Publication metadata

Title: Influence of thermo-mechanical properties of coatings on friction in elastohydrodynamic lubricated contacts.

Author(s): W. Habchi

Journal: Tribology International

DOI/Link: https://doi.org/10.1016/j.triboint.2015.04.020

Influence of Thermo-Mechanical Properties of Coatings on Friction in Elastohydrodynamic Lubricated Contacts

W. Habchi*

Lebanese American University, Department of Industrial and Mechanical Engineering, Byblos, Lebanon

*Corresponding author: wassim.habchi@lau.edu.lb

Abstract

This paper presents a numerical investigation of the influence of thermo-mechanical properties of coatings on friction in elastohydrodynamic contacts. In a previous work by the author, it was shown that thermal properties of coatings had a significant influence on friction. In fact, under high sliding speeds, friction was found to increase with the thermal inertia of coatings. The current work reveals that mechanical properties of coatings also have a significant impact on friction. Friction actually exhibits an increase with the rigidity of the coating. Furthermore, a combination of soft coatings with low thermal inertia is shown to maximize friction reduction while hard coatings with high thermal inertia maximize friction increase. These effects are found to increase with the coating thickness.

Keywords: Elastohydrodynamic Lubrication; Surface Coatings; Thermo-mechanical properties; Friction

1. Introduction

The influence of surface coatings on the performance of elastohydrodynamic lubricated (EHL) contacts has been a subject of interest for the tribological community over the last few decades. From the earliest works on the topic, the focus was on the effect of mechanical properties of coatings on pressure, film thickness and stress distribution within the solid components of these contacts. The early numerical models for coated EHL contacts assumed rigid substrates and elastic coatings such as the work of Bennett and Higginson [1] or Elsharkawy and Hamrock [2] [3] for the case of isothermal line contacts or also the work of Jin [4] [5] for isothermal circular contacts. One of the first numerical models to account for the elasticity of the substrates in the elastic deformation calculation for the solid components is that of Elsharkawy et al. [6] followed by that of Liu et al. [7] [8] or also more recently that of Wang et al. [9]. All of the aforementioned works assumed isothermal operating conditions and focused on the effects of the coating's rigidity and its thickness on lubrication performance. It was found that stiffer (or harder) coatings lead to increased central contact pressure and pressure spike and reduced contact area. The reverse effect was observed with soft coatings. It was also found that all these effects increase with the thickness of the coating layer. In addition to considering the effects of the mechanical properties of coatings on pressure and film thickness, Fujino et al. [10] and Chu et al. [11] also looked at the influence on stress distribution within the contacting solid elements. The former considered standard rolling conditions whereas the latter assumed a pure squeeze motion. Both works reached the same conclusion that for an enhanced fatigue life of the components, thicker and softer coatings are preferable as they lead to lower maximum stresses within the solids. This being said, surface coatings for elastohydrodynamic lubrication applications have always been selected based on their mechanical properties so as to reduce contact pressures and stresses and increase film thickness with the purpose of attaining an improved fatigue life for the corresponding components.

Interest in selecting coatings so as to control friction in EHL contacts has only appeared recently after several experimental works reported reduced friction in Diamond-Like-Carbon (DLC) coated EHL contacts such as Evans et al. [12] or Kalin et al. [13] [14]. Originally, it was thought that the observed friction reduction is a consequence of boundary slip at the lubricant-solid interfaces. However, Björling et al. [15] reported friction reduction in their measurements done on DLC coated contacts with operating conditions under which boundary slip is very unlikely to occur. This led them to assume that the root cause for friction reduction might be thermal effects within the lubricating film. This assumption was later verified in [16] by validating the observed friction reduction measurements against numerical results. The employed numerical model did not include any boundary slip effects. It is based on the finite element full-system approach for thermal elastohydrodynamic lubrication developed by Habchi et al. [17]. Details of the incorporation of the surface coatings in the finite element model developed by the author for thermal non-Newtonian EHL contacts can be found in [18]. In this work, the author showed that friction in EHL contacts may be controlled by a suitable choice of surface coatings based on the thermal properties of their material. It was found that low thermal inertia coatings can significantly reduce friction and high thermal inertia coatings increase it under high sliding speed conditions. These effects were shown to increase with the coating thickness which was later verified by the experiments of Björling et al. [19]. The origins of these findings were investigated in [20] and it was found that the underlying mechanisms are purely thermal. In fact, a low thermal inertia coating acts as an insulator, trapping the generated heat within the lubricant film and confining it to the central area of the contact. This leads to an overall temperature increase within the central part of the film. The temperature rise is associated with a viscosity decrease leading to reduced friction coefficients. The exact opposite takes place when high thermal inertia coatings are employed leading to increased friction. Most importantly, since temperature variations are restricted to the central area of the contact and do not propagate towards the inlet, friction variations were attained without any noticeable effect on film thickness.

In [20], only the influence of thermal properties of coatings on friction in EHL contacts was investigated along with the underlying heat transfer mechanisms. The current work investigates the possibility of controlling friction by a suitable choice of coatings based on the mechanical properties of their material or also the combination of mechanical and thermal (or thermo-mechanical) properties. The underlying physical mechanisms governing friction variations are also investigated in detail. Next, the numerical model employed in this work is briefly recalled.

2. Numerical Model Description

The numerical model employed in this work has been described in detail in [18]. Here, only its main features are recalled. The model is based on the finite element full-system approach introduced by Habchi et al. [17] for the solution of the thermal non-Newtonian EHL problem under steady-state considerations. The problem is reduced to that of an equivalent contact between a ball of radius *R* and a flat plane subject to an external applied load *F* as shown in Figure 1. Solid surfaces are assumed to be smooth and moving at constant unidirectional velocities in the x-direction.

