Modelling of friction for high temperature extrusion of aluminium alloys

Liliang Wang
Modelling of friction for high temperature extrusion of aluminium alloys

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Liliang WANG

Master of Engineering
Harbin Institute of Technology, China
geboren te Liaoning, China
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1.1 BACKGROUND

Extrusion is a process in which a cast billet of solid metal is converted into a continuous length of generally uniform cross-section by forcing it to flow through a shaped die opening. Generally, the extrusion process is a hot working operation, in which the metal billet is heated to a proper temperature, at which a relatively high ductility and low flow stress can be achieved. Figure 1.1 shows the principle of direct extrusion. The extrusion die is located at one end of the container, and the billet to be extruded is pushed towards the die.

![Schematic working principle of direct extrusion process.](image)

Hot extrusion is widely used for the manufacturing of near-net-shape solid and hollow sections [1-5]. In recent years, the increasing demands of such profiles in automobile and aircraft industries have led to a demand for a better understanding of the process. On the other hand, hot aluminium extrusion involves complex thermo-mechanical and chemical interactions between hot aluminium and tool-steel tooling [5-7] (mainly extrusion die and
container) and the local contact conditions at the work piece/tooling interfaces are of great influence on process parameters, such as productivity, product quality and scrap rate. In recent years, finite-element (FE) simulations have been extensively used in scientific research and industrial practice to analyse the process and to aid in process optimization. A basic issue of FEM simulations is the accuracy of the results, which is mainly determined by the viscoplastic material behavior of aluminium alloys at elevated temperatures (temperature and strain rate sensitive); and the assignment of boundary conditions, especially the friction boundary condition [8-20]. However there remain some uncertainties in the selection of friction models and the determination of friction coefficients, because the friction phenomenon, especially the friction at elevated temperatures, is not fully understood yet.

1.2 DETERMINATION OF FRICTION COEFFICIENTS

In the past years, some efforts have been made to study the tribological phenomenon of the extrusion process and the experiments conducted can be classified as three different types, namely, field tests, e.g. extrusion friction tests [12, 13, 15, 16, 18, 21, 22]; physical simulation tests, e.g. block on disc tests [6, 23, 24]; and tribological tests, e.g. ball-on-disc tests [25-29]. The three types of tribological tests were not compared yet and this is the subject within this research. Figure 1.2 summarizes the friction characterization techniques for the extrusion processes.

Friction at billet/container interface:
- Extrusion test + FE simulation
- Forward extrusion with different billet lengths + Theory
- Billet with rod markers embedded

Friction in the bearing channel of extrusion dies:
- Extrusion tests: sticking and slipping lengths on the bearing surface
- Block on disc test
- Ball/Pin on disc test

Figure 1.2 Summary of the friction characterization techniques for extrusion processes.
1.3 THESIS LAYOUT

This thesis deals with the assignment of friction boundary conditions for hot aluminium extrusion process. Issues addressed are the high-temperature ball-on-disc tests, friction modelling, double action extrusions and computer simulations of the hot aluminium extrusion process. The layout of the thesis is illustrated in Figure 1.3.

In Chapter 1, the background of the present research is introduced.

In Chapter 2, the basic theories of friction are introduced and the techniques for the friction characterization of extrusion processes are reviewed. In addition, the commonly used friction models for extrusion processes are reviewed.

In Chapter 3, a model for high-temperature ball-on-disc test is developed. The individual contributions of shearing and ploughing friction are studied, and the evolution of wear track or mean contact pressure during the ball-on-disc tests is characterized.

In Chapter 4, the friction stress between hot aluminium and H11 tool steel is determined by using short sliding distance ball-on-disc tests. Based on the testing results, a physically based friction model for the bearing channel of hot aluminium extrusion die is developed.

In Chapter 5, a novel extrusion process, double action extrusion (DAE), is developed to highlight the friction in the bearing channel of aluminium extrusion dies. Both theoretical and FE modelling of this novel process are conducted and the working mechanism of the DAE is analysed. In addition, the adhesive strength friction model (developed in Chapter 3) is implemented into the FE simulation of hot aluminium extrusion process and this model is experimentally verified.

In Chapter 6, the most important conclusions of this thesis are summarized. The friction testing techniques for extrusion processes and the nature of friction in the bearing channel of hot aluminium extrusion process are discussed. Finally, recommendations for further research are proposed.
Chapter 1: Introduction

Chapter 2: Literature review

Chapter 3: Modelling of high temperature ball-on-disc tests

Chapter 4: Determination of friction coefficient for the bearing channel of the hot aluminium extrusion die

Chapter 5: Double action extrusion - a novel extrusion process for the friction characterization at the billet-die bearing interface and friction model verification

Chapter 6: Conclusions, discussions and future recommendations

Summary

Figure 1.3 Layout of the thesis.
References


Chapter 2
LITERATURE REVIEW

2.1 THE ORIGINS OF FRICTION

“Friction is the resistance of motion during sliding or rolling that is experienced when one solid body moves tangentially over another with which it is in contact. [1]” Friction is a highly complicated phenomenon, which can be attributed to many mechanisms, such as formation and break-down of asperity junctions [2], ploughing of hard asperities over the softer surface [1], entrapment of hard wear particles [3], adhesive force due to chemical reaction or inter-atomic diffusion [4] etc. In this section, the development of the theories on the origins of friction is briefly reviewed.

2.1.1 The classic friction laws

In ancient times, our ancestors started to think about reducing friction by using wheels or lubricants, e.g. the earliest record of using wheels was from 3500 BC and the earliest record of using lubricant was in 1880 BC approximately, by Egyptians [1]. However, the detailed scientific understanding was not setup then. The pioneer work in the field of friction was conducted by Leonardo da Vinci (1452-1519), who for the first time proposed the concept of friction and deduced the rules of friction. However, da Vinci’s work was not published for hundreds of years, until 1699. A French physicist, Guillaume Amontons re-discovered the rules of friction and proposed the two well-known friction laws:

The 1st friction law: Friction force is proportional to the normal force between the surfaces in contact;

\[ f = \mu N \]

where \( f \) is friction force, \( \mu \) is friction coefficient and \( N \) is normal load.

The 2nd friction law: Friction force is independent of the apparent contact area;
Many years later, in 1781, a French physicist, Charles-Augustin Coulomb summarized da Vinci and Amontons’s work and contributed the 3rd friction law, namely, the kinetic friction force is independent of the sliding velocity, and Coulomb clearly separated the concepts of static and kinetic friction for the first time [2].

2.1.2 The origins of friction – a brief review of the theories of friction

2.1.2.1 Interlocking of the surface asperities

It was realized hundreds of years ago that surfaces are not perfectly flat and characterized by micro-hills and valleys. When two surfaces are placed together, the upper surface is supported on the hills or summits of the lower surface, as shown in Figure 2.1 a and b. These hills or summits are called asperities. Since the two mating surfaces are only supported by asperities, the contact area (real area of contact) is much smaller than the apparent contact area. According to Coulomb’s theory, the friction was due to the interlocking of the surface asperities and riding of rigid asperities of one surface over the other, as shown in Figure 2.1 (c). Therefore, if the average asperity angle is \( \alpha \), the friction coefficient is approximately \( \tan \alpha \) and is independent of normal load or apparent contact area, which explains the Amonton’s friction laws.

![Figure 2.1 Asperities contact between mating surfaces.](image)

2.1.2.2 The adhesion-ploughing theory
Bowden and Tabor proposed an adhesion-ploughing friction theory, which is the most widely accepted theory in recent decades [5]. According to Bowden and Tabor, due to the intense contact pressure on the asperities, localized adhesion and welding of metal surfaces occurs, when a surface is sliding over the other one, work is required to shear or separate these welding junctions, meanwhile, ploughing of the softer metal occurs [2]. Therefore the friction force can be expressed as the sum of two terms: the adhesive or shearing term \( \left( f_s \right) \) and ploughing term \( \left( f_p \right) \).

The shearing term \( \left( f_s \right) \)

As discussed in the previous section, when two surfaces are placed together, the real contact area is much smaller than the apparent contact area. In other words, on the mating surfaces, only asperities contact occurs, i.e. the mating surfaces are supported by a number of asperities. If the normal load applied is \( N \), yielding pressure of soft material is \( p \), then the real contact area can be expressed as:

\[
A_r = \frac{N}{p} \quad (2.2)
\]

Assuming the mean shear strength of welding junction is \( \tau \), then the force required to move the asperities in the direction of parallel to the contact surfaces, i.e. the shearing friction force \( f_s \) is:

\[
f_s = A_r \tau \quad (2.3)
\]

Substitute Equation (2.2) into Equation (2.3):

\[
f_s = \frac{N\tau}{p} \quad (2.4)
\]

and

\[
\mu_s = \frac{\tau}{p} \quad (2.5)
\]

According to Equations 2.2, 2.3 and 2.4, the real contact area increases with increasing normal load, consequently, the shear friction force is independent of apparent contact area, which meets Amontons’s 1st friction law. In addition, as can be seen from Equation 2.5, the shear friction coefficient is determined by mean shear strength of the welding junctions and yielding strength of softer material. Therefore it is independent of normal load, which meets
Amontons’s 2\textsuperscript{nd} friction law. According to the data provided in [2], the mean shearing strengths of the welding junctions are slightly higher than the shear strengths of pure metals. Therefore, the maximum shearing friction coefficients should be about 0.5-0.6, assuming that the shearing strength of a metal is typically half of its yielding strength. Nevertheless, it is difficult to explain some experimental results, in which friction coefficients greater than 1 were observed. In fact, in most cases with plastic contact, particularly in the case of ductile metal contact, the ploughing term of friction plays an important role.

The ploughing term \( (f_p) \)

When a hard asperity slides over a soft surface, the asperity indents into the soft surface to take the normal load and in the meanwhile ploughing force is required to remove the soft material in front of the asperity. Bowden et. al. was among the first to attempt to model the ploughing term of friction [6]. Many researchers have tried to model the ploughing effect of asperities with different simplified tip shapes, such as cones, spheres and pyramids [7, 8]. Taking sphere shape asperity as an example:

![Figure 2.2 The indenting area of a sphere tip asperity.](image)

Figure 2.2 schematically shows the contact between the sphere tip asperity and the soft material, with their geometric relationships indicated. The tangential force and the normal force acting on an elemental area \(dA\) are given as:
\[
\begin{align*}
\{dF_x &= p \sin \beta \cos \gamma dA + f \sin \gamma dA \quad (2.6a) \\
\{dF_z &= p \cos \beta dA \quad (2.6b)
\end{align*}
\]

where \(dA = r^2 \sin \beta d\beta d\gamma\), \(0 \leq \beta \leq \xi\), \(0 \leq \gamma \leq \pi\), \(\beta\) and \(\gamma\) are integration angle as shown in Figure 2.2. \(\xi\) is the upper integral limit of the angle \(\beta\). \(p\) and \(f\) are the normal pressure and friction stress, respectively. The overall friction coefficient is designated as \(\mu\), while the shear friction coefficient acting on the contact interface as \(\mu_s\). Then, the shear friction stress can be expressed as: \(f = \mu_s p\).

Integrating Equations 2.6a and 2.6b leads to:

\[
\begin{align*}
\{F_x &= pr^2 (\xi - \sin \xi \cos \xi) + \mu_s p 2r^2 (1 - \cos \xi) \quad (2.7a) \\
F_z &= \frac{\pi pr^2}{2} \sin^2 \xi \quad (2.7b)
\end{align*}
\]

It can be seen from Equation 2.7a that the tangential force \(F_x\) is composed of two terms: the first term concerns the ploughing friction that results from the deformation of the soft material in front of the asperity; the second term is the shear friction stress component where plastic deformation is absent.

If the normal load applied on this asperity is \(L\),

Then, \(F_z = L\) and \(p = \frac{2L}{\pi r^2 \sin^2 \xi}\)

With the normal pressure \(p\) inserted into \(F_x\), Equation 2.7a can be reorganized with the overall friction coefficient \(\mu\) expressed as:

\[
\mu = \frac{F_x}{L} = \frac{2(\xi - \sin \xi \cos \xi)}{\pi \sin^2 \xi} + \frac{4(1 - \cos \xi)}{\pi \sin^2 \xi} \mu_s
\]

(2.8)

The geometric relationship in the indenting area shown in Figure 2.2 may be expresses as:
\[
\sin \xi = \frac{w}{2r}, \cos \xi = \left[1 - \left(\frac{w}{2r}\right)^2\right]^{\frac{1}{2}}, \xi = \sin^{-1}\left(\frac{w}{2r}\right)
\]

where \(w\) is the width of the indentation and \(r\) represents the radius of the asperity.

\(\mu\) can then be expressed as a function of the width of the indentation:

\[
\mu = \frac{2\left(\sin^{-1}\left(\frac{w}{2r}\right) - \frac{w}{2r}\left[1 - \left(\frac{w}{2r}\right)^2\right]^{\frac{1}{2}}\right)}{\pi\left(\frac{w}{2r}\right)^2} + \frac{4\left(1 - \left(\frac{w}{2r}\right)^2\right)^{\frac{1}{2}}}{\pi\left(\frac{w}{2r}\right)^2} \mu_s \quad (2.9)
\]

In Figure 2.3, the overall friction coefficients are plotted against the ratio of the width of the indentation to the diameter of the asperity at different shear friction coefficients. It becomes clear that the overall friction coefficient increases markedly with the increase in the width of the indentation \(w\) (related to the extent of deformation) and the shear friction coefficient \(\mu_s\).

When the deformation is severe, resulting in ploughing, the overall friction coefficient will be greater than the friction coefficient resulting from the shear friction alone. Therefore, the ploughing term could contribute significantly to the overall friction force, which may explain the high friction values observed in some of the experimental data.

![Figure 2.3 Variation of overall friction coefficient with the width of the indentation.](image-url)
2.2 FRICTION CHARACTERIZATION TECHNIQUES FOR EXTRUSION PROCESSES

Friction in extrusion processes has drawn much attention in recent decades due to the tremendous development of the FE analysis of extrusion processes. This is because the accuracy of simulation results is highly dependent on the sensible assignment of friction coefficient as boundary conditions. In the past years, much research work has been conducted to determine the friction coefficient between work piece and toolings and to develop friction models for extrusion processes. Some of the previous findings will be reviewed in this section.

Friction in the extrusion process is a complex phenomenon, because the mutual sliding between work piece and tooling takes place under high contact pressures, which could be a few times greater than the flow stress of the work piece material, and sometimes severe surface enlargement and temperature effects are involved [9]. Consequently, sensible selection of friction testing techniques is of great importance in order to obtain reliable friction coefficients or factors for extrusion processes.

2.2.1 Ring compression test

One of the most widely used friction testing techniques used in bulk metal forming processes is the ring compression test, which was first introduced by Kunogi in 1956 [10], and developed by Male and Cockcroft in 1963 [11], making it an effective and efficient way of characterizing friction and evaluating lubricants for metal forming processes. In ring compression tests, the inner diameter of the ring may increase, decrease or remain constant, depending on the magnitude of friction at the tool / work piece interfaces. For instance, under extremely low friction conditions, or when the friction between the work piece and tool is lower than a critical value, the material flows outwards, and both inner and outer diameters of the ring increase. If the friction at the contact interfaces is higher than a critical value, the material close to the inner diameter flows inwards, which decreases the inner diameter of the ring, and the remainder material flows outwards, which enlarges the outer diameter of the ring (as shown in Figure 2.4).

Since the size of inner diameter is highly sensitive to the friction at contact interfaces between the work piece and dies, under various friction conditions, the reduction in the size of inner diameter as a function of the amount of compression in height can be summarized as friction
calibration curves (FCCs), given in Figure 2.5 [11], which can be used to identify friction coefficient quantitatively.

Figure 2.4 Typical shapes of inner and outer surfaces that are normally observed after a ring compression test: ring compression test results under a. low friction condition and b. high friction condition [12].

Figure 2.5 Typical calibration curves for ring compression tests: the decrease in inner diameter of a ring vs. the reduction in height [11].
Table 2.1 Examples of applications of ring compression tests.

<table>
<thead>
<tr>
<th>Ring compression tests</th>
<th>Work piece material</th>
<th>Testing temp. (°C)</th>
<th>Lubricant(s)</th>
<th>Friction coef./factor</th>
<th>Applications</th>
</tr>
</thead>
<tbody>
<tr>
<td>1978 Tabata et al.[13]</td>
<td>sintered copper powder metals</td>
<td>25</td>
<td>11 Lubs.</td>
<td>0.02-0.1</td>
<td>Lubs. evaluation</td>
</tr>
<tr>
<td>1998 Petersen et al. [14]</td>
<td>CP Al</td>
<td>25</td>
<td>MoS₂</td>
<td>m=0.105-0.125</td>
<td>Alternative shaped rings</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>kerosene</td>
<td>m=0.25-0.275</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>No lub.</td>
<td>m=0.375-0.85</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>MoS₂</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>2000 Hu et al. [16]</td>
<td>CP Al</td>
<td>25</td>
<td>Shell Tellus 23 Oil</td>
<td>μ=0.01-0.08</td>
<td>Metal forming</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1963 Male et al. [11]</td>
<td>Aluminium</td>
<td>-200-600</td>
<td>No lub.</td>
<td>μ≈0.15-0.57</td>
<td>Industrial metal-working processes</td>
</tr>
<tr>
<td></td>
<td>Copper</td>
<td>-200-1000</td>
<td>No lub</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>α-Brass</td>
<td>-200-800</td>
<td>No-lub</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Mild steel</td>
<td>-200-1000</td>
<td>No-lub</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>CP Titanium</td>
<td>0-1000</td>
<td>Graphite</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1989 Pawelski et al.[17]</td>
<td>C45, X40CrMoV5, X210Cr12</td>
<td>990-1160</td>
<td>Graphite + ester</td>
<td>m=0.12-0.8</td>
<td>Hot rolling</td>
</tr>
<tr>
<td></td>
<td></td>
<td>990-1160</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1990 Sadeghi et al. [18]</td>
<td>Forging steel</td>
<td>700-1200</td>
<td>Graphite</td>
<td>m=0.1-0.6</td>
<td>Hot forging</td>
</tr>
<tr>
<td>1992 Shen et al. [19]</td>
<td>Al-Li alloy</td>
<td>357</td>
<td>Lub A: MoS₂</td>
<td>m=0.2</td>
<td>Lubs. evaluation for hot forging</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Lub B</td>
<td>m=0.1-0.2</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Lub C</td>
<td>m=0.05</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Lub D: Oil</td>
<td>m=0.07</td>
<td></td>
</tr>
<tr>
<td>1996 Rudkins et al. [20]</td>
<td>Medium carbon steel and cutting steel</td>
<td>800-1000</td>
<td>No lub.</td>
<td>m=0.75-0.9</td>
<td>Hot metal forming</td>
</tr>
<tr>
<td>2005 Cho et al. [21]</td>
<td>6061-T6</td>
<td>200&amp;400</td>
<td>No lub.</td>
<td>m ≈ 0.6</td>
<td>Warm forming</td>
</tr>
<tr>
<td>2006 Sagar et al. [22]</td>
<td>CP Al, Al-Zn alloy</td>
<td>30-500</td>
<td>No lub.</td>
<td>m=0.3-0.9</td>
<td>Metal forming</td>
</tr>
</tbody>
</table>
In the past years, ring compression tests have been used by many researchers to evaluate lubricants or to determine friction coefficients. Some previous applications of the ring compression tests have been summarized in Table 2.1.

