

IMPERIAL COLLEGE LONDON

**INTERACTIVE MODELLING OF THE
EVOLUTIONARY TRIBOLOGICAL
PHENOMENA IN COLD STAMPING**

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Declaration of Originality

This thesis hereby presented is based on research by the author at the Department of Mechanical Engineering of Imperial College London. I declare the work contained in this thesis is only author's own work. The work of this thesis has not been submitted elsewhere for any other degree or qualification.

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Abstract

Tribological contacts are responsible for approximately 23% of world's total energy consumption with the majority of energy consumption involved in manufacturing operations. Manufacturing is one of the main energy consuming sectors and the optimizing of its friction and wear performance can significantly increase the life-time and reduce the maintenance work of equipment. In cold stamping, friction and wear that are generated between the tool and work-piece are important tribological responses influencing the material flow, surface quality and tool-life. It has been found that these responses are not material properties but highly nonlinear phenomena, which vary spatially and historically with operation parameters, such as contact pressure, sliding speed and sliding distance of forming processes.

It has been widely observed that galling and lubricant breakdown phenomena that occur during cold stamping cause evolutionary surface morphology changes and thus involve the simultaneous interaction between friction and wear: the galling generated on a tool surface during forming can cause coefficient of friction (COF) to increase and wear rate to decrease; the lubricant film breakdown with time during single path sliding leads to a rapid increase in the COF and wear at the contact.

In this thesis, interactive friction models were developed to predict the evolution of COF with the effect of galling and lubricant breakdown. In these models, the historical and pressure dependencies of these phenomena combining the effects of operation parameters were addressed. These interactive friction models could be applied to situations involving galling and lubricant breakdown in a dynamic environment such as the metal forming industry, where the use of traditional constant COF values present limits in predictive accuracy.

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Nomenclature

Variables (English)

A contact area	D model constant
a_1 model constant	E^* parameter of the elasticity modulus
a_2 model constant	F friction
A_f fractional contact area	$\bar{f}(t)$ normalised galling area at time t
A_H area of the Hertzian circle	$f(t)$ galling area fraction at time t
B model constant	$f(t_0)$ galling area fraction at the initial of the contact
b model constant	$f(t_{sat})$ saturated value of the galling area fraction
b_1 model constant	F_f frictional force at the contact
b_2 model constant	F_n Normal force at the contact
b' half contact width	G^* parameter of material
B' length of cylinder	H hardness
c model constant	h lubricant film thickness
C model constant	$h(0)$ lubricant film thickness at time 0
C_s model constant	$h(t)$ lubricant film thickness at time t
C_w function of ejection speed of wear particles	H_{min}^* parameter of minimum film thickness
d model constant	k wear coefficient

NOMENCLATURE

l length of the line contact	v sliding velocity
L sliding length	V wear volume
L' lubricant number	V_{avg} average volume of third body debris
m model constant	V_e ejected wear volume
M_i mass of the third body entrapped inside of the contact	V_i wear volume of the internal third body
M_i^{max} model constant	V_s source wear volume
M_i^{start} mass of third body entrapped at the contact	V_t wear volume at the running-in state
n experimental constant	W contact load
N number of elements at the contact	W^* parameter of contact load
P contact pressure	x sliding distance
P_0 reference pressure	YP yield stress
Q_s flow of third body detached from the first body	
Q_w wear flow of third body ejected outside the contact	
R^* parameter of radius of the contact	
Ra surface roughness	
SD standard deviation	
t time	
U^* parameter of velocity	
u_{dif} differential velocity	

Variables (Greek)

α pressure exponent of viscosity
β lubricated area ratio
η viscosity of lubricant
η_0 atmospheric viscosity
κ_1 lub model constant
κ_2 lub model constant
λ_1 model constant

NOMENCLATURE

λ_2 model constant	ρ_s density of the source of third body
λ_3 model constant	σ surface roughness
λ_4 model constant	τ_0 eyring shear stress
μ coefficient of friction	τ_a adhesion stress component
μ_{BL} friction parameter of boundary lubrication	τ_f frictional stress
μ_d dry coefficient of friction	τ_h hydrodynamic friction stress component
μ_{EHL} friction parameter of elastohydrodynamic lubrication	τ_p ploughing stress component
μ_{Al-CI} coefficient of friction generated by aluminium-cast iron contact	
μ_L coefficient of friction at the fully flooded lubrication condition	
μ_P contribution of contact pressure to the coefficient of friction	
μ_{Al-Al} coefficient of friction generated by aluminium-aluminium contact	
μ_v contribution of sliding velocity to the coefficient of friction	
ρ third body density	
ρ_e density of the ejected third body	
ρ_i density of internal third body	
$\bar{\rho}_i$ normalised density of internal third body	

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List of Publications

Parts of the work presented in this thesis have been disseminated through a number of written publications; these are listed below, as of August 2018.

Peer-reviewed journal publications

- [p1] **Hu, Y.**, Wang, L., Politis, D. J., & Masen, M. A. (2017). Development of an interactive friction model for the prediction of lubricant breakdown behaviour during sliding wear. *Tribology International*, 110, 370-377.
- [p2] **Hu, Y.**, Yuan, X., Ma, G., Masen, M. A., & Wang, L. (2015). Tool-life prediction under multi-cycle loading during metal forming: a feasibility study. *Manufacturing Review*, 2, 28.
- [p3] **Hu, Y.**, Zheng, Y., Politis, D. J., Masen, M. A, Cui, J., & Wang, L.(2019). Development of an interactive friction model to predict aluminum transfer in a pin-on-disc sliding system, *Tribology International*, 130, 216-228.
- [p4] **Hu, Y.**, Zheng, Y., Politis, D. J., & Wang, L., Interaction between friction and galling during sliding of an aluminium alloy against cast iron under contact load changing conditions: Experiments and modelling, (submitted).
- [p5] **Hu, Y.**, Zheng, Y., Politis, D. J., & Wang, L., Interactive mechanisms and modelling of the evolutionary tribological phenomena in cold stamping processes: a review, (submitted).
- [p6] Zheng, Y., **Hu, Y.**, Wang, A., Politis, D. J., Fakir, O. E., Said, R., & Wang,

L., Data mapping for hot aluminium stamping towards data-driven tribological characterization schemes. Unpublished manuscript.

Refereed conference publications

- [c1] **Hu, Y.**, Gharbi, M. M., Liang, V., Zheng, Y., Politis, D. J., & Wang, L. (2018). The Galling Behavior of Advanced Coating Contacts with Aluminium Alloy during Sliding Wear. In *Key Engineering Materials* (Vol. 767, pp. 117-123). Trans Tech Publications.
- [c2] **Hu, Y.**, Zheng, Y., Politis, D. J., Wang, L., & Gharbi, M. M. (2017). Investigation of Galling Behaviors Between an Aluminum Alloy and Metal Forming Tool. In *Advanced High Strength Steel and Press Hardening: Proceedings of the 3rd International Conference on Advanced High Strength Steel and Press Hardening (ICHSU2016)* (pp. 438-442).
- [c3] **Hu, Y.**, Zheng, Y., Politis, D. J., & Wang, L., FE simulation of aluminium galling induced forming tool failures under multi-cycle loading conditions: a feasibility study, accepted by *The 4th International Conference on Advanced High Strength Steel and Press Hardening (ICHSU2018)*.
- [c4] Chan, T. Y., **Hu, Y.**, Gharbi, M. M., Politis, D. J., & Wang, L. (2016, August). The friction coefficient evolution of a MoS₂/WC multi-layer coating system during sliding wear. In *Journal of Physics: Conference Series* (Vol. 734, No. 3, p. 032081). IOP Publishing.
- [c5] Zheng, Y., **Hu, Y.**, Gharbi, M. M., Politis, D. J., & Wang, L., Development of coating design guidelines for a sheet metal forming tool—a feasibility study, In *Proceedings of the 10th International Tool Conference 2016* (pp. 83-91).

Chapter 1

Introduction

1.1 Challenges of galling and lubricant breakdown in cold stamping

1.1.1 Introduction of galling and lubricant breakdown

The challenging tribological conditions of cold stamping, such as fracture of work-piece and short tool-life, can be caused by large press capacities and poor lubrication conditions during the forming process of high strength materials with complex forming geometries. The tribological conditions are commonly evaluated by consideration of friction and wear during the forming process. Friction describes the tangential force generated at the contact point, which affects the material flow and drawability of a forming process. In comparison, wear occurs on both work-piece and tool and can cause damage to both surfaces, including material scratching, scoring and galling, which can reduce tool-life and the surface quality of products.

Friction and wear are not material properties but are two evolutionary variables of an identical tribo-system (Kato, 2000). These variables are identified as the result of a combination of multiple influencing parameters, which can be classified into 3 categories: 1) material properties of the contact pair, 2) contact conditions, such as surface roughness and lubrication and 3) operation parameters (or process variables), such as temperature, sliding speed, sliding distance, contact pressure

and forming cycles. As friction and wear are simultaneously affected by the same parameters, they are reasonably related to each other and interactions are expected between these tribo-responses (Kato, 2000). These interactions have been observed in many tribological phenomena, including adhesive wear, lubricant breakdown, coating breakdown (Pujante et al., 2015), oxide film (Kato, 2000, Dohda et al., 2015) and reaction layer (Kato, 2000, Holmberg and Matthews, 1994).

In cold stamping, the tribological interactions commonly occur in:

- Gallings,
- Lubricant breakdown.

Galling is defined by standard of American Society for Testing and Materials (ASTM) G40, as a form of surface damage arising between sliding solids distinguished by macroscopic, usually localized, roughening and creation of protrusions above the original surface; it includes plastic flow or material transfer or both. It is more often mentioned in industrial field than academic field. In cold stamping, it is commonly used to describe the severe adhesive wear and characterised by the material transfer from the soft work-piece to the tool by adhesion. A transfer layer is formed after repeated stamping operations, especially at some critical positions, such as radii of a tool, which can result in high local friction, scratching on the work-piece and misalignment of a tool. Material transfer caused by adhesive wear may refer to other industrial terms, e.g. material pick-up, scuffing, localized welding and seizing (seizure). Gallings is the most widely used and thus adopted in the present work.

Lubricant breakdown is a form of lubrication failure where it is not possible to maintain completely hydrodynamic or mixed conditions and there may be a considerable amount of boundary lubrication during which wear of the surfaces may occur (Bowden and Tabor, 2001). In cold stamping, this lubrication failure is caused by lubricant layer uneven transportation and thinning during forming and may lead to lubrication starvation and direct contact/wear of work-piece and tool. In many cases of stamping, lubricant is squeezed out towards regions with lower pressure and side leakage occurs. Different from bearing and gear systems, an adequate quantity

of lubricant applied prior to the stamping operation does not guarantee that lubrication will be effective at all locations or at all stages of a forming operation. Therefore, the term, lubricant breakdown, is adopted in the present work to distinguish the stamping tribo-system from cyclic systems.

It has been found that both galling and lubricant breakdown cause dramatic morphology changes at the contact, which simultaneously affects friction and wear. Due to 1) the potential industrial audience and 2) aim of distinguishing the tribological failures in stamping tribo-system from other systems (e.g. bearing and gear), in this work, the industrial terms, galling and lubricant breakdown, are used together with scientific terms (e.g. aluminium transfer, adhesive wear, starvation, boundary and flooded lubrications) to describe the observed phenomena.

1.1.2 Characteristics of operation parameters in cold stamping

The tribological conditions of cold stamping, such as contact pressure, sliding distance and sliding speed, have the characteristics of being:

- Uneven in spatial distribution
- Uneven in historical distribution

which indicates that these influential parameters will vary with location on the tool/work-piece surface as well as forming phases/number of cycles.

Contact pressure is one of the decisive factors to a tribo-system. It is generated at the interface between the work-piece and tool during stamping and has strong spatial and historical dependency. First of all, the variation is largely determined by the geometry of tool and the deformation process of the work-piece. Protrusion features on the tool, such as draw beads and corner radii, are likely to generate higher contact pressure during the forming process than flat areas, such as the flange and wall (as shown in Fig. 1.1). As the forming process progresses, the deformation and strain hardening of the work-piece may also result in a redistribution of local pressure and finally generate a spatial distribution of the contact pressure. Another

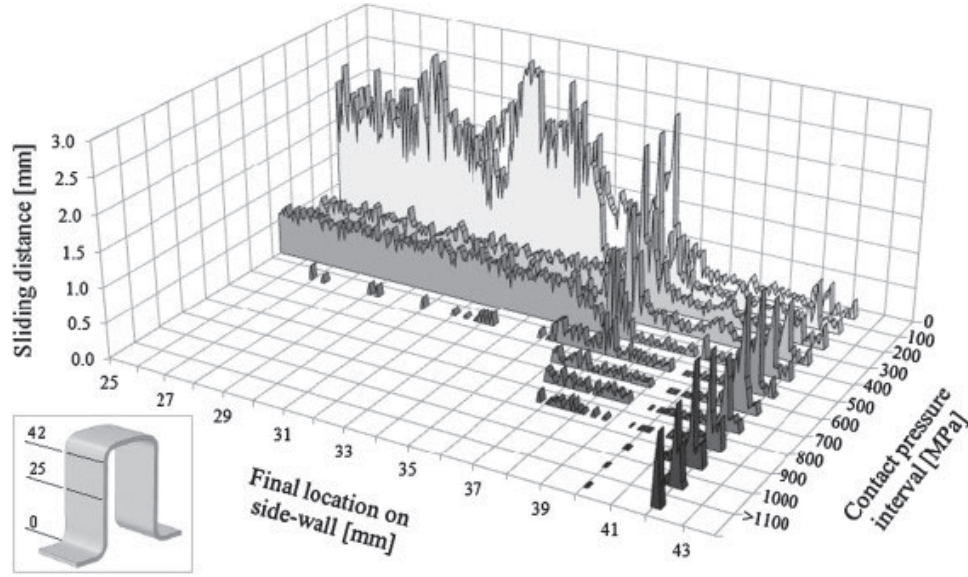


Figure 1.1: Spatial distribution of contact pressure: contact conditions occurring on the blank surface during the U-shape bending of DP600 (Pereira et al., 2010).

characteristic of contact pressure in stamping is the historical dependency where the forming load varies with each increment of a forming cycle (as shown in Fig. 1.2), which is determined by die geometry, material and friction condition. It should be noted that stamping provides some unique dynamic characteristics due to the sudden contact and detachment of tools in multi-phase forming resulting in a rapid change of contact load, or ‘load jump’, and the variation of friction and wear with each forming cycle which has not been modelled in most sheet metal forming studies.

Sliding distance itself is a historical parameter that indicates the accumulation of the displacement between the work-piece and tool during a forming process. Instead of punch displacement, the material flow is also determined by the geometry of tool and friction condition and thus makes it a spatial parameter. As sliding speed is the derivation of sliding distance, it is not only decided by punch speed but also inherits the historical and spatial dependencies, where it varies at different locations during the forming process.

The variation of contact pressure, sliding distance and speed will further lead to the variation of tribological responses during a forming process, which challenges

both the existed tribological models as well as FE techniques. Based on the statistical study of the contact condition by Zheng et al. (2018), many of the critical positions of stamping processes experience a relatively high contact pressure and a low sliding speed (as shown in Fig. 1.3). This means the tribological condition in stamping process is located in the active zone of galling for many materials (Deuis et al., 1997, Okonkwo et al., 2016) and also has high risk of lubricant breakdown (Begelinger and De Gee, 1982).

1.1.3 Challenges of tribology modelling

During the cold stamping process, the severe tribological damage of tools and work-piece is caused by a lubricant film breakdown and further tendency to galling has been subjected to considerable attention and study. However, the influencing factors of the tribological damage or their non-linear evolutionary mechanisms have not been well studied. As both galling and lubricant breakdown phenomena lead to dramatic morphology changes at the contact interface and simultaneously affect both friction and wear progress, an interaction between them exists which has been predicted and observed in the literature but surprisingly attracts limited attention in modelling. The dynamic tribological condition during cold stamping also challenges the present friction models and a holistic tribology model that can simulate both the historical and spatial distribution of friction and wear under multi-cycle loading condition is required.

1.2 Aim and objectives

The main aim of this research is to quantitatively model the evolutionary tribological phenomena, specifically galling and lubricant breakdown, in cold stamping process. To achieve this aim, the following objectives have been met:

1. Investigate the tool failure mechanisms of galling and lubricant breakdown in cold stamping.

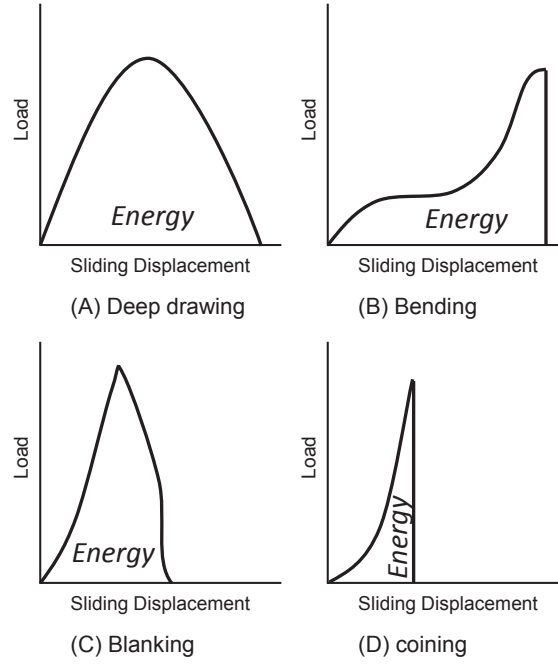


Figure 1.2: Historical distribution of contact load: schematic of load-displacement curves for various sheet forming operations (Altan and Tekkaya, 2012).

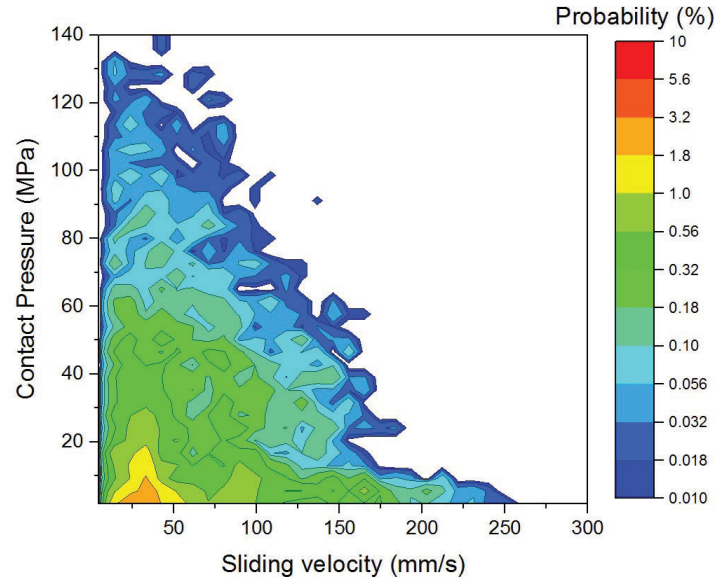


Figure 1.3: Probability of contact pressure and sliding velocity on the tool surface of a limited-height-dome-shape of AA6082; each data was obtained from one element at a forming stage on the tool in FE simulation (Zheng, Y et al., 2018).

2. Identify the interactive relations between friction and wear in the failure mechanisms.
3. Investigate the origin and influential factors of the interactions.
4. Study the effect of operation parameters of cold stamping on the interactive relations.
5. Quantitatively measure the evolutionary tribological failure process with the effect of operation parameters.
6. Develop mechanisms-based friction models to describe the interactive relations.

1.3 Major tasks

To achieve the objectives, the research programme has been divided into a number of distinct tasks, which are listed below:

1. Review of the state of the art study for galling and lubricant breakdown forming. Focusing on 1) mechanisms; 2) modelling techniques.
2. Quantitatively study the galling and lubricant breakdown phenomena by conducting simulative tests.
3. Study the evolutionary phenomena of galling and lubricant breakdown with the effect of operation parameters.
4. Develop interactive models between friction, wear and morphology changes.

1.4 Thesis structure

In this thesis, the tribological challenges in cold stamping, specifically galling and lubricant breakdown, are modelled. The materials are presented as follows:

Chapters 1 and 2 propose the challenges of present tribological modelling technique, the characteristics of tribological conditions and the methodologies of tribological study in cold stamping. Gallling and lubricant breakdown phenomena are reviewed by their mechanisms, interactions and their modelling techniques in sheet metal forming.

Chapter 3 presents the experimental methods for quantitative study of the evolutionary process of gallling and lubricant breakdown. The materials that represent the workpiece and tools are presented and the experimental setup with key procedures and the characterizing approaches are introduced.

Chapter 4 studies the evolutionary forming process of gallling with the interaction between friction and wear by a simulative test. An interactive friction model is developed.

Chapter 5 presents the application of this model to the load change conditions and a good agreement between experiments and model results are presented.

Chapter 6 investigates the evolutionary study of the lubricant breakdown process. An interactive friction model that describes the effects of operational parameters on the variation of lubricant breakdown phenomena at room temperature are presented.

Chapter 7 summarises the conclusions of this research and presents the future work.

Chapter 2

Literature review of galling and lubricant breakdown phenomena in cold stamping

2.1 Methodologies of tribological study in stamping

2.1.1 Tribological test methods

¹Tribological tests in sheet metal forming have been reviewed by Dohda et al. (2015), Bay et al. (2008) and Podgornik et al. (2004). Based on the literature, it was found that no specialist rig has been designed for galling or lubricant tests although there are specific test procedures and evaluation methods that were applied. Based on Bay et al. (2008), the tribological test methods in sheet metal forming can generally be classified into two classes:

¹This chapter is based on paper work ‘Y. Hu, Y. Zheng, D. J. Politis, L. Wang, *Interactive mechanisms and modelling of the evolutionary tribological phenomena in cold stamping processes: a review*, (submitted).’

- Simulative tests
- Process tests

The simulative tests model the tribological conditions with the attempt to study friction, wear and lubrication in a controlled way. They can be design for simulating the targeted tribological location on the tool.

In the review by Bay et al. (2008), an overview of the most frequently applied simulative tests in a typical deep drawing setup has been presented and is shown in Fig. 2.1. In this figure, the flange region can be simulated using the representative tests: 1) strip drawing with a flat die, 2) draw bead test and 3) drawing with tangential compression test. The conditions on the die curvature are represented by test 4) bending under tension test and 5) bending with tangential compression test. Test 6 simulates the ironing in the die. Tests 7 and 8 represent stretching over the punch curvature and the punch nose.

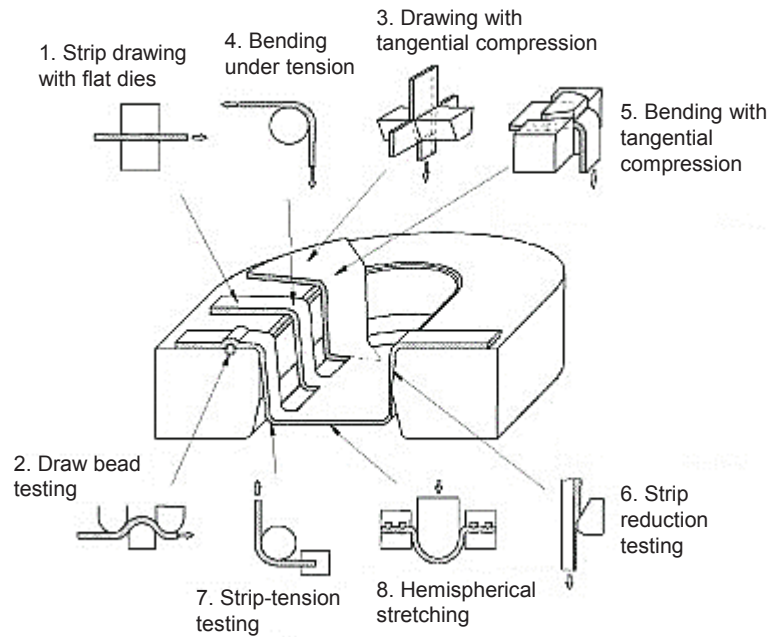


Figure 2.1: Schematic illustration of simulative tests for stamping process (Bay et al., 2008).

Pin-on-disc tests are widely adopted to observe friction and wear between rigid sliding friction pairs and are well applied in the tribological test of galling and

lubricant breakdown (Riahi and Alpas, 2007, Wang et al., 2010, Descartes and Berthier, 2002, Wang and Yang, 2012, Wang et al., 2012). A schematic setup of this arrangement is shown in Fig. 2.2A, which consists of a stationary pin in contact with a rotation disc. A normal load is imposed by dead weights on the top of the pin and friction is recorded by spinning the disc against the pin. Although this test is rather convenient and accurate, the test conditions may not be representative of real sheet metal forming conditions and thus the results are difficult to implement into FE simulations.

Strip reduction tests are designed to simulate the tribological conditions in ironing. As shown in Fig. 2.2B, the thickness of the strip is reduced between a non-rotating hardened tool pin that simulates the cylindrical punch and a supporting plate. This test is very useful for testing stamping lubrication and tool materials/coatings by quantifying the amount of galling on the roller or the strip (Altan and Tekkaya, 2012, Bay et al., 2008, Olsson et al., 2004).

Bending under tension test is another popular simulative test (Kim et al., 2012, Bay et al., 2008), which simulates the plastic deformation condition at the curvature area of tools. As shown in Fig. 2.2C, the tool pin can be set as fixed or free. One strip is drawn over a fixed, circular cylindrical tool-pin and another strip is drawn over a free rotating pin, implying that no sliding takes place. The difference in tension measured by the two tests gives an estimate of the friction. Similar to the strip reduction test, tribological conditions can be evaluated by the morphology of the tool-pin.

Process tests are characterised as replicating typical forming operations and thus have many variations. Benchmarks have been developed to replicate the most common applications in a controlled way. The cup drawing test (Fig. 2.3A) is a simple deep drawing process that is widely used to test lubricating conditions, and to develop and verify tribological models (Kim et al., 2012, Tamai et al., 2016). In this test, the work-piece is radially drawn into a forming die by a round cup cylindrical punch to form a hollow cup (Altan and Tekkaya, 2012, Olsson et al., 2004) and the tribological condition is measured by evaluating the fracture of the work-piece

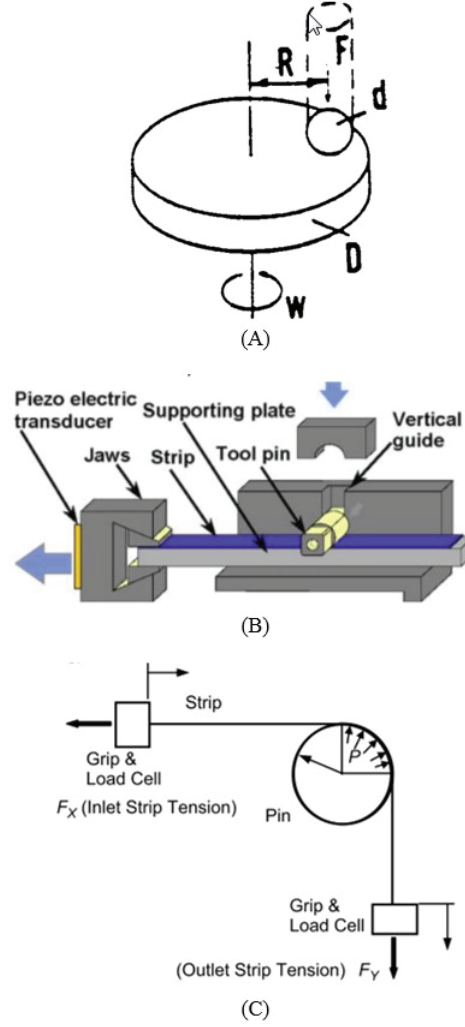


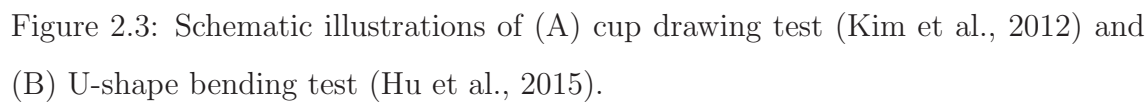
Figure 2.2: Schematic illustrations of (A) pin-on-disc test (ASTM G99, F is the normal force on the pin, d is the pin or ball diameter, D is the disk diameter, R is the wear track radius, and ω is the rotation velocity of the disk). (B) strip reduction test (Olsson et al., 2004). (C) bending under tension test (Kim et al., 2012).

and damage on the tool. U-shape bending tests have been developed as a simplified version of a round cup deep drawing test (Altan and Tekkaya, 2012) (Fig. 2.3B) and are used to provide experimental data to develop tribological models (Dong et al., 2017, Wang et al., 2016, Lee et al., 2015, Pereira et al., 2010) and verify FE simulations (Hu et al., 2015).

The identification of galling on engineering tool surface, especially in an effort to qualitatively rank the galling resistance of material couples, was reviewed by Podgornik et al. (2004). In this work, the galling tendency was quantified by the coefficient of friction increase exceeding a critical value of 0.5. In the pin-on-disc test, a safe sliding distance before galling occurred could be determined from the COF evolution. The severity of galling based on microscopy assessments was quantified by the galling area fraction, which is defined as the contact area that is covered by the transfer layer. This parameter has been proven as a reliable factor to measure the galling formation process (Rabinowicz, 1965, Schedin, 1994). In the studies of the equilibrium and the built-up process of the transfer layer, the measurement of thickness (de Rooij and Schipper, 2001b, Österle et al., 2012), mass (Fillot et al., 2007) and volume (Dong et al., 2017) of the transfer layer was also performed.

The test standard ASTM G98-02 is used to determine the galling resistance of materials, as was employed in the literature (Wen and Huang, 2012, Foster et al., 2008). The test is conducted under a constant compressive load by rotating one sample 360° relative to the other. The criterion for galling is the visual appearance of this phenomenon on the specimen surface. The threshold galling stress is determined by the highest non-galling stress from the test. This traditional test takes into consideration the contact pairs and contact loads; however, the historical transfer process is neglected. As the experimental data clearly demonstrate, a low severity of transfer and a low COF at the initial stage of a test is not a reliable indicator for galling prevention (Van der Heide et al., 2001), and hence long-term testing that recognises the different transfer modes and their transition should be adopted.

The lubricant thinning and breakdown are always identified by the simultaneous increase of friction and wear. Due to the variation of COF being easy to record in



real-time, and the development of a corresponding relationship between lubricant film thickness and friction (Wen and Huang, 2012, Bhushan, 2013), the variation of COF has been widely used to evaluate lubrication condition transition. This method is also extended to wear map (Begelinger and De Gee, 1982, Schipper and De Gee, 1995, Salomon, 1976) and an interaction between friction-wear-lubricant thinning/breakdown is developed, in which the lubricant mode transition can directly lead to wear mode transition and corresponding friction variation. Also, the wear-caused morphology change is direct evidence of the failure of lubricant which can be used to evaluate the lubricant performance (Tamai et al., 2016, Hu et al., 2017a) by identifying the wear mode. The real-time observation of wear and morphology is relatively difficult and some transparent and in-situ rigs have been developed (Hector and Wilson, 1991, Heinrichs et al., 2012).

2.1.2 Tribological modelling methods

Models have been developed in the literature to quantitatively analyse the variation of tribological responses during sheet metal forming. Although the Coulomb model ($\mu = F/N$, where μ is coefficient of friction, F is the friction force and N is the normal load) is still the most commonly used friction model in sheet metal forming, the variation of COF and the corresponding wear rate during a forming process have resulted in evolutionary models being increasingly adopted (Nielsen and Bay, 2017).

On one hand, the development of these models has focused on an understanding of tribological phenomena inherent in sheet metal forming. Constitutive equations are used to model the variations of friction and wear over a range of operation parameters such as speed and pressure by calibrating to experimental data (Lee et al., 2015). Multi-scale models that are based on the micro-scale asperity contact and macro-scale modelling by statistical methods have also been developed (Hol et al., 2012, Karupannasamy et al., 2014, Ma et al., 2010, Karupannasamy et al., 2013). With the interaction between friction and wear, unified models that analyse different tribological responses have also been developed (Diomidis and Mischler, 2011, Iordanoff et al., 2002). On the other hand, with the variation process of tribological

responses have been studied, the multi-cycle modelling technique that can predict the variation of tribological condition due to the increase of forming cycles was developed by Karupannasamy et al. (2014), which is close to the industrial application. By implementing advanced tribological models into FE simulation, the distribution of friction and wear on both work-piece and tool can be visually analysed (Kim et al., 2012) which makes further study of the friction effect on springback (Wang et al., 2016, Lee et al., 2015, Gil et al., 2016), prediction of formability (Tamai et al., 2016) and prediction of tool-life (Hu et al., 2015) possible.

2.2 Galling in cold stamping

2.2.1 The mechanism of galling

The galling mechanism can be classified into two processes, namely 1) wear (detachment) process and 2) transfer process.

The wear process is the initial start of galling as the source of the transfer material has been identified from the original contact pairs (Schedin, 1994, Heilmann et al., 1983). The adhesive wear mechanism has been well reviewed by the literature (Bhushan, 2013, Wen and Huang, 2012, Kato, 2002, Kato, 2000, Holmberg and Matthews, 1994, Suh, 1986a), in which Holm-Archard's and Suh's theory are the most influential and well accepted.

The Holm-Archard model (Eq. 2.1) and its modifications (Bhushan, 2013, Pereira et al., 2010, Archard, 1953) provide the basic linear relations between the factors:

$$V = \frac{kWx}{H} \quad (2.1)$$

where V is the wear volume, k is the wear coefficient, W is the contact load, x is the sliding distance and H is the hardness of the soft material. In this model, increasing the contact load and sliding distance will increase the wear volume, whereas the increase of the hardness of the soft material will decrease the wear volume. Most of the material follows the trend of wear volume predicted by this model but the proposed proportionality with the contact load and material hardness was not always

observed in sliding wear systems in reality.