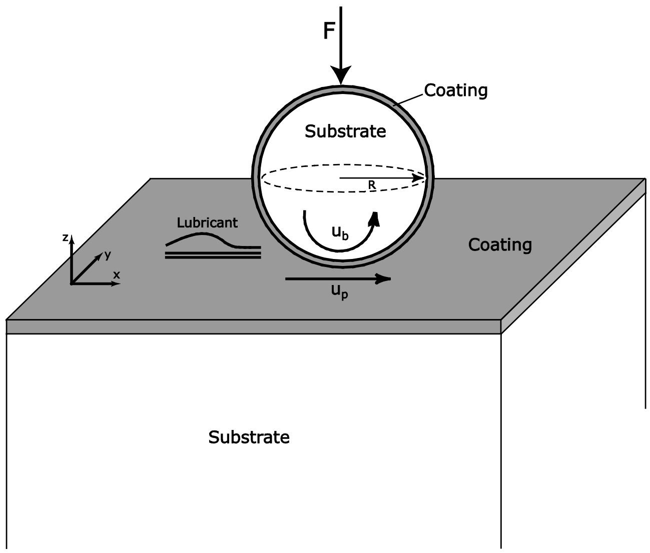

Figure 1: Geometry of the coated TEHD circular contact

The numerical model entails two main sub-problems: EHL and thermal. The first consists in solving the *2D* generalized Reynolds equation [21] applied to the fluid domain in the contact area, the *3D* linear elasticity equation applied to the solid domains (substrates + coatings) and the load balance equation in a full-system fashion (simultaneously). The second problem consists in solving the *3D* energy equation applied to the solid components (substrates + coatings) and the lubricant film. All equations are discretized using non-structured finite elements. The mesh size is specifically adjusted to meet the precision requirements of the TEHD problem. That is, sufficiently small size elements are employed in the central part of the contact and the mesh size continuously increases with the distance from the central region. Lagrange second order tetrahedral elements are used for the 3D thermal problem and the 3D linear elasticity part of the EHL problem while Lagrange fifth order triangular elements are used for the 2D hydrodynamic part. The use of higher order elements for the hydrodynamic part, as an alternative to refining the mesh, allows having a good precision for its solution without inducing any unnecessary increase in the number of elements of the 3D elastic problem. In fact, the 2D mesh used in the contact area where the hydrodynamic equation is applied is nothing else but the projection of the 3D tetrahedral elements (employed for the solution of the elastic part) on that surface. The boundary conditions for the generalized Reynolds equation are zero-pressure on the boundaries of the contact area and zero-pressure as well as zero-pressure-gradient on the free exit boundary. The treatment of the latter is handled by the use of a penalty method. The pressure field generated within the lubricant film is used to define a normal pressure load boundary condition on the surface of the solid domain for the solution of the elastic part. In addition, displacement as well as stress continuity conditions are imposed across the coating-substrate interface. As for the thermal problem, an ambient temperature boundary condition is applied to all inlet boundaries. Temperature as well as heat flux continuity conditions are imposed on all solid-solid and solid-lubricant interfaces. For further details, the interested reader is referred to [18].

The overall numerical procedure consists in defining an initial guess for dimensionless pressure P (with its corresponding film thickness H) and temperature distribution T. Starting with the initial guess, the EHL problem is solved for a given fixed temperature distribution defined by the initial guess. Then, the thermal problem is solved for given pressure and film thickness profiles obtained from the solution of the EHL part. The obtained temperature solution is used to update the initial guess for T and the entire procedure is repeated until convergence is attained.

3. Friction in Coated EHL Contacts

The lubricant selected for this work is a mineral oil (Shell T9) which properties have been characterized in [22]. Its viscosity and density have been measured as a function of pressure, temperature (and shear stress for the former) and appropriate rheological models were derived. The dependence of its thermal properties (thermal conductivity and heat capacity) on pressure and temperature has also been characterized and fitted to adequate mathematical expressions. These expressions and the rheological models will not be detailed here for brevity. Only the ambient pressure and temperature values for the different properties are reported in Table 1. The interested reader is referred to [22] for further details. It is important to mention however that this lubricant has been successfully used without any alteration of its measured rheological and transport properties in predicting friction and the results were matched against experiments [22]. This makes it a good candidate to be used in the current work. In [22], Habchi et al. highlighted the importance of accounting for the pressure-temperature dependence of the lubricant's thermal properties for an accurate prediction of friction. In fact, they showed that failing to account for this dependence leads to an underestimation of friction in the "thermoviscous regime" [23].

Operating conditions	Lubricant Properties	Solid material properties			
T_0=303K	μ_0=0.0135Pa.s	Substrate	Coatings		
u_m=1m/s	α^*=20GPa^{-1}	E_s=210GPa	Mechanical properties	Thermal properties	
SRR=0.0-0.5	ρ_0=872kg/m^3	υ_s=0.3	$E_c = \begin{cases} 105\,\text{GPa (Soft)} \\ 420\,\text{GPa (Hard)} \end{cases}$	Low I	High I
F=25;100N	k_0=0.11W/m.K	ρ_s=7850kg/m^3		ρ_c=3500kg/m^3	ρ_c=10000kg/m^3
R=12.7mm	c_0=1790J/kg.K	k_s=46W/m.K		k_c=5W/m.K	k_c=90W/m.K
t_c=20;40;80;160μm		c_s=470J/kg.K	υ_c=0.3	c_c=200J/kg.K	c_c=1000J/kg.K

Table 1: Operating conditions, lubricant properties and solid material properties

The operating conditions as well as the solid material properties (substrates + coatings) are summarized in Table 1. Note that the considered range of slide-to-roll ratios (SRR) is 0 - 50% since in a non-controlled mechanical system, such as a ball bearing for instance, SRR rarely exceeds 50% as one of the contacting surfaces is driven by the other by traction at a relatively similar surface speed. For the coatings, two categories are considered based on mechanical properties: “Hard” and “Soft”. The former has a Young’s modulus that is greater than that of the substrate ($E_c = 2E_s$) while that of the latter is lower ($E_c = E_s/2$). Two other categories are also considered based on thermal properties, particularly thermal inertia $I = \sqrt{kC}$: “Low I” and “High I”. The former has a lower thermal inertia than that of the substrate while that of the latter is higher. The reader is reminded that thermal inertia is a property that characterizes the ability of a material to transport heat by conduction and advection. In fact, thermal conductivity k characterizes the ability of a material to transport heat by conduction whereas volumetric heat capacity $C = \rho c$ corresponds to its ability to store and transport heat by advection (heat transport by mass). Note that the thermal properties of the materials selected for this work are not intended to represent any specific existing materials but were rather selected to provide a range of values of thermal inertia, which is the relevant property when it comes to heat removal from the contact [20]. Similarly for the mechanical properties, these were simply chosen to cover a range of values for coating rigidity which governs the central contact pressure. The latter is of significant relevance when it comes to friction generation, as shall be discussed later. Seven different coating configurations are used in the following:

- “Uncoated”: Mechanical and thermal properties similar to substrate
- “Soft”: Soft with thermal properties similar to substrate
- “Hard”: Hard with thermal properties similar to substrate
- “Low I”: Low thermal inertia with mechanical properties similar to substrate
- “High I”: High thermal inertia with mechanical properties similar to substrate
- “Soft + Low I”: Soft with low thermal inertia
- “Hard + High I”: Hard with high thermal inertia

Coatings are assumed to be bonded to the substrates and all solid materials are considered to be elastic. Thin coatings have been investigated by the author and his collaborators in [16] and [20]. These are only successful in influencing friction owing to their thermal properties. However, the effect of mechanical properties of coatings on friction is negligible for thin coatings [16]. This is why for the current work, which aim is to investigate the effects of mechanical or thermo-mechanical properties of coatings on friction, a range of relatively thick coatings has been considered so as to emphasize the effect of coating rigidity on friction. Both contacting elements (ball and plane) are taken to have the same material properties. That is, both substrates are similar as well as both coatings. In addition, for coatings, the same thickness is employed on both contacting surfaces. The “Uncoated” configuration will be used as a reference whereas the “Soft” and “Hard” configurations allow isolating the effects of mechanical properties of coatings on friction. The “Low I” and “High I” allow isolating the effects of thermal properties and finally the “Soft + Low I” and “Hard + High I” will provide an insight on the combined effects of thermal and mechanical properties of coatings on friction. The “Soft + High I” and “Hard + Low I” configurations will not be considered in the following since in practice, it is either desired to increase friction or decrease it as much as possible, depending on the application at hand. Therefore, if friction is to be controlled by a suitable choice of surface coating, it wouldn't make sense to choose a coating material with thermal properties conflicting with mechanical properties in terms of friction influence as shall be discussed later.

