Gesellschaft für Tribologie	Rolling Bearing Lubrication	GfT Work sheet 3

Replaces GfT-Work sheet 2.4.1/1984

Contents

1 Preface

The purpose of this work sheet is to explain the theoretical and practical knowledge of today on bearing lubrication and assist users in design offices and workshops. In addition to the proved calculation methods for reliable designing of rolling bearing arrangements, the contained design and maintenance instructions shall guarantee a fail-safe operation in the sense of a planned maintenance. For bearing lubrication, oils, greases and in exceptional cases solid lubricants can be used.

With identical operating conditions, the required lubricant amounts are less than those for sliding bearings. It must, however, be guaranteed that a sufficient lubricant quantity is available at all times under all operating conditions and at each functional area. Only a proper lubricant, carefully adapted to the operating conditions and the bearing type, in connection with appropriate lubricant conveyance, may yield a long service life.

Therefore the lubricant has become a vital machine element.

Details on and properties of the various rolling bearing types have been mentioned in so far as necessary.

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2 Principles

The tribo-system "Rolling Bearing" is characterized by heavily loaded areas in rolling contact and in the normal case only slightly loaded surfaces in sliding contact (rolling element/cage, rolling element face/lip, rolling element/rolling element, cage/cage guiding areas). Therefore its service life depends on fatigue phenomena, in certain cases also on wear processes (sliding wear, mixed friction). Both damage types as well as frictional losses are influenced by lubrication.

The correct selection of bearing type and size (dimensioning) is checked by calculating the nominal rolling bearing life according to the well-known principle of the dynamic load carrying capacity (see DIN ISO 281).

Bearing life calculation assumes formation of a sufficiently thick grease film between the surfaces in rolling contact, which depends on lubricant type and lubrication method. It is predominantly described by the EHD theory, i.e. the theory of the elastohydrodynamic lubrication. To which extent a load-carrying EHD film will develop depends on bearing size, rotational speed, primarily on the operating viscosity of the lubricant and to a small extent on the magnitude of load. A minimum loading is necessary to ensure a correct cycling process in the bearing.

The EHD theory does not take into account lubricant changes depending on operation and time and exceptional operating conditions (temperature, loads) nor environmental conditions (media, vacuum). The long-term behaviour of a lubricant is governed by other laws.

3 Lubrication theory and influence of the lubrication condition on bearing life

[1], [2], [12], [18], [30]

3.1 General on lubrication theory

Bearing life is also influenced by the lubricant film. Its thickness is determined by the lubricant itself and its properties adapted to the operating conditions as well as by the macro- and micro-geometry of the contact surfaces. The separation of the contact surfaces is aimed at.

A direct comparison between theoretical lubricant film thickness and roughness depth of the surfaces may lead to a misinterpretation of the lubricating condition. The lubricant film thickness required for a wear-free operation, that means for a separation of the elements in rolling contact, does not only depend on the surface roughness but also on the profile form [40]. As practice and tests show, a film thickness of a few 10ths µm will be sufficient (Fig. 1).

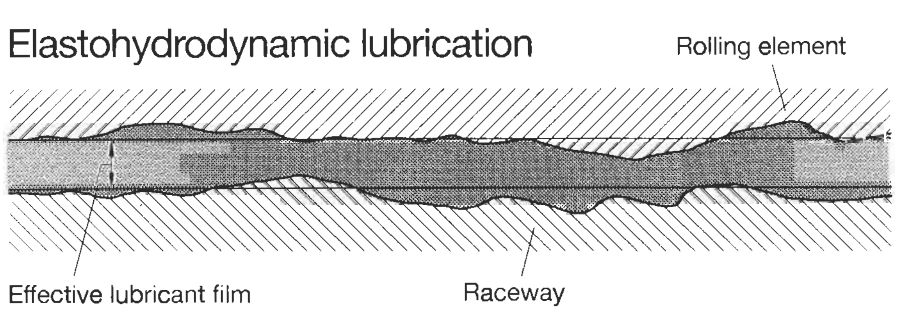

Fig. 1: Complete separation of the surfaces by a lubricating film of thickness h_0.