Suh's delamination adhesive wear theory (Suh, 1986a) explains the deformation and detachment mechanism inside soft materials. It states that large plastic strains are developed inside of the work-piece and result in cracks and void nucleation in the deformation layers, which causes the formation of thin, long wear particles. In macro-scale, the increase in sliding distance or the magnitude of the applied load can cause a greater critical depth of the deformation layer. The metallurgical structure is considered in this theory but hard to qualitatively measure (Deuis et al., 1997).

Moreover, both Holm-Archard model and Suh's delaminated theory of wear neglect the formation of the tribo-layer and the role of wear particles in their treatment of adhesive wear (Heilmann et al., 1983, Deuis et al., 1997), which have been found to form a running-in state during sliding. In sheet metal forming, considerable forming cycles occur in the running-in state and thus its effect cannot be ignored. The study of Burwell and Strang (1952) examined the running-in of wear evolutions. It was found that, at the initial stage, wear increased in a nonlinear fashion with sliding distance due to galling. As shown in Fig. 2.4, in the running-in state, the increase of wear volume gradually slows down corresponding to the decrease of wear rate. In the steady state, the wear volume reaches a linear relation with a constant wear rate, which can be explained by static wear models.

The detached materials may transfer to the counterpart by direct and indirect transfer mechanism.

The direct transfer mechanism was proposed by Bowden and Tabor (2001) where the asperities of the work-piece can directly adhere to the counterpart by mechanical and chemical bonding. In this situation, the shear strength generated at the interface is smaller than the stress inside the work-piece material and asperity break and transfer occurs during sliding. Rabinowicz (1980) stated a similar explanation that when an adhesive junction has been formed between work-piece and tool surfaces, the junction area is likely to break inside the softer material, which generates least resistance.

The indirect transfer mechanisms were developed by recognizing the important

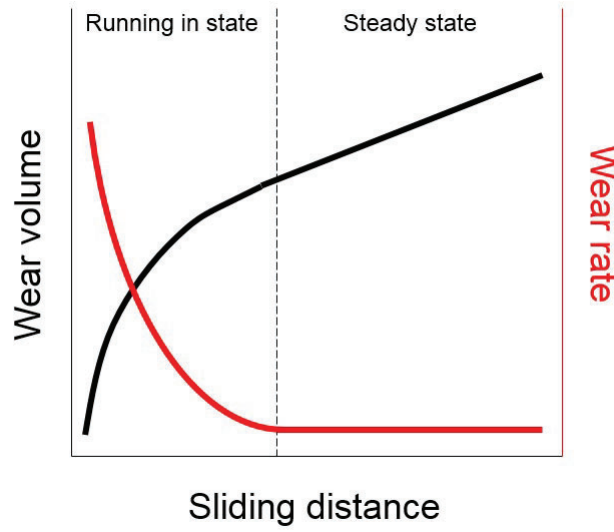


Figure 2.4: Typical relation of wear volume and wear rate *vs.* sliding distance with a transition from running-in to steady state in an adhesive wear tribo-system.

role of free wear particles and their effect on the tribo-system. In this case, the wear particles may freely exist at the contact interface and transfer to the counterpart with the help of external forces during sliding (Godet, 1990, Berthier, 2005). Otherwise, the wear particles may finally be ejected. On the counterpart, the transferred wear particles may also back transfer to the original bulk material (Dwivedi, 2010, Heilmann et al., 1983, Deuis et al., 1997, Riahi and Alpas, 2007, Sawyer et al., 2014).

The material transfer process is triggered by both mechanical and chemical resistance.

Mechanically, the tool surface condition is of great importance in the initial transfer process. In the wear process, in general, a certain range of the surface roughness increases the wear rate; in the transfer process, the rough surface triggers the transfer process by entrapping wear particles. In the work of Groche and Köhler (2006), cross-sectional pictures showed that aluminium transfer lumps were likely formed at the spheroidal graphite defects on the cast iron surface with protruding features that activate aluminium transfer. Schedin (1994) reported that, in sheet metal forming, the tool surface defects were the initiation sites for the material transfer. Lumps, which were observed as the individual components of the transfer

layer, were built up during successive forming operations. Olsson et al. (2002) carried out a similar study by a punching test between stainless sheet and AISI M3:2. It showed that the initial pick-up of the work-piece filled up the surface texture of the punch consisting of microscopic valleys. De Rooij and Schipper (2001b) studied the severity of galling as a function of tool roughness for DLC coating against AA6016 T4. From the study, it was clear that a lower surface roughness reduced material transfer and it was also noticed that the transition from low material transfer to significant material transfer occurred within a narrow range of surface roughness value. Hanson et al. (2009) further examined the roughness effect on work-piece transfer of powder metallurgical tool steel against an austenitic stainless steel and aluminium alloy. A general conclusion was given that the risk of work-piece adhesion and transfer increased with the roughness of the tool material; however, it was still largely affected by the work-piece material and their oxidation process. Also, many patterns designed at the contact with high surface roughness can also reduce material transfer by storing wear particles (Bowden and Tabor, 2001).

Chemically, the transfer process is affected by material pairs as well as the tribo-materials that are generated by the mechanical/chemical reaction. Many commonly formed work-piece materials were reported causing galling on tool steels used in sheet metal forming, including carbon steel (Landheer and Zaat, 1974, Schedin, 1994, Okonkwo et al., 2016), stainless steel (Schedin, 1994, Hanson et al., 2009, Okonkwo et al., 2016, de Rooij et al., 2013) and aluminium alloy (Heinrichs and Jacobson, 2011, Heinrichs et al., 2012, Schedin, 1994, Hanson et al., 2009, Olsson et al., 2004). According to Rabinowicz (1965), the possibility of galling between contact pairs can be explained by the influence of metallurgical compatibility, which was the degree of intrinsic attraction of the atoms between the contacting metals. A high metallurgical compatibility indicated a high possibility of forming strong metallic bonding. Bhushan (2013) stated that when high contact load was applied to soft metals like aluminium and copper, plastic deformation occurred at the contact region that led to small separations and formation of metallic bonds between the work-piece and tool. Dwivedi (2010) made a similar conclusion, where surface peaks

were known to deform under external load and increased the real area of contact, which in turn increased metallic intimacy.

The composition of the reinforcement additions of the work-piece material may play a minor role on influencing adhesive wear behaviours. Deuis et al. (1997) reviewed the effect of the addition of silicon in aluminium alloy and the size and fraction of reinforcement in aluminium-based composites. Generally, the addition of silicon in aluminium alloy improved the wear resistance and an optimum percentage of silicon that provided lowest wear rate was obtained. However, Dwivedi's review (Dwivedi, 2010) concluded this influence of microstructure on wear behaviour was very much debatable because phases lost their individuality as they approached sliding interface and a mechanically mixed layer was formed and the role of grain refinement, modification and heat treatment was not yet decisive.

The chemical composition of the generated tribo-material is usually more complicated than the original contact pair, which may change in both micro-structure and chemical composition. Kim et al. (2007a, 2007b) studied the structure and composition of aluminium wear debris generated by dry sliding on a pin-on-disc tribometer against 440C stainless steel at room temperature. The structural and chemical analyses of the wear particles suggested that the mechanical mixing lead to the formation of an aluminium-oxygen solid solution and hydroxylated material. It should be noted that these constituents are not normally formed in the absence of external forces and do not have an equilibrium structure and composition.

Anti-galling coatings were introduced with the purpose of separating the original contact pair and could partly reduce galling (Heinrichs and Jacobson, 2011, Heinrichs et al., 2012, Dong et al., 2017). It was found that the galling formation process also existed in coating system by similar mechanisms with tool steel. Heinrichs et al. (2012) conducted in-situ experiments of sliding contact by aluminium alloy (AA) 6082 against DLC coating and tool steel. Immediate pick-up of aluminium has been found in AA6082 against tool steel, compared to a slower build-up process of aluminium transfer layer in AA6082 against DLC. In this paper, with the help of in-situ microscope, detailed physical transfer mechanisms were observed and three

transfer stages were identified:

- Primary transfer
- Secondary transfer
- Damage activated transfer

These stages were both observed in coated and uncoated contact. With the stage transition, the COF increased and finally stabilized with an initial COF=0.16 and a saturated COF=0.42. Hu et al. (2017b) examined the galling phenomenon between AA6082 against coatings of CrN and AlCrN. The primary, secondary and damage activated transfer were observed in these coating systems, which were triggered by the minor coating defects. Dong et al. (2017) studied the anti-galling property of CAPVD WC:C coated tool against AA6082 in a U-shape bending test. It showed that WC:C coating had a much lower adhesion rate compared to the uncoated G3500 tool steel. The evolution of COF of WC:C against aluminium was relatively flat, which indicates both a low severity and a rapid saturation of galling were formed at the contact.

Sliding distance effect

It is well observed that galling leads to historical variations of friction and wear at the contact. That is, the roles, 1) transmit tangential force, 2) trigger further wear and transfer and 3) accommodate velocity difference, are played by the transfer layer and wear particles (third body) instead of the original contact pair (Berthier, 2005). Corresponding to the transition mode of friction and wear, the build-up of a transfer layer experiences a transition from a running-in to a steady state as sliding distance increases.

A friction increase due to galling is widely reported. With the generation of the transfer layer, the friction mechanism transitions from the original contact pair to a transfer layer contact and results in high adhesion and mechanical resistance. In (Olsson et al., 2002), friction increase with increasing stroke number was reported

between stainless steel work-piece and tool steel in punching test. A slow down to stable trend of the evolution of friction coefficient was observed. Dong et al. (2017) showed the results of friction coefficient gradually increasing from 0.05 to a stable value of 0.45 in the tribo-systems of AA 6082 against cast iron G3500, coating with PN and PNC. Heinrichs et al. (2012) conducted in-situ experiments of sliding contact by AA 6082 against DLC coating and tool steel. The COF increased and finally stabilized with the formation and saturation process of the transfer layer. An initial $\text{COF} = 0.16$ and a saturated $\text{COF} = 0.42$ were observed. This friction increase process with the build-up of the transfer layer is shown in Fig. 2.5. Van der Heide et al. (2001) conducted friction tests between low carbon steel sheet and tool steel and obvious galling was formed under lubricated conditions. The increase of COF was recorded from an initial stable regime of 0.11, which was under the protection of lubricant, to a high unstable regime of 0.3 connected by a transition regime. This result showed a complex tribological condition transition, where galling was formed due to lubricant breakdown. It also showed that the application of lubricant could extend the running-in stage of galling thus extending the safe sliding distance and tool-life.

The wear mechanism of galling is a combination of adhesive and abrasive wear. With the formation of transfer layer, wear mechanisms that generated between the asperities of the work-piece and tool are partly replaced by cohesion (or self-adhesion) and abrasion between the work-piece and soft transfer layer. The tribo-system tends towards an equilibrium with active self-mating by flatting and polishing and a transition from severe wear to mild wear occurs. This transition largely reduces the wear rate (de Rooij et al., 2013, Landheer and Zaat, 1974, Amiri and Khonsari, 2010, Heinrichs et al., 2012, Deuis et al., 1997) to a stable value and thus slows down the increase of wear volume to a linear relation that can be predicted by the Holm-Archard equation (Burwell and Strang, 1952).

The build-up of the transfer layer from a running-in to a steady state can be characterised on micro and macro scale. On micro-scale, for material transfer on a single asperity, de Rooij et al. (2013) suggested that the growing transfer lump

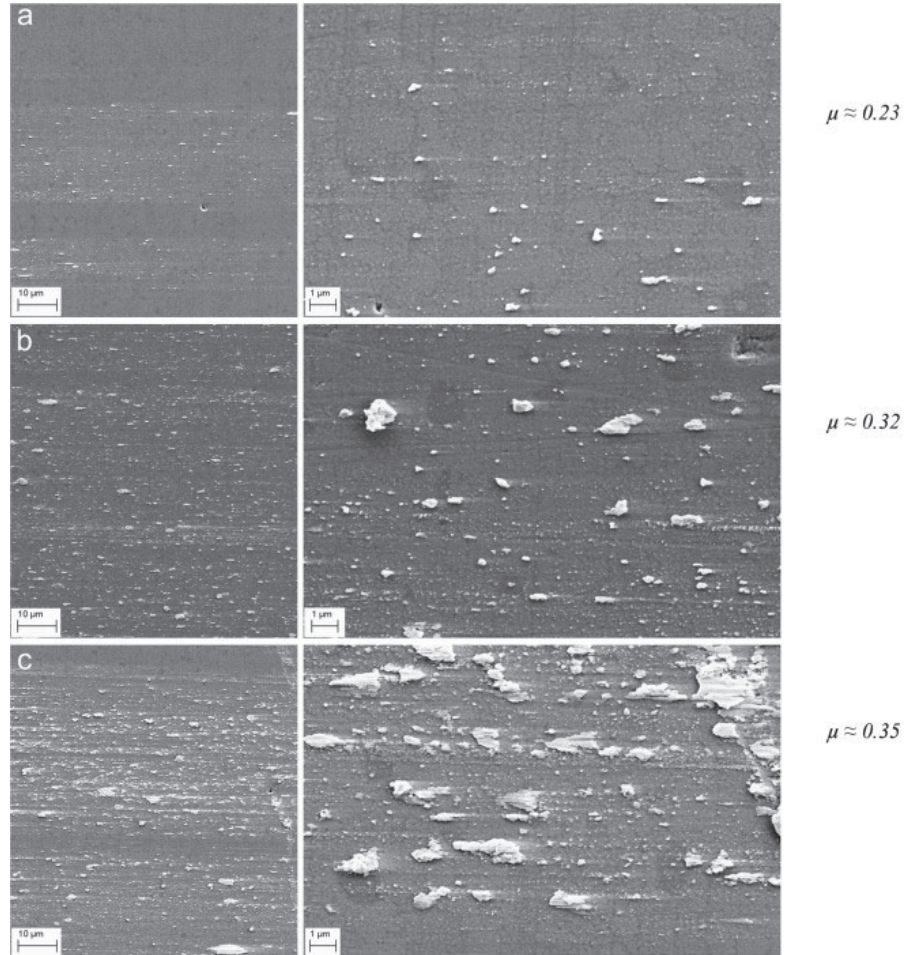


Figure 2.5: Appearance of polished DLC coating applying on Vancron 40 tool steel disc after different number of tip passages (from left to right), resulting in increasing number and size of transferred fragments. The corresponding friction levels are given. SEM with acceleration voltage of 3 kV. (a) 1 passage, (b) 5 passages and (c) 10 passages (Heinrichs et al., 2012).

should be able to withstand the stress acting on it with increasing length and height. If the continuous growth made it unstable, material was removed from the lump to make it stable again. On macro-scale, de Rooij and Schipper (2001b) conducted material transfer tests of aluminium alloys and tool steel with coated DLC on a deep drawing simulator. It was observed that the lumps grew to a limited size. Deus et al. (1997) reported that a critical transfer layer thickness existed for a given sliding situation. Rabinowicz (1965) presented a theory of transfer competition between the original contact pair. It was suggested that there were different transfer rates of the contact pair; for the material with a preponderant high transfer rate, it would generate a high fraction of transfer area on the counterpart at the steady stage. By combining the phenomenon of friction, wear and material transfer, the evolution of general mechanisms transition can be plotted as shown in Fig. 2.6. In the running-in state, the COF increased and wear rate decreased corresponding to the formation of the transfer layer; in the steady state, the COF and wear rate become stable corresponding to a dynamic equilibrium when the transfer layer is formed.

Contact load effect

It is well known that the increase of contact load at the die/work-piece interface will increase the severity of galling and reduce tool life. The increase of contact load increases the friction energy inputted at the contact and thus increases the severity of both mechanical anchor and chemical diffusion bonding (Amiri and Khonsari, 2010). In macro-scale observations, the increase of contact load will:

- Shorten the duration of the running-in stage
- Generate a high severity transfer layer

The shortened duration of running-in was observed in Podgornik et al. (2004). In this study, the comparison of the COF evolutions *vs.* number of strokes under different contact loads was conducted. In the evolutions under 150 N, the running-in stage was identified as a plateau of low COF value ($=0.13$) which was followed by a rapid increase that was caused by galling. With the increase of contact load to 1300

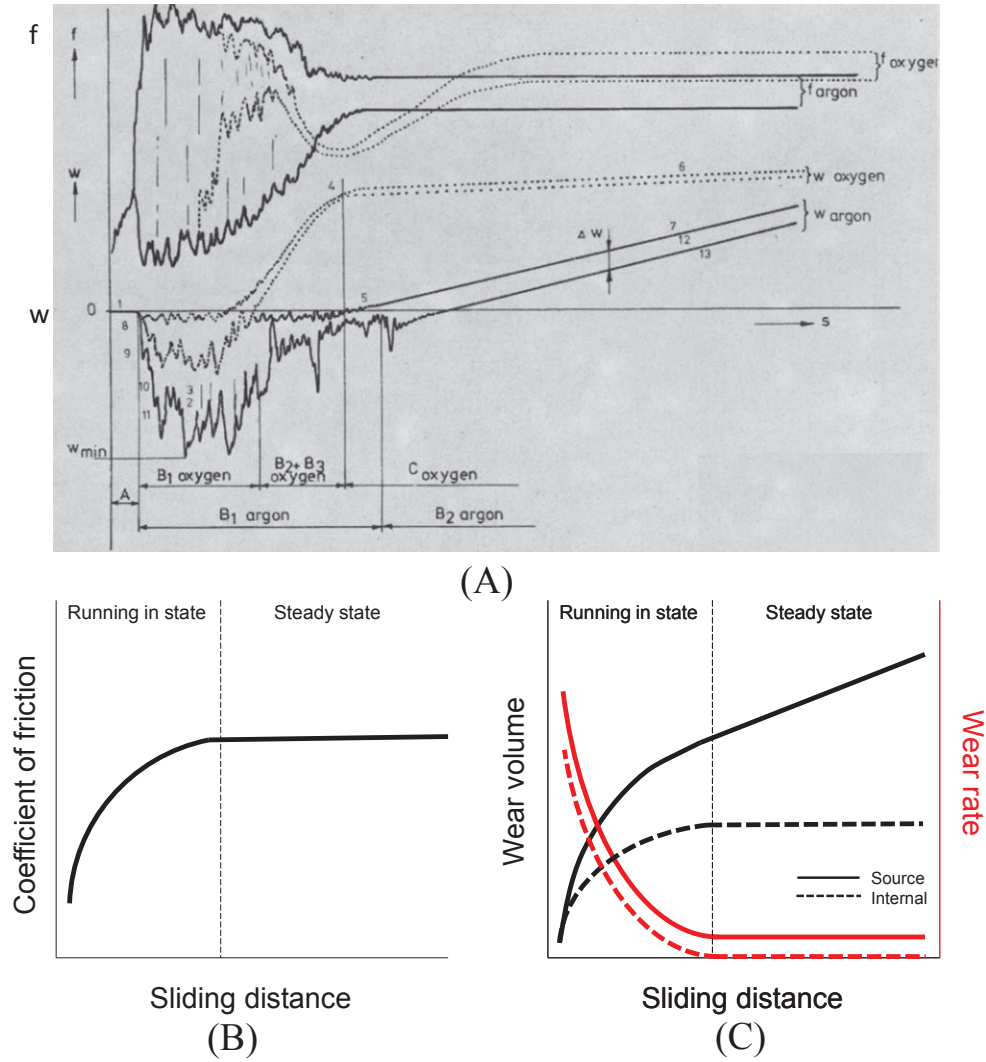


Figure 2.6: (A) Experimental result of wear (w) and coefficient of friction (f) as a function of sliding distance (s) for annealed plain carbon steel specimen in argon and oxygen respectively (contact load: 50 N; sliding speed: 0.5 m/s) (Landheer and Zaat, 1974). Schematic diagram of (B) evolution of coefficient of friction, (C) wear and transfer volume and their wear rates of an adhesive wear tribo-system

N, the running-in state was shortened from 25 strokes to 5 (Fig. 2.7). The group of evolutions obtained in tool steel against lubricated austenitic stainless steel were compared with the group of evolutions obtained in DLC coated tool steel against the same counterpart. It showed that the contact pair of DLC *vs.* stainless steel was more sensitive of contact load, which presented a faster rate of shortening the running-in stage with contact load increase as well as a higher stable COF value (0.48 *vs.* 0.55). A similar result was shown in the study of Dong et al. (2017), in which the friction tests of AA 6082 against cast iron G3500 under different contact pressure were conducted. It was clearly shown that, under 1.8 MPa, the running-in stage is finished in approximate 6 strokes instead of 10 at 0.07 MPa.

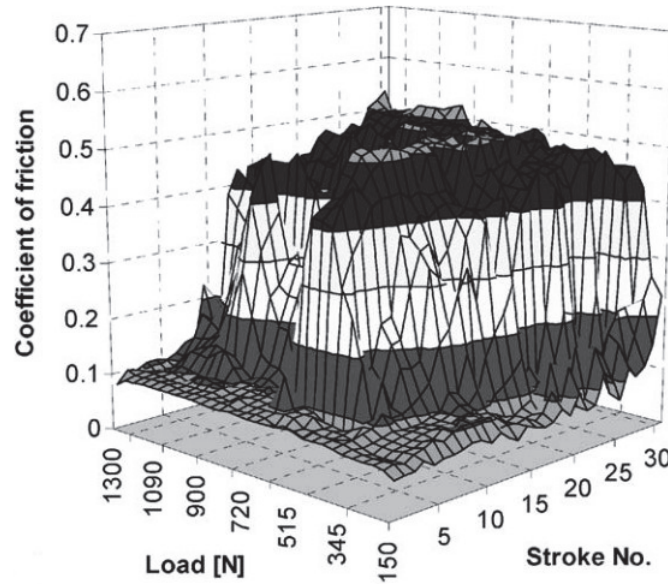


Figure 2.7: Representative friction maps for polished VANADIS 6 tool steel, sliding against PAO lubricated austenitic stainless steel (Podgornik et al., 2004).

The relation between severity of galling and contact load was studied by de Rooij and Schipper (2001b) between DLC coating and AA1050 MF (Fig. 2.8A). A rapid increase in range of galling severity was observed between 400 N and 500 N. This was explained by a transition from the ploughing regime to the wedge-formation regime on the tool surface, where the attack angle of the asperities of the tool surface increased with increasing load. The phenomenon that galling rapidly increases in a

certain range of contact pressure was also observed in (Wen and Huang, 2012). A similar dependency between the transfer layer and applied pressure was reported in (Österle et al., 2012, Podgornik et al., 2004). In (Österle et al., 2012), a proportional relation between the transfer layer thickness and contact pressure was obtained by numerical simulation. In (Podgornik et al., 2004), a proportional relation between the galling area and contact load was obtained by experiment (Fig. 2.8B).

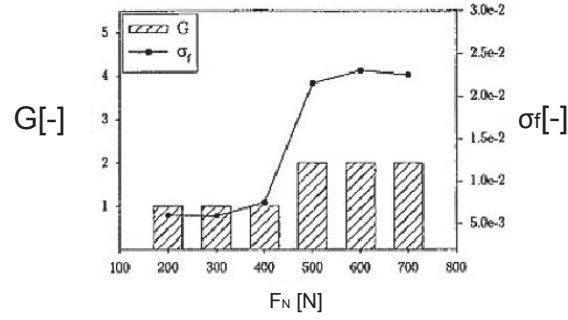
Sliding speed effect

According to the classic Coulomb's model, the effect of sliding speed is negligible between metal contact. However, it was found that galling was sensitive to sliding speed in different speed ranges. Liu et al. (1991) generated a map for wear mechanisms in aluminium alloys (Fig. 2.9). The galling phenomenon was described as melt wear that mass transfer from the aluminium surface to a harder substrate, which was likely occurred at relatively high speeds and high loads. Okonkwo et al. (Okonkwo et al., 2016) studied the effect of sliding speed on the wear of steel-tool steel pairs using the pin-on-disc test. It was found that the adhesive wear rate was very sensitive to sliding speed at slow speed conditions (<35 mm/s) but relatively insensitive during higher speed conditions. It was concluded that at a low sliding speed, material transfer was dominated by adhesive wear mechanism; whereas at a high sliding speed resulting in high frictional heating, the effect of increasing bulk temperature played an important role on material transfer and resulted in a transition from adhesive wear to material removal-dominated mechanisms (Okonkwo et al., 2016).

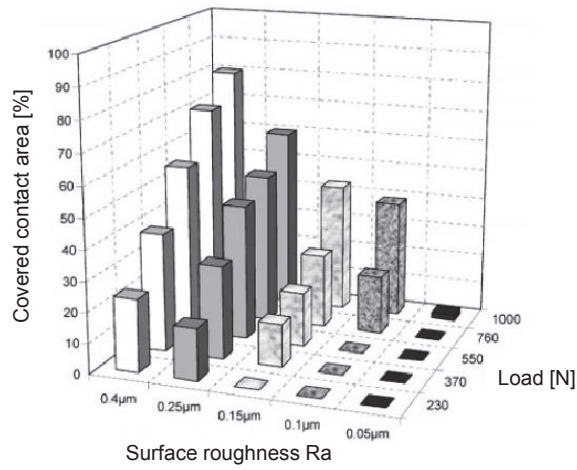
Interaction between friction and wear

As both the variation of friction and wear are related to the formation of the transfer layer, an interaction between them is built (as shown in Fig. 2.6), which has been predicted and observed in many literature studies but surprisingly attracts limited attention for further investigation.

The third body theory was developed to provide a holistic approach to study



(A)



(B)

Figure 2.8: (A) The severity of galling, indicated by G and σ_f , as a function of applied normal force F_N for DLC coatings sliding against AA 1050 MF material, $v=0.005$ m/s (de Rooij and Schipper, 2001b). (B) Galling intensity map showing the proportion of the VANADIS 6 contact area covered by transferred stainless steel as a function of the initial surface roughness of the tool steel and contact load (Podgornik et al., 2004).

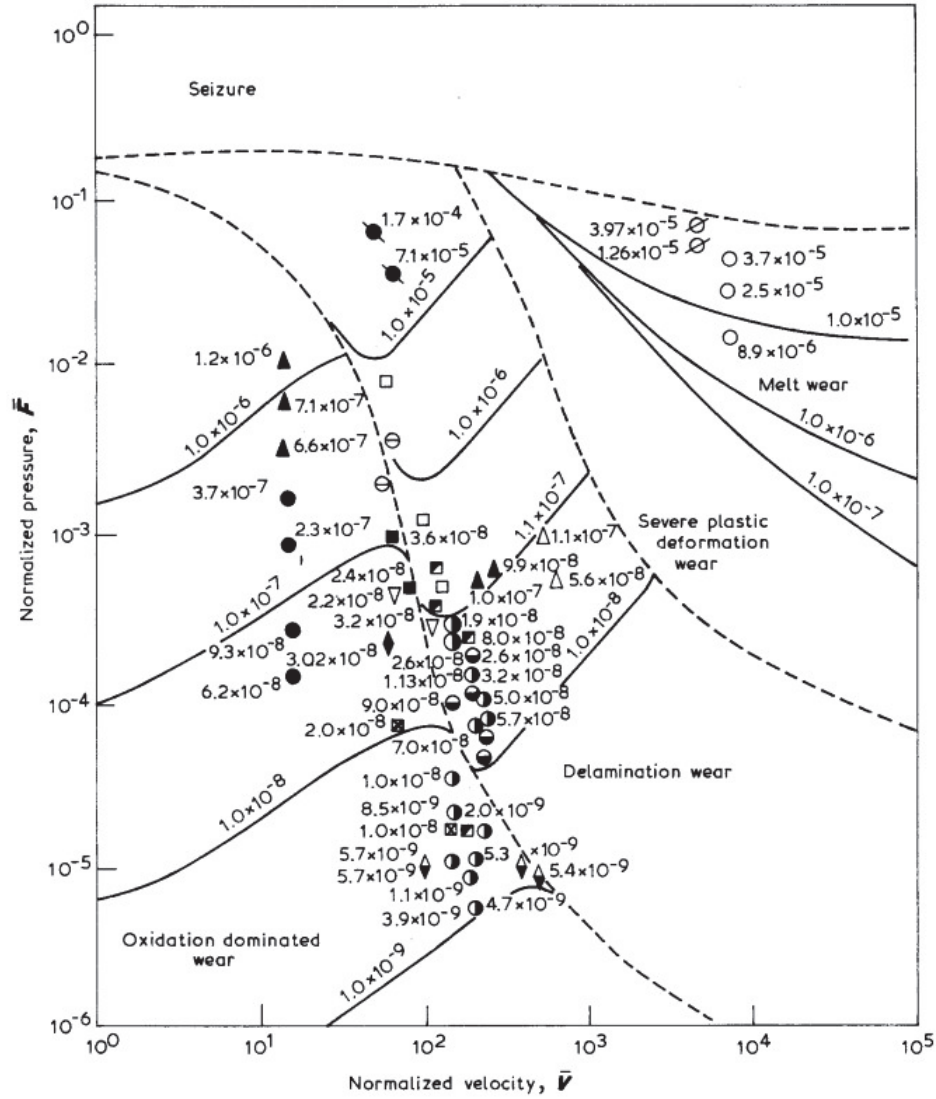


Figure 2.9: A map for wear mechanisms in aluminium alloys. The numbers in the figure represent normalized wear rate. Galling is referred to melt wear located in the high velocity and pressure zone (Liu et al., 1991).

the galling phenomenon by focusing on the role of the interface on friction and wear. Godet (1990) defined the third body by two senses. In a material sense, the third body is a zone, which exhibits a marked change in composition from the original contact pair, i.e. first body. In a kinematic sense, it could be defined as the thickness across which the difference in velocity between solids is accommodated. The third body theory provides a global view of the whole tribo-system including the original contact pair and the generating of transfer layer and wear particles at the interface, thus making it a powerful approach to analyse the adhesive wear tribo-system. Berthier et al. attempted to develop a global adhesive wear mechanism based on the third body theory. In (Descartes and Berthier, 2002), they introduced the rheology and flows of solid third bodies in the application of MoS_{1.6} coating to enable the analysis of a dynamic structure formation process. This work was adopted into a series of work including later adhesive wear analysis. They further developed the modelling of third body flows. In this work, the relation between the particle detachment mechanism, the flow of these particles inside the contact and their ejection from the contact was represented by a mass equilibrium equation. Fillot, Iordanoff and Berthier (Fillot et al., 2007) simulated the dynamic process of adhesion between the detached particles and the running surface. The wear process was considered by three material states: the detachment of particles, their flow at the contact interface and their ejection. General evolutions of the three states of materials *vs.* time were obtained with the effect of adhesion force and ejection process. It was found that 1) a thicker third body layer was formed as particles adhesion increased and 2) an immediate ejection at the contact would provoke severe detachment and ejection, which resulted in a thin third body layer. Diomidis and Mischler (2011) studied the third body effects of fretting, which is defined by Budinski (2007) as oscillation motion of small amplitude, of a steel contact. It was found that the generation of the transfer layer could affect both the frictional and wear response. The running in state and steady state were recorded in the COF evolutions with an initial low COF=0.6 and a stable COF=0.8. In addition, the running-in duration decrease with increasing pressure was observed in this work.

2.2.2 Modelling technique of galling

The running-in state that was caused by the formation of transfer layer was the focus of Yang's adhesive wear model (Yang, 2005, Yang, 2003). In the running-in state, it was assumed that the rate of volume removed per unit sliding distance was a function of the volume of metal available at the junctions. The integration of wear volume is shown in Eq. 2.2. This model adopted the Holm-Archard equation to model the steady state and thus the whole wear evolution could be predicted.

$$V_t = A(1 - \exp^{-BL}) \quad (2.2)$$

In this model, V_t denotes the wear volume at the running-in state, L is the sliding distance, A and B are constants that are determined by contact load and surface conditions.

Fillot et al. (2007) developed a wear model (Eq. 2.3) based on the third body concept that modelled the wear volume variations from the running-in to the steady state. In this model, an equilibrium of the mass inside the contact is developed as

$$\frac{dM_i}{dt} = Q_s - Q_w \quad (2.3)$$

where Q_s is the flow of third body detached from the first body, Q_w is the wear flow of third body ejected outside the contact and M_i is the mass of third body entrapped inside of the contact. It could be seen that the wear process constituted a form of competition between the source flow, Q_s , and wear flow, Q_w , which were modelled respectively by Eq. 2.4 and 2.5. This model successfully modelled the variation of the volumetric changing rate of detached, entrapped and ejected wear particles from the running-in state to the steady state.

$$Q_s = C_s(M_i^{\max} - M_i) \quad (2.4)$$

$$Q_w = C_w(M_i - M_i^{\text{start}}) \quad (2.5)$$

In Eq. 2.4 and 2.5, C_s and $M_i^{\max}$ are constants of the experiments. M_i^{start} is the mass of third body entrapped in unevenness of the surfaces and C_w is a function of ejection speed of wear particles.

Models that focused on the growth of transfer layer were also developed. Based on the experimental results of aluminium alloy-DLC contact (de Rooij and Schipper, 2001b), de Rooij developed a lump growth model (de Rooij and Schipper, 2001a) for general macro-scale applications. This model focused on the modelling of summit height distribution of the tool surface in the transfer layer build-up process. The transfer mechanism was modelled by hard wedges, moving through a soft and plastically deforming surface. The model showed that the physical contact behaviour between tool and sheet would change during lump growth because of a height distribution change. Instead of a continuous transfer layer, Rooij et al. (2013) developed a material transfer model on a single asperity scale and presented the geometrical development of a single lump over time. The shape of the lump was modelled as mechanically stable and thus could resist the forces acted on it. The orientation of the lumps and the multi-layer structure were also considered in this model.

Hol et al. (2012) developed a macro-scale friction model for sheet metal forming, in which the effect of surface roughness and deformation under pressure were taken into account. In this model, the real area of contact was used to determine the influence of ploughing and adhesion effects between contacting asperities on the COF. A statistical approach was then adapted to translate the microscopic models to a macro-scope level enabling the friction model to be implemented into a FE code and applied to a full-scale sheet metal forming simulation. The proposed friction model was able to predict the relation between the interface change and friction variation although did not take into account the galling phenomenon.