3.1.Influence of Thermal Properties of Coatings

The influence of coatings' thermal properties on friction in coated EHL contacts has been investigated in [18]. It was found that a "Low I" coating leads to lower friction coefficients whereas a "High I" one increases friction in the "thermoviscous regime" [23] (under high sliding speeds). This effect was shown to increase with coating thickness but a limiting value was observed. In fact, beyond a coating thickness of t_c=80μm, friction coefficients were not further influenced. The author later showed in [20] that the origins of these observations are purely thermal. A low thermal inertia coating acts as an insulator, trapping the generated heat within the lubricant film and confining it to the central area of the contact. This leads to an overall temperature increase within the central part of the film. The temperature rise is associated with a viscosity decrease leading to reduced friction coefficients. The exact opposite takes place when high thermal inertia coatings are employed leading to increased friction. A very important aspect of these observations is the fact that temperature variations are restricted to the central part of the contact and do not propagate towards the inlet. As such, friction variations are attained without any noticeable effect on film thickness. The detailed results and analysis of this study will not be repeated here as the main objective of this work is to study the influence of coatings' mechanical properties, or the combined effects of thermal and mechanical properties on friction. The interested reader is referred to [18] and [20] for more details about the influence of thermal properties of coatings on friction.

3.2.Influence of Mechanical Properties of Coatings

The influence of the mechanical properties of coatings' material on friction is shown in Figure 2. Friction coefficients for the "Soft" and "Hard" coating configurations are reported as a function of slide-to-roll ratio (SRR) for a coating thickness of 40μm. The "Uncoated" case is also shown as a reference for comparison.

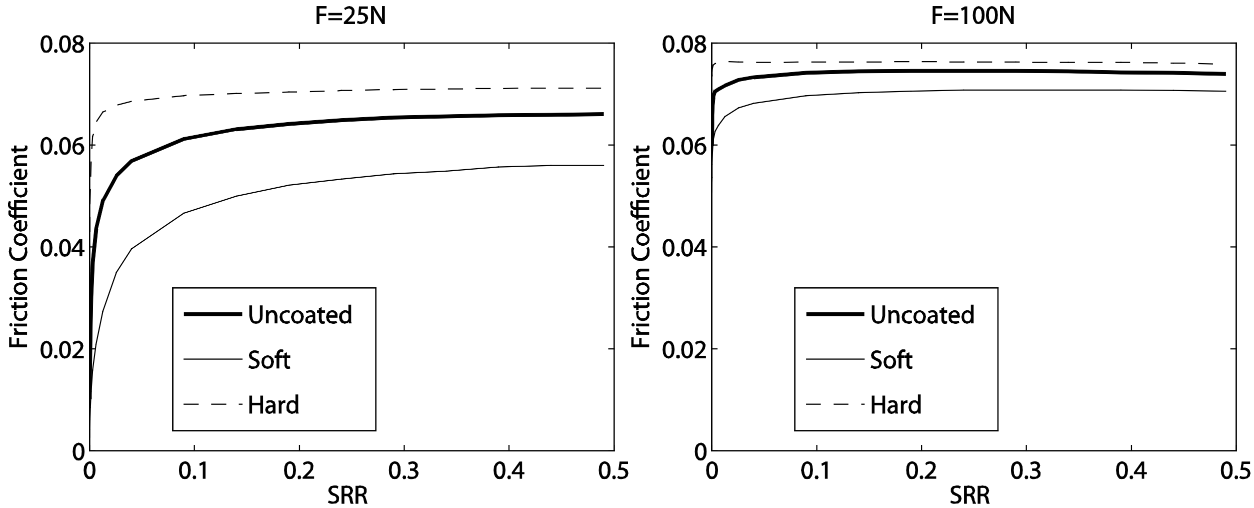

Figure 2: Effect of coating's mechanical properties on friction in coated circular TEHD contacts (t_c=40μm)

Figure 2 clearly shows that a soft coating leads to reduced friction whereas hard coatings tend to increase friction. In other words, friction coefficients increase with the rigidity of the employed surface coating. Moreover, friction coefficients are affected over the entire considered range of SRR. The deviations from the uncoated case are not restricted to any particular friction regime. For a detailed discussion on friction regimes and their significance, the reader is referred to [23]. Also note that for the higher load case (F=100N) friction is less affected by the mechanical properties of coatings. The underlying physical mechanisms behind the observations reported here will be discussed later.

3.3. Influence of Thermo-Mechanical Properties of Coatings

The influence of the thermo-mechanical properties of coatings' material on friction is shown in Figures 3 and 4. Figure 3 shows the "friction-reducing" coating configurations while the "friction increasing" ones are reported in Figure 4.

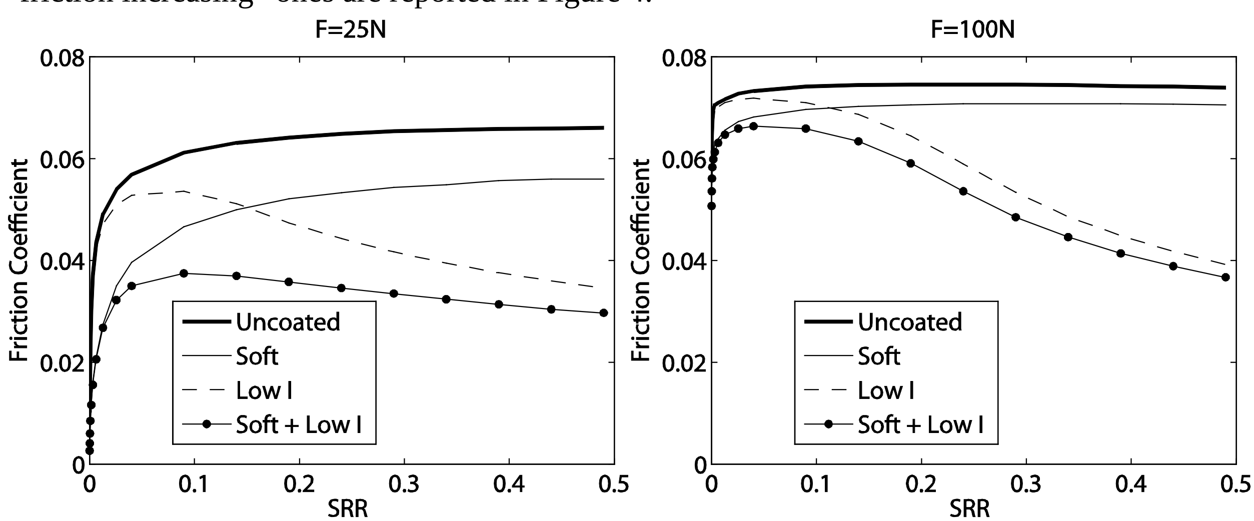

Figure 3: Effect of soft coating's thermo-mechanical properties on friction in coated circular TEHD contacts (t_c=40μm)

Figure 3 shows the friction curves for the combination of soft coatings with a low thermal inertia (“Soft + Low I”). For comparison, the “Uncoated”, “Soft” and “Low I” configurations are also reported. The considered coating thickness is 40μm for all these cases. Note that both the “Soft” and “Low I” configurations reduce friction (as discussed earlier) but in a different way. In the “Low I” case, friction is only reduced in the “thermoviscous regime” while for the “Soft” configuration, friction is affected over the entire considered range of SRR. As it would be expected, the “Soft + Low I” configuration combines both effects leading to a significant friction decrease over the entire range of SRR. Also, the extent of friction decrease exceeds the individual reductions observed for the “Soft” and “Low I” cases separately reaching around 50% over a significant portion of the considered range of operating conditions.

