Chapter 4
Surface finish and friction in cold metal rolling

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4.1 INTRODUCTION

The cold metal rolling industry needs reliable and accurate models to improve predictions of surface finish and friction and thus increase productivity and improve quality. These models are used in setup programs for schedule optimisation, for on-line control algorithms and to help determine process improvements.

In many cold rolling operations, lubricant is used to reduce frictional forces, to protect the roll and strip surfaces, and to act as a coolant. The amount of oil drawn into the roll bite and the initial surface roughness are the critical factors determining friction in the contact and surface finish of the product (Schey, 1983a and 1983b). Figure 4.1(a) illustrates schematically the inlet to the bite during a strip rolling operation. Neglecting for the moment the surface roughness on the roll and strip, a 'smooth' film thickness $h_s$ at the end of the inlet can be determined by integrating Reynold's equation. Wilson and Walowit (1972) derive an expression for $h_s$ as

$$h_s = \frac{6\eta_0 a \bar{u}}{\theta_0 (1 - \exp(-aY))}$$

(4.1)

where $\bar{u}$ is the average entraining velocity, $\theta_0$ is the inlet angle between the strip and roll, $Y$ is the plain strain yield strength of the strip and $\eta_0$ is the viscosity of the lubricant at ambient pressure. The pressure viscosity coefficient $\alpha$ in the Barus equation $\eta = \eta_0 \exp(\alpha p)$ is used to describe the variation of viscosity $\eta$ with pressure $p$.

In practice both the roll and the strip surfaces are rough. In many rolling processes, the roll grinding process leads to a roll roughness with a pronounced lay, with asperities running along the rolling direction. This longitudinal roughness is in turn transferred to the strip. Isotropic roughness is also common, for example where the surface has been produced by shot blasting, while transverse roughness, with asperities
running perpendicular to the rolling direction, seems to be less common. In the presence of roughness, the ratio \( \Lambda_s = h_s/\sigma_0 \) of the smooth film thickness \( h_s \) to the combined roll and strip roughness \( \sigma_0 \) (\( \sigma_0^2 = \sigma_r^2 + \sigma_s^2 \)) is used to characterise the lubrication regime. For large \( \Lambda_s \), the surfaces are kept apart by a continuous film of oil. However, the surface tends to roughen in these circumstances, probably due to differential deformation of grains in a manner akin to the surface roughening observed in a tensile test (Schey, 1983a). Various authors have derived models of friction under these full film conditions (e.g. Sa and Wilson, 1994). However, to generate a good surface finish, most cold rolling operates in the 'mixed lubrication' regime, where there is some hydrodynamic action drawing lubricant into the bite, but also some contact between the asperities on the roll and strip. First asperity contact occurs with \( \Lambda_s \) below about 3, so that to give substantial asperity contact and a good imprint of the smooth rolls onto the strip, \( \Lambda_s \) needs to be below about 1. Figure 4.1(b) shows schematically the contact in these circumstances, with areas of 'contact' between the roll and strip and areas separated by an oil film. The ratio of the areas of close contact to the nominal contact area is termed the contact area ratio \( A \).

*Figure 4.1* (a) Schematic of lubrication mechanisms in rolling, (b) details of contact

Prediction of the area of contact ratio \( A \) requires detailed models of how the asperity contacts deform (reviewed in section 4.2) and the role of the pressurised oil separating the roll and strip. The average friction stress through the bite can then be determined using appropriate expressions for the frictional stress on the contact areas and in the valleys. Section 4.3 describes such models, and corresponding experimental findings.

Figures 4.2(a) and (b) illustrate the way in which the roll imprints its roughness onto aluminium foil in the mixed lubrication regime. It seems probable that the pits observed on the foil surface are associated with surface roughening in regions where there is a significant oil film separating the roll and foil.
Although it is frequently assumed that the interface between the roll and strip can be separated into 'contact' and valley areas, in fact it is an open question as to the nature of the contact, which may be dry, or separated by a boundary or thin hydrodynamic film. The answer probably depends on rolling and local conditions. The mechanism of lubrication under these asperity contacts has been termed micro-plasto-hydrodynamic lubrication (MPHL). A similar mechanism can occur in stainless steel rolling, where the shot-blast finish on the hot band prior to rolling tends to generate pits on the surface during subsequent deformation. Oil trapped in the pits can be drawn out due to the sliding between the roll and strip, as illustrated in Figure 4.1. Further details of the MPHL mechanism are given in section 4.4, while section 4.5 describes work on boundary lubrication. Finally the significant effect of transfer films formed on roll surfaces is reviewed in section 4.6.

4.2 UNLUBRICATED ROLLING

Although it is sometimes possible to roll without lubrication, in general lubricant is applied during cold rolling to act as a coolant, to prevent surface damage and to reduce friction. Nevertheless the deformation of surfaces under unlubricated conditions provides a useful starting point for lubricated rolling, and points up the key features in the mechanics of asperity deformation.

4.2.1 Without bulk deformation

Bowden and Tabor (1950) introduced the idea of asperity junctions as a mechanism for friction between rough surfaces. In the simplest case each asperity contact acts as a small indentation, so that the area of each contact equals the ratio of the
force at the contact to the material hardness $H$. Summing up all the contacts, the ratio $A$ of the true to the nominal area of contact is related to the average contact pressure $\overline{p}$ by

$$A = \frac{\overline{p}}{H}$$  \hfill (4.2)

This expression would indicate an area of contact ratio close to one third at a typical mean contact pressure equal to the yield stress of the strip. Modifications to this theory at higher pressure, for example by Childs (1977) and Bay and Wanheim (1976), allow for interaction between adjacent contacts, which tends to limit the amount of asperity flattening. Wave models have also been proposed to account for sliding between the tool and surface (Challen and Oxley, 1979). Although these models have been widely used in metal working, they do not allow for deformation of the bulk material, so that only the surface of the workpiece undergoes large strains. As we shall see in the following section, this seriously limits the applicability of these models to metal forming problems.

4.2.2 With bulk deformation

The effect of bulk plasticity was highlighted by the results of Greenwood and Rowe (1965) and Fogg (1968), who showed that the presence of sub-surface deformation allows the asperities to be crushed considerably more than in the absence of bulk deformation. The deformation throughout the substrate means that it is at the point of yield and acts as a soft 'swamp'; differences in normal velocity at the surface associated with differences in contact pressure can easily be accommodated by a perturbation to the uniform plastic strain field. Sheu and Wilson (1983), Wilson and Sheu (1988) and Sutcliffe (1988) show how the evolution of the workpiece roughness topography depends on the bulk strain of the material. Depending on the orientation of the roughness relative to the bulk strain direction, either upper-bound or slip line field solutions are used to derive the velocity field at the surface and the corresponding relationship between contact pressure and asperity flattening rate. Figure 4.3 shows Sutcliffe’s idealisation of the contact geometry for transverse roughness with a periodic array of flat indenters of width $A\lambda$ spaced a distance $\lambda$ apart, loading the surface of the workpiece with a mean contact pressure $\overline{p}$. Figure 4.4 shows the corresponding slip line field. The local field under the indenter is that due to Hill (1950) for indentation of a strip of limited height. This can be matched up with a uniform deformation pattern in the centre of the strip, given by a series of square rigid blocks separated by velocity discontinuities. The region at the surface between the asperities is rigid and undeforming. The flattening rate is expressed in terms of a dimensionless flattening rate $W$

$$W = \frac{2\Delta v}{\lambda\dot{e}}$$  \hfill (4.3)
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where \( \Delta v = v_c - v_v \) is the relative flattening velocity between the plateau and valley of each asperity and \( \dot{\varepsilon} \) is the bulk strain rate.

Figure 4.3 Schematic of asperity contact model (Sutcliffe, 1988)

Figure 4.4 Slip line field solution for transverse roughness (Sutcliffe, 1988)
The flattening rate $W$ is a function of the real area of contact area ratio $A$ and the ratio $\bar{p}/Y$ of the mean contact pressure to the plane strain yield stress of the strip, i.e.

$$W = W\left(\frac{\bar{p}}{Y}, A\right)$$  \hspace{1cm} (4.4)

The theoretical flattening rate $W$ is in the range 1–8, increasing with increasing mean pressure and falling with increasing area of contact ratio.

