyela_mail@yahoo.com,
daniela.pana@ugal.ro

ABSTRACT

Slip bearings are machine components that provide support for rotating parts, usually rotating shafts or axles, take on the forces that load the parts concerned and work under conditions of relative sliding of the shaft journal surface onto the bearing (bearing shell) surface, while the two surfaces are separated by a film of lubricant. For the correct operation of the slip bearings, without systematic wear of the Journal and bearing shell surfaces, they must be separated by a sufficiently thick lubricant film to completely exclude direct contact of the two surfaces in relative movement.

KEYWORDS: Oil film, Journal, bearing shell, lubricant, radial bearing, friction

1. INTRODUCTION

Lubrication, a process absolutely necessary for any machine or mechanical construction with moving elements, concerns technicians of all categories, from the study, research and design offices to simple or very complex industrial systems.

The correct selection of a lubricant is a complex action, due to a number of factors that must be taken into account: constructive factors (the type of the trisystem, its accessibility for lubrication, quality and quantity of lubricant, method of lubrication, etc.), operating factors (load, speed, etc.), environmental factors (ambient temperature and pressure, presence of contaminants, etc.).

The characteristic of the lubricating materials market is today the existence of a wide range of lubricants, mineral or synthetic, which are intended for general use or for highly specialized use. The properties and quality of lubricating materials depend on the nature of the raw material and the viscosity range of the base oil, on the processes and processes of obtaining and the degree of purification, and on the quantity and quality of the additives (additives)

2. DEVELOPMENT OF THE OIL FILM FORMATION

In the rest position, the Journal rests on the bearing shell, the center of the Journal and the bearing shell are in the direction of force, as shown in figure 1, a. At start-up due to the frictional force between the Journal and the crankpin - the fusset tends at first to climb on the bearing shell, figure 1,b).

A long state of rest can lead to a metallic contact, which causes the Journal the tendency to mount on the bearing shell at the first moment when starting. In its movement, the Journal drives the lubricant that adheres to it, and due to the viscosity, it forms the bearing film. As seen from the theory of flat plates, the carrier's resultant is off-center, which is also observed in this situation, and the consequence is the off-center setting of the spindle and bearing shell.

This trend is accompanied by a counteracting action, namely: with the rotation of the Journal the oil is driven (due to adherence and viscosity) into the wedge-shaped interstit (resulting from the bearing play).

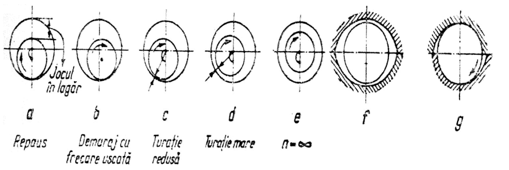

Fig. 1. The evolution of the formation of the oil film in the radial bearings.

The purpose of the lubricant is to create a film between the Journal and the bearing shell, thus reducing friction, which no longer occurs between two metals, but between metal and a fluid substance.

The lubricant coating in contact with the bearing shell has a zero speed and that in contact with the Journal has its speed v; the lubricating interlayers have different speeds, ranging from zero to v. When turning the spindle, the lubricant is driven (displaced) in the gap between the Journal and the bearing shell, in the gradually narrowing area, increasing its pressure; this results in a hydrodynamic (self-supporting) force, which interrupts the direct contact between the Journal and the bearing shell, defeating the force F which loads the shaft.

For functional situations in figure 1, a continuous lubricant film is formed in which a hydrodynamic pressure is stored that is capable of balancing the outer load. The formation of the hydrodynamic load film is due to the achievement of functional conditions: 1. The relative speed between the Journal and the bearing shell; 2. The play between the Journal and the bearing shell leads to the formation of a "grease stop"; the existence of a sufficiently large quantity of lubricant between the journal and the bearing shell evolution of oil film formation in radial bearings.

Mineral oils are the most important lubricating materials, due to the replacement of the friction between the Journal and the bearing shell by the internal friction of the oil, the coefficient of friction can be reduced up to 100 times and in addition, the oil carries the heat released to the bearing.

The most important oil properties are: Viscosity – the property of opposing the movement of one oil layer relative to another; lubricity – the ability of the oil to adhere to the surfaces of the parts and form thin films on them. The viscosity characterizes the internal friction in the oil film and depends on the temperature and pressure.

It is understood that for the oil to pass from the input to the output area, it is necessary to develop sufficient pressure in the oil film to support the spindle. The pressure in the input area will always be higher than the pressure in the oil outlet area.

Due to this difference in pressure, the spindle is raised and moved to the left (figures 1, c and 1, d in the direction of movement). It is thus noted that in the case of lubrication of the bearings, the distribution of pressures is asymmetrical with respect to half the length of the lift surface and flat surfaces (figure 2).