Figure 4 shows the friction curves for the combination of hard coatings with a high thermal inertia (“Hard + High I”). For comparison, the “Uncoated”, “Hard” and “High I” configurations are also reported. The considered coating thickness is also 40μm for all these cases.

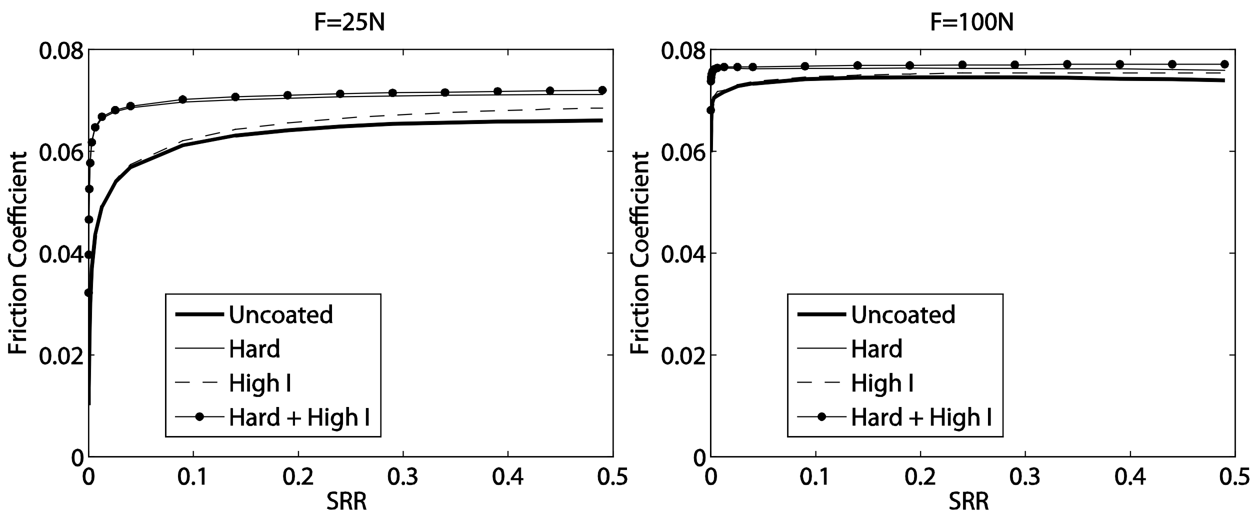

Figure 4: Effect of hard coating’s thermo-mechanical properties on friction in coated circular TEHD contacts (t_c=40μm)

First, note that similarly to what is observed in Figure 3, the effects of thermal properties of coatings on friction (“High I”) are restricted to the “thermoviscous regime” while those of mechanical properties (“Hard”) extend to the entire range of SRR. However, in this case friction is increased. As for the “friction-reducing” cases, the extent of friction increase for the “Hard + High I” case exceeds the individual increases observed for the “Hard” and “High I” cases separately.

3.4.Influence of Coating Thickness

The influence of the thickness of coating layers on friction is shown in Figure 5 for all coating configurations considered in this work under an external applied load of 25N. Friction curves are reported for all considered coating thicknesses and for comparison purposes, the “Uncoated” case is shown in all figures.

For the “Soft” configuration, friction continuously decreases with increasing coating thickness over the entire considered range of SRR. As for the “Hard” configuration, friction exhibits an increase with increased coating thickness; though in this case, the effects are more pronounced in the “linear” and “non-linear viscous” regimes [23] (at low to moderate SRR) than they are in the “Plateau” regime where friction reaches an asymptotic value (the “thermoviscous” regime where friction starts decreasing with increasing SRR is not reached in this particular case).

For the “Low I” case, as mentioned earlier, in the “thermoviscous” regime, friction decreases with increasing coating thickness but a limiting value is reached when the thickness exceeds 80μm. As for “High I” coatings, though friction variations are less pronounced (for reasons that will be discussed later), friction increases with coating thickness in the “thermoviscous” regime, but a limiting value is also observed.

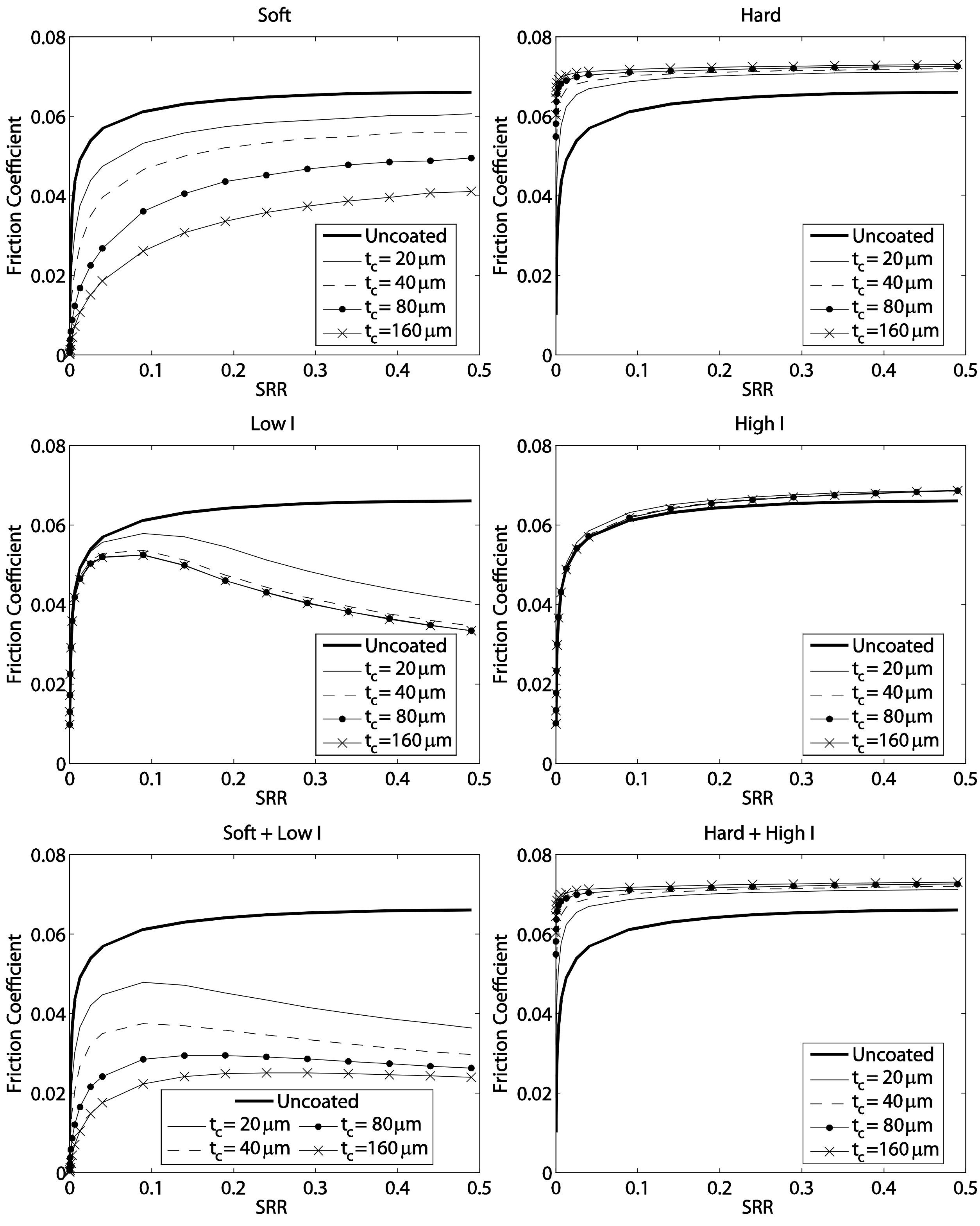

Figure 5: Effect of coating thickness on friction in coated circular TEHD contacts (F=25N)

