Characterization of interfacial shear strength and its effect on ploughing behaviour in single-asperity sliding

Tanmaya Mishra\textsuperscript{a,}\textsuperscript{*}, Matthijn de Rooij\textsuperscript{a}, Megshyam Shisode\textsuperscript{b}, Javad Hazrati\textsuperscript{b}, Dirk J. Schipper\textsuperscript{a}

\textsuperscript{a} Surface Technology and Tribology, Faculty of Engineering Technology, University of Twente, 7500, AE Enschede, the Netherlands
\textsuperscript{b} Nonlinear Solid Mechanics, Faculty of Engineering Technology, University of Twente, 7500, AE Enschede, the Netherlands

ARTICLE INFO

Keywords:
Friction model
Boundary layer
Ploughing
Interfacial shear
Material point method

ABSTRACT

The shear strength at the interface contributes to the overall friction force experienced by the contacting bodies sliding against each other. In this article, an experimental technique to characterize the shear strength at the interface of metallic bodies in sliding contact has been developed. The boundary layers formed at interface in a lubricating contact have been varied by using two different types of lubricants in combination with both zinc coated and uncoated steel sheets. The empirical relations between the experimental parameters such as contact pressure and sliding velocity and the interfacial shear strength have been expressed by fitting the experimental results. These expressions have been incorporated in the Material Point Method (MPM) based ploughing model. The coefficient of friction and ploughing depth obtained from the numerical simulations have been validated relative to the experimental results with a good agreement for both lubricated and unlubricated substrates, different loads and spherical indenter sizes. Furthermore, the interfacial shear strength has been varied in the MPM-based ploughing model and ploughing experiments to study the contribution of interfacial shear strength to overall friction, deformation and wear.

1. Introduction

Most metallic surfaces are naturally covered by an oxide layer as well as a boundary layer when lubricated. Shear takes place at these surface layers when a tangential load is applied. The stress required to shear off these layers at the sliding contact interface is defined as the ‘interfacial shear strength’ or specifically ‘boundary layer shear strength’ for boundary layers. In the absence of a lubricating boundary layer, metallic oxide films are typically formed at the contact which contributes to a higher interfacial shear strength. In the absence of any interfacial layer, direct contact between sliding bodies results in a very high interfacial shear strength which might almost equal the bulk shear strength of the deforming substrate. The interfacial shear strength along with the resistance of the substrate to plastic deformation of the substrate contributes to the overall friction and wear in sliding of a rigid asperity through a metallic substrate [1].

Initial work on functioning of lubricated boundary layer was done in Refs. [1–3]. The presence of lubricant between two bodies sliding against each other prevents direct contact of the metallic asperities, thereby greatly reducing friction and wear. The lubricant does so by forming boundary layers [4] of low shear strength, either by physical or chemical adsorption on the surface of the contacting body(s) in boundary lubrication regime. A non-polar lubricant adsorbs (attaches) itself to the inactive metallic surfaces by weak Van der Waals forces. In the presence of functional groups such as acids, amines or esters, the lubricant’s polar head adsorbs itself on the metallic surface while the long hydrocarbon tail forms parallel chains which shear during loading and sliding of the contacting bodies [22]. The polar functional heads in the physically-adsorbed boundary layers might further react and chemically bond with the activated metallic surfaces to form metal-hydrocarbon based chemically-adsorbed boundary layers. The boundary layers typically fail as the severity of contact increases at high contact pressure and temperature. Boundary layers have been studied using Langmuir-Blodgett (LB) films by depositing them on surfaces of metals, glass and mica as vertically adsorbed monolayers [5].

Typically, the shear strength of the lubricant boundary layer has been measured by sliding large spheres at low loads on smooth-lubricated surfaces to avoid plastic deformation. By eliminating the friction due to plastic deformation of the substrate, the shear strength of the boundary layer is given as the ratio of the measured friction force and the real area of contact. In elastic deformation, the contact area is a function of the applied load. Hence, the boundary layer shear strength...
is typically load-dependent. It has been shown to be directly proportional to the applied load \( [6] \), in the sliding experiments using glass spheres on LB mono- and multilayers of stearic acid and calcium stearate deposited on glass plates under a range of contact pressures \([7]\). The shear strength of the lubricant monolayers of fluorides and metal soaps (stearetes) with long chain fatty acids, polymeric films and anthrancene deposited on glass, mica and platinum \([6–8]\) have also shown a linear relationship with applied load, above a critical value of contact pressure. The boundary layer shear strength remains constant at low loads due to the constraining of the contact pressure by the Van der Waals attraction between the contacting surfaces and/or due to the gradual orientation of the adsorbed molecular chains along the direction of sliding under the critical contact pressure \([8]\). Hence, for the shear stress to increase with load, the contact pressure must exceed the energy barrier (activation energy) required to cause shear \([15]\). For higher contact pressures, the increase in shear stress is due to squeezing of the molecular chains and consequent hardening of the boundary layers \([8]\). The ‘hardening’ of the boundary layers has been also associated with the increase in activation energy with contact pressure \([15]\).

In certain lubricants like calcium carbonate in dodecane colloidal films boundary layer shear strength increased linearly within two given pressure ranges, while staying constant for the pressures in between \([9]\). The boundary layer shear strength has been shown to decrease exponentially with temperature in Refs. \([7,8]\). The slope of the plot of logarithmic boundary layer shear strength and temperature is the activation energy. For low loads and sliding velocities, the boundary layer shear stress has been shown to linearly decrease with temperature. Initial studies in Refs. \([10,11]\) with lubricants such as stearic acid have showed an increase in the boundary layer shear strength with sliding
velocity. This has been explained due to the increase in strain rate of the boundary layers with sliding velocity and its subsequent ‘hardening’. However, other studies \cite{8,12,13} with lubricants such as calcium stearate have observed a decrease in boundary layer shear strength with the sliding velocity \cite{8}. This has been explained by the visco-elastic behaviour, where the response time to an applied load, \( i.e. \) the apparent pressure felt by the boundary layer over a given time period, decreases with sliding velocity. The shear strength of boundary layers has also been shown to marginally decrease with increasing film thickness with multiple monolayers in Refs. \cite{13,14}. However, since shear only occurs at the surface of boundary layers, the shear strength is often assumed to be unchanged with additional layers.

Challen and Oxley \cite{33} have shown, using slip line field solutions, the effect of interfacial shear strength on the friction and wear behaviour of two-dimensional wedge shaped asperity. The theoretical solutions to compute friction and wear in two dimensional asperities have been extended for three dimensional spherical asperities by single-asperity sliding experiments on lubricated and unlubricated metallic contacts by Hokkirigawa and Kato \cite{34}. The various wear regimes have been mapped in a wear mode diagram as ‘ploughing’, where substrate material is displaced to the sides of the track due to the sliding asperity, ‘wedging’, where substrate material is removed and accumulated in front of the sliding asperity and ‘cutting’, where substrate material is removed as chips. The increase in interfacial shear strength \( e.g. \) by absence of boundary layers in the contact can result in a transition from ploughing to wedging. The wedging wear mode results in formation of wedges of deformed substrate material stacked in front of the sliding asperity and subsequently, a possible transfer of the stacked substrate material to the surface of the asperity due to high adhesive forces. The slip-line field theory has also been used to study the influence of pressure and boundary layer shear strength for rigid cylindrical asperities ploughing through a soft substrate in Ref. \cite{24}.

Friction and wear in boundary lubrication regime have been modelled using particle-based, molecular dynamics (MD) simulations for rough surfaces in contact during loading and sliding \cite{36–39}. The effect of applied load, lubricant amount and chain length of molecules for different long chain-alkanes on friction has been studied in Ref. \cite{38} for boundary lubrication, while their effect on contact area has been studied in Ref. \cite{36} for different lubricated conditions. Polarisable lubricant such as polyethylene oxide polymer has been shown in Ref. \cite{37} to form films in the contact between charged, oxidised metallic surfaces, thereby preventing direct asperity contact and reducing friction force. Recently, MD simulation has also been used to investigate the reduction in friction and wear in water lubricated contact between inert polymers and metals compared to dry contact in Ref. \cite{39}. While MD simulations of boundary layers has been helpful in understanding of friction mechanisms in boundary lubrication at an atomistic scale, the up-scaling of the MD results can be challenging.