Sheu and Wilson (1983) and Sutcliffe (1988) used upper-bound theory to address the closely related and industrially more relevant problem of roughness running along the rolling direction. Wilson and co workers adopt an alternative formulation\(^1\), expressing the problem in terms of an equivalent asperity hardness $H$, given by the mean asperity pressure $\bar{p}/A$ divided by the workpiece shear yield stress $k$ ($k = Y/2$ for plane strain conditions),

$$H = \frac{\bar{p}}{Ak}$$  \hspace{1cm} (4.5)

The hardness $H$ is expressed as a function of a dimensionless strain rate $E$, where

$$E = \frac{\lambda\dot{\varepsilon}}{2\Delta v}$$  \hspace{1cm} (4.6)

Note that $E$ is equal to $1/W$. Hence the problem has the functional form

$$H = H(E, A)$$  \hspace{1cm} (4.7)

Where there is bulk deformation, so that $E$ is not zero, the hardness $H$ is much less than the normal material hardness $H$ without bulk deformation. The hardness $H$ falls with increasing $E$ and rises with increasing area of contact ratio $A$.

To use these asperity flattening models to predict the change in asperity geometry and contact area during rolling, consider the schematic geometry of Figure 4.5, with an array of triangular asperities of slope $\theta$ which it is assumed does not change during the subsequent deformation (experiments and finite element calculations suggest that this is a reasonable assumption). For longitudinal roughness (where the asperities run along the rolling direction), the rate of change of contact area with bulk strain is given by

\(^1\) Note that Wilson uses the semi-wavelength $A/2$ in his work.
while for transverse roughness the equation is modified to take into account the increase in indenter spacing:

\[ \frac{dA}{dε} = \frac{1}{1+ε} \left( \frac{W}{\tan θ} - A \right) \]  \hspace{1cm} (4.9)

Equations (4.8) and (4.9) show how the increase in contact area, and hence the reduction in surface roughness, depends on the bulk strain. Increasing the normal pressure corresponds to a larger value of the crushing rate \( W \), and so an increase in the rate at which the contacts grow. Increasing asperity slope reduces the crushing rate.

Sutcliffe (1988) describes a series of experiments in which he deformed copper blocks containing machined triangular asperities using flat rigid platens. Figure 4.6 shows the side of one such specimen, for the case where the roughness lay is transverse to the bulk straining direction. The bright areas show regions where a film of dark die applied to the surface has been disrupted due to straining. The pattern of deformation, with non-deforming regions between indenters and a square pattern of intense shear lines in the middle of the block, is remarkably similar to the corresponding theory shown in Figure 4.4.
A comparison of experimental measurements with theoretical predictions of the evolution with bulk strain of the contact area is shown in Figure 4.7(a). Good agreement is seen for high normal pressures. Note that the area of contact ratio rapidly rises above the value of about 1/3 associated with zero bulk strain. The excellent conformance achievable between the surfaces explains the remarkable efficiency of rolling as a process to produce good surface finish by imprinting a bright roll finish. At lower normal pressures, the applied end tension (equivalent to coiling tensions in rolling) must be correspondingly higher to cause the bulk material to yield. In these circumstances the agreement is not so good, perhaps related to work hardening in the experiments or to changes in the geometry of the asperities during the deformation (see also the calculations of Korzekwa et al, 1992). Wilson and Sheu (1988) show excellent agreement between the predictions of their model and measurements of the change in contact area after rolling aluminium strips with model asperities, see Figure 4.7(b). The change in contact area is the same for either single or multiple pass rolling, depending only on the total bulk strain.
Various researchers have addressed the asperity crushing problem using finite elements. Makinouchi et al (1987) and Ike and Makinouchi (1990) considered transverse roughness, finding a similar pattern of non-deforming and deforming regions to that predicted by Sutcliffe. Korzekwa et al (1992) have undertaken a comprehensive series of numerical simulations to investigate the effect of strain direction on the asperity flattening rate. Their numerical results for transverse roughness show good agreement with the slip line field solutions of Sutcliffe for relatively high flattening rates. At the lowest flattening rates the slip line field solution predicts higher flattening rates. Probably the rigid-perfectly plastic assumption used in the slip line field leads to an overestimate of the flattening rate in these cases. For longitudinal roughness Korzekwa’s results show similar quantitative results to the upper-bound calculations of Wilson and Sheu (1988), although they differ quantitatively. Korzekwa’s results for this case have been fitted using appropriate polynomial functions by Sutcliffe (1999).

In the above analyses the contact geometry is idealised by assuming that the tool is smooth and all the roughness resides on the strip. In practice the roll tends to have a very similar roughness to the strip, not least because the strip surface conforms to the tool roughness after the first pass of a multi-pass schedule. Where the roughness is entirely longitudinal, the smooth tool and equivalent strip approximation is appropriate. However for transverse roughness there is an additional ploughing effect, as the relative motion between the surfaces induces continual deformation (see Figure 4.8). Wilson and Sheu (1988) and Wilson (1991) consider this case for a rough tool and smooth strip. Now the flattening rate only depends on the local tool slope at the contact, rather than the history of pressure and strain.

![Figure 4.8 Schematic of boundary friction model of transverse roughness with sliding; (a) partial contact, (b) complete contact (Wilson, 1991)](image)
4.2.3 Random rough surfaces

The above studies relate to asperity geometries consisting of a series of identical triangular asperities. Real rough surfaces generally contain a random profile. By choosing the initial shape of the asperity with the same height distribution as the real surface, the distribution in asperity heights can be modelled, for example using the pseudo-Gaussian distribution of Christensen (1970). The variation in asperity spacing is also important, since the crushing rate is inversely proportional to the asperity slope.

Steffensen and Wanheim (1977) show how the inclusion of short wavelength asperities, superimposed on long wavelength asperities, can reduce significantly the predicted area of contact ratio between tool and workpiece in the case where there is no bulk deformation. Although this may make relatively little difference to the roughness amplitude, section 4.3.3 shows that it can impact significantly on predictions of friction. Wilson (1991) describes a model including bulk deformation, with long and short wavelength asperities on the strip tool, respectively.

Sutcliffe (1999) treated a two-wavelength longitudinal surface profile, applying the asperity flattening model to each wavelength in turn. The long wavelength components are predicted to be eliminated more quickly that the short wavelength, high slope components, c.f. Eq. (4.8). Figure 4.9 shows the excellent agreement obtained between predictions and measurements of the change with bulk strain of the amplitude of the two components of roughness. Measurements were taken from as-received aluminium strips which were cold rolled to various reductions. The amplitudes of the roughness components were extracted by Fourier analysis. One important conclusion from this work was that the short wavelength components (typically with wavelengths less than 30 µm) can be expected to have a significant impact on friction, while the longer wavelength components, although making a greater contribution to the initial amplitude of roughness, are nevertheless crushed rather quickly so are less important. It is also important to note that, for these typical roughness slopes, the asperities have almost entirely conformed to the roll roughness after a bulk strain of only 10%.

![Figure 4.9 Change with bulk strain in strip roughness amplitudes \( \sigma \) and \( \Sigma \) for short and long wavelength components respectively, normalised by their initial amplitudes \( \sigma_0 \) and \( \Sigma_0 \) (Sutcliffe, 1999)](image-url)
4.3 MIXED LUBRICATION

The previous section has revealed the importance of bulk deformation in allowing a close conformance of the strip surface to that of the roll. In practice lubricant is applied to act as a coolant and to reduce friction. In this section we show how the presence of lubricant reduces the extent to which the strip and tool surfaces can come into contact. This section considers with 'mixed lubrication', where the amount of oil drawn into the contact is determined by oil entrainment at the inlet to the bite. Montmitonnet (2001) provides a useful summary of recent modelling approaches in this regime. Section 4.4 deals with an alternative mechanism, whereby oil is trapped in pits at the inlet, and then drawn out from the pits due to sliding action in the bite.

This chapter considers lubrication with neat lubricants. In practice the lubricant is frequently applied as an emulsion of water and oil. A review of how this affects the tribological conditions is given by Schmid and Wilson (1995).

4.3.1 Modelling

This section reviews the key elements in a tribological rolling model. The various approaches adopted by different authors are summarised in section 4.3.1.5, experimental methods are described in section 4.3.2 and the predictions are compared with experiments in 4.3.3. Further experimental findings are reviewed in section 4.3.4.