Finally, for the "Soft + Low I" case, friction is also continuously reduced with increasing coating thickness over the entire considered range of SRR. As observed in the previous section, for a given coating thickness, the extent of friction decrease in this case is more pronounced than individual reductions observed for the “Soft” and “Low I” cases separately reaching about 64% in the most extreme case (t_c=160μm and SRR=0.5). As for the “Hard + High I” configuration, friction increases with coating thickness and as with the “Hard” case, variations are more pronounced in the “linear” and “non-linear viscous” regimes than in the “Plateau” regime where the frictional response of the contact is governed by the limiting-shear-stress behavior of the lubricant. Also note that friction variations are less pronounced than in the “Soft + Low I” case. In fact, in the most extreme case (t_c=160μm and SRR=0.5) friction is increased by about 11% with respect to the “Uncoated” configuration. But as noted earlier, the combined influence of thermal and mechanical properties of coatings in the “Hard + High I” case exceeds the individual effects of “Hard” and “High I” configurations taken separately.

4. Discussion

The physical mechanisms behind the observations made in the previous section are investigated here. As far as coating thermal inertia effects are concerned, the underlying thermal mechanisms have been thoroughly discussed in [20] and the focus in this section will be turned towards the effects of the mechanical properties of coatings or the combined effects of thermal and mechanical properties.

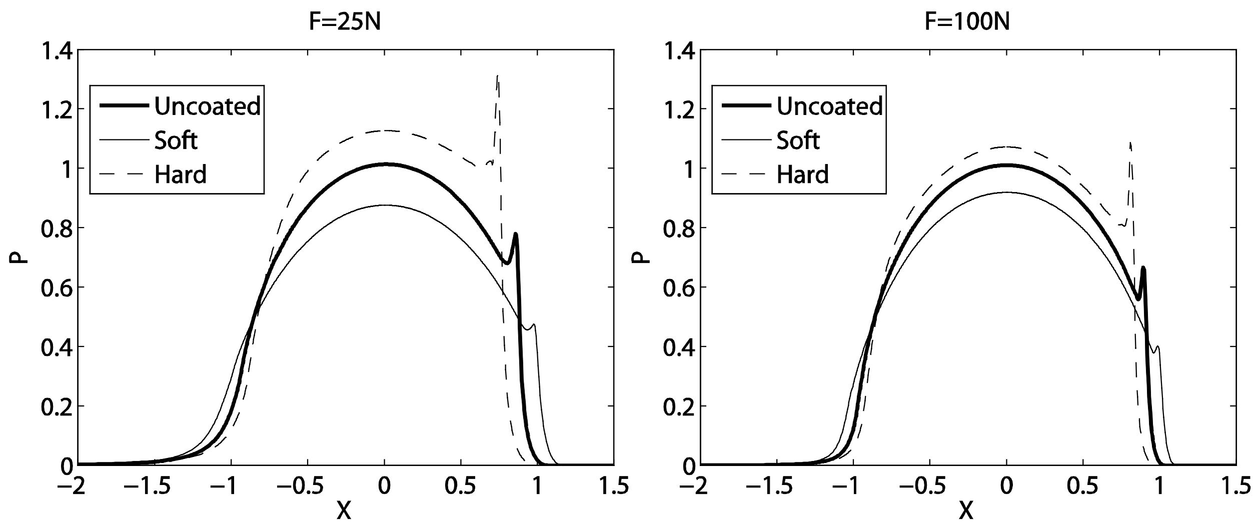

Figure 6: Influence of coating’s mechanical properties on dimensionless pressure distribution along the central line of the contact in the *x*-direction for t_c=40μm and SRR=0.5 (left: F=25N; right: F=100N)

First, in order to understand why friction coefficients increase with the rigidity of the employed surface coating, it is essential to have a look at pressure distribution in these contacts and how it is affected by the rigidity of the coating. Figure 6 shows the dimensionless pressure distribution along the central line of the contact in the *x*-direction for t_c=40μm and SRR=0.5 for the “Soft” and “Hard” coating configurations. The “Uncoated” case is also shown for comparison. Clearly, the central pressure as well as the pressure spike height increase with the rigidity of the coating while contact area is reduced. These observations are in agreement with the classical coated EHL literature [1][2][3][4][5][6][7][8][9]. At first glance, one might be misled to think that friction coefficients should not be affected as the effect of pressure increase (leading to an increase in lubricant viscosity) should be offset by the reduction in the contact area. However, friction depends linearly on the contact area (all other parameters being the same) whereas viscosity-pressure dependence is usually exponential. As such, pressure variations have a stronger impact on friction than the extent of the contact area. Therefore, a soft coating which leads to reduced contact pressures (and corresponding reduced lubricant viscosities and shear stresses) would lead to reduced friction. The exact opposite holds for hard coatings.

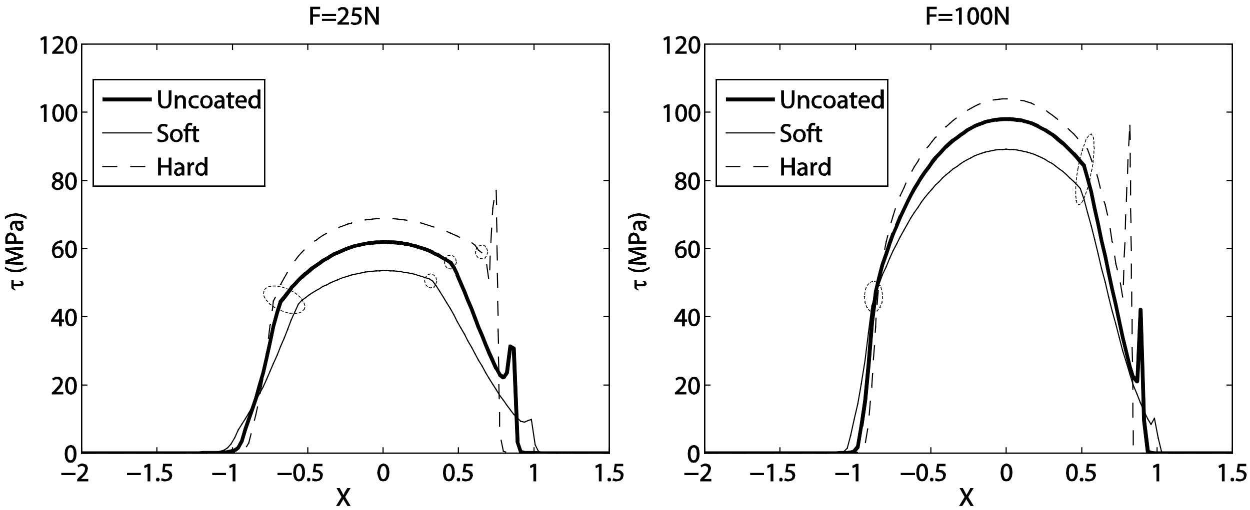

Figure 7: Influence of coating's mechanical properties on shear stress distribution along the central line of the contact in the middle-layer of the lubricant film in the *x*-direction for t_c=40μm and SRR=0.5 (left: F=25N; right: F=100N)