So far the characterization of boundary layer shear strength has been done for long chain fatty acid based lubricants on smooth glass and mica substrates. The boundary layer shear strength for mineral oils on metallic substrates has been determined for aluminium and gold coated glass and steel in Refs. \cite{18,19,21} respectively. Moreover, the effect of boundary layer shear strength on ploughing friction has not been investigated using numerically models so far. Most manufacturing systems use coated metallic tools and workpiece, working in the boundary lubrication regime where the shearing of boundary layers occurs under varying operating conditions. The presence of a coated system adds to the complexity of deformation behaviour of the substrate in measuring boundary layer shear strength. Furthermore, the use of large spherical balls to characterize the boundary layer shear strength also poses challenges in designing the required experimental set up.

Hence, the research has focussed on characterizing the boundary layer shear strength of both lubricated and un lubricated zinc coated and uncoated steel sheets under varying loads and sliding velocities using an in-house developed experimental set up. The effect of interfacial shear strength in overall friction and wear modes of a single-asperity ploughing through metallic substrate is investigated by the material point method (MPM)-based numerical ploughing model developed by Ref. \cite{32}. The MPM model \cite{32} is used to investigate the effect of the interfacial shear strength on the ploughing behaviour of a single asperity sliding through a steel substrate. By incorporating experimentally determined relationships for the interfacial shear strength in the ploughing model, the numerical results have been validated and are found to be in good agreement with the ploughing experiments using spherical tip pins.

2. Calculation of friction due to interfacial shear strength

The current section elaborates on the theory behind calculation of the interfacial shear strength \( \tau \) and the contact area \( A_c \) whose product results in the interfacial friction force \( F_{\text{int}} = \tau A_c \). The section also introduces on an algorithm to calculate the contact area in loading of a roller on a zinc-coated steel sheet.

2.1. Calculation of interfacial shear strength

The activation energy based Eyring model describes boundary layer shear with discrete movement (dislocation) of a small number of molecules \cite{15}. The dislocation movement is resisted by the neighbouring molecules due to a potential barrier, which must be overcome with shear and/or thermal stresses. The height of the barrier increases linearly with applied pressure as shown schematically in Fig. 1. Fig. 1 describes the energy of the potential barrier \( Q \) affected by pressure \( P \) and shear stress \( \tau \). The average time reciprocal, \( 1/\tau' \) to overcome this barrier, for a mobile unit of molecules, is the product of their effective vibration frequency \( \nu \) and the Boltzmann factor, \( \exp(-E/kT) \) as given in equation (1.1). Here \( E \) is the height of the energy barrier, \( k \) is the Boltzmann constant and \( T \) is the temperature. Chugg and Chaudri \cite{16} have expressed (equation (1.1)) the average time reciprocal as the shear rate \( \gamma \) and the effective vibration frequency as \( \nu/k \) where constant \( A \) is \( \sim 5 \times 10^{12} \text{s}^{-1} \). The Boltzmann factor shows the probability of a system being in a state with energy \( E \), where \( E/kT \) is the entropy per molecule. Here \( E = Q + P \Omega - \tau \Phi \) is the height of the barrier, \( kT \) is the heat required for increasing thermodynamic entropy of system. In macroscopic system with large number of molecules, \( RT \) is used instead of \( kT \) with units \( J/mol \) (\( R \) is the gas constant).

In equation (1.1), \( \Omega \) and \( \Phi \) are used for dimension correction with units of volume in the order of \( 0.1-1 \text{ nm}^3 \). \( \Omega \) can be physically interpreted as the pressure activation volume, causing a local increase in volume at a given lattice, permitting molecular motion. Physically, \( \Phi \) is the stress activation volume, which is the change in molecular/dislocation volume due to unit shear. The energy barriers are periodically separated by a distance \( c \) with allowable transition in both directions.

![Fig. 1. Energy barrier for dislocation movement during shearing of a boundary layer \cite{15}.](image-url)
Taking the ratio of the sliding velocity \( v \) and a reference sliding velocity \( v_0 \) equal to the ratio of the average molecular velocity \( u = c_0/\sqrt{\pi} \) and \( u_0 \) which is the product of molecular vibration frequency \( \omega_0 = 10^{11}/s \) and lattice constant \( c_0 = 0.22\text{nm} \), equation (1.2) is written as equation (1.3).

In equation (1.3), we approximate \( 2 \sin h \frac{\pi v}{\Lambda_0} \approx \exp -\frac{\pi v}{\Lambda_0} \) taking \( \phi/kT > 1 \) for low temperatures. The shear stress \( \tau \) is then expressed in equation (1.4) as a function of \( P, T \) and \( v \) where the values of \( \Phi, \Omega \) and \( \Theta \) can be determined using given values of \( v, P \) and \( T \) for various lubricant monolayers [8]. At high temperatures, \( (\phi/kT < 1) \), we approximate \( \sin h \frac{\pi v}{\Lambda_0} \approx \phi/kT \) by expanding power series of hyperbolic sine and neglecting its higher order terms. The values of \( \Phi, \Omega, \phi \) and \( v/v_0 \) range between 1 and 100 \( \text{kJ/mol} \), 0.01–1 \( \text{nm}^3 \), 1–10 \( \text{mm}^3 \) and 10^-6–10^-4 respectively for long chain hydrocarbon based lubricants [8]. Hence, the value of \( \text{v}kT/v_0\Phi \) does not vary largely with temperature and remains nearly constant. Rearranging terms in equations (1.2) and (1.3) would lead to an exponential temperature dependence of the shear strength as shown in equation (1.5) [8].

\[
\frac{1}{t'} = \omega_0 \exp \left( -\frac{Q + P - \pi \Phi}{kT} \right)
\]

(1.1)

\[
\frac{1}{u} = 2u_0 \exp \left( -\frac{Q + P - \pi \Phi}{kT} \right) \sinh \frac{2\Phi}{kT}
\]

(1.2)

\[
v = v_0 \exp \left( -\frac{Q + P - \pi \Phi}{kT} \right)
\]

(1.3)

\[
\tau = \frac{kT}{\Phi} \ln \left( \frac{v}{v_0} + \frac{Q + P - \pi \Phi}{kT} \right)
\]

(1.4)

\[
\tau = \frac{kT}{\Phi} \exp \left( \frac{Q + P - \pi \Phi}{kT} \right)
\]

(1.5)

Based on the observations and fitting of experimental data, empirical linear-relationships between boundary layer shear strength and contact pressure, temperature and logarithmic sliding velocity have been proposed as shown in equations (2.1), (2.2) and (2.3) respectively Refs [8,17]. In the work of Briscoe and Evans [8] and Chugg and Chaudri [16], the expressions of shear stress obtained from the Eyring model (equation (1.4)) have been compared with the empirical relations in equations (2.1)-(2.3) to obtain the values of intrinsic shear strength \( \omega_0, \eta \) and \( \zeta_0 \) and the proportionality constants \( \alpha, \beta, \phi \) and \( \psi \) in equation (2.4) [8].

\[
\tau = \eta + \alpha P
\]

(2.1)

\[
\tau = \beta \theta
\]

(2.2)

\[
\tau = \zeta_0 + \psi \ln v
\]

(2.3)

where,

\[
\omega_0 = \frac{kT \ln \left( \frac{v}{v_0} + \frac{Q + P - \pi \Phi}{kT} \right)}{\phi} \quad \alpha = \frac{Q}{\phi} \quad \beta = \frac{P}{\phi} \quad \phi = \frac{kT}{\Phi} \quad \psi = \frac{2\Phi}{kT}
\]