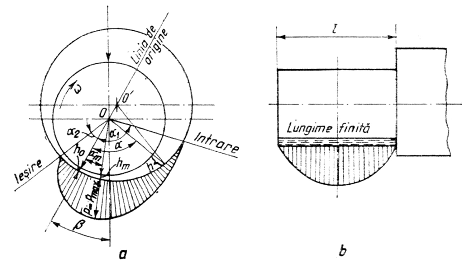

Fig. 2. The curve of the pressure distribution in the radial bearings.

With heavy loads, lateral movement is smaller. If the load increases and the pressure in the oil film is not sufficient the angle is reduced to 0 and the oil film will break, the spindle will rest directly on the bearing shell. The dry rubbing will then occur with its consequences.

When the pressure in the oil film is sufficient and the spindle speed tends toward infinity, the center of the Journal tends to have a concentric position with the bearing shell (Figure 1,e).

Such position causes the parallelism between the two surfaces to appear, and therefore the bearing to decrease accordingly. As a result, the Journal will tend to return to its off-center position, which provides a wedge-shaped gap. It follows that at very high engine speeds and low or variable loads, near to the coincidence of the Journal axis with what the bearing shell will have an area of instability of the Journal position and therefore the possibility of vibrations, which should be avoided. Among the breeding solutions used, the following are quoted:

ß minimize the relative play or use of lemon-shaped bore or offset-play bearing shells (figure 1, f, figure 1, g, respectively).

The lemon shape is obtained by turning to a larger diameter of the bore the two half-shells, but with adjustment plates between the separating planets. After turning, the plates are removed and so the bearing shell of the desired shape and dimensions is obtained, observing the relative play only in the direction perpendicular to the separation plane of the bearing shells.

The shape of the offset play bearing shell is obtained by turning the bore off-center, after which a half-shell is assembled turned 180 degrees in the horizontal plane.

Outside the unstable area, hydrodynamic lubrication ensures that the relative spindle position is self-adjusting and that the coupler operates correctly. The path described by the spindle center during operation is similar to half a circle (Figure 1, e).

Figure 2, a and b shows the pressure curve, the relative position of the Journal and the geometrical reference elements required for the calculation of the hydrodynamic lubrication of the bearing. The following symbols and designations are hereby adopted:

R - bearing shell bore radius
R - Journal radius
Δ_d - bearing play $=2\Delta_r$
$\Delta_r = R - r$ radius play
$\Psi = \frac{R-r}{r}$ level of the vehicle
e - journal eccentricity, in the bearing
$\chi = \frac{e}{\Delta_r}$ relative eccentricity
$\delta = \frac{h_o}{\Delta_r} = \frac{(R-r)-e}{R-r} = (1-\chi)$ the minimum relative thickness of the oil film
h_m - the thickness of the oil film in the direction of maximum pressure; this direction closes with the line of the center of the angle α_m.
$h_0 = (R\text{-}r) - e$ - absolute minimum thickness of the oil film.

The relative position of the Journal is characterized by the relative eccentricity of the Journal and the angle β. The origin angle β is the angle closed by the direction of the force P with the straight line containing the center of the Journal and the shell called the origin line. The angles α1 and α2 define the beginning and the end of the load area respectively. It is noted that only part of the edge of the bearing shell is involved in taking up the load. It should be noted, however, that, at the same bearing, as the speed increases up to the point close to the incidence of the center of the Journal and the bearing shell when the oil is introduced through the upper part, the pressure curve will be in the shape shown in figure 3.

The existence of oil-scaled portions of fus that do not effectively contribute to taking up the load should be avoided, as friction is increased under these conditions.

The example in figure 3 is relevant for determining the location of the inlet, its dimensions, and the oil inlet pressure.

In figure 2, show that the minimum thickness of the oil film is measured in the direction of the origin line. Figure 2.b shows that the pressure distribution along the Journal length (in z direction) when its axis is parallel to that of the shell is approximately parabolic. The pressure is at the highest half of the Journal length and at the ends, the pressure is equal to the outside. When the axis of the Journal is tilted, the pressure has another distribution as shown in Figure 4 (curves II and III). Curve IV gives the pressure distribution when the spindle is tilted and has contact with the edges of the bearing shell, a situation which should be avoided because friction and heating are increasing rapidly, bearing and spindle heat up and can lead to binding.

Figure 5 shows the pressure distribution in the case of a curved spindle (in the case of intermediate journals). Depending on the curvature, the pressure distribution is more or less flattened. Touching the edges of the shell brings the same negative consequences.