4.3.1.1 Asperity deformation. Section 4.2 describes appropriate models of asperity deformation. As indicated there, it is essential to include the effect of bulk deformation, modelling the evolution of contact area with bulk strain. The hydrodynamic oil pressure \( p_v \) generated in the valleys tends to reduce the crushing rate, since it is the difference between the plateau and valley pressures \( p_p \) and \( p_v \) that drives asperity crushing. This oil pressure can simply be accounted by subtracting \( p_v \) off the mean pressure \( \bar{p} \), to be used in Eqs. (4.4) or (4.5) for the crushing rate. Longitudinal roughness, transverse and isotropic roughness orientations have been considered in the various models of mixed lubrication.

Several models of rolling consider the contact area as a function only of the local pressure, ignoring the effect of bulk strain on the evolution of the geometry (as indicated in Tables 4.1 and 4.2). Although simpler, this approach is not appropriate where there is bulk strain.

4.3.1.2 Hydrodynamic equations. A hydrodynamic model is needed to predict the change through the contact in the oil pressure \( p_v \), generated in the valleys. The basis for all the models is Reynolds' equation, which for steady flow with smooth surfaces separated by a film of thickness \( h \) takes the form

\[
\frac{dp_v}{dx} = 12\eta \bar{u} \frac{h - h^*}{h^3}
\]  

(4.10)
where \( h^\ast \) is a constant. In many cases \( h^\ast \) is equal to the film thickness at the start of the bite, where the pressure gradient approaches zero. Used in this form, Wilson and Walowit's equation would be recovered for a straight inlet wedge. A 'correction' to include roughness has been used by making some assumption about the height of the asperities in the work zone, and hence the oil trapped there. In fact adherents to this method do not allow for the critical effects of hydrodynamic pressure and bulk deformation on the mechanics of asperity crushing, limiting its accuracy.

Two approaches have been widely used to include roughness in the Reynolds' equation. For an assumed deterministic roughness geometry, an 'averaged Reynolds' equation' can be derived (Christensen, 1970) to give the variation in the rolling direction, along the \( x \) axis, of the reduced hydrodynamic pressure \( q_v \) of the lubricant as

\[
\frac{d(\alpha q_v)}{dx} = 12\eta_0 \alpha \Pi \frac{h_v - h^\ast}{h_v^3 \left[ 1 + 3\sigma^2 / h_v^2 \right]}
\]  

(4.11)

where \( h_v \) is the mean film thickness, averaged across the valley, and \( \sigma^2 \) is the local combined variance of the roll and strip roughnesses. As the asperities are crushed through the bite the geometry parameters \( h_v \) and \( \sigma^2 \) fall. The reduced and actual lubricant pressures \( q_v \) and \( p_v \) are related by \( \alpha q_v = 1 - \exp(\alpha p_v) \).


\[
\frac{d}{dx} \left( \phi_t \frac{h_t^3}{12\eta} \frac{dp_t}{dx} \right) = \mu \frac{dh_t}{dx} + \frac{\Delta u}{2} \sigma \frac{d\phi_t}{dx}
\]  

(4.12)

where \( h_t \) is the mean film thickness, \( \Delta u \) is the sliding speed and \( \phi_t \) and \( \phi_s \) are pressure and shear flow factors. These flow factors depend on the Peklenik surface orientation parameter \( \gamma \), which is the ratio of correlation lengths in the rolling and transverse directions. Values of Peklenik parameter of infinity, 0 and 1 correspond to longitudinal, transverse and isotopic roughness, respectively. The flow factors are fitted with semi-empirical equations by Wilson and Marsault (1998) and given by Lin et al (1998) as explicit functions of the depth of the pits and the orientation parameter \( \gamma \).

4.3.1.3 Method of solution. Models of rolling need to couple the asperity mechanics and hydrodynamic equations. As the asperities are crushed in the bite, the film thickness falls, thus leading to an increase in valley oil pressure. Conversely this increase in valley pressure limits the crushing process. For 'high speed' conditions, the crushing process takes up only a short part of the entry region. Sheu and Wilson (1994) introduce short entry and transition regions at the inlet, prior to and after the onset of bulk plasticity, respectively. At the end of the transition region, the oil and plateau pressures
are effectively the same. In the work zone there is some subsequent reduction in asperity height, as elongation of the strip leads to stretching of the oil film and a corresponding reduction in film thickness. Solution of the coupled ordinary differential equations for the asperity crushing and oil pressure mechanisms gives the oil flow rate through the bite and hence the film thickness in the bite. This approach has been widely used (e.g. Lin et al., 1998, Marsault, 1998). Sutcliffe and Johnson (1990b) adopt a similar approach, but only consider an inlet analysis, in which the bite region is not modelled explicitly.

Le and Sutcliffe (2000a, 2001a) extend this approach to include more than one wavelength of roughness. A first iteration considered the oil pressure to be the same in each valley. This was subsequently enhanced to model variations in pressure across the strip width, depending on the local asperity height.

For 'low speed' conditions, it is no longer appropriate to make the assumption that there is a short region in the inlet during which the oil pressure approaches the mean pressure. Instead the oil pressure is significantly below the mean pressure throughout the bite, and the whole bite needs to be modelling using the coupled equations (Chang et al, 1996). Qui et al (1999) describe an alternative approach, using a finite difference method, which allows them to generate a more stable solution method which can be applied to both the 'high' and 'low' speed cases. Examples of these solutions are given in section 4.2.3 below.

4.3.1.4 Friction modelling. The tribological models described above allow the contact geometry, area of contact ratio, contact pressure and oil hydrodynamic pressure to be found through the bite. Noting that the valley and asperity pressures equal the mean pressure in the work zone for high speed conditions, the effective friction coefficient $\mu$ is given

$$\mu = A\mu_c + (1-A)\mu_v$$

where $\mu_c$ and $\mu_v$ are friction coefficients for the areas of contact and valley regions respectively. An alternative approach uses a friction factor $m$, where the local shear stress $\tau$ equals $mk$, with $k$ the shear yield stress of the workpiece ($Y = 2k$ for plane strain conditions). Then the effective friction factor $m$ is given by

$$m = Am_c + (1-A)m_v$$

where $m_c$ and $m_v$ are friction factors for the contact and valley regions.

The contribution from the contact areas tends to dominate, since the contact area is normally greater than the valley area and the frictional stresses are higher on the contact areas. Hence it is important to use an accurate value for $\mu_c$ or $m_c$. Unfortunately it is not clear what an appropriate value should be, or how to model it. Clearly this will depend on the mechanism of friction on the contact regions, but this still remains an open question, with boundary lubrication or some form of thin hydrodynamic film
perhaps governing friction. Guidance on this has to come from experimental measurements, and to a certain extent it remains a fitting parameter. Further details of this point are discussed in sections 4.4, 4.5 and 4.6 below. A typical value used in current models seems to be of the order $\mu = 0.1$ or $m_v = 0.2$. One refinement to the model allows the friction coefficient or factor to depend on the smooth film thickness or rolling speed. The rationale behind this approach is that there is some hydrodynamic component to friction under these contact areas. Montmitonnet et al (2001) infer an appropriate variation in friction coefficient from rolling mill trials, while Le and Sutcliffe propose a 'semi-empirical' friction law (2001b), using the simpler strip-drawing test to derive a variation of contact friction coefficient with smooth film thickness.

The frictional stress $\tau$ in the valley regions, and hence the valley friction coefficient $\mu_v$ or friction factor $m_v$, can be estimated using the Newtonian viscosity:

$$\tau = \dot{\gamma}$$

where the strain rate $\dot{\gamma}$ is estimated from the sliding speed and valley film thickness in the bite and the viscosity is taken at the pressure in the bite.