(2.4)

Earlier work of Briscoe and Tabor [13] have shown the shear strength of organic and polymeric films to decrease exponentially with temperature as given in equation (2.5) taking gas constant \( k \) and activation energy \( Q' \). Further, under isothermal and isobaric conditions, the shear strength is given to increase with increase in strain rate of the boundary layer (of thickness \( h \)) i.e. \( v/h \). Hence the intrinsic shear strength term \( \tau_0 \), which is independent of contact pressure, can be expressed as a function of strain rate (sliding velocity) and temperature. However, the boundary layer shear strength for some lubricants have been shown to decrease with increase in sliding velocity \( v \). The increase in \( v \) reduces the mean contact time \( t_c = v/2a \) for the boundary layer. Here, \( a \) is the contact radius. This effect, termed as 'visco-elastic retardation in compression' [13], reduces the response time of the boundary layer to applied pressure, thereby reducing \( \alpha \). Equation (2.5) corrects the value of \( \alpha \) taking the 'visco-elastic' effect into account [13].

\[
\eta_0 = K \exp \left( -\frac{Q'}{kT} \right)
\]

\[
K = \zeta_0 \ln \left( \frac{v}{\phi} \right)
\]

\[
\alpha = n_0 \exp \left( -\frac{v}{\phi} \right)
\]

(2.5)

The characteristic frequencies \( \phi \) and \( \theta \) in equation (2.5) corresponds to high strain rate and low strain rate processes respectively [13]. Hence the 'strain rate effect' dominates the boundary layer shear strength at high sliding velocity while the viscoelastic effect dominates boundary layer shear strength at reduced sliding velocity. The 'strain rate term' \( K \) corresponds to shear strength at high sliding velocity and varies between 0 to \( \zeta_0 \) for velocities ranging from \( (v_x, e_0) \) where \( v_x = h^2 \) and \( e_0 = 2.72 \). Similarly, the visco-elastic term \( \alpha = \zeta_0 \) corresponds to shear strength at low sliding velocity and varies between \( n_0 \) to 0 for velocity ranging between \( (0, \infty) \). It can be seen that for high sliding velocity, the 'visco-elastic' term diminishes while for low sliding velocity the 'strain rate term' diminishes.

Based on experimental data from Refs. [7,8] it has been observed that for large pressure ranges, the boundary layer shear strength doesn't vary linearly with contact pressure. Hence a more general power-law relationship between shear strength and contact pressure can be used for fitting the experimental data. Similarly, by taking positive and negative exponents in the power-law relationship between shear strength and sliding velocity the effect of visco-elastic retardation due to compression and strain rate can be accounted for. Taking these observations into account, Westeneng [20] proposed a power law relationship between \( \tau-P \) and \( \tau-v \), where \( D \) and \( G \) are proportionality constants and \( p \) and \( q \) are exponents. The exponential relation between \( \tau \) and \( T \) is used in equation (3.3), where \( \zeta_0' \) is a constant. Assuming the shear stress in equations (3.1), (3.2) and (3.3) to be independent of each other and taking the natural logarithm of \( \tau \) in all the equations, the logarithmic \( \tau \) is combined to obtain equation (3.4).

\[
\tau = D P + \ln \eta = \ln D + p \ln P
\]

(3.1)

\[
\tau = G v + \ln \eta = \ln E + q \ln v
\]

(3.2)

\[
\tau = \zeta_0' \exp \left( \frac{Q'}{kT} \right) + \ln \eta = \ln \zeta_0' + \frac{Q'}{kT}
\]

(3.3)

\[
\ln \tau = F + \frac{Q'}{kT} + p \ln P
\]

(3.4)

In equation (3.4), \( F \) can be taken as equivalent to the parameter \( K \) in equation (2.5) which is influenced by the 'strain rate' effect of sliding velocity \( v \). Similarly \( P \) can be taken as equivalent to the parameter \( \alpha \) in equation (2.5) which is influenced by the 'visco-elastic' effect of the sliding velocity \( v \). Expanding the expressions for \( F \) and \( p \), logarithmic shear stress is expressed in equation (4.1). Taking \( Q'/R \) as \( n_x \), \( \eta \), \( n_0 \) as \( n_x \), \( \eta \), and \( v = h^2 \) combined with other constants as \( K' \) in equation (4.2), boundary layer shear strength is given as a function of contact pressure, sliding velocity and temperature in equation (4.3).

\[
\ln \tau = \zeta_0' \ln \left( \frac{v}{\phi} \right) + \frac{Q'}{kT} + n_0 \exp \left( -\frac{v}{\phi} \right) \ln P
\]

(4.1)

\[
\ln \tau = n_x \ln v + \frac{n_x}{T} + n_x \ln P + K'
\]

(4.2)

\[
\tau(P, T, v) = CP^h v^n \exp \left( \frac{n_x}{T} \right)
\]

(4.3)

In order to measure the interfacial shear strength, experiments can be done where the shear strength will be calculated from the measured friction force and calculated contact area. The details about the procedure for calculation of contact area are explained in section 2.2 below.

2.2. Calculation of contact area and contact pressure for a coated line contact

The contact width \( b \) for a cylinder of radius \( r \) and length \( l \) pressing into an uncoated flat surface is given in equation (5.1) in terms of its...
normal load $F_n$ and effective modulus of elasticity $E^*$ for the substrate and the tool. In order to improve friction and wear behaviour, materials can be coated with surface layers. The effective elastic modulus $E^*$ is then defined by the Young's modulus and the Poisson's ratio of the coated system $E_{cs}$ and $\nu_c$, and of the indenter $E_{in}$ and $\nu_i$, as given in equation (5.2). The effective Young's modulus $E_{cs}$ for a multilayered system with the substrate $s^n$ covered with $n$ number of layers is given in equation (5.3), see Refs. [27,28]. Equation (5.3) uses influence factors $I_0$ and $I_i$, given in equations (5.4) and (5.5) respectively with $i = 1$ as the bottom layer and $i = n$ as the top layer. The relative depth of the layer $i$ $t_i$ in equation (5.6) is the ratio of the total layer thickness up to layer $i$ ($t_i$, $t_2$, ... $t_n$) and the contact radius $a$. The average Poisson’s ratio $\bar{\nu}$ of all the layers is given in equation (5.7).

$$b = \frac{4F_n r}{\pi E^*}$$

(5.1)

$$\frac{1}{E^*} = \frac{1 - \nu_i^2}{E_{in}} + \frac{1 - \nu_c^2}{E_{cs}}$$

(5.2)

$$\frac{1 - \nu_c^2}{E_{cs}} = \frac{1}{E_{cs}} - \sum_{i=1}^{n} I_i (\nu_i - \nu_{i+1}) - I_i (\nu_c - \nu_i)$$

(5.3)

$$I_0 = \frac{2}{\pi} \arctan \left( \frac{(1 - 2\nu)\bar{t}_i}{2(1 - \bar{\nu})} \right)$$

(5.4)

$$I_i = \frac{2}{\pi} \arctan \left( \frac{1 + \bar{t}_i}{\bar{t}_i} \right)$$

(5.5)

$$\bar{t}_i = \frac{1}{n} \sum_{k=1}^{n} t_k$$

(5.6)

$$\bar{\nu} = \frac{1}{n} \sum_{i=1}^{n} \nu_i$$

(5.7)

$$E_{cs} = E_i + (E_c - E_i) I_0 \ \forall \ n = 1, \ \nu_i = \nu_c$$

(5.8)

The equations to estimate the effective modulus of a coated system is based on spherical indentation of the multi-layered substrate. The influence factors $I_0$ and $I_i$ are functions of the contact width (radius for point contact) $a$ used in the expression of $E^*$. The contact width is typically computed from the effective elastic modulus as shown in equation (5.1).

