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Determination of the transition speed in journal bearings under consideration of bearing deformation

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ABSTRACT

This paper discusses the further development of an analytical model to calculate the minimum film thickness and the transition speed in journal bearings with consideration of bearing deformations. The focus of the development was on generalising an existing model to include any relative eccentricities and width/diameter ratios. Furthermore, the influence of different bearing types and the layer structure of plain bearings on the deformation are taken into account. Using two white metal journal bearings as examples, the results of the new approach are discussed and compared with previous analytical models and full numerical calculations.

1. Introduction

When designing hydrodynamic journal bearings, the eccentricity of the shaft within the bearing shell and the resulting minimum film thickness are two essential target values of the bearing calculation. The eccentricity of the shaft depends on the hydrodynamic pressure development in the plain bearing. The main influencing factors for the pressure development are the diameter and the width of the bearing, the load, the bearing clearance, the viscosity of the lubricant and the sliding velocity between shaft and bearing shell. These variables are summarised in the Sommerfeld number, which is important for the plain bearing calculation [1–4]. Other influencing factors are the friction-induced increase in temperature in the bearing, the elastic pressure-induced deformation of the shaft and bearing shell, as well as the occurrence of cavitation. Due to the complex relationships, numerical calculations are often used to determine the conditions in the plain bearing [5–7] or [8]. However, there are also various analytical calculation methods used for the design of journal bearings [1,4,9–11]. With these methods it is fairly easy to calculate the eccentricity for an operating point of a rigid bearing with perfectly smooth shaft and bearing shell surfaces. Analytical methods are especially useful if a time-efficient design and optimisation of plain bearings are the main aim. Optimisation algorithms can be found in [12,13].

Through the design of journal bearings, it must be ensured that the occurring eccentricities do not exceed certain limits. If the selected bearing parameters result in a too low eccentricity, unwanted vibrations may occur during operation and the frictional losses of the bearing are higher than necessary (upper operating limit). However, if the eccentricity of a bearing in operation is too large, mixed lubrication occurs, which should be avoided in many bearing applications. This so-called lower operating limit can be defined using the minimum permissible oil film thickness h_{lim} and the transition speed.

Various analytical methods exist for determining the transition speed of statically loaded journal bearings. An empirical equation based on rigid bearings was developed by Vogelpohl in [14]. With the force F acting on the bearing, the dynamic viscosity η, the bearing volume V and an empirical transition coefficient C_T the following equation arises:

$$n_{tr} = \frac{F}{C_T \eta V} \quad \text{with} \quad V = \frac{\pi}{4} D^2 B \tag{1}$$

In [14] Vogelpohl specifies a value of 0.1 m^{-1} for C_T. Lu and Khonsari directly derive the following equation by solving Reynolds differential equation, which is also based on rigid bearings [4,15]:

$$n_{tr} = \frac{\overline{p} h_{lim}}{4.678 \frac{C}{2} \left(\frac{B}{D}\right)^{1.044} \eta \left(\frac{D}{C}\right)^2} \quad \text{with} \quad h_{lim} = 3\sqrt{\left(1.25 Ra_J\right)^2 + \left(1.25 Ra_B\right)^2} \tag{2}$$

But close to the transition point from hydrodynamic lubrication to mixed lubrication with very small minimum oil film thicknesses, the elastic deformation of the shaft and bearing shell can no longer be ignored. Therefore, Spiegel developed a method in [16] based on the work by Vogelpohl [14], which can be used to

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Nomenclature

B	bearing width [m]
C	radial bearing clearance $= D_B - D_J$ [m]
C_T	factor defined by Vogelpohl [m^{-1}]
D	bearing diameter [m]
E'	reduced Young's modulus [N/m²]
F	radial force [N]
h_{min}	minimum oil film thickness [m]
h_{lim}	minimum permissible oil film thickness [m]
K_E	elasticity factor [–]
n_{tr}	transition speed [min^{-1}]
Ra	average roughness [μm]
Rz	average roughness height [μm]
Rq	root mean square roughness (RMS) [μm]
$\overline{p}$	specific bearing load $= F/(BD)$ [N/m²]
So	Sommerfeld number $= \overline{p}\psi^2/(\eta\omega)$[–]
U_{tr}	transition velocity [m/s]
V	bearing volume [m³]
ε	relative eccentricity [–]
η	dynamic viscosity [Pa s]
ω	angular velocity [1/s]
ψ	relative bearing clearance $= C/D$ [‰]

calculate the transition point into mixed lubrication taking into account elastic deformations. This method is described in the section below. An additional approach to calculate the transition point into mixed lubrication under consideration of elastic deformations is described by Landheer et al. [17]. The method is based on the approach by Spiegel [16], although the equations are generalised such that the transition speed can be illustrated in a so-called transition diagram, depending on the load. In addition, the temperature dependence of the viscosity is considered in the calculation by means of approximation equations. Assuming a constant viscosity, the equation for a single load reads:

$$n_{tr} = \frac{60}{\pi D}\frac{K_E \overline{p}\psi h_{lim}}{\eta} \quad \text{with} \quad h_{lim} = 2.5\sqrt{Rq_J^2 + Rq_B^2} \tag{3}$$

The used elasticity factor K_E represents the reciprocal value of the elasticity factor derived by Spiegel [16]. Explanations about the used minimum permissible oil film thickness h_{lim} are provided in [18].