2.3 Lubricant breakdown in cold stamping

2.3.1 The mechanism of lubricant breakdown

Lubricant is applied during aluminium stamping to reduce the contact between workpiece and tool and thus reduce the wear of them. A suitable lubricant can be chosen by the operation condition. The traditional lubricant for sheet metal forming is widely used in aluminium stamping process, e.g. mineral oil, synthetic

oil, polymer solution with additives for special requirement. The viscosity under the forming condition is a key factor of lubricant. During the forming process, the viscosity of lubricant is mainly influenced by temperature, contact pressure, shear stress etc. A suitable viscosity can provide both low friction and protection the surface at the contact and choosing lubricant with anti-wear and good load-carrying capacity can reduce galling due to lubricant breakdown in aluminium forming.

Lubricant breakdown is caused by a shortage of lubricant at the contact interface, which can be affected by the operation parameters (including contact load and sliding speed) and contact conditions (including surface roughness and lubricant properties). With the thinning of the lubricant layer, direct contact between work-piece and tool occurs, which results in surface scratches and friction to increase. In a situation with lubricant transport, the initial lubricant amount and sliding distance can also affect the film thickness. In most forming cases, a moderate quantity of lubricant applied between the work-piece and the tool can provide a separation barrier during the metal forming process (Foster et al., 2008, Wilson, 1978). This amount of lubricant typically involves the assumption of ideal hydrodynamic lubrication conditions with a low COF and little wear due to moving objects. However, in many situations, it is not always possible to maintain ideal hydrodynamic lubrication conditions and there may be considerable levels of non-hydrodynamic lubrication, e.g. boundary lubrication, that results in galling or wear on tooling and the product. This is especially essential for sheet metal forming processes, where the transportation of lubricant is uneven due to the non-uniform distribution of relative sliding distance, strain and contact pressure at the work-piece-tooling interfaces. Moreover, in many cases, the lubricant is squeezed out towards regions with lower pressure and side leakage occurs, which will cause further loss of lubricant from the contact and lead to lubricant film breakdown. Therefore, an adequate quantity of lubricant applied prior to the forming operation does not guarantee that lubrication will be effective at all locations or at all stages of a forming operation (Bowden and Tabor, 2001, Wilson, 1978).

The lubricating condition can be classified into three general conditions by

the lambda ratio (λ =mean film thickness/composite standard deviation of surface heights of the two surface) (Bhushan, 2013, Wen and Huang, 2012, Holmberg and Matthews, 1994), they are:

- Hydrodynamic lubrication ($\lambda > 5$, typical thickness: 1-100 μm)
- Mixed lubrication ($1 < \lambda < 5$, typical thickness: 0.05-1 μm)
- Boundary lubrication ($1 < \lambda$, typical thickness: $< 0.05 \mu\text{m}$)

The variations of these effect factors will lead to the transition of the lubricating conditions and simultaneously influence the friction and wear conditions.

The Stribeck curve phenomenon (Holmberg and Matthews, 1994, Bhushan, 2013) (Fig. 2.10) connects the lubricant condition transition with the variation of coefficient of friction. The effect of operation parameters is described by the Hersey number $\eta U/P$, where η is the lubricant viscosity, U is the sliding velocity and P is the average contact pressure. The decrease of contact pressure and increase of sliding speed will lead to an increase of the lubricant film thickness, which reduces the surface asperity contact caused by the enhancement of the elastohydrodynamic effect, where the elastic deformation of the contacting solids plays a significant role in the lubrication process. When the interface gradually enters the full-film hydrodynamic lubrication regime, friction may reach its minimum and then slowly rise due to the increase in lubricant shear strain rate in the film. It was found that the prediction of the thinning trend of the lubricant film by the Hersey number had wide engineering applications including stamping processes (Gelinck and Schipper, 2000, Haar, 1996).

The lubricant film thickness under elasto-hydro lubricant conditions can be numerically solved by Reynolds equation in the hydrodynamic lubrication in sheet metal forming. Based on the numerical calculation, Dowson and Higginson (2014) proposed a minimum film thickness formula in line contacts as Eq. 2.6

$$H_{min}^* = \frac{2.56 G^{*0.54} U^{*0.7}}{W^{*0.13}} \quad (2.6)$$

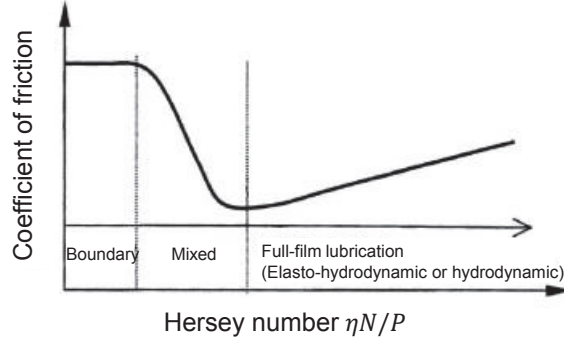


Figure 2.10: Schematic of the Stribeck curve (Zhu et al., 2015).

where $H_{\min}^*$, G^* , U^* and W^* are dimensionless parameters of minimum film thickness, material, velocity and load and are presented in Eq. 2.7

$$\begin{aligned}
 H_{\min}^* &= h/R \\
 G^* &= \alpha E^* \\
 U^* &= \eta_0 U / E^* R^* \\
 W^* &= W / E^* R^* l
 \end{aligned} \tag{2.7}$$

where h is the film thickness, R^* is the radius of the contact, α is the pressure exponent of viscosity, E^* is the elasticity modulus for the solid materials at the contact, η_0 is the atmospheric viscosity, W is the contact load and l is the length of the line contact. It should be noted that the conventional Reynolds equation is suitable for analysing lubrication problems with relative smooth surfaces under hydrodynamic lubrication condition but it can hardly be applied to critical conditions in sheet metal forming, where lubricant transport is influenced by lubricant amount, sliding distance and surface roughness (Yang and Lo, 2004) and lubricant breakdown may occur under critical conditions.

The F_n - v Transition diagram of lubricant condition (Fig. 2.11) is another method of connecting the thinning of lubricant film with friction and wear. Salomon (1976) studied the failure criteria in thin film lubrication and concluded the studies into the lubrication condition transition diagram. In the study, the interaction between friction and wear were connected by the thinning of lubricant layer and thus increased possibility of direct contact between the rubbing pair resulted in increased severity

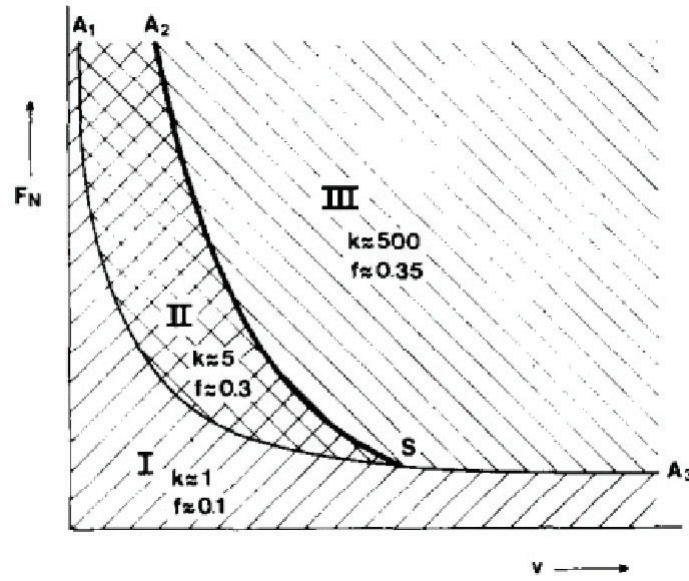


Figure 2.11: F_N - v Transition diagram of first, second and third primary transitions at constant oil bath temperature. f : friction coefficient; k : wear rate (Salomon, 1976).

of wear. The F_N - v diagram of contact load *vs.* sliding speed was drawn, and three stages were presented in the diagram:

- Stage I, partial or full lubrication
- Stage II, boundary lubrication
- Stage III, scuffing or pre-seizure due to complete failure of lubrication

It showed that either increasing the sliding speed or the contact load will lead to the increase of friction and wear. Schipper and de Gee (1995) studied the F_N - v transition diagram and proposed that, in reality with high sliding speed, the pronounced decrease in load carrying capacity with increasing speed of sliding was due to the effect of frictional heating in the contact zone causing an appreciable decrease in effective lubricant viscosity. Begelinger and De Gee (1982) presented a detailed F_N - v diagram, in which it showed that the load-carrying capacity gradually increased with sliding speed until a critical speed at 2 m/s was reached, after which the capacity would decrease.

Lubricant transportation is another reason that causes a shortage of lubricant. A certain amount of lubricant will breakdown during the transportation process with the thinning of the lubricant film. Kingsbury (1978) studied the effect of the initial amount of lubricant on lubricant breakdown on a ball-on-disc tester. It showed that the effect of lubricant quantity diminution was to decrease the running time before lubricant breakdown during lubricant transportation.

2.3.2 Modelling technique of lubricant breakdown

Wilson et al. (1995) developed friction models for sheet metal forming under lubricant condition with the consideration of lubricant breakdown. The friction under hydrodynamic lubrication condition was calculated by Reynolds equation (Wilson and Hector, 1991, Hsu and Wilson, 1994, Hector and Wilson, 1991). The transition of lubrication was determined by comparing the calculated lubricant film thickness to the surface roughness, σ . Once the lubricant film thickness was lower than 3σ , the lubrication condition was transferred to mixed lubrication condition, in which the frictional stress is calculated by Eq. 2.8:

$$\tau_f = \tau_a A_f + \tau_p A_f + \tau_h (1 - A_f) \quad (2.8)$$

where A_f is the fractional contact area and τ_a , τ_p , τ_h are the adhesion, ploughing and hydrodynamic friction stress components, respectively. This model was applied to model a simple stretch forming process.

Gelinck and Schipper (2000) developed a mixed lubrication model for line contact by which Stribeck curve can be calculated. By the interaction, the transitions from boundary lubrication to mixed lubrication and further to elasto-hydrodynamic lubrication could be predicted and subsequently the lubrication regime in which a particular contact operates can be predicted. In this model, the friction force was generated by integral of the shear stress at the asperity contact and the hydrodynamic components (Eq. 2.9).

$$F_f = 2B'b'\tau_0 \operatorname{arcsinh}\left(\frac{\eta u_{dif}}{\tau_0 h}\right) \quad (2.9)$$

In this equation, F_f is friction, B' is length of cylinder, b' is the half contact width, τ_0 is eyring shear stress, η is viscosity, u_{dif} is differential velocity and h is the film thickness.

Kim et al. (2012) applied a COF model in the modelling of cup drawing, which was a function of contact pressure, sliding velocity and sliding direction. The model was calibrated by experimental data obtained from a draw-bend friction test and implemented into FE simulation. Circular cup drawing experiments and simulations were performed to validate the methodology. Kim's friction model was based on the modelling of a Stribeck curve (Lugt, 1992), which can be described as the following arctangent function of lubricant number L' (Eq. 2.10):

$$\mu(L) = 0.5((\mu_{BL} + \mu_{EHL}) + (\mu_{BL} - \mu_{EHL})\tanh(\frac{\lg(L'^2/L_{BL}L_{EHL})}{\lg(L'/L_{BL}L_{EHL})})) \quad (2.10)$$

where μ_{BL} and μ_{EHL} are the parameters decided by experiments. The lubricant number L is a function given as Eq. 2.11.

$$L = \frac{\nu\eta}{PR_a} \quad (2.11)$$

where ν is the sliding velocity, P is the contact pressure, η is the lubricant viscosity and R_a is the surface roughness.

Lee et al. (2015) developed a constitutive friction model for the springback analysis of the advanced high strength steel sheet forming. The experimental result showed that an increase of the sliding velocity led to a decrease of the COF, rapidly at low velocities and rather slowly at high velocities. Also, the friction coefficient tended to increase with increasing contact pressure up to 10 MPa, after which, it decreased and saturated at higher contact pressures. These relations were modelled by a phenomenological model, in which it was assumed that the contributions of sliding velocity and contact pressure were multiplicative (Eq. 2.12).

$$\mu(v, p) = \mu_v(v)\mu_p(p) \quad (2.12)$$

Then, $\mu_v(v)$ (Eq. 2.13) and $\mu_p(p)$ (Eq. 2.14) can be characterised using the obtained experimental data in

$$\mu_v(v) = a_1 + a_2 \exp(-a_3 v) \quad (2.13)$$

$$\mu_p(p) = b_1(1 - \exp(-b_2p)) \quad (2.14)$$

where a_1 , a_2 , a_3 , b_1 and b_2 are model constants.

Tamai et al. (2016) developed a nonlinear friction model that considered contact pressure, sliding velocity and sliding length. In the experiment, it was found that the friction coefficient decreased with increasing sliding velocity. With long sliding distance, the lubricant in the sheet surface concavities gradually decreased and resulted in a shortage of lubricant. The frictional condition became dry friction and led to an increase of COF. Based on the experimental results, the friction coefficient can be calculated by Eq. 2.15

$$\mu(p, V, L) = a \ln(V) + b \ln\left(\frac{p}{YP} 10^3\right) + c + dL \quad (2.15)$$

where a , b and c are model constants and YP is yield stress of the sheet material. L is the sliding length.

Wang et al. (Wang et al., 2016) developed a pressure-dependent variable friction model for the sheet forming of advanced high strength steel. The model was calibrated by experimental results obtained on full lubricated condition, which showed a decrease of friction coefficient with the increase of contact pressure and thus it may not be suitable for dry and mix lubrication conditions. This trend of COF *vs.* contact pressure was expressed as an exponential function as follows (Eq. 2.16):

$$\mu = \mu_0 \left(\frac{P}{P_0}\right)^{n-1} \quad (2.16)$$

where P_0 is the reference pressure and μ_0 is the friction coefficient at P . n is an experimental constant. The same model was also used in the study of deep drawing springback prediction by Gil et al. (2016).

2.4 Conclusions

The purpose of this chapter was to highlight the current research focus involving the tribological failures in cold stamping:

- Galling

- lubricant breakdown

and the interactive responses between friction and wear and their contributions to this tribo-system.

In galling, the built-up of transfer layer on the tool always leads to an increase of friction due to the high cohesion between the work-piece and the transfer layer, and a decrease of wear rate due to the hard tool asperities covered by relatively soft transfer layer.

In lubricant film breakdown, the thinning of lubricant film will lead to the tribo-system transferring to a dry sliding condition, in which friction increases due to the increased percentage of metal/metal contact, and wear rate increases due to the direct contact between contact pairs.

The effects of operation parameters including contact pressure, sliding speed, sliding distance and lubricant amount are especially reviewed. For galling, the material transfer layer is formed in a certain sliding distance, which can be shortened by increasing of contact pressure. The transfer volume is also increased with contact pressure. For lubricant breakdown, high contact load and low sliding speed will lead to lubricant breakdown. In the transportation of lubricant, the increase of lubricant amount can delay the breakdown.

Although the variation of COF due to galling and lubricant breakdown has widely been observed in lab-scale tests, it is still hard to be implemented into FE software. It is because 1) the historical effect of galling under dynamic loading condition/between different forming cycles are still unknown and 2) there is still no galling model that addresses operation parameters of stamping.

The lubrication condition in stamping is relative better understood. However, in many stamping lubrication models, it is still assumed that the lubrication condition is under hydrodynamic or mixed lubrication condition, which makes the COF varies in a low range and cannot address the consequence tribological failures, e.g. galling due to lubricant breakdown.

As the tribological condition becoming a challenge to aluminium stamping, there is an urgent requirement to develop a holistic friction model that can address the

distribution of evolutionary tribological failures under a dynamic multi-cycle condition. Focusing on these aims, interactive friction models has been developed and the dependencies of operation parameters and forming history will be addressed in the holistic modelling of galling and lubricant breakdown in the later chapters.

Chapter 3

Experimental methods for quantitative study of the evolutionary process of galling and lubricant breakdown

3.1 Summary

Based on the literature review of chapter 2, adequate test methods and characterizing approaches should be chosen to investigate the questions raised in chapter 1. This chapter presents the experimental work conducted on aluminium alloy and tool materials to study the evolutionary process of galling and lubricant breakdown. Firstly the materials that represent the workpiece and forming tools are presented. Secondly, the experimental setup and key procedures to characterise the coefficient of friction are introduced. From these tests, the safe sliding distance before the tribological failures could be determined under different contact conditions .

3.2 Test materials

The material used for tests was aluminium alloy (AA) 6082 in the T6 condition (solution heat-treated and artificially aged) due to its industrial potential and lubrication difficulty, which makes the material easy to adhere and be worn on tools (Fan and Wang, 2015). In dry sliding tests, cast iron G3500, which is a grey cast iron for stamping tools, was chosen as the material of the frictional counterpart. In lubricated sliding tests, the ball material used as the counterpart in the friction tests was Tungsten Carbide WC-6% Co ball due to its good abrasion resistance and low adhesion with aluminium as a potential coating material for aluminium forming (Dong et al., 2015). The lubricant used for the tests was a lubrication grease, OMEGA 35, made from polyethylene glycol, silicon dioxide and graphite. This lubricant features adequate performance in a high temperature environment application (up to 700°C).

3.3 Test methods

The experiments were performed on an Anton Paar tribometer (THT) (Fig. 3.1). The variables, contact load, sliding speed and sliding distance, can be controlled. Test standard ASTM G 99 for determining the wear and friction of materials during sliding using a pin-on-disc setup covers the general methods used in the present research. For the pin-on-disc test, two specimens are required. As shown in Fig. 3.2, a pin with a flat or radiused tip is positioned perpendicular to a flat circular disk. In THT, the disc specimen revolves about the disc centre horizontally. Weights were loaded on the pin specimen to press against the disc.

In dry sliding tests, the material used for the pin was aluminium alloy AA6082 T6 (flat contact surface with a diameter of 2 mm, $R_a = 486 \pm 35$ nm, hardness = 115 ± 5 HV) representing the workpiece material. The material used for the disc was cast iron G3500 ($R_a = 126 \pm 19$ nm, hardness = 209 ± 18 HV), the same material used for the forming tools. The hardness measurements were taken on each sample using Vickers hardness machine with 5kg load and 10-second dwell time (HV5/10). In lubricated sliding tests, AA6082 sheet at T6 condition with a thickness of 1 mm was



Figure 3.1: Anton Paar pin-on-disc test setup.

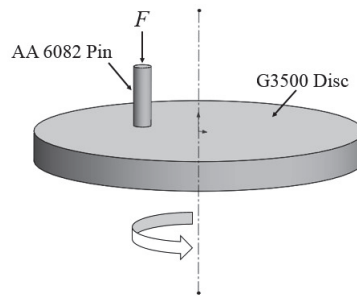


Figure 3.2: Schematic illustration of a pin-on-disc test.

utilised to produce the disc samples. The ball material used as the counterpart in the friction tests was Tungsten Carbide WC-6% Co ball, which was mounted on the top of the pin holder.

The test procedures of testing standard ASTM G 99 were followed for the pin-on-disc tests including the cleaning and drying procedure of the specimens prior to testing. Non-film-forming cleaning agent, acetone, was used to remove all dirt and foreign matter from the specimens. Surface roughness of specimens were measured before testing. The disc was securely inserted into the disc holder of the THT rig to be perpendicular to the axis of the revolution. The pin was adjusted perpendicularly to the disc surface in order to maintain the necessary contact condition. After testing, the loose wear particles near the wear track were carefully protected to further study.

Two lubricant application methods were used in the friction tests: 1) a precisely controlled quantity of lubricant was applied on the ball to simulate the non-

hydrodynamic lubrication (insufficient lubricant) condition and 2) lubricant was evenly applied to the disc to simulate a fully flooded lubrication condition. For condition 1), the lubricant was applied by a dedicated rig designed and manufactured by the authors' group with micro volume lubricant reservoirs. For the fully flooded lubrication condition, 100 mg of grease was applied on the disc's surface at a thickness of $500 \pm 50 \mu\text{m}$, which was measured by a plastic wet film comb (Elcometer 3238).

The detailed test conditions with adopted test method and test plan will be given in the later chapters.

3.4 Characterisation of galling and lubricant breakdown

The COF was calculated from the measured friction force, which can be read from the tribometer software. The wear rate was recorded using a linear variable differential transformer (LVDT) measuring the penetration depth or approach of the two surfaces during the experiment. The morphology of the cast iron surfaces was studied using a Hitachi S-3400N scanning electron microscope (SEM) and a Veeco WykoNT9100 white light interferometer (WLI). SEM provided micrographs of the transfer layer with high magnification and resolution. Cast iron discs were carefully transported and mounted rigidly to the holder SEM. WLI was used to measure the difference of height distribution before and after the test on the cast iron and aluminium specimens, which were caused by material transfer and wear. The galling area fraction covering the cast iron was calculated from the WLI 3D surface plot by height (Appendix A). To reduce the effect of the original surface on the calculation of galling area, 3σ method is applied. As the original surface is a Gaussian surface, after test, asperities with height of $3\sigma \approx 500 \text{ nm}$ above the mean surface were identified as the transfer layer. σ is the standard deviation of the original cast iron surface and the standard of 3σ makes 99.7% of the asperities on the original surface can exclude from the calculation. SEM with an in-situ focused ion beam (FIB) was

used to generate cross-section micrographs of the transfer layer and revealed the formation history. The formation process was indicated by COF and galling area variation, whilst the lubricant breakdown was indicated by coefficient of friction variation and wear. The severity of these phenomena was quantitatively studied by comparing the sliding distance.

Due to 1) the application of the interactive friction models developed in the present study focus on the tool wear and tool-life prediction and 2) significant difference between the pin used in the present work and the fresh work-piece applied in real forming, the adhesive wear/material transfer process are mainly studied instead of aluminium pin. As discussed in the present work, the main role of aluminium pin in the aluminium/cast iron tribo-system is the source of third body (generating aluminium wear particles), the wear volumes (wear depth) of aluminium under different condition of contact load become an important parameter and is presented and modelled by the Holm-Archard equation.

In real sheet metal forming, new sheet is applied at each forming cycle. However, in pin-on-disc test, the pin can hardly be changed during sliding. Instead, it was changed after each test. This result may overestimate the generation rate of aluminium wear particles at the contact comparing to real forming test. In the pin-on-disc test, the damage mode of pin is identified as secondary transfer mechanisms which is partly triggered by the damage of aluminium pin (chapter 4). In real forming test, the effect of secondary transfer could be reduced by fresh aluminium but a similar evolution would be expected due to the accumulation of aluminium on the tool surface.

Instead of conducting industry-scale forming test, the author tries to develop a new numerical method to solve the problem of accurate tool-life prediction. In the present study, experiments are designed to obtained quantitative responses (friction and galling area) from operation parameters (contact pressure and sliding distance/speed), whose ranges were obtained in real forming conditions. The differences caused by test rig can be calibrated by fitting the interactive friction model with experimental data obtained in industry-scale forming test in the future work.

Chapter 4

Development of an interactive friction model for the prediction of galling behaviour during sliding wear

4.1 Summary

¹ This chapter presents the investigation of the formation of aluminium galling under constant load conditions. In aluminium cold stamping processes, it is observed that the COF increases and an observable aluminium transfer layer is formed on the tool surfaces. In this chapter, this phenomenon is studied via pin-on-disc dry sliding tests with an aluminium alloy 6082 pin used to represent the workpiece material sliding against a cast iron G3500 disc representing the tool material, for a range of contact loads and sliding distances. The results showed that the COF gradually increases until a steady state value is reached. By analysing this phenomenon, the aluminium transfer layers that generated at the sliding interface were identified as

¹This chapter is based on paper work ‘Y. Hu, Y. Zheng, D. J. Politis, M. A. Masen, L. Wang, *Development of an interactive friction model for the prediction of galling behaviour during sliding wear*, (submitted).’

the origin of this behaviour that affects both friction and wear. To model this phenomenon, an interactive friction model was developed enabling the prediction of friction and the evolution of the transfer layer from the running-in to the steady state. This mechanism-based model can be used for representing friction variations and material transfer in metal forming operations.

4.2 Introduction

During aluminium sheet forming processes, adhesion of aluminium on the tool surface after repeated stamping operations can result in unacceptable part quality. Typical examples of poorly formed surfaces and geometries include scratches on the product, damage of functional surfaces such as coatings and surface textures as well as misalignment (de Rooij and Schipper, 2001b, Pujante et al., 2015, D’Amato et al., 2014, Chen et al., 2016). Therefore, there is a requirement to model aluminium adhesion mechanisms, to enable the prediction of friction variations in the tool-workpiece contact, to model tool wear life and improve the product’s surface quality.

Previous research has predominantly focused on the three main tribological responses of the adhesive wear system, namely: 1) particle transfer onto the counterpart, also called galling, transfer layer/film and material pick-up; 2) progressive loss of material or ‘*wear*’; and 3) friction.

Over the years, advancements in adhesive wear have resulted in improved dynamic analyses in addition to static studies. Early investigations focused on the transfer layer as it presents obvious static evidence for adhesion, and have been well studied in metal forming processes, with one-cycle adhesive wear mechanism models being well developed both mechanically (Landheer and Zaat, 1974, Burwell and Strang, 1952, Schedin, 1994, Heilmann et al., 1983, Olsson et al., 2002, Hanson et al., 2009) and chemically (Pujante et al., 2015, Riahi and Alpas, 2007). With the improvement of observation and modelling techniques, multi-cycle repeated transfer mechanisms have been studied (Heinrichs et al., 2012, de Rooij and Schipper, 2001b,

de Rooij et al., 2013, Podgornik et al., 2004, Olsson et al., 2002). Meanwhile, the evolution of wear rate and COF as opposed to an assumed constant value throughout forming cycles have been gradually introduced (Suh, 1986a, Meierhofer et al., 2014, Ma et al., 2015, Hu et al., 2017a, El Mansori and Paulmier, 1999, Xu et al., 2012).

Another development in recent years has been the combination of the above mechanisms into a single model to enable a holistic investigation of adhesive wear (Ma et al., 2015, Hu et al., 2017a, Kato, 2000, Diomidis and Mischler, 2011, Österle et al., 2012, Stoyanov et al., 2013, Colas et al., 2013, Fillot et al., 2007, Meierhofer et al., 2014, Berthier, 2005). The interaction between friction and adhesive wear was suggested by Bowden and Tabor (2001), who states that the break of adhesive junctions may simultaneously generate tangential stress as well as detachment and transfer of fragments from one surface to another. It was found that the transfer layer affects friction and wear (Bhushan, 2013) and hence a stable transfer morphology would be expected to correspondingly give a stable response. The relationship between wear and material transfer has been identified in early studies, where the source of the transfer layer was found to be the result of loose wear debris generated from adhesive wear (Heilmann et al., 1983). The influence of the transfer layer on the COF was also well observed (Heinrichs et al., 2012, Podgornik et al., 2004, Hanson et al., 2009). The responses, namely, transfer, wear and friction, were observed to reach a steady state value as sliding progressed. For friction in adhesive wear, the COF increased from an initial low value to a higher constant value (Kato, 2000, Podgornik et al., 2004, Diomidis and Mischler, 2011, Stoyanov et al., 2013, Colas et al., 2013, Roizard et al., 2009) with the terms 'running-in state' and 'steady state' widely used to describe the adhesive wear condition from unsteadiness to stabilisation (Yang, 2005, Yang, 2003, Pereira et al., 2010). Although the variation of surface morphology is difficult to make in real time in-situ recordings and can be represented by a range of parameters for various applications, evidence and analysis demonstrates that the steady state of mating surfaces are characterised by a dynamic equilibrium of the surface morphology and a transfer layer is formed with

a certain pressure-dependent galling area (Rabinowicz, 1965), height (Österle et al., 2012, de Rooij and Schipper, 2001b, Schipper, 2001), volume and mass (Schedin, 1994, de Rooij et al., 2013, Heilmann et al., 1983, Schipper, 2001, Fillot et al., 2007).

Evidence of the interactions between friction, wear and material transfer has been presented by Landheer and Zaat (Landheer and Zaat, 1974) and has further been described by the third body theory (Berthier, 2005, Fillot et al., 2007, Colas et al., 2013, Berthier et al., 2002, Godet, 1990). Applying this theory to the problem of galling, the transfer layer and wear particles are the so-called third body, which are generated as a result of the interaction between the original mating pairs (first bodies). Both friction and wear are simultaneously derived from the interaction between first bodies and third bodies. According to this theory (Landheer and Zaat, 1974, Colas et al., 2013, Yang, 2003, Yang, 2005, Fillot et al., 2007, Burwell and Strang, 1952), a transfer layer develops during the running-in state, with the friction and wear rate slowly developing to a constant steady state value. This behaviour is schematically shown in Fig. 4.1, showing the evolution of the coefficient of friction as well as the volumetric wear (V_s), the build-up of the transfer layer (V_i) and the volume of ‘ejected’ wear particles (V_e) that are removed from the system. Coefficient of friction, wear volume (V_s) and volume of third bodies (V_i) are functions of sliding distance and can be divided into running-in state and steady state. At the steady state, coefficient of friction and V_i become constants, and the variation of V_s becomes linear. V_e , which indicates the volume of wear particles ejected from the contact, equals to the wear volume subtracts by the volume of transferred third bodies.

In this chapter, the interaction between friction, wear and transfer, is characterised in a unified manner to evaluate the evolution mechanisms of aluminium adhesive wear. Experimental work was performed through pin-on-disc tests to quantitatively investigate the friction and transfer phenomena in an aluminium-cast iron tribo-system by two critical operation parameters, contact load and sliding distance (Dwivedi, 2010, Pereira et al., 2010). The interactive mechanisms of this tribo-system were analysed and the formation progress of the tribo-system from a

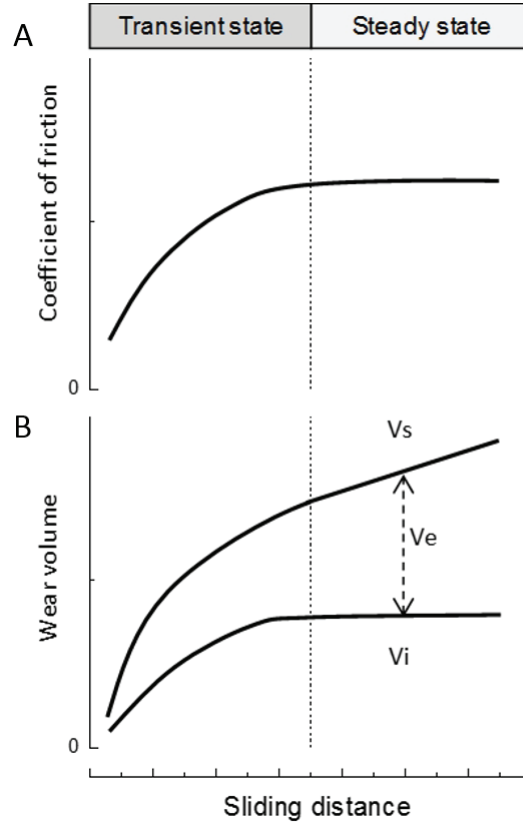


Figure 4.1: Schematic diagram of (A) evolution of COF, (B) wear (V_s), ejection (V_e) and transfer volume (V_i) evolutions of an adhesive wear tribo-system.

running-in state to a steady state was studied. An interactive friction model was developed based on the experimental data, which enables the modelling of the evolution of the friction as well as the galling area fraction of the cast iron surface.

4.3 Materials and methods

4.3.1 Materials

A pin-on-disc test was designed to study three main characteristics of aluminium adhesive wear: (i) to determine the COF evolution, wear rate evolution and galling area of aluminium adhesive wear; (ii) to quantitatively measure the effect of contact load and sliding distance on material transfer; and (iii) to develop and calibrate the interactive tribological model. The material used for the pin was aluminium

alloy AA6082 T6 (flat contact surface with a diameter of 2 mm, $R_a = 486 \pm 35$ nm, hardness = 115 ± 5 HV) representing the workpiece material. The material used for the disc was cast iron G3500 ($R_a = 126 \pm 19$ nm, hardness = 209 ± 18 HV), the same material used for the forming tools.

4.3.2 Test methods

The experiments were performed on an Anton Paar tribometer (THT) at room temperature with a linear sliding speed of 50 mm/s at a wear track radius of 5.0 mm. Sliding speed changes at a low speed range (below 100 mm/s) only has minor effect on galling which was confirmed by previous investigations and thus was set as a constant. A small radius of 5 mm was set for the advantage of generating 1) homogenous galling patches 2) stable COF curves and 3) reasonable sliding distance before the equilibrium. The flat-on-flat contact couples were self-aligned at the initial cycles sliding during the experiment. Tests were conducted according to the test matrix shown in Table 4.1. Two variables, sliding distance (test no. 1-6) and contact load (test no. 8-10), were studied and a reference of aluminium-aluminium contact (test no. 7) was conducted. Tests evaluating the effect of contact load were terminated after the COF had reached a steady state. Each test condition was repeated three times and an average value with standard deviation (SD) was measured. Due to the fact that unloading and reloading changed the distribution of wear particles, instead of interrupting, tests evaluating the sliding distance were conducted independently with discrete distance steps increasing from 100 to 600 laps (test no. 1-6).

4.3.3 Characterisation of galling

The COF was calculated from the measured friction force, whilst the wear rate was recorded using an LVDT measuring the penetration depth or approach of the two surfaces during the experiment. These values were both recorded by the tribome-

Table 4.1: Test matrix of galling test under constant load

Test No.	Load(N) /Pres-sure(MPa)	Contact couple		Sliding laps/distance(m)	Notes
1	5/1.59	AA6082	G3500	100/3.14	
2	5/1.59	AA6082	G3500	200/6.28	
3	5/1.59	AA6082	G3500	300/9.42	
4	5/1.59	AA6082	G3500	400/12.56	Sliding
5	5/1.59	AA6082	G3500	500/15.71	distance
6	5/1.59	AA6082	G3500	600/18.85	
7	5/1.59	AA6082	AA6082	600/18.85	Reference
8	2/0.64	AA6082	G3500	1200/37.70	
9	5/1.59	AA6082	G3500	600/18.85	Load
10	10/3.18	AA6082	G3500	500/15.71	

ter. The morphology of the cast iron surfaces was studied using a Hitachi S-3400N scanning electron microscope (SEM) and a Veeco WykoNT9100 white light interferometer (WLI). The electron source of SEM was operated at accelerating voltages of 10.0 kV and 15.1 kV and the secondary electron signal was collected. The area of aluminium transfer layer was calculated from the WLI 3D surface plot. The disturbance of wear particles was minimized during transporting and storing of the samples to enable the observation of loose wear particles.