In order to compute the contact width in a line contact for the case of a cylindrical roller in contact with a flat surface, using the effective elastic modulus of the coated system, an initial guess for the contact width $b_0$ is used to calculate the line contact width $b$. The calculated contact width $b$ is then corrected by adding the difference $|b - b_0|$ to $b_0$ and using the new value of contact width $b^*$ to recalculate $b$. The steps are iterated until the difference $|b - b_0|$ is minimized below a given tolerance as shown by the algorithm in Fig. 2. The algorithm calculates the line contact area for a single layer of (hot dip galvanized) zinc-coated steel sheet using the effective Young’s modulus $E_{cs}$ computed by equation (5.8), deduced from equation (5.3) for $n = 1$ and $\nu_i = \nu_c$.

The mean nominal contact pressure, for a Hertzian line contact, is calculated using equation (6.2) from the normal load $F_n$ and the Hertzian contact area $A_c = 2b h$ where $b$ is the contact half width and $l$ is the contact length. The maximum nominal contact pressure $P_0 = 4P_{nom}/\pi$ should be maintained below the yield stress of the substrate sheet to avoid any plastic deformation [10]. The yield strength for perfectly plastically deforming substrate can be approximated by $\sigma_y = H_c/2.8$ as per [31]. The effective hardness of the coated system is given based on the experimental fit relations by Refs. [29,30]. For a soft zinc coating of thickness $t$ on a hard steel substrate the is expressed in equation (6.1). The applied loads should be chosen such that $b_0 < \sigma_y$.

$$P_{nom} = \frac{N}{A_c} = \frac{N}{2b}$$

(6.2)

$$H_c = H_i + (H_c - H_i) \exp \left( -\frac{125t}{r} \right)$$

(6.1)

Fig. 3a shows the variations in effective hardness and Young’s modulus of the zinc coated steel sheet with the coating thickness that is calculated using equation (5.8) and equation (6) and material data from Table 1. The indentation hardness of the bulk zinc, zinc coating and the steel sheets were measured Berkovich indenters at 100 mN load. Both
the hardness and Young's modulus of the zinc coated steel sheet decrease with increase in coating thickness from that of the steel substrate to that of the zinc coating. The effective hardness of the coated system is independent of the applied load. However, the effective Young's modulus of the coated system increasingly approaches towards the Young's modulus of zinc with increase in coating thickness for lower applied loads. For high loads the Young's modulus of the stiffer substrate's modulus of the coated system is 1.046 GPa is obtained from equations (5.8) and (6.1) for the given zinc coating thickness of 20 μm and applied load of 1–16 N.

3. Experimental method

This section describes the design of the experimental set up for determining the boundary layer shear strength. The preparation of the sheets specific to characterize the boundary layer shear strength and ploughing experiments have been explained. The experimental set-up has also been elaborated.

3.1. Design of experiments

The friction in sliding of a rigid asperity through a soft substrate is attributed to the resistance to plastic deformation of the substrate and the shearing of the interface. The key to experimentally characterize the interfacial shear strength is to eliminate the component of friction due to plastic deformation of the sheet. Hence, the measured friction solely results from shearing of the interface. Typically, large glass spheres have been slid on smooth (glass, mica, metal coated) surfaces in experiments pertaining to characterize the boundary layer shear strength [7]. Since a line contact distributes the load over a larger contact area compared to a point contact using pins of similar dimensions, it is easier to limit the generated nominal contact pressures below the yield stress of the sheet. This helps to avoid any macro-scale plastic deformation of the substrate. Hence, the curved surface of a hard cylindrical roller pin is slid against a soft, flat sheet in a lubricated line contact to characterize the interfacial shear strength.

Further, flattening and polishing of the substrate (sheet) is done to maximize the real contact area, avoid unwanted friction and formation of wear particles due to asperity interlocking. However, at the micro-scale, the local pressure on the asperities in the line contact exceeds the yield point. To prevent the local plastic deformation of the asperities and optimize the conformity of contact, multiple traverses are performed on the sliding track until a steady state friction is obtained. During the initial sliding traverses, the harder asperities on the pin tool plastically deforms the surface of the polished sheet during the ‘running-in’ phase. During the subsequent traverses, repeated re-loading and unloading of the sheet results in metal work hardening of the sheet surface thereby increasing its yield strength. As the surface of the sheet work hardens, the surface of the sheet deforms elastically during sliding and a ‘steady state’ friction is reached. The friction measured in the ‘steady-state’ is predominantly due to shearing of the boundary layers. Subsequently lower applied loads are applied during sliding of the roller.

The pressure distribution under the cylindrical roller is determined by the shape of the roller and its surface roughness. The discontinuity in contact at the edge of the cylindrical roller results in high stress concentration. The roller acts like a punch along the length of its contact with the sheet. The stress concentration at edges combined with bending of the sheet localizes the roller-sheet contact and results in ploughing instead of shearing on the sheet surface. Typically, the edges of cylindrical rollers are crowned with different geometries to avoid the punching (edge) effect leading to stress concentration at the edges. In the experiments for the current study, the cylindrical roller has been provided with a logarithmic crowned profile towards the edges [23].

3.2. Preparation of experimental specimen

Mirror polishing of rough DX46 steel sheets, shown in Fig. 4a, with roughness of 1.24 μm was done. For mirror-polishing, the sheets are laser cut into 46 mm diameter circles and mounted on bakelite discs of 50 mm diameter either by hot mounting or by using the Loctite industrial glue. Using an automatic lapping/polishing machine, initial coarse grinding of the mounted sheet specimens is performed with 320 (46 μm size) grade sandpaper under 30 N load to remove any unevenness. The specimens are then fine-polished using diamond suspension of different particle sizes on metal disc in 3 steps, each lasting for 3 min under a load of 30 N. The size of the diamond particles at each step is reduced from 9 μm to 3 μm-1 μm. Finally, an OPS (oxide polishing suspension) with 0.04 μm grain size is used to obtain a scratch free surface. A final mean surface roughness of 6–8 nm is obtained and the sheet is degreased using ethanol. The polished sheet is shown in Fig. 4b with a clear view of its grain boundaries.

A set of sheet specimens with substrate roughness of 20–30 nm, shown in Fig. 4c, were also prepared by ‘shine polishing’, where the final polishing steps of 3 μm, 1 μm size diamonds suspension and OPS in ‘mirror polishing’ were skipped. The polished sheet specimen was degreased and stored at room temperature for 1000 h to allow formation of a stable oxide film on the surface. The presence of an oxide film and higher surface roughness of the specimens helped prevent material transfer to the surface of the pin during characterization of the interfacial shear strength of unlubricated steel sheets. The fresh, mirror polished specimen with surface roughness of 6–8 nm had higher possibility of material transfer due to the high interfacial shear strength resulting from direct contact between the metallic asperities.