An improved estimate of the frictional stress considers non-Newtonian behaviour. Measurements of lubricant behaviour, for example in a pressured cell (Bair and Winer, 1982), or inferred from twin-disc experiments (Evans and Johnson, 1986), show that the effective viscosity falls at high shear rate, and may reach a plastic limit in some cases. Evans and Johnson follow the Eyring model,

$$\tau = \tau_0 \sinh \left( \frac{\eta \dot{\gamma}}{\tau_0} \right)$$

where $\tau_0$ is the Eyring stress at which significant non-linearity sets in. A limit on the shear stress is imposed, associated with plastic behaviour. Bair and Winer propose a relationship which gives a smooth transition from Newtonian to perfectly-plastic behaviour. Even without such a non-Newtonian model, it seems sensible to set an upper limit on shear stress, for example equal to the corresponding boundary friction coefficient. In general the hydrodynamic component will be a small contribution compared with the contact friction. An exception to this occurs when the pressure or strain rate is very high, as for example occurs in foil rolling or under asperity contacts.

4.3.1.5 Summary of models

The above sections have described the key factors included in modelling of mixed lubrication. Tables 4.1 and 4.2 summarise the actual methods adopted by various authors, for strip and foil rolling respectively. The foil models include significant roll elasticity, as described in section 4.3.5. Examples of the results are given in section
4.3.3 below. The second column of each table indicates whether the effects of bulk deformation and hydrodynamic pressure generated in the valleys are included in the model of asperity flattening.

Table 4.1 Summary of mixed lubrication models for strip rolling

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<tr>
<th></th>
<th>Bulk deformation and valley pressure?</th>
<th>Reynolds' equation</th>
<th>Rolling 'speed'</th>
<th>Notes</th>
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<tr>
<td>Tsao and Sargent, 1977</td>
<td>No</td>
<td>Smooth</td>
<td>–</td>
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<td>Keife and Sjögren, 1994</td>
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<td>Smooth</td>
<td>–</td>
<td></td>
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<tr>
<td>Li, 1995</td>
<td>–</td>
<td>Smooth</td>
<td>–</td>
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<td>Two wavelength roughness</td>
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<td>Le and Sutcliffe, 2000a, 2001a</td>
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<td>High</td>
<td>Two wavelength roughness</td>
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<tr>
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<td>Averaged</td>
<td>High</td>
<td>Inlet analysis</td>
</tr>
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<td>High or low</td>
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<td>Flow factor</td>
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<td>Qui et al, 1999</td>
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Table 4.2 Summary of mixed lubrication models for foil rolling

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<td>Flow factor</td>
</tr>
<tr>
<td>Le and Sutcliffe, 2002a</td>
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<td>Averaged</td>
</tr>
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</table>

4.3.2 Experimental methods

Although there is an abundance of measurements of surface finish and friction in rolling, it can be difficult to apply the findings with confidence without information about the tribological mechanisms involved. Since experimental mills are invariably unable to replicate the industrial conditions, laboratory measurements must always be treated with caution. One way to overcome the slower speeds generally found in experimental mills is to use more viscous oils, although the boundary lubrication conditions may not then be matched. Alternative metal deformation processes, notably strip drawing and plane strain compression, have been widely used to investigate the
tribology of metal rolling. Again these experiments can give valuable insight, but care needs to be taken in applying them to rolling.

Measurements of surface finish in rolling are relatively easy to obtain, by taking samples either from lab experiments or industrially rolled coils. In the later case it is important to take the samples from a portion of the coil which has been rolled at normal speed, to ensure that the hydrodynamic conditions are appropriate. The surface finish can then be measured on a profilometer. Replica techniques are valuable to measure roughness of the roll. A recent development by Testex corporation, in which a crushable foam film replicates the roll profile, seems to be preferable to other replica techniques. Stout and Blunt (2000) give a comprehensive review of profilometry methods. Traditional line-trace profilometers are adequate to extract values of $R_q$ roughness. Some judgement needs to be made about appropriate filtering, to eliminate long wavelength components associated with roll form error or lack of flatness in the samples. Although a filter cut-off of 800 µm is often used, the results of Sutcliffe (1999) for dry rolling (see section 4.2.3 above) suggest that this may over-emphasise the long wavelengths of roughness. A shorter wavelength cut-off, of say 300 µm, may be more appropriate. Modern three-dimensional profilometers give valuable additional information about the shape of the surface topographic features. They have the additional advantage of scanning more of the surface for a given reading, so that the surface roughness tends to be averaged more effectively. To attain the required resolution, three dimensional measurements typically cover an area with side length of a few hundreds of microns, so that there is an inherent filtering effect in these measurements. An example of more sophisticated analysis of the surface is given in section 4.4.2, to characterise pits formed on stainless steel strip. Images using a scanning electron microscope (SEM) are also essential to confirm that the mechanisms of friction are similar to those found in the corresponding industrial application.

Azushima (1978) pioneered the use of the 'oil drop' experiment to estimate the amount of oil drawn into the bite. A small measured drop of oil of known weight is placed onto the cleaned strip which is rolled using cleaned rolls. As it passes through the bite the drop spreads over the surface of the strip. The cigar-shaped area of the drop can easily be identified because of differences in surface roughness between lubricated and unlubricated regions. Knowing this area and the lubricant density, the mean film thickness can be found. Sutcliffe and Johnson (1990a) show that this measurement corresponds well to the film thickness found from surface roughness measurements, and that starvation effects due to the limited oil drop size can be easily avoided. Li (1995) describes an alternative method using ellipsometry, while recent work has used laser fluorescence to measure film thickness directly (Azushima et al, 2000 - see also Chapter 5).

A direct measurement of friction in the roll bite can be found by using a pin embedded in the roll (e.g. Al-Salehi, 1973 and Jeswiet, 1998). Although the inferred measurements seem reasonable, some concern arises in the mixed regime that the presence of the pins may disturb the oil film found at the measurement point.
A widely used method is to infer friction from measurements of rolling parameters. In principle a measurement of the rolling load can be used to deduce the friction coefficient in the bite. However results will be sensitive to errors in the assumed yield stress. A more accurate inverse method is to use the additional information of the measured forward slip (i.e. the relative slip between the strip and roll at the exit, divided by the roll speed). This can be inferred from the distance between imprints transferred from the roll onto the strip or from direct measurements of the roll and strip speed. The strip speed can be measured using optical methods or from a bridle roll speed. Direct measurements of strip speed are more relevant to industrial rolling, where the roll and strip speeds are relatively stable, while it is not appropriate to mark the rolls. Inversion of an appropriate rolling model can then give the mean friction coefficient through the bite. Differences in inferred friction arise from different models of rolling. The method can be made more accurate by adjusting the coiling tensions so that the neutral point moves to the exit plane (zero forward slip). In this case the inferred frictional stress is less sensitive to the assumed distribution of friction stress through the bite.

4.3.3 Theoretical results and comparison with experiments

Figure 4.10 shows a typical result from Sheu and Wilson (1994) for the variation in pressure and contact area at the inlet to the bite. The mean pressure rises in the inlet region, for large $X$, due to hydrodynamic effects. At a dimensionless position $X \approx 2$, there is some contact between the roll and strip asperities, leading to a non-zero area of contact ratio $A$ and a mean pressure $p$ above the valley pressure $p_b$. At the end of the inlet region the mean pressure has reached the yield stress $\sigma$, so that bulk deformation begins. Further asperity flattening occurs in the transition zone, driven by the difference between the asperity and valley pressures. For the case illustrated, the increase in contact area is relatively small in this transition region; a more significant increase will occur at slower speeds.

Figure 4.10 Variation at the inlet to the bite of the mean pressure $p$, valley pressure $p_b$ and contribution from asperity contacts $Ap_a$ with corresponding increase in area of contact ratio $A$ (Sheu and Wilson, 1994)
Figure 4.11, from the predictions of Montmitonnet et al (2001), shows how the pressure variation through the bite varies with roll speed for typical rolling conditions. At high speed, \( v_r = 10 \text{ m/s} \), the results are similar to those of Sheu and Wilson, with the hydrodynamic pressure \( p_b \) reaching the mean pressure \( p \) in a very short inlet zone. The variation of pressure through the rest of the bite then shows the familiar friction hill. For a speed of 0.1 m/s, conditions are in the 'low speed' regime, where the hydrodynamic pressure lies below the mean pressure throughout the work zone. At very low speed, \( v_r = 0.001 \text{ m/s} \), the hydrodynamic effects are very small. Instead the asperities crush very quickly in the inlet, and oil is trapped in isolated pockets, reaching the mean pressure due to hydrostatic action. The subsequent asperity deformation would be governed in these circumstances by the MPHL mechanism (section 4.4.2).