4.4 Results and Discussion

4.4.1 Microstructure

In Fig. 4.2, the transfer layers generated from the interaction of aluminium-cast iron contact are observed. The chemical composition of the wear particles and transfer layer was studied by Kim et al. (2007b) who conducted aluminium sliding against stainless steel at room temperature and showed that the aluminium wear particles were mainly soft aluminium with oxygen in its solid solution and hydroxylated

material. In the present research, the generated wear particles and transfer layer, are simply referred to as ‘aluminium wear particles’ and ‘aluminium transfer layer’. Fig. 4.2B, compared to the original surface of Fig. 4.2A, shows: (i) the large area of material transfer patches, which was composed of individual aluminium lumps firmly accumulated on the cast iron surface, and (ii) the entrapped loose wear particles. The lumps and wear particles show brighter edges that may be caused by enhanced emission of electrons in the SEM images. The dark areas encircled by bright edges indicate large transfer patches, which were elevated structures with the height of several micrometres measured by WLI (Fig. 4.3). The scale of these lumps varies from $1\text{ }\mu\text{m}$ to $500\text{ }\mu\text{m}$; the diameter of loose wear particles ranges in the scale from $1\text{ }\mu\text{m}$ to $100\text{ }\mu\text{m}$ (Fig. 4.2B). With the relative motion of the contact couple, some of the aluminium particles were ejected out of the wear track and contributed to the ejection volume (Fig. 4.2C). The transfer layer appeared to be built up of lumps that grew in a multi-layer ellipsoidal shaped structure, with their major axis in the direction of sliding (Heinrichs et al., 2012, de Rooij et al., 2013, Hu et al., 2016). In the enlarged view of transfer patches (Fig. 4.2D), edges of lumps are fitted with ellipse and overlaps can be observed. Both the small size and dynamic characteristic made wear particles difficult to be observed and quantitatively measured thus the measured morphology by WLI was mainly the transfer layer.

The surface morphology of aluminium pins are shown in Fig. 4.4. Due to the low hardness of aluminium pin comparing to the cast iron counterpart, the aluminum pin shows clear abrasive damage from the beginning of the contact. As shown in Fig. 4.4 , an uneven abrasive damage can be observed at 100 laps. The damage on aluminum becomes severer with sliding distance increasing and aluminium deformation becomes clearer at the exit of wear particles in the sliding direction. It is believe that the severity of damage on the pin largely depends on the surface roughness of the counterpart. When aluminum transfer layer gradually grows on the counterpart, the surface roughness increases on the cast iron surface, which will further damage the aluminium pin and generate more wear particles.

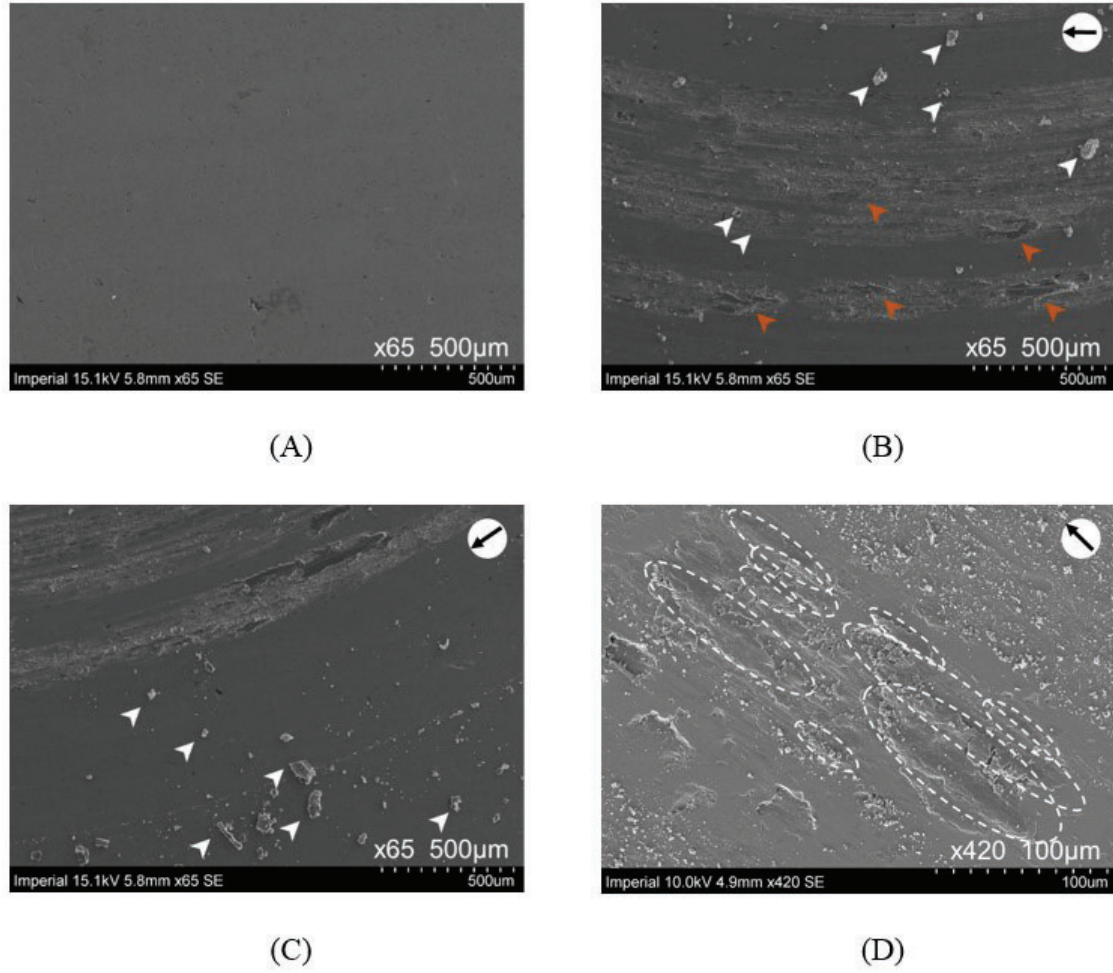


Figure 4.2: SEM topography of aluminium transfer layer captured from test 10: (A) reference: clean cast iron surface; (B) aluminium transfer layer (bright tracks in the direction of sliding), large aluminium transfer lumps (red arrow heads) and entrapped wear particles (white arrow heads); (C) ejected aluminium wear particles outside the track (white arrow heads); (D) enlarged view of the aluminium lumps (white ellipses). Sliding direction: black arrows (top right).

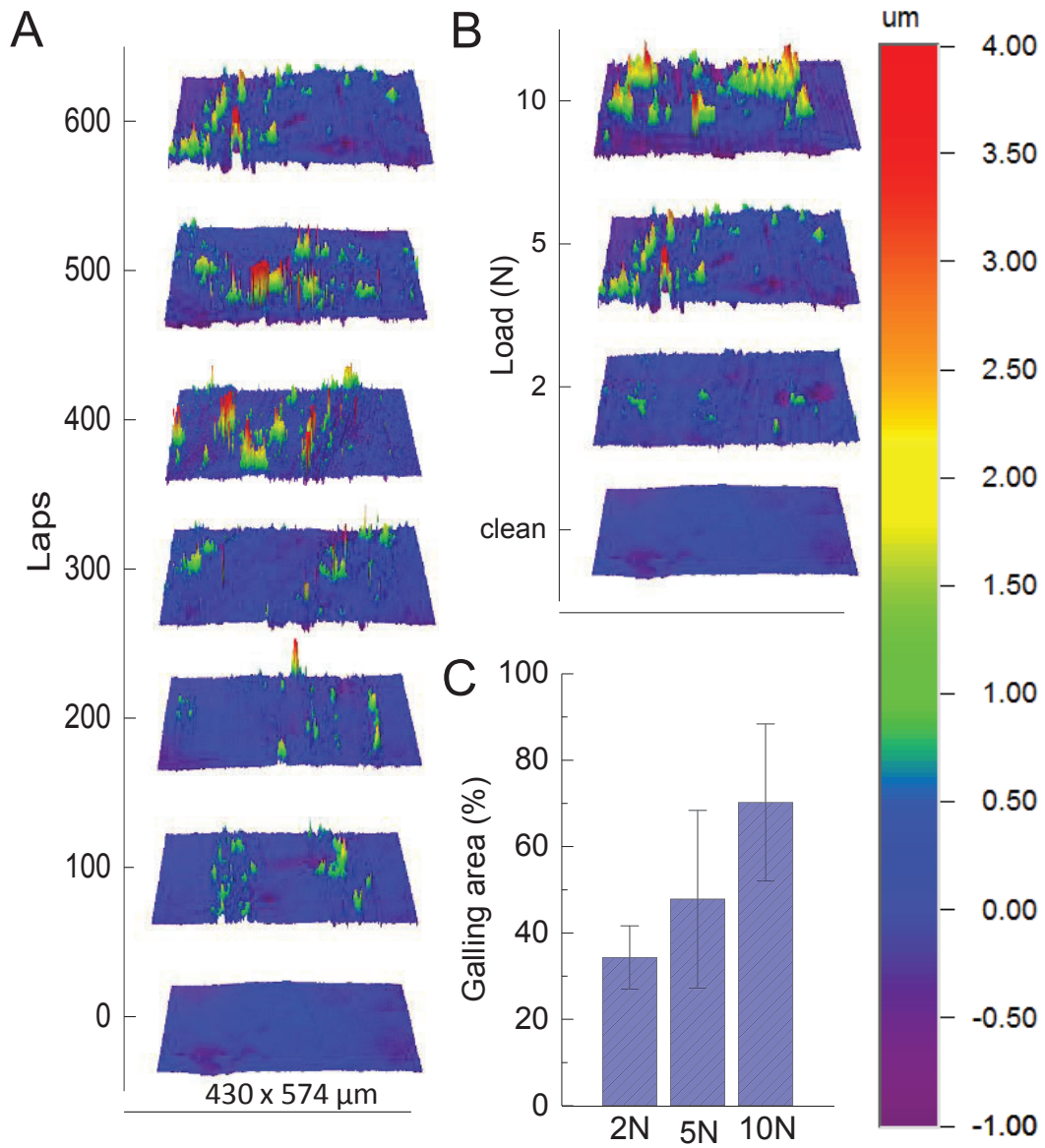


Figure 4.3: WLI surface morphology with the effects of (A) sliding distance at a load of 5 N and (B) contact load of 2, 5 and 10 N at the steady state. (C): the galling area (%) under different loads at the steady state.

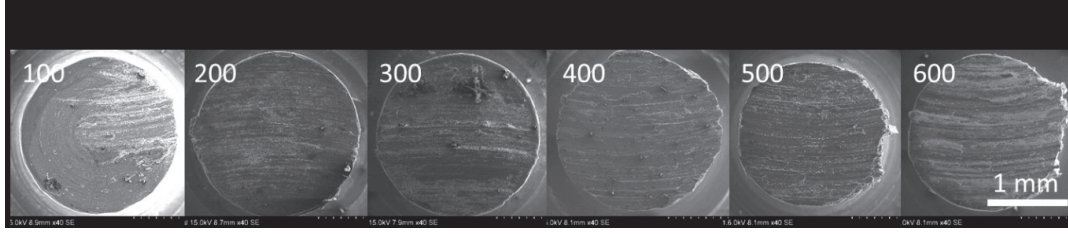


Figure 4.4: SEM topography of aluminum pins with sliding distance increase (Laps: 100 to 600, captured from test 1 to 6).

4.4.2 Interaction between friction, wear and galling behaviour

In Fig. 4.3, the development of the transfer layers with increasing sliding distance and contact pressure are presented. In Fig. 4.3A, under the contact load of 5 N, the wear particles accumulated in the direction of sliding and transfer patches with elevated height were gradually formed with increasing sliding distance. The accumulation of transfer layer did not progress infinitely and eventually a steady state was reached. The evolution of the transfer layer can be quantified by the galling area fraction, defined as the fraction of the cast iron surface covered by the aluminium transfer layer. As the contact morphology change plays the main role of affecting tribological responses in the present study, galling area fraction becomes a suitable parameter for modelling the formation process. The relationship between contact load and the steady state value is shown in Fig. 4.3B where the saturated galling area increased with increasing load and an approximate linear relationship between contact load and galling area was obtained in the studied load range (2 to 10N). The dependency of the galling area fraction with load is shown in Fig. 4.3C and is 34, 48 and 71% for 2, 5 and 10 N respectively. In Fig. 4.5B, the increase rate of galling area was clearly reduced after approximately 400 laps and fluctuates around 48%, which is defined as the saturated galling area under 5 N and indicates a steady state has been reached.

Comparing the increasing galling area with the increase of COF and wear depth as shown in Fig. 4.5, an interaction between material transfer, friction and wear can be observed where these three parameters experience a transient state and reached a

steady state. This is indicated by the constant COF, galling area and wear rate. In the transient state, which is widely named as running-in state, the coefficient began with a value of 0.18 at zero sliding distance and increased to a stable value of 0.55 at approximately 400 laps of sliding. Difference can be observed between this COF evolution and the evolutions obtained in chapter 6 where smooth evolutions were obtained under fully flooded condition. The aluminium-cast iron tribo-system at dry sliding condition experiences magnificent vibrations and generates noise on the COF evolutions. The noise may generated by factors including original surface roughness of the contact pair, accumulated wear particles, formed aluminium aluminium lumps and instrument vibrations. It is believed that, as the aluminium transfer layer and wear particles gradually formed at the contact, the initial noise that caused by initial asperity contact is replace by the vibration caused by third bodies. The wear depth evolution (Fig.4.5C) shows an initial steep slope of the evolution that gave a high wear rate at the beginning of the sliding test and caused severe oscillation of the evolution. As the tribo-system transferred to the steady state, a linear increase of the wear depth, giving a constant wear rate, can be observed. Combining the analysis of Fig. 4.5A, B and C, an interactive relationship between friction, wear and transfer in the Al-cast iron adhesive wear system has been demonstrated: the formation of aluminium transfer layer affects both friction and wear; and these three phenomena all experience an initial running-in state and transfer to a steady state simultaneously.

The effects of contact load on the evolution of the coefficient of friction and the wear depth are shown in Fig. 4.6. The increasing load shortens the distance of the running-in state and increases wear rate by increasing the slope of the wear depth. For applied loads of 10 N, 5 N and 2 N, the steady state was reached after approximately 220, 400 and 780 laps, respectively. The initial and stable coefficients of friction of the tribo-system were not significantly influenced by the load in the studied range and can therefore be treated as pressure-independent constants. The coefficient of friction started from a value of 0.18 at zero distance to a steady state value of 0.55. In the present study, the wear coefficient can be

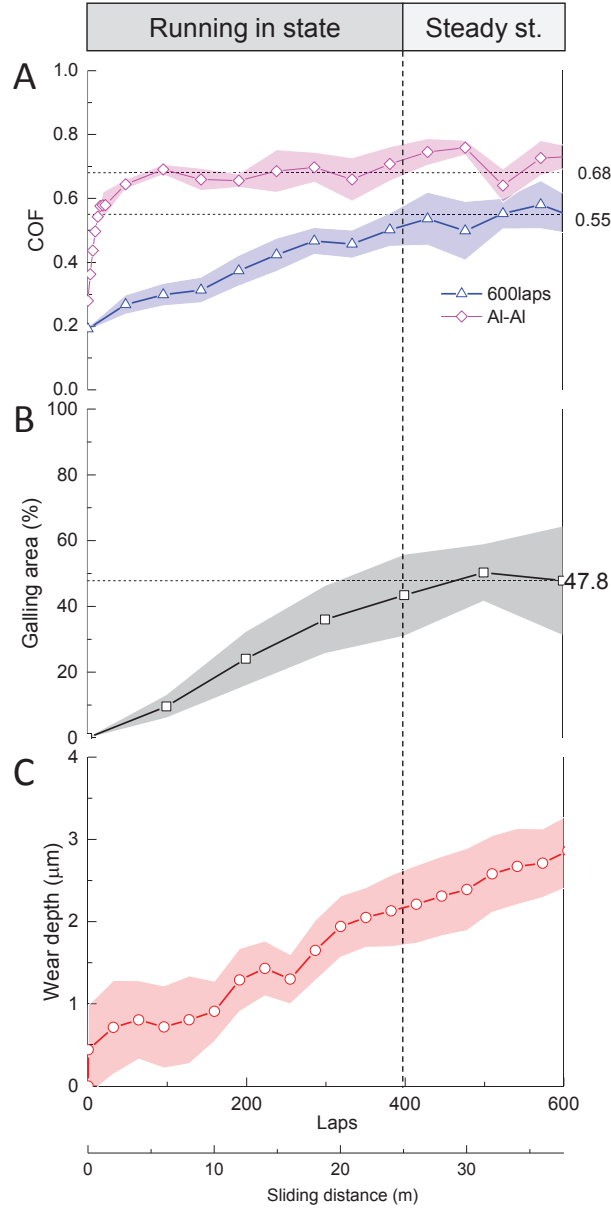


Figure 4.5: Effects of sliding distance under 5 N:(A) COF evolutions of Al-cast iron with Al-Al contact as reference , (B) galling area evolution and (C) wear depth evolution. Dash line, running-in state; symbols, mean coefficient of frictions; envelopes, SDs.

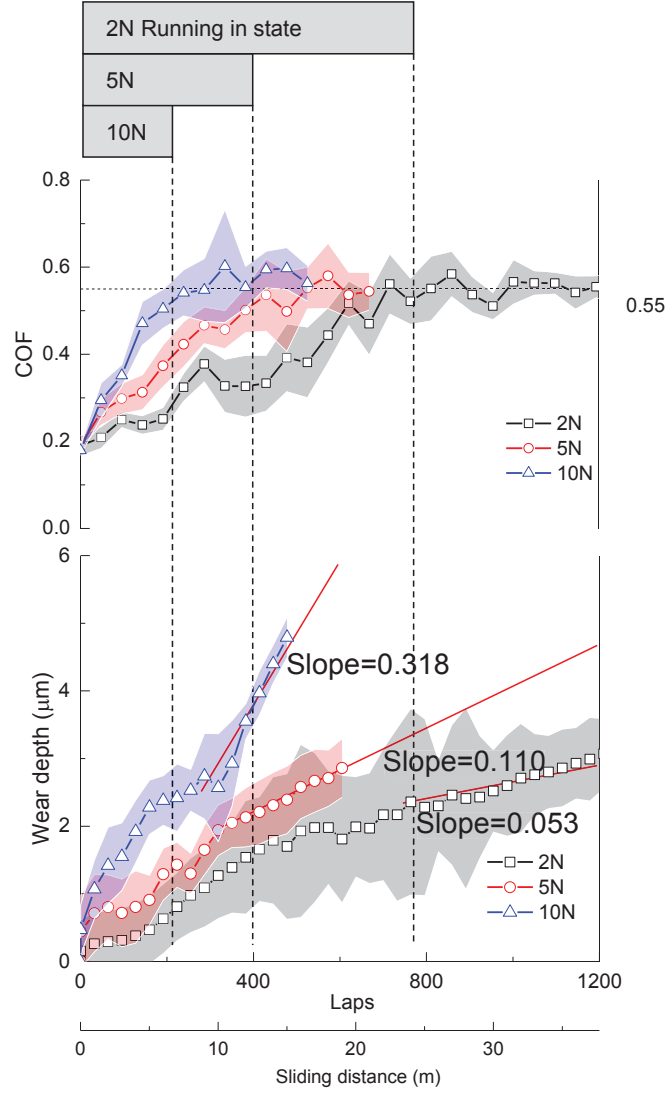


Figure 4.6: Effects of contact load on the evolution of (A) coefficient of friction and (B) wear depth. Dash lines, running-in states for different loads; symbols, mean coefficient of frictions; envelopes, SDs. Wear rates (wear depth *vs.* sliding distance [$\mu\text{m}/\text{m}$]) are indicated by slopes (fitted by single linear regression in the steady states) in (B).

calculated using the Holm-Archard equation (Eq. 2.1, $h = kPx/H$, where h is the penetration depth caused by wear, k is the wear coefficient, P is the contact pressure, x is the sliding distance and H is the hardness of the soft material) from the slopes of wear depth evolutions that obtained in Fig. 4.6B. The mean wear coefficient is $8.40\text{E-}9$ ($\text{SD}=1.54\text{E-}9$). It should be noted that, different from the evolutions of wear depth obtained under 2 and 5 N, the evolution obtained under 10 N experienced a short plateau at the beginning of steady state. After this plateau, the wear rate increased again, which has a high wear rate similar to the initial of contact. This phenomenon indicates that the wear rate may not linearly vary with contact load in the steady state, which is predicted by Holm-Archard equation. Instead, due to material properties, step change of wear rate may occur.

4.4.3 Relationship between friction, wear and the development of a transfer layer

The tribological behaviour of the system is strongly affected by the generation of a transfer layer and the effects include 1) a change of the interface characteristics through physical-chemical interactions; and 2) a change from tribology between solid aluminium against solid cast iron to a much more complex system. This causes an increased number of interrelationships and interdependencies.

As an aluminium transfer layer as well as loose wear particles form at the interface and partly separate the contact couples, they present a new physical/structural and chemical morphology of the interface compared to the initial aluminium-cast iron contact. The roles, 1) separating the aluminium-cast iron contact couple, 2) transmitting contact load and shearing, and 3) triggering further wear (Heinrichs et al., 2012), are gradually determined by the third body transfer layer instead of the original aluminium-cast iron contact and thus give a different response to friction and wear. The phenomenon of material transfer has been confirmed to be a chemical and mechanical interaction (Pujante et al., 2015, Kim et al., 2007b). Severe galling and ploughing grooves were observed in Fig. 4.2 and 4.3 on the transfer layer and the dominant friction mechanism can be identified as a mixture of adhesion

and ploughing whereas the dominant wear mechanism is a mixture of adhesive and abrasive wear.

At the beginning of the sliding process, the aluminium pin is in direct contact with the cast iron surface and friction is generated by adhesion and ploughing between asperities of aluminium and cast iron. The relative motion shears the contact junctions and breaks the soft aluminium asperities, which may result in detachment of solid aluminium and subsequent attachment to the cast iron surface. This initial state gives a low COF and a high wear rate (Fig. 4.6).

After the layer of aluminium third body has been generated, the friction mechanism partly transfers from the initial adhesion to cohesion (self-adhesion) between the solid aluminium, loose particles and the transfer layer. As schematically illustrated in Fig.4.7, with the transfer layer covering the asperities on the cast iron specimen, ploughing occurs between the pin, lumps and loose particles of aluminium. The aluminium particles may also roll at the interface to accommodate the velocity difference between the contact couple, reducing friction. Moreover, the increase of real contact area (Stempflé et al., 2010) and surface roughness (Heinrichs and Jacobson, 2011) increases the cohesion and mechanical force, respectively. The transfer layer will prevent the direct contact of aluminium against the hard cast iron asperities, thus limiting ploughing and abrasion and, combined with the polishing mechanism, the top surfaces of aluminium lumps and asperities may become smooth. As a result, the wear rate decreases from the maximum value to a constant at the steady state (shown in Fig. 4.6B). At the steady state, the COF is independent of applied contact load and is about 0.55, which is lower than the 0.68 observed in an aluminium-aluminium contact as shown in Fig. 4.6A. Regardless of the morphology difference between different materials, this phenomenon can be explained by the incomplete coverage of the aluminium transfer layer in the steady state: the metallic bonds at the interface of an aluminium-aluminium contact are stronger than those in an aluminium-steel contact (Rabinowicz, 1971, Bhushan, 2013). As an incomplete coverage of aluminium transfer layer is observed, a mixture of aluminium-aluminium and aluminium-cast iron contact occurs, giving a reduced value for the COF of 0.55

at the steady state region when compared to 0.68 for aluminium-aluminium contact.

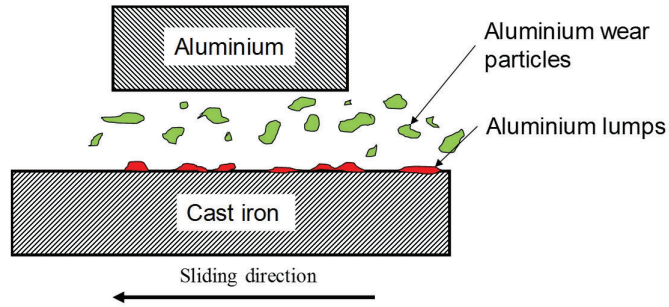


Figure 4.7: Schematic of Al-cast iron contact with two states of aluminium: particles and lumps.

4.4.4 A theoretical model for the formation of a transfer layer

The formation of the aluminium third body layer on the cast iron surface determines the sliding distance dependent friction and wear evolution. In the aluminium-cast iron tribo-system, this layer is formed gradually with increasing sliding distance. Loose wear particles entrapped at the interface are moved following the relative motion of the contact couple. With the combined effects of high contact pressure and shearing, surface defects and the high adhesion characteristic of aluminium, these wear particles are likely to adhere to the cast iron and form aluminium lumps, which further accumulates as a transfer layer (Heinrichs et al., 2012). The transfer and back-transfer mechanism may exist between wear particles, solid aluminium and transfer lumps of aluminium, that is, on one hand, wear particles detach from solid aluminium and transfer to cast iron surface; on the other hand, the lumps may detach from the cast iron surface and become wear particles or even transfer back on the solid aluminium (Rabinowicz, 1965, Berthier, 2005). This mechanism may be active through the whole sliding process even when steady state is reached. When subjected to loading and shearing, the interface underwent mixed elastic/plastic modes of deformation.

At the beginning of the contact, the small real contact area gave an extremely high contact pressure, which is beyond the elastic regime (Shankar and Mayuram, 2008, Karupannasamy et al., 2014). Plasticity occurred at the asperities, lumps and loose wear particles of aluminium, which leads to geometry changes and balance including the growth and saturation of contact area and transfer volume on a macroscopic scale. This automatic pressure reducing mechanism maintains the contact pressure at the critical value and thus presents a mechanically stable new built surface morphology.

At the equilibrium state, although wear particles are continually generated, the morphology of the interface is relatively stable as observed in Fig. 4.3B by galling area fraction and in the literature (Rabinowicz, 1965, Stempflé et al., 2010, de Rooij and Schipper, 2001, de Rooij et al., 2013, Österle et al., 2012, Schipper, 2001, Wang et al., 2010). An important conclusion is drawn from these observations where the wear particle generation and ejection mechanisms play a critical role in balancing the wear particles and lumps at the contact. To describe this dynamic balance, the concept of third body density, ρ , is introduced, where it is a measurement of the number of third body elements in a unit contact area.

During the aluminium transfer process, the wear volume of cast iron is relatively minor. It can be assumed that all the third body is generated by detachment from solid aluminium and entrapped at the contact in the forms of loose wear particles and aluminium transfer layer. At the equilibrium state, density of the entrapped third body at contact interface, ρ_i , achieved a dynamic balance between the new generated wear particles ρ_s and the third body ejected from the contact ρ_e , where a critical third body density at the contact, which is pressure dependent at room temperature, is reached. The three densities are concluded as:

- ρ_i : density of the entrapped third body at contact interface
- ρ_s : density of the source of third body
- ρ_e : density of the ejected third body

This mechanism is described by a tribological circuit of the third body changing rate

(Fillot et al., 2007, Berthier, 2005), which relates the function of third bodies with their derivatives and provides a dynamic view of the life of the third body layer. The density rate at the interface, $(\dot{\rho}_i)$, is the changing rate of the density of third body that is entrapped at the contact and is determined by the rate of source density of wear particles (source rate, $(\dot{\rho}_s)$) that convects into the contact and the rate of density of wear particles that ejects (ejection rate, $(\dot{\rho}_e)$) from the contact. Based on the conservation of mass, an equilibrium of the changing rates of the transfer layer at the contact can be written as Eq. 4.1.

$$\frac{d\rho_i}{dt} = \dot{\rho}_i = \dot{\rho}_s - \dot{\rho}_e \quad (4.1)$$

Before equilibrium, the wear progress constituted a competition between $\dot{\rho}_s$ and $\dot{\rho}_e$ (Fillot et al., 2007, Colas et al., 2013) and gave various $\dot{\rho}_i$. This sequence is presented in Fig. 4.8 and is analysed from the running-in to the steady state.

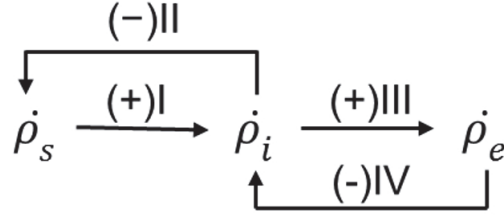


Figure 4.8: Feedback loop of source density (detachment of Al wear particles), ejection density (ejection of these particles) and third body internal density (entrapped particles at the contact) with outputs (I, III) and feedbacks (II, IV).

State 1 running-in state This state began at the beginning of sliding and ended when the system reached equilibrium. At the beginning, the volumetric wear from the unit surface (ρ_s) increases rapidly from zero with a maximum source rate of generating wear particles $(\dot{\rho}_s)$. The main consequence of ρ_s increase is the instantaneous development of third body at the contact, characterised by an increase in ρ_i (de Rooij et al., 2013, Heilmann et al., 1983)). At the beginning, the maximum $\dot{\rho}_s$ leads to a maximum accumulation rate of wear particles $(\dot{\rho}_i)$ at the contact (process I). When sliding continues, the increase of ρ_s is gradually suppressed by the saturation of the transfer layer. That is, with the increase of ρ_i toward saturation, less

detachment from the solid aluminium is expected, which slows down the increase of ρ_s , i.e. a decrease of $\dot{\rho}_s$ (negative feedback process II). Meanwhile, the amount of (ρ_s) , which keeps increasing until the equilibrium state is reached, also leads to a rise of the ejection density, (ρ_e) , by increasing the ejection rate $\dot{\rho}_e$ (process III). It is certain that both the volume of ejected wear particles V_e and its variation rate $\dot{\rho}_i$ starts from a low initial value, which should be close to zero, and gradually increase at the running-in state (Fillot et al., 2007). The steady increase of $\dot{\rho}_e$ will further suppress the increase of ρ_i by decreasing the internal density rate $(\dot{\rho}_i)$ (negative feedback process IV).

State 2 Steady state This state starts after the equilibrium is reached, where the friction has stabilised. A transfer layer has developed and shows minimal wear. When the $\dot{\rho}_s$ decreases and the $\dot{\rho}_e$ increases to the same value, i.e. $\dot{\rho}_e = \dot{\rho}_s = \text{constant}$, and $\rho_i = \text{constant}$ with $\dot{\rho}_i = 0$, a dynamic equilibrium is reached, which demonstrates that the amount of ejected particles equalled to the amount of newly generated particles and gives zero net change of the density of the transfer layer at the contact interface. The schematic diagram of Fig. 4.9 shows the relationship between different wear density and their changing rates. In this figure, the changing rate of the source and the internal density decreased in the running-in state as the ejection rate gradually increased to the value of generation rate when equilibrium is achieved.

Galling is a pressure-dependent phenomenon. When the high contact load combined with shearing are applied, high deformation energy is inputted at the contact that will intensify both the detachment and transfer of aluminium wear particles. As observed in Fig. 4.3 and 4.6, the increasing contact load increased the galling area and transfer volume as well as shortened the duration of the running-in state. The galling area fraction increased linearly with the contact load in the studied range (2 N to 10 N), and may cover the whole contact when the contact load continues increasing. The major reason should be: higher contact load leads to more wear particles to be generated and increases the amount of collision between wear particles and the counterpart, thus increases the probability of material transfer. Secondary, the higher contact load increases the real contact area and also enhance the diffusion

bonding between aluminium and cast iron. In addition, the local cold welding may be enhanced by higher mechanical energy converting into thermal energy through deformation of the transfer patches and releases frictional heat, generating firm adhered junctions (Landheer and Zaat, 1974). However, comparing to mechanical anchoring and metallic bonding as well as considering the test condition at room temperature, this factor could be minor.

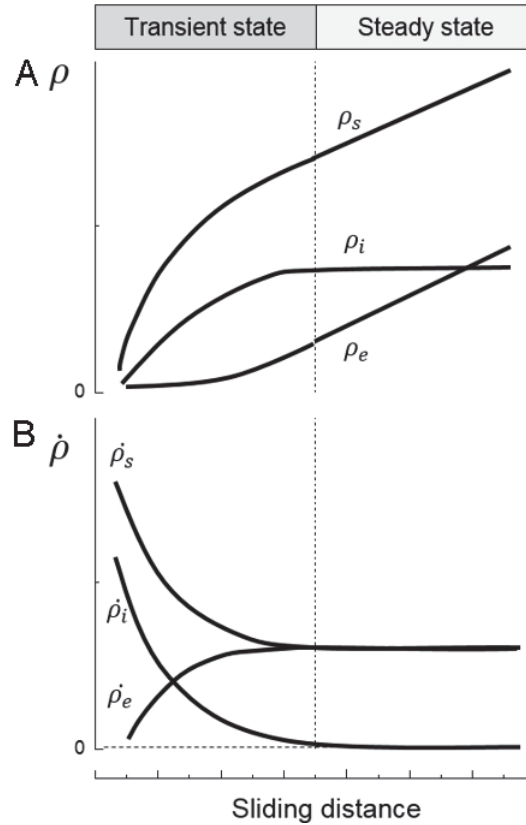


Figure 4.9: Evolutions of the density of third bodies (A) and their variation rates (B).

4.5 Interactive friction model

A mechanism-based model was developed to represent the interaction of friction, wear and transfer through a transfer layer formation progression in the aluminium-cast iron tribo-system. In this progression, the COF and galling area are the vari-

ables that can be experimentally measured and modelled. The variation of these variables is driven and modelled by the accumulation of third body at the contact interface.