![Fig. 3. (a) The effect of coating thickness $t$ on effective hardness $H_c$, and Young's modulus $E_c$, of a coated system and (b) the effect of applied load on $E_c$ obtained using equation (5.8) and equation (6.1) and material data from Table 1 ($H_f = 0.5GPa$, $H_t = 1.4GPa$, $E_f = 70GPa$, $E_t = 210GPa$).](image-url)
Table 1
Material parameters.

| Parameters                  | Substrate                  | Coating                              | Tool/Pin                          |
|-----------------------------|----------------------------|--------------------------------------|-----------------------------------|
| Geometry                    | Circular sheet of thickness 1 mm and diameter 50 mm | Logamtic crown steel roller          |                                   |
| Radius at the centre of roller $r$ | $=$                       | $=$                                  | $=$                               |
| Uncrowned/contact length of roller $l$ | $=$                       | $=$                                  | $=$                               |
| Total length of roller      | $=$                       | $=$                                  | $=$                               |
| Material                    | DX56 steel                | Zinc coated/galvanized steel          | AISI 52100 bearing steel          |
| Coating thickness $t$       | $=$                       | $=$                                  | $=$                               |
| Mean surface roughness $R_s$| 8 nm                      | 10 nm                                | 100 nm                            |
| Young’s modulus, $E$         | 210 GPa                   | 70 GPa [25]                          | 210 GPa                           |
| Hardness of sheet $H$       | 1.4 GPa                   | 0.5 GPa [42]                         | 8.2 GPa                           |
| Poisson’s ratio $\nu$       | 0.3                       | 0.3                                  | 0.3                               |
| Lubricant A                 | Viscosity: $\eta_{\text{np}}$ | Lubricant B Viscosity: $\eta_{\text{yp}}$ | Fuchs Anticorit PL5100T 90 mPas    |
| Quaker FERROCOAT N6130      | 23 mPas                   |                                      |                                   |

3.3. Experimental set up

The shear experiments on DX56 steel sheet lubricated with 2 different lubricants were done using the linear friction tester, shown in Fig. 5, with 3 repetitions. The linear friction tester consists of a XY linear positioning stage driven separately by actuators as shown in Fig. 5c. A horizontal beam supports the loading tip and moves the Z-stage using a linear piezo actuator for coarse and fine displacement respectively while applying a normal load. The normal load is applied using a force controlled piezo actuator, connected to a PID control loop feedback system so the system can operate load controlled. The friction forces are measured by a piezo sensor along the loading tip as shown in Fig. 5b. An example of the friction signal divided by the normal load, the coefficient of friction, is shown as a function of the sliding distance in Fig. 5a.

A cylindrical roller of diameter 10 mm and length 10 mm is mounted on a self-aligning pin as shown in Fig. 5b and c. The pin holder consists of a joint which allows for rotation of the roller in the axis along the sliding direction such that proper alignment of the roller over a slightly tilted surface results in a continuous line contact. Different loads were applied on these rollers to perform load controlled shear experiments. The plots for the roller pin were obtained from the linear sliding tester and the average coefficient of friction for the steady state was obtained for all applied loads. The wear track was studied under both optical and confocal microscopes to measure the contact area on the wear track. The applied loads for characterizing the interfacial shear strength were 1, 2, 4, 7, 11 and 16 N.

4. Computational method

The ploughing of asperities in the substrate is modelled using material point method (MPM) which has been introduced and implemented in Mishra et al. in Ref. [32]. The MPM-based ploughing simulation models the MPM particles in the substrate using the ‘mpp-linear pair style’ code which has been further explained in Ref. [32]. The indenter/asperity has been modelled using triangular mesh as an STL file with no self-interaction. The asperity interacts with the substrate using contact algorithm defined in the ‘tri-smd-pair style’ code [32] which includes the contact and friction algorithm between the triangular mesh and MPM particles. The friction algorithm computes the contact area and the overall friction using the interfacial shear strength which will be measured in section 5.2. Fig. 6 shows the MPM-ploughing model set up.

The flow stress is computed from the physically based isothermal Bergström van Lijemt hardening relation [40]. The relation was modified by Vegter for sheet metal forming processes [41], leading to the following formulation where the flow stress $\sigma_{\text{pl}}$ is decomposed into a static-work/strain hardening stress $\sigma_{\text{sh}}$ and dynamic stress $\sigma_{\text{dy}}$, which takes into account the strain-rate and the thermal effects as shown in equation (7.1). This flow stress model has been included in the current MPM numerical set up to account for the interaction processes between dislocations in cell structures including the changing shape of dislocation structures. The Bergström van Lijemt material model constants are listed in Table 3 with their characteristic values obtained for the DX56 steel sheet [32]. The model parameters $\varepsilon$, $\dot{\varepsilon}$ and $T$ represent the strain, strain rate and working temperature respectively.

\[
\sigma_{\text{pl}} = \sigma_{\text{sh}} + \sigma_{\text{dy}} = \sigma_{\text{sh}} + \sigma_{\text{dy}}(\beta(\varepsilon + \dot{\varepsilon}) + \{1 - \exp[-\omega(\varepsilon + \dot{\varepsilon})]\})
\]

\[
+ \sigma_{\text{dy}}\left(1 + \frac{kT}{\Delta v_0} \ln \frac{\varepsilon}{\dot{\varepsilon}_0}\right)^n
\]

(7.1)

A power-law expression for boundary-layer shear stress as a function of the nominal pressure $P$, sliding velocity $v_0$ and the contact temperature $T_c$ has been implemented in the triangles-particles interaction pair style following from equation (4.3), as given in equation (7.2) [32].

\[
r = C_p P^{\eta_p} v_0^{\eta_v} T_c^{\eta_T} \exp - \frac{\eta_r}{T_c} = C_p P^{\eta_p} v_0^{\eta_v} \exp - \frac{\eta_r}{T_c}
\]

(7.2)

where $C_p$ is the pressure constant, $C_v$ is the velocity constant, $\eta_p$ is the pressure exponent, $\eta_v$ is the velocity exponent and $\eta_T$ is the temperature exponent. The constant term $C$ is the product of $C_p$, $C_v$ and $C_T$. The C’s and $n$’s are experimentally fitting factors. The contact pressure for experiments is taken as the nominal contact pressure $P_{\text{nom}} = F_0/A_0$. In the MPM model, contact pressure is obtained between each indenter’s triangular mesh in contact with the MPM particle of the sheet. The

![Fig. 4. Surface of mounted sheet before (a) polishing $R_p = 1.24 \mu m$ (b) after mirror polishing $R_p = 0.006 \mu m$ and (c) shine polishing $R_p = 0.028 \mu m$ seen under confocal microscope at 50x magnification.](image-url)
coefficients and exponents in equation (7.2) can be arranged to reduce the expression to the Coulomb friction law. In equation (7.2), the contact temperature is obtained from the simulations for sliding as the wall temperature of indenter \( T_c \). The relative tangential velocity of the indenter's triangle with respect to the MPM particle in contact \( v_{\text{tan}} \), is taken locally as \( v_s \). The coefficients \( C_p \) and \( C_v \) and exponents \( n_p \), \( n_v \), and \( n_T \) can be obtained through experiments in a linear sliding friction experiments by varying the contact pressure, sliding velocity and temperature.

5. Results and discussion

The interfacial shear strength of both lubricated and unlubricated, zinc coated and uncoated steel sheet have been determined by the experimental method, discussed in this section. The subscripts \( UL \) and \( QL \) correspond to unlubricated (no lubricant applied on the shine polished sheet) and Quaker lubricated sheets while the superscripts \( MP \) and \( SP \) correspond to mirror polished and shine polished sheets (specimen preparation explained in section 3.2) respectively. Firstly, the contact area in the line contact has been calculated for varying loads based on the model discussed in section 2.2. Using the calculated contact area, the interfacial shear strength has been measured for the range of applied loads and sliding velocities and expressed as empirical equations. The equations have been implemented in the MPM-based numerical model described in section 4. Further, the effect of interfacial shear strength on the friction forces and wear mode in the sliding of the asperity/indenter is highlighted using results from MPM-based ploughing simulation.

![Fig. 5.](image.png)

Fig. 5. (a) Linear friction tester for boundary shear characterization and ploughing experiments with (b) its (Fig. 5a) schematic showing substrate specimen (B), (c) the loading set-up and sliding tool (A).