The shear stress distributions along the central line of the contact in the middle-layer of the lubricant film in the *x*-direction for the cases considered in Figure 6 are shown in Figure 7. Obviously, shear stresses increase with coating rigidity leading to increased friction coefficients. The dashed ellipses in Figure 7 mark the onset and offset of the limiting-shear-stress (LSS) mechanism characterized by a discontinuity in the gradient of the shear stress distribution. In fact, in regions of the contact where LSS is reached, shear stress dependence on pressure becomes linear unlike the remainder of the contact. In fact, as long as LSS is not reached, shear stresses vary linearly with viscosity which has an exponential dependence on pressure. Note that the extent of the contact region where LSS is reached increases with coating rigidity (though the contact area is reduced) because of the increased pressures. These lead to higher lubricant viscosities and as a consequence, higher shear stress levels are attained exceeding the LSS over a wider region of the contact. This explains why friction coefficients are less influenced for the "Hard" or "Hard + High I" configurations than they are for the "Soft" or "Soft + Low I" ones. The LSS region is also larger for the 100N load case than the 25N one which explains why friction coefficients exhibit less variation in the former case. The mild friction variations observed in the "High I" case of Figure 5 are attributed to a similar effect as discussed in [20]. In fact, in this case, the extent of the LSS region also becomes larger than in the "Low I" case. However, the reason in this case is temperature-related rather than pressure-related. In fact, the higher thermal inertia coating material leads to reduced contact temperatures. These lead to increased lubricant viscosity and as a consequence shear stress which exceeds the LSS over a wider region of the contact.

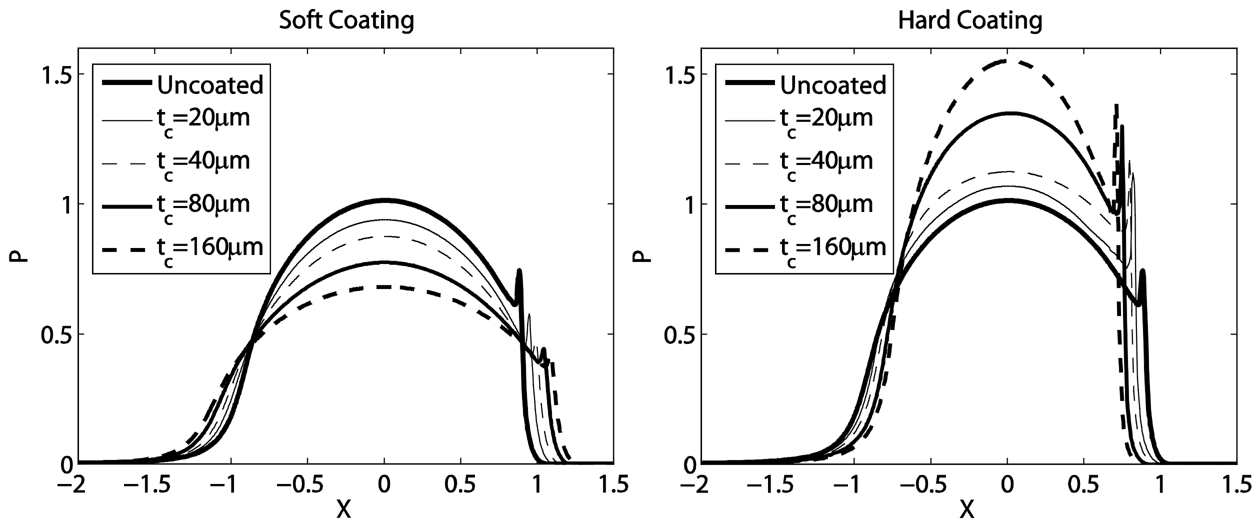

Figure 8: Influence of coating thickness on dimensionless pressure distribution along the central line of the contact in the *x*-direction for F=25N and SRR=0.5 (left: "Soft" coating; right: "Hard" coating)

Next, in order to understand how the above discussed effects are influenced by the coating thickness, it is sufficient to have a look at the pressure profiles obtained for different coating thicknesses. These are reported in Figure 8 for the case F=25N and SRR=0.5 along with the "Uncoated" case (shown for comparison). Obviously, pressure distributions exhibit more variations with increased coating thickness. Given the pressure-friction relationship discussed earlier, the stronger pressure is affected the more friction coefficients will be influenced (as evidenced by the friction curves of Figure 5). Though a limiting friction value is not observed in this case when coating thickness is increased, it is to be expected that beyond a certain thickness, friction would no longer be affected. This would be reached when the coating itself becomes a half-space. Then, any additional increase in thickness would no longer affect the pressure distribution and as a consequence friction. But in practical engineering applications, coating thicknesses of such magnitude are never employed. In the case of thermal inertia effects, a limiting value for friction was observed beyond a coating thickness of 80μm as shown in Figure 5 ("Low I" and "High I" cases). This is simply because the thermal boundary layer has a thickness that does not exceed 80μm and as a consequence it lies entirely within the coating. Therefore, any additional increase in the coating thickness would have no impact on the temperature distribution within the lubricant film which is the root cause for friction variations in this case. In order to illustrate this point, the penetration depth of heat within the solids is shown in Figure 9 (for the ball) for the "Low I" and "High I" cases with different coating thicknesses. Note that Z=0 corresponds to the ball's surface and that the coating-substrate interface is located at Z=1 and that in the ratio T/T_0, both temperatures are taken in Kelvin. Clearly, for coating thicknesses exceeding 80μm the thermal boundary layer of the solids is entirely contained within the coating and heat no longer reaches the substrate which remains at an ambient temperature T_0.

Note that the temperature at a given depth for the cases t_c=80μm and t_c=160μm is the same even though the corresponding temperature profiles in Figure 9 do not overlap. This is simply due to the definition of the dimensionless space coordinate *Z* ($Z = z/t_c$ in the coating and $Z = z/a$ in the substrate) which is function of the coating thickness. The corresponding dimensionless temperature distributions across the lubricant film are shown in Figure 10. Note that *Z*=0 corresponds to the plane's surface while *Z*=1 corresponds to the ball's surface ($Z = z/h$ within the lubricant film). As expected, beyond a coating thickness of 80μm, the lubricant film temperature is no longer influenced leading to the limiting friction values observed in Figure 5.