The origins of the COF can be classified as i) aluminium-cast iron contact and ii) aluminium-aluminium contact. The initial COF (μ_{Al-CI}) stems from the contact of the clean aluminium pin and cast iron interface. After the tribo-system reaches the equilibrium, the stable COF (μ_{Al-Al}) is generated by a saturated galling interface between solid aluminium and aluminium third body. As discussed, these two coefficients of friction can be considered as pressure-independent constants, which have an initial value of $\mu_{(Al-CI)}=0.18$ and a saturated value of $\mu_{Al-Al}=0.55$. Thus, the variation of COF becomes a function of the galling area and can be modelled by Eq. 4.2.

$$\mu(t) = (1 - \bar{f}(t))\mu_{Al-CI} + \bar{f}(t)\mu_{Al-Al} \quad (4.2)$$

In this equation, to represent the variation of galling area in the formation progression of transfer layer, a normalised galling area, $\bar{f}(t)$ is introduced. The normalised galling area can be defined as the galling area at a certain sliding time ($f(t)$) by the saturated galling area at the steady state ($f(t_{sat})$) (Eq. 4.3). During the transfer layer forming progress, $f(t)$ starts from an initial clean contact ($f(t_0) = 0$) and gradually increases to the maximum when the contact points are fully covered by an aluminium transfer layer ($f(t_{sat}) = max$). Therefore, the normalised galling area varies from 0 to 1 and indicates the transition of the tribo-system from a running-in to a steady state. $f(t_{sat})$ is a load (contact pressure) dependent parameter that can be determined experimentally. With the formation progress described by $\bar{f}(t)$, the COF as a function of the transfer progress can be predicted, where the COF generated at the clean surface is gradually replaced by the coefficient of friction at the saturated aluminium-aluminium contact with the increase of normalised galling area.

$$\bar{f}(t) = \frac{f(t)}{f(t_{sat})} \quad (4.3)$$

The variation of $f(t)$ is the main consequence of the accumulation of third body at the interface, which can be presented by the variation of the density of third

body. By definition, the expression of the density of third body, ρ , can be written as Eq. 4.4, where N is the number of third body elements and A is the contact area. The total wear volume can be expressed as the sum of average volumes of third body particles (Eq. 4.5) and thus the expression of the density of third body can also be written as Eq. 4.6.

$$\rho = \frac{N}{A} \quad (4.4)$$

$$V = \sum_{n=1}^N V_{avg} = NV_{avg} \quad (4.5)$$

$$\rho = \frac{1}{A} \frac{V}{V_{avg}} \quad (4.6)$$

As the galling area is generated by entrapped aluminium wear particles, it is certain that there is a positive correlation between the galling area and the density of entrapped third body, ρ_i . However, this relationship has not yet been well discussed by macro-scale physical models. The assumption and calculation, which were made in modelling material transfer on a single asperity scale where the galling area $f(t)$ is a function of V_i (de Rooij et al., 2013), is extended to macro contacts and can be simplified into Eq. 4.7. The derivation of this equation is shown in Appendix B. This equation constructs a relationship between the density of entrapped third body and galling area and can be applied to calculate the normalised galling area fraction, $\bar{f}(t)$. In this equation, b and λ_1 are model parameters and the load-dependent parameter, $f(t_{sat})$, can also be calculated by the the density of entrapped third body at the steady state, $\rho_i(t_{sat})$ (Eq. 4.8).

$$f(t) = b(\rho_i(t))^{\lambda_1} \quad (4.7)$$

$$f(t_{sat}) = b(\rho_i(t_{sat}))^{\lambda_1} \quad (4.8)$$

The calculation of $\rho_i(t)$ is shown by Eq. 4.9 to 4.11. To address the variation of both the amount of entrapped third body and the running-in time, a concept of normalised density of entrapped third body ($\bar{\rho}_i(t)$) is introduced. Similar to the definition of the normalised galling area ($\bar{f}(t)$), the normalised density of entrapped third body is defined as the density of entrapped third body at a certain sliding time by the saturated density of entrapped third body at the steady state (Eq. 4.9).

Therefore, the normalised density of entrapped third body is a variable between 0 and 1, where $\bar{\rho}_i(t_0)=0$ and $\bar{\rho}_i(t_{sat})=1$. The two variation characteristics of $\rho_i(t)$ can be modelled by solving Eq. 4.9. Firstly, the running-in time can be determined by the evolution of $\bar{\rho}_i(t)$ from 0 to 1. Secondly, the amount of entrapped third body can be calculated by $\bar{\rho}_i(t_{sat})$. To solve the equation, $\dot{\rho}_t$ can be expressed as Eq. 4.10 by considering the definition of $\dot{\rho}_i$. Combining Eq. 4.9 and 4.10, the change rate of $\bar{\rho}_i$ can be directly connected to the change rates, $\dot{\rho}_s(t)$, $\dot{\rho}_e(t)$ and the entrapped third body at the steady state, $\rho_i(t_{sat})$ (Eq. 4.11). Integrating $\dot{\bar{\rho}}_i(t)$ and multiplying by the entrapped amount variation, $\rho_i(t_{sat})$, the density of entrapped third body ($\rho_i(t)$) can be calculated.

$$\bar{\rho}_t(t) = \frac{\rho_i(t)}{\rho_i(t_{sat})} \quad (4.9)$$

$$\dot{\rho}_i = \dot{\rho}_s - \dot{\rho}_e \quad (4.10)$$

$$\dot{\bar{\rho}}_t(t) = \frac{\dot{\rho}_i(t)}{\rho_i(t_{sat})} \quad (4.11)$$

The calculation processes of $(\dot{\rho}_s(t), \dot{\rho}_e(t))$ and $\rho_i(t_{sat})$ are shown in Eq. 4.12 and 4.13, Eq. 4.14 and Eq. 4.16, respectively.

The expressions of $\dot{\rho}_s(t)$ and $\dot{\rho}_e(t)$ can be determined by modelling the relationships shown in Fig. 4.8. By the calculation of Appendix D, the expression of $\dot{\rho}_s(t)$ can be given in the form of Eq. 4.12. In this equation, $\dot{\rho}_s(t_{sat})$ is the source rate at the steady state and n is a model parameter, which physically represents the ratio between the initial source rate and its saturated value (Eq. 4.13). This equation presents the mechanism that, with the wear mechanism transits from the running-in to the steady state, and $\dot{\rho}_s(t)$ gradually reduces to a constant value when the tribo-system reaches equilibrium. At the beginning of the running-in state, due to the rough surface, it is likely to damage the surface of solid aluminium and generate third body in a high rate. When the transfer process approaches the steady state, the amount of entrapped third body is large, however the interface between the solid aluminium and soft flat transfer layer is unlikely to effectively damage their counterparts and thus a low generation rate of third body is expected.

$$\dot{\rho}_s(t) = \dot{\rho}_s(t_{sat})(n\dot{\bar{\rho}}_i(t) + 1) \quad (4.12)$$

$$n = \frac{c}{\dot{\rho}_s(t_{sat})} \quad (4.13)$$

The ejection rate, $\dot{\rho}_e(t)$, which is originally generated by $\dot{\rho}_s(t)$, starts from zero and keeps increasing until it reaches a saturated value ($\dot{\rho}_s(t)=\dot{\rho}_e(t)=\text{constant}$) at the steady state. As shown in Fig. 4.8, the increase of $\bar{\rho}_i(t)$ will increase the ejection rate, $\dot{\rho}_e(t)$, and on the other hand, this ejection process decreases $\dot{\rho}_i(t)$. Thus, $\dot{\rho}_e(t)$ can be expressed by Eq. 4.14, in which $\dot{\rho}_s(t_{sat})\bar{\rho}_i(t)$ indicates the variation of the source of the ejected wear particles and negative in Eq. 4.10 indicates the effect of ejection that leads to decrease on the density of third body at the interface. This is further demonstrated in Appendix E.

$$\dot{\rho}_e(t) = \dot{\rho}_s(t_{sat})\bar{\rho}_i(t) \quad (4.14)$$

The term, $\dot{\rho}_s(t_{sat})$ in Eq. 4.12 and 4.14 can be modelled by building the relation between $\dot{V}_s(t_{sat})$ and $\dot{\rho}_s(t_{sat})$. As shown in Appendix F, $\dot{V}_s(t_{sat})$ can be expressed by the instantaneous form of the Holm-Archard equation. Instituting the instantaneous form of Holm-Archard equation in the definition of ρ_s , $\dot{\rho}_s(t_{sat})$ can be written as Eq. 4.15.

$$\dot{\rho}_s(t_{sat}) = BPv(t_{sat}) \quad (4.15)$$

The saturated density of third body at the interface, $\rho_i(t_{sat})$, can be expressed by Eq. 4.16. $\rho_i(t_{sat})$ has the characteristics that i) when the system reaches the steady state, $\rho_i(t_{sat})$ is a fraction of $\rho_s(t_{sat})$; ii) both $\rho_i(t_{sat})$ and $\rho_s(t_{sat})$ are pressure-dependent parameters where the increase of contact will increase their values; and iii) due to saturation characteristics of the contact interface, the increase of $\rho_i(t_{sat})$ is reduced by the increase of contact pressure. As shown in appendix H, the expression of $\rho_i(t_{sat})$ can be written as Eq. 4.16, where C and λ_2 are model constants and $1 > \lambda_2 > 0$. Constant C represents the numerical relation between $\rho_i(t_{sat})$ and $\rho_s(t_{sat})$. The rate reduction of contact pressure is presented by the term, $1 - \lambda_2$, where the increase rate of $\rho_i(t_{sat})$ is lower than $\rho_s(t_{sat})$.

$$\rho_i(t_{sat}) = CP^{1-\lambda_2} \quad (4.16)$$

By substituting the expression of $\dot{\rho}_s(t)$ (Eq. 4.12), $\dot{\rho}_e(t)$ (Eq. 4.14) and $\rho_i(t_{sat})$ (Eq. 4.16) and the ratio $m = B/C$, in Eq. 4.11, the final expression of $\dot{\bar{\rho}}_i(t)$ can

be written as (Eq. 4.17). The COF and galling area can be modelled by calibrating experimental results with the model parameters shown in Table. 2.

$$\dot{\bar{\rho}}_i(t) = mP^{\lambda_2}v(t)((n\dot{\bar{\rho}}_i(t) + 1) - \bar{\rho}_i(t)) \quad (4.17)$$

Table 4.2: Model constant of galling test under constant load

Parameters	m	n	b	μ_{Al-Al}	μ_{Al-CL}	λ_1	λ_2
value	0.1	15	0.43	0.18	0.55	1.6	0.7

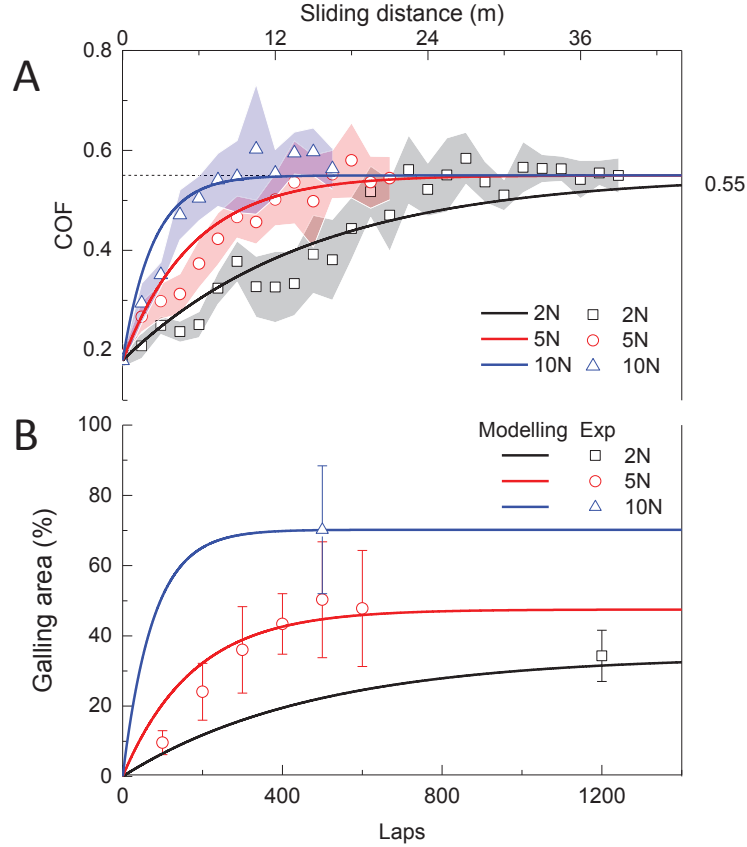


Figure 4.10: Modelling result of (A) coefficient of friction evolution and (B) galling area at different contact loads comparing to experimental results.

The fitting results of the COF and galling area fraction at different contact loads are shown in Fig. 4.10. It can be seen that this model provides a good agreement

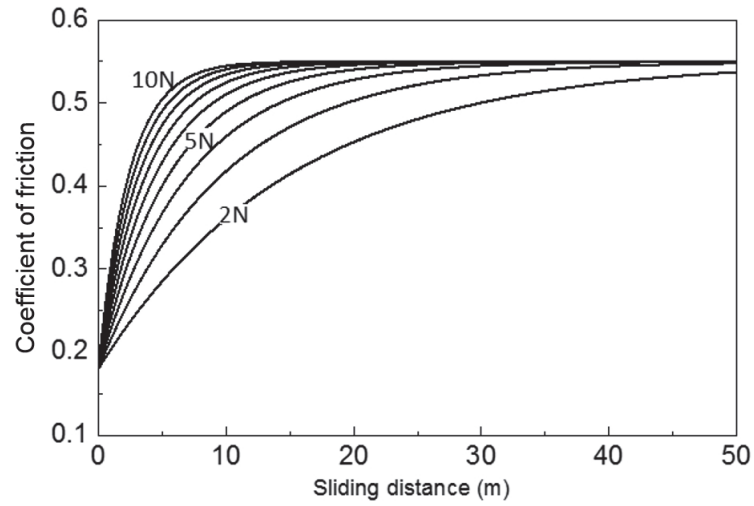


Figure 4.11: Coefficient of friction evolutions with the effects of contact load and sliding distance.

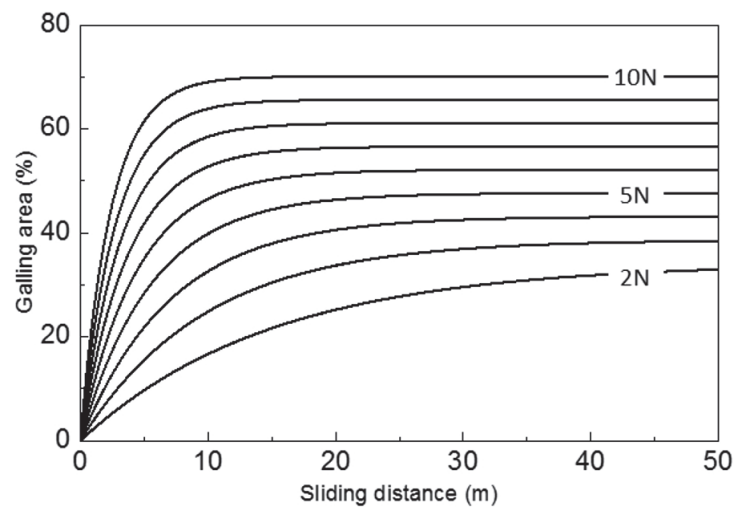


Figure 4.12: Galling area evolutions with the effects of contact load and sliding distance.

with experimental results to model the evolutions at different loads. Moreover, the modelling of COF and galling area fraction can be modelled as a function of contact load and sliding distance (Fig. 4.11 and 4.12). These figures are plotted in the studied range of load (from 2 to 10 N) and sliding distance (from 0 to 50 m). In these figures, the development of the equilibrium state in the COF and galling area is demonstrated in the friction (Fig. 4.11) and galling (Fig. 4.12) evolutions at various contact conditions. It can be seen that when the contact load increases from 2 N to 10 N, the increase rate of coefficient of friction gradually rises. The evolutions of COF reach a pressure-independent value of 0.55. In galling evolutions, the increase of contact load not only increases the formation rate of galling area, but also increases the saturated galling area at the equilibrium.

4.6 Conclusions

During the dry sliding between an aluminium alloy and cast iron, the aluminium transfer process was found to be dependent on sliding distance and contact load. Furthermore, it was revealed that this transfer process had induced the transition in the frictional mechanisms and thus affected the evolution of the COF. The interaction between friction, wear and material transfer on aluminium adhesive wear was analysed and an interactive friction model was developed to enable the prediction of the evolutions of the COF and the galling area from the running-in state to steady state. This model can be used to estimate friction variation and aluminium transfer caused by adhesive wear in the aluminium sliding process. These dependencies and interactions can be drawn as follows:

- Sliding distance dependency: the aluminium transfer experiences a transition from a running-in state, in which the transfer amount gradually increases until to a dynamic equilibrium, namely, the steady state. Meanwhile, COF begins at a low initial value and increases to a constant when the tribo-system reaches the steady state; wear rate begins at a high initial value and gradually reduces to a low constant at the steady state.

GALLING UNDER CONSTANT CONTACT LOAD

- Contact load dependency: a high contact load will shorten the running-in state and increase the saturated transfer amount as well as wear rate at the steady state.

Chapter 5

Application of the interactive friction model to the prediction of galling on load change conditions

5.1 Summary

¹The galling process remains one of the least understood mechanics of boundary conditions in metal forming. While it has long been established that coefficient of friction (COF) is a system parameter rather than a material parameter, the interaction in aluminium transfer, that is generated on a tool surface can cause evolutionary COF increase, has remained elusive. Inaccuracy is caused by using a constant COF in simulations. The aluminium transfer process, which has history and pressure dependency, is determined by a dynamic balance between the generation and ejection of third body at the contact. The application of sheet metal forming provides dynamic pressure variation due to rapid contact of tools, which will disequilibrate and rebuild the dynamic balance and interactively affects the evolution of galling and COF. To address this dynamic interactive phenomenon, pin-on-disc dry sliding

¹This chapter is based on paper work ‘Y. Hu, Y. Zheng, D. J. Politis, L. Wang, *Interaction between friction and galling during sliding of an aluminium alloy against cast iron under contact load changing conditions: Experiments and modelling*’.(submitted)

tests with an aluminium alloy 6082 pin sliding against a cast iron G3500 disc were conducted under a step load change condition. The COF evolutions, morphologies of the transfer layer and its cross-section were studied. To validate the interactive model that developed in Chapter 4, it is applied to the dynamic formation process of the aluminium third body layer under load change. A good agreement between experiment and modelling result has been obtained to advance multi-cycle simulation of friction variation and galling distribution.

5.2 Introduction

Galling is a severe adhesive wear progress in industry whereby during the sliding contact of a tool material against a work piece, the soft material may adhere to the counterpart. This phenomenon has widely existed in aluminium sheet metal forming processes where after repeated operations, a transfer layer originating from an aluminium workpiece was formed on the tool surfaces (Pujante et al., 2015, Podgornik et al., 2004, Olsson et al., 2002, Riahi and Alpas, 2007). This transfer layer can result in damage of workpieces and tool surfaces including scratches on the product, damage of functional surfaces such as coating and surface texture, misalignment of tools and low drawability. Although the application of lubricant and coating can partly suppress aluminium transfer (Dong et al., 2017, Olsson et al., 2002, Heilmann et al., 1983), there is still no effective solution to prevent galling in aluminium forming. To enable the prediction of galling distributions and friction variations within a tool and to further predict tool life, there is an urgent requirement to study and model the progress of aluminium galling.

Galling is a system response rather than a material property. It depends on contact conditions, such as counterpart material, surface roughness, sliding distance, contact pressure and temperature, as well as the contact history, which cause numerous interrelationships and interdependencies. The transfer mechanism has been identified as wear particles that detach from the workpiece and transfer to the counterpart by either adhering as lumps or covering as loose wear particles (Landheer

and Zaat, 1974, Schedin, 1994, Heilmann et al., 1983, Moshkovich et al., 2010). These transferred tribo-materials, were called ‘third body’, are produced by mechanical and chemical interactions of the contact couple during sliding, and can be both macro/micro structurally and chemically different from the source material (Rigney and Karthikeyan, 2010, Mishra et al., 2007, Scharf et al., 2010, Moshkovich et al., 2010). The chemical composition of the aluminium third body formed at room temperatures with steel was suggested as a combination of aluminium-oxygen solid solutions and hydroxylated materials (Kim et al., 2007a, Kim et al., 2007b). The initial surface roughness of the contact pair was found as an important trigger for transfer where a high surface roughness may result in severe galling (de Rooij and Schipper, 2001, de Rooij et al., 2013, Podgornik et al., 2004, Hanson et al., 2009, Schedin, 1994). History and pressure dependency of galling were studied in the former research. The formation of a transfer layer experiences two historical states with sliding distance increase: 1) in the running-in state, the mass of aluminium transfer layer gradually increases until 2) a critical level (Schedin, 1994), namely the steady stage, which is characterised by a dynamic equilibrium of the mass/volume (Fillot et al., 2007), the thickness (Österle et al., 2012, de Rooij and Schipper, 2001, de Rooij et al., 2013) and the galling area of the transfer layer (Rabinowicz, 1965, Podgornik et al., 2004); It was also found that both the formation rate and the saturated volume of the transfer layers increase with the contact load increase (Österle et al., 2012). In the early research by Bowden and Tabor (2001), adhesive wear was described as forming and breaking of adhesive junctions, which suggested that tangential stresses may be simultaneously generated with detachment and transfer of fragments. In aluminium against tool steels, this interaction between aluminium transfer and friction was widely observed as the coefficient of friction (COF) was increased by the formation of aluminium transfer layer at the running-in state (de Rooij and Schipper, 2001, de Rooij et al., 2013, Olsson et al., 2002, Heinrichs et al., 2012, Dong et al., 2017) to a stable COF value at the steady state (Dwivedi, 2010). It was predicated that the study of friction, wear and transfer in adhesive wear would be developed into a holistic interactive investigation (Dwivedi, 2010,

Kato, 2000, Bhushan, 2013).

There are three main tribological responses of an adhesive wear system, namely, adhesive wear, material transfer and friction. The evolution of adhesive wear is a well-discussed response from tribo-systems. The Holm-Archard equation and its variants provided great flexibility for modelling wear volumes at the equilibrium state (Kato, 2002, Dwivedi, 2010, Deus et al., 1997, Fillot et al., 2007, Hu et al., 2015, Ma et al., 2015, Bhushan, 2013), and the transition of wear evolution in the running-in state was modelled by Yang (Yang, 2003, Yang, 2005). Based on these adhesive wear models, the interaction between the responses of friction and transfer were modelled: a statistical model for describing the growth of lumps and the generating of adhesion force has been developed (de Rooij and Schipper, 2001); and subsequently a model for material transfer on a single asperity scale was developed by the same author (de Rooij et al., 2013), in which the morphology variation and the interactive resisting force generated by a lump was modelled. The concept of solid third body was gradually applied into galling analysis to holistically investigate the three responses (Godet, 1990, Fillot et al., 2007, Diomidis and Mischler, 2011), in which it was explained that friction and wear are simultaneously derived from the interaction between the original mating pairs and the transfer layer. In the third body model (Fillot et al., 2007), the formation of transfer layer was described as a competition between the generation and ejection of wear particles until a dynamic equilibrium is reached. However, although these physics-based models provide insights into the chemical and mechanical nature of galling, the complexities limit their further industrial application.

On the other hand, modelling of the friction evolution caused by galling can be treated as a nonlinear dynamic problem in tribological modelling. Instead of understanding the sophisticated interactions between friction, wear and transfer generations, dynamic models focus on accurate modelling of responses in order to model the correct operation of a system. The history and pressure dependencies of friction were addressed in many dynamic friction models (Marques et al., 2016, Berger, 2002) and many were even specially designed for sheet metal forming (Kim

et al., 2012, Lee et al., 2015, Gil et al., 2016, Tamai et al., 2016, Wang et al., 2016, Hu et al., 2017a). However, these non-linear dependencies were always caused by lubrication, abrasive running-in and the start/stop of the tribo-system rather than galling. Although the interaction between friction and galling has been confirmed in the literature, there are almost no dynamic models developed to model the formation process of galling. Nevertheless, the application of sheet metal forming also provides some unique dynamic characteristics compared to other dynamic systems: 1) due to the sudden contact and detachment of tools, rapid change of contact load, or ‘load jump’, which widely exists in sheet forming (Karupannasamy et al., 2014, Deng et al., 2017); 2) the shape complexity of forming tools combined with the flow of workpiece material generates uneven spatial and historical distribution of sliding distance and contact pressure and thus leads to an uneven distribution of galling and friction on tools (Pereira et al., 2010). These characteristics provide challenges both to the existing friction/wear models as well as the previous study of galling mechanisms.

In the present study, the pressure and history dependencies as well as the dynamic effect of load jump on aluminium adhesive wear and its interaction between the aluminium transfer and the COF increase in the running-in state at room temperature are presented. Experiments were conducted to explore the transfer mechanisms under load change conditions. Results were analysed by COF evolutions and micrographics. An interactive friction model that addressed both the galling mechanism and the dynamic responses was applied to predict the galling area and COF variations with the formation of aluminium transfer layer over time.

5.3 Materials and methods

5.3.1 Materials

Pin-on-disc tests were designed to determine the effect of contact load change at the running-in state on aluminium adhesive wear in order to: 1) quantitatively measure its effect on the COF and galling area, and 2) to verify an interactive

friction model. The material used for the pin was aluminium alloy AA6082 T6 (flat contact with a diameter of 2 mm, $R_a = 486 \pm 35$ nm, hardness = 115 ± 5 HV) representing the workpiece material. The material used for the disc was cast iron G3500 ($R_a = 126 \pm 19$ nm, hardness = 209 ± 18 HV) representing the tool material.

5.3.2 Tests methods

The experiments were performed on an Anton Paar tribometer via a pin-on-disc setup at room temperature with a linear sliding speed of 50 mm/s and a wear track radius of 5 mm. The flat contact between pin and disc was self-aligned in less than ten laps at the beginning of tests. Tests were performed according to Table 5.1. The step change of contact load was characterised by stages with a certain sliding distance. Two conditions of contact load change, load increase (test 1) from 2 to 10 N and decrease (test 2) from 10 to 2 N, were conducted. The change in load was conducted semi-automatically on the Anton Paar tribometer. In the control software, the load and sliding distance of each stage was set in advance. Then different dead weights were applied manually by following the instruction of the software during experiment. In order to obtain clear results of the effects of load change, the sliding distance of stage 1 was designed to reach approximately the middle of the running-in state; and stage 2 to reach the equilibrium state. For stage 1, 200 laps of sliding distance under 10 N and 100 laps under 2 N were set. COF evolutions were recorded in real time with an acquisition rate 160 Hz. Tests for generating COF evolutions were repeated for three times. The obtained COF data were smoothed and presented by average values with standard deviations (SDs). In addition, a series of tests conducted under constant load (Table 5.2) were designed to compare with tests 1 and 2. To minimise the disturbance of transporting samples, test 4 and 6 were conducted to represent the morphology condition at the end of stage 1 in tests 2 and 1 respectively. Tests 5 and 7 were conducted under constant load of 2 and 10 N respectively and ended at the equilibrium state enabling a comparison with the COF evolutions under load change conditions. The load effect on wear particles was studied by comparing test 3 (2 N) and 6 (10 N).

5.3.3 Characterisation of galling

After tests, surface morphologies were captured by moving samples from the tribometer. The morphology of the cast iron surfaces was studied by a Hitachi S-3400N scanning electron microscope (SEM), a Veeco WykoNT9100 white light interferometry (WLI) and a Zeiss Auriga-45-24 SEM with an in-situ focused ion beam (FIB). The electron source of Hitachi SEM was operated at accelerating voltages of 10.0 kV and secondary electron signals were collected. The cross-section was prepared by FIB operated at an energy of 30 kV and a probe current of 50 pA. The observation area was chosen at the central line of the wear track to minimise the effect of radius difference. The galling area fraction covering the cast iron was calculated from the WLI 3D surface plot by height. As the original surface is a Gaussian surface, after test, asperities with height of $3\sigma \approx 500$ nm above the mean surface were identified as the transfer layer. σ is the standard deviation of the original cast iron surface and the standard of 3σ makes 99.7% of the asperities on the original surface can exclude from the calculation. Particles with the major dimension smaller than 500 nm may be neglected by this standard. However, they are beyond the scope of the present chapter and it will be proved that have very limited effect on the macro-scope galling area. The disturbance of wear particles was minimized during transport and storage which made the observation of loose wear particles possible.

5.4 Results

5.4.1 COF evolution

Fig. 5.1 shows the COF evolution under load increase from 2 to 10 N. The COF was low at the beginning 0.18 and gradually increased to approximately 0.3 at the end of stage 1 (S1). Immediately after the abrupt increase of contact load to 10 N, a steeper slope was observed which gave a rapid increase of COF to a stable value of 0.55 at approximately 400 laps. Fig. 5.2 shows the COF evolution of load decrease from 10 to 2 N. Under 10 N, the COF began with a value of 0.2 and increased to

Table 5.1: Test matrix of load change test

Test No.	Stage I		Stage II		Notes	
	Load (N) /contact pressure (MPa)	Sliding laps /distance (m)	Load (N) /contact pressure (MPa)	Sliding laps /distance (m)		
1	2/0.64	200/6.28	10/3.18	600/18.85	Load S2(inc)	increase,
2	10/3.18	100/3.14	2/0.64	700/21.99	Load S2(dec)	decrease,

Table 5.2: Test matrix of constant load test

Test No.	Load (N) /contact pres- sure (MPa)	Sliding laps/distance (m)	Notes
3	2/0.64	100/3.14	
4	2/0.64	200/6.28	S1(inc)
5	2/0.64	1200/37.70	
6	10/3.18	100/3.14	S1(dec)
7	10/3.18	500/15.71	

approximately 0.42 at the end of stage 1 (S1). After the contact load decreased to 2 N, the increase of COF slowed down to a stable value of 0.55 after approximately 570 laps. It should be noted that the COF evolution experiences an unstable period after the decrease of contact load, which lasted approximately 500 laps with large fluctuation as shown in the frame of Fig. 5.2. Comparing to the COF evolutions obtained under constant contact loads, the change of increase rate before and after load change are clearly presented. In Fig. 5.3, the COF curve of load change and constant load almost coincide in the first stage under 2 N; they separate after 200 laps due to the rapid increase of COF caused by the load increase. The load change

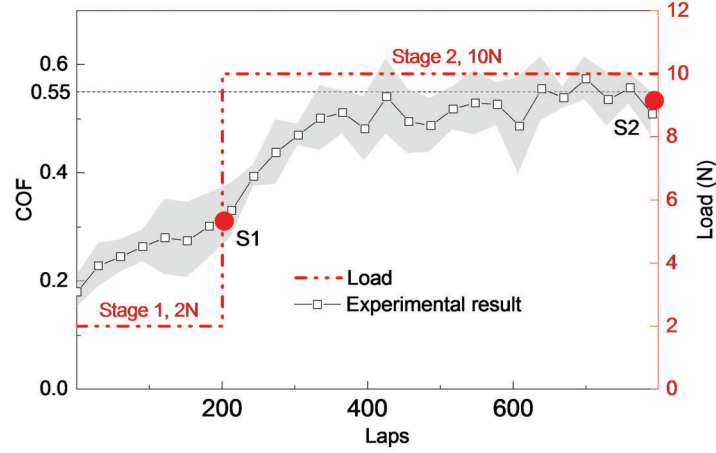


Figure 5.1: Effect of contact load increase on the evolution of COF. Symbols, represent COFs; envelopes, SDs.

system reached its equilibrium at 400 laps, which is between the equilibrium distance of 2 N (780 laps) and 10 N (220 laps). In Fig. 5.4, in the first stage under 10 N, the COF curve shows a rapid increase similar to the curve under the load of 10 N. The mismatch between curves of 10 N and 10-2 N at the initial 100 laps may be caused by system errors between different test batches. After a reduction of the increase rate of COF by decreasing the load, this system reached the equilibrium at a sliding distance of 600 laps, which is between the evolutions under 10 (220 laps) and 2 N (780 laps). The initial COF and the saturated COF at equilibrium obtained from different conditions were close to the value of 0.18 and 0.55, respectively, and thus they can be treated as pressure-independent constants in the studied load range.

5.4.2 Aluminium transfer layer analysis

Fig. 5.5 shows the topography of the ‘clean’ cast iron surface before the test. Compared with the topographies after tests (Fig. 5.6 and 5.7), some characteristics of the aluminium transfer which is independent of contact load can be observed. In Fig. 5.6B, the height distribution of the clean surface was mainly in the range of $0 \pm 0.5 \mu\text{m}$. Compared with the WLI pictures in Fig. 5.6 and 5.7, transfer layers

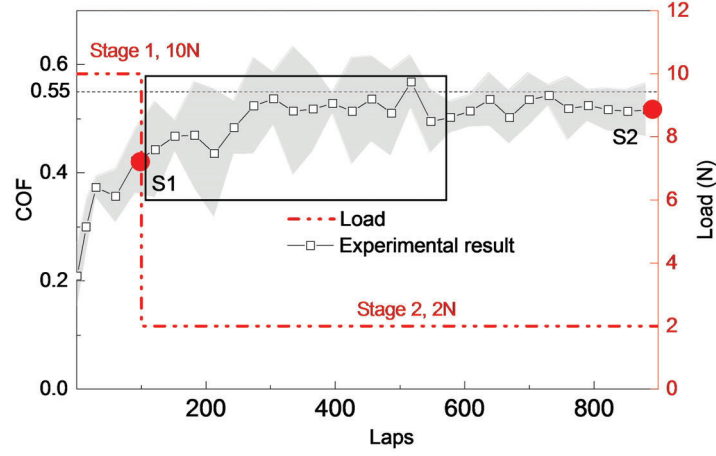


Figure 5.2: Effect of contact load decrease on the evolution of COF. Symbols, represent COFs; envelopes, SDs.

with a clear height increase were formed on the original surface. The top view of the transfer layers is presented by the SEM pictures in Fig. 5.6 and 5.7, which show that they were comprised of contact patches and loose wear particles. The formation direction of the transfer layer can be clearly observed in both WLI and SEM pictures from ploughing grooves and transfer tracks.