2.2.1.1 Materials effects on friction coefficient

The influence of testing material on friction has been studied by using ring compression tests in some previous work. Pawelski et al. investigated the effects of work piece material on the friction factor under both lubricated and unlubricated conditions. It was found that, under unlubricated conditions, friction factor ranged from 0.8 to 0.9 and was insensitive to work piece materials [17]. Similar results were observed in Rudkins et al.’s research, and the friction factors did not vary with work piece materials [20]. In contrary, Sagar et al. found that the alloy composition affected friction significantly [22] and similarly, Sofuoglu et al. suggested that the use of a generalized friction calibration curves without considering material types would lead to pronounced error for testing results [12].

Friction is not a material property [1], thus it is not determined by testing materials. However, the material properties may affect friction, particularly when clean metal and alloy surfaces contact each other, and strong inter-atomic bonds are formed at the contact interface. As explained by Rabinowicz [23], the interaction of mating materials depends on the mutual solubility of them and varies significantly with different material combinations. For the material pairs with a solid solution less than 0.1% solubility at liquid phases, they tend to produce low adhesion, thus low friction. The contact of two materials with over 1% solubility at liquid phases generally leads to higher adhesion. Friction is highly dependent on the mechanical properties of testing materials. Soft and ductile metals tend to produce higher friction. For instance, when a metal is in contact with Pb and Sn, the real contact area tends to be high even at low normal pressures, thus a high friction coefficient can be observed. The oxide film of testing materials can influence friction, i.e. the metals which tend to form a tough oxide film under ambient condition usually produce low friction. For instance, the oxide film on the surface of Chromium is responsible for the low friction. Therefore, when ring compression tests were conducted with different material combinations, the friction test results could be different, and different friction calibration curves should be used for different testing materials. However, the friction test results are not only affected by materials, but also
testing conditions, such as contact pressure, temperature and sliding velocity. This is because
the friction is a system response rather than a material property [1].

2.2.1.2 Contact pressure effects on friction coefficient

It is rather difficult to study the influence of contact pressure on the friction by using standard
ring compression tests, because the normal pressure at the contact interface is always greater
than the flow stress of the work piece material [14]. Therefore, alternative ring geometries (as
shown in Figure 2.6) were developed to achieve different contact pressures [14, 15], namely,
concave-shaped ring for low contact pressure (Figure 2.6 a), rectangular-shaped ring for
medium contact pressure (Figure 2.6 b) and convex-shaped ring for high contact pressure
(Figure 2.6 c). It is shown in Tan et al.’s work [15], different normal pressures were obtained
by using rings with different geometries. Due to the contact pressure difference, the concave-
shaped rings resulted in the lowest friction, the rectangular-shaped rings in medium friction
and the convex-shaped rings in the highest friction, suggesting that the friction increased with
increasing contact pressure.

Figure 2.6 Schematic of (a) Concave-, (b) rectangular- and (c) convex- shaped ring
geometries to obtain different contact pressures [15].

According to classic friction laws, the friction coefficient cannot be affected by contact
pressure or normal load, and the friction force increases linearly with rising normal load.
However, this may not be applicable in extrusion processes, due to the excessively high
contact pressure. Under high contact pressures, apparent contact area is approaching real
contact area, and the friction stress is equivalent to or higher than the shear strength of the work piece material. Therefore, the shear deformation occurs in the work piece material rather than at the contact interface, thus the corresponding friction stress cannot be further increased with increasing contact pressure, but equals to the shear strength of the deformed material, which is the upper limit of friction stress. As such, the friction coefficient could decrease with increasing contact pressure. In addition, high contact pressure and fast sliding would produce massive frictional heat. In the case of low melting point metals, the frictional heat may cause softening or local melting of the material and lead to a low friction. Formation of oxide films at high temperatures might be responsible for low friction. In contrary, under high contact pressure conditions, the oxide or lubricant film can be penetrated, which leads to the contact of pure metals, and normally a high friction is observed. In general, in extrusion processes, friction decreases with increasing contact pressure. However, for highly oxidized or lubricated surfaces, results could be different, which depends on the surface conditions and magnitude of contact pressure. In Tan et al.’s research, the high contact pressure led to the penetration of lubrication film and the partial contact of pure metals consequently. The extent of penetration increased with increasing contact pressure, therefore the friction increased with increasing contact pressure.

2.2.1.3 Temperature effects on friction coefficient

Work piece/die interface temperature plays an important role in metal forming processes. The ring compression tests have been used to study the effects of temperature on friction. However, inconsistent results among previous studies were obtained. Pawelski et al. found that under unlubricated conditions, friction factor was independent of temperatures, ranging from 990 to 1160 °C [17]. Cho et al. studied the temperature effects on friction at temperatures of 200 and 400°C. AA6016-T6 aluminium alloy was work piece material. It was found that the value of friction factor was about 0.6 and was temperature in-sensitive [21]. Rudkins et al. studied the temperature effects on the dry friction coefficient of two types of steel. It was found that with the increasing temperature, friction coefficient increased from 0.75 to 0.9 [20]. Sagar et al. investigated the effect of temperature on frictional properties of CP aluminium. They found a sharp increase of friction when temperature was higher than 500 °C [22]. In Sadeghi and Dean’s work [18], ring compression tests were performed at temperatures ranging from 700 to 1200 °C, to evaluate the friction between steel work piece and die, which was lubricated by a graphite based lubricant. It was found that, the friction factor increased linearly with increasing billet temperature, ranging from 0.1 at 700 °C to 0.6
at 1200 °C. Male et al. investigated the temperature effects on the dry friction of aluminium, copper, α-brass, mild steel and titanium specimens [11]. It was found that below 120-140 °C, temperature had little effect on friction coefficient. Above this temperature range, there was an increase in friction coefficient. When the temperature was further increased, the friction coefficient increased (up to μ=0.57) with increasing temperature for aluminium and α-brass specimens; and the friction coefficient decreased with increasing temperature for copper and mild-steel specimens. For pure titanium specimens, however, temperature had no effect on the friction coefficient in a temperature ranging from 200 to 1000 °C.

In extrusion processes, temperature affects friction in different ways. An increasing temperature generally results in the softening of materials, thus the real contact area is increased, which leads to a high friction. In addition, more active atomic interdiffusion and intensive creep may occur at elevated temperatures, which result in a high adhesive friction. Lubricants may lose their effects when overheated, thus an increase of friction occurs. However, high temperatures may cause severe oxidation, which reduces the friction. If the temperature approaches the melting temperature of the testing material, a drastic decrease of friction occurs [2]. The viscosity of some lubricants can be reduced at elevated temperatures, which enhances the lubricant effect. Therefore, the rising temperature leads to quite different friction test results, depending on the extent of temperature and the material response to it. In some of the ring compression tests, the effect of oxide films may have compensated the effect of rising temperature, thus a constant friction can be observed. The combined effects of several factors could lead to various results, as observed in the reviewed ring compression tests.

2.2.1.4 Sliding speed effects on friction coefficient

In standard ring compression tests, the mutual sliding speed between the work piece and die is highly dependent on the friction conditions of the mating surfaces and varies from point to point. The study of the effect of sliding speed on friction may be achieved by applying different compression speeds or strain rates during ring compression tests. Hot ring compression tests have been conducted under different forming speeds. For example, Pawelski et al. investigated the effect of compression speed on the friction [17]. The results of ring compression tests without lubrication have shown that the friction factors lied between 0.8 and 0.9 and were not affected by speed. Under lubricated condition, friction factor was reduced with increasing forming speed. Cho et al. studied the effect of forming speed on the
friction at warm forming temperatures of 200 and 400°C with the compression speeds of 0.05 and 0.4 mm/s. It was found that, under dry sliding condition, the forming speed had limited effect on the magnitude of friction factor, and the friction factor at the tool/work piece interface was identified to be 0.6 [21].

According to the classic friction laws, sliding speed does not affect friction coefficient. However, in metal forming, the influence of sliding speed on friction could become explicit. The effect of sliding speed is mainly achieved through the increase of temperature in the contact region. A high sliding speed generally leads to the temperature rise due to the frictional heat, which may affect friction significantly, as explained in the previous section. Therefore, when the sliding speed is high enough, the material properties around the contacting area would be changed. For instance, the formation of oxidation films, decrease of viscosity of lubricants and drastic softening of the testing material could occur at high sliding rates, which reduce the friction. On the other hand, the failure of lubricants when overheated could result in the increase of friction.

The major advantage of using ring compression tests for the friction characterization is that only the measurement of shape change is involved [24], which is easy to conduct in practice. Nevertheless, in ring compression tests, the oxidation layer is normally trapped between the contacting surfaces, and the severity of deformation is low, thus the obtained friction results may not be comparable to real metal forming operations [25-27], in which new surfaces generation is large and deformation is severe, e.g. the friction in extrusion processes. In addition, the interface conditions during ring compression tests are hardly adjustable. For instance, it is difficult to evaluate the effects of sliding speed or contact pressure on friction by using standard ring compression tests, because the sliding speed at the work piece/tooling interface is mainly determined by friction and varies from point to point in an uncontrollable way; also, the contact pressure is mainly determined by material strength and cannot be adjusted, unless alternative shaped ring shapes are used [14, 15].

Friction in extrusion processes is a highly complex phenomenon, which can be affected by many factors, such as material properties and testing conditions. Furthermore, the interface conditions in the extrusion process may differ from point to point. For instance, the local temperature and sliding velocity in the bearing channel area could be much greater than those found on the container wall. Therefore, contact conditions in the ring compression tests have to be considered very carefully in order to emulate real contact conditions.
### Table 2.2 Example of applications of extrusion friction tests.

<table>
<thead>
<tr>
<th>Extrusion test</th>
<th>Work piece material</th>
<th>Tool material</th>
<th>Billet temp. (°C)</th>
<th>Die temp. (°C)</th>
<th>Speed (mm/s)</th>
<th>Lub(s)</th>
<th>Friction factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>1992 Shen <em>et al.</em>[19]</td>
<td>Al-Li alloy</td>
<td>FX-2</td>
<td>357</td>
<td>349-366</td>
<td>8.4</td>
<td>Lub A</td>
<td>m=0.15-0.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Lub B</td>
<td>m=0.15</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Lub C</td>
<td>m=0.1-0.15</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Lub D</td>
<td>m=0.1</td>
</tr>
<tr>
<td>1992 Buschhausen <em>et al.</em>[27]</td>
<td>AISI 1006</td>
<td></td>
<td>25</td>
<td>25</td>
<td>10</td>
<td>Lub</td>
<td>m=0.08-0.2</td>
</tr>
<tr>
<td>1997 Nakamura <em>et al.</em>[25]</td>
<td>6061</td>
<td>High speed steel</td>
<td>-</td>
<td>-</td>
<td>80</td>
<td>Ca-Al</td>
<td>μ=0.3-0.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>VG26</td>
<td>μ=0.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>MoS&lt;sub&gt;2&lt;/sub&gt;</td>
<td>μ=0.5-0.6</td>
</tr>
<tr>
<td>1998 Nakamura <em>et al.</em>[26]</td>
<td>6061</td>
<td>High speed steel, cemented carbide</td>
<td>-</td>
<td>-</td>
<td>80</td>
<td>VG2</td>
<td>μ&lt;sub&gt;d&lt;/sub&gt;=0.017-0.05 μ&lt;sub&gt;LP&lt;/sub&gt;=0.37-0.42</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>VG26</td>
<td>μ&lt;sub&gt;d&lt;/sub&gt;=0.005-0.048 μ&lt;sub&gt;LP&lt;/sub&gt;=0.15-0.19</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>VG1000</td>
<td>μ&lt;sub&gt;d&lt;/sub&gt;=0.001-0.039 μ&lt;sub&gt;LP&lt;/sub&gt;=0.15-0.28</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>MoS&lt;sub&gt;2&lt;/sub&gt;</td>
<td>μ&lt;sub&gt;d&lt;/sub&gt;=0.088-0.105 μ&lt;sub&gt;LP&lt;/sub&gt;=0.07-0.18</td>
</tr>
<tr>
<td>2002 Bakhshi-Jooybari [28]</td>
<td>CP Al</td>
<td>H13</td>
<td>25</td>
<td>25</td>
<td>-</td>
<td>No Lub.</td>
<td>m=0.84</td>
</tr>
<tr>
<td></td>
<td>Steel</td>
<td></td>
<td>900</td>
<td>900</td>
<td>-</td>
<td>Graphite</td>
<td></td>
</tr>
<tr>
<td>2003 Flitta <em>et al.</em>[29]</td>
<td>AA2024</td>
<td>Al-Cu ally</td>
<td>-</td>
<td>300-450</td>
<td>3 &amp; 8</td>
<td>No Lub.</td>
<td>m=0.654-0.92</td>
</tr>
<tr>
<td></td>
<td>AA6060</td>
<td></td>
<td>430</td>
<td>360-382</td>
<td>2 &amp; 5</td>
<td>No Lub.</td>
<td>Full sticking</td>
</tr>
</tbody>
</table>

### 2.2.2 Extrusion friction test for billet/container interface

It has been found that the ring compression tests are unable to reflect the real condition in some metal forming operations, in terms of contact pressure, deformation and material flow severity [9, 19, 25, 26]. In the 1990s, extrusion friction tests were proposed to estimate the global friction factor on the work piece/die interface. Table 2.2 shows examples of applications of extrusion friction tests. In the extrusion friction tests, two effects of friction...
have been used for the friction identification, namely: (I) the friction effects on material flow and (II) the friction effects on extrusion load.

2.2.2.1 Friction characterization based on the friction effects on the material flow

During metal forming operations, the material flow is significantly affected by the magnitude of friction force on the work piece/die interface, because the friction force generally restricts the metal flow. The material constrained by lower friction force normally flow faster than that constrained by higher friction force. Based on this principle, extrusion friction tests with great friction sensitivity have been developed and conducted. Buschhausen et al. proposed a combined backward extrusion process, named double backward extrusion test [27]. The principle of the double backward extrusion test is shown in Figure 2.7 (a). During the tests, the upper punch moved at a constant speed of 10 mm/s, while the lower punch and the die were stationary. The relative velocities between the punch, the work piece and the container led to different friction conditions, thus the height or length of the extruded cups was highly friction sensitive, particularly when low extrusion ratio was selected. FEM simulations of the double backward extrusion process was performed, and based on the simulation results, calibration curves were established. By using these curves, the friction can be determined quantitatively by only measuring the cup heights and punch stroke. Similarly, Nakamura et al. developed two new friction testing methods, namely, combined forward rod-backward can extrusion (as shown in Figure 2.7 b) [25] and combined forward conical can-backward straight can extrusion / combined forward straight can-backward straight can extrusion (as shown in Figure 2.7 c) [26]. In both friction testing techniques, the heights of the extrudates were sensitive to friction conditions and the friction could be estimated from the calibration curves obtained from FEM simulations. It was found from recent studies of the double cup (backward) extrusion test that, the interface pressures and surface generation in double cup extrusion may not be comparable to those found in cold forging. Therefore, process parameters of the double cup extrusion tests were studied by using FEM simulations [9]. It was found that the contact pressure at the billet–container interface and surface generation increased with increasing extrusion ratio, suggesting that double cup extrusion test with smaller extrusion ratio is suitable for friction determination, because of its high friction sensitivity. The test with higher extrusion ratio should be used for lubricants evaluation without finding a friction value, due to the higher similarity of contact conditions to those of real forging operations, in terms of contact pressure and surface enlargement.
Figure 2.7 The design of (a) double backward extrusion \cite{27}; (b) combined forward rod-backward can extrusion \cite{25} and (c) combined forward conical / straight can-backward straight can extrusion \cite{26}.

Recently, the effect of friction on the sliding velocity has been used in a different way for friction estimation. Schikorra et al. investigated the friction at the container wall during hot aluminium extrusion process. In their tests, hot extrusion of AA6060 billet with 19 AA4043
(AlSi5.5) rod markers embedded were performed and then the node displacement at the container wall was studied. Figure 2.8 shows the schematic design of the test. The node displacement at the container wall was measured. Strong experimental evidence has shown that, at the billet temperature of 430 °C, almost perfect sticking occurred at the billet/container interface [30].

![Figure 2.8 Process sketch (axis symmetry) [30].](image)

2.2.2.2 Friction characterization based on the friction effects on the extrusion load

In the forward extrusion process, the total extrusion load can be expressed as:

\[
F_{\text{total}} = f_c + f_d + F_d
\]

(2.10)

where \(F_{\text{total}}\) is the total extrusion load; \(f_c\) is the friction force between billet and container wall; \(f_d\) is the friction force between extrudate and die bearing and \(F_d\) is the force required for the plastic deformation of work piece material, which depends on the flow stress of work piece material, and is a function of total stain, stain rate and temperature.

According to Bakhshi-Jooybari’s research work [28], friction between the billet and container can be expressed as:

\[
f_c = \tau \pi dL
\]

(2.11)

where \(\tau\) is the frictional shear stress between billet and container wall, which was assumed to be constant over the entire contacting interface, and is a function of the shear strength of work piece material. \(d\) is the inner diameter of the container and \(L\) is the remaining length of the billet in the container. According to Equations 2.10 and 2.11, the total extrusion load is
affected by the friction force between the billet and container, in the meanwhile, it is mainly determined by the remaining length of the billet in the container. As such, the global friction force on the billet/container interface can be estimated by changing initial billet lengths [28, 31].

In backward extrusion process, there is no relative movement between the billet and container. The total extrusion load can be expressed as:

$$F_{total} = f_d + F_d$$  \hspace{1cm} (2.12)

Compared with the forward extrusion process, the difference in total extrusion force is mainly caused by the disappearance of friction force on the billet/container interface. It thus provides an alternative possibility to estimate the friction on the container wall.

A combined FEM simulation and forward hot extrusion method was employed by Flitta et al. [29] to estimate the friction on the container wall. The friction factor was estimated by adjusting the friction settings in the corresponding FE simulations to fit the experimentally obtained extrusion loads at particular ram displacements. It was found that the friction transformed from sliding at the initial stage of extrusion to almost full sticking at the steady state extrusion and the use of a constant friction factor for the whole hot aluminium extrusion cycle was incorrect. Shen et al. [19] developed a backward extrusion-type forging, named “Bucket” tests, to evaluate lubricants for forging process. In the “bucket” tests, the plastic deformation was more severe and contact pressure was higher than those found in the ring compression tests, which represented real forging conditions. The forging load was friction sensitive: when the friction was low, a lower forging load could be obtained and vice versa.

Compared to ring compression tests, extrusion friction tests have the following advantages: first, the geometry is more complex and thus is more similar to the real forming operations. Consequently, the estimated friction coefficients or factors would be more reliable. Second, during the extrusion friction tests, high hydrostatic pressures and severe surface enlargement can be achieved, which are highly favourable to simulate severe deformation conditions.

Similar to ring compression tests, for qualitative evaluation of the lubricants, only the extrusion friction tests would be sufficient, which is convenient for industrial practice. Nevertheless, in order to quantify the friction factor/coefficient, friction calibration curves are required for both tests, thus either theoretical analysis or FEM simulations is needed to
generate those curves. However, in terms of tooling cost and experimental complexity, the ring compression test is usually less than extrusion friction tests [32].