When evaluating the lubricant film thickness it should be borne in mind that only the lubrication condition in the rolling contact is concerned. The physical principles do not take into account the lubrication conditions at other contact surfaces with higher amounts of sliding friction. Therefore the lubricant selection should not exclusively take into account the principles of the EHD theory but additionally field experience as well as the overall lubricating condition in the bearing. This especially applies when using lubricants with extreme pressure additives*) which might release chemical reactions whose effect cannot be covered by the EHD theory [41].

Lubrication based on such additive reactions should be allocated to boundary lubrication. Mixed friction is characterized by partial metal-to-metal contact. This is accompanied by a run-in effect. In the case of boundary lubrication, the oil film

*) Code letter acc. to DIN 51502

thickness will not be larger than one oil molecule. This happens, if the amount of lubricant is insufficient or with a too low relative movement between the contact surfaces. The friction coefficient will then amount to ~ 0.1 and may increase to 0.5 with beginning metal-to-metal contact (Fig.2).

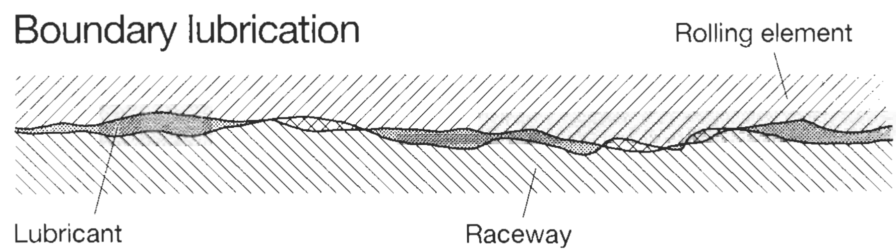

Fig. 2: Local metal-to-metal contact with mixed friction

Also in case of synthetic oils with polar (physically)-acting properties the actual lubricant film thickness may differ from the rated value.

As a standard value for the minimum loading a C_0/P_0 coefficient of 60 can be accepted, depending on the bearing type. EHD preconditions an elastic deformation of the elements in rolling contact (higher load).

For determining the comparable lubricating condition for grease lubrication, the viscosity of the base oil and the influence of the thickener have to be taken into account. The field tested, wear-resisting influence of some thickeners on the contact surface separation can presently not yet be confirmed by theory, but pertinent tests are planned.

The viscosity of a lubricating oil drops with rising temperature. To ensure build-up of a load-carrying lubricant film between the contact surfaces of rolling elements and raceways the oil needs a definite viscosity at operating temperature. If this viscosity is known from practice or has been defined according to Section 3.2.3, the pertinent ISO VG grade (40 °C) can be read off figure 6b, Section 3.2.3.

There is a natural upper limit to the oil viscosity. At higher rotational speeds functional limits result from increasing mechanical energy losses, especially in case of higher friction under idle running and from the higher bearing temperatures, but also from the reduced pumpability of high-viscosity oils.

The viscosity of the lubricating oil changes with the pressure. Hertzian pressures at the rolling contact amount to 30 000 bar under heavy loads, in the run-in phase up to 7 000 bar [49].

$$\eta = \eta_0 \cdot e^{\alpha p}$$

η_0 = dynamic viscosity at atmospheric pressure
η = dynamic viscosity at pressure p
p = pressure $[N \cdot m^{-2}]$

For lubricants based on mineral oils this has been taken into account in the calculation of the lubrication condition according to the EHD theory. For other lubricants viscosity-dependent changes of the lubrication condition may occur. The differing pressure-viscosity behaviour depending on viscosity is shown in Fig. 3. The values listed for paraffine base mineral oils form the a_{23} diagram (Fig. 7). Mineral oils with EP additives show α values between the values measured for paraffine and naphtenic base mineral oils. Furthermore, the pressure-viscosity coefficients α substantially depend on the lubricant molecule structure and/or the lubricant doping.

The published α values have been determined under nearly static conditions. Under rolling contact the pressure conditions rapidly change, the period during which the high pressure acts, generally is very short and therefore it is doubtful whether the high viscosity increase can be expected at all. Moreover, the pressure increase entails increasing internal friction which leads to a higher temperature and therefore to a corresponding viscosity decrease, see [53].