GREASE LUBRICATION OF ELASTOHYDRODYNAMIC CONTACTS: A CRITICAL REVIEW

Luis Andrade-Ferreira
Dept. of Mechanical Engineering and Industrial Management, Faculty of Engineering
University of Porto - Porto, Portugal
lferreira@fe.up.pt

Keywords: Grease, Lubrication, Elastohydrodynamic, Rheology, Bearings.

ABSTRACT
In practical situations, a large proportion (about 80%) of all rolling bearings is lubricated by some sort of grease.
Several experimental studies have shown that greases present an erratic behaviour as a function of time, the measured film thickness varies significantly with the time of the experiment. Also, it was not possible to identify in a definitive way which are the lubricant rheological parameters that control grease lubrication.
One important feature that has been identified for grease lubrication is the reduction of grease in the inlet zone during the experiments, what leads to a starved film formation. It is important to realize that the formation of film thickness is completely different for fully flooded and starved regimes.
Because of all these problems with grease lubrication, it is not possible to know the correct value of the lubrication parameter in order to calculate the expected life of a grease lubricated rolling bearing. Usually, this expected life is calculated with the lubricant parameter of the base oil, then the obtained value is corrected considering the greater initial apparent viscosity of the grease and, most times, considering the reduction of film thickness for long running justified by the starved regime.
In this paper we propose to review the grease lubrication of elastohydrodynamic contacts, to make a critical analysis of the present status of this type of lubrication and to show some of the results achieved in theoretical and experimental studies.

1. INTRODUCTION
Greases are defined by the NLGI [1] as solid to semi-fluid products that result from the dispersion of a thickening agent in a lubricant fluid, to which can be added additives with particular properties.
Most of the greases use as lubricant fluid oil, mineral or synthetic, which represents 75 to 96% of its volume. Also, most of greases use as thickener some kind of soap that are metallic salts of fatty acids and represent 4 to 20% of their volume. Nowadays, more than 60% of the world production of greases is made with some sort of lithium soaps, conventional or complex (Todd [2]). There are also greases whose thickener is not a metallic soap, but clay or a polymer. The additives improve the greases lubricant properties, like their stability to oxidation, corrosion protection, anti-wear, surface adhesion, particle inhibitors, among others.
As they are convenient to use, greases lubricate more and more types of mechanisms and in almost all environments. Also, a large proportion (about 80% or more) of all rolling bearings are lubricated by some sort of grease.
Even if there is a growing interest on greases, one cannot say that there has been an important research effort on this subject. The fact is that there still is a lack of knowledge about its behavior in what concerns friction, film thickness formation or the useful life of the lubricant (Vernge [3], Cann [4]). There is not sufficient knowledge about the mechanisms of the film formation and how the grease feeds the contact, and what is the relationship between its composition and internal structure and its lubricant properties. That happens because greases are fragile materials, with a complex behavior when comparing with liquids (Ferreira [5]).
Also, the development of grease research has been inhibited by economical factors, the grease market is very small when compared with other lubricants.
However, the grease research is absolutely needed if one wants to understand how it works in an elastohydrodynamic (EHD) contact, as the one existing in a rolling bearing, because it is proved that the lubrication mechanisms are different for oils and for greases (Godfrey [6]). The original idea that the base oil was the sole lubricant and the solid phase would act as a reservoir (Bondi [7]) is abandoned for long.
In spite of all these difficulties, several research works have been developed to better understand the grease behavior in EHD contacts. Among these works one can distinguish those that try to relate the grease compo-
sition with their rheological properties, trying to define the fundamental parameters that rule their behavior, and those that for different material compositions it was studied the grease behavior in lubricated mechanisms that somehow simulated an EHD contact, measuring the film thickness and the friction force in fully flooded and starved contacts and its dependence on the time of the experiment. Also, it is worth to look at the works leading to the calculation of the useful life of grease in a rolling bearing.