### 2.2.3 Localized friction measurement techniques

Ring compression and extrusion friction tests are general testing techniques for the estimation of global friction and evaluation of lubricants. For a local area of particular concern, the friction has to be evaluated by using specialized techniques.

#### 2.2.3.1 Direct stress measurement techniques

Many direct stress measurement techniques, such as pressure transmitting pins, split tools and ridged metallic sheets etc. have been used to measure the stress distribution on the work piece/die interface in metal forming operations. Among these techniques, the pressure transmitting pins are probably the most commonly used. The system comprises a "pin head" or rod with a small diameter, e.g. 2 mm [33], which is embedded into the body of the tool so that local contact pressures can be measured [34]. Recently, this technique was used to measure the friction at the contact interface [33]. The pins were embedded in different orientations to the die surface. The pin vertical to the die surface measures the axial or vertical component of stresses (Figure 2.9 a). The inclined pin detects the combined normal and tangential (friction) force (Figure 2.9 b). As such, both normal and frictional stresses at the interface were obtained from this design.

![Figure 2.9 The orientation of (a) vertical and (b) inclined pins [33].](image)

The testing results of Lupoi and Osman are shown in Figure 2.10 [33]. It is of great interest to see that during the simple compression tests of CP aluminium cylinders, friction coefficient
varied significantly along the mating interface and throughout the whole process. These results have confirmed that the use of a constant friction coefficient for the entire contacting interface throughout the whole forming process is incorrect.

Mori et al. investigated the pressure distribution on the extrusion die surface by using the pressure transmitting pin technique [35]. It was found that the pressure decreased with the increasing distance to the die centre, which was caused by the friction at billet/container interface.

![Graph showing variation of the friction coefficient along the interface at (a) 20mm and (b) 8mm billet heights [33].](image)

Figure 2.10 Variation of the friction coefficient along the interface at (a) 20mm and (b) 8mm billet heights [33].

![Schematic of split tool technique used in metal cutting process [36].](image)

Figure 2.11 Schematic of split tool technique used in metal cutting process [36].
Split tool technique is to use a tool composed of two parts separated by a gap. The gap should be small enough, ranging from 0.06 to 0.075 mm, to avoid the work piece material flowing into it. This technique has been successfully applied in the metal cutting process [37-39] (shown in Figure 2.11), in which, the dynamometer and charge amplifier were used to obtain the cutting force [39], and the results have confirmed the existence of the shear stress plateau under high normal pressure conditions [38], which is due to the limit of the shear strength of the work piece material.

2.2.3.2 Extrusion friction test for extrudate/bearing interface

In the extrusion process, the friction in the bearing channel region is of great importance, since it determines the surface quality of final products. However, this region is small and its effects on the total extrusion pressure generally can hardly be detected. This has brought difficulties in the study of friction in this region. In the past years, the friction in the bearing channel region has been studied experimentally by using extrusion dies with a tiny choke angle. A transition of friction from full sticking to sliding was observed (as shown in Figure 2.12), and the friction can be characterized from the lengths of full sticking and sliding zones [40-42]. It was found that the friction in the full sticking region was almost constant and in the sliding region, friction increased with increasing die angle and decreasing exit speed [41].

![Friction transition from sticking to slipping in the extrusion die](image)

Figure 2.12 Friction transition from sticking to slipping in the extrusion die [43].

2.2.3.3 Block on cylinder test
Block on cylinder test was developed by Björk et al. [43, 44] to simulate tribological interactions on the bearing surface of hot extrusion dies. The principle of the block on cylinder test equipment is shown in Figure 2.13. Prior to testing, the block and cylinder were heated by a resistance heater to a temperature of about 550 °C, to reproduce the typical temperature in the bearing channel region of hot aluminium extrusion processes. The temperature of the block was continuously monitored by a thermal couple. All the tests were conducted in an argon atmosphere to simulate the absence of air at the extrudate/die interface. During block on cylinder tests, the normal force between the block and cylinder was applied by using a spring, which gradually increased from an initial magnitude of 20N to its final value of 60N in one minute. As shown in Figure 2.13, the rotating Al cylinder represented the extruded profile. The friction force was continuously recorded by a load cell attached to the block. Intensive sticking friction was found in their results, leading to excessively high friction coefficients. Similar tests were conducted by Tercelj et al. [45] and Pellizzari et al. [46]. Their results have confirmed that excessive chemical reactions led to the severe die wear and high friction coefficients.

![Figure 2.13 Schematic of block on cylinder test equipment [43].](image)

### 2.2.3.4 Ball-on-disc test

Ball/pin-on-disc test is a widely used laboratory testing technique for the quantitative study of tribological behaviour of materials. A typical ball-on-disc tester is shown in Figure 2.14, which consists of a stationary pin in contact with a rotating disc (Figure 2.14 b). During the tests, a normal load is imposed by dead weights on top of the pin. In the meanwhile, the pin rubs on the same wear track repeatedly on the top surface of the rotating disc. The friction force between the ball and disc is transmitted to the end of the T-shaped arm (Figure 2.14 b).
in the form of displacement, which can be accurately measured and recorded. The testing conditions, such as, normal load, sliding speed and temperature etc. can be adjusted easily in a ball-on-disc tester and the individual effect of each factor on the friction can be studied accurately.

Although ball-on-disc tests are considered to be rather convenient and accurate, the testing results are mostly used for the evaluation and comparison purposes and few results have been implemented as friction boundary conditions in the FE simulations of extrusion processes. This is probably due to the lack of knowledge about the evolution of contact conditions during ball-on-disc tests. During ball-on-disc tests, a relatively high contact pressure can be achieved in a small contact area between the ball and rotating disc. If a soft material is sliding over a harder one, severe plastic deformation may occur, which could lead to the removal of oxide layers and contact of pure metal. In the meanwhile, the contact pressure may drop with the increasing sliding distance. Therefore, ball-on-disc tests are favourable to the friction characterization of the regions, in which local contact pressure is high and new surface generation is severe.

Ball-on-disc tests have been used to identify the friction coefficient for metal cutting processes [47-49]. In the work conducted by Bonnet et al. [48] and Rech et al. [49], high
contact pressure (up to 2 GPa) and sliding velocity (60-600 m/min) were achieved by using a modified ball-on-disc test and friction under metal cutting conditions was determined. It was found that the apparent friction obtained from ball-on-disc tests was contributed by ploughing friction (generated from the plastic deformation in front of the spherical pin head) and adhesive friction. The adhesive friction decreased with increasing sliding velocity and interface temperature. It has been confirmed that the friction coefficients determined by ball-on-disc tests can be used in the FEM simulation of a metal cutting process.

The first attempt of using ball-on-disc tests to simulate the interactions between aluminium and steel on the bearing surface of the extrusion dies was conducted by Ranganatha et al. [50]. A spherical tipped pin made from aluminium was in contact with a rotating steel disc. It was found that the friction increased with increasing temperature when the temperature was higher than 300 °C. The values of friction were excessively high due to the material transfer and back transfer between the hot aluminium and steel.

### 2.2.4 Comparisons of friction testing techniques for extrusion processes

In the preceding sections, six friction testing techniques have been reviewed. These techniques can be classified into three different groups, namely, field test (extrusion friction tests for container and bearing channel regions; direct stress measurement techniques); simulative test (ring compression test and block on cylinder test) and tribological test (ball-on-disc test). In this section, these friction testing techniques will be compared in different aspects, such as the interface conditions (contact pressure, test temperature, new surface generation and sliding speed), implementation of the test (calibration and cost aspects) and application of the test results.

#### 2.2.4.1 Contact pressure

Mori et al.’s results have provided a strong experimental evidence about the pressure distribution in the extrusion process [35], in which hot extrusion of AA1015 was performed at the temperature of 300 °C, the normal pressure on the die face was about 150 MPa. Of course, the contact pressure in the extrusion process may vary significantly from point to point, which is influenced by many factors, such as temperature, extrusion speed, extrusion ratio, work piece material properties and friction. Since the field test is to use real extrusion process to estimate the friction coefficients on the container wall or bearing surface, the contact pressure
is rather close to the real extrusion process, if the correct process parameters, such as
temperature, ram speed and extrusion ratio, are used.

The contact pressure in the ring compression tests is in the same level as the flow stress of the
work piece material, which might be lower than that found in an extrusion process and it can
hardly be adjusted. Similarly, in the block on cylinder tests, the contact pressure might be low
[45], especially when the testing temperature of the Al cylinder is close to its melting
temperature and the block tends to sink into the hot Al. Nevertheless, the use of two discs on
the side faces of the Al cylinder was helpful to achieve a higher hydrostatic pressure [45].

During the ball-on-disc tests at elevated temperatures, the initial contact pressure can be very
high, due to the small contact area between the spherical pin head and flat disc surface.
However, when a soft material is sliding over a hard one, severe plastic deformation or wear
of the softer material may occur under such a high contact pressure, which enlarges the
contact area significantly, consequently, reduces the contact pressure. Therefore, during the
sliding of the pin over the rotating disc, the contact pressure may drop in an uncontrollable
way, which strongly depends on the diameter of the spherical pin head, sliding distance and
the strength of the testing materials. In general, the contact pressure increases with decreasing
ball size [51] and decreases with increasing sliding distance [52]. It is worth noting that, the
selection of the pin and disc materials could affect test results. If the pin is made from a soft
material, and the disc is made from a hard one, severe plastic deformation would occur on the
tip of the pin, which leads to a significant enlargement of the contact area and a steep decrease
of contact pressure, after a short distance of sliding. Therefore, the contact pressure during the
steady-state sliding is probably in the same level as the yield strength of the soft material. On
the other hand, if the disc is made from a soft material, while the pin is made from a hard one,
plastic deformation tends to occur in the disc, but the material flow is most likely constrained
by the remaining disc material, which is much larger than the size of the wear track. Hence a
relatively high hydrostatic pressure which is greater than the strength of the disc material
would be imposed onto the spherical pin head. As such, different materials combinations
would result in different contact pressures, hence the selection of pin and disc mating
materials need to be considered carefully prior to testing, especially when the strengths of the
pin and disc materials are different. In the meanwhile, the selection of ball size and sliding
distance is of great importance.

2.2.4.2 Test temperature
Ring compression test is suitable for all ranges of test temperatures which do not vary significantly during the test, because the heat generation caused by friction or plastic deformation normally can be ignored, due to the low severity of plastic deformation. The block on cylinder tests are carried out under isothermal conditions, and the test temperature can be continuously monitored through the whole testing cycle, therefore, this technique is suitable for all ranges of test temperature which was rather stable during the tests with an error of less than 2 °C, as indicated in [45].

During the extrusion friction tests, only two process parameters can be adjusted: the initial billet temperature and extrusion speed [53]. The test temperature can be affected by these two factors but varies significantly throughout the extrusion process [54] in an uncontrollable way. Therefore, it is probably not sensible to study the temperature effects on the friction by means of extrusion friction test.

In the ball-on-disc tests, the frictional heat can be normally ignored. In most of the ball-on-disc test equipment, the test temperature can be controlled accurately by a conduction heater. However, one possible exception is the ball-on-disc tests under an excessively high sliding speed and normal load, in which the frictional heat should be considered [48, 49].

2.2.4.3 Sliding speed

In the ring compression tests, the mutual sliding speed between the work piece and tool surface cannot be controlled, which varies with the friction conditions and compression speed considerably. Therefore, ring compression tests are not suitable for studying the sliding speed effects on the friction.

In the block on cylinder tests, the sliding speed can be accurately controlled through the adjustment of rotating speed of the Al cylinder, thus it can be used to investigate the sliding speed effects.

In the extrusion friction tests, the sliding speed of work piece over the tooling varies remarkably with local conditions and generally cannot be controlled. For instance, in the regions, such as the container wall, when full sticking occurs, the mutual sliding speed is nearly zero [30]. On the other hand, in the area such as the bearing channel, the mutual sliding speed between the work piece and tooling surface can be very high, typically up to 90 m/min [45]. The local sliding speed is strongly influenced by extrusion ratio, extrusion speed and
frictional conditions. Therefore extrusion friction tests may not be suitable for studying the effects of sliding speed on the friction.

In the ball-on-disc tests, the sliding speed can be controlled accurately via adjusting the rotating speed of the disc. Hence it can be used to study the sliding speed effects on the friction.

2.2.4.4 New surface generation

As indicated in many previous research, one of the limitations of ring compression tests is that the new surface generation is low, which cannot emulate the metal forming operations, in which severe plastic deformation occurs. Moreover, the oxide layer is normally trapped between the mating surfaces, which may act as a lubricant to reduce friction. Therefore, ring compression tests are suitable for simulating the contact conditions which involve severe new surface generation. In the block on cylinder tests, the new surface generation is strongly affected by the applied normal contact pressure. At high normal pressures, fully or partially contact of pure metal may occur and the sticking phenomenon can be observed which leads to an excessively high friction coefficient.

In the extrusion friction tests, new surface generation is rather intensive, especially when a high extrusion ratio is selected. Therefore, they can be used to simulate the metal forming operations, in which pure metal contact is dominant.

In the ball-on-disc tests, a large amount of new surface generation is normally involved, but probably only during the initial run-in period, suggesting that to emulate the extrusion process, short sliding distance ball-on-disc tests could be used.

2.2.4.5 Calibration

Ring compression tests can be used for the lubricant evaluation and friction characterization. For the former purpose, no calibration is required. Nevertheless, in order to determine friction coefficient quantitatively, friction calibration curves must be generated by using FEM simulation or theoretical analysis, with different values of friction as input parameter. In addition, theoretically, a set of friction calibration curves is only corresponding to one particular work piece material and under a certain test conditions. In the block on cylinder tests, a load cell is used to measure the friction force, thus the standard routes for load cell calibration would be sufficient, which is easier than that of ring compression tests.
Similar to the ring compression tests, in the extrusion friction tests, the height of extrudates, load-stroke curves or the lengths of sticking/slipping zone do not provide an explicit sign of friction coefficient, thus the calibration procedure including extrusion tests and intensive FEM simulations is required and the truthfulness of the calibration relies on the accuracy of the material model used in the FEM simulations. When the pressure transmitting pin technique is used, calibration has to be conducted prior to testing, in which dead weights are normally used to impose a normal load. However, the stress condition in the calibration tests may differ from those found in real metal forming operations, in which both tangential and normal forces exist. The tangential force over the pin head could cause the friction force between the pin and its bore, which reduces the movement of the pin head. Consequently, inaccurate testing results may be obtained if the friction between the pin and its bore is ignored.

Most of the ball-on-disc tests are conducted in a standard tribometer, in which a sophisticated sensor is used to measure the friction. Therefore, the calibration of the test rig can be performed following the standard calibration routes of a tribometer.

2.2.4.6 Cost aspects

To determine the friction coefficient, only the measurement of the dimensions of the ring after compression is required. Therefore, the cost of ring compression tests is low. However, to generate friction calibration curves is time consuming, which requires intensive FEM or theoretical analysis. Block on cylinder test is conducted in a novel test rig. Therefore, the construction of the rig might be expensive and time consuming. During testing, the contact pressure is low, thus a longer testing period is probably required to compensate the unfavourable effects of low contact pressure [45].

Extrusion friction tests are relatively complicated to perform, and the manufacture of the extrusion die could be expensive. The testing procedure of extrusion friction tests is complicated, which may involve the preheating of the die and billet and the ejection of formed testpiece etc.

Ball-on-disc tests are easy to perform, but the cost of a tribometer might be high. In addition, the post-processing of test data involves a large amount of modelling work, which is time consuming. This will be explained in section 2.2.4.7.

2.2.4.7 Accuracy and application of the test results
The accuracy of the ring compression test depends on the friction calibration curves. The generation of these curves is based on the assumption of a constant friction at the mating interface, which should be sufficient for an estimation of global friction between the mating materials. As a simulative test, there is always a hot debate on the transfer of testing results from ring compression tests to the extrusion process, although the ring compression test was originally developed for the friction characterization of cold extrusion process [10]. The accuracy of block on cylinder test depends on the load cell attached to the block, thus the measured friction force should be highly precise, and the high friction coefficient obtained from the tests reflects strong chemical interactions between aluminium and steel at elevated temperatures, therefore the application of the test results into extrusion process where the contact pressure is relatively low (e.g. bearing channel region) is feasible.

The friction coefficients obtained from extrusion friction tests are estimated average values over the entire container or bearing surface. Therefore, these values can be transferred into the corresponding real extrusion process directly [55]. However, to transfer these friction data into another extrusion process when the test parameters are changed is doubtful, because the geometrical and process parameters of the extrusion friction tests affect the similarity of the friction tests to the real forming operations, in terms of surface expansion and contact pressure [9]. In the bearing channel of the extrusion die, the friction transition occurs from full sticking at the extrusion die entrance to slipping at die exist, and the friction is estimated by means of measuring the lengths of the two zones. The accuracy of this method is probably dependent on the lengths of these two zones on the bearing surface [41, 56]. However, the transfer of the friction test results from one extrusion test to another is not feasible. The friction results obtained from direct stress measurement techniques are highly accurate, provided a proper calibration is conducted prior to testing and the implementation of the friction test results into FE simulations as frictional boundary conditions is feasible. Again, the test results obtained from direct stress measurement techniques cannot be transferred to other extrusion processes.

The results of friction obtained from ball-on-disc tests are highly accurate. However, the test results cannot be transferred into a metal forming operation directly. This is due to the build-up of metal in front of the ball [48, 49, 57], which causes ploughing friction, and leads to an overestimation of the friction between the mating materials, when a ball made from a hard material is sliding against a disc made from a soft material. As such, the ploughing friction and shear/adhesion friction have to be discriminated by means of FEM simulations or
theoretical analysis, and only the adhesive part of apparent friction representing the real friction between the ball and disc should be used in metal forming operations [48, 49].

2.3 FRICTION MODELS FOR EXTRUSION PROCESSES

It is widely accepted that the friction model is one of the key input boundary conditions in FE simulations of extrusion process. In the past years, the traditional friction laws including the Coulomb [58, 59] and Shear friction laws [29, 54, 60, 61] have been implemented into the FE simulations of extrusion process and new friction models were developed [41, 62, 63]. In this section, different friction models for extrusion processes are reviewed and compared in detail.

2.3.1 Coulomb friction model

In Coulomb friction law, friction stress (force) is assumed to be proportional to the normal pressure (force), which can be expressed in the form of Equation 2.1.

\[ f = \mu N \left( \frac{2}{\pi} \arctan \left( \frac{\mu}{\mu_0} \right) \right) \frac{u_s}{u_s} \]

where \( u_s \) is the sliding velocity, and \( u_0 \) is a positive constant which is smaller than \( u_s \). By using this equation, when sliding velocity is zero, the friction force goes to zero. This avoids the jump of friction force at neutral points.

2.3.2 Shear friction model

The Shear friction model, also named as Tresca friction model, assumes that the friction stress is proportional to the shear flow stress of the deformed material, and it can be expressed as:

\[ f = mk \]

where \( m \) is the friction factor, which normally ranges between 0 (frictionless condition) and 1 (full sticking condition). \( k \) is the shear flow stress of the deformed material. The friction factor \( m \) is considered as a ‘fit factor’ in the FE simulations of aluminium extrusion processes.
2.3.3 Temperature based friction model for the billet/container interface

Flitta and Sheppard investigated the effects of initial billet temperature on friction in extrusion process [29]. In their study, the Shear friction model (Equation 2.14) was used and the friction factor was adjusted in the FE simulations to achieve good agreements between the simulated and experimentally measured extrusion loads at two locations of particular interest. The first location was selected to determine the friction at the initial stage of extrusion, after the peak extrusion pressure has been achieved. The second location is the pressure of steady state extrusion to investigate the influence of temperature rise. It was found that, the assumption of a constant value of friction factor for all extrusion temperatures was incorrect and a transition from sliding to sticking friction was observed when initial billet temperature was increased from 300 to 450 °C. A linear relationship between friction factor and temperature was obtained, which can be expressed by:

\[ \bar{m} = A + BT \]  

(2.15)

where \( \bar{m} \) is the average friction factor; \( T \) is temperature; \( A \) and \( B \) are constants.