![Fig. 6.](image.png)

Fig. 6. MPM simulation of an spherical indenter (asperity) with radius 0.1 mm ploughing through a substrate along sliding \( x \)-direction with equivalent plastic strain being shown.

| Parameters                                      | Symbol | Values/expression |
|-------------------------------------------------|--------|-------------------|
| Substrate and indenter material density          | \( \rho \) | 7900 kg/m\(^3\) |
| Substrate and indenter specific heat capacity    | \( c_p \) | 502 J/(kg K) |
| Substrate and indenter thermal conductivity      | \( k \) | 502 W/(m K) |
| Young's Modulus of substrate                     | \( E \) | 210 GPa |
| Poisson's ratio of the substrate                 | \( \nu \) | 0.3 |
| Ambient temperature                              | \( T_{\text{room}} \) | 294 K |

| Parameters                                      | Symbols | Value/Expression |
|-------------------------------------------------|---------|-----------------|
| Initial static stress                            | \( \sigma_0 \) | 82.988 MPa |
| Stress increment parameter                       | \( d_{\sigma_m} \) | 279.436 MPa |
| Linear hardening parameter                       | \( \beta \) | 0.482 |
| Remobilization parameter                         | \( \omega \) | 6.690 |
| Strain hardening exponent                        | \( \sigma \) | 0.5 |
| Initial strain                                   | \( \varepsilon_0 \) | 0.005 |
| Initial strain rate                              | \( \varepsilon_{00} \) | \( 10^{0.4} \) |
| Maximum dynamic stress                            | \( \sigma_{\text{dy}} \) | 1000 MPa |
| Dynamic stress power                             | \( m \) | 3.182 |
| Activation energy                                | \( \Delta G_0 \) | 0.8 |
| Boltzmann's constant                             | \( k \) | \( 8.617 \times 10^{-5} \) eV |
5.1. Calculation of contact area and contact pressure

The sliding of the cylindrical pin on the lubricated, mirror polished sheet surface at an applied load of 16 N increases its roughness from 6 nm to 93 nm as shown in Fig. 7b and d. As the asperities on the surface of the pin slide (plough) through the polished sheet, there is transfer of roughness from the pin to the surface of the smooth sheet. Now, with increased conformity between the contacting surfaces, the boundary shear characterization experiments are performed on the same wear track with lower applied loads. On the other hands, the surface of the pin remains unchanged for most lubricated experiments. However, there is transfer of material from sheet to tool due to galling/adhesive wear for some unlubricated experiments on steel and lubricated experiments on zinc coated steel specimen at high loads as shown in Fig. 7a. Results pertaining to experiments with material transfer are avoided in calculation of boundary layer shear strength.

The apparent contact width and the nominal contact pressure for the coated system are calculated from the algorithm in Fig. 2, and validated against a finite element model (MSC Marc) as listed in Table 4. An element size of 0.2 μm is taken in the FE model for line contact. Both the analytical model and the finite element model give good agreement in calculation of Hertzian line contact and hence contact pressure for coated systems. For lower loads the finite element method requires a very fine mesh resolution to give an accurate line contact width. The layer hardness of 800.141 MPa with a standard deviation of 18.788 MPa is obtained for the zinc coating using Berkovich indentation at 100 mN load.

5.2. Calculation of boundary layer shear strength

The coefficient of friction vs sliding distance plots, obtained for different loads, are given in Fig. 8a. The mean friction force is calculated over a sliding length of 14 mm after the first 3 mm of sliding for a total sliding distance of 20 mm. The coefficient of friction due to shearing of the boundary layer decreases with increasing load. The steady state friction force is obtained after multiple traverses as shown in the steadying of the average coefficient of friction after initial decline due to running-in with each traverse in Fig. 8b. The mean steady state friction forces measured from the sliding experiments are divided by the computed Hertzian contact area to obtain the boundary layer shear strength.

The mean coefficient of friction for the shear experiments are plotted against various sliding velocities in Fig. 9. It can be seen that the boundary layer shear strength is not greatly affected by the change in sliding velocity. The boundary layer shear strength, τ, for the Quaker lubricant marginally increases with sliding velocity while that for the Anticorit lubricant marginally decreases with sliding velocity, as shown in Fig. 9a and b respectively. For experiments done with Quaker lubricant, the effect of shear strain rate, i.e. \( \dot{\gamma} = \frac{dv}{dt} \) for boundary layer thickness \( t \) seems to dominate, as a result of which \( \tau \) increases with \( v \).

For experiments done with Anticorit lubricant, the visco-elastic effect seems to dominate, where the effect contact pressure \( P \propto P/\exp(v/a) \) in response to application to normal stress (contact radius \( a \)) reduces with \( v \) as a result of which \( \tau \) decreases with \( v \) [13]. The relationship between the interfacial shear strength and sliding velocity is given in equations (8.1) and (8.2) for both the lubricants. The experiments were done at constant load of 7 N and at room temperature. Corresponding to equations (4.3) and (7.2), \( n_a = 0 \) and \( n_r = 0 \) are taken for equations (8.1) and (8.2).

\[
\tau_0^Q = 13.13 e^0.018
\]

(8.1)

\[
\tau_0^A = 9.32 e^0.015
\]

(8.2)

The characterization of boundary layer shear strength for ‘Quaker Ferrocoat N136’ and ‘Fuchs Anticorit PLS100T’ lubricated and unlubricated -DX56 steel sheets with varying contact pressures was done using the linear sliding friction experimental set-up, shown in Fig. 5. Fig. 10 shows the variation of boundary layer shear strength with (applied load) nominal contact pressure for both lubricated and unlubricated steel sheets. As explained in the literature [8,9], a higher contact pressure increases the potential barrier for the shearing of the boundary layers, thereby increasing the interfacial shear strength. It can be seen from Fig. 10a, b and c that the slope of the boundary layer shear strength plots decreases with increase in nominal contact pressure. Hence, the boundary layer shear strength (at constant sliding velocity of 1 mm/s and at room temperature) has been plotted against the contact pressure by fitting the experimental data with the power law relation (based on equation (7.2)) in equations (9.1)–(9.3). Both the lubricants seem to result in boundary layers with different shear strengths. The shear strength of the Quaker lubricated interface is slightly lower than that of the Anticorit lubricated interface. The shear strength of the unlubricated interface has been measured for experiments with stable friction behaviour where galling is absent. The absence of a lubricant boundary layer results in formation of stronger metallic junctions with high shear strength. So the unlubricated contact has a higher interfacial shear strength than the lubricated contact, see Fig. 10d. Corresponding to equations (4.3) and (7.2), \( n_a = 0 \) and \( n_r = 0 \) are taken for equations (9.1), (9.2) and (9.3) and equations (10.1) and (10.2).

\[
\tau_0^Q = 1.34 P^{0.88}_{nom}
\]

(9.1)

\[
\tau_0^A = 0.64 P^{0.93}_{nom}
\]

(9.2)

\[
\tau_0^U = 2.05 P^{0.88}_{nom}
\]

(9.3)

The characterization of the boundary layer shear strength for experiments done with zinc coated DX-56 steel was also carried out with sheets lubricated with Quaker and Anticorit lubricants. The unlubricated shear experiments done with zinc coated steel were avoided as steel pins sliding through zinc coated sheets have shown a higher
affinity for material transfer or galling. The mean friction forces obtained from the shear experiments were divided by the contact area calculated for the coated system in section 5.1. The boundary layer shear strengths were plotted against the applied nominal pressure for experiments done with lubricated GI sheets (at constant temperature and velocity) and fitted with power law expressions derived in equation (8.1) in Fig. 11a and b. The curve fit expressions for both the lubricants are listed in equations (10.1) and (10.2).

\[
\tau_{Bl}^{G} = 0.32P_{nom}^{0.32} \\
\tau_{Bl}^{A} = 7.34P_{nom}^{0.78}
\]

5.3. Effect of boundary layer shear stress on single asperity sliding behaviour

The effect of the boundary layer shear strength on the friction and wear in sliding of a rigid asperity has been studied in this section. Both experiments and simulations have been performed using "asperities" of 1 mm and 3 mm diameter under Quaker lubricated and un lubricated contact. The MPM-based ploughing model incorporates the relationship between the boundary layer shear strength and the contact pressure given in equations (9.2) and (9.3) for lubricated and unlubricated steel sheets to compute the interfacial friction. The ploughing model also uses material model (equation (7.1)) parameters listed in Tables 2 and 3 to compute the substrate deformation. The change in wear regime and wear volume in the presence and absence of a lubricant is highlighted using ploughing experiments and simulations.

