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Influence of mass temperature on gear scuffing

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ABSTRACT

A new scuffing parameter for gears lubricated with mineral base oils is developed, proposing the existence of gear mass temperatures which are critical for each lubricant (viscosity grade).

The introduction of the total cycle of the gear for mass temperatures calculation is supported by a theoretical background that was clearly established without empirical formulation and without any geometrical gear specificity which provides consistent physical support to the proposed scuffing parameter.

The oil dynamic viscosity at the critical mass temperatures is constant, allowing the prediction of critical temperatures for other viscosity grades of the mineral gear oils without the need of additional scuffing tests.

1. Introduction

In previous works [1,2] it was possible to verify that the traditional scuffing criteria are not able to explain many of the phenomena observed, for instance, how different tooth geometries (eg. FZG types A and C) and oil bath temperature affects the scuffing load carrying capacity of gears lubricated with additive free base mineral oils. For restricted and particular operating conditions, all scuffing criteria seem to be valid, however, they fail for different operating conditions. These studies showed that the oil bath temperature (or the oil viscosity) and the friction coefficient between gear teeth have a very significant influence on the scuffing load carrying capacity of the FZG gears.

On the other hand, there are some faults in the explanation of some relevant phenomena for gears scuffing. For example, the parameter T, which represents the distance from the pitch point to the start or the end point of meshing, also known as the Almen's “T” parameter [3,4], explains the behaviour of different gear geometries in scuffing, but does not provide its true physical meaning. The importance of parameter “T” in gear scuffing was confirm in previous studies [1], but a clear physical meaning of this parameter and a convincing explanation of the gear scuffing behaviour have not just been provided.

It is fundamental that a scuffing criterion for gears should not depend on a specific gear geometry and the concept should be applicable to other mechanisms, such as those found in rolling bearings or cam-tappet mechanisms. Furthermore, it is important to note that in the case of gears, such as other mechanical elements, the operation is discontinuous, that is, each teeth pair is in contact only a time fraction of the total period, the shorter the greater the number of gear teeth.

This approach that proposes the existence of gear mass temperatures which are critical for each lubricant (viscosity grade) was also started in a previous work [5], but in that case the theoretical background was week. Now it became clear that the mass temperature depends on the gear power loss during gear meshing but also on tooth flank cooling between meshing periods. So, this study highlights the relevance of the parameter “T”, which results from the discontinuous operation of the teeth. The concept of integral temperature [6,7] is also introduced in order to improve the physical perspective of the phenomenon.

There are not many recent publications in the scope of gear scuffing. Recently, scuffing becomes once again a relevant subject, due to the increasing number of failures that have occurred due to the tendency to use increasingly thinner oils. But, even so, is reduced the amount of work published on this subject. The gears scuffing load capacity is been updated through refining some parameters that interferes with scuffing, like the dynamic loads and transient thermal elastohydrodynamic lubrication [8], that affect the flash temperature [9], or the power loss [10,11]. Other recent scuffing studies are performed in twin disk test machine [12] or ball on disk test machine [13] intended to perform scuffing studies with simpler mechanisms, intended to characterize the lubricants but with no direct repercussion for gears. Another recent study [14] analyses the bulk temperatures and flash temperatures in bevel gears defines also the convective heat transfer coefficient for spur gears that shows the convective coefficient increasing with the rotational speed and depends on lubricant properties, but shows the friction heat flux depending also of the cooling time that is inversely proportional to

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rotational speed.

By other side, in this work, near scuffing conditions the oils characteristics are similar, because the viscosity at bulk temperature is similar, and this could justify the constant value of the global heat transfer coefficient.

The bulk temperature (or mass temperature) is here assumed as the temperature of the tooth flank before teeth meshing occurs. It is also assumed that this temperature might be different below and above the pitch line, and this depends on tooth cooling after meshing. In fact, during the gear meshing cycle the power dissipated in the tooth flank is not equal below and above the pitch point, even in under steady state conditions. Furthermore, the teeth flanks are heated during a short period (the meshing time) and cooled during most of the total cycle time when heat convection plays an important role.

This approach assumes that under scuffing conditions the mass temperature or bulk temperature is high enough to reduce the oil viscosity to a certain level, where there are no conditions to generate a lubricating film. In order to validate this approach, oils without the extreme pressure and anti-wear additives were selected.

In a lubricated mechanism like the gears, some layers prevent that scuffing or other adhesive wear occurs. The tribo-films that are present in particular when there are anti-wear or extreme pressure additives improving the scuffing resistance. Understanding very well the influence of oil viscosity or the influence of the hydrodynamic role of the lubricant film in scuffing, demands starting to analyse the oils without this kind of additives. After this, is possible to introduce the additives, but now having a base that specifies which is the minimum capacity or scuffing resistance of this kind of oil. Now is possible to understand why this happens and is possible to identify the conditions above them the role of additives is crucial where the lubricant film is not sufficient to prevent scuffing.