A generalized form of the friction model was proposed, to make it applicable for all extrusion variables [29, 55]:

\[ \bar{m}_{\Delta_s} = \left[ a + \alpha n \ln \left( \frac{Z_d}{A} \right) + b \right]_{\Delta_s} \]  

(2.16)

where \( \bar{m}_{\Delta_s} \) is the average friction factor, which varies with ram stroke; \( Z_d \) is the average of the Zener–Hollomon parameter; \( \alpha, n \) and \( A \) are the constants related to mechanical behaviours of the work piece material; \( a \) and \( b \) are constants for the friction model.

2.3.4 Empirical friction models for the bearing channel of extrusion dies

Based on the experimental observations, i.e. a full sticking zone at the die entrance region; and a sliding zone at the die exit area [41, 42, 65], Abtahi developed an empirical friction model for the bearing channel of hot aluminium extrusion process [41]. In this model, the shear/friction stress was extrusion speed and die angle dependant. In the sticking zone, the friction was of sticking type, which decreases slightly with the increasing distance from die
entry, as shown in Equation 2.17. In the sliding region, a linear interpolation between the friction at the slipping point and that at the die exit was used, as shown in Equation 2.18.

\[
\tau_\mu = \tau_0 + kx \quad (2.17)
\]

\[
\tau_{sl} = m \ln \left( \frac{\alpha - \alpha' + n}{v} \right) (L - x) + \tau_e \quad (2.18)
\]

where \(\tau_\mu\) is the shear stress generated in the sticking zone; \(\tau_0\) is the shear stress at the extrusion die entry; \(x\) is the distance from die entry; \(k\) is a function of exit speed; \(\tau_{sl}\) is the friction in the slipping zone; \(\alpha\) is the choke angle of the extrusion die; \(\alpha'\) is the current rotation of the bearing (deflection); \(v\) is the extrusion speed in the bearing channel; \(L\) is the bearing length; \(\tau_e\) is the friction stress at the extrusion die exit; \(m\) and \(n\) are constants. Figure 2.15 shows the predicted shear stress (as a function of distance from die entry) in the bearing channel by using Abtahi’s friction model.

![Image showing shear stress as a function of distance from die entry](image)

Figure 2.15 Predicted shear stress as a function of distance from die entry using Abtahi’s friction model [41].

Saha developed another friction model based on the same experimental observations as Abtahi’s [65], in which, the friction force was calculated as the sum of friction forces generated from sticking and slipping zones (Equation 2.19). In the sticking zone, full sticking
friction was assumed; in the slipping zone, the friction was assumed to be of shear type with a constant friction factor:

\[ F_r = m_1 k A_1 + m_2 k A_2 \]  \hspace{1cm} (2.19)

where \( F_r \) is the total friction force within the bearing channel; \( m_1 \) and \( m_2 \) are the friction factors in the sticking and slipping zones, respectively; \( A_1 \) and \( A_2 \) are the real area of contact in the sticking and slipping zones, respectively; \( k \) is the material shear strength.

### 2.3.5 Physical friction model for the bearing channel of extrusion dies

Ma et al. developed a novel physical friction model for unlubricated aluminium extrusion processes [56, 63], which is capable of predicting the sticking/slipping lengths in the bearing channel of aluminium extrusion die.

**The asperity shape model:**

In Ma’s model, the calculation of the local friction distribution was based on the asperity ploughing, thus the friction is significantly affected by the local contact geometry. The extrudate was assumed as a smooth and perfectly plastic surface. The bearing surface was modelled as a rigid rough surface which consists of individual summits with a power law generatrix:

\[ w = Sh^{\chi} \]  \hspace{1cm} (2.18)

\[ S = \chi \omega ^{-\frac{1}{2}} \]  \hspace{1cm} (2.19)

where \( h \) and \( w \) are the indentation depth and generatrix width; \( S \) is the shape variation coefficient, which reflects the sharpness of an asperity, and is different from one asperity to another; \( \omega \) is the tip curvature of an asperity; \( \chi \) and \( \lambda \) are the constants to characterize the shape of asperities.

**The friction model:**

Challen and Oxley’s slipline friction model [66] was used to calculate the total friction force and normal force acting on all the asperities in contact per unit area. It was assumed that only the front half of the asperities sustained the normal load, which generated friction correspondingly.
\[ F = \frac{1}{2} \pi H \chi^2 \sum_{i=1}^{n} f \left( \theta_i, f_{
u k} \right) \omega_i^{-1} \delta_i^{2l} \]  \tag{2.20}

\[ P = \frac{1}{2} \pi H \chi^2 \sum_{i=1}^{n} \omega_i^{-1} \delta_i^{2l} \]  \tag{2.21}

where \( \delta_i \) is the effective indentation depth of an asperity at a given cut-off height of \( h \).

Sticking friction at the extrudate/bearing interface was assumed to occur when the friction stress \( (\tau) \) reached the shear strength of aluminium \( (k) \).

### 2.3.6 Comparison of different friction models

#### 2.3.6.1 Determination of model parameters

Coulomb friction model assumes that the friction force (stress) is proportional to normal load (pressure), which can be directly measured by using standard equipment, such as load cell, pressure transmitting pins or dynamometer; therefore the friction coefficient can be determined accordingly.

In the Shear friction model, the friction factor is a function of the shear flow stress of the deformed material, which is rather difficult to measure directly due to the uncertainty of the shear flow stress at the contact interface. Therefore, the assignment of friction factors in the FE simulations was, sometimes, pure guess work, although some friction testing techniques, such as ring compression tests and extrusion friction tests, could be used to provide some guidelines for the assignment of friction factors.

In Flitta’s friction model, the parameters in the model have to be determined via extrusion tests and the corresponding FE simulations. If the extrusion parameters, such as extrusion speed and initial billet temperature \( etc. \) have been changed, the parameters for the friction model may have to be re-determined accordingly, because any change in extrusion parameters would lead to the change of extrusion temperatures.

In Abtahi’s and Saha’s empirical friction models and Ma’s physical friction model, the model parameters have to be determined from extrusion tests using chocked extrusion dies. However, it is unlikely to identify all the model parameters accurately. However, these models represent the complex nature of friction in the bearing channel to extrusion dies.
2.3.6.2  Application of the friction models

The Coulomb friction model was considered to reflect the real behaviour of friction [67]. However, in the FE simulations of extrusion processes, Coulomb friction model is not widely used, because this model tends to overestimate the friction stress, particularly under high contact pressures, unless the shear strength of the work piece material is taken into account [68, 69]. As such, the Coulomb friction law is normally used in combination with a limiting stress in many FE codes, namely, when the friction stress is lower than the shear strength of the deformed material, the friction stress follows the Coulomb friction law. When the calculated friction stress is greater than the shear strength of the deformed material, shear flow stress is used to replace the friction stress. This is shown in Equation 2.22:

\[
f = \mu \sigma_n, \text{ when } \mu \sigma_n < k; \quad f = k, \text{ when } \mu \sigma_n \geq k
\]  

(2.22)

The Shear friction model relates the friction stress to the shear strength of the deformed material, which may not reflect the real behaviour of friction in some cases, e.g. under the contact pressures much lower than the deformed material, but this model has been almost exclusively used in the FE simulation of the extrusion processes due to its theoretical simplicity and numerical rigidity [67]. Generally, the Shear friction model can be used for the metal forming processes or regions in which severe plastic deformation occurs at mating interface.

Since the extrusion process is a highly complicated thermo-mechanical process, the local contact conditions, such as temperature, contact pressure and strength of the work piece material may vary considerably throughout the extrusion cycle, the use of a constant friction coefficient or factor may not be sensible. Undoubtedly, the use of state variables dependent friction coefficients or factors would be a rather effective way to reflect the complex nature of friction in the extrusion processes.

References


Chapter 3
MODELLING OF HIGH TEMPERATURE BALL-ON-DISC TESTS

ABSTRACT

During high-temperature ball-on-disc tests of aluminium against steel, hot aluminium deforms and wear track evolves. The individual contributions of ploughing and shearing to the apparent friction and the contact pressure are unknown. The aim of this chapter was to develop a model capable of determining these parameters. It was found that during high-temperature ball-on-disc tests, the ploughing friction accounted for only about 1% of the apparent friction, although the ploughing friction coefficient increased with increasing wear lap, while the shear friction played a dominant role in determining the apparent friction measured. The mean contact pressure decreased significantly as the test proceeded.
3.1 INTRODUCTION

Ball-on-disc tests have been widely used in research to describe the tribological behaviour of materials in contact and to determine friction coefficient [1, 2]. It is however important to note that ball-on-disc tests, being similar to scratch tests [3-11] but different from ring compression tests and pin-on-disc tests, generate friction, or apparent friction that may contain two integrated components, i.e. ploughing friction and shear friction [12]. This is certainly the case in ball-on-disc tests at elevated temperatures, where the contact pressure between the ball and disc is considerably high relative to the yield strength of the disc material. As such, the ball indents the disc, creating an extra resistance due to the plastic deformation of the disc in front of the ball, i.e. ploughing friction. The existence of ploughing friction is thus a phenomenon in ball-on-disc tests under these conditions. The friction coefficient so obtained may lead to an overestimate when applied to the situation where no ploughing is involved. In the case of aluminium extrusion, for example, only shear friction at the die and aluminium interface is present. To be able to make use of the results from high-temperature ball-on-disc tests for modelling of the interfacial contact in aluminium extrusion, the ploughing friction component should be deducted from the apparent friction measured during high-temperature ball-on-disc tests. The method to distinguish these two friction components in ball-on-disc tests has not been established. Hegadekatte et al. [13] and Jiang and Arnell [14] modelled the wear track on the cross-section in ball-on-disc tests and obtained results in agreement with experimental measurements. However, no differentiation between the ploughing friction and shear friction was made. In recent years, FE simulations [4], analytical models [5, 6] and mechanical models [7-11, 15] have been utilised to evaluate the ploughing friction and shear friction in scratch tests. However, these models cannot be directly applied for high-temperature ball-on-disc tests, because the tribological behaviour of the evolving wear track in ball-on-disc tests is much more complex than that of a single wear track in scratch tests.

The objective of the present chapter was, on the basis of the existing models for scratch tests, especially the model of Tayebi et al. [11], to develop a model capable of distinguishing the ploughing friction and shear friction during high-temperature ball-on-disc tests and determining the mean contact pressure on the deformed disc.
3.2 MODEL DEVELOPMENT

3.2.1 Existing models for scratch tests

The model of Goddard [16] has been widely used in the mechanical analysis of scratch tests. It is based on the force balance of a spherical indenter to reproduce the forces acting on the contact surface. In this model, the shear stress (friction stress) is assumed to lie on the horizontal planes. The forces acting on the elemental area $dA$ are expressed as:

\[
\begin{align*}
  dF_x &= \left( pr^2 \sin^2 \beta \cos \gamma + fr^2 \sin \beta \sin \gamma \right) d\gamma d\beta \\
  dF_z &= pr^2 \cos \beta \sin \beta d\gamma d\beta
\end{align*}
\] (3.1)

The assumption has been found to underestimate the apparent friction coefficient [9].

To compensate for the underestimation, Tayebi [11] modified Goddard’s model. In Tayebi’s model, the incline angle of the friction force is considered and the friction force assumed to be in the direction opposite to the relative velocity at the contact point. Figure 3.1 gives a close-up view of the friction forces acting on the elemental area $dA$. The forces acting on the elemental area are expressed by Equation 3.2. In both Equations 3.1 and 3.2, $p$ and $f$ are the mean contact pressure and friction stress, respectively, and $r$ represents the radius of the ball.

Figure 3.1 Close-up view of the friction force and velocity on the elemental area $dA$. 
\[
\begin{aligned}
\begin{cases}
    dF_x = \left( pr^2 \sin^2 \beta \cos \gamma + fr^2 \sin \beta \sqrt{\cos^2 \gamma \cos^2 \beta + \sin^2 \gamma} \right) d\gamma d\beta \\
    dF_z = \left( pr^2 \cos \beta \sin \beta - fr^2 \frac{\cos \gamma \sin^2 \beta \cos \beta}{\sqrt{\cos^2 \gamma \cos^2 \beta + \sin^2 \gamma}} \right) d\gamma d\beta
\end{cases}
\end{aligned}
\] (3.2)

These two models developed for scratch tests are not directly applicable for ball-on-disc tests. This is because, in scratch tests, there is no disc material removal in front of the indenter, the same as in ball-on-disc tests during the first lap of wear. However, as the ball-on-disc test proceeds, the disc material in the wear track is gradually removed, which is the dominant tribological phenomenon and should be taken into consideration in the model for ball-on-disc tests.

### 3.2.2 Extension of the models to ball-on-disc tests

In the model for ball-on-disc tests, it is assumed that the disc material deforms homogeneously and behaves as a perfectly plastic material or a perfectly elasto-plastic material, and the bulging of the disc material in front of the ball is ignored; while the ball material is perfectly rigid. The mean contact pressure over each lap is assumed to be unvaried. As the surface condition of the disc in contact with the ball in the first lap of wear is totally different from that in any of subsequent laps, the model will address these two situations individually. In the first instance, the elastic recovery of the disc at the rear of the ball is ignored. To allow the model to cover a wide range of material matings, the elastic recovery of the disc at the rear of the ball is also taken into account, when the disc material has a relatively low elastic modulus, such as polymer.

#### 3.2.2.1 First lap of wear with no elastic recovery

For a perfectly plastic deforming disc in contact with a perfectly rigid ball, the contact interface during the first lap of wear is the same as that of the scratch test, as shown schematically in Figure 3.2. The horizontal and vertical forces can easily be obtained by integrating Equation 3.2, i.e.,

\[
\begin{aligned}
\begin{cases}
    F_x = 2\int_0^{\tilde{\xi}_1} \int_0^{|\tilde{\xi}_1|/2} dF_x \\
    F_z = 2\int_0^{\tilde{\xi}_1} \int_0^{|\tilde{\xi}_1|/2} dF_z
\end{cases}
\end{aligned}
\] (3.3)

where \( \tilde{\xi}_1 \) is the upper integral limit of the angle \( \beta \) (see Figure 3.2). \( \tilde{W}_1 \) is the width of the wear track after the first lap of wear.
Figure 3.2 Schematic drawing showing the contact interface with no elastic recovery of the disc in the first lap of wear during ball-on-disc tests.

3.2.2.2 *Arbitrary* \((i+1)th\) *lap of wear with no elastic recovery*

For a perfectly plastic disc mating with a perfectly rigid ball in an arbitrary \((i+1)th\) lap after the first lap, disc wear occurs in the wear track where the disc material has been removed down to a certain depth. Figure 3.3 schematically shows the contact interface in the \((i+1)th\) lap of wear. The horizontal and vertical forces can be obtained by integrating Equation 3.2:

\[
\begin{align*}
F_x &= 2 \int_0^{\delta_i} \int_0^{\pi/2-\alpha_i} dF_x + 2 \int_0^{\delta_i} \int_{\pi/2-\alpha_i}^{\pi/2} dF_x \\
F_z &= 2 \int_0^{\delta_i} \int_0^{\pi/2-\alpha_i} dF_z + 2 \int_0^{\delta_i} \int_{\pi/2-\alpha_i}^{\pi/2} dF_z
\end{align*}
\]

Combining Equation 3.2 and Equation 3.4 leads to:
Figure 3.3 Schematic drawing showing the contact interface with no elastic recovery of the disc in an arbitrary \((i+1)\)th lap of wear during ball-on-disc tests.

\[
F_s = 2r^2 \int_0^{f(y)} \int_{\eta/2-\eta_0}^{\eta/2} (p \sin^2 \beta \cos \gamma) d\gamma d\beta + 2r^2 \int_0^{\xi} \int_{\eta/2-\eta_0}^{\eta/2} (p \sin^2 \beta \cos \gamma) d\gamma d\beta \\
+ 2r^2 \int_0^{f(y)} \int_{\eta/2-\eta_0}^{\eta/2} (f \sin \beta \sqrt{\cos^2 \gamma \cos^2 \beta + \sin^2 \gamma}) d\gamma d\beta \\
+ 2r^2 \int_0^{f(y)} \int_{\eta/2-\eta_0}^{\eta/2} (f \sin \beta \sqrt{\cos^2 \gamma \cos^2 \beta + \sin^2 \gamma}) d\gamma d\beta \\
F_\zeta = 2r^2 \int_0^{f(y)} \int_{\eta/2-\eta_0}^{\eta/2} (p \cos \beta \sin \beta) d\gamma d\beta + 2r^2 \int_0^{\xi} \int_{\eta/2-\eta_0}^{\eta/2} (p \cos \beta \sin \beta) d\gamma d\beta \\
- 2r^2 \int_0^{f(y)} \int_{\eta/2-\eta_0}^{\eta/2} \left( f \frac{\cos \gamma \sin^2 \beta \cos \beta}{\sqrt{\cos^2 \gamma \cos^2 \beta + \sin^2 \gamma}} \right) d\gamma d\beta \\
- 2r^2 \int_0^{f(y)} \int_{\eta/2-\eta_0}^{\eta/2} \left( f \frac{\cos \gamma \sin^2 \beta \cos \beta}{\sqrt{\cos^2 \gamma \cos^2 \beta + \sin^2 \gamma}} \right) d\gamma d\beta
\]

Although Goddard’s model results in some errors because the incline angle of the friction force is not taken into consideration, it is relatively simple and has been widely used. For comparison with the present model (Equation 3.5), Goddard’s model may also be modified to
make it applicable for the ball-on-disc test. Integrating Equation 3.1 gives the expressions of the horizontal and vertical forces acting on the ball:

\[
\begin{align*}
F_x &= r^2 \int_{\gamma_0}^{\pi/2-\gamma_0} p \left( f_1(\gamma) - \frac{1}{2} \sin \left( 2f_1(\gamma) \right) \right) \cos \gamma d\gamma + pr^3 \left( \xi_1 - 2 \sin \left( 2\xi_1 \right) \right) (1 - \cos \omega_i) \\
&\quad + 2r^2 \int_{\gamma_0}^{\pi/2-\gamma_0} f_1(1 - \cos \left( f_1(\gamma) \right)) \sin \gamma d\gamma + 2r^2 f_1 \sin \omega_i (1 - \cos \xi_1) \\
F_z &= \frac{1}{2} p \left( \frac{a_{\omega_1}}{2} \right)^2 (\sin 2\omega_i + 2\omega_i) 
\end{align*}
\]

As shown in Figure 3.3c, in the area COD, \( f_1(\gamma) \) is the upper integral limit of \( \beta \), while in the areas AOD and COB, \( \xi_i \) is the upper integral limit of the angle \( \beta \). \( \omega_i \) is the angle to specify the position of the front contact boundary during the \((i+1)th\) lap of wear [8, 9].