5.3.1. Effect on friction in ploughing

The shear strength of the interface is varied by using a clean, unlubricated or lubricated 'shine polished' substrate prepared as per the method explained in section 3.2. Indenters having tips fitted with spherical balls of 3 mm diameter are used to plough the substrate. MPM-based ploughing simulations are done using same parameters as the experimental set-up. The MPM-ploughing simulations have incorporated interfacial friction models from equations (9.1) and (9.3) for the 'Quaker' lubricated and unlubricated contact respectively. The coefficient of friction is plotted against the sliding distance for ploughing experiments and ploughing simulations on both lubricated and unlubricated substrates in Fig. 12a and b respectively. The mean value of the coefficient of friction is measured at the steady state and plotted against applied loads ranging from 1 N to 46 N. MPM-based ploughing simulations are also performed for applied loads of 1–46 N and for particle-cell sizes of 5, 10 and 20 μm. The coefficient of friction for different resolutions are interpolated to 0 cell size to obtain the converged coefficient of friction.

Fig. 8. (a). Friction measurements with the linear sliding friction tester for various normal loads $F_n = 2, 11$ and 22 N. (b) Coefficient of friction with each traverse and running-in of the sheet to obtain friction due to boundary layer shear at 16 N load.

Table 4

| Load (N) | Contact width $b$ (Analytical) (μm) | Contact width $b$ (FEA) (μm) | Nominal contact pressure $P_{nom}$ (MPa) |
|----------|-----------------------------------|-------------------------------|----------------------------------------|
| 1        | 8.44                              | 11.8                          | 23.7                                   |
| 2        | 11.68                             | 13                            | 34.4                                   |
| 4        | 16.01                             | 17.2                          | 30.3                                   |
| 7        | 20.40                             | 23                            | 68.3                                   |
| 11       | 25.32                             | 28.8                          | 87.3                                   |
| 16       | 30.02                             | 34.2                          | 106.7                                  |

Fig. 9. Variation of boundary layer shear strength with sliding velocity at an applied load of 7N for DX56 sheet lubricated with (a) Quaker Ferrocoat N136 (b) Fuchs Anticorit PLS100T.
substrate are validated against the ploughing experiments for the applied load range. The MPM-based ploughing simulation results are in good agreement with the experimental results for sliding of indenter of 3 mm diameter through the lubricated substrate. Previous results [32] have also shown agreement between MPM-based ploughing simulation and ploughing experiments for an indenter of 1 mm and 3 mm diameter (Fig. 13). The MPM-based ploughing simulations on unlubricated substrate also show good agreement with the ploughing experiments on unlubricated substrate for loads ranging from 4 to 29N (Fig. 13a). For loads on either side of the range 4–29 N, the experimental coefficient of friction is higher than that numerical ones. Since the substrate is not mirror polished (see section 3.2) and the roughness of the substrate is kept at 20–30nm, which is close to the roughness of the spherical indenter, the asperities of the indenter and substrate may interlock while sliding at low penetration depth. The asperity interlocking (AI) results in the additional coefficient of friction for ploughing experiments at low loads.

Also, at high loads, the substrate-material is transferred onto the indenter-surface resulting in increased wear and friction in the ploughing experiments. This phenomena, termed ‘wedging’ is further explained in section 5.3.3.

Having validated the numerical coefficient of friction results for various loads, indenter size and interfaces, MPM-based ploughing simulations have been used to study the contribution of interfacial shear strength to the total coefficient of friction. The ratio of the coefficient of friction due to interfacial shear strength and the total coefficient of friction $\mu = \mu_s/\mu$, for various loads is shown in equation (11.1). The friction force due to ploughing has been calculated by running MPM simulations without any interfacial shear. This had been done theoretically by taking the coefficients of expressions in equations (9.1) and (9.3) as zero. The resulting friction force has been attributed to the plastic deformation of the substrate only and termed as ‘ploughing friction’. The coefficient of ploughing friction $\mu_p$ has been subtracted from the total coefficient of friction $\mu$ and then normalized by $\mu$ to obtain $\mu$ as shown in equation (11.1).

$$\frac{\mu}{\mu} = 1 - \frac{\mu_s}{\mu} = \mu_p = (1 - \mu)\mu$$

(11.1)

5.3.2. Effect on deformation of substrate

The substrate deformation due to ploughing also depends on the interfacial shear strength. The wear track on the substrate is observed under the confocal microscope to obtain the ploughed profile as shown in Fig. 14a. The cross-section of the ploughed profile is plotted in Fig. 14b in the $\gamma\gamma$-plane where the sliding direction of the indenter is along $x$-axis. It can be seen that the increase in interfacial shear strength, in the absence of lubricant, increases the deformation of the substrate and hence the ploughing depth. The ploughing depth $d$ is calculated as the sum of the groove depth $d_g$ and the pile-up height $h_{pu}$ as shown in Fig. 14b.

The ploughing depths are then computed for loads ranging from 1 to 46 N for both lubricated and unlubricated substrates. The ploughing depth for numerical simulations are converged for particle-cell resolutions of 5–20μm.

The ploughing depths obtained from the MPM-based ploughing simulations for ploughing of lubricated and unlubricated steel substrate by a 3 mm diameter indenter have been validated against ploughing
experiments in Fig. 15a. The ploughing simulation results agree well with the experimental results for the given range of loads both in lubricated and unlubricated conditions. The agreement also extends the validation of ploughing depths in Ref. [32] for different interfaces and indenter sizes. An increase in ploughing depth increasing with increasing the applied load can also be seen in the absence of lubricant.

The MPM-ploughing simulations done in the absence of interfacial shear strength by taking coefficients in equations (9.1) and (9.3) as zero have been used to compute contribution of the deformation in the substrate to the ploughing depth. The increase in ploughing depth due to interfacial shear has been characterized by normalizing the ploughing depth for (lubricated and unlubricated) substrates with interfacial friction $d_{\mu}$ to the ploughing depth of substrate without interfacial friction $d_{0}$, given as $\frac{d}{d_{0}}$ in equation (11.2) and shown in Fig. 15b. The ratio for all loads, indenter sizes and interfaces exceeds unity. An additional component of force acts on the substrate, due to shearing at the interface, in the direction opposite to the direction of plastic flow. For plastic flow beneath the indenter, the interfacial friction force on the substrate also has components acting in the $z$ direction. This component of interfacial friction along with the applied load in $z$ direction, causes a biaxial stress state on the substrate elements in contact which assists the deformation of the substrate and increases the ploughing depth of the substrate. Also, the resistance of friction force due to interfacial shear to the plastic flow of the deformed substrate around the indenter which acts reduces the ploughing depth.

Fig. 11. Variation of boundary layer shear strength with nominal contact pressure at a sliding velocity of 1 mm/s for zinc-coated GI sheet lubricated with (a) Quaker Ferrocoat N136 (b) Fuchs Anticorit PLS100T and (c) their comparison with each other and (d) with uncoated steel sheet.