Fig. 5.6 shows the SEM and WLI topographies of the cast iron surface at end of sliding under 2 N (S1(inc)) and later 10 N (S2(inc)) in the load increase test. In stage 1, small peaks with the average height of $1\text{ }\mu\text{m}$ were generated on the cast iron surface in the direction of sliding (Fig. 5.6A). A slight transfer layer composed of transfer patches can be observed in the SEM picture at this stage (Fig. 5.6B). The enlarged picture (Fig. 5.6C) shows the wear layer was formed by individual transfer lumps (dark coloured transfer patch surrounded by bright boundary) and wear particles (bright particles). The transfer lumps appear to be built in an ellipsoidal shape with the major dimension from 10 to $50\text{ }\mu\text{m}$ being aligned in the direction of sliding with micro-scale wear particles scattering around. Overlaps between lumps were rare but can be observed. After sliding under stage 2 of 10 N, significant height increase of the transfer layer (average of $3\text{ }\mu\text{m}$) can be observed in Fig. 5.6D. Large transfer patches (dark coloured area with major dimension from 50 to $300\text{ }\mu\text{m}$ in the

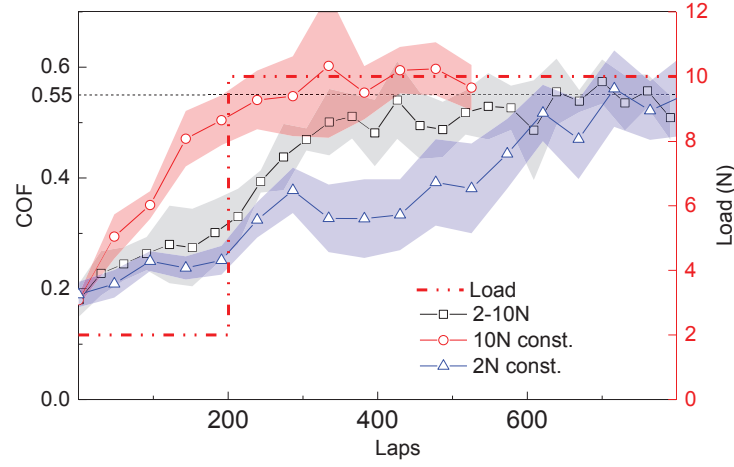


Figure 5.3: Comparing of COF evolutions between load increase and constant loads. Symbols, represent COFs; envelopes, SDs.

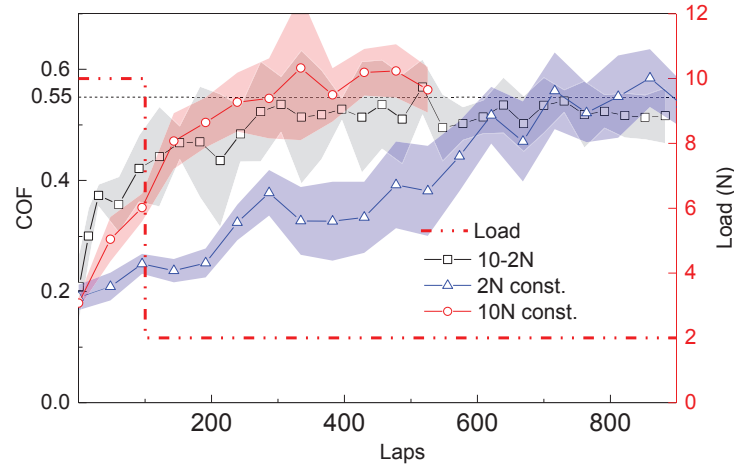


Figure 5.4: Comparing of COF evolutions between load decrease and constant loads. Symbols, represent COFs; envelopes, SDs.

direction of sliding) can be observed (Fig. 5.6E). It is shown in the enlarged view (Fig. 5.6F) that, the dark coloured transfer patches were composed of large transfer lumps (major dimension approximately $100\text{ }\mu\text{m}$). Boundaries between lumps and on the top of transfer patches can be observed, which indicates the formation process of either accumulating or overlapping of wear particles. Scattered small wear particles

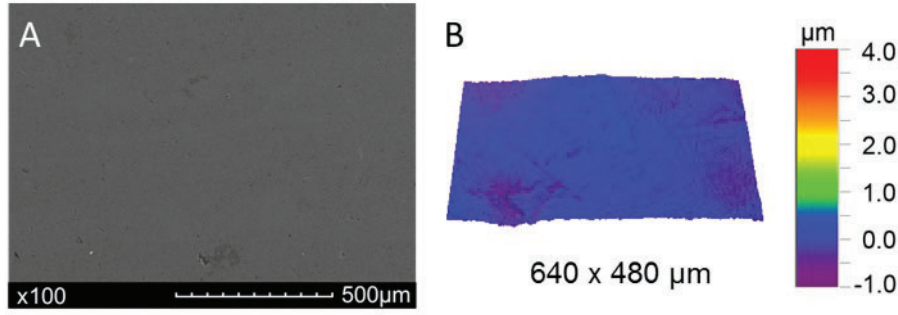


Figure 5.5: (A) SEM and (B) WLI topography of the original cast iron surface.

can also be observed around the transfer patches.

Fig. 5.7 shows the SEM and WLI topographies of the cast iron surface at the end of sliding under 10 N (S1(dec)) followed by 2 N (S2(sec)) in the load decrease test. In the WLI picture of state 1 (Fig. 5.7A), a large amount of high transfer layer was generated with an average height of approximately $2\ \mu\text{m}$. In the SEM picture (Fig. 5.7B), transfer patches were formed in the direction of sliding with the size and pattern close to the observation of S2 (inc) but with low density and height. In the enlarged picture (Fig. 5.7C), boundaries on the top surface of the transfer patches formed by overlapping and accumulation can also be observed, which is similar to S2 (inc). After sliding of stage 2, the height of high peaks was close to the condition of S1 (dec), but a few small peaks in the range lower than $1.5\ \mu\text{m}$ grew at this stage (Fig. 5.7D). In the SEM picture (Fig. 5.7E), these layer of small peaks were observed as a bright transfer layer composed of small wear particles that covered the whole wear track. In the enlarged view (Fig. 5.7F), a large transfer lump (major dimension of $80\ \mu\text{m}$) can be observed under the layer of small wear particles. This layer particularly accumulated in the edge area towards the direction of sliding as shown in Fig. 5.7F by a purple shadow.

5.4.3 Wear particle analysis

Pictures of wear particles generated under 10 N (test 6, Fig. 5.8A-D) and 2 N (test 3, Fig. 5.8E-H) were captured on the outer boundary of the wear track. Under the contact load of 10 N, large wear particles (with major dimensions between 50

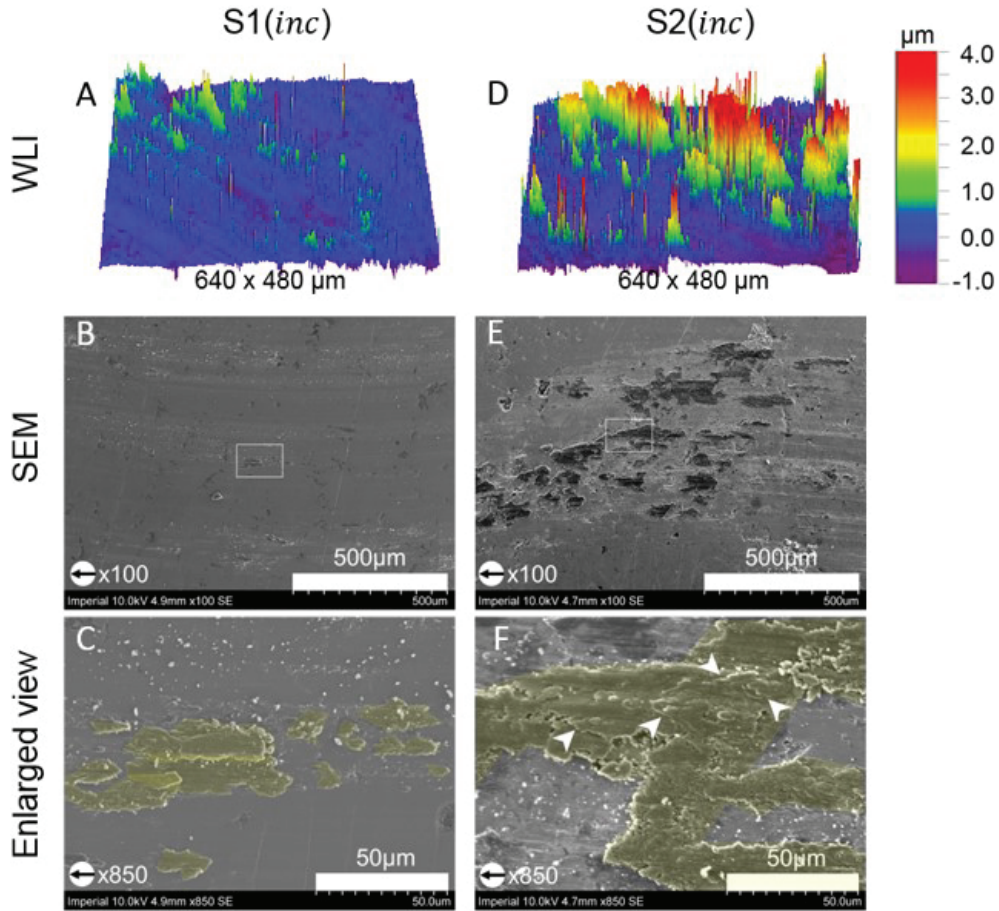


Figure 5.6: Topographies of wear tracks at load increase condition: (A-C) WLI and SEM topography at the end of sliding under 2 N (S1(inc)); (D-F) WLI and SEM topography at the end of sliding under 10 N (S2(inc)). White arrow: boundaries between individual lumps; yellow area: transfer patches; black arrow: sliding direction.

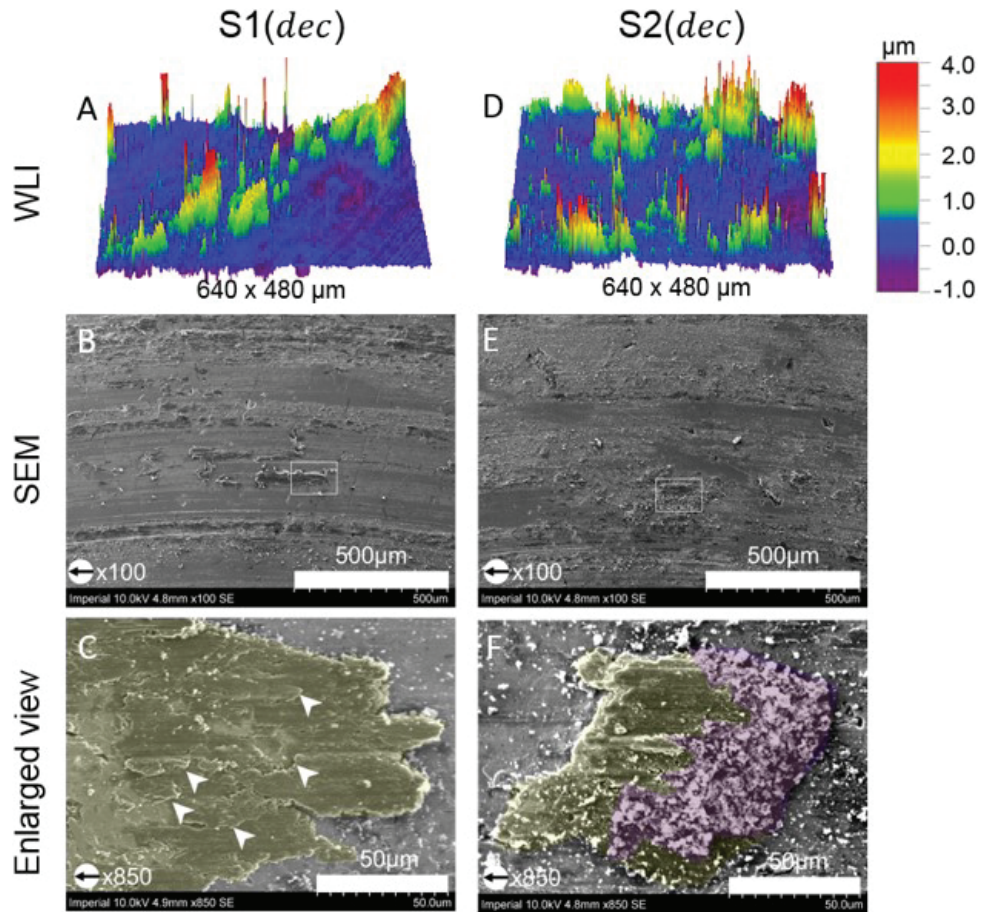


Figure 5.7: Topographies of wear tracks at load decrease condition: (A-C) WLI and SEM topography at the end of sliding under 10 N (S1(dec)); (D-F) WLI and SEM topography at the end of sliding under 2 N (S2(dec)). White arrow: boundaries between individual lumps; yellow area: transfer patches; purple area: layer of small wear particles; black arrow: sliding direction.

and 300 μm) were generated; under 2 N, the single wear particles are small (with major dimensions less than 30 μm) and likely to cluster together (Fig. 5.8F and G). Independent of contact load, certain shapes of single wear particles were observed throughout the tests and can be classified into two categories: 1) chip shape particles (Fig. 5.8A, B and H) and 2) typical adhesive wear particles (Fig. 5.8D, E, F and G). Chip shape, which may adhere on the counterpart but detach, is characterised by its ellipsoidal or semi-ellipsoidal outline with ploughing grooves or transfer layer on the top surface in the direction of sliding. For example, in Fig. 5.8A, a chip-shape wear particle with contours of small transfer lumps can be observed; its former sliding direction can be inferred by both the direction of its outline major axis as well as the grooves and small transfer lumps. In Fig. 5.8B and H, chip shape particles with relatively flat surfaces and shallow ploughing grooves in the direction of sliding when compared to Fig. 5.8A may indicate their top surfaces shown in the pictures, were adhered on the cast iron surface instead of contacting the counterpart. In contrast, adhesive wear equiaxed particles present irregular shapes with rough regions of typical shear fracture (Stachowiak et al., 2008). For the small irregular wear particles observed at 2 N, the edges present a flake-like shape, which is more likely to accumulate together and form loss structures as shown in Fig. 5.8D and F. In Fig. 5.8D, a special cluster was observed, which was formed by a chip-shape wear particle in the middle clustered with adhesive wear particles. In Fig. 5.8F, a rope-like cluster was formed with different size and type of small wear particles.

5.4.4 FIB cross section analysis

Fig. 5.9 presents tilted and cross-sectional SEM images of aluminium transfer patches formed on the wear track of test 1. In Fig. 5.9A, transfer patches covered most areas of the cast iron wear track. Boundaries illuminated by yellow shadows separated a few large transfer lumps. A cross section with the width of 4 μm was prepared by FIB at a joint of the boundaries to enable the observation of the historical formation process of the laminar transfer layers. In the enlarged view (Fig. 5.9B), the boundary between the cast iron substrate and the transfer layer can be clearly

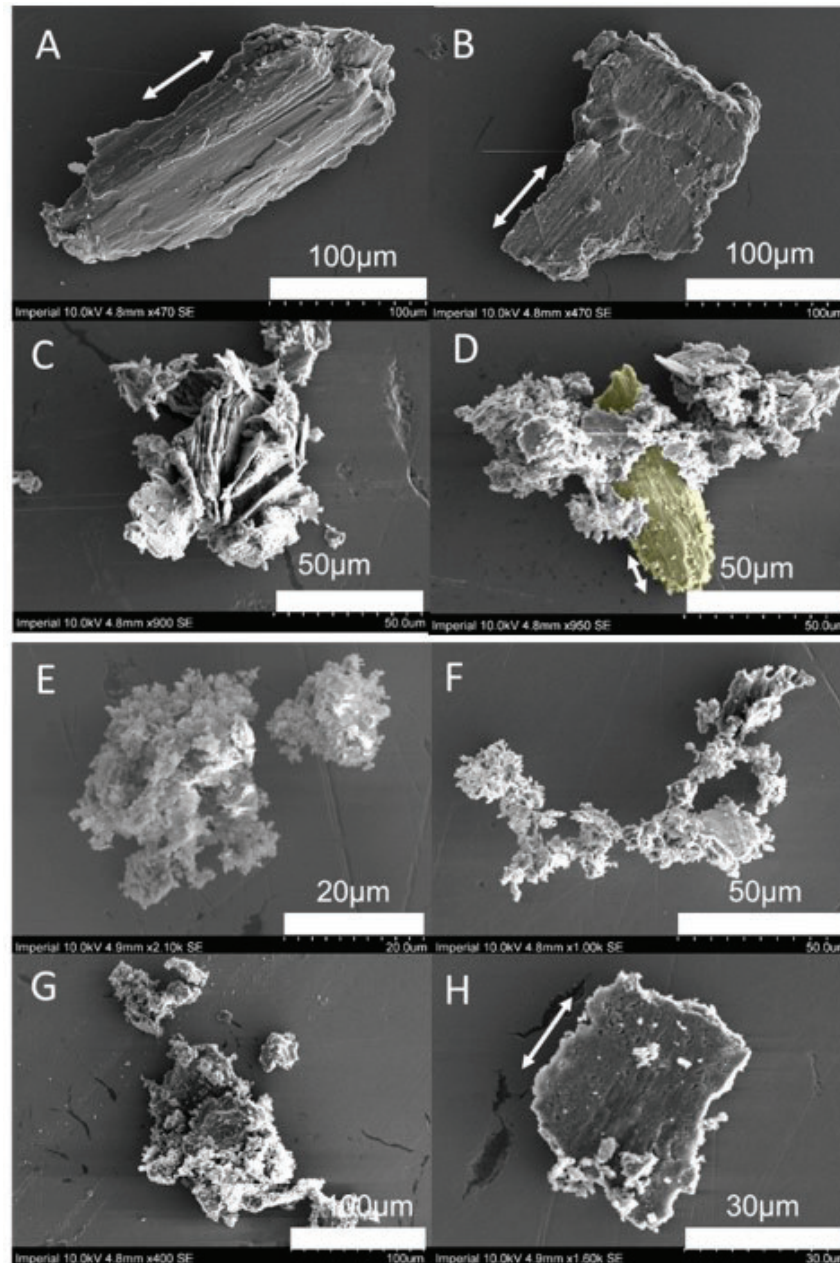


Figure 5.8: SEM images of aluminium wear particles obtained at the contact load of (A-D) 10 N and (E-H) 2 N. Arrows: sliding directions shown by ploughing grooves; yellow area: ship shape particles.

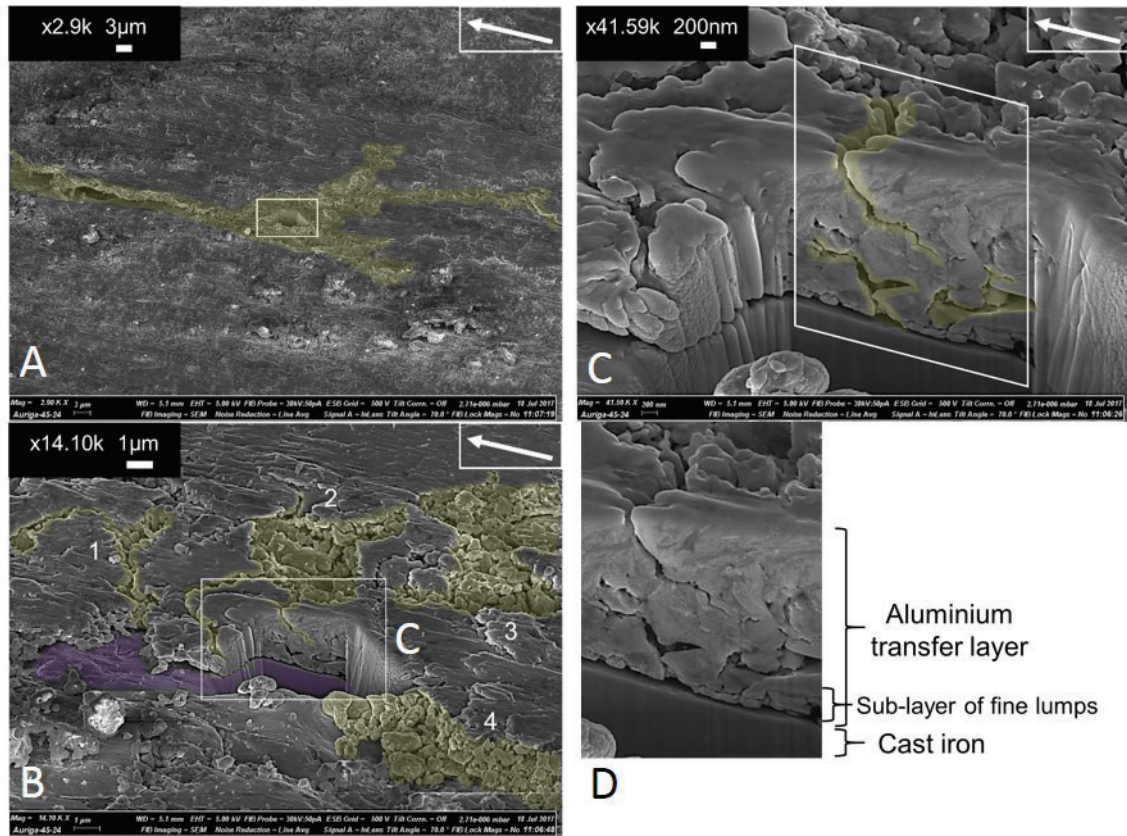


Figure 5.9: Tilted topographies of the structure of aluminium transfer layers taken on test 1: (A, B) cross-sectional images through FIB sample preparation, (C) enlarged view of the cross-section and (D) schematic diagram of layers in the cross-section. Yellow areas: boundaries between wear particles; purple area: cast iron substrate; white arrows: sliding direction.

observed in the cross-section area. The cross-section of the cast iron presents homogeneous properties with no clear deformation or crack being observed. In contrast, boundaries between wear particles gives an insight of the deformation history where the wear particle experienced plastic deformed from loose wear particles of irregular shapes shown in Fig. 5.8 into chip shape transfer lump during sliding. As shown in Fig. 5.9B, transfer lumps may be crushed and accumulated compactly and leave small spaces at the edge (e.g. Fig. 5.9B boundary 1). Otherwise, lumps may accumulate loosely with large gaps formed between them, which were later filled by small wear particles (Fig. 5.9B boundary 2, 3 and 4). A similar phenomenon can

also be observed beneath the top surface in Fig. 5.9C, where the boundaries between wear particles extended on the cross section and void spaces were formed by loose accumulation. A schematic diagram of the laminar structure of aluminium transfer layer is shown in Fig. 5.9D, in which a sub layer of fine wear particles can be observed at the bottom of the transfer layer.

5.4.5 Galling area calculation

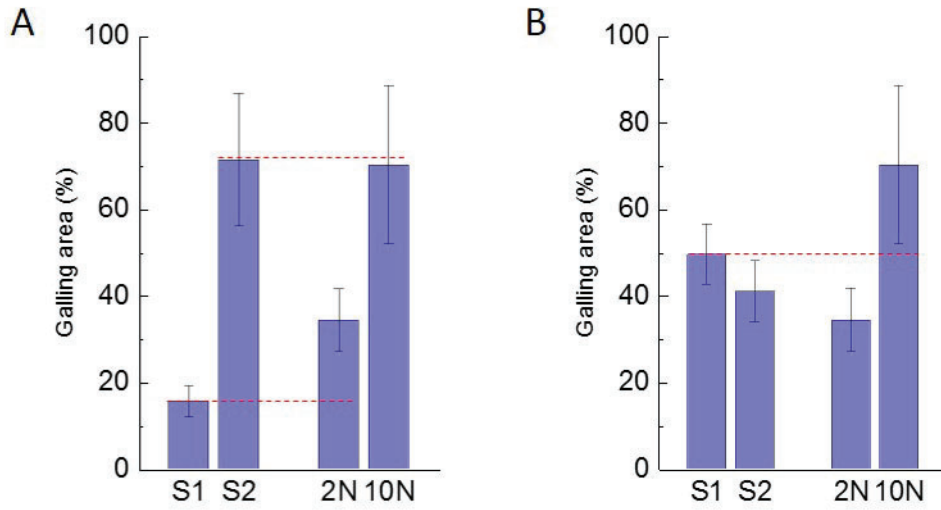


Figure 5.10: Galling area (%) at S1 and S2 under the load increase (A) and decrease (B) conditions compared to the saturated values obtained at constant conditions.

The results of the galling area at different stages under the load increase and decrease conditions are shown in Fig. 5.10. It was found that galling area fraction, which is defined by the area of cast iron covered by aluminium transfer layer, is a reliable and feasible parameter of measuring the formation process of a transfer layer as well as modelling the contact area between solid aluminium and its transfer layer. The galling area starts from zero at the beginning and gradually increases to a pressure-dependent saturated value with the tribo-system reaching its equilibrium (Rabinowicz, 1965, Podgornik et al., 2004). The saturated galling areas obtained under constant contact load of 2 and 10 N are plotted in Fig. 5.10 and compared

with results obtained from load change tests. In Fig. 5.10A, stage 1 stopped before equilibrium and showed a galling area of 15.6%, which is lower than the saturated value (34.3%) under the same load of 2 N. After sliding at stage 2 under 10 N when the system reached, a final galling area of 71.4% was obtained, which is almost equal to the saturated galling area under constant load of 10 N (70.2%). In Fig. 5.10B, stage 1 stopped before equilibrium under 10 N and showed a galling area of 49.5%. This value is lower than the saturated galling area under a constant load of 10 N (70.2%) but higher than the saturated galling area under a constant load of 2 N (34.3%). After sliding at stage 2 under 2 N, the system reached equilibrium, and a galling area of 41.1% was obtained. This value is lower than the galling area achieved at stage 1 but still higher than the saturated galling area under a constant load of 2 N.

5.4.6 COF angular map

Fig. 5.11 shows the angular maps of COF evolution from test 1 and 2. The x-axis is set as the angle from 0 to 360 corresponding to a cycle of rotation of the cast iron disc. Number of laps were plotted in the y-axis. An acquisition rate of 160 Hz was set to generate approximately 100 points per lap. The coloured bar shows the COF. These angular maps show a spatial and historical distribution of aluminium lumps along the wear track. As the COF increase stems from aluminium transfer, a red spot in the angular maps of COF evolution indicates a high COF points where a cluster of aluminium patches or wear particles were accumulated at a certain angle on certain laps on the cast iron surface.

The 2 N and 10 N loads demonstrated different features on the angular maps . As shown in Fig. 5.11A from 0 to 200 laps and Fig. 5.11B from 100 to 350 laps, high COF points generated under 2 N load presented a dense but short period of high COF, which lasted less than 10 degrees rotation angle. Comparing to Fig. 5.11A from 200 to 550 laps and Fig. 5.11B from 0 to 100 laps, high COF points generated under 10 N present a sparse pattern with a relatively long period. This can be caused by: 1) 2N was likely to generate small and dense transfer lumps, which is

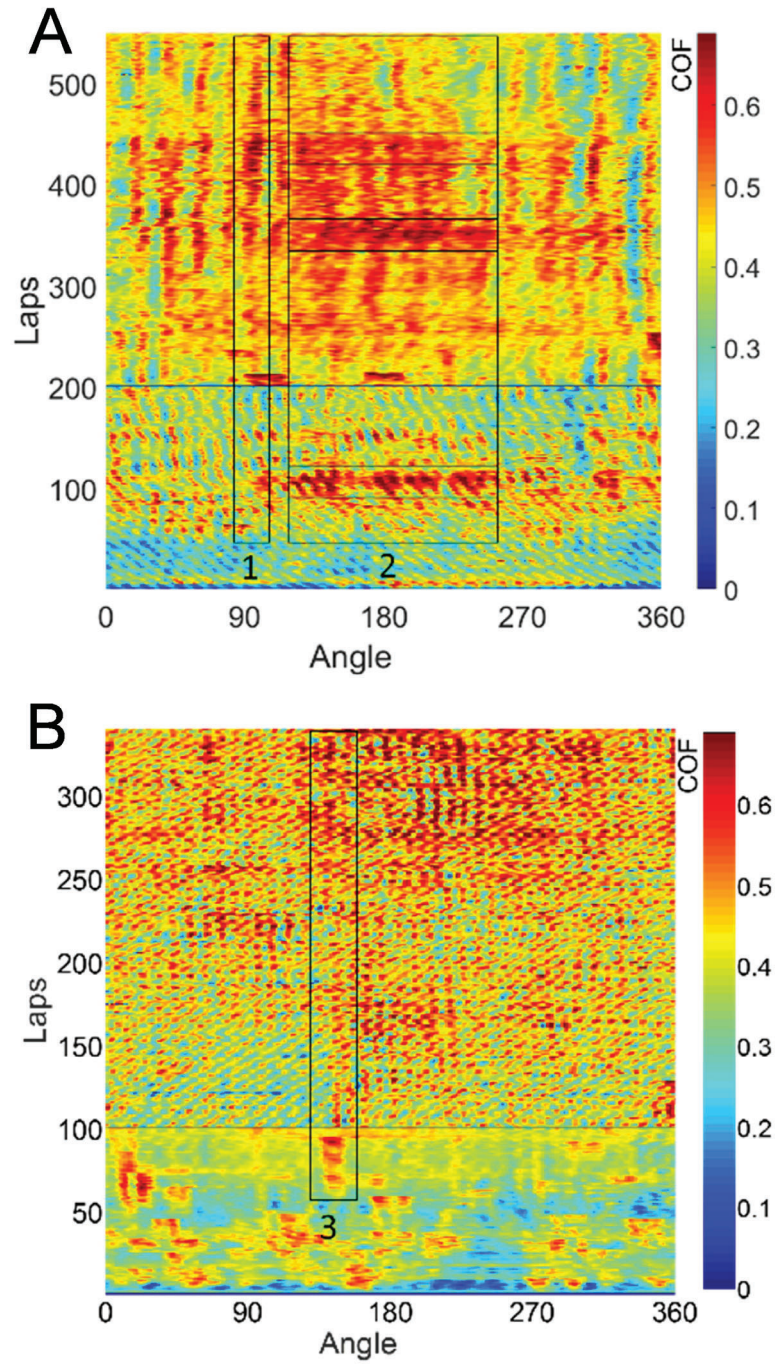


Figure 5.11: Angular maps of COF evolutions: (A) load increase test; (B) load decrease test.

illustrated by SEM micrographs (Fig. 5.6 and 5.7), and 2) the small inertia of the weight of 2 N made it easy to generate a signal by a small disturbance on the wear track.

When an individual lump was formed, it was likely to survive and accumulate more wear particles and finally formed a large transfer patch (Heinrichs et al., 2012, Schedin, 1994). This phenomenon can be illustrated with the angular maps by chains of high COF points formed in the direction of the axis of laps, such as area 1 in Fig. 5.11A, which is a cluster of lumps that were initially formed at approximately 45 laps in the angle of 95° under 2 N, grown under 10 N at the same degree until the end of the test. This survival feature is the most common characteristic of the galling phenomenon and can be observed all over the angular maps. Instead, some lumps may not survive during the whole sliding progress. As shown in Fig. 5.11A in area 2, there is a zone of high COF formed at 120 laps from the angle of 110 to 250. This zone disappeared (or known as healing) under 2 N load condition but regenerated at the 10 N load condition, which indicates that lumps may be ejected as wear particles but regenerated by secondary transfer based on the former transfer patches. Also, these mechanisms can continuously be observed through load changes. As shown in Fig. 5.11A area 1 and 5.11B area 3, continuous red lines can be observed from increase and decrease of contact load.

5.5 Discussion

5.5.1 Relation between friction, wear and aluminium transfer layer

It is well observed that the aluminium transfer layer strongly affects the aluminium-steel tribo-system by its responses. The effects include a change of the interface characteristics through physical-chemical interactions; and a further change from tribology between solid aluminium against solid cast iron to a much more complex system including third body. That is, when an aluminium third body layer forms on the cast iron surface, it partly separates the original contact between solid aluminium

and cast iron and presents a new physical/structural and chemical morphology of the interface compared to the initial ‘*clean*’ contact. The roles, 1) separating the aluminium-cast iron contact couple, 2) transmitting contact load and shearing, and 3) triggering further wear and transfer, are determined by the tribo-system of third body instead of the original contact and thus cause an increased number of interrelationships and interdependencies between friction, wear and transfer.

The friction and wear mechanism in the aluminium-cast iron tribo-system is identified as a mixture of adhesion and abrasion by the observation of transfer patches and ploughings caused by severe adhesive wear and plastic deformation as shown in Fig. 5.6 and 5.7. At the beginning of the sliding process, friction and wear is generated by the direct contact between solid aluminium and cast iron. Contact junctions as initial transfer were formed, which may break by shearing inside the soft aluminium asperities and result in subsequent attachment of aluminium to the cast iron surface and simultaneously generate friction and wear. Transfer patches can also be formed by adhesion of loose wear particles, which contributed to friction and wear by shearing and pressing. As a result, this initial aluminium-cast iron contact generated a COF of 0.18 (shown in Fig. 5.1, 5.2, 5.3 and 5.4) and a high initial wear rate (Moshkovich et al., 2010). After a layer of aluminium had been generated and considerably covered the cast iron asperities, friction and wear should partly stem from cohesion and abrasion between aluminium transfer layer, loose wear particles and solid aluminium. Several factors may contribute to the COF values. At this stage, aluminium wear particles tended to deform plastically, be compressed and agglomerated into large wear particle clusters (Fig. 5.8), which may roll at the interface to accommodate the velocity difference between the contact couple and partly reduce friction and wear (Wang et al., 2010, Zhou et al., 2010). Shearing stress may also be reduced by sliding between soft aluminium instead of cast iron asperities. On the other hand, friction may be increased by cohesion of aluminium, which builds stronger metallic bonds than aluminium-cast iron bonding and thus gives a high self-mated COF (Bhushan, 2013). Also, it should be noted that, although the asperities of aluminium lumps were subjected to a flat-

tening process and reached stable structures with a pressure-dependent height and galling area, the integral surface roughness increased due to the material transfer and increased the contribution of ploughing friction. Experimental results of COF increasing show that the mixture of adhesional and ploughing friction may play a dominant role in determining friction than simple mechanical interaction. As shown in the COF evolutions in Fig. 5.1, 5.2, 5.3 and 5.4, COFs were increased to and maintained a pressure-independent value of 0.55 when the transfer layer reached the equilibrium. The wear evolution in the aluminium-cast iron tribo-system follows a general adhesive wear model where the initial contact between rough cast iron and aluminium generates a high wear rate on solid aluminium. This wear rate reduced significantly after the soft aluminium transfer layer is formed and gradually polished during the running-in stage, which limits ploughing and abrasion. The wear rate finally became a constant value at the steady state, which can be predicted by the Holm-Archard equation ($h = kPx/H$, where h is the wear rate, k is the wear coefficient, P is the contact pressure, x is the sliding distance and H is the hardness of the soft material) (Yang, 2003, Yang, 2005).