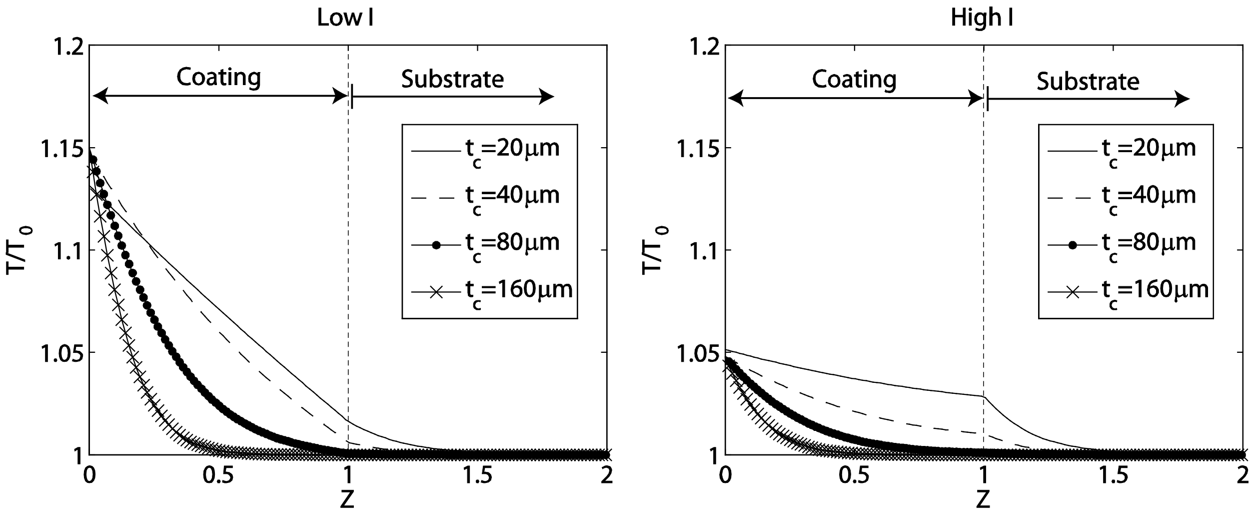

Figure 9: Temperature distribution within the ball in the *z*-direction along a line through the contact center for F=25N and SRR=0.5

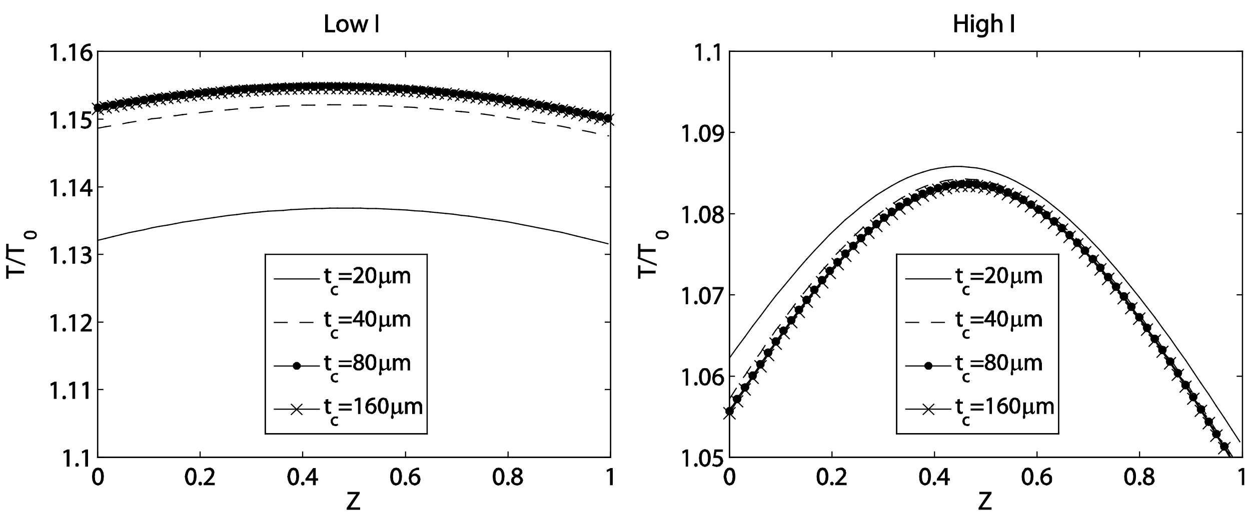

Figure 10: Temperature distribution across the lubricant film thickness in the *z*-direction along a line through the contact center for F=25N and SRR=0.5

The interpretation of the effects of coating rigidity on friction in coated EHL contacts discussed in this section holds for any slide-to-roll ratio as the latter has very little influence on pressure distribution (all other parameters being the same) which is the root cause of friction variations. This explains why friction variations associated with coating rigidity span over the entire considered range of SRR contrarily to those associated with thermal inertia. In fact, as stated previously, coating thermal inertia effects on friction are restricted to the “thermoviscous regime” (at high SRR). This is because these originate from thermal effects which manifest and become significant or dominant only at high sliding speeds.

To summarize, soft coatings reduce friction in EHL contacts as a consequence of reduced pressure and thus shear stresses. The opposite is observed with hard coatings. Low thermal inertia coatings reduce friction as a consequence of increased contact temperatures. The opposite holds for high thermal inertia coatings. A combination of soft and low thermal inertia coating material obviously leads to a superposition of corresponding friction-reducing mechanisms. As such, friction reductions exceed the individual reductions observed with “Soft” or “Low I” configurations taken separately. Similarly, a combination of hard and high thermal inertia coating material leads to a superposition of corresponding friction-increasing mechanisms. As such, friction increase exceeds the individual increase observed with “Hard” or “High I” configurations taken separately. All these effects increase with coating thickness and most importantly, all are achieved without any significant influence on film thickness. In fact, in the most extreme case considered here, film thickness variations do not exceed 5%. Usually, central film thicknesses are barely affected by the addition of coating layers, and film thickness variations in coated EHL contacts with respect to uncoated ones (though relatively small) are observed in the vicinity of the minimum film thickness [18]. Given that the minimum film thickness for the uncoated case with F=25N is about 68nm and for the 100N case it is about 48nm, the maximum variation in film thickness does not exceed 5nm in absolute value in the most extreme case considered here. This is because thermal inertia effects are confined to the central region of the contact and temperature variations do not propagate towards the inlet as detailed in [20]. Therefore, lubricant film build-up which is governed by the hydrodynamic behavior of the lubricant in the contact inlet is not affected. As for the effects of the mechanical properties of coatings, these are also known not to significantly affect lubricant film thickness (see for instance [2], [5] and [18]).

5. Conclusion

This work investigates the effects of thermo-mechanical properties of coatings on friction in EHL contacts. It is found that friction coefficients increase with the rigidity of the employed surface coating irrespective of the friction regime. The underlying mechanism is found to be an increase in contact pressure with the coating rigidity leading to increased lubricant viscosity and as a consequence increased shear stress and friction. In an earlier work by the author [18], friction was also found to increase with the thermal inertia of the coating. However, it was found in [20] that thermal effects are the root cause of friction variations in this case and that these are restricted to high sliding speeds. In the current work it is shown that a combination of “friction-reducing” thermal and mechanical coating properties (soft with low thermal inertia) leads to significantly more pronounced friction reductions than in the corresponding individual cases irrespective of the friction regime. Friction reduction as high as 64% is attained in the most extreme case considered. The same was found for a combination of "friction-increasing" thermo-mechanical coating properties (hard with high thermal inertia). Friction increase was shown to be more pronounced than in the corresponding individual cases reaching about 11% in the most extreme case considered. Friction variations were also shown to increase with coating thickness. Most importantly, all reported friction variations are attained without affecting film thickness and as such the risk of component failure.