Fig. 12. Coefficient of friction vs sliding distance for a 3 mm diameter spherical tip indenter sliding through ‘Quaker’ lubricated and unlubricated DX-56 steel substrate for (a) ploughing experiments and (b) MPM-based ploughing simulations at an applied load of 16 N (smaller sliding distance in ploughing model is to reduce the computation time).
Both the effects of interfacial friction force in resulting in ‘biaxial stress-state’ and ‘resistance to plastic flow’ act against each other. However, the increase in ploughing depth with increasing interfacial shear indicates the dominance of the ‘multiaxial stress-state’ effect. As the contact area increases with the applied load, the magnitude of interfacial friction force increases and hence the ploughing depth ratio increases. At higher loads the substrate not only flows underneath the indenter, but also piles up and flow around the indenter. The section of piled up substrate experiences interfacial friction force along the (loading) \(z\)–direction. Hence the ‘biaxial stress-state’ effect diminishes for the piled-up substrate at higher loads and the rate of increase of \(\bar{d}\) decreases and becomes almost constant at higher loads (Fig. 15b). The initial rate of increase of \(\bar{d}\) is higher for 3 mm diameter indenter where the interfacial shear strength is a major contributor to total friction force as shown in equation (11.1). Hence the total friction force is given as a factor of the total friction force due to plastic deformation of the substrate (ploughing), i.e. \(F_f = k_\mu F_p\). This corresponds to the theory in Bowden and Tabor [1] where the friction force due to adhesion \(F_a\), given using factor \(\tilde{\mu}\) as \(\tilde{\mu}F_f\), and ploughing \(F_p\) are independent of each other and their sum is the total friction force. Hence, the Bowden and Tabor relation for total friction force holds for large indenters at large loads. Thus the total friction force for large indenters at high loads is given in equation (11.3). So, the factors \(k_\mu\) and \(\tilde{\mu}\) are related as \(k_\mu = (1 - \tilde{\mu})^{-1}\) and \(\tilde{\mu} = \tilde{\mu}\).

\[
F_f = F_a + F_p = \tilde{\mu}F_f + F_p \Rightarrow F_f = \frac{F_p}{1 - \tilde{\mu}} 
\]

(11.3)

5.3.3. Effect on the wear behaviour

The increase in interfacial shear strength results in transition of the wear mode from ‘ploughing’ to ‘wedging’ based on the ‘wear mode diagram’ for sliding of a single-asperity [34]. The absence of lubricant at the sliding metallic interface can result ‘wedging’ and possible material transfer [26]. The wear mode diagram plots the wear modes as a function of the ‘angle of attack’ and the ‘interfacial friction factor’. The angle of attack for a spherical asperity is defined as the angle made by the tangent to the sphere at the point of contact with the substrate in the \(xz\) plane with the sliding \(x\)-direction. The angle of attack \(\chi\) for a sphere of radius \(r\) and ploughing depth \(d\) is computed from equation

\[
\bar{d} = \frac{d_f}{d_0} 
\]

(11.2)
The ‘interfacial friction factor’ \( f_{hk} \) is the ratio of the interfacial shear strength \( \tau \) to the bulk shear strength of the substrate \( \tau_{cu} \). If the angle of attack exceeds the critical value of \( \gamma_{pl,ul}^{\text{MP}} \) (see equation (12.2)) for \( f_{hk} \in (0.5, 1) \) there is transition from ploughing to the ‘wedging’ wear mode. The transition to the cutting wear mode is achieved for an angle of attack \( \gamma_{pl,ul}^{\text{MP}} \) are approximated by equation (12.3) [35].

\[
\tan \gamma = \frac{a}{b} = \frac{\sqrt{d(2b - d)}}{2b - d} \tag{12.1}
\]

\[
\gamma_{pl,ul}^{\text{MP}} = \arccos f_{hk} \quad \forall f_{hk} \in (0.5, 1) \tag{12.2}
\]

\[
\gamma_{pl,ul}^{\text{MP}} = \left( \pi - \arccos f_{hk} \right) / 4 \tag{12.3}
\]

The surface of the spherical tip of the indenter was observed under the confocal microscope after each experiment to check for material transfer. For lubricated ploughing experiments, the value of the interfacial friction factor is less than 0.5 which is why the wear mode is generally ploughing or cutting. The maximum ploughing depth obtained was 30 μm, for an (maximum) applied load of 46 N on the 1 mm diameter indenter. The corresponding maximum angle of attack computed was \( \gamma_{pl,ul}^{\text{MP}} = 19.6^\circ \). From the wear mode diagram in Fig. 16, it can be seen that the angular friction factor corresponding to the maximum angle of attack for ploughing on a lubricated surface \( \gamma_{pl,ul}^{\text{MP}} \) corresponds to \( f_{hk} < 0.8 \) (Fig. 16). Ploughing experiments on lubricated substrates have been done on steel sheets prepared by shine polishing and by mirror polishing as explained in section 3.2. The shine polished substrate formed a stable oxide film which prevents direct metal-metal contact during ploughing. Hence transition into the wedging wear mode occurs at a ploughing depth of 8.6 μm which corresponds to an attack angle \( \gamma_{pl,ul}^{\text{MP}} \) of 6.6°. The interfacial friction factor corresponding to the transition attack angle \( \gamma_{pl,ul}^{\text{MP}} \) for shine polished substrate is \( f_{hk} = 0.95 \). For a mirror polished substrate, the transition to wedging occurs almost immediately at low ploughing depth of 1.6 μm. This transition corresponds to an attack angle \( \gamma_{pl,ul}^{\text{MP}} = 1.3^\circ \) and an interfacial friction factor of \( f_{hk} = 0.99 \). This implies that the interfacial shear strength for a fresh, clean ‘mirror polished’ substrate (defined in section 3.2) is close to its bulk shear strength.

The surface of the 3 mm spherical pins, for experiments done using the mirror polished and shine polished substrate after the transition to wedging wear mode has been shown in Fig. 17. The material transferred from the groove of the substrate is stacked and hardened as wedges on the surface of pins as shown in Fig. 17b and c. It can be seen that the transition to the wedging wear mode occurs at a low ploughing depth of 1.1 μm for an unlubricated, clean mirror polished substrate as compared to a high ploughing depth of 9.8 μm for a shine polished substrate (defined in section 3.2) which has been used approximately after 1000 h of the polishing process. The ploughing depths of 1.1 μm and 9.8 μm correspond to the applied loads of 4 N and 46 N respectively. This transition in wear mode corresponds to the computed attack angle and interfacial friction factor mapped in Fig. 16.

The modelling of the wedging wear mode is challenging using the MPM-based ploughing model. This is because, to model material removal from the substrate, its transfer and adhesion onto the surface of the asperity on unloading requires a robust damage model and adhesion model. Such models have not been implemented in the current MPM-based ploughing model. However, the friction and deformation of the substrate in the MPM-based ploughing model has been compared to the experiments for the same experimental parameters resulting in wedging as shown in Fig. 18. The friction plot in Fig. 18a shows the fluctuations in the friction with sliding distance due to pile-up and stacking of the wedges (lumps of substrate material) in front of the indenter during wedging. This pile-up of the substrate material can be also seen by checking the mean position of groups of particles along the sliding length of the ploughed track as shown in Fig. 18b. It can be seen that the particle pile-up to subsequently higher height (mean + z coordinate) as the asperity moves along the substrate. However, due to absence of adhesion between the asperity and substrate the MPM particles do not stick to the asperity as in case of wedging in experiments. Thus the MPM-based ploughing model can be used to compute friction and ploughing depth for various interfaces given their interfacial shear strength.
6. Conclusion

An experimental method to characterize the interfacial shear strength of lubricated and unlubricated contacts (coated and uncoated) has been developed. The interfacial shear strength has been fitted with empirical relations to express it as a function of contact pressure and sliding velocity. The fitted relations have been successfully implemented in the MPM-based ploughing model and the results have been validated against ploughing experiments with good agreement. The MPM-based ploughing model has been further used to understand the effect of interfacial shear strength of the friction and wear behaviour of a single asperity sliding through a steel substrate. The Bowden and Tabor [1] relationship for friction in sliding contacts has been shown to hold for large indenters at large applied loads. In these cases, the contribution of the interfacial shear strength and plastic deformation to the total friction force and the ploughing depth is shown to be independent and additive. An experimental study on the transition from ploughing to wedging wear mode has been carried out by varying the interfacial shear strength and the transition has been explained using the wear mode diagram for spherical indenters [34].

Acknowledgments

This research was carried out under project number S22.1.14520a.