Predictions of film thickness from the various models are in broad agreement, showing that the film thickness depends primarily on the rolling speed, oil properties and inlet geometry (Sutcliffe and Johnson, 1990a, Sheu and Wilson, 1994, Lin et al, 1998, Montmitonnet et al, 2001). The effects of yield stress, strip thickness, asperity geometry and entry tension are of secondary importance. Figure 4.12(a) shows reasonable agreement between experimental measurements of film thickness (Sutcliffe and Johnson, 1990b), measured using an oil drop experiment and theoretical predictions (Sutcliffe and Johnson, 1990a). The independent variable \( \Lambda \) is the ratio of the smooth film thickness, Eq. (4.1), to the initial combined surface roughness, so that values of this speed parameter below one correspond to significant asperity contact. The dependent variable \( \Lambda \) is the corresponding ratio of measured or predicted mean film thickness to the initial surface roughness. The modification to theory given in Figure 4.12(a) is associated with the asymmetrical lubrication conditions in these experiments.
Note the small dependence of results on the yield stress $Y$ of the strip. Corresponding results from Sheu and Wilson (1994) are given in Figure 4.12(b) for cold rolling of aluminium strip. Again agreement is good. Both the results shown in Figure 4.12 are for longitudinal roughness, i.e. the roughness lay runs in the rolling direction. Marsault (1998) shows reasonable agreement between experiments of Lin (1998) and theory, where the roughness is transverse, isotropic or longitudinal.

Rasp and Häfele (1998) provide a comprehensive series of measurements of film thickness under mixed lubrication conditions, using the oil drop technique on cold rolled aluminium, as illustrated in the results of Figure 4.13.

2 Experimental points differ from those given by Wilson and Sheu because inaccurate pressure and temperature viscosity coefficients used by them have been corrected.
Figure 4.13 shows how the entrained film thickness increases with increasing rolling speed and roll roughness. The effect of roughness orientation is also explored. By considering modifications to the smooth film thickness formula, Eq. (4.1), associated with roughness and thermal effects, Rasp and Häfele are able to show a good correlation between predicted and measured film thicknesses. However, their model does not include the details of asperity flattening described in section 4.2.2 above, limiting its predictive capability.

As shown in section 4.2.3, roughness wavelength has a significant effect on asperity flattening. To investigate this effect, Sutcliffe and Le (2000) performed a series of cold rolling experiments on as-received aluminium strip, which initially had a typical industrial longitudinal roughness. The changes in lubrication parameter $\Lambda_s$ were achieved by varying the rolling speed and by using three mineral oils spanning a wide range of viscosities. Figure 4.14 compares theoretical predictions using a two-wavelength model with measurements of the effect of lubrication parameter $\Lambda_s$ on the change in surface roughness amplitudes $\sigma$ and $\Sigma$ for the short and long wavelength components. Predictions are in good agreement with measurements in this case, for a strip thickness reduction of 25%. Note that, for $\Lambda_s > 1$, the surfaces roughen on the scale of the grain size during rolling, because the roll and strip are no longer in sufficiently close contact. For a higher reduction of 50%, agreement is still good for the long wavelength components, but less good for the shorter wavelengths. This difference is ascribed to difficulties in modelling what happens to the newly created surface and to the effect of sliding between the roll and strip. As with un lubricated rolling, inclusion of a short wavelength component reduces significantly the predicted true contact area between the roll and strip. This confirms the supposition that short wavelengths must be included to develop an accurate friction model for cold rolling.

Figure 4.14 Change in amplitude of short and long wavelength strip roughness components $\sigma$ and $\Sigma$, normalised by their initial amplitudes $\sigma_0$ and $\Sigma_0$, as a function of speed parameter $\Lambda_s$ (Le and Sutcliffe, 2001a)
Trials performed at an industrial aluminium foil rolling plant illustrate the practical importance of this conclusion, showing how the change in roughness spectrum due to roll wear is reflected in the change in foil roughness (Le and Sutcliffe, 2000b). The observed reduction in short wavelength components on the roll would significantly reduce the friction coefficient and hence strip reduction. To maintain the same friction level as the roll wears, the mill speed needs to fall to draw less oil into the bite.

Figure 4.15 shows the predictions of Montmitonnet et al (2001) for the change in mean friction coefficient $\mu$ and forward slip $G$ with roll speed. The friction curve takes on the typical Stibbeck form. At very low speed the contact area ratio approaches the limiting value of 1, and the mean friction coefficient is governed by the assumed boundary friction law on the contacts. As speed increases the amount of oil entrained into the bite rises and the area of contact ratio falls. This leads to a drop in friction coefficient. At very high speeds friction is dominated by shearing of the oil, explaining the small rise in friction at the end of the plot. At the highest speeds the value of forward slip become negative, indicating skidding which would lead to poor mill control.

![Figure 4.15 Predicted variation of average Coulomb friction coefficient $\mu$ and forward slip $G$ with roll speed (Montmitonnet et al, 2001)](image)

Tabary et al (1996) describe a series of experimental measurements of strip roughness and corresponding friction coefficient when cold rolling aluminium using smooth or rough rolls in the mixed lubrication regime. A comparison between the inferred friction coefficient from these trials and the predictions of the two-wavelength model of Le and Sutcliffe (2001a) is given in Figure 4.16. Three approaches are used to model friction. In the simplified 'semi-empirical' approach, the surface roughness is modelled using only a single wavelength of roughness, but the boundary friction coefficient is allowed to vary with smooth film thickness according to measurements on a strip drawing rig. The two-wavelength model tracks the change in roughness amplitude for two wavelengths of roughness, using a simple Coulomb friction
coefficient of $\mu_b = 0.09$ on the contact areas. Both theories give reasonable agreement with experiments, showing the transition from high to low friction coefficient as the speed and hence lubrication parameter $\Lambda_s$ increases. The semi-empirical model benefits from being simple to use, but requires some input from strip-drawing or similar experimental data. It is expected that this would be the most suitable model for commercial exploitation. The two-wavelength model, while being more difficult to implement, does not require input from empirical data, so that it serves as a valuable research tool to justify and calibrate the semi-empirical model. By comparison, the theoretical predictions shown on Figure 4.16 for a constant friction coefficient and a single wavelength are not able to predict the measured friction values satisfactorily. Montmitonnet et al (2001) compared predictions of their model with these trial data, and found that good agreement could be obtained by assuming that the boundary friction factor fell with increasing rolling speed. It was suggested that this fall was due to increasingly effective micro-hydrodynamic lubrication of the plateaux (see section 4.4.1).

Tabary et al (1996) found that the variation of friction with rolling speed was not significantly different when using rough rolls and smooth strip, or vice versa, Figure 4.17. This implies that the use of a composite roughness for the roll and strip roughnesses is appropriate, and that ploughing of the strip by asperities on the roll does not make a significant contribution to friction, for the conditions of these experiments with longitudinal roughness.
Li (1995) describes a series of rolling experiments on aluminium and low carbon steel in the mixed lubrication regime. They find that the inferred friction coefficient increases with increasing reduction for both materials and decreases with rolling speed. Friction correlates well with the lubrication parameter \( \Lambda \), allowing them to derive a simple empirical friction model. As with the film thickness model of Rasp and Häfele (1998), this model does not include the details of hydrodynamic lubrication and asperity flattening described in section 4.3.1 above.

**4.3.4 Other experimental results**

An abundance of early measurements of strip surface roughness and friction in cold rolling is reviewed by Schey (1970, 1983b) and Roberts (1978), although it can be hard to interpret these measurements without corresponding observations and measurement of surface finish. As expected, where conditions are in the mixed lubrication regime, frictional conditions fall with increasing rolling speed. The dependence of friction on strip reduction is not clear. Lenard (2000) presents typical results for a low carbon steel, showing a fall in friction coefficient with increasing rolling speed and reduction. By contrast, in common with the findings of Tabary et al (1996), he shows an increase in friction coefficient with increasing reduction for various aluminium alloys. It is not clear why these two metals should show such differing behaviour. Perhaps boundary friction and the role of transfer films are significantly different in these cases (see sections 4.5 and 4.6 below).
4.3.5 Foil and temper rolling

For thin foil or temper rolling, elastic deformation of the rolls needs to be included to predict the shape of the bite region (Fleck et al, 1992). Because the bite is very long compared with the thickness of the strip, the rolling load or reduction is particularly sensitive to friction. Table 4.2, section 4.3.1.5, summarises a number of tribological foil models. Coupled solutions including roll deformation are described by Sutcliffe and Montmitonnet (2001) and Le and Sutcliffe (2002a). Luo and Keife (1997), Keife and Jonsäter (1997) and Pawelski et al (1998) present foil models coupled with simplified tribological models, in which the change in contact ratio does not take into account the hydrodynamic pressure in the valleys or the effect of bulk straining. Domanti and Edwards (1996) perform dry calculations for temper and foil rolling to predict frictional and rolling parameters in the bite.