2. Analytical model for calculating the transition speed according to [16]

The transition speed can initially be derived for the rigid journal bearing. Based on the fact derived in [19] that the value $k(\varepsilon) = So(1-\varepsilon)$ for all B/D ratios and for $\varepsilon \to 1$ ($h_{min} \to 0$) tends toward $k(\varepsilon) = 1.224$ in rigid bearing calculations and with the Sommerfeld number $So = \overline{p}\psi^2/(\eta\omega)$ and the minimum oil film thickness $h_{min} = 0.5C(1-\varepsilon)$ the following equation can be written:

$$k(\varepsilon \to 1) = \; 1.224 \; = \frac{\overline{p}\psi^2}{\eta\omega}\frac{2h_{min}}{C} = \frac{F\psi^2}{\eta UB}\frac{h_{min}}{C} \tag{4}$$

Assuming that the transition point to mixed lubrication in journal bearings occurs at $\varepsilon \approx 1$, the minimum oil film thickness h_{min} is equated with the minimum permissible oil film thickness $h_{lim.}$ The sliding velocity U corresponds to the transition velocity U_{tr}. Thus, the transition speed for the rigid bearing can be calculated as follows:

$$n_{tr} = \frac{60}{\pi D}U_{tr} = \frac{60}{\pi D}\frac{F\psi^2}{1.224\eta B}\frac{h_{lim}}{C} = \frac{60}{\pi D}\frac{\overline{p}\psi h_{lim}}{1.224\eta} \tag{5}$$

The minimum permissible oil film thickness h_{lim} can be calculated with the following equation, ignoring inclinations and waviness [3]:

$$h_{lim} = Rz_J + Rz_B \tag{6}$$

Provided that the minimum oil film thickness is constant, a greater force can be transmitted from an elastic bearing than from a rigid bearing [20]. In so doing, the increase in load-carrying capacity is dependent on the elastic properties of the bearing and the shaft as well as on the forces to be transmitted. The ratio between rigid and elastic load-carrying capacity can be described with an elasticity factor K_E. The elasticity factor K_E can be calculated using dimensionless parameters, which are described in more detail in [20]. To determine K_E, rigid calculations of journal bearings with finite width are put in relation to elastic calculations of a cylindrical/plane contact with an infinite width (without side flow) in [16]. The calculations used in [16] can be gathered from [20–22]. The resulting values of K_E are only presented in form of a diagram in [16]. According to [11], the following equation applies to the resulting elasticity factor:

$$K_E = \left[1 + \sqrt{2}\frac{\overline{p}D}{E' h_{lim}}\right]^{2/3} \tag{7}$$

The knowledge of the material properties of the shaft and bearing shell, which can be used to determine a reduced Young's modulus E', is required to determine the elasticity factor:

$$\frac{1}{E'} = \frac{1}{2}\left(\frac{1-\nu_J^2}{E_J} + \frac{1-\nu_B^2}{E_B}\right) \tag{8}$$

Considering the elasticity factor K_E in the calculation of the transition speed, Eq. (5) becomes:

$$n_{tr} = \frac{60}{\pi D}U_{tr} = \frac{60}{\pi D}\frac{F\psi^2}{1.224K_E\eta B}\frac{h_{lim}}{C} = \frac{60}{\pi D}\frac{\overline{p}\psi h_{lim}}{1.224K_E\eta} \tag{9}$$

However, the assumptions made for Eqs. (4)–(9) often result in an incorrect prediction of the transition speed. This is essentially due to the following:

- inaccurate calculation of the elasticity factor K_E, because the side flow is disregarded in the used elastic bearing calculations and the description of the geometry according to [20] is simplified
- assumption that the transition into mixed lubrication occurs at $\varepsilon \approx 1$
- inaccurate determination of the minimum permissible oil film thickness h_{lim}
- depending on the design of the journal bearing and the housing as well as on the operating point, it is possible that the load-carrying capacity is reduced by elastic deformations

Therefore, an enhanced model is presented below, which achieves greater prediction accuracy.

3. Improved calculations of the elastic journal bearing

Elastic calculations of a cylindrical/plane contact with an infinite width were used in [16] for a lack of suitable solutions. Nowadays it is