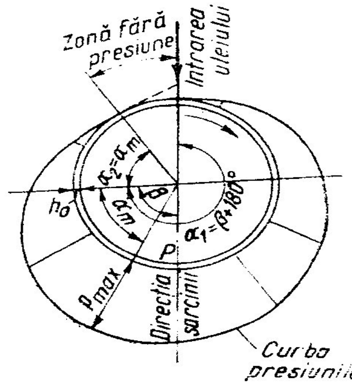

Fig. 3. Bearing with oil-scaled Journal and non-load side, highlighting the location of the feed hole and the quantity of feed oil.

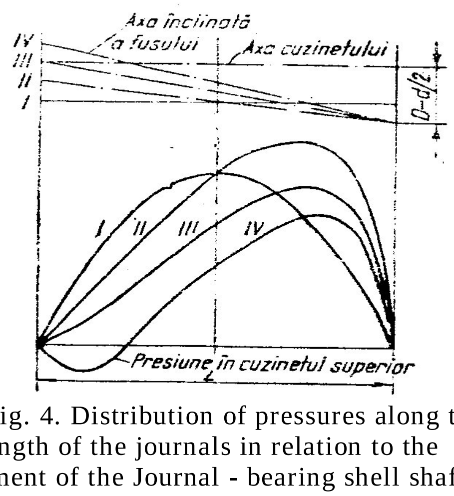

Fig. 4. Distribution of pressures along the length of the journals in relation to the alignment of the Journal - bearing shell shafts.

The examples shown indicate that the shorter spindles behave better, the effect of the bending is less, that it is necessary to carefully check the spindles and shafts when bending so that the spindle tilt angle and the arrow do not exceed the permitted limits.

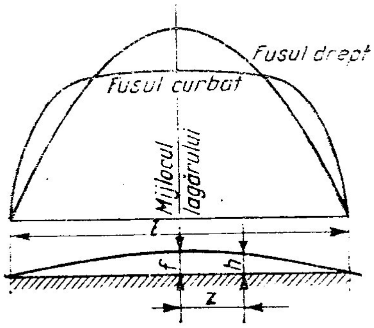

Figure 5. Modification of the pressure distribution due to the formation of the Journal.

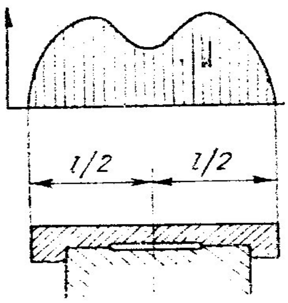

Figure 6. Modification of the Journal pressure distribution due to deformation in the bearing shell.

Figure 6 shows the pressure distribution in the case of a bearing shell which may become deformed as a result of its design and bearing.

3. THE OIL FILM HOLDER

The bearing load, its capacity to wear the Journal on which the load operates, is given by the pressure in the lubricant film. By solving the Reynolds equation, the pressure for finite length and low eccentricity journals ($e \approx \frac{r}{1000}$) is obtained:

$$P = \frac{\eta \cdot \omega}{\psi^2} \cdot d \cdot l \cdot \phi_p, \qquad (1)$$

or

$$p_m = \frac{P}{d \cdot l} = \frac{\eta \cdot \omega}{\phi^2} \cdot \phi_p \qquad (2)$$

In which P is the force (load) taken over by the oil film ϕ_p and I have a dimensionless parameter called Sommerfeld's number.

The inverse of Sommerfeld’s number, C_p, shall be called a load factor and shall be determined with the relationship:

$$C_p = \frac{1}{\phi_p} = \frac{1}{\frac{p_m \cdot \psi^2}{\eta \cdot \omega}} = \frac{\eta \cdot \omega}{p_m \cdot \psi^2} \qquad (3)$$

As the pressure is dependant on many factors p = p(η, r, ω, ψ, δ, α, l) plus the influence of the characteristics of the coupling materials (resistance to deformation), of the surface qualities and of the execution accuracy, etc., tables or diagrams are used for the current design of the bearings, in which the values $\frac{1}{C_p} = \phi_p$ are already calculated, depending on the ratio $\frac{l}{d}$ and δ or χ.

The determination of the viscosity of the lubricant which provides fluid lubrication for C_p is done with:

$$\eta \geq \frac{p_m \cdot \psi^2}{\omega} \cdot C_p \qquad (4)$$

Other factors of the relationship may also be determined when other factors are known. The choice of lubricant must be taken into account that a high viscosity also gives high friction.

4. CONCLUSIONS

The thickness of the oil film is dependant on the relative clearance but also on the roughness of the Journal and the bearing shell surfaces.

The weight of the lubricating film is dependant on the factors on which the value is determined, but also on the relative ratio $\frac{l}{d}$ and thickness δ of the lubricating film.

The ratio $\frac{l}{d}$, influences the load, lubricant flow and heating, so all the basic factors for the coupler to function in good condition.

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