The predictions of Sutcliffe and Montmitonnet (2001) highlight the critical importance of friction in foil rolling. Figure 4.18 illustrates typical results for the variation of rolling load with rolling speed, at constant reduction. Note that the variation in load is plotted on logarithmic axes. As the speed falls, the area of contact ratio rises leading to a rise in friction and a very large increase in rolling load. Le and Sutcliffe compare model predictions with measurements from mill trials, in which aluminium foil was reduced in thickness from about 30 to 15 µm under typical industrial conditions, see Figure 4.19. Because of the considerable roll flattening in this pass, the predicted load is very sensitive to the details of both the friction and elastic roll deformation models, so that the reasonable agreement found confirms the accuracy of these models. The differences between predictions and measurement of forward slip are explained by variations in friction coefficient through the bite, perhaps arising from thermal or tribo-chemical effects. Good agreement between measured and predicted loads is achieved at the less demanding earlier passes where there is less roll deformation.

![Figure 4.18 Predicted variation of rolling load with speed for thin foil (Sutcliffe and Montmitonnet, 2001)](image-url)
4.3.6 Thermal effects

Although thermal effects are much less significant in cold than in hot metal rolling, nevertheless there are significant temperature gradients in the bite, with both deformation work and friction generating heat. From a tribological point of view, temperature rises are important because they can give a substantial reduction in viscosity and because they can affect the tribo-chemistry in the bite, particularly in the presence of additives. In the inlet the oil temperature will be close to that of the roll and strip at that location. However these surfaces, in turn, will tend to be close in temperature to that of the lubricant as it is applied in abundance as a coolant. Some viscous heating of the oil will occur in the inlet region. The magnitude of this effect can be estimated from the results of Wilson and Murch (1976). Significant rises in oil temperature experienced in the bite will not affect the amount of oil entrained, but will affect friction, via changes in viscosity, or possibly via changes in additive performance. Sa and Wilson (1994) present an appropriate model for full film lubrication. In the mixed lubrication regime changes in heat transfer coefficient at the interface will significantly affect the apportioning of heat between the strip and roll.

4.4 MICRO-PLASTO-HYDRODYNAMIC LUBRICATION (MPHL)

For concentrated contact of elastic solids, so-called micro-elasto-hydrodynamic lubrication (micro-EHL) theory has been developed to investigate the tribological conditions under asperity contacts. By analogy, micro-plasto-hydrodynamic lubrication (MPHL) addresses this issue for the case where the workpiece is deforming plastically.

The importance of the MPHL mechanism can be judged from the work of Mizuno and Okamoto (1982), who measured sliding friction between a steel tool and a copper workpiece. The material was first deformed using the plane strain compression test (see Figure 4.27) and then the top platen was slid relative to the workpiece.
Figure 4.20 Variation of friction stress $\tau_m$ with product of viscosity $\eta$ and sliding speed $V$ for a copper workpiece slid relative to a flat platen, after being compressed (Mizuno and Okamoto, 1982)

Although the material was not deforming during sliding, nevertheless the deformation prior to sliding ensured that it conformed closely to the tool and was at the point of yield. They observed that the measured friction stress could be correlated with the product $\eta V$ of the viscosity and the sliding speed, as shown in Figure 4.20. Below a value of $\eta V$ of about 100 Pa mm the friction stress is independent of sliding speed, suggesting that lubrication is due to shearing of adsorbed boundary molecules (see section 4.5). Above this value, the increase of friction with $\eta V$ indicates that hydrodynamic effects control friction. It was suggested that the hydrodynamic film was created by outflow of oil from the pits during the sliding process. By assuming that friction is all due to bulk shearing of the fluid under the contacts, a film thickness can be inferred (Sutcliffe 1989). For values of $\eta V$ equal to 100 and 1000 Pa mm the corresponding film thicknesses are 8 and 53 nm. Note that these inferred film thicknesses are larger than the length of a typical additive (Schey, 1970, gives a value of 3nm for calcium stearate), so that it seems reasonable to suppose that friction could be due to shearing of such a MPFL film.

Although micro-EHL modelling has become a relatively well-filled field of research, the same is not true of MPFL. Montmitonnet (2001) notes that the range of finite element techniques used for micro-EHL could usefully be brought to bear on MPFL, although complications arising from the need to model the effect of bulk plasticity make this task appear formidable.

4.4.1 Micro-plasto-hydrodynamic lubrication in the mixed lubrication regime

In section 4.3 on mixed lubrication, it was supposed that the interface between roll and strip could be divided into 'contact' areas and valleys. In practice, it may be possible to generate a very thin film even in the contact areas. Sutcliffe and Johnson
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(1990b) present a simple analysis for MPHL at the entry to the bite. They adopt a simple geometric model for an asperity which is at the point of making contact with the roll. Using a finite difference method they calculate the 2-dimensional pressure distribution and film thickness under the asperity contact.

A more sophisticated model, tracking the evolution of valley pressure and asperity geometry around the point where the asperities first make nominal contact, would be needed to provide more confidence in the above approach. It is also likely that non-Newtonian viscous behaviour of the lubricant (see section 4.3.1.4) will play a significant role, because of the high strain rates associated with the thin films (Sheu and Wilson, 1982).

4.4.2 Micro-plasto-hydrodynamic lubrication of pits

Micro-PHL theory has been much more successfully applied to lubrication from isolated pits, probably because of the simpler geometry in this case than in the mixed lubrication problem. Such pits are typically found for isotropic surfaces, such as generated from the shot-blast finish on stainless steel strip. Build-up of hydrostatic pressure in the pits as they are reduced in volume tends to prevent the pits being eliminated (Kudo, 1965). However, in the presence of sliding between the tool and strip, this oil can be drawn out of the pits due to hydrodynamic action. Lo (1994) identifies a 'percolation threshold', below which isolated pits will form. Various experimental studies have considered MPHL, frequently using artificial indents to help observe the phenomenon (Fudanoki, 1997, Kudo and Azushima, 1987, Kihara et al, 1992, Wang et al, 1997). An extension to this idea is provided by Sheu et al (1998, 1999), who propose a two-scale topography. The coarse scale (e.g. deep pits) allows transport of lubricant into the work zone, while the finer scale roughness delivers lubricant throughout the surface.

4.4.2.1 Measurement of pit geometry. Pitting on a cold-rolled strip surface tends to be irregular, with a range of pits of different sizes and shapes. Here 3-D profilometry is invaluable in gathering the data, but it is not obvious how to synthesise it. Ahmed and Sutcliffe (2000) and Sutcliffe and Georgiades (2001) show how the raw data can be used to extract statistics of the pit geometry and hence a 'characteristic' pit diameter, spacing, depth and slope. The method is illustrated for samples of industrially-rolled bright annealed stainless steel, which had a rough shot-blast finish prior to cold rolling. The pit identification method is illustrated in Figure 4.21, for a strip reduction of 43% relative to the initial shot-blast hot band. Figure 4.21(a) shows the raw data from the profilometer, while Figure 4.21(b) shows the individual pits which have been identified, along with the 'characteristic' pit diameter and spacing. The method is effective at identifying pits seen visually, while the characteristic values of pit diameter and spacing seem reasonable.
Figure 4.21 Example of pit identification on stainless steel strip; (a) Profilometry data. The greyscale gives the surface height, with darker colours being pits, (b) Pits identified in black. Values of a characteristic pit diameter \( d_c \) and spacing \( D_c \) for this sample are indicated on the figure (Sutcliffe and Georgiades, 2001)

Figure 4.22(a) shows the change though the pass schedule in total pit area and roughness amplitude with overall strip thickness reduction. As the strip is reduced in thickness, both the percentage pit area and the strip roughness fall, with a good correlation between the area and roughness. The change in characteristic pit diameter and spacing through the pass schedule is shown on Figure 4.22(b). The characteristic pit diameter falls from about 75 µm after the first pass to only a few microns at the end of the schedule, as the large features coming from the shot-blasting process are progressively flattened.