In the case of material mating in high-temperature ball-on-disc tests between hot aluminium and steel, the elastic recovery at the rear of the ball may be ignored and then Equations 3.5 and 3.6 can be used. For the solutions of Equations 3.5 and 3.6, the integral limits must be determined and special ball-on-disc tests were performed to determine these integral limits from wear track widths after different laps of wear. However, for a generic model also suitable for materials with low elastic modulus, the elastic recovery of the disc should be considered.

### 3.2.2.3 First lap of wear with elastic recovery

Figure 3.4 schematically shows the contact interface with elastic recovery of the disc at the rear of the ball in the first lap of wear. The horizontal and vertical forces can be obtained by integrating Equation 3.2:

\[
\begin{align*}
F_x &= 2 \int_{\theta_1}^{\pi/2+\theta_1} \int_{\xi_1}^{\pi/2+\xi_1} dF_x + 2 \int_{\xi_1}^{\pi} \int_{\pi/2+\theta_1}^{\pi} dF_x \\
F_z &= 2 \int_{\theta_1}^{\pi/2+\theta_1} \int_{\xi_1}^{\pi/2+\xi_1} dF_z + 2 \int_{\xi_1}^{\pi} \int_{\pi/2+\theta_1}^{\pi} dF_z 
\end{align*}
\]

where \( \xi_1 \) is the upper integral limit of the angle \( \beta \) in the areas AOD and COB and the front half contact surface, \( f_1(\gamma) \) the upper integral limit of \( \beta \) in the area AOB (Figure 3.4c) and \( \theta_1 \) the angle to specify the position of the rear contact boundary after the first lap of wear.
3.2.2.4 *Arbitrary* \((i+1)th\) *lap of wear with elastic recovery*

The contact interface of any arbitrary \((i+1)th\) lap with the consideration on elastic recovery at the rear of the ball is presented in Figure 3.5. The horizontal and vertical forces can be obtained by integrating Equation 3.2:

\[
\begin{align*}
F_x &= 2\int_0^{\phi_{(i+1)}} \int_0^{\pi/2 - \phi_i} dF_x + 2\int_0^{\phi_{(i+1)}} \int_{\pi/2 + \theta_i}^\pi dF_x + 2\int_0^{\phi_{(i+1)}} \int_{\pi/2 + \theta_i}^\pi dF_x \\
F_z &= 2\int_0^{\phi_{(i+1)}} \int_0^{\pi/2 - \phi_i} dF_z + 2\int_0^{\phi_{(i+1)}} \int_{\pi/2 + \theta_i}^\pi dF_z + 2\int_0^{\phi_{(i+1)}} \int_{\pi/2 + \theta_i}^\pi dF_z
\end{align*}
\]

where, as illustrated in Figure 3.5c, \(\phi_i\) and \(\theta_i\) are the angles to specify the positions of the front and rear contact boundary during the \((i+1)th\) lap of wear and \(\xi_i\) is the upper integral.
limit of the angle \( \beta \) in the areas \( COB \) and \( AOD \). In the areas \( COD \) and \( AOB \), the upper limit of \( \beta \) is the functions of \( \gamma \), namely, \( f_i(\gamma) \) and \( f_{i+1}(\gamma) \), respectively.

Figure 3.5 Schematic drawing showing the contact interface with elastic recovery in an arbitrary \(((i+1)th)\) lap of wear during ball-on-disc tests.

### 3.3 EXPERIMENTAL DETAILS

A CSM® high-temperature tribometer with a ball-on-disc configuration was used to perform a series of high-temperature friction tests over short distances for the solutions of the models, i.e. Equations 3.5 and 3.6. The disc was made of a high-strength aluminium alloy, AA7475. It had a thickness of 5 mm, a diameter of 49 mm, and a polished surface. The ball was made of an austenite stainless steel and had a diameter of 6 mm. The radius of the wear track was 6 mm. The tests were carried out under a normal load of 2 N and at 450 °C.
To provide the data to determine the integral limits needed for the model, five ball-on-disc tests with different wear laps (10, 20, 30, 40 and 50 laps) were carried out. The wear tracks were examined using an optical microscope. The average width of the wear track in each lap was determined from 12 measurements.

### 3.4 EXPERIMENTAL RESULTS

Figure 3.6 shows the variation of the width of the wear track with wear lap. Over the range of laps between 10 and 50 during the ball-on-disc tests, the width seems to increase linearly with lap, i.e.

\[
a_i = 0.0133 \times i + 0.51792
\]  

(3.9)

where \(a_i\) is the width of the wear track after the \(i\)th lap of wear. The linear relationship has an error of less than 4% over a range from 10 to 50 laps.

![Figure 3.6 Widths of the wear track at different laps of wear.](image)
3.5 DETERMINATION OF THE INTEGRAL PARAMETERS IN THE MODEL

The high-temperature friction tests of aluminium against steel allows the elastic recovery in the rear of the ball to be omitted. Then, Equations 3.5 and 3.6 may be used, i.e., \( a_{i+1} = W_{i+1} \).

According to the geometric relationships as shown in Figure 3.3a and c, the parameters \( \xi_i \) and \( \omega_i \) in these equations can be determined from the widths of the wear track, i.e.,

\[
\omega_i = \cos^{-1} \left( \frac{a_i}{W_{i+1}} \right) = \cos^{-1} \left( \frac{a_i}{a_{i+1}} \right) \quad (3.10)
\]

\[
\xi_i = \cos^{-1} \left( \frac{W_{i+1}/2}{r} \right) = \cos^{-1} \left( \frac{a_{i+1}}{2r} \right) \quad (3.11)
\]

As shown in Figure 3.7, during the integration in the area COD (Figure 3.7a), the upper integral limit of angle \( \beta \) is not a constant, and it moves along the arc CED (Figure 3.7b). In this area, a function \( f_i(\gamma) \) may be employed to describe the relationship between \( \gamma \) and the upper limit of \( \beta \) for the purpose of integration. The function \( f_i(\gamma) \) can be derived from the geometric relationship as shown in Figure 3.7b.

\[
r \sin \left( f_i(\gamma) \right) = a_{i+1} \sin \omega_i / 2 \cos(\gamma) \quad (3.12)
\]

Thus,

\[
f_i(\gamma) = \sin^{-1} \left( \frac{a_{i+1} \sin \omega_i}{2r \cos(\gamma)} \right) = \sin^{-1} \left( \frac{W_{i+1} \sin \omega_i}{2r \cos(\gamma)} \right) \quad (3.13)
\]

Thus, all the integral parameters, i.e. \( \xi_i, \omega_i \) and \( f_i(\gamma) \) needed for the solution of the model (Equation 3.5) can be obtained by using Equations 3.10, 3.11 and 3.13.
3.6 APPLICATION OF THE MODEL

In both Equations 3.5 and 3.6, the horizontal force $F_x$ is composed of four components, of which the first two represent the contribution of the contact pressure to the friction force, namely, the ploughing friction, and the last two components are actually the shear friction. For the vertical force $F_z$, the influence of the friction force is considered in the model (Equation 3.5), but it is neglected in Equation 3.6. As a result, Equation 3.6 is greatly simplified. In the ball-on-disc tests, the vertical force $F_z$, i.e., the normal load was 2 N. The horizontal force $F_x$ applied on the ball is actually the friction force measured during the test. Therefore, the mean contact pressure $P$ and friction stress $f$ in the present model, can be obtained from $F_x$ and $F_z$ after all the integral limits are input into Equation 3.5. The mean contact pressure $p$ in Equation 3.6 can be simply obtained directly from $F_z$. 

Figure 3.7 Geometry relationship between the upper integral limit of $\beta$ and $\gamma$. 

![Diagram](image-url)
3.6.1 Ploughing and shear friction coefficients

Figure 3.8 shows the shear friction coefficient values obtained from the model based on Equation 3.5 at different laps of wear. The differences between the apparent friction coefficients measured during the tests and the shear friction coefficients calculated give the values of the ploughing friction coefficient. It is obvious that the ploughing friction accounts for a very small percentage (about 1 %) of the apparent friction, while the shear friction plays a dominant role in determining the apparent friction. With increasing wear lap, both shear friction and ploughing friction coefficients increase gradually. Figure 3.8 also compares the shear friction coefficients obtained from Equation 3.6. There are indeed differences in the shear friction coefficient values obtained from Equation 3.5 and Equation 3.6, although these are not remarkable.

![Graph showing friction coefficients over laps of sliding]

Figure 3.8 Shear friction coefficients and ploughing friction coefficients calibrated from Equations 3.5 and 3.6.
### 3.6.2 Mean contact pressure

Figure 3.9 shows the variation of the mean contact pressure with the lap of wear over a range between the 10th and 50th laps. It is clear that regardless of the model used (Equation 3.5 or Equation 3.6), the mean contact pressure decreases significantly with increasing wear lap or the distance covered during the ball-on-disc tests. Under the same normal load of 2 N, the mean contact pressure decreased, for example, from 47 MPa at the 10th lap to 19 MPa at the 50th lap. It is also obvious from Figure 3.9 that the differences in the mean pressure values obtained from Equation 3.5 and Equation 3.6 are negligible.

![Figure 3.9 Mean contact pressure determined from Equations 3.5 and 3.6.](image)

### 3.6.3 Comparison between Equation 3.5 and Equation 3.6

From Figures 3.8 and 3.9, it appears that Equations 3.5 and 3.6 give quite similar shear friction coefficient values and mean pressure values. Actually, the horizontal force in both Equations 3.5 and 3.6 can be simplified to be:

\[
F_x = f_{\text{plowing}} + f_{\text{shear}} = f_{\text{plowing}} + \mu_s F_z = F_{\text{plowing}} + \mu_s \mu_s p s_n \tag{3.14}
\]

where \( s_n \) is the normalized area. In principle, the values of \( F_z \) calculated using Equations 3.5 and 3.6 should both be consistent with the normal load applied in the tests, i.e., 2 N. Figure
3.10 shows the normal load values calculated from these two models. The model extended from Tayebi’s model, i.e. Equation 3.5, gives the values in the range of 2.03 to 2.05 N, which is in good agreement with the normal load applied. The error is about 3.5 %, which may stem from the limit of the numerical integration tolerance. However, the normal load values obtained from Equation 3.6 are in the range of 1.82 to 1.87 N. The larger deviations of almost 10% from the normal load applied may be attributed to the underestimation of the normalised contact area \( s_n \) [15]. Therefore, the model (Equation 3.5), although more complex than Equation 3.6, is preferable for the determination of the ploughing friction and shear friction as well as mean pressure during high-temperature ball-on-disc tests.

![Graph showing normal load values](image)

Figure 3.10 Normal load calculated using Equation 3.5 and Equation 3.6.

### 3.7 CONCLUSIONS

A model capable of determining the ploughing friction and shear friction as well as the mean contact pressure during high-temperature ball-on-disc tests was developed on the basis of Tayebi’s model for scratch tests. Considering the ball perfectly rigid and the disc perfectly plastic or elasto-plastic in ball-on-disc tests, the integral limits for the solution of the model could be obtained from the evolving wear track. The forces acting on the ball surface could be reproduced by integration. During the ball-on-disc tests with a steel ball sliding on an
aluminium disc at 450 °C, the ploughing friction accounted for only about 1 % of the apparent friction, although the ploughing friction coefficient tended to increase with increasing wear lap, while the shear friction played a dominant role in determining the apparent friction. The mean contact pressure decreased significantly over a range of wear laps till 50. The model extended from Tayebi’s model for scratch tests gives quite similar values of the shear friction coefficient and the mean pressure values to those from Goddard’s model. However, the former is preferable, as the latter underestimates the normalised contact area.

References


Chapter 4

DETERMINATION OF FRICTION COEFFICIENT FOR THE BEARING CHANNEL OF THE HOT ALUMINIUM EXTRUSION DIE

ABSTRACT

Appropriate specification of the frictional boundary condition for the finite element (FE) simulation of metal-forming processes is of great importance to the trustworthiness of the results. The research reported in this chapter aimed at understanding the interfacial contact between aluminium and steel at elevated temperatures and determining friction coefficients at this material mating. A series of high-temperature ball-on-disc tests were carried out with the AA7475 aluminium alloy as the material of disc and the hardened H11 steel as the material of ball. A mathematical model for high temperature ball-on-disc test developed in the preceding chapter was employed to account for the evolution of the contact interface during ball-on-disc tests. Friction coefficients at different temperatures and over a number of laps were determined. The shear friction stresses and mean contact pressures along with the progress of the tests at 350 – 500 °C were calculated. It was found that the friction coefficients obtained from ball-on-disc tests alone were insufficient to represent the frictional interaction between deforming aluminium and steel at elevated temperatures. The evolution of the contact interface with increasing sliding distance must be taken into consideration and the friction behaviour can be reasonably characterized by using friction stress. In addition, a physically based friction model was developed based on the temperature dependent friction stress results.

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4.1 INTRODUCTION

The contact at the billet/die bearing interface where the extruded product is shaped is of vital importance for the product quality and die life. Preceding research using both the experimental approach and FE simulation [1-3] have shown that, being similar to the machining process [4], a transition from sticking to sliding takes place in the die bearing channel and thus the friction coefficients representing the change in friction mode must be incorporated into FE simulation. In addition, physical simulations, such as ring compression tests at room temperature [5] and block-on-disc tests at elevated temperatures [6] have been performed to characterize the tribological interactions between aluminium and steels. More recently, first attempts have been made to determine the friction coefficients between hot aluminium and steel by means of ball-on-disc tests during which conform contact between the deforming disc and ball is gradually established, which is closer to the actual contact at the die bearing during extrusion [7, 8].

Ball-on-disc test is one of the most commonly used methods for characterizing the tribological properties of the materials at contact. However, the friction coefficients obtained from ball-on-disc tests have not yet been implemented in the FE simulation of aluminium extrusion, due to a lack of the methodology for translating the data from ball-on-disc tests to the extrusion process. In the preceding chapter, a model capable of revealing the features of the contact interface during high-temperature ball-on-disc tests and discriminating between the ploughing friction and shear friction was developed [7]. The objective of the present chapter was to implement this model in practical contact situations between hot aluminium and H11 tool steel during ball-on-disc testing, and the results were expected to be extracted to define the frictional boundary condition for the FE simulation of the aluminium extrusion process.

4.2 MATERIALS AND EXPERIMENTAL PROCEDURE

In the present chapter, the high-strength AA7475 aluminium alloy disc had a thickness of 5 mm and a diameter of 49 mm. The surface of the disc with a hardness value of 53 HRA at room temperature was polished to an average roughness Ra of 33 nm. The H11 hot-work tool steel hardened to a hardness value of 53 HRC at room temperature was selected as the mating material. The ball had a diameter of 5 mm, and an average roughness of 209 nm. The compositions of the materials used in the present chapter are given in Table 4.1.
Table 4.1 Compositions of H11 hot-work steel and AA7475 aluminium alloy (wt.%).

<table>
<thead>
<tr>
<th></th>
<th>C</th>
<th>Cr</th>
<th>Mn</th>
<th>Mo</th>
<th>Si</th>
<th>V</th>
<th>Fe</th>
</tr>
</thead>
<tbody>
<tr>
<td>H11 steel</td>
<td>0.40</td>
<td>5.0</td>
<td>0.30</td>
<td>1.30</td>
<td>1.0</td>
<td>0.50</td>
<td>Balance</td>
</tr>
<tr>
<td>AA7475</td>
<td>0.10</td>
<td>0.12</td>
<td>1.9</td>
<td>0.06</td>
<td>2.4</td>
<td>0.18</td>
<td>5.2</td>
</tr>
</tbody>
</table>

A CSM® high-temperature tribometer with a ball-on-disc rig was used for a series of short-distance high-temperature friction tests in order to reach the solutions of the model developed. The radius of the wear track was 6 mm and the linear speed was 2 mm/s. The tests were carried out under a constant normal load of 6 N at 350, 400, 450 and 500 °C in the ambient atmosphere. Three tests with different wear laps (1, 5 and 10 laps) were carried out at each temperature. The short-distance tests under the relatively high normal load were desired to produce high contact pressures up to 120 MPa, which would resemble the situation in the bearing channel of the extrusion die during lab-scale extrusion experiments where the normal pressure varied from two to six times of the flow stress of the billet material. The friction coefficient was continuously registered during the test. Thereafter, wear tracks were examined using an optical microscope. The average width of the wear track in each lap was determined from 12 measurements.

**4.3 RESULTS AND DISCUSSION**

**4.3.1 Evolution of friction coefficient with sliding distance**

Figure 4.1 shows the evolution of the friction coefficient over a sliding distance of 10 laps at different temperatures. It is of interest to note that friction coefficient increases with the sliding distance. At 500 °C, in particular, the friction coefficient increases even by 50%. The significant variation of the friction coefficient with sliding distance during the ball-on-disc tests was also observed by other researchers [8], and this phenomenon was attributed to the material transfer and back transfer which significantly altered the contact interface topography and changed the real contact area. However, in the present chapter, short distance friction tests
(10 laps of sliding) were performed and the friction coefficients were found to increase steadily with the sliding distance, and thus the explanation [8] may not applicable. At the beginning stage of testing, the contact pressure was very high, and severe plastic deformation and drastic removal of surface material occurred. As shown below, the increase of friction coefficient with sliding distance appears to be accompanied by the increase in apparent contact area during the running-in period. Their correlation appears to be peculiar and occurs in the material mating at high temperatures with involvement of strong adhesion. The evolution of the friction coefficient with sliding distance, as shown in Figure 4.1, leads to the uncertainty as to the exact value to be put into FE simulation. It is therefore necessary to have a model with which the friction coefficient and sliding distance are correlated with each other. In the present chapter, the friction coefficients and residual widths of wear tracks determined during and after the ball-on-disc tests were used as input data of the model, i.e. Equations 3.2, 3.3 and 3.4. The evolutions of the contact area and shear friction stress were obtained.

4.3.1.1 Evolution of wear track width

An average value of the widths of wear tracks was determined from 12 measurements by using an optical microscope. The residual width of the wear tracks can be fitted:

$$W'_i = a \times i^b$$

(4.1)

where $W'_i$ is the width of the wear track after the $i$ th lap of wear, and $a$ and $b$ are constants. Figure 4.2 shows the experimental and fitted results of the widths of wear tracks. As can be seen, the widths of wear tracks increase steadily with increasing sliding distance and testing temperature. This can be explained by Holm and Archard’s equation as given below [9, 10]. During the ball-on-disc tests between aluminium and hardened tool steel at elevated temperatures, it is reasonable to assume the ball material (hardened steel) behaves as a rigid material. Therefore, the amount of wear on the disc surface is proportional to the normal load and sliding distance/laps of sliding and inversely proportional to the surface hardness of the disc material, i.e.

$$v = \frac{kWx}{H}$$

(4.2)

where $v$ is the total volume of wear; $W$ the applied normal load; $x$ the total sliding distance and $H$ the surface hardness of the material. Obviously, the more laps of sliding, the larger
amount of wear, or wider and deeper wear track will be. In addition, the hardness of the disc material decreases significantly with increasing temperature, thus wider wear tracks are formed at higher temperatures, as the experiments show (Figure 4.2).