2. RHEOLOGICAL BEHAVIOUR OF GREASES

As far as one can know, grease rheology has been the subject of several studies for very long. The knowledge of the rheological behavior is essential to know the ability of grease to circulate in a dispensing system, the friction force, and the film formation, among other important properties for lubrication. In 1951, ASTM ("American Society for Testing Materials") had introduced the penetration test by a cone (now ISO 2137 or ASTM D217) as the reference test for grease (ASTM [8]). As it is well known, by this test it is estimated the grease consistency, through the NLGI number and somehow it represents the rheological behavior of greases. However, its use and above all the interpretation of the obtained results have always been contested (Gow [9], Hammelid [10]). It is a test in which the analysis of all the parameters that can influence the results is very complex, in spite of the simplicity to perform it.

Also, it is not possible to establish a rheological law able to be extrapolated to practical situations. For a better knowledge of grease properties, several rheometric studies have been performed, considering the flow of greases in tubes ("Poiseuille" flow) or in cone and plate systems (constant stress) or in coaxial cylinders ("Couette" flow).

From all these studies it can be concluded that greases present a high non-newtonian behavior. Several rheological models have been tried to describe the grease behavior and among them the most used are shown in Table 1. Those rheological models were developed between 1950 and 1970.

### Table 1: Rheological models

<table>
<thead>
<tr>
<th>Model</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Herschel–Bulkey [11]</td>
<td>( \tau = \tau_\ell + \left( \mu_b \frac{dv}{dh} \right)^n )</td>
</tr>
<tr>
<td>Eyring and all [12]</td>
<td>( \tau = \mu_b \frac{dv}{dh} + \beta \sinh^{-1} \left( \frac{dv}{\lambda \frac{dv}{dh}} \right) )</td>
</tr>
<tr>
<td>Sisko [13]</td>
<td>( \tau = \mu_b \frac{dv}{dh} + \alpha \left( \frac{dv}{dh} \right)^n )</td>
</tr>
<tr>
<td>Bauer and all [14]</td>
<td>( \tau = \tau_\ell + \mu_b \frac{dv}{dh} + \alpha \left( \frac{dv}{dh} \right)^n )</td>
</tr>
</tbody>
</table>

where:  
- \( \tau_\ell \) - yield stress  
- \( \mu_b \) - dynamic viscosity of the base oil  
- \( \frac{dv}{dh} \) - strain rate  
- \( \alpha, \beta, \lambda, \alpha \) - lubricant parameters

None of these laws is able to describe the grease behavior in all lubricant situations. So, Newton’s formula is used many times, as:

\[ \tau = \mu_a \frac{dv}{dh} \]

where \( \mu_a \) is the apparent viscosity.

One must note that the value of \( \mu_a \) is time dependent when grease is submitted to deformation cycles, and \( \mu_a \) is also dependent on the shear rate and on the temperature.

In all these studies the viscoelastic, sometimes plastic, behavior of greases was noticed, with the appearance of normal stresses (Hutton [15]).

To characterize the viscoelastic phenomena of greases, rheometric measurements have been made to evaluate the elastic and viscous components of greases, measuring the complex shear modulus \( G^* \):
\[ \tau = G' + iG'' \]

where \( G' \) represents the elastic behavior and \( G'' \) the viscous behavior. The loss angle \( \delta \) allows the comparison between the two components:

\[ \tan \delta = \frac{G''}{G'} \]

If \( \tan \delta < 1 \) then the elastic component is predominant, if \( \tan \delta > 1 \) then the viscous component is the most important.

This type of tests, performed by different authors (Vergne [3], Godfrey [6], Hammelid [10], Jonkiz [11], Kersznan [17]), made possible to have an objective idea about the rheologic parameters that are more important to lubrication and what is the influence of the grease composition on these parameters, Table 2.