5.5.2 Effect of sliding distance

In the aluminium-cast iron system, the aluminium transfer layer is formed gradually with increasing sliding distance. At the initial stage of sliding, adhesive and abrasive wear are highly active and were observed by the high initial wear rate. Combined with the effects of external force of shearing and pressure, as well as the high adhesion characteristic of aluminium against steel (Suh, 1986, Bhushan, 2013, Dong et al., 2017), the generated wear particles are likely to transfer to the cast iron by two main methods: 1) transfer against rough surface and 2) transfer by adhesive bonding (Heinrichs et al., 2012, Hu et al., 2017b). These initial transfers may trigger secondary transfer by adhering and accumulating loose wear particles on the formed lumps and further building multi-layer structures as shown in Fig. 5.6, 5.7, 5.8 and 5.9. Instead of infinite growth, the formation of aluminium transfer layer will reach a critical condition. In a single lump scale, an aluminium lump may

reach a critical size and shape and be mechanical stable rather than breakdown (de Rooij et al., 2013). However, this mechanical stability may be broken by detachment and ejection. As observations show clear boundary and void spaces between the transfer layer and the cast iron substrate (Fig. 5.9), as well as the detachment of integral aluminium lumps once adhered on the cast iron surface (Fig. 5.8), a firm bonding between aluminium and cast iron may not always exist. In a macroscopic view, the generation and ejection of wear particles and transfer lumps may be active throughout the sliding process; transfer and back transfer may also exist between solid aluminium, transfer layer and loose wear particles (Rabinowicz, 1980, Dwivedi, 2010, Deuis et al., 1997, Riahi and Alpas, 2007, Korres et al., 2012). As revealed by the angular map of COF (Fig. 5.11), transfer patches were gradually accumulated at certain positions on the wear track by initial transfer and may trigger secondary transfer at the same position. Some transferred lumps may not survive during sliding, but the rough transfer patches they leave can still trigger a further secondary transfer. Eventually, a pressure-dependent steady state was reached as the COF, wear rate and the galling area become stable. At the equilibrium state, although wear particles were continually generated, the morphology of the interface was relatively stable. Redundant wear particles were ejected out of the contact and a dynamic equilibrium of transfer layer was maintained.

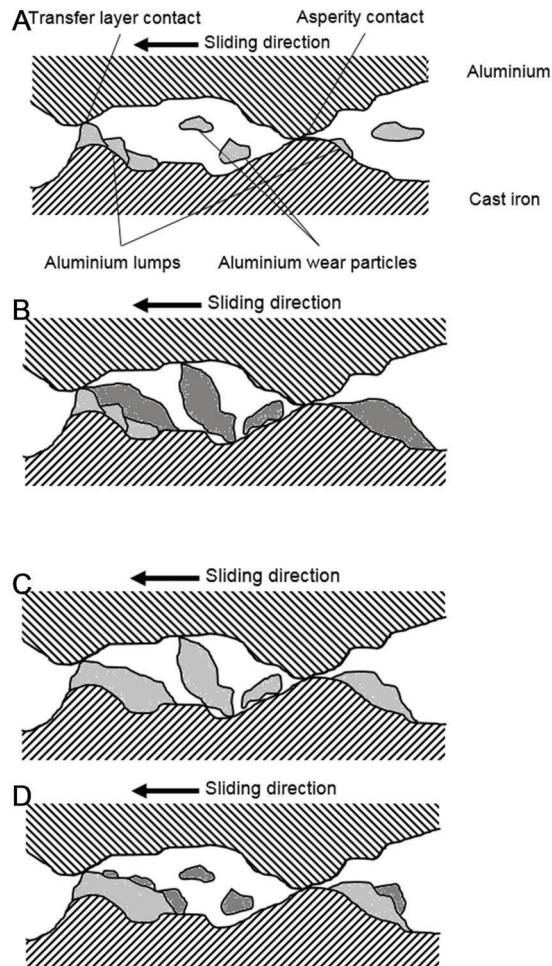
This dynamic balance between generation and ejection can be described by a tribological circuit (Fillot et al., 2007, Berthier, 2005). The instantaneous internal volume, $\dot{\rho}_i$, is the changing rate of the wear particle density of the transfer layer entrapped at the contact and determined by the rate of source density of wear particles (source rate, $\dot{\rho}_s$) which convects into the contact and the density rate of ejected wear particles (ejection rate, $\dot{\rho}_e$) from the contact. Based on the conservation of mass, an equilibrium of the third body changing rates at the contact can be written as Eq. 4.1. The rate of source density is correspondingly equal to the wear rate of the tribo-system. In the running-in stage, driven by a maximum source rate ($\dot{\rho}_s$), the source density (ρ_s) increases steeply from zero at the beginning of sliding and the source rate decreases as the sliding bodies gradually match with each other.

Meanwhile, the ejection rate increases from zero at the beginning and gradually approaches the value of source rate. This stage gives a net increase of instantaneous internal volume, $\dot{\rho}_i$. When reaching equilibrium, the decrease of $\dot{\rho}_s$ meets the increase of $\dot{\rho}_e$, i.e. $\dot{\rho}_s = \dot{\rho}_e = \text{constant}$, and $\rho_i = \text{constant}$ with $\dot{\rho}_i = 0$, a dynamic equilibrium is reached, which demonstrates that the amount of ejected particles were equal to the volume of new generated particles and give a zero net change of the volume of transfer layer at the contact interface.

5.5.3 Effects of load changing

Another important characteristic of the transfer layer formation process is its pressure dependency, which simultaneously affects the wear rate and transfer rate of the tribo-system. The effect of pressure on wear rate is well modelled by the Archard's equation and its derivations (Fillot et al., 2007, Dwivedi, 2010, Kato, 2002, Deus et al., 1997, Hu et al., 2015, Ma et al., 2015, Bhushan, 2013) where the contact load increase will lead to a rapid wear rate increase, which can be observed in experiments as large quantities and size of wear particles were generated after load increased (Costa et al., 2017, Xie and Bhushan, 1996, Zhang and Alpas, 1997). In addition, large aluminium lumps will be formed from large wear particles under high contact load, which increase both the transfer area and surface roughness and thus increase the possibility of secondary transfer and, furthermore, increase the transfer rate. Under constant loads, the effect of load is concluded as 1) the saturated galling area increases with contact pressure increase and 2) the duration of the transfer layer formation process is shortened by increasing contact pressure.

Schematic illustrations of the mechanisms under contact load changing are shown in Fig. 5.12. In the load increase condition (test 1), in stage 1 under 2 N, small wear particles were generated under a low wear rate, which led to a slow formation process of the transfer layer. The formed transfer layer had low height and small transfer patch and was unlikely to trigger further transfer, which caused the slow increase of COF in this stage. In the stage 2 under 10 N, with higher energy input on the contact, the detachment of aluminium became more severe. The wear particle size



Contact load increase:

- Under 2 N, small wear particles were generated.
- Small transfer patches were formed by loose wear particles

- When the contact load was increased from 2N to 10N, small wear particles were ejected and replaced by large ones at the contact.
- Large transfer patches were formed by large wear particles based on the existed transfer patches.

Contact load decrease:

- Under 10 N, large wear particles were generated.
- Large transfer patches were formed.

- When the contact load decreased from 10N to 2N, wear particles with smaller sizes were generated.
- Small transfer patches were formed on the existed large transfer lumps. Some large transfer lumps may be rejected during sliding.

Figure 5.12: Schematic illustrations and descriptions of the mechanisms of the transfer of aluminium to cast iron surface during sliding: (A-B) contact load increase; (C-D) contact load decrease. Light particle: initial transfer; Dark particle: secondary transfer.

dramatically increased and led to a wear rate increase. Large wear particles were likely to adhere on the initial transfer patches formed at 2 N by secondary transfer processes as shown in Fig. 5.12. Large transfer lumps were built and gradually accumulated more loose wear particles and finally formed a dense transfer layer. The galling area and COF showed a steep increase in this stage and finally, the tribo-system reached the equilibrium with a saturated condition under 10 N, which is indicated by the equality of the saturated galling areas in Fig. 5.10A.

In the load decrease condition (test 2), in stage 1 under 10 N, combining with a high wear rate, large wear particles were generated immediately and a relatively dense transfer layer with a galling area of 49.5% at the end of stage 1 was rapidly formed. Consequently, COF experienced a rapid increase under 10 N. The COF began at 0.2 and increased to 0.42 within 100 laps of sliding. After a load of 2 N was applied, small wear particles were generated, which gradually replaced the large particles generated under 10 N by ejection, and eventually dominated the transfer on cast iron surface. The transfer of small wear particles was more likely to be triggered by the large aluminium lumps formed under 10 N through the mechanism of secondary transfer which caused a complete change of the morphology of the transfer layer with a galling area of 41.1% (Fig. 5.10B) at the end of stage 2. This drop of galling area indicates a self-healing process occurred after the load change. In galling phenomenon, healing is reflected by removal of the formed transfer layer and reset of a new stable surface. When the contact load changed from high to low, the transfer layer might experience a process of self-mating to the low contact load, which led to a morphology change with the reduction of the galling area. This process may include recovering of the elastic deformation inside of the transfer lumps formed at high contact load (Karupannasamy et al., 2014), polishing/flattening the high asperities of aluminium lumps, accumulating small wear particles and ejection of unstable large transfer lumps. The increase of COF was largely reduced under stage 2, and due to the active evolution of the morphology, pronounced oscillations of COF were recorded during this healing process (Fig. 5.2). Estimated from friction measurement, it was found that the healing process under the contact load of 2 N

when dropped from 10 N, required about 470 laps sliding over the cast iron surface. The finally galling area at the equilibrium is between the saturated galling area of 2 and 10 N, which indicates some of the aluminium lumps formed under high contact load may survive. Note that the particular healing distance is only valid for this specific test. The healing distance should be affected by the contact load before and after change, while a general mechanism was expected to a similar condition.

The change of contact load will lead to a rapid change of wear rate ($\dot{\rho}_s$), which will simultaneously rise or reduce the ejection rate ($\dot{\rho}_e$) and further influence the instantaneous volume of the transfer layer ($\dot{\rho}_i$). At the load increase condition, in stage 1 under 2 N, $\dot{\rho}_i$ gradually increases towards the equilibrium with the converging of $\dot{\rho}_s$ and $\dot{\rho}_e$ to a constant at a relatively low rate. When the load of 10 N is applied, this running-in process is replaced by a faster process of converging where $\dot{\rho}_s$ jumps to a high value due to the rapid wear rate increase and is followed by an increase of $\dot{\rho}_e$. When $\dot{\rho}_s = \dot{\rho}_e$, the equilibrium ($\dot{\rho}_i = 0$) occurs. At the load decrease condition, in stage 1 under 10 N, $\dot{\rho}_s$ begins at a high level and thus generates an initial steep increase of $\dot{\rho}_e$. After load decreases to 2 N, $\dot{\rho}_s$ and $\dot{\rho}_e$ are reduced to a low rate generated at 2 N and finally reach the equilibrium slowly.

Based on the results and discussions, three conclusions of the interaction between COF, wear and transfer layer in the running-in state under load changing condition can be drawn: 1) increase of contact load will lead to an increase of the source rate, the formation rate of transfer layer and the saturated galling area and vice versa. 2) If the decrease of the saturated galling area caused by the decrease of the corresponding contact load was lower than the already formed galling area, the galling area will decrease by self-healing process. Otherwise, the galling area will continue increase. 3) COF will keep increasing under any contact load until a saturated value; the increase rate depends on the contact load.

5.6 Interactive friction model

Based on the interactive friction model developed in chapter 4, the interaction of friction, wear and transfer through a transfer layer formation progression in the aluminium-cast iron tribo-system can be modelled. The equations are discussed under a step change of contact load condition and the evolutions of COF and galling area under load changing conditions can be modelled.

The definition equations shown in chapter 4 can still be used under the contact load change condition. The origins of the coefficient of friction can be classified as i) aluminium-cast iron contact and ii) aluminium-aluminium contact. After the tribo-system reaches the equilibrium, the stable coefficient of friction (μ_{Al-Al}) is generated by a saturated galling interface between solid aluminium and aluminium third body, which is independent of contact pressure but galling area as shown in experiments. Thus, the variation of coefficient of friction can be expressed by Eq. 4.2. The variation of galling area, $f(t)$, is the main consequence of the accumulation of third body at the interface, which can be presented by the variation of the density of third body. By definition and derivation, the relation between galling area and density of entrapped third body is still valid under the load change condition and can be written as Eq. 4.7.

In the contact load change condition, the entrapped third body density $\rho_i(t)$ is also determined by the accumulating history of third body. The accumulating rate of third body at time t is determined by comparing the equality between the entrapped third body density at time t ($\rho_i(t)$) with the saturated density under the contact load at time t ($\rho_i(t_{sat})$). The possible variation rate of entrapped third body, $\dot{\rho}_i(t)$, can be separated into 3 cases: $\rho_i(t_{sat}) > \rho_i(t)$, $\rho_i(t_{sat}) = \rho_i(t)$ and $\rho_i(t_{sat}) < \rho_i(t)$. For $\rho_i(t_{sat}) > \rho_i(t)$, the tribo-system is unsaturated. The potential amount of entrapped third body is larger than the present amount and the tribo-system will accumulate more third body at the contact. When $\rho_i(t_{sat}) = \rho_i(t)$, the tribo-system reaches the steady state and $\dot{\rho}_i(t) = 0$. In the case of $\rho_i(t_{sat}) < \rho_i(t)$, the saturated entrapped third body is smaller than the present amount of entrapped third body due to contact pressure reduction. The tribo-system is under an ‘oversaturation’ state and

the entrapped third body will be ejected from the interface with a negative rate of $\dot{\rho}_i(t)$ until a new equilibrium is built. This comparing is conducted by calculating Eq. 5.1 (Appendix. C). Integrating this equation, the density of entrapped third body ($\rho_i(t)$) can be calculated.

$$\dot{\rho}_i(t) = \dot{\rho}_i(t) \left(\frac{\rho_i(t_{sat}) - \rho_i(t)}{1 - \bar{\rho}_i(t)} \right) \quad (5.1)$$

The calculation processes of $\dot{\rho}_s(t)$, $\dot{\rho}_e(t)$ and $\rho_i(t_{sat})$ are shown in Eq. 4.12 to 4.16. The parameter for the aluminium-cast iron tribo-system in the present study are the same and shown in Table 4.2.

The term, $\dot{\rho}_s(t_{sat})$ can be modelled by building the relation between $\dot{V}_s(t_{sat})$ and $\dot{\rho}_s(t_{sat})$. As shown in Appendix D, $\dot{V}_s(t_{sat})$ can be expressed by the instantaneous form of the Holm-Archard equation. Instituting the instantaneous form of Holm-Archard equation in the definition of ρ_s , $\dot{\rho}_s(t_{sat})$ can be written as Eq. 5.2. By applying the function of $\dot{P}(t)$ (Appendix G), this equation can be expressed as Eq. 5.3.

$$\dot{\rho}_s(t_{sat}) = B[P(t)v(t) + \dot{P}(t)x(t)] \quad (5.2)$$

$$\dot{\rho}_s(t_{sat}) = BP(t)v(t) \quad (5.3)$$

The formation rate of the transfer layer directly affects the evolutions of COF and galling area. Rapid changes of COF and galling area evolutions are modelled with load change under the contact load increase (Fig. 5.13A) and decrease (Fig. 5.13B) conditions. In Fig. 5.13A, the modelling result shows that the contact load increases from 2 to 10 N causing the instantaneous rapid increase of the evolutions of COF and galling area, which reaches the steady state earlier than under a constant load of 2 N. In Fig. 5.13B, the modelling result shows a decrease of the rate increase of COF evolution and a drop of galling area caused by a decrease of contact load from 10 to 2 N. These modelling results are compared with results modelled under constant loads, between which the effects of load change are clearly presented. Experimental results are presented in scatter charts with standard deviations, with a good agreement being observed.

To study the effect of contact load change, a series of modelling conditions were designed in Table 5.3. Two stages of load with 9 conditions were set: in stage 1, the

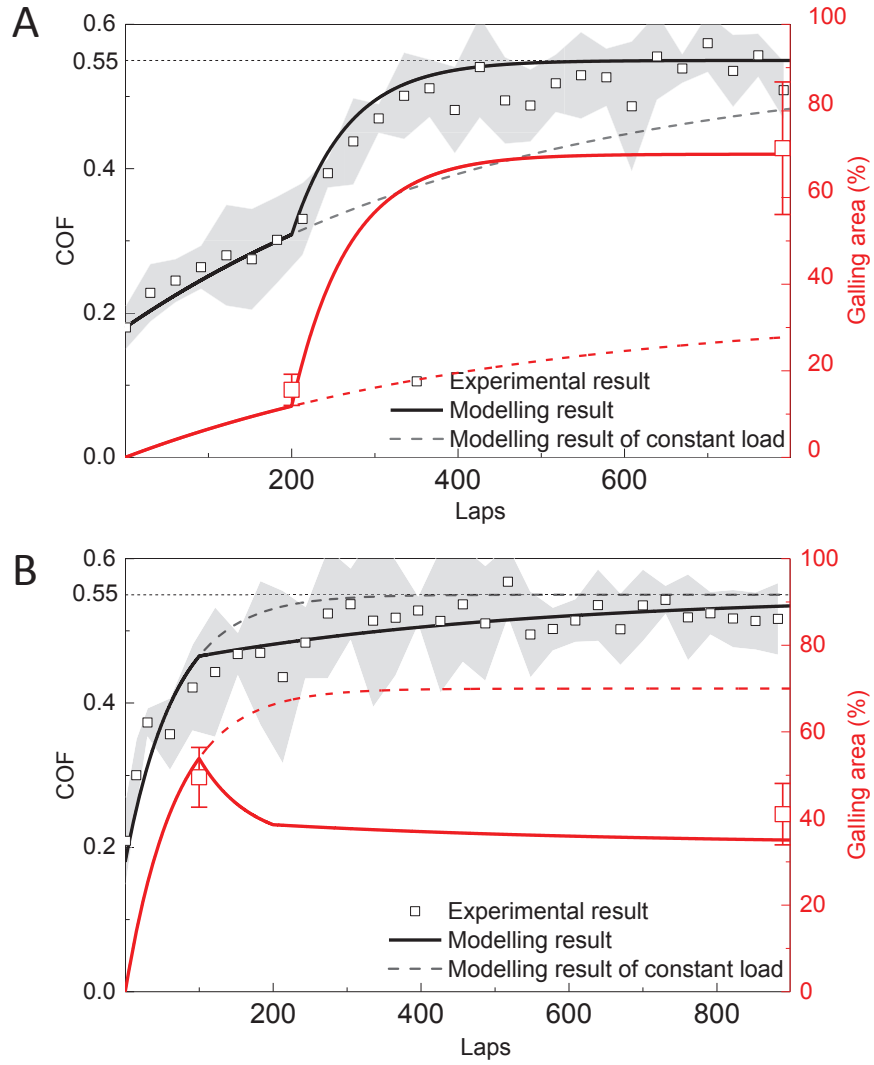


Figure 5.13: Modelling result of COF and galling are evolutions under (A) load increase condition and (B) load decrease condition comparing to experimental results.

load gradually decreased from 10 N to 2 N with a sliding distance of 200 laps; in stage 2, the load increased from 2 N to 10 N with a sliding distance of 600 laps. The modelling results of COF and galling area are shown in Fig. 5.14. In Fig. 5.14A, the variation of the rate increase of COF evolutions caused by contact load change before and after load changing can be observed, where high contact load will lead an increased rate of COF and vice versa. Finally, with continuous sliding, the COF evolutions converge to the saturated COF of 0.55 at the steady state. Fig. 5.14B presents the load changing effect on the evolution of the galling area. According to galling area evolution, not only the increase rate but also the saturated galling area are affected by load change. In test 1-4, due to the high contact load at stage 1, the galling areas generated at stage 1 are already higher than the saturated galling areas of the contact load of stage 2, which leads to self-healing processes in stage 2. From tests 6-9, as the saturated galling areas of stage 2 are not reached by stage 1, the final galling areas were dependent on the saturated galling area of the contact load under stage 2. Test 5 with a constant contact load of 6 N though two stages separates these two groups. Clear changes of increase rate of COF and galling area before and after load change can be observed with contact load of stage 2 gradually increasing from lower to higher than stage 1, where a higher increase rate is obtained.

5.7 Conclusions

In this chapter, experiments with a step change of contact load were conducted during the dry sliding between an aluminium alloy and cast iron, in which contact pair galling widely exist. It has been found that the contact load change led to a rapid change of the formation rate of the aluminium transfer layer and, furthermore, interactively changed the evolutions of COF. It was also found that the pressure and history dependency which was discovered under constant load condition were applicable at load changing conditions. That is, in the running-in state, the formation rate and the saturated amount of the transfer layer are pressure dependent, where a high contact pressure will shorten the transition period as well

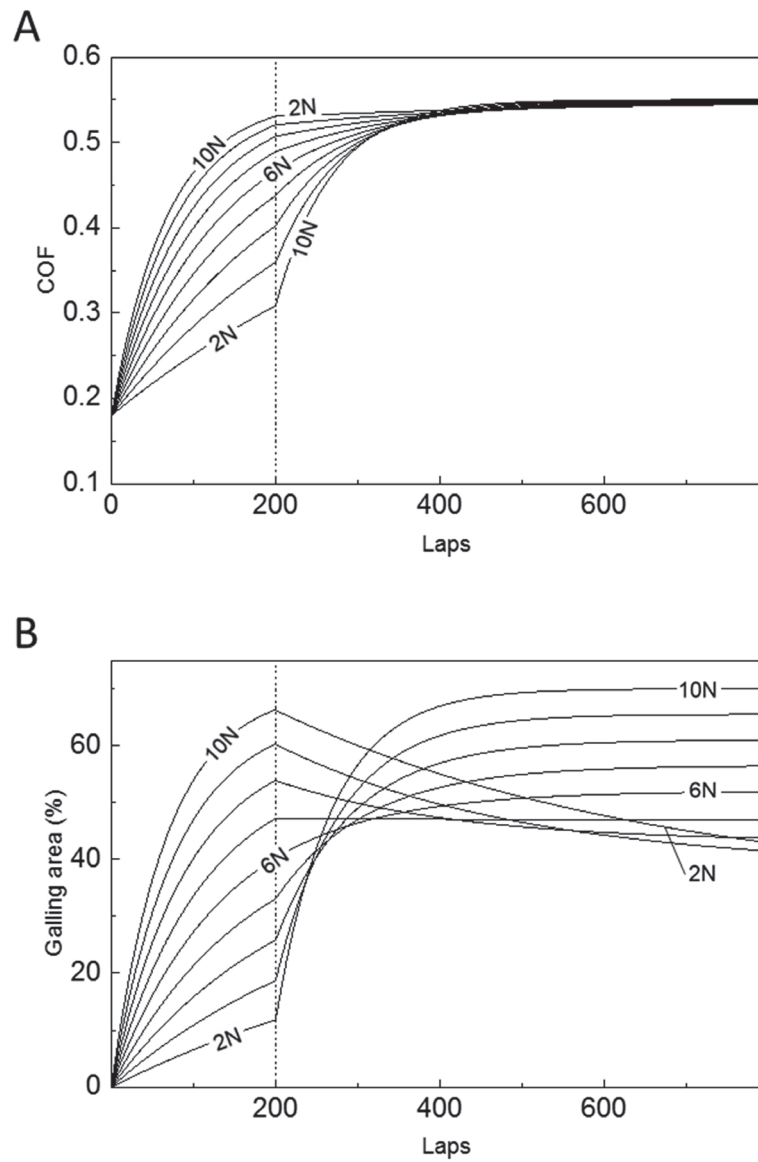


Figure 5.14: Modelling result of evolutions of (A) COF and (C) galling area under test condition of table 4.

Table 5.3: Modelling conditions of change load

Test	Stage 1		Stage 2	
No.				
	load	Sliding dis-	load	Sliding dis-
	(N)	tance	(N)	tance
1	10	200 laps	2	600 laps
2	9	200 laps	3	600 laps
3	8	200 laps	4	600 laps
4	7	200 laps	5	600 laps
5	6	200 laps	6	600 laps
6	5	200 laps	7	600 laps
7	4	200 laps	8	600 laps
8	3	200 laps	9	600 laps
9	2	200 laps	10	600 laps

as increase the saturated transfer amount at the steady state. In addition, a new phenomenon was found, where the formed transfer layer may partly survive after the decrease of contact load and affect the final galling area. This phenomenon led to a period of self-healing of the morphology where a series of secondary transfer mechanisms were triggered and a mixed transfer layer of aluminium transfer lumps and fine equiaxed particles were built, which resulted in interactive oscillations of COF evolution and decrease of galling area. A mechanical-based model for load change has been described in the chapter and showed reasonable agreement when compared to the experimental results. This model could be used in prediction of tool damage caused by galling in aluminium metal forming, in which multi-cycle operations with long sliding distance and rapid load changes are common.

Chapter 6

Development of an interactive friction model for the prediction of lubricant breakdown behaviour during sliding wear

6.1 Summary

¹In this chapter, a novel interactive friction-lubricant thickness model was developed to predict the evolution of coefficient of friction and the useful life of lubricant film. The developed model was calibrated by experimental results determined from pin-on-disc tests. For these experiments, a grease lubricant was applied on a Tungsten Carbide ball which slides against a disc made from AA6082 Aluminium alloy. In the pin-on-disc tests, the lubricant film thickness decreased with time during single path sliding leading to a rapid increase in the coefficient of friction. The breakdown of lubricant was divided into three stages, namely, the Stage I low and stable coefficient of friction region, Stage II region in which the coefficient of friction sees a rapid rise,

¹This chapter is based on paper work ‘Y. Hu, L. Wang, D. Politis, M. Masen, *Development of an interactive friction model for the prediction of lubricant breakdown behaviour during sliding wear*, Tribology International, 110 (2017) 370-377.’

and Stage III in which the coefficient of friction reaches a plateau with a value similar to that of dry sliding. In order to characterise the evolution of coefficient of friction throughout these stages, a novel interactive friction model was developed combining the effects of sliding distance, sliding speed, contact pressure and initial lubricant amount on the evolution of the coefficient of friction. This interactive friction model can be applied to situations involving lubricant breakdown in a dynamic environment such as the metal forming industry, where the use of traditional constant coefficient of friction values present limits in predictive accuracy.

6.2 Introduction

The study of lubrication breakdown in a lubricated contact has received some attention amongst metal forming researchers due to the growing demand for accurate FE simulation of boundary conditions. In most forming cases, a moderate quantity of lubricant applied between the workpiece and the tool can provide a separation barrier during the metal forming process (Foster et al., 2008, Wilson, 1978). This amount of lubricant typically involves the assumption of ideal fully flooded lubrication conditions with low coefficient of friction and little wear due to moving objects. However, in many situations, it is not always possible to maintain ideal fully flooded lubrication conditions and there may be considerable levels of non-hydrodynamic lubrication, e.g. boundary lubrication, that result in galling or wear on tooling and the product. This is especially essential for stamping processes, where the transportation of lubricant is uneven due to the non-uniform distribution of relative sliding distance, strain and contact pressure at the workpiece-tooling interfaces. Moreover, in many cases, lubricant is squeezed out towards regions with lower pressure and side leakage occurs, which will cause further loss of lubricant from the contact and lead to lubricant film breakdown. Therefore, an adequate quantity of lubricant applied prior to the forming operation does not guarantee that lubrication will be effective at all locations or at all stages of a forming operation (Bowden and Tabor, 2001, Wilson, 1978).

In recent years, FE simulation has been widely used by metal forming engineers to analyse and optimise forming processes. The coefficient of friction, as one of the key inputs for an FE model, is normally assigned as a constant value (Ma et al., 2015, Yang and Lo, 2004, Hsu and Wilson, 1994, Wang et al., 2016). However, in practice, lubricant film breakdown might dramatically increase the coefficient of friction, due to the direct contact between the workpiece and dies (Zhang and Ou, 2016, Clarke et al., 2016). Classic models that do not take into account changes in the lubrication consistency, may cause inaccuracies in the FE simulation results. Therefore, understanding and modelling the lubricant breakdown behaviour, and the interaction with the evolution of the coefficient of friction and lubricant service life are of great practical importance. Previously, the phenomenon of lubricant film breakdown has been studied in many fields, including mechanical transmissions (Castro and Seabra, 1998, Bhushan, 2013, Fang et al., 1993), internal combustion engines (Bowden and Tabor, 2001, Tomanik, 2008), bearings (Kingsbury, 1978, Sander et al., 2016, Shi et al., 2016) and metal processing (Bhushan, 2013, Holmberg and Matthews, 1994, Wen and Huang, 2012, Wilson and Carpenter, 1973). The influencing factors of lubricant film breakdown have been identified and quantitatively studied. They can be classified into two groups: 1) the operation parameters, including geometry of the contact, sliding speed, load and lubricant amount; and 2) the interface characteristics, including lubricant properties, surface roughness, surface plastic deformation, boundary lubrication and squeezing/side leakage. Bowden and Tabor (Bowden and Tabor, 2001) reviewed the effects of speed, lubricant viscosity and temperature on the lubricant breakdown phenomenon in various industrial applications. The sliding speed effects on the mechanism of breakdown were also studied by Begelinger and De Gee (Begelinger and De Gee, 1982), in which friction-time diagrams as a function of sliding speed were presented and two important conclusions were drawn: 1) the breakdown time is viscosity dependent and 2) in the low speed region (velocity < 2 m/s), the load-carrying capacity of the lubricant film increases with increasing sliding speed. In Kingsbury's work (Kingsbury, 1978), the effect of increasing lubricant quantity, as extending the running life time before lubricant film breakdown, was

observed in ball bearing tests. This effect is also studied by Groche et al. (Groche et al., 2013b, Groche et al., 2013a) and similar conclusions were drawn in metal forming.

As a fundamental study of mechanisms of lubricant film breakdown with the effect from operation parameters, the present chapter is concerned with the lubrication of bodies in normal point contact. The aim of this chapter is to develop an interactive friction model to characterize the breakdown of the lubricant during sliding point contact and its interaction with the evolution of coefficient of friction. The parameters of lubricant film diminution and breakdown as a function of time and sliding distance due to lubricant transport, sliding speed, load, and the quantity of entrapped lubricant were studied experimentally at room temperature. Based on these results, the interactive relationship between the evolution of the coefficient of friction and the reduction of the lubricant film was modelled, enabling the coefficient of friction and lubricant breakdown to be predicted through a novel friction/film thickness interactive model.

6.3 Experimental set-up and test programme

Aluminium sheet is studied due to its industrial potential and also lubricated difficulty, which is easy to adhere and be worn (Fan and Wang, 2015). For the production of the disc samples, AA6082 sheet at T6 condition with a thickness of 1 mm was utilised. The mechanical properties of the tested metal are: Young's modulus 72 GPa, Poisson's ratio 0.33, and Vickers hardness 100 HV. The test piece material was cut into squares with dimensions of 50 mm by 50 mm. All samples were ground by silicon carbide emery paper to obtain uniform surface roughness. The arithmetic average surface roughness, R_a , was $0.50 (\pm 0.30) \mu\text{m}$, which was measured through a 3D white light interferometry surface profilometer (Veeco Wyko NT9100). The ball material used as the counterpart in the friction tests was Tungsten Carbide WC-6% Co ball (Young's modulus 630 GPa, Poisson's ratio 0.23, Vickers hardness 1780HV), 6 mm in diameter, due to its good abrasion resistance and low adhesion

with aluminium as a potential coating material for aluminium forming (Dong et al., 2015). To prevent contamination, both ball and disc were cleaned with acetone and dried in air before the application of lubricant. The lubricant used for the tests was a lubrication grease, OMEGA 35, made from polyethylene glycol, silicon dioxide and graphite. This lubricant features adequate performance in a high temperature environment application (up to 700°C). Although the viscosity of OMEGA 35 decreases with temperature increase, its proportion of synthetic carrier and micronized megalite solid lubricant gives it good ‘spreadability’ and enhanced lubricating qualities. The key physical parameters of OMEGA 35 are shown in Table 6.1. Two lubricant

Table 6.1: Lubricant data of OMEGA 35

Kinematic Viscosity (cSt)		Specific gravity(dimensionless)		Dropping point($^{\circ}\text{C}$)
40°C	35	15°C	1.33	260
100°C	6			

application methods were used in the friction tests: 1) a precisely controlled quantity of lubricant was applied on the ball to simulate the non-hydrodynamic lubrication (insufficient lubricant) condition and 2) lubricant was evenly applied to the disc to simulate a fully flooded lubrication condition. For condition 1), the lubricant was applied by a dedicated rig designed and manufactured by the authors’ group with micro volume lubricant reservoirs of $0.16\text{ }\mu\text{L}$, $0.24\text{ }\mu\text{L}$, and $0.4\text{ }\mu\text{L}$, corresponding to an average mass of lubricant applied on the ball of 4 mg, 10 mg and 14 mg, respectively. For the fully flooded lubrication condition, 100 mg of grease was applied on the disc’s surface at a thickness of $500\pm 50\text{ }\mu\text{m}$, which was measured by a plastic wet film comb (Elcometer 3238). The initial lubricated area was assumed to be the projected area of the ball.