Figure 4.1 Evolution of the friction coefficient with increasing sliding distance at different temperatures.
4.3.1.2 Evolution of contact area and mean contact pressure

Figure 4.3 shows the evolution of the apparent contact area during ball-on-disc testing. Due to the removal of the surface material, the wear track becomes wider and the contact area indeed increases with the sliding distance. According to Bowden and Tabor’s classical theory of friction [11], the friction coefficient increases with increasing real contact area. In the present research, aluminium was so soft that severe plastic deformation occurred on the contact interface and the real contact area increases as the apparent contact area increases. Therefore, it is the increasing contact area with the laps of sliding that results in the increasing friction coefficient as shown in Figure 4.1.

Figure 4.4 shows the evolution of mean contact pressure on the contact interface. As a result of the increasing contact area, the mean contact pressure decreases with increasing sliding distance.

During the ball-on-disc tests, contact area, mean contact pressure and the friction coefficient varied considerably. Therefore, friction coefficient alone may be insufficient to characterize the friction properties of the mating materials. Friction stress that is the friction force per unit area may be a better option, because it is convenient for the characterization of the friction at the interface involving strong adhesion without specifying the real contact area. It is certainly necessary to understand the evolution of the friction stress further during prolonged pin-on-disc tests.
Figure 4.3 Evolution of the apparent contact area with sliding distance.

Figure 4.4 Evolution of the mean contact pressure with sliding distance.
4.3.2 Evolution of shear friction stress

Figure 4.5 Evolution of the shear friction stress with sliding distance.

Figure 4.5 shows the evolutions of the calculated shear friction stress at different temperatures and over a sliding distance of 10 laps. It is interesting to see that the shear friction stress starts from a relatively low value, and then becomes quite stable at each of the temperatures, while the friction coefficient increases considerably (Figure 4.1). The low shear friction stress at the initial stage may be due to the oxide layer on the disc and ball surfaces, which tends to lower the adhesion between aluminium and steel [12, 13]. After the initial stage of sliding, the oxide layer may be broken up and metal-to-metal contact occurs, leading to the increases in friction stress. In addition, the severe plastic deformation on the surface material may generate a considerable work-hardening effect [11, 14], which may also lead to the rise of shear friction stress. As can be seen from Figure 4.5, the shear friction stress differs markedly at different temperatures, and it is therefore necessary to reveal the influence of temperature on the friction stress.

4.3.3 Influence of temperature on the shear friction stress

Figure 4.6 shows the correlation of the mean shear friction stress with temperature. It has been used as a base to develop a novel friction model. From Figure 4.6, it is clear that the friction stress decreases steadily with increasing temperature. This is consistent with the observations
made during the machining process [15] and the results of the friction tests carried out by Bowden and Tabor [11]. It is however inconsistent with the results obtained by other researchers from ball-on-disc tests at elevated temperatures [8]. Therefore, more efforts are needed to explain the contradictory results and to reveal the real effect of temperature on friction.

![Figure 4.6 Mean shear friction stress as a function of temperature.](image)

It is commonly understood that the friction force at elevated temperatures stems from the deformation of surface material and the adhesive bonding of the contact joints [11, 16, 17]. As temperature increases, the friction force due to the deformation of the asperities decreases significantly because of the softening effect of the surface material. On the other hand, the adhesive friction plays an increasing important role in determining the overall friction, because the mating materials tend to be more active at elevated temperatures.

At elevated temperatures, the mating materials tend to be more active to generate adhesive bonding between each other, but an increase in friction coefficient may not necessarily appear, because the overall friction coefficient will be determined by many factors, for example, the adhesive strength and real contact area. As temperature increases, the strength of the adhesive joints decreases significantly and thus the friction coefficient tends to decrease. It is however important to note that during the high-temperature ball-on-disc tests the area of the contact
interface increases with the sliding distance, which may be different from other types of friction tests. This may increase the number of adhesive joints due to strong adhesion and thus the friction force. Further studies are needed on the physical and chemical tribology of the mating surfaces at elevated temperatures to clarify the correlation between the apparent contact area and friction coefficient under this special circumstance. In addition, the increase of the friction coefficient may be partly caused by hard wear debris generated due to oxidation and entrapped at the interface, because the tests were carried out in the ambient atmosphere and as such the influence of a hard aluminium oxide layer on friction coefficient would be inevitable. The combination of these factors complicates the results of ball-on-disc tests. As a consequence, incomparable results might be obtained from the ball-on-disc tests, as compared to other friction testing methods and friction coefficient alone is most likely insufficient to characterize the interface friction property, when ball-on-disc tests are used. It would be necessary to take the evolution of the contact area/normal pressure into account as well and friction stress might be a better option.

4.4 PHYSICALLY-BASED ADHESIVE STRENGTH FRICTION MODEL (ASFM) FOR THE BEARING CHANNEL OF HOT ALUMINIUM EXTRUSION DIE

According to the classic theory of tribology, friction force mainly stems from ploughing (due to hard asperities and trapped wear debris) and adhesive (due to atomic or chemical interactions) forces [11, 14, 17]. At high temperatures, the adhesive friction plays an important role, due to the strong atomic or chemical interactions, especially under high contact pressures [6, 18]. This is mainly due to the following reasons: at elevated temperatures, the atoms in both of the contact materials are highly activated, which aids interdiffusion at the surface layers of the materials and a strong chemical bonding tends to be established. On the other hand, the mating materials and surface oxides are relatively soft and easy to be deformed. In addition, the high contact pressure (several times higher than the flow stress of the work piece material) further aids the severe plastic deformation of the asperities or oxidation scale, and brings the faying material highly close to each other to a distance of atomic level. Therefore, in the bearing channel of hot aluminium extrusion, the strong chemical or diffusion bonding is the dominant friction mechanism. Consequently, the strength of adhesive junctions can be determined by two factors, namely, the rate of atomic interaction and the strength of adhesive joints (Equation 4.3). The interaction rate at atomic level
increases with increasing temperature, which can be modelled by using an Arrhenius type equation [19] (Equation 4.4). Meanwhile, the bonding strength decreases with increasing temperature, due to the decrease of material strength with increasing temperatures, and the bonding strength drops to zero at the melting temperature of AA7475. In the present study, in order to determine the constants of the model, short distance ball-on-disc test results were used as the friction stress at different temperatures. Table 4.2 lists the determined material constants. As can be seen from Figure 4.7, a good agreement between the model and experimental results was obtained.

\[ f(T) = f_0 D \left( 1 - \frac{T}{T_m} \right)^\eta \]  

(4.3)

\[ D = D_0 \exp \left( - \frac{Q_D}{RT} \right) \]  

(4.4)

where \( f(T) \) is temperature dependant bonding strength or friction stress; \( f_0 \) is the initial adhesive strength at room temperature (300 \( K \)); \( D \) is the inter-diffusion coefficient; \( T \) is the mean contact temperature (\( K \)); \( T_m \) is the melting temperature of the work piece material (\( K \)); \( Q_D \) is the activation energy; \( R \) is the universal gas constant; \( D_0 \) and \( \eta \) are constants;
Figure 4.7 Evolution of friction stress between AA7475 and H11 steel at different temperatures.

Table 4.2 Material constants of the ASFM for hot AA7475 and H11 steel.

<table>
<thead>
<tr>
<th>$f_0$ (MPa)</th>
<th>$T_m$ (K)</th>
<th>$Q_0$ (J/mol)</th>
<th>$R$ (J/K/mol)</th>
<th>$D_0$</th>
<th>$\eta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>378</td>
<td>811</td>
<td>2400</td>
<td>8.314</td>
<td>3.62</td>
<td>0.7</td>
</tr>
</tbody>
</table>

As can be seen from Figure 4.7, the bonding strength is determined by the combined effects of atomic interaction and material strength. Under the high temperature condition, the bonding strength decreases with increasing temperature, because the drop of material strength plays a dominant role, although the atoms are highly activated and atomic interactions are more intensive. Therefore a decreased adhesive strength with the increasing temperature can be observed, which is consistent with other research results [11, 15].

4.5 CONCLUSIONS

A series of ball-on-disc tests were carried out at different temperatures. The friction coefficients were found to increase with increasing sliding distance. The individual friction coefficient data could not be utilized directly for FE simulation of the aluminium extrusion process. A model for ball-on-disc tests, developed in the preceding chapter, was used to reveal the contact between aluminium and tool steel at elevated temperatures. The calculated shear friction stress and mean contact pressure showed that, during the running-in period, the shear friction stress was quite stable, while the friction coefficient increased with increasing sliding distance significantly. Therefore, a fundamental understanding of the evolution of the contact interface must be gained, before the results of ball-on-disc tests can be used as the frictional boundary conditions for FE simulation.

References


Chapter 5
DOUBLE ACTION EXTRUSION - A NOVEL EXTRUSION PROCESS FOR FRICTION CHARACTERIZATION AT THE BILLET DIE BEARING INTERFACE

ABSTRACT

A novel extrusion testing method, double action extrusion (DAE), to highlight the effect of friction at the die bearing in aluminium extrusion was developed. It was found that the lengths of the extrudates and extrusion force were indeed sensitive to the die bearing length and thus to the friction. FEM simulations of DAEs were carried out to evaluate two commonly used friction models. It was found that, when the extrusion dies with a 15’choke angle were used, full-sticking friction represents the experimental results the best. In addition, the physically-based friction model (adhesive strength friction model: ASFM) developed in Chapter 3 was implemented into the simulations of DAEs for model verification. Good agreements between the FE predictions and experiments were obtained, indicating that ball-on-disc test is an effective way of characterizing the friction for the bearing channel of extrusion dies. For a further understanding of the DAE, a theoretical model was developed, and a good agreement between the modelling results and experiments was obtained. The theoretical modelling results revealed that the length difference of the extrudates was caused by the combined effects of friction and material rate dependence at elevated temperatures.

5.1 INTRODUCTION

For hot aluminium extrusion, the bearing channel area is of great importance, as it determines the quality of the final products, but the assignment of friction coefficients within the bearing channel remains a pure guess work, because the bearing area is relatively small compared to the container area; therefore the change of friction condition within the bearing channel can hardly be observed from the variation of the total extrusion load or the slope of the load-displacement curve. In the past few years, the friction coefficient within the bearing channel was studied by using extrusion tests [1-3]. Being similar to the different contact zones observed from the metal cutting process [4-7], the presence of two or three different zones were observed on the bearing surface of the extrusion dies, namely a full sticking zone close to the die entrance and a sliding zone close to the die exit; when extruding at a lower die temperature, a transition zone between the two zones could be observed. In some research work [1-3], the lengths of sliding and sticking zones were considered friction sensitive, and friction coefficients were thus obtained by measuring the lengths of sliding and sticking zones. To avoid complications, the contact within the bearing channel has been simplified to be of the sliding type and a constant friction factor ranging from 0.3 to 0.6 was used [8-11]. In the present chapter, a novel simulative test, double action extrusion (DAE), highlighting the friction in the bearing channel was developed [12], the tests were conducted on a Gleeble 3800 thermo-mechanical simulator, in which an aluminium billet was pressed against two dies with different bearing lengths (2 mm and 6 mm). FEM simulations of DAE were carried out to evaluate the Shear and Coulomb friction models over a wide range of friction factors/coefficients from 0.2 to 1.

Although the DAE test has been found to be a friction sensitive process, the fundamental mechanisms of this novel simulative test remain unclear. One of the main objectives of the present chapter is to address the above issue by theoretical analysis, and to evaluate the different friction models by using DAE tests and FEM simulations.

5.2 EXPERIMENTAL AND SIMULATION DETAILS

Figure 5.1 shows the schematic of DAE and the experimental setup. During the experiments on a Gleeble thermomechanical simulator, an aluminium billet was pressed against two extrusion dies with different bearing lengths (8 and 2 mm; 6 and 2 mm) simultaneously. A 15° chock angle was assigned to the bearing of the dies in order to enhance the effect of the friction on the DAE results. It was expected that the differentiated lengths of the extrudates
passing these two bearing channels as well as the extrusion force could be used to characterize the friction at the die bearing.

Aluminium alloy 7475 was used as the billet material. The extrusion tooling, i.e. two extrusion dies with different bearing lengths and one container, was made of H11 hot-work tool steel. Their physical properties are listed in Table 5.1, which were also used for the parameters of FE analysis. Figure 5.2 shows the aluminium billet and extrusion tooling used in the DAE experiments. The dimensions of the billet, extrusion dies and container, together with the main process parameters, are listed in Table 5.2. The DAE experiments were carried out at 350, 400 and 450 °C, which are the typical temperatures for hot aluminium extrusion. The speed of the moving anvils was 1 mm/s.

![Figure 5.1](image1.png)  
(a) Schematic and (b) experimental setup of double action extrusion (DAE).

![Figure 5.2](image2.png)  
Figure 5.2 Aluminium alloy billet and steel tooling used in the DAE experiments.
The materials of the aluminium billet shows thermo-viscoplastic behaviour as the forming temperature was above 350 °C, while the material of extrusion tooling was considered as thermo-rigid. The elastic behaviour of the materials was neglected. The flow stress/strain data of the AA7475 alloy were determined from hot compression tests using a Gleeble 3800 thermo-mechanical simulator, after flow stresses at high strain rates were corrected for deformational heating [13]. Flow stress – strain data over a temperature range of 250 – 500 °C and a strain rate range of 0.01 – 10 s\(^{-1}\) were imported into DEFORM-3D as the material model of the AA7475 alloy. The flow stress data was also used for the mechanical behaviour modelling of AA7475 alloy in section 5.3.

Table 5.1 Physical properties of the AA7475 work piece and H11 tooling.

<table>
<thead>
<tr>
<th>Property</th>
<th>AA 7475</th>
<th>H11 tool steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat capacity (J/kg/K)</td>
<td>2.43369</td>
<td>3.2 at 315 °C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.5 at 540 °C</td>
</tr>
<tr>
<td>Thermal conductivity (W/m/K)</td>
<td>180.181</td>
<td>24</td>
</tr>
<tr>
<td>Heat transfer coefficient between tooling and</td>
<td>11</td>
<td>11</td>
</tr>
<tr>
<td>billet (W/K/m(^2))</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat transfer coefficient between tooling/billet and air (W/K/m(^2))</td>
<td>0.02</td>
<td>0.02</td>
</tr>
<tr>
<td>Emissivity</td>
<td>0.7</td>
<td>0.7</td>
</tr>
</tbody>
</table>

Table 5.2 Dimensions of the billet and DAE tooling as well as the main process parameters.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Billet length (mm)</td>
<td>15</td>
</tr>
<tr>
<td>Billet diameter (mm)</td>
<td>9.8</td>
</tr>
<tr>
<td>Container inside diameter (mm)</td>
<td>10</td>
</tr>
<tr>
<td>Container outside diameter (mm)</td>
<td>20</td>
</tr>
<tr>
<td>Die bearing length (mm)</td>
<td>2 and 8; 2 and 6</td>
</tr>
<tr>
<td>--------------------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>Choke angle (min)</td>
<td>15</td>
</tr>
<tr>
<td>Reduction ratio</td>
<td>11</td>
</tr>
<tr>
<td>Initial billet temperature (°C)</td>
<td>350, 400 and 450</td>
</tr>
<tr>
<td>Initial tooling temperature (°C)</td>
<td>350, 400 and 450</td>
</tr>
<tr>
<td>Anvil speed (mm/s)</td>
<td>1</td>
</tr>
</tbody>
</table>

Figure 5.3 shows the FE model for the DAE. All the objects in the FEM model were meshed with tetrahedral elements. In order to enhance the efficiency of the FEM simulations and the accuracy in the areas of particular interest, a number of mesh windows with an increased element density were applied around the die orifices to generate local finer elements. A relative interference depth of 0.2 was defined to trigger the remeshing procedure.

The friction at the billet-container and billet-die face interfaces was considered to be of shear type and a friction factor $m = 1$ used. Friction windows were applied at the work piece-die bearing interface to make the friction boundary condition adjustable. Both the Shear friction model and Coulomb friction model with friction factors/coefficients ranging from 0.2 to 1 were used in the FEM simulations to evaluate these friction models and determine the friction factor/coefficient. The ASFM was implemented into DEFORM 3D V6.1 via user defined subroutine. In DEFORM 3D version 6.1, to avoid the overestimation of the friction stress, the value of the friction stress calculated from the Coulomb friction model was compared with the shear flow stress of the work piece material at each iteration step and automatically changed to the shear flow stress, if the calculated friction stress was larger than the shear flow stress. In this way, the Coulomb friction model could be used in the FEM simulation of the aluminium extrusion process at elevated temperatures.
5.3 THEORETICAL MODELLING OF DOUBLE ACTION EXTRUSION

5.3.1 Theoretical background

Figure 5.4 shows the schematic of double action extrusion. The process can be considered as a combination of two indirect extrusions. The work piece is divided into 5 zones according to different deformation modes, namely, bearing area zone (zone 1 and zone 5), severe deforming zone (zone 2 and zone 4) and rigid zone (zone 3). Plastic deformation occurred in the bearing area zone and severe deforming zone due to the compression effect of the extrusion dies and dead metal zone. During the double action extrusion, zone 3, moved together with the container, thus it can be considered as a rigid zone.

Figure 5.3 FE model for the DAE.
Figure 5.4 Theoretical analysis of double action extrusion.

(a) Zone 1 and zone 5: bearing area

An elemental material of width $dx$ was analysed (Figure 5.4 a). The forces acting on this elemental material can be expressed as:

$$(\sigma_{sl} + d\sigma_{sl})\pi\left(\frac{D + dD}{2}\right)^2 = \sigma_{sl}\pi\left(\frac{D}{2}\right)^2 + P_i\pi D\frac{dx}{\cos\alpha}\sin\alpha + \tau_i\pi D\frac{dx}{\cos\alpha}$$

(5.3.1-1)

According to the geometric relationship, $dx = \frac{dD}{2}\cot\alpha$. The Shear friction model was used, i.e. $\tau \approx \frac{mY}{2}$. Thus,

$$\sigma_{sl} = Y_i\left(2 + m\cot\alpha\right)\ln(D) + C_i \ (i=1 \text{ or } 5)$$

(5.3.1-2)
(b) Zone 2 and zone 4: deforming zone

The dead metal zone builds up in the corners of the dies, which acts as a conical die surface, therefore it can be considered as rigid during extrusion. An elemental material of width \(dx\) was analysed (Figure 5.4 b). The forces acting on the elemental material can be expressed as:

\[
(\sigma_{x2} + d\sigma_{x2})\pi \left(\frac{D + dD}{2}\right)^2 = \sigma_{x2}\pi \left(\frac{D}{2}\right)^2 + P_2\pi D\frac{dx}{\cos\beta} + \tau_2\pi D\frac{dx}{\cos\beta} \sin\beta + \tau_2\pi D\frac{dx}{\cos\beta} \cos\beta
\]

(5.3.1-3)

According to the geometric relationship, \(dx = \frac{dD}{2} \cot\beta\) and during extrusion, shear deformation occurred at the zone 2/dead metal zone interface, and thus \(\tau \approx \frac{Y}{2}\). Therefore,

\[
\sigma_{xj} = Y_j \left(2 + \cot\beta\right)\ln(D) + C_j \ (j = 2 \text{ or } 4)
\]

(5.3.1-4)

(c) Zone 3: rigid zone

The work piece material moved together with the container and no plastic deformation occurred, therefore the axial stresses acting on the zone 2/zone 3 and zone 3/zone 4 interfaces should maintain a dynamic balance throughout the extrusion process.