Figure 4.22 Variation of surface parameters with overall strip reduction during rolling of stainless steel (a) fractional pit area and \( R_q \) roughness, (b) pit diameter and spacing (Sutcliffe and Georgiades, 2001)
4.4.2.2 Modelling and comparison with experiments. The starting point for the analysis is the work of Kudo (1965), who shows how hydrostatic pressure can build up during bulk deformation. Azushima (2000) uses the finite element method (FEM) to analyse the generation of hydrostatic pressure in the pits. He takes into account oil compressibility but ignores oil leakage via the interface. Lo and Wilson (1999) and Lo and Horng (1999) derive a theoretical model using hydrodynamic theory, showing how pit crushing depends on pit geometry and hydrodynamic conditions. A schematic model of the problem is shown in Figure 4.23. The rise in oil pressure at the trailing edge of the pit can be found by hydrodynamics. The oil flow rate out of the pit is also determined by the asperity crushing rate, which in turn is governed by the difference between the valley and plateau pressures. The solution for the change in asperity geometry and corresponding oil film thickness at the trailing edge is given by coupling the hydrodynamic and mass flow equations for the oil with the mechanics model for deformation of asperities. This model was further developed and applied to the rolling and drawing processes by Sutcliffe et al (2001) and Le and Sutcliffe (2002b). Sheu et al (1999) developed a FEM model to investigate work-piece material extrusion into artificial cavities on a tool surface with trapped oil. The fluid cavity is modelled as a porous foam filled with oil that can permeate via a porous media in the interface between the tool and work-piece. This allows for the coupling of hydrostatic permeation and surface deformation.

![Figure 4.23 Schematic of MPHL mechanism drawing oil out of pits due to relative sliding between tool and workpiece (after Lo and Wilson, 1999)](image-url)
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Figure 4.24 shows Le and Sutcliffe's (2002b) comparison of theoretical predictions of pit area with measurements from industrial samples of cold rolled strip with an initial shot-blast surface. In general agreement is good; the difference between results would probably be closed up by considering the statistical variation in pit geometries.

![Figure 4.24 Comparison of predicted and measured variation in pit area with bulk strain for rolling of stainless steel strip (Le and Sutcliffe, 2002b)](image_url)

The effect of oil entrainment at the inlet can be estimated by the lubrication parameter $\Lambda_i$, here denoted as $\Lambda_i$. Ahmed and Sutcliffe (2001) suggest a corresponding parameter $\Lambda_m$ to characterise the MPHL lubrication mechanism, as the ratio of the initial pit volume to an estimate of the volume drawn out during sliding. Regime maps have been constructed both from theoretical simulations and from experimental measurements of surface roughness, by plotting the change through the pass schedule in the lubrication parameters $\Lambda_i$ and $\Lambda_m$. Figure 4.25 illustrates such a map for a theoretical simulation of a multi-pass schedule, with each pass having a strip reduction of 15% (Le and Sutcliffe, 2002b). Initially the inlet parameter $\Lambda_i$ is very much less than one, indicating that there is negligible oil film entrained at the inlet. The value of $\Lambda_m$ is also less than one, showing that it is relatively difficult to draw oil out of the pit in the initial pass. Further down the schedule the inlet and MPHL parameters rise, as the pit depth reduces. The increase in $\Lambda_m$ above one indicates that the oil is very rapidly drawn out by the MPHL mechanism, so that this is not a limiting factor. Instead conditions can be expected to be similar to that for dry rolling. Towards the final pass, particularly for the shallowest pits and at the higher rolling speed, the inlet parameter $\Lambda_i$ approaches one, indicating that there is now a significant oil film entrained at the inlet, which tends to inhibit the elimination of the pits. Corresponding regime maps derived from experiments are shown in Figure 4.26 (Sutcliffe and Georgiades, 2001). For each pit on a given sample, values of the lubrication parameters $\Lambda_m$ and $\Lambda_i$ are calculated and
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plotted on the figure. Deeper pits are towards the bottom left of each line. These results are in good qualitative agreement with the theoretical predictions.

An alternative mechanism for oil flow from pits has been described by Bech et al (1999) and Shimizu et al (2001), in which the gradients of hydrostatic pressure can lead to oil flow. This micro-hydrostatic lubrication mechanism is more relevant to very low speed rolling; at high speeds hydrodynamic lubrication dominates. Further details of these MPHL mechanisms are described in Chapter 7.

![Theoretical regime map showing the predicted change in the lubrication parameters $\Lambda_m$ and $\Lambda_i$ for MPHL and inlet lubricant entrainment of pits. The trajectories show the change in lubrication regime during a typical pass schedule associated with the progressive reduction in pit depth (Le and Sutcliffe, 2002b)](image1)

**Figure 4.25** Theoretical regime map showing the predicted change in the lubrication parameters $\Lambda_m$ and $\Lambda_i$ for MPHL and inlet lubricant entrainment of pits. The trajectories show the change in lubrication regime during a typical pass schedule associated with the progressive reduction in pit depth (Le and Sutcliffe, 2002b)

![Experimental regime map showing the change in lubrication parameters $\Lambda_m$ and $\Lambda_i$ for MPHL and inlet lubricant entrainment of pits on strip samples (Sutcliffe and Georgiades, 2001)](image2)

**Figure 4.26** Experimental regime map showing the change in lubrication parameters $\Lambda_m$ and $\Lambda_i$ for MPHL and inlet lubricant entrainment of pits on strip samples (Sutcliffe and Georgiades, 2001)
4.5 BOUNDARY LUBRICATION

The original definition of boundary lubrication was for 'lubrication in which friction depends not only on the properties of the lubricant but also on the chemical nature of the solid boundaries' (Hardy and Doubleday, 1922). However, as Ike (1999) notes, the distinction between hydrodynamic and boundary lubrication has become increasingly blurred as both measurement and modelling have reached down to molecular dimensions. Interferometry measurements on oil films which are only a few molecules thick show quasi-hydrodynamic behaviour; atomic force microscopy (AFM) measurements show solidification of liquid near a crystalline solid and models of lubricant flow are increasingly concerned with simulation at a molecular level (see reviews by Spikes, 2000 and Bushan, 1995). On a practical level, it is in any case very difficult to distinguish between MPHL and boundary lubrication. For elastic contacts, this problem can be avoided by using atomically smooth surfaces (e.g. surface force apparatus or AFM, see Bushan, 1995), but during metal-forming processes new surface is continually being created, so that this approach is not feasible. Probably, as suggested by Montmitonnet et al (2001), the situation in the contact areas in much metal forming is a mixture of MPHL and boundary lubrication, with perhaps some unlubricated regions as well. Some authors refer to this as the thin film regime. In this section we follow the original definition of boundary lubrication as depending on the physico-chemical interactions at the interface, although it is unlikely that the experimental results discussed below are free from MPHL influences.

An estimate of the film thickness under these 'thin film' conditions is made by Ike (1999), based on the roughness, measured using AFM, of surfaces deformed by a very smooth gauge block. He postulated that the measured roughness of the strip after deformation was equal to the film thickness separating the surfaces. Results suggest that the film thickness is of the order of 1 nm, comparable to several molecular widths.

Although boundary friction is critical in cold rolling, there is rather scant information on how to model it. The great bulk of published research on boundary lubrication, both experimental and theoretical, considers the more common case where there is little if any deformation of the surfaces in contact and cannot be applied directly to modelling of boundary lubrication in metal working. Early work on boundary lubrication concentrated on the role of adsorbed molecules (Rowe, 1966). However it is clear that reactive molecules will absorb chemically on the surface, particularly where, as in metal rolling, there is considerable fresh metal surface formed in the bite. Wakabayashi et al (1995) illustrated this for gas-phase lubrication of orthogonal machining, where they showed that frictional behaviour can only be understood in terms of a chemisorption of the lubricant at the tool-workpiece interface. Mori et al (1982) present a study of chemisorption by organic compounds on a clean aluminium surface prepared by cutting. They find that the rate of absorption was proportional to the cutting speed, with absorption taking place both during and after cutting. It was suggested that the absorption rate was related to the formation rate of new aluminium surface. Equally
it might be expected that the oxide film behaviour will be important (Milner and Rowe, 1962, Li et al, 1999).