<table>
<thead>
<tr>
<th>Rheological parameter</th>
<th>Base oil viscosity</th>
<th>% of soap</th>
<th>Type of soap</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \tau_I )</td>
<td>0</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>( \mu_k ) at low ( (dv/dh) )</td>
<td>2</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>( \mu_k ) at high ( (dv/dh) )</td>
<td>0</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>( G_i )</td>
<td>0</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>( (dv/dh)_c )</td>
<td>0</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>( G' )</td>
<td>0</td>
<td>4</td>
<td>2</td>
</tr>
<tr>
<td>( G'' )</td>
<td>1</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>( \tau_{CO} )</td>
<td>1</td>
<td>3</td>
<td>4</td>
</tr>
</tbody>
</table>

where 0 means a negligible effect and 4 a very strong effect; \( \tau_I \) - yield stress; \( \mu_k \) - apparent viscosity; \( G_i \) - instantaneous elastic modulus; \( G_m \) - mean elastic modulus; \( (dv/dh)_c \) - critical elastic strain; \( G' \) - elastic modulus; \( G'' \) - dissipative modulus; \( \tau_{CO} \) - cross-over stress.

Another characteristic that is suggested is the “cross-over” stress, \( \tau_{CO} \), that corresponds to the transition between an elastic behavior to a viscous one, when \( G'' = G' \) (Hammelid [10]). The practical application of this parameter is not generalized yet.

From all these rheology studies come up that when the shear rate increases the grease behavior tends to become similar to the one of liquid (Couronne [16]).

3. SIMULATION IN EHD MACHINES

Several studies have been performed, simulating real elastohydrodynamic contacts. The experimental studies based on test performance with two disc machines and with different technologies (Dyson [18], Poon [19], Aihara [20]) have shown that in grease lubricated contacts that are reeved during the tests, the grease film thickness is at first greater if compared to the one obtained with the base oil, decreases with time and, in most situations, it stabilizes at about a value 30% less to the one obtained with the base oil, or at least to a value rather inferior to the initial value.

These values are confirmed by measurements made in real rolling bearings (Karbacher [21]). The final value of the grease film thickness depends on the grease composition: the thickener and the base oil and their percentage in volume.

Another type of tests was developed to understand what happens inside an EHD contact, making use of optical techniques based in interferometry. These techniques allow the measurement of the film thickness with very good accuracy. These tests confirmed the results obtained with the two disc machines, that is that film thickness of grease contacts is at first bigger than the one obtained with the base oils, then decreases with time (Cann [4], Ferreira [5], Muenich [22], Palacios [23], Cann [24]). If the contact is continually fed with grease, than the grease film thickness will always be greater then the one for base oil, behaving like an oil lubricant, where the film thickness is a function of the rolling speed (Hurley [25]). This suggests that the thinning of the grease film may be explained by starving phenomena, and also by the disruption of the grease structure.

Several attempts have been made to explain and model this behavior. So, the fact that greases are viscoelastic materials implies that important normal stresses come up in the inlet zone of the contact, where the lubricant film is formed, preventing the admission of grease into the contact (Hutton [15], Kempe [26]).

Infra-red spectroscopic analysis of the greased lubricated contacts have shown a complex behavior of greases in lubricant film formation, a kind of grease reservoir being formed around the track corresponding
to the contact zone (Hurley [25] and [27]). However, the formation of such track and reservoir, Figure 1, is dependent on the grease composition, above all it depends on the thickener.

\begin{figure}[h]
\centering
\includegraphics[width=0.8\textwidth]{GreaseReservoir.png}
\caption{Grease reservoir in grease lubricated EHD contact (Hurley [27]).}
\end{figure}

The starving phenomena in the grease lubricated contacts have been object of theoretical modeling for film thickness calculations. However, it is not possible to introduce this phenomena as a controlled parameter, because is time dependent. As it depends on grease composition, the contact feeding of grease can vary with time and, sometimes, this variation is somehow erratic (Chevalier [28], Jacob [29]).