In the present research, \(D_5 = 10mm\) and \(D_0 = 3mm\), hence the extrusion ratio was 11 approximately. In addition, a constant ram speed \((v = 1mm/s)\) was employed during the tests, thus the extrusion speeds in the two bearing channels should always follow the rule:

\[
v_s(t) - v_i(t) = 11mm/s
\]

(5.3.1-5)

where \(t\) is the extrusion time. \(v_s(t)\) and \(v_i(t)\) are the extrusion speeds at the die entrances of zone 5 and zone 1, respectively. At the very initial stage of extrusion \((t=0)\), we can safely assume the initial extrusion speeds as : \(v_s(0) = -v_i(0) = 5.5mm/s\).

The lengths of extruded profiles can be expressed as:

\[
L_{xS} = \int_0^t v_s(t)dt
\]

(5.3.1-6)
\[ L_{x1} = \int_{0}^{t} v_1(t) \, dt \quad (5.3.1-7) \]

### 5.3.2 Integral constants determination

When the length of the extruded profile in zone 1 is \( L_{x1} \), (0 \( \leq \) \( L_{x1} \leq 8 \), and for the boundary condition determination: \( L_{x1} = 8 \) when \( L_{x1} \geq 8 \), because the axial stress drops to zero as soon as the profile is extruded out of the die) the diameter of the end of the profile in zone 1 is: 
\[
D = D_1 - 2L_{x1} \tan \alpha ,
\]
and the boundary condition is: \( \sigma_{x1} = 0 \), when \( D = D_1 - 2L_{x1} \tan \alpha \).

Consequently, \( C_1 \) can be determined and the axial stress can be expressed as:
\[
\sigma_{x1} = Y_1 (2 + m \cot \alpha) \ln \left( \frac{D}{D_1 - 2L_{x1} \tan \alpha} \right) \quad (5.3.2-1)
\]

Identical situation happens in zone 5, when the extruded profile length is \( L_{x5} \) (0 \( \leq \) \( L_{x5} \leq 2 \), and for the boundary condition determination: \( L_{x5} = 2 \) when \( L_{x5} \geq 2 \), thus \( C_5 \) can be determined and the axial stress is:
\[
\sigma_{x5} = Y_5 (2 + m \cot \alpha) \ln \left( \frac{D}{D_1 - 2L_{x5} \tan \alpha} \right) \quad (5.3.2-2)
\]

In zone 2, \( \sigma_{x2} = \sigma_{x1} \), when \( x = -L_2 - L_3 - L_4 \) and \( D = D_1 \), hence \( C_2 \) can be determined:
\[
\sigma_{x2} = Y_2 (2 + \cot \beta) \ln \left( \frac{D}{D_1} \right) + Y_5 (2 + m \cot \alpha) \ln \left( \frac{D_1}{D_1 - 2L_{x1} \tan \alpha} \right) \quad (5.3.2-3)
\]

Similarly, in zone 4, axial stress can be expressed as:
\[
\sigma_{x4} = Y_4 (2 + \cot \gamma) \ln \left( \frac{D}{D_1} \right) + Y_5 (2 + m \cot \alpha) \ln \left( \frac{D_3}{D_3 - 2L_{x5} \tan \alpha} \right) \quad (5.3.2-4)
\]

In zone 3, a dynamic balance is established throughout the whole process of extrusion, which is,
\[
\sigma_{x2 \rho = D_2} = \sigma_{x4 \rho = D_2} \quad (5.3.2-5)
\]
5.3.3 Material model for AA7475

The hyperbolic sine function proposed by Sellars and Tegart [14] and modified by Sheppard and Wright [15] has been widely used to model the evolution of steady-state flow stress of aluminium alloys. Therefore, Sellars and Tegart’s model was employed:

\[ Y(\dot{\varepsilon}, T) = \frac{1}{\alpha} \sinh^{-1}\left(\frac{\dot{\varepsilon}}{A} \exp\left(\frac{Q}{RT}\right)\right)^{1/n} \]  \hspace{1cm} (5.3.3-1)

where \(Q\) is the activation energy, \(R\) is the universal gas constant and \(T\) is the temperature (K). Table 5.3 shows the values of the material constants. Figure 5.5 shows the comparison between the calculated and experimental stress-strain curves at different testing temperatures and strain rates, in which good agreements were obtained.

![Figure 5.5 Comparison of computed (solid curves) and experimental (symbols) stress-strain relationship at different strain rates and testing temperatures.](image)

Table 5.3 Material constants for AA7475.

<table>
<thead>
<tr>
<th>(1/\alpha) (MPa)</th>
<th>(A) (s(^{-1}))</th>
<th>(n)</th>
<th>(Q) (J/mol)</th>
<th>(R) (J/K/mol)</th>
</tr>
</thead>
<tbody>
<tr>
<td>87.72</td>
<td>1027094727</td>
<td>5.41</td>
<td>129400</td>
<td>8.314</td>
</tr>
</tbody>
</table>
### 5.3.4 Strain rate determination

Strain rate can be determined by following the method introduced in [16]:

\[
\dot{\epsilon} = \frac{d\epsilon}{dt} = -\frac{vD_i^2}{D^3} \frac{dD}{dx} \quad (i=1 \text{ or } 3)
\]  

(5.3.4-1)

In zone 1: \(\frac{dD}{dx} = 2 \tan \alpha \).

Therefore,

\[
|\dot{\epsilon}_1| = 2 \frac{v_i(t)D_i^2}{D^3} \tan \alpha
\]

(5.3.4-2)

In zone 2: \(\frac{dD}{dx} = 2 \tan \beta \).

Therefore,

\[
|\dot{\epsilon}_2| = 2 \frac{v_i(t)D_i^2}{D^3} \tan \beta
\]

(5.3.4-3)

In zone 5: \(\frac{dD}{dx} = -2 \tan \alpha \).

Therefore,

\[
|\dot{\epsilon}_5| = 2 \frac{v_5(t)D_5^2}{D^3} \tan \alpha
\]

(5.3.4-4)

In zone 4: \(\frac{dD}{dx} = -2 \tan \gamma \).

Therefore,

\[
|\dot{\epsilon}_4| = 2 \frac{v_4(t)D_4^2}{D^3} \tan \gamma
\]

(5.3.4-5)

During the DAE, two indirect extrusions occurred simultaneously, thus it is reasonable to assume that \(\beta = \gamma\). The DAE tests were carried out on a Gleeble thermo-mechanical simulator, and the aluminium billet temperature was accurately controlled by a feedback control system, therefore the temperature of the work piece was assumed as constant during the tests.
5.3.5 Governing equations

The governing equations were solved using a FORTRAN program with full sticking condition \((m=1)\) along the container wall and the extrusion die assumed. The instantaneous extrusion speeds \(v_1(t)\) and \(v_5(t)\) were determined from (5.3.5-1) to (5.3.5-6), and a time increment of 0.0001 s was used to update the lengths of the extrudates by using (5.3.5-7) and (5.3.5-8).

\[
\sigma_{x2|\partial=\partial_2} = \sigma_{x4|\partial=\partial_2} \quad (5.3.5-1)
\]

\[
\sigma_{x2} = Y_2 (2 + \cot \beta) \ln \left( \frac{D}{D_1} \right) + Y_5 (2 + m \cot \alpha) \ln \left( \frac{D_1}{D_1 - 2L_{x1} \tan \alpha} \right) \quad (5.3.5-2)
\]

\[
\sigma_{x4} = Y_4 (2 + \cot \gamma) \ln \left( \frac{D}{D_4} \right) + Y_5 (2 + m \cot \alpha) \ln \left( \frac{D_3}{D_3 - 2L_{x5} \tan \alpha} \right) \quad (5.3.5-3)
\]

\[
\dot{\varepsilon} = \frac{d\varepsilon}{dt} \quad (5.3.5-4)
\]

\[
Y(\dot{\varepsilon}, T) = \frac{1}{\alpha} \sinh^{-1} \left( \dot{\varepsilon} \exp \left( \frac{Q}{RT} \right) \right)^{\frac{1}{n}} \quad (5.3.5-5)
\]

\[
v_5(t) - v_1(t) = 11 \text{mm/s} \quad (5.3.5-6)
\]

\[
L_{x1} = \int_0^t v_1(t) dt \quad (5.3.5-7)
\]

\[
L_{x5} = \int_0^t v_5(t) dt \quad (5.3.5-8)
\]

5.4 RESULTS AND MODEL VERIFICATION

5.4.1 Typical DAE results

Figure 5.6 shows a typical result of DAE. During the DAE tests, the aluminium billet was pressed against two extrusion dies and extrusion in the indirect mode took place simultaneously through these two dies. The friction force for the extrudate to flow through the die with a bearing length of 8 mm was greater than that through the die with a bearing length
of 2 mm, as soon as the extrudate flew through the die with a shorter bearing length. As a result, the lengths of the extrudates were significantly different due to the sensitivity of extrusion speed to the friction force at the die bearing.

![Figure 5.6 A typical DAE result with different extrudate lengths.](image)

Figure 5.6 A typical DAE result with different extrudate lengths.

Figure 5.7 shows the measured extrusion forces, when DAE was performed at 350, 400 and 450 °C. As can be seen in the figure, the extrusion forces decrease markedly with increasing temperature, mainly due to decreasing flow stress of the work piece material with rising temperature. The extrusion forces at these temperatures show a similar trend, i.e. a small plateau at the very early stage, followed by a sharp increase in extrusion force and then a gentle decrease as the process proceeded further. The small plateau corresponds to the initiation of extrusion toward both of the dies (upsetting), and the sharp force increase corresponds to breakthrough. In DAE, there is no relative movement and hence no dynamic friction between the billet and container and therefore the extrusion force in the steady state reflects the dynamic balance of the billet material (work hardening and dynamic recovery or dynamic recrystallization) which is governed by temperature and influenced by the temperature evolution during DAE.
Figure 5.7 Extrusion forces measured during the DAE experiments at different temperatures.

5.4.2 Steady-state extrusion force

Figure 5.8 shows the steady-state extrusion forces at different extrusion temperatures. The extrusion force decreases with increasing temperature as a result of material softening at higher temperatures. Of more interest is the comparison in the experimentally measured extrusion forces and those predicted on the basis of the Shear and Coulomb friction models at different friction factors/coefficients. It can be seen that both of the models show a similar trend as the experimental results in terms of the effect of temperature on the extrusion force. However, the extrusion forces predicted vary over a wide range, as a result of different friction conditions assigned. From Figure 5.8, it appears that the shear friction model at m = 1 yields the extrusion forces the closest to the experimental measurements, although the predicted value is 12% higher than the experimental one at 450 °C. The predicted steady-state extrusion force decreases with decreasing friction factors, and the FEM results with a friction factor of 0.6 is presented in Figure 5.8, which shows a deviation of over 16% from the experimental data, suggesting that the widely used constant friction factor ranging between 0.3 and 0.6 might be too low in terms of extrusion force prediction. It can be seen from Figure 5.8 that the prediction from the ASFM shows the same trend as the experimental results in terms of the temperature effect on the extrusion force and a fairly good agreement between the FE predictions and experimental results was achieved.
A deviation of 12% or smaller can be observed when Coulomb friction model at \( \mu = 1 \) used, which is still acceptable. At the other friction conditions, however, the predicted extrusion forces are all much lower than the experimental results. It is clear that the steady-state extrusion force is indeed highly sensitive to the friction at the die bearing in DAE. In the present DAE tests, the full sticking friction describes the friction boundary condition at the die bearing the best.

5.4.3 Exudate lengths and validation of theoretical model

Figures 5.9, 5.10 and 5.11 show the comparison of the DAE experiments, FE simulations and the theoretical model, in terms of the extrudate lengths. The relative lengths of the extrudates are not very sensitive to temperature. As soon as the extrudate is out of the 2 mm long die bearing, the extrudate lengths start to diverge. In other words, when the extrusion process proceeds, the difference in extrudate length becomes greater. Obviously, the friction force at the bearing channel plays a decisive role in the DAE process, and DAE is indeed sensitive to friction in terms of extrudate lengths. On the other hand, AA7475 shows rate dependence at elevated temperatures and this was the main factor to diminish the extrudate length difference. During the DAE testing, since the extrusion speed is higher in the 2 mm bearing die, the higher strain rate increases the flow stress of the material around the die orifice. Therefore a
higher force is required to deform the material around this area. On the other hand, the extrusion speed in the 8 mm bearing die was lower compared to that in the 2 mm bearing die, consequently, the material around the 8 mm bearing die was deformed at a relatively lower strain rate, thus the material was soft and easy to deform. Therefore, the combined effects of friction and material rate dependence led to a dynamic balance during DAEs.

As can be seen from Figures 5.9, 5.10 and 5.11, the FEM predictions of the extrudate lengths based on the ASFM and the Shear and Coulomb friction models over a wide range of friction factors/coefficients are presented together with the experimental results. At these three extrusion temperatures, the Coulomb friction model at $\mu = 1$ gives the most accurate predictions of the extrudate lengths. The predicted lengths of the extrudates are not very sensitive to the friction coefficients since the shear flow stress of the work piece is low compared to the calculated friction stress. Also, the novel friction model (ASFM) leads to highly accurate results in terms of extrudate lengths. The predictions of the shear friction model at $m = 1$ are quite accurate as well, although small deviations from the experimental measurements can be found at high temperatures. These deviations may be partly caused by the errors of numerical iterations. Nevertheless, the deviations from the experimental data increase markedly when lower friction factors are selected and the results of the present research clearly indicate that a friction factor in the range from 0.3 to 0.6 often assumed at the die bearing during aluminium extrusion may be too low and the friction in the die bearing channel may be better represented by using the sticking boundary condition.

The extrudate lengths predicted by the theoretical model have shown great agreements with the experimental data and the fundamental understanding of this novel process was obtained. During DAE, the material flow is controlled by two main factors, namely, the friction force within the die bearings, and the deforming force of the work piece material. At the initial stage of extrusion, the work piece is extruded at the same extrusion speed in both of the dies. With the increasing length of the extrudates, the friction forces increase at the same rate, due to the increasing contact area. As soon as the profile has been fully extruded out of the 2 mm bearing die, the friction within the 2 mm bearing die reaches its maximum value and will not be further increased; on the other hand, the friction force increases in the 8 mm bearing die as the extrudate length increases, therefore the extrusion speed in the 8 mm bearing die is slowed down. Since the extrusion ratio always remains constant (Ratio $\approx 11$) during the DAEs, the extrusion speed in the 2 mm bearing die must be increased, which enhances the strength of the work piece material around the 2 mm bearing die, and the increased deforming force balances
the extra friction force from the 8 mm bearing die. In the present study, at steady stage of DAEs, the extrusion speed in the 2 mm bearing die increased to almost 11 mm/s, while a very low extrusion speed, which was very close to zero, occurred in the 8 mm bearing die. The dynamic balance between the friction forces and deforming forces is maintained throughout the DAE process. In the present theory model, a shear type friction model with a friction factor of 1 (m=1) was used, which confirms that during DAEs, the friction feature in the bearing channel with a 15° choke angle can be accurately represented by full sticking boundary condition.

The DAE results of 2 and 6 mm bearing dies [12] are shown in Figures 5.9, 5.10 and 5.11 for a comparison with the results of DAE tests with 2 and 8 mm bearing dies. The reason for choosing a longer die bearing was to highlight the influence of friction and a more remarkable length difference was expected. As described in Abtahi’s work, sliding contact occurred following the sticking contact near the die entrance, and the length of the sticking zone was generally more than 3-4 mm [2], suggesting that the greater the bearing length is, the more obvious the sliding effect should be. As can be seen from Figures 5.9, 5.10 and 5.11, no obvious difference can be observed between the results of 2 and 8 mm bearing dies and those of 2 and 6 mm bearing dies. One of the main reasons for this phenomenon is the extremely low friction stress generated from the die exit area of the 8 mm bearing die. At the final stage of extrusion, especially when the profile length in the 8 mm bearing die is over 6 mm, the extrusion speed (strain rate) within this die is very slow, which significantly decreases the (shear) flow stress of the work piece material, consequently, the friction stress generated from the die exit area of the 8 mm bearing die is decreased correspondingly, although full sticking between the work piece and die occurs. Therefore, the increase of bearing length from 6 mm to 8 mm does not make significant improvement to the friction sensitivity of DAEs and the combination of 2 and 6 mm bearing dies probably shows better friction sensitivity.
Figure 5.9 Comparison of DAE experiments, FEM simulations and theoretical model in terms of extrudate lengths from DAE tests at 350 °C.
Figure 5.10 Comparison of DAE experiments, FEM simulations and theoretical model in terms of extrudate lengths from DAE tests at 400 °C.
Figure 5.11 Comparison of DAE experiments, FEM simulations and theoretical model in terms of extrudate lengths from DAE tests at 450 °C.
5.5 CONCLUSIONS

A novel extrusion testing method, double action extrusion (DAE), to highlight the effect of friction at the die bearing in aluminium extrusion was developed. The DAE experiments, FEM simulations and theoretical modelling of DAE were carried out. It was confirmed that the measurable parameters of the DAE experiments, i.e. extrudate lengths and extrusion force, were both sensitive to the friction at the die bearing. Comparisons between the FEM simulation and DAE experiments were carried out, and the results indicated that for an extrusion die with a 15° choke angle, the commonly assumed friction factor values over a range of 0.3 to 0.6 in the shear friction model at the billet-die bearing interface might be inappropriate and the full sticking condition would represent the interfacial contact better. In terms of the extrudate lengths, the Coulomb model at $\mu = 1$ yielded the results the closest to the experimental measurements. In terms of the steady-state extrusion force, the shear friction model at $m = 1$ was in agreement with the experiments reasonably well.

The physically based friction model (ASFM) was implemented into the FE simulation of hot aluminium extrusion process. Good agreements between the FE simulations and experiments were achieved, in terms of extrudate length and extrusion force, indicating that the ASFM obtained from ball-on-disc tests can represent the friction conditions in the bearing channel of the hot aluminium extrusion dies.

A theoretical model for the novel DAE process was developed and good agreements with experimental data were achieved in terms of the extrudate lengths. It was found that both the friction force in the bearing channels and the rate dependence of work piece material significantly influenced the material flow during DAE tests.

References


Chapter 6
CONCLUSIONS, DISCUSSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

In this thesis, the assignment of friction boundary conditions for hot aluminium extrusion process was studied. The success of friction modelling for the bearing channel of hot aluminium extrusion die relies on three innovations: (1) A mathematic model for high temperature ball-on-disc tests was developed, and this model can be used to determine the friction coefficient for hot aluminium extrusion process. (2) A novel physically based friction model was developed based on the ball-on-disc test results. (3) A novel extrusion process, double action extrusion (DAE), to highlight the friction in the bearing channel of extrusion dies was developed and the modelling of DAE was conducted.

In Chapter 3, a model capable of determining the ploughing friction and shear friction as well as the mean contact pressure during high-temperature ball-on-disc tests was developed on the basis of Tayebi’s model for scratch tests. Considering the ball perfectly rigid and the disc perfectly plastic or elasto-plastic in ball-on-disc tests, the integral limits for the solution of the model could be obtained from the evolving wear track. The forces acting on the ball surface could be reproduced by integration. During the ball-on-disc tests with a steel ball sliding on an aluminium disc at 450 °C, the ploughing friction accounted for only about 1% of the apparent friction, although the ploughing friction coefficient tended to increase with increasing wear lap, while the shear friction played a dominant role in determining the apparent friction. The mean contact pressure decreased significantly over a range of wear laps till 50. The model extended from Tayebi’s model for scratch tests gives quite similar values of the shear friction coefficient and the mean pressure values to those from Goddard’s model. However, the former is preferable, as the latter underestimates the normalised contact area.