Other work on boundary lubrication in metal forming highlights the role of additives. Chambat et al (1987) showed that organo-metallic compounds were produced during industrial foil rolling of aluminium. Increasing the concentration of an additive package from 0.1 to 4% was found to bring about a significant reduction in friction factor during plane strain compression tests on steel (Kubié, 1980). Oleic acid at a concentration greater than 1% was effective for aluminium and stearyl amine was effective at much lower concentrations (Kubié, 1980, Johnston and Atkinson, 1976). Stearic acid at 1% in mineral oil was effective at preventing metal transfer in twist compression tests using aluminium while the shorter chain lauric acid was ineffective, as was the unsaturated oleic acid (Nautiyal and Schey, 1990). In these tests a dark non-metallic appearance to the tool developed with the stearic acid, through which details of the anvil could be observed in the SEM. It was suggested that this film was due to reaction products (see section 4.6 below). Increasing the chain length of saturated acids at a 1% concentration in hexadecane increased the effectiveness of boundary lubrication for ball-on-cylinder tests (both components being steel), with a loss of effectiveness for the stearic acid at about 130°C (Jahanmir, 1985). A transition temperature was observed in these tests at low speed, while none was observed at higher speeds for the 16 and 18 carbon atom additives. From this evidence it was suggested that adsorbed monolayers controlled friction at low speeds, while at higher speeds partial hydrodynamic lubrication occurred, with the influence of the additives arising from the presence of ordered monolayers. Matching of the additive and base oil chain lengths was not found to influence the friction coefficient. In the twist-compression and ball-on-cylinder tests there was no bulk plastic deformation. However, the high local pressures at the asperity tops will lead to deformation there, particular for the twist-compression tests with aluminium. It is not clear how results without bulk deformation can be used for metal forming processes. This will depend on the mechanisms controlling boundary friction, and hence the importance of the bare metal surface created during metal forming.

Finally, leaving aside for a moment the question of the frictional stress at the interface, Wilson (1991) presents a series of models for boundary lubrication where roughness interactions between the roll and strip play a role (for example generating a ploughing term). The contact ratio depends on a balance between the flattening mechanism discussed in section 4.2 and opening up of gaps between the roll and strip as asperities slide past one another, as illustrated in Figure 4.9.

4.6 TRANSFER LAYERS

Montmitonnet et al (2000) have recently highlighted the role of transfer films in metal rolling. Transfer films have been observed in the laboratory both in rolling and plane strain compression testing of a variety of metals. The formation of these films provides the key to understanding the evolution of friction during both types of tests.
Moreover similar films are seen in industrial practice; rolls used for aluminium develop a blueish sheen, while a brown film adheres to rolls used for stainless steel. The most extensive series of tests have been undertaken by Kubié (1980) (see Delamare et al, 1982), using the plane strain compression test shown schematically in Figure 4.27. Flat strip is compressed between flat parallel platens. For appropriate strip and platen geometries, the deformation mode is similar to that pertaining in cold metal rolling, justifying the use of such a test to model rolling, although tribological conditions may differ.

In Kubié's studies, a series of indentations was made on the strip (using a fresh part of the strip for each indentation). Figure 4.28 shows a typical variation in friction stress with indentation number on stainless steel. As a 'low friction slurry' transfer layer builds up on the initially clean tool, the friction stress drops significantly. After 12 indentations the tool was cleaned with an absorbent paper, removing the transfer layer and restoring the frictional stress to its original value. AES studies showed this film to be composed of strip metal (plus additives and lubricant, as deduced from its slurry-like behaviour). The slurry tended to reduce the roughness of the tool by filling in the valleys. For Zircaloy alloys a much more solid transfer layer was formed on the tool, giving rise to a significant rise in friction factor with indentation number.
Recent work by Dauchot et al (1997, 1999, 2001) compared rolling and plane strain compression of both aluminium and steel, and related results to ToF-SIMS analysis (Time-of-Flight Secondary Ion Mass Spectroscopy). Rolls used to deform aluminium alloy show an accumulation of a relatively thick transfer film, with thicker films formed at the edges of the strip where lubrication is poor, suggesting that the formation of the transfer film is a function both of the lubrication conditions and the chemistry at the interface.

Sutcliffe et al (2002) describe a detailed series of plane strain compression tests on aluminium strip to investigate the tribology in the thin film regime. Both a formulated lubricant containing ester as additive and a lubricant comprised of hexadecane plus stearic acid additive in concentrations of 0, 0.05 and 0.5% were used. The friction factor $m$ was estimated for a variety of process conditions, and correlated with surface analysis techniques, including SEM, profilometry and ToF-SIMS. The strip surface conformed very closely to the tool except at the middle of the indent, suggesting that only a very thin oil film, if any, could separate the tool and strip surfaces. Figure 4.29 shows typical results for the variation in friction factor with indentation number, using hexadecane plus stearic acid as lubricant. For the first few indents the friction factor is close to one, implying shearing of the strip metal surface. Results are independent of additive concentration. After a few indents a transfer layer built up on the tool, leading to a significant fall in friction factor. This transfer layer was of the ‘low friction slurry’ type, and could be easily removed with an absorbent paper. The reduction in friction associated with this transfer layer increased with increasing additive concentration and more effective hydrodynamic or hydrostatic lubrication (as evidenced by effects of speed, roughness orientation and oil starvation). It seems that, in this case, the role of the additive lies not in protecting the surfaces in the first instance, but in generating a better transfer layer.

![Figure 4.29 Variation of friction factor with indentation number for plane strain compression of aluminium strip using hexadecane and stearic acid as lubricant (Sutcliffe et al, 2002)](image)
Figure 4.30 shows an SEM image of a strip after 9 indents. An imprint of the tool surface roughness is clearly seen running horizontally on the strip, indicating thin film conditions. It appears that dark areas seen on the strip correlate with regions where the tool roughness is not imprinted on the strip. Presumably both the dark areas and the change in strip topography are due to a transfer layer forming on the tool. ToF-SIMS analysis revealed the presence of stearate and aluminium stearate ions on both the tool and strip, spreading from the edge of the indent towards the middle with increasing indentation number. The presence of aluminium stearate gave evidence of adsorption and reaction of the stearic acid with the bare aluminium surface. Aluminium stearate was not found due to mere contact of the aluminium with stearic acid, in the absence of bulk deformation.

4.7 CONCLUSIONS

The results described in this chapter show how the mechanics of lubrication in cold rolling in the industrially-relevant mixed lubrication regime is now reasonably well understood. Section 4.2 identifies the importance of bulk deformation of the strip in facilitating asperity crushing. Hydrodynamic pressure is induced in the lubricant due to the entraining action of the roll and strip. Modifications to Reynolds' equation to include the effect of roughness are described. The interaction between the deformation of asperities and the hydrodynamic pressure has been effectively modelled and predictions of film thickness and surface roughness are in good agreements with measurements. Details of the roughness topography are seen to be important; in particular the importance of short wavelengths of roughness is highlighted.

Frictional models have also been successfully developed for the mixed lubrication regime, both for strip and foil rolling. However the mechanisms of friction on the contact areas are still not clear. Although traditionally this has been supposed to
be due to boundary lubrication, empirical models suggest some hydrodynamic action, probably due to a micro-plasto-hydrodynamic lubrication (MPHL) oil film. Models for such an MPHL film generated in the inlet are not well established.

However useful models of MPHL have been developed for the alternative scenario, where oil trapped in isolated pits in the inlet is drawn out in the bite due to sliding between the roll and strip. Measurements and analysis of the pit geometry using 3D profilometry are described, to show how the pits change during a roll schedule.

The chapter finishes with two sections on boundary lubrication and transfer films. Our knowledge of this area is largely empirical, with details expected to depend on the metal and lubricant chemistry. However the application of modern surface analysis techniques, in conjunction with further modelling, shows promise in developing our understanding of these phenomena. It is likely that appropriate models will need to include the asperity mechanics and MPHL models described above, along with the tribo-chemistry of the contact.

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