2. Gear scuffing tests

In previous works [1,2], gear scuffing tests were performed on the FZG test rig [15], using type A and type C FZG gear teeth geometries and for wide ranges of the operating conditions: torque, tangential speed, base oil viscosity and bath oil temperature. Forty different tests were performed and very significant volume of experimental data was gathered concerning the scuffing load stage and the scuffing temperatures.

Table 1 shows the range of operating conditions used and Appendix 1 shows the operating conditions and the scuffing load stage of each test. Two FZG gear geometries were used: type A geometry, 20 and 10 mm wide, and type C, 14 mm wide [15]. Six base oils, with four different viscosity grades, were tested. Appendix 2 presents the most relevant characteristics of the gears and oils tested.

Four different test procedures are used:

(i) Increasing torque (at constant oil bath temperature and rotating speed),
(ii) Increasing bath oil temperature (at constant torque and rotating speed),
(iii) Increasing speed tests (at constant torque and oil bath temperature) and
(iv) Standard load carrying capacity tests, according to the DIN 51354 [16] or ISO 14635 [17].

The standard tests (increasing torque without cooling the oil bath) are unsuitable to evaluate the influence of each one of the parameters individually. There is almost total compatibility between these different test procedures, meaning that the results are independent of the test procedures.

Table 2 shows the range of three parameters usually important in scuffing analysis, under scuffing operating conditions. The maximum value of the product of the maximum Hertzian pressure by the sliding speed (P_0 x V_S), along the gear meshing line, the oil dynamic viscosity (η_0) at the bath oil temperature (T_0) and the minimum specific film thickness (Λ) along the gear meshing line.

The P_0 x V_S product has a large range in scuffing conditions, which is related to bath oil viscosity as can be observed in Fig. 1 a). Higher viscosity means higher scuffing load capacity. The specific film thickness, minimum value in the gear meshing line, calculated according Dowson and Higginson [18] using the thermal correction factor according Gohar [19] is another important parameter, clearly showing that the gears are operating under mixed film lubrication when scuffing occurs. In appendix 3 there is the definition of the specific film thickness and other variables related that are used in present work. Previous works [20,21], state the importance of this parameter in scuffing, but it is not sufficient to explain the scuffing behaviour, as can be observed in Fig. 1 b). However, for all tests with good running-in conditions, scuffing never occurs for $\Lambda > 1$.

Appendix 4 shows some common scuffing criteria, applied to the gear scuffing results, showing the limitations of these criteria.

3. Friction energy generation and mass temperature

3.1. Analysis in a point of the gear meshing line

During a large part of the gear meshing cycle, convection heat transfer from the gear teeth surfaces (at the mass temperature, T_{Mi}) to the oil bath (at the oil bath temperature, T_0) prevails. Starting from the global equation for convection heat transfer [22], and analysing the heat energy balance during the gear meshing cycle it is possible to relate the heat energy loss (HE) with the mass temperatures and the bath oil temperature, for each contact point (i):

$$HE_i = \alpha_{ht}\, A_i (T_{Mi} - T_0) t_T \tag{1}$$

where α_{ht} is the global heat transfer coefficient (mainly convection), A_i is the contact area and t_T is the total cycle time of the gear, different for the pinion and the wheel. The contact area is defined by the gear tooth width (L_w) multiplied by the length of contact ($2b_i$).

$$Ai = L_w \cdot (2b_i) \tag{2}$$

The friction energy generation, FE, in any point (i) of the gear meshing line is given by Ref. [11]:

$$FE_i = \mu_i\, Fn_i\, V_{Si}\, tc_i \tag{3}$$

where μ_i is the friction coefficient, F_{ni} is the normal force, V_{Si} is the sliding speed and tc_i is the contact time.

When the system reaches an energetic balance, in any point of gear meshing line, the heat generated by friction, FE, is equal to the heat energy loss, HE. Under these conditions it is possible to state that:

Table 1
Test conditions.

Torque (load)	up to 628Nm
Motor speed	500 to 3000 rpm (tangential speed, pitch point: 2,8 to 17,3 m/s)
Bath oil temperature	75 to 153 °C
Oil type	6 mineral base oils with 4 viscosity grades–68 to 680 cSt @ 40 °C
Gear geometry	type A (10 e 20 mm wide) and type C (14 mm)

Table 2
Parameters under scuffing conditions.

P_0V_S	2 to 15 GW/m^2 (maximum value in gear meshing line)
Bath oil viscosity	3,6 to 18,2 mPa s
Specific film thickness	0,23 to 1,14 (minimum value in gear meshing line)