In Chapter 4, a series of ball-on-disc tests were carried out at different temperatures. The friction coefficients were found to increase with increasing sliding distance. The individual
friction coefficient data could not be utilized directly for FE simulation of the aluminium extrusion process. A model for ball-on-disc tests, developed in Chapter 2, was used to reveal the contact between aluminium and tool steel at elevated temperatures. The calculated shear friction stress and mean contact pressure showed that, during the running-in period, the shear friction stress was quite stable, while the friction coefficient increased with increasing sliding distance significantly. Therefore, a fundamental understanding of the evolution of the contact interface must be gained, before the results of ball-on-disc tests can be used as the frictional boundary conditions for FE simulation.

In Chapter 5, a novel extrusion process, double action extrusion (DAE) was developed. The DAE experiments, FEM simulations and theoretical modelling of DAE were carried out. It was confirmed that the measurable parameters of the DAE experiments, i.e. extrudate lengths and extrusion force, were both sensitive to the friction at the die bearing. Comparisons between the FEM simulation and DAE experiments were carried out, and the results indicated that the commonly assumed friction factor values over a range of 0.3 to 0.6 in the shear friction model at the billet-die bearing interface might be inappropriate and the full sticking condition would represent the interfacial contact better. In terms of the extrudate lengths, the Coulomb model at $\mu = 1$ yielded the results the closest to the experimental measurements. In terms of the steady-state extrusion force, the shear friction model at $m = 1$ was in agreement with the experiments reasonably well. The novel physically based friction model (ASFM) was implemented into the FE simulation of hot aluminium extrusion process. A good agreement between the FE simulations and experiments has been achieved, in terms of extrudate length and extrusion force.

A theoretical model for the novel DAE process was developed and good agreements with experimental data were achieved in terms of the extrudate lengths. It was found that both the friction force in the bearing channels and the rate dependence of work piece material significantly influenced the material flow during DAE tests. It was confirmed that the full sticking condition is able to represent the friction condition on the work piece-die interface.
6.2 DISCUSSIONS

6.2.1 Friction characterization for the bearing channel of hot aluminium extrusion die by using ball on disc tests

During the friction tests, the large variety of contact conditions, such as contact temperature, pressure, sliding distance, sliding velocity and oxidation scale should be considered very carefully [1], because these factors may influence the friction coefficients considerably. In general, it is very unlikely to emulate all the contact conditions or reflect all the tribological conditions by using one single friction testing technique, because one testing technique is only able to reflect one specific or a few tribological conditions, i.e. the tribological conditions of a particular region of the work piece / tooling interface. Therefore, a combination of different testing methods should be used, for instance the combination of ring compression tests, extrusion friction tests and short sliding distance ball-on-disc tests.

Ring compression test is one of the most widely used friction testing techniques for the friction characterization of bulk metal forming process. During ring compression tests, the contact pressure on the work piece / tooling interface is roughly the same to the flow stress of the work piece material and is, normally, not adjustable, unless using alternative geometries [2, 3]. Due to the low severity of plastic deformation [4], the sliding distance and velocity between the work piece and tooling are relatively low, which are friction dependant and vary in an uncontrollable way. The oxidation scale is trapped at the contact interface, which acts as a barrier, and prevents the formation of strong chemical bonding. As a result, the ring compression test is probably not suitable for the friction determination of the regions where surface enlargement is severe or where adhesive friction is predominant. Therefore the ring compression tests were mostly used for the friction characterization of bulk metal forming process, in which surface oxides are trapped between the faying surfaces and new surface generation is low. For hot aluminium extrusion process, ring compression tests might be able to emulate the contact conditions between the dummy block and rear surface of the billet.

The extrusion friction tests were developed to overcome the drawbacks of ring compression tests, in which high contact pressure and more intensive surface enlargement can be achieved [5-9]. Most recent research results have shown that different contact conditions in the extrusion friction tests can be achieved by adjusting the extrusion ratio[8]: low contact pressure and surface enlargement can be achieved when low extrusion ratio is used, thus a
high friction sensitivity can be obtained. If a high extrusion ratio is used, high contact pressure and surface enlargement are obtained, which resemble the real contact condition of forging or extrusion processes, but sacrifice fiction sensitivity. The combination of extrusion friction tests and FEM simulations is an effective way of estimating global friction at the billet and container interface.

Ball/pin-on-disc test is a widely used laboratory testing technique for the quantitative study of tribological behaviour of materials. Although ball-on-disc tests are considered to be rather convenient and accurate, the testing results are mostly used for the evaluation and comparison purposes and few results have been implemented as friction boundary conditions in the FE simulations of extrusion processes. This is probably due to the lack of understanding about the evolution of contact conditions during ball-on-disc tests.

During ball-on-disc tests, a high contact pressure can be achieved in a small contact area between the ball and rotating disc. If a soft material is sliding over a harder one, severe plastic deformation may occur, which could lead to the removal of oxide layers and the contact of pure metals. In the meanwhile, the contact pressure may drop with the increasing sliding distance. Therefore, short sliding distance ball-on-disc tests are favourable to the friction characterization of the regions, in which local contact pressure is high and new surface generation is severe, such as the bearing channel of hot aluminium extrusion dies. Because during hot aluminium extrusion, fresh aluminium is extruded out from the container, and in the die bearing, a pure metal contact takes place. It is well known that the presence of chemical stable surface oxides or scale prevents the strong atomic interactions [10]. However, under most of the circumstances, chemical stable surface oxides are not removable from aluminium alloys. Therefore, in order to reproduce the friction conditions in the bearing channel, it is vital to choose a friction testing technique being able to remove the surface oxides. Obviously, short sliding distance ball-on-disc test is one of the best friction testing techniques over the other ones, because during the ball-on-disc tests, severe plastic deformation occurs at the ball / disc interface [11-13], especially during the run-in period. Therefore short sliding distance ball-on-disc test is highly suitable for the study of the friction between fresh metals, which is very much similar to the contact condition in the bearing channel or welding chamber of the extrusion dies. However, the friction test results cannot be transferred into FE simulations of extrusion processes directly as the friction boundary conditions, due to the complicated nature of the evolution of contact conditions during the tests. Therefore, the selection of testing parameters, such as pin and disc materials, sliding
distance and size of the ball has to be considered carefully. Furthermore, friction data processing has to be conducted through FEM simulation or theoretical analysis.

The selection of the pin and disc materials could affect ball-on-disc test results. If the pin is made from a soft material, and the disc is made from a hard one, severe plastic deformation and wear would occur on the tip of the pin, which leads to a significant enlargement of the contact area. After the run-in period, a steep decrease of contact pressure occurs and the contact pressure during the steady-state sliding is close to the yield strength of the soft material. On the other hand, if the disc is made from a soft material, while the pin is made from a hard one, plastic deformation tends to occur in the disc, but the material flow is most likely constrained by the remainder disc material, which is much larger than the size of the wear track. Hence a relatively high hydrostatic pressure which is greater than the strength of the disc material would be imposed onto the spherical pin head. As such, different materials combinations would result in different contact pressures, hence the selection of pin and disc mating materials need to be considered carefully prior to testing, especially when the strengths of the pin and disc materials are different. In the meanwhile, the selection of ball size and sliding distance is of great importance. In general, the contact pressure increases with decreasing ball size [14] and decreases with increasing sliding distance [15].

When a hard pin is sliding over a soft disc, the apparent friction coefficient obtained from the test is normally composed of ploughing and shearing/adhesive friction [12, 16, 17]. The ploughing friction is caused by the plastic deformation of the disc material in front of the pin, which depends on the size of the ball, sliding distance and the material strength. Consequently, the test results cannot be transferred into a metal forming operation directly, when such a material pair is used, because the existence of ploughing friction leads to an overestimation of the friction between the mating materials. As such, the ploughing friction and shear friction have to be discriminated by means of FEM simulations [12, 17] or theoretical analysis [18, 19], and only the shear component of apparent friction representing the real friction between the two mating materials should be used in metal forming operations [12, 17]. However, when the material combination of soft pin and hard disc is used, the friction coefficients obtained from the tests mainly attribute to shearing/adhesive friction. Therefore, with the knowledge about the contact pressure evolution, the results can be used as friction boundary conditions in the FE simulations. To simulate the tribological conditions at the work piece/bearing interface, short sliding distance ball-on-disc tests are recommended, with the disc made from the work piece material and ball made from the die material.
6.2.2 Nature of friction in the bearing channel of hot aluminium extrusion dies

In the hot aluminium extrusion process, it is widely accepted that the change of friction mode from sticking to slipping is caused by the different contact pressure within the bearing channel, i.e. at the die entrance where the contact pressure is high and the overall friction stress is higher than the shear flow stress of the work piece material, thus full sticking friction takes place. On the other hand, the contact pressure is lower at the die exit, where the overall friction stress is lower than the shear flow stress of the work piece material, and hence sliding friction occurs. According to the classic theory of tribology, the mating surfaces are supported by the plastically deformed asperities, and the friction force is generated from ploughing and/or adhesive force when the asperities sliding over each other. From the micro-scale point of view, the contact condition does not vary significantly from different asperity interfaces, because the contact junctions are already in the yielding state and the value of the contact pressure is equivalent to the hardness of the work piece material. On the other hand, the only influence of contact pressure is to change the real contact area, i.e. the number of asperities in contact, but it cannot change the local contact conditions on the tip of an asperity. Therefore, the nature of different friction modes within the bearing channel may be summarized as:

Formation of isolated adhesive junctions → Adhesive junctions growth → Coalescence of adhesive junctions.

Formation of isolated adhesive junctions. At low contact pressure conditions, full sticking occurs on the tips of plastically deformed asperities, but the faying surfaces are only supported by a small number of asperities [20, 21], thus only a few isolated adhesive junctions are formed and normally too small to be observed by the naked eye. At this stage, no adhesive layers can be observed in the bearing channel of the extrusion dies. Therefore a so-called slipping zone can be observed experimentally.

Adhesive junctions growth. As the increase of contact pressure, the number of plastically deformed asperities is increased [20, 21], to support the increased contact pressure. In some regions with a higher asperity density, adhesive junctions growth takes place due to the plastic deformation and some of the adhesive junctions may coalesce to each other locally [20]. Consequently, some of the work piece material or intermetallic wear debris may transfer from the extrudates to the bearing surface of the die, due to the strong adhesive bonding [22-24]. It
is worth noting that, during the hot aluminium extrusion, the temperature of the extrusion die is normally lower than that of the billets, and thus the strength of the adhesive junctions is enhanced once they are formed and adhere onto the extrusion die, due to the decrease of temperature. Therefore, the size of the adhesive junctions might “grow” bigger and bigger, due to the aggregation of wear debris. This is the so-called “lump growth” [22, 25], which might be one of the reasons for the generation of extrusion surface defects, such as die line and pick-up. At this stage, an in-continuous tribo-layer might be visible on the die land, and the so-called transition zone between the slipping and sticking zone may be observed.

*Coalescence of adhesive junctions.* At the die entrance, where a high contact pressure is achieved, the real contact area is maximized, thus the number of adhesive junctions is significantly increased and they are close to each other, thus tend to coalesce with each other. Moreover, a great amount of work piece material may transfer from the extrudates to the bearing surface of the die, due to the strong adhesive bonding. Therefore there exists a great chance for the adhesive junctions within a large area to coalesce with each other and thus the adhesive junctions can be observed by the naked eye. At this stage, a continuous tribo-layer, i.e. the so-called adhesive zones, can be observed on the die land.

### 6.3 RECOMMENDATIONS

#### 6.3.1 Short sliding distance ball-on-disc tests

During the run-in period of ball-on-disc tests, the apparent friction force is composed of the ploughing part and adhesive part. The ploughing part is caused by the plastic deformation of the disc material in front of the ball, which only exists in the first few laps of sliding. In the bearing channel of aluminium extrusion dies, plastic contact of fresh metals occurs, thus short sliding distance ball-on-disc tests should be used and probably the first lap of sliding is of particular interest, because due to the limitation of the ball-on-disc tests, cyclic loading occurs on the same wear track, thus fresh surface contact disappears after a few laps of wear. In addition, after a few rotations, the contact between the steel ball and aluminium disc transforms from plastic contact to elastic contact. A ball-on-disc tester with a dynamically changeable radius of wear track is probably favourable, in which cyclic load on the same wear track can be avoided. Alternatively, a soft pin with constant cross-sectional area can be used
to slide against a hard disc. The constant cross-sectional area is to avoid the contact pressure changes during the tests and thus no changeable radius of the wear track is necessary.

6.3.2 Double action extrusion tests

In the current work, extrusion dies with a 15°choke angle were used and full sticking friction seems to be able to represent the friction condition in the bearing channel of the extrusion dies. In the future work, extrusion dies with a smaller choke angle could be used so that the distinct sticking and slipping zones can be observed. In addition, a relatively low extrusion speed (up to 11 mm/s) was applied in the current DAE tests to correspond to the linear speed of the ball-on-disc friction tests, which might be another reason for the full sticking friction observed in the bearing channel. A higher value of extrusion speed should be used for the future studies to simulate the real extrusion conditions in the industry.

References


In recent years, finite-element (FE) simulations have been extensively used in scientific research and industrial practice to analyse the extrusion process. A basic issue of FE simulations is the accuracy of the results, which is mainly determined by the viscoplastic material behaviour of aluminium alloys at elevated temperatures and the determination of boundary conditions, especially the friction boundary condition. In this thesis, the determination of friction boundary conditions for hot aluminium extrusion process was done by using the short sliding distance ball-on-disc test at elevated temperatures.

A mathematical model for high-temperature ball-on-disc tests was developed. This model is capable of discriminating the individual contributions of ploughing and shearing friction to the apparent friction. It was found that during high-temperature ball-on-disc tests, the friction coefficients obtained from ball-on-disc tests alone were insufficient to represent the frictional interaction between deforming aluminium and steel at elevated temperatures.

Based on the ball-on-disc test results, a novel physically based friction model (adhesive strength friction model: ASFM) was developed for the bearing channel of an aluminium extrusion die. To verify this friction model, a novel extrusion testing method, double action extrusion (DAE), to highlight the effect of friction at the die bearing in aluminium extrusion was developed. The ASFM was implemented into the FE simulation of DAE tests, and good agreements between the FE predictions and experiments were obtained, indicating that ball-on-disc test is an effective way of characterizing the friction for the bearing channel of extrusion dies. For a further understanding of the DAE, a theoretical model was developed, and a good agreement between the modelling results and experiments was obtained. The theoretical modelling results revealed that the length difference of the extrudates was caused by the combined effects of friction and material rate dependency at elevated temperatures.
SAMENVATTING

In de afgelopen jaren, zijn eindige-elementen (FE) simulaties veelvuldig gebruikt bij wetenschappelijk onderzoek en in de industriële praktijk om het extrusie-proces te analyseren. Een fundamentele kwestie van FE simulaties is de nauwkeurigheid van de resultaten, die voornamelijk wordt bepaald door het viscoplastisch gedrag van aluminium legeringen bij hoge temperaturen en de toegepaste randvoorwaarden, in het bijzonder de wrijving randvoorwaarde. In dit proefschrift zijn de wrijving randvoorwaarden voor het aluminium extrusie-proces bepaald aan de hand de short sliding distance ball-on-disc-test bij verhoogde temperaturen.

Een wiskundig model voor de ball-on-disc proeven was ontwikkeld. Dit model is in staat de individuele bijdragen aan de schijnbare wrijving van ploegen en afschuivingswrijving te onderscheiden. Het bleek dat bij hoge temperatuur ball-on-disc tests, de verkregen wrijvingscoëfficiënten alleen onvoldoende waren om de wrijvingsinteractie tussen aluminium en staal bij hoge temperaturen te beschrijven.

Op basis van de ball-on-disc testresultaten is een nieuw fysisch wrijvingsmodel (kleefkracht wrijving model: ASFM) ontwikkeld. Om dit wrijvingsmodel te verifiëren is een nieuwe extrusie testmethode (double action extrusion (DAE)) ontwikkeld met de nadruk op de wrijving in de matrijs. De ASFM werd geïmplementeerd in de FE simulatie van de DAE tests, en goede overeenkomsten tussen de FE voorspellingen en de experimenten werden verkregen, wat aangeeft dat ball-on-disc test is een effectieve manier is om de wrijving in de matrijs te beschrijven. Voor een beter begrip van de DAE, is een theoretisch model ontwikkeld, waarvan de resultaten goed overeenkwamen met de experimenten. De resultaten van het model lieten zien, dat het lengte verschil van de extrudaten werd veroorzaakt door de gecombineerde effecten van wrijving en deformatiesnelheidsafhankelijkheid bij verhoogde temperaturen.
LIST OF PUBLICATION

Journal publications:


L. Wang, J. Zhou, J. Duszczyk, and L. Katgerman, Friction in aluminium extrusion - part 1: A review of friction testing techniques for aluminium extrusion, accepted for publication.


L. Wang, J. Zhou, J. Duszczyk, and L. Katgerman, Friction modelling for the bearing channel of hot aluminium extrusion process, accepted for publication.
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I would also like to express my sincerest appreciation to my parents and my wife. You always support me and help me. Thank you!
Appendix A

Flow stress of AA7475 at different temperatures

The flow stress of AA7475 was determined from compression tests conducted on a Gleeble 3800 thermo-mechanical simulator.

Table G.1 Flow stress data of AA7475 at strain rate of 0.01/s.

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Table G.2 Flow stress data of AA7475 at strain rate of 0.1/s.

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Table G.3 Flow stress data of AA7475 at strain rate of 1/s.

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Table G.4 Flow stress data of AA7475 at strain rate of 10/s.
Appendix B
Constitutive parameters for aluminium alloys

In this thesis, the Sellars –Tegart’s equation was used to model the evolution of steady-state flow stress of AA7475, which is a function of temperature and strain rate:

\[ Y(\dot{\varepsilon}, T) = \frac{1}{\alpha} \sinh^{-1}\left(\frac{\dot{\varepsilon}}{A} \exp\left(\frac{Q}{RT}\right)\right)^{\frac{1}{n}} \]  

Constants in the above equation were determined from compression tests and are shown in Table G.5.

| Material constants for AA7475. |
|-----------------------------|----------------|--------------|--------------|--------------|
| \(\frac{1}{\alpha}\) (MPa) | A (s\(^{-1}\)) | n | Q (J/mol) | R (J/K/mol) |
| 87.72 | 1027094727 | 5.41 | 129400 | 8.314 |

This equation can be used to model the evolution of steady-state flow stress of varies aluminium alloys. Table G.6 shows the values of the constants determined for a range of aluminium alloys obtained from torsion tests.

<p>| Material constants for different aluminium alloys [1]. |
|-----------------------------|----------------|--------------|--------------|--------------|
| Alloy | (\frac{1}{\alpha}) (MPa) | A (s(^{-1})) | n | Q (J/mol) | R (J/K/mol) |
| 1100 | 22.2 | 5.18\times10^{10} | 5.66 | 1.58\times10^{5} | 8.314 |
| 2024 | 62.5 | 3.25\times10^{8} | 4.27 | 1.49\times10^{5} | 8.314 |</p>
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References