Nomenclature

α^*	: Pressure-viscosity coefficient (Pa^{-1})
μ_0	: Ambient pressure and temperature lubricant's viscosity (Pa.s)
υ_c	: Coating's Poisson coefficient
υ_s	: Substrate's Poisson coefficient
ρ_c	: Coating's density (kg/m^3)
ρ_s	: Substrate's density (kg/m^3)
ρ_0	: Ambient pressure and temperature lubricant's density (kg/m^3)
τ	: Shear stress (Pa)
a	: Hertzian contact radius (m)
c_c	: Coating's heat capacity (J/kg.K)
c_s	: Substrate's heat capacity (J/kg.K)
c_0	: Ambient pressure and temperature lubricant's heat capacity (J/kg.K)
C_c	: Coating's volumetric heat capacity ($J/m^3.K$)
E_c	: Coating's Young's modulus of elasticity (Pa)
E_s	: Substrate's Young's modulus of elasticity (Pa)
F	: Contact external applied load (N)
h	: Lubricant film thickness (m)
H	: Dimensionless lubricant film thickness
I	: Coating's thermal inertia ($J/m^2.K.s^{1/2}$)
k_c	: Coating's thermal conductivity (W/m.K)
k_s	: Substrate's thermal conductivity (W/m.K)
k_0	: Ambient pressure and temperature lubricant's thermal conductivity (W/m.K)
p	: Pressure (Pa)
P	: Dimensionless pressure
p_h	: Hertzian contact pressure (Pa)
R	: Ball's radius (m)
SRR	: Slide-to-Roll ratio = $(u_b-u_p)/u_m$
t_c	: Coating's thickness (m)
T	: Temperature (K)
T_0	: Ambient temperature (K)
u_b	: Ball's surface velocity (m/s)

u_p : Plane’s surface velocity (m/s)
u_m : Mean entrainment speed = $(u_b+u_p)/2$ (m/s)
x, y, z : Space coordinates (m)
X, Y, Z : Dimensionless space coordinates

Subscripts

c : Coating
s : Substrate
0 : Value of parameter at ambient pressure and temperature

Dimensionless Parameters

$$H = \frac{hR}{a^2} \qquad P = \frac{p}{p_h} \qquad X = \frac{x}{a} \qquad Y = \frac{y}{a} \qquad Z = \begin{cases} z/a & : \text{substrates} \\ z/t_c & : \text{coatings} \\ z/h & : \text{lubricant} \end{cases}$$

References

[1] Bennett A. and Higginson G. R. – Hydrodynamic Lubrication of Soft Solids, *J. Mech. Eng. Sci.*, 1970, vol. 12 (3), pp. 218-222.

[2] Elsharkawy A. A. and Hamrock B. J. – EHL of Coated Surfaces: Part I – Newtonian Results, *ASME Journal of Tribology*, 1994, vol. 116, pp. 29-36.

[3] Elsharkawy A. A. and Hamrock B. J. – EHL of Coated Surfaces: Part II – Non-Newtonian Results, *ASME Journal of Tribology*, 1994, vol. 116, pp. 786-793.

[4] Jin. Z. M. – Elastohydrodynamic Lubrication of a Circular Point Contact for a Compliant Layered Surface Bonded to a Rigid Substrate, Part 1: Theoretical Formulation and Numerical Method, *IMechE J. Eng. Trib.*, 2000, vol. 214 (3), pp. 267-279.

[5] Jin. Z. M. – Elastohydrodynamic Lubrication of a Circular Point Contact for a Compliant Layered Surface Bonded to a Rigid Substrate, Part 2: Numerical Results, *IMechE J. Eng. Trib.*, 2000, vol. 214 (3), pp. 281-289.

[6] Elsharkawy A. A., Holmes M. J. A., Evans H. P. and Snidle R. W. – Micro-Elastohydrodynamic Lubrication of Coated Cylinders Using Coupled Differential Deflection Method, *IMechE J. Eng. Trib.*, 2006, vol. 220, pp. 29-41.

[7] Liu Y., Wayne Chen W., Zhu D., Liu S. and Jane Wang Q. – An Elastohydrodynamic Lubrication Model for Coated Surfaces in Point Contacts, *ASME Journal of Tribology*, 2007, vol. 129, pp. 509-516.

[8] Liu Y., Jane Wang Q. and Zhu D. – Effect of Stiff Coatings on EHL Film Thickness in Point Contacts, *ASME Journal of Tribology*, 2008, vol. 130, 031501.

[9] Wang Z., Yu C. and Wang Q. - Model for Elastohydrodynamic Lubrication of Multilayered Materials, *ASME Journal of Tribology*, 2014, vol. 137, doi: 10.1115/1.4028408.

[10] Fujino T., Iwamoto K., Tanaka K. and Shima M. – Stress Distribution of Coated Film with a Range of Coated Film Thickness and Elastic Properties under a Single EHL Operating Condition, *Tribology International*, 2007, vol. 40, pp. 1638-1648.

[11] Chu L. M., Yu C. C., Chen Q. D. and Li W. L. - Elastohydrodynamic Lubrication Analysis of Pure Squeeze Motion on an Elastic Coating/Elastic Substrate System, *ASME Journal of Tribology*, 2014, vol. 137, doi: 10.1115/1.4028916.

[12] Evans R. D., Cogdell J. D., and Richter G. A. - Traction of Lubricated Rolling Contacts Between Thin-Film Coatings and Steel, *Tribology Transactions*, 2009, vol. 52 (1), pp. 106–113.

[13] Kalin M. and Polajnar, M. - The Correlation Between the Surface Energy, the Contact Angle and the Spreading Parameter, and their Relevance for the Wetting Behaviour of DLC with lubricating oils, *Tribology International*, 2013, vol. 66, pp. 225–233.

[14] Kalin M., Velkavrh I. and Vižintin J. - The Stribeck Curve and Lubrication Design for Non-Fully Wetted Surfaces, *Wear*, 2009, vol. 267 (5-8), pp. 1232–1240.

[15] Björling M., Isaksson P., Marklund P. and Larsson R. – The Influence of DLC Coating on EHL Friction Coefficient, *Tribology Letters*, 2012, vol. 47, pp. 285-294.

[16] Björling M., Habchi W., Bair S., Larsson R. and Marklund P. - Friction Reduction in Elastohydrodynamic Contacts by Thin Layer Thermal Insulation, *Tribology Letters*, 2014, vol. 53, pp. 477-486.

[17] Habchi W., Eyheramendy D., Vergne P. and Morales-Espejel G. – Stabilized Fully-Coupled Finite Elements for Elastohydrodynamic Lubrication Problems, *Advances in Engineering Software*, 2012, vol. 46, pp. 4-18.

[18] Habchi W. – A Numerical Model for the Solution of Thermal Elastohydrodynamic Lubrication in Coated Circular Contacts, *Tribology International*, 2014, vol. 73, pp. 57-68.

[19] Björling M., Larsson R. and Marklund P. - The Effect of DLC Coating Thickness on Elastohydrodynamic Friction, *Tribology Letters*, 2014, vol. 55, pp.353-362.

[20] Habchi W. – Thermal Analysis of Friction in Coated Elastohydrodynamic Circular Contacts, *Tribology International*, 2015, 10.1016/j.triboint.2015.01.017.

[21] Yang P. and Wen S. - A Generalized Reynolds Equation for Non-Newtonian Thermal Elastohydrodynamic Lubrication, *ASME Journal of Tribology*, 1990, vol. 112, pp. 631-636.

[22] Habchi W., Vergne P., Bair S., Andersson O., Eyheramendy D. and Morales-Espejel G. E. – Influence of Pressure and Temperature Dependence of Thermal Properties of a Lubricant on the Behavior of Circular TEHD Contacts, *Tribology International*, 2010, vol. 43, pp. 1842-1850.

[23] Habchi W., Bair S. and Vergne P. – On Friction Regimes in Quantitative Elastohydrodynamics, *Tribology International*, 2013, vol. 58, pp. 107-117.