Several experimental studies have shown in a fully flooded situation for the same basic composition, the greater the percentage in volume of the thickener, the greater is the film thickness (Kendall [30], Caan [31]). Also, when the resistance to structural degradation is higher, the behavior of the grease in the contact will be more stable and the film thickness will be greater (Hurley [32], Couronne [33]). These results confirm the practical results, that is the tested greases that give a higher regularity of film thickness and a greater film thickness are those that give a higher life in mechanisms.

Another important feature in EHD lubrication is the friction force generated in the contact. As it is well understood, the heat generated on the contact is dependent on the friction. It is also well known that the higher the temperature in the contact the faster will be the lubricant degradation through an excessive oxidation and in grease lubrication through the change of the chemical composition (Couronne [33]).

In the fully flooded situation, in pure rolling or in slip and rolling, greases give usually a lower friction coefficient than the respective base oils (Caan [34]). However, in the starved regime, it seems that the friction coefficient gets bigger with the time of the experiment and becomes greater then the one of the base oils (Kaneta [32]).

As the lubricants used in these results are quite different, these results must be taken with caution.

4. GREASE LIFE FORECAST
The grease lifetime calculation is of great importance, especially in what concerns to rolling bearings. The grease quality is essential for the reliability of mechanisms.

The roller bearing companies show tables where, as a function of the bearing type and the service conditions and environment, the optimized intervals for re-lubrication are given (e.g. FAG [36]). However, the methodologies to obtain this diagrams demand for the use of important experimental resources and are based on the statistical analysis of the experimental results. On the other hand, the extrapolation of the obtained results is very difficult or even impossible (Kroner [37]). The analysis of the results of tests performed with equipment for contact simulation still doesn’t permit a reliable extrapolation to practical situations (Ferreira [38] and [39]).

Also, it is very difficult to model the life of grease lubricated rolling bearings, because of the erratic behavior of greases. In practice the lubricant parameters are unknown. Usually the expected life is calculated with the base oil lubricant parameter, then its value is corrected taking into account the greater initial value for the film thickness and, in most situations, considering the film thickness reduction with time, based on the starving of the contact (STLE [40]).

The calculations for grease life are based on the equation developed by Booser (1974) [41], that in a generalized from trials to calculate the grease life taking into account the temperature gradient, speed, grease composition and the load that the rolling bearing supports. The equation is given as:
\[ \log L = -2.30 + \frac{2450}{273 + T} - 0.301S \]

where:
- \( L \) – geometric mean grease life for 50% bearing failures, h
- \( T \) – temperature (in °C) of the external ring
- \( S = S_G + S_N + S_W \) - half-life subtracting factor
- \( S_G \) - half-life subtracting factor for grease properties
- \( S_N \) - half-life subtracting factor for speed
- \( S_W \) - half-life subtracting factor for load-speed conditions

This equation has been improved through time, to better describe the working conditions of the mechanisms and the lubricant evolution (e.g. Kawamura [42]).

5. DISCUSSION
Grease lubrication still is more an art then a truly science. Nowadays, there still is lack of knowledge about how the lubricant feeds the contact.
However, this situation is about to change, as the number of applications of grease as a lubricant is increasing rapidly. The relubrication systems are relatively simple, at relatively low cost and there is the possibility to formulate a great variety of different lubricants, answering to the user needs.
The employment of grease centralized dispensing systems is also being improved and they are more simple then the ones for liquid lubricants (Litters [43]). So, it is needed to develop the necessary scientific work that will allow, from simple and reliable product characterization tests, to formulate greases as a function of the required performance. This scientific work has only begun, as it was shown in this paper.

6. CONCLUSION
Greases are more and more attractive for lubrication mechanisms. But:
- Only now we begin to understand the need to know the relationship between the grease composition and its performance, in what concerns the film thickness, the friction force in the contact and the life of grease lubricated mechanisms.
- The traditional evaluation of greases by measuring its consistency is insufficient.
- It is necessary to develop simple methodologies to analyze the grease behavior whose results can be readily extrapolated to real contacts.

7. BIBLIOGRAPHY


[36]. FAG, Rolling Bearings. Cat. WL 41 520/2 EA, 1996