The frictional behaviour was investigated on an Anton Paar pin-on-disc tribometer under a single direction sliding. The design, measurement and evaluation of tests were partly based on ASTM standards G99. Three sets of tests were designed, aimed at investigating the influence of lubricant transportation and film breakdown phe-

nomenon. The variables are lubricant amount, sliding speed and load. Testing conditions are shown in Table 6.2 and each condition was repeated three times. The dry sliding test and the full lubricant test (tests no. 1 and 5) were designed for comparing with the steady state coefficients of friction in non-hydrodynamic lubrication. Loads of 0.5, 2 and 5 N were used which corresponded to the mean contact pressure, calculated using Hertz contact theory as, 0.25, 0.4 and 0.55 GPa, respectively. All friction tests were conducted in an ambient environment, at a temperature of 24 °C. The wear track created was analysed by a white light interferometry profilometer (WLI) and an optical microscope to investigate the wear tracks obtained under lubricated and dry conditions to identify the dominant friction mechanism. The coefficient of friction revolution for each test was smoothed. For each condition, different tests were combined and the averages (solid lines) and standard deviations (error bars) are given.

6.4 Results and discussion

6.4.1 Lubrication, friction and wear mechanisms

In the case of insufficient lubrication, the coefficient of friction was low at the initial stage and followed by an abrupt increase of coefficient of friction indicating the breakdown of the lubricant film; finally, the coefficient of friction increases to a stable value similar to the metal/metal contact situation. The results of experiment No. 3 are analysed in Figures 6.1-6.3. The coefficient of friction evolution is shown in Fig. 6.1 and the wear track is shown in Fig. 6.2 with the surface topography shown in Fig. 6.3 after removing the wear debris and the residual lubricant. It is found that the evolution of friction can be divided into 3 stages according to the different coefficient of frictions.

In stage I, the coefficient of friction is low and stable, with an average value of approximately 0.1. No clear wear scar was observed in this stage (point 1) suggesting that the two surfaces were separated by the lubricant film. The friction in

LUBRICANT BREAKDOWN

Table 6.2: Test matrix of lubricant breakdown							
Effect	Test No.	Temp.	Speed	Load	Pressure	Lubricant	
	Unit	$^{\circ}C$	mm/s	N	GPa	Quantity mg/mm^3	Application method
Lubricant Amount	1	24	10	5.0	0.55	0.0 (dry)	
	2	24	10	5.0	0.55	0.2	on the ball
	3	24	10	5.0	0.55	0.5	on the ball
	4	24	10	5.0	0.55	0.7	on the ball
	5	24	10	5.0	0.55	4.0 (fully flooded)	on the disc
	6	24	30	5.0	0.55	0.5	on the ball
Speed	7	24	50	0.5	0.55	0.5	on the ball
Load	8	24	10	0.5	0.25	0.5	on the ball
	9	24	10	2.0	0.40	0.5	on the ball

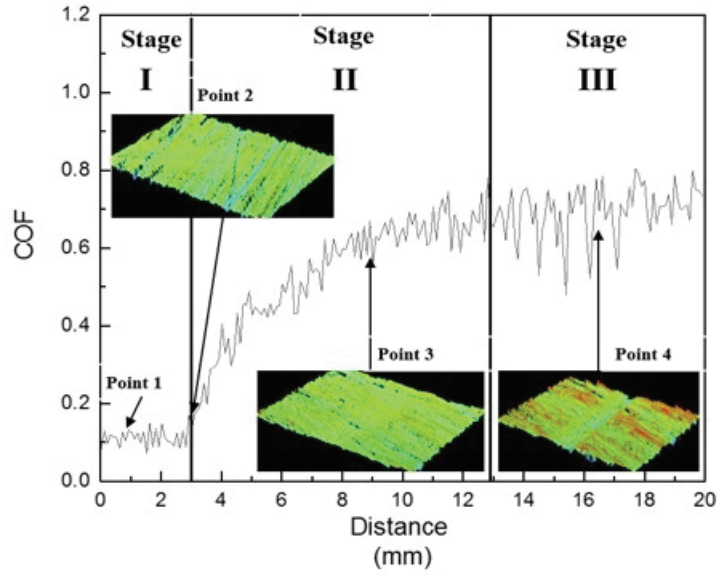


Figure 6.1: Evolution of the coefficient of friction (10 mm/s, 5N/0.55GPa and 0.2 mg/mm²) with the local wear track images obtained using WLI at different sliding distances. The image shows 4 points chosen from different stages in the friction curve. The wear track becomes more obvious with increasing sliding distance and COF.

stage I may be primarily generated by the internal fluid shear stress of the lubricant at the interface (Wen and Huang, 2012) and can be partly generated under mixed lubrication condition. During sliding, the thickness of the lubricant film gradually decreases due to lubricant transfer from the ball to the aluminium disc and lubrication mode changes from fully flooded lubrication to mixed lubrication regime, which is defined as a transition state between fully flooded lubrication and boundary lubrication and in which two lubrication mechanisms may be functioning (Bhushan, 2013). In this mode, the coefficient of friction can be regarded as a constant (Wen and Huang, 2012, Bowden and Tabor, 2001).

In stage II, the coefficient of friction starts rising rapidly from about 0.1 and gradually slows down at a value of 0.65. In this regime, the coefficient of friction is highly variable and unstable because the friction stems from fracture phenomena at the surface (Bowden and Tabor, 2001). At the beginning of stage II, the thickness of lubricant decreases to the height of the peaks on the aluminium surface and direct

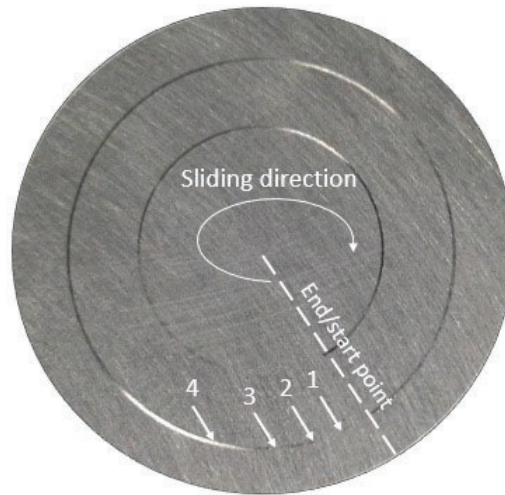


Figure 6.2: Wear tracks on the disc within one lap of sliding wear, with the locations of points 1-4 marked on the disc.

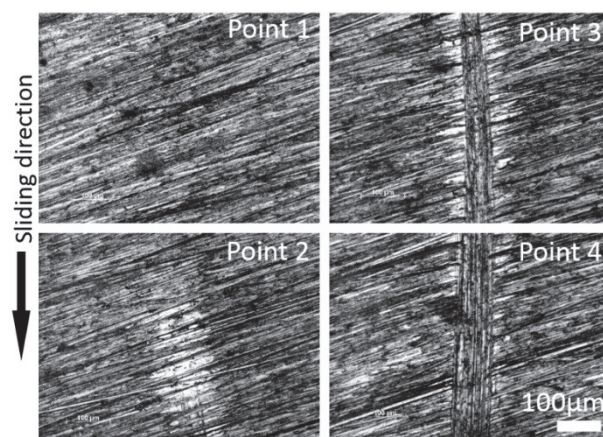


Figure 6.3: Wear track topography at different sliding distances.

contact under boundary condition may occurs. In that case, the normal force is supported by both the residual lubricant trapped in the contact and the surface asperities. A wear track develops on the surface, as shown in Fig. 6.3 point 2 and 3, initially the wear track is almost invisible and becomes wider and deeper with increasing sliding distance. The friction force of this mixed lubrication is supposed to consist of two components: the friction force generated from interacting asperities and the shearing of the remaining lubricant (Gelinck and Schipper, 2000). The interacting asperities friction was mainly caused by deformation and explained as the asperities deform and fracture into wear particles, which are entrapped and generate ploughing tracks as the ball penetrated into and moved along the wear track. It was found that the adhesive friction between the WC-Co 6% ball and the aluminium alloy disc did not play an important role in the present research since the material transfer between them was hardly observed by a microscope in the tested sliding distance. Both stage II and III were dominated by ploughing friction due to the large hardness difference between WC and aluminium: the disc surface was ploughed by the hard asperities on the WC ball as well as the wear debris. The width of the wear track on the aluminium surface increases in this stage (Fig. 6.3 point 2, 3, 4) because the effects of residual lubricant become less significant as the sliding distance increases.

In stage III, the coefficient of friction reaches a plateau with an average value of 0.65, which was close to that obtained under dry sliding. The fluctuation of the coefficient of friction was severe. In this stage, the lubricant is almost completely removed from the contact interface and hence ploughing friction played an important role in the overall friction force. With the severe ploughing damage on the surface (Fig. 6.3 point 4), many large sized wear particles were generated and entrapped in the wear track during sliding wear. It was expected that the quantity and size of the particles generated on the track reached a dynamic balance, which may indicate that the quantity of entrapped particles was equal to that being ejected from the wear track, leading to a relative stable third body condition and a stable coefficient of friction (Ma et al., 2015).

6.4.2 Evolution of coefficient of friction during the sliding wear between lubricated contact pairs

The effects of applied lubricant quantity, load and sliding speed on the lubricant breakdown distance and coefficient of friction were studied experimentally. The experimental data for different quantities of applied lubricant are shown in Fig. 6.4(A). As can be seen, it was found that the increase in lubricant quantity (from 0.2 mg/mm^2 to 0.7 mg/mm^2) leads to an increased sliding distance at which low friction is experienced. As expected, in the dry sliding and fully lubricated tests, there was no coefficient of friction transition observed. It is therefore indicated that the length of the stage I region was influenced by the initial lubricant quantity that was entrapped at the interface, that is, the reduced lubrication quantity diminished the operating time before lubricant breakdown. This conclusion agrees with previous researchers' work (Kingsbury, 1978, Wilson and Carpenter, 1973, Wilson, 1978).

An examination of the effect of load, as shown in Fig. 6.4(B), demonstrates that the distance of lubricated stage I becomes shorter as the pressure increases from 0.25 to 0.55 GPa. This breakdown phenomenon agrees with the trend predicted from the generalised Stribeck curve and the Sommerfeld number, $\eta v/P$ (where η is the viscosity of lubricant, v is the sliding velocity and P is the load) (Bowden and Tabor, 2001, Schipper, 1988), that is, the increasing load will result in thinner lubricant film and thus premature breakdown. According to the International Research Group (IRG) transition diagram (Salomon, 1976, Schipper and De Gee, 1995, Begelinger and De Gee, 1982), a similar conclusion can be drawn: an increase of the load will lead to a decrease of the load-carrying capability of a lubricant and cause earlier onset of the primary transition in film failure.

As shown in Fig. 6.4(C), the effect of sliding speed (10mm/s, 30mm/s and 50mm/s) on the evolution of coefficient of friction was investigated. An elongation of the breakdown distance with increasing sliding speed was observed in the film breakdown test: the longest low-friction regime is obtained from the test of 50 mm/s. This may be due to the fact that an increased sliding speed will increase the film thickness (Wen and Huang, 2012, Holmberg and Matthews, 1994, Chiu, 1974,

Wedeven et al., 1971) as the contact fully immersed in lubricant. This analysis of film breakdown mechanism is probably only virtue of the fact that the speed of sliding is lower than a threshold value due to the temperature influence. Begelinger and De Gee (1982) found that, at a low speed ($v < 2$ m/s), the increase of sliding speed will increase the load-carrying capability and thus extend the film breakdown distance; however, a contrasting phenomenon, showing a load-carrying capability decrease with higher speed ($v > 2$ m/s). A similar phenomenon that the increase of sliding speed will decrease wear of the contact was recorded by Bowden and Tabor (2001) in a piston/cylinder reciprocation test. It was explained as the amount of hydrodynamic lubrication is greatly increased with increasing speed. The contrast experimental observations may be caused by the temperature increase during high speed sliding and lubricant failure.

6.5 Interactive friction modelling of lubricant behaviour

Based on Bowden and Tabor (2001) and later Azushima's models (Azushima, 1978), the overall coefficient of friction, μ , generated between the WC-6% Co ball and the lubricated aluminium disc stems from two mechanisms (Eq. 6.1), namely the coefficient of friction at the fully flooded lubrication condition, μ_L , and the dry coefficient of friction, μ_d . In the present work, the contribution of these two components are decided by a lubricated area ratio, β , which varies from 0 (the initial full lubricated state) to about 1 (dry sliding state).

$$\mu = (1 - \beta)\mu_L + \beta\mu_d \quad (6.1)$$

As discussed, the contact evolved from the initial fully flooded lubrication to eventual dry sliding during the sliding process. This process could be separated into three components: 1) the initial full lubrication state with a low coefficient of friction, 2) the transition to mixed lubrication condition or eventually boundary lubrication with the increasing of coefficient of friction and 3) the dry sliding condition, in which

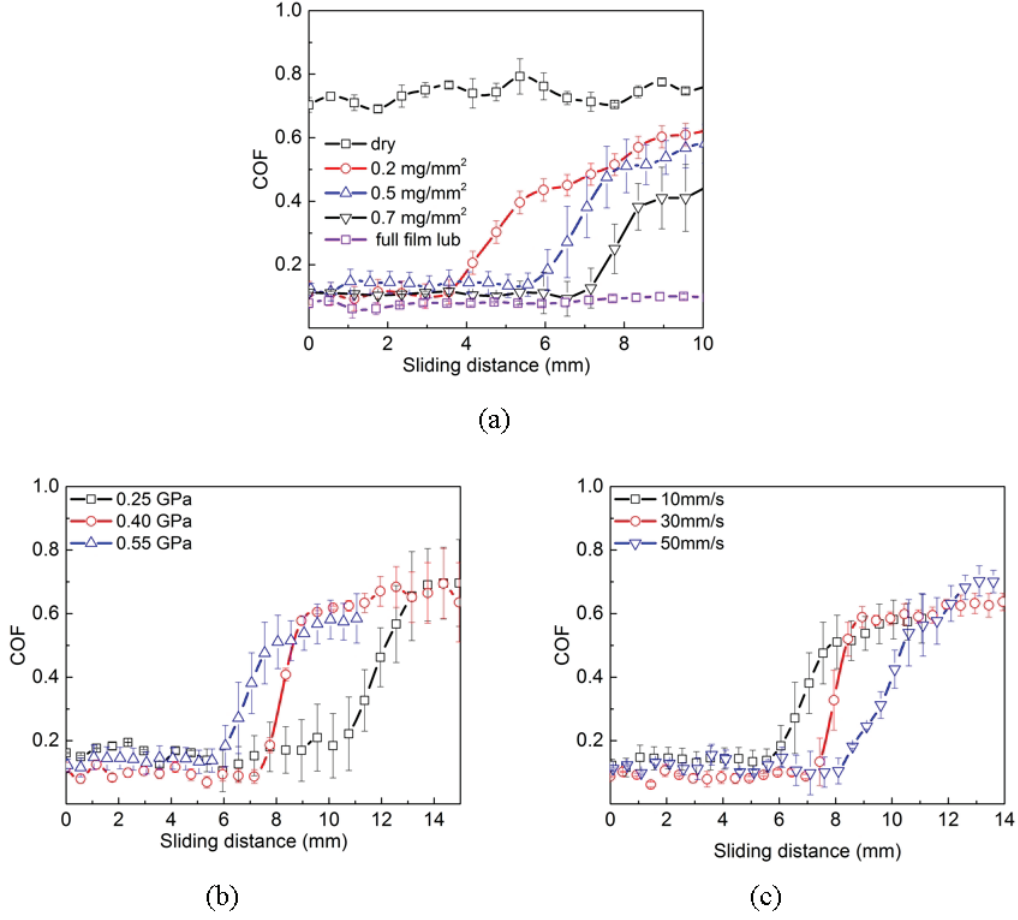


Figure 6.4: Coefficient of friction evaluations showing the effects of: (A) quantity of lubricant, (B) contact pressure (C) sliding speed.

the coefficient of friction stabilized at a high value. Therefore, it may be reasonable to assume constant values of coefficient of friction, i.e. $\mu_L=0.13$ and $\mu_d=0.6$, for the initial full lubricated and final dry sliding states, whose values can be determined from the friction tests under fully flooded lubrication and dry sliding conditions. It is claimed that the lubrication states, determined by the film thickness, give different coefficients of friction and the decrease of lubricant film thickness gives more chance of solid contact and thus increase the friction (Wen and Huang, 2012, Bhushan, 2013). This relationship, between the film thickness, $h(t)$ and the lubricated area ratio, β , is modelled by Eq. 6.2, where $h(t)$ is the instantaneous film thickness of the lubricant to represent the friction transferring from a steady stage to a breakdown stage, and λ_3 and λ_4 are breakdown distance parameters for the lubricant. This

equation is an empirical equation and was developed considering the physical mechanisms of the lubricant film thickness and the relative variation of lubricated area ratio in the transition from mixed to boundary lubrication condition. The height distribution of the contact surface morphology was assumed as a normal distribution and the steady state was considered as the aluminium surface can mechanically support the counterpart under a boundary condition. To improve the numerical integration efficiency, the equations have been simplified considerably.

$$\beta = \exp(-(\lambda_3 h(t))^{\lambda_4}) \quad (6.2)$$

In the experiment, the thickness of the lubricant film, $h(t)$, is primarily influenced by the transportation of lubricant during sliding, entrapped lubricant quantity, lubricant squeezed out, sliding speed, lubricant properties and contact geometry. By incorporating the operational parameters, and assuming a constant volume case where the volume of initial entrapped lubricant is equal to the volume smeared on the track, a lubricant film thickness model with variable sliding speed v , Hertz contact radius r , initial film thickness $h(0)$, and instantaneous film thickness $h(t)$ can be developed and is presented in Eq. 6.3.

$$Ah(0) = Ah(t) + \sqrt{A} \int_0^t f(P, \eta) h(t) v^{k_1} dt \quad (6.3)$$

where A is the area of the Hertzian circle and the boundary condition is found when $t = 0$, $h(t) = h(0)$. As the amount of entrapped lubricant is determined by the thickness and the lubricant properties, and is nearly independent of the size of the deformed region (Archard and Kirk, 1961), a constant Hertzian contact area is assumed. $Ah(0)$ is the initial volume of the entrapped lubricant that stored between ball and aluminium disc before sliding. $Ah(t)$ is the entrapped volume at time t and $\sqrt{A} \int_0^t f(P, \eta) h(t) v^{k_1} dt$ is the volume of lubricant transported onto the wear track during sliding (Fig. 6.5). In reality, considerable volume of lubricant at the contact may be squeezed out at the initial contact. The lubricant covered on the wear track may largely come from stored lubricant between the side of ball and aluminium sheet, which may also a pressure-dependent value.

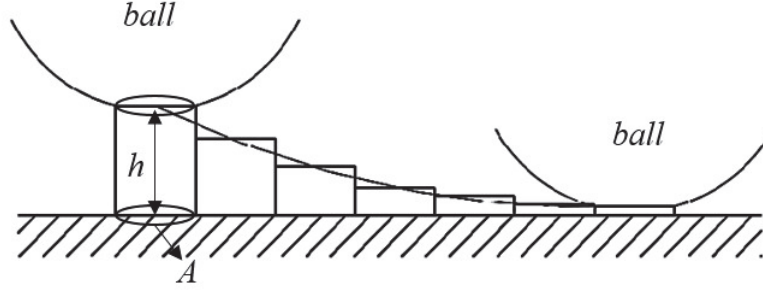


Figure 6.5: Schematic model of lubricant transportation with an initial thickness of h and area of A .

The volume of lubricant smeared on the track during sliding is calibrated by the function, $f(P, \eta)$, where P is the mean contact pressure calculated by the Hertz contact equations (Johnson and Johnson, 1987); and η is the viscosity of the lubricant, which is assumed as a constant at room temperature. Thus $f(P, \eta)$ can be written as $f(P, \eta) = bP^{\kappa_2}$, where b is a constant. Therefore, Eq. 6.3 incorporates additional operating parameters compared to equations used in previous studies (Dowson and Whomes, 1966, Wedeven et al., 1971), who's efforts focused on the effect of lubricant squeezed out of the contact surface only as a function of pressure and load for a given lubricant. This equation can be transferred into a first order linear differential equation (Eq. 6.4) (the area A is integrated into constant c).

$$\dot{h}(t) = \frac{dh}{dt} = -h \exp(cP^{\kappa_1}v^{\kappa_2}t) \quad (6.4)$$

In Eq. 6.4, c , κ_1 and κ_2 are breakdown time parameters that can be determined from the friction evolution curve; c is a friction constant; P is the contact pressure and v is the sliding speed. This instantaneous film thickness equation can be integrated into the form of Eq. 6.5 by solving the boundary conditions: $h(t) = h(0)$ at time $t = 0$.

$$h(t) = h(0) \exp(-cP^{\kappa_1}v^{\kappa_2}t) \quad (6.5)$$

The right side of Eq. 6.5 may be divided into two components: $h(0)$ and $\exp(-cP^{\kappa_1}v^{\kappa_2}t)$, which indicate the initial film thickness and the film diminution rate respectively. The variables of diminution rate (P and v) are represented by Power Law, with similar form of Dowson's equations (Hamrock and Dowson, 1976, Hamrock and

Dowson, 1977a, Hamrock and Dowson, 1977b). The breakdown time was calculated using Eq. 6.4. The highest asperity peak was approximately $1\ \mu m$. Thus, when the lubricant thickness, $h(t)$, is smaller than the asperity's height, breakdown would occur (Bhushan, 2013, Wilson et al., 1995) and the parameters, c , κ_1 and κ_2 , can be determined. It has to be mentioned that this assumption is simple and relatively accurate. Some advanced studies show complex effects of surface roughness on lubricant breakdown (Krupka et al., 2015, Lundberg, 1995), which are not intended to be covered in this study. $h(0)$ is assumed to be determined only by the quantity of applied lubricant. In the present work, the initial lubricant film thickness can hardly be measured due to 1) non-transparent lubricant being used and 2) contact pair being set. Thus, the initial lubricant thickness are set based on the fitting of model and data of literatures. The initial lubricant film thickness, $h(0)=12\ \mu m$, is set to the condition of $0.5\ mg/mm^2$ lubricant applied. Then the $h(0)$ values for application volumes of $0.3\ mg/mm^2$ and $0.7\ mg/mm^2$ are calculated from this pre-setting value, which are 4.5 and $28\ \mu m$. The friction model was calibrated using the obtained experimental data. The calibration consists of two steps:

1. The breakdown time was calibrated using Eq. 6.4. The parameters, c , κ_1 and κ_2 , can be determined from the experimentally observed breakdown time.
2. With the film thickness obtained from step 1, the coefficient of friction evolution is fitted by Eqs. 6.1 and 6.2 with the parameters λ_3 and λ_4 determined from the breakdown distance.

The processes of determining these parameters and the modelling results of lubricant film thickness with respect to time are shown in Fig. 6.6. Comparisons between modelling and experimental results are shown in Fig. 6.7, indicating that the interactive friction model enables the prediction of breakdown time of lubricant film. The model parameters and constants are listed in Tab. 6.3, which were determined by using a dedicated algorithm developed in the authors' group. After the constants have been determined, the model can be used to predict the evolution of coefficient of friction under a range of conditions. The interactive friction model developed in

the present chapter is valid for the prediction of the coefficient of friction evolutions for loads between 0.5 and 5 N and sliding speed between 10 mm/s and 50 mm/s.

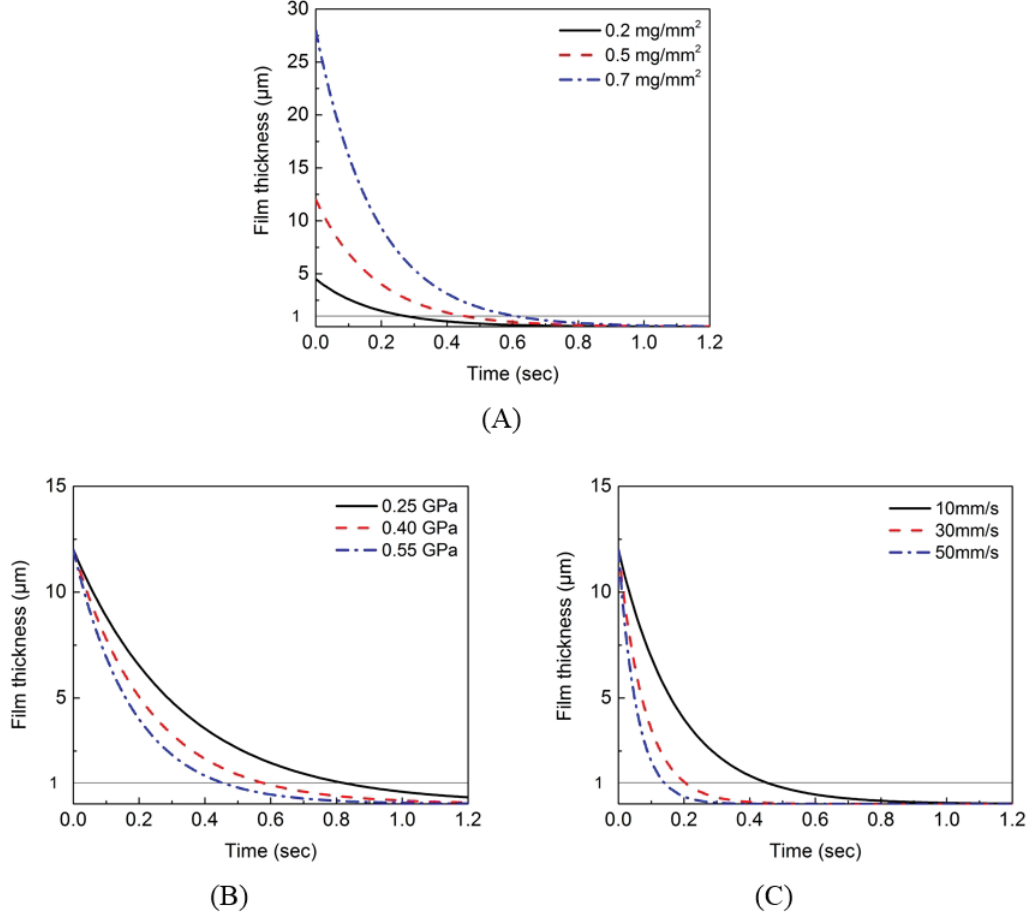


Figure 6.6: Simulation results of lubricant film thickness diminution under various (A) lubricant quantity, (B) load and (C) sliding speed.

Table 6.3: Model constant of galling test under constant load

Parameters	λ_3	λ_4	k_1	k_2	c
Values	3	2.3	0.75	0.73	1.6

Comparison with other lubricant film thickness results are discussed in this section. The Eq. 6.4 is similar to the Wilson's lubricant film thickness model (Wilson and Wang, 1984, Hector and Wilson, 1991, Hsu and Wilson, 1994, Wilson et al., 1995)

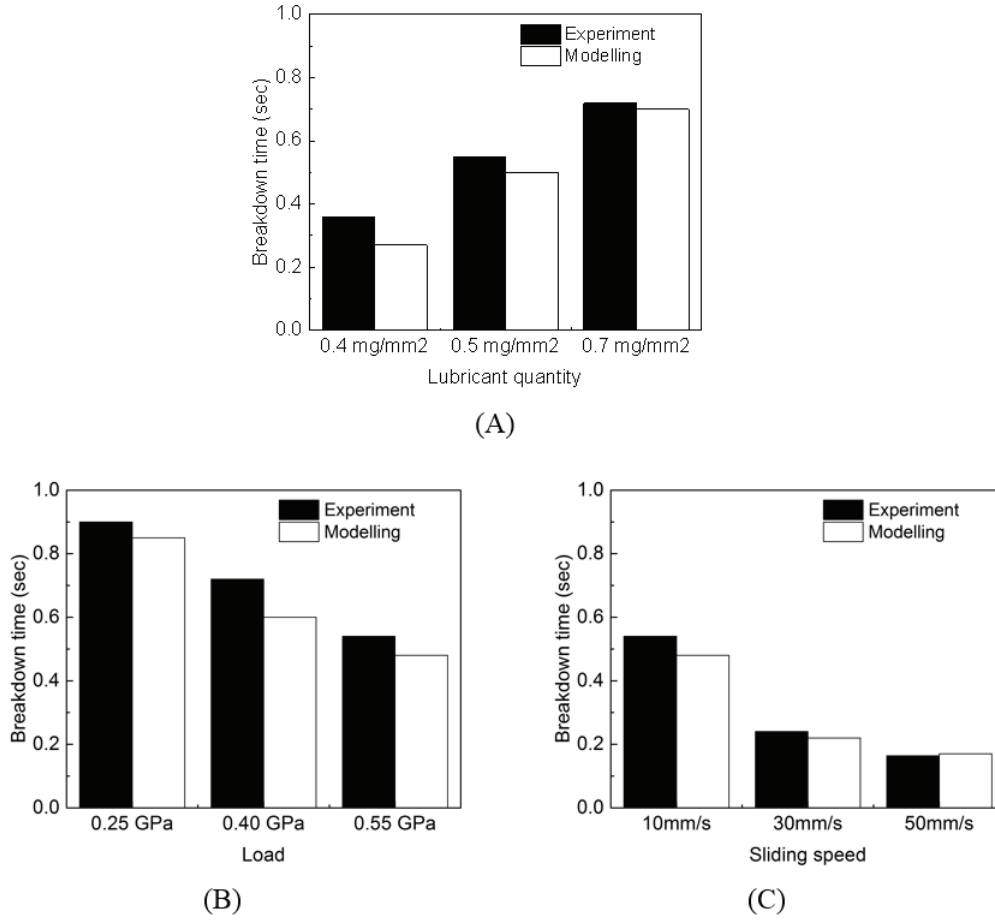


Figure 6.7: Comparison of simulation and experiment result of breakdown time under various (A) lubricant quantity, (B) load and (C) sliding speed.

in the form of ODEs. Wilson's model was developed on Reynolds equation in sheet metal forming simulation. The film thicknesses calculated from Eq. 6.4 have been compared with experimental measurement (in dome test) and various theoretical predictions shown in Fig. 6.8. The model developed in this chapter enables the description of typical evolution curves observed in previous research, i.e. an initial rapid decrease in film thickness followed by a gradual reduction in thickness. The present film thickness model shows its flexibility for modelling lubricant transportation and squeezing out phenomena. Also, this model shows the ability that the lubricant film can diminish to the height of asperity as a result of non-hydrodynamic lubrication, comparing to other results and models under the fully flooded lubrication condition and assumption that the lubricant film thicknesses finally reduce to

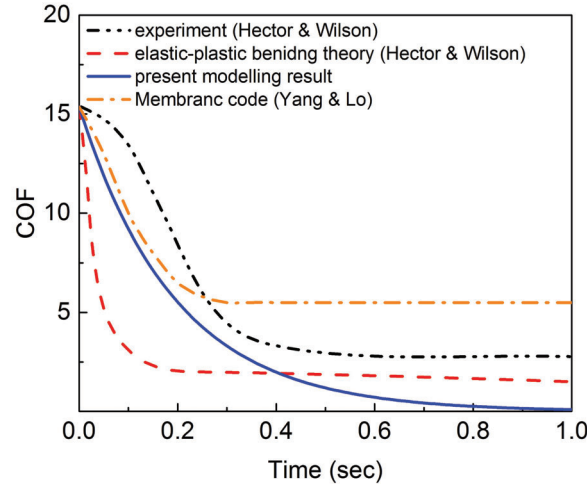
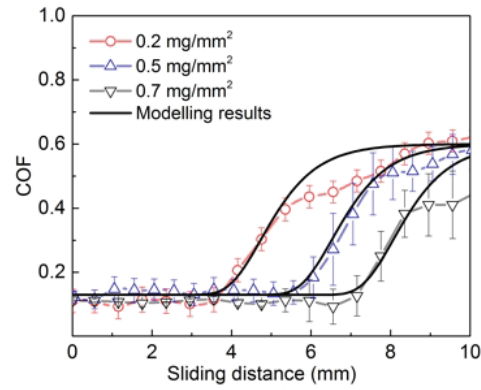
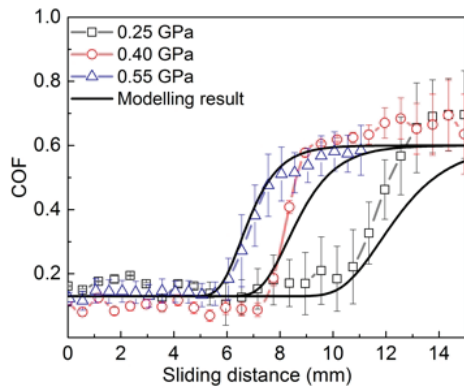


Figure 6.8: Comparison of various theoretical predictions (elastic-plastic bending theory (Hsu and Wilson, 1994); Membrane code (Yang and Lo, 2004)) with Hector and Wilson's measurements (Hector and Wilson, 1991) of film thickness.

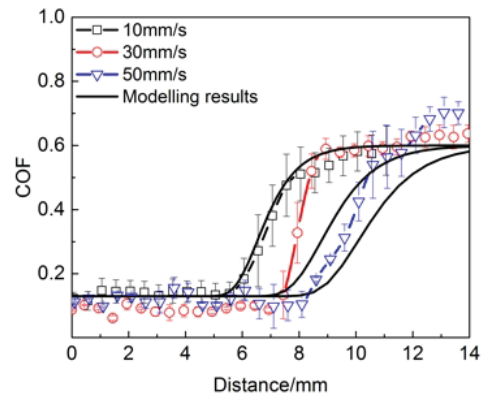
constant values as fully flooded lubrications are obtained. The fitting results between predicted and experimentally obtained coefficient of friction evolutions are shown in Fig. 6.9. Good agreements have been achieved. In addition, a three dimensional friction/lubricant regime transfer diagram has been generated, as shown in Fig. 6.10. The stage transfer and relative coefficient of friction can be predicted with certain operation parameters (lubricant amount, sliding speed and load). The breakdown distance is also determined as a function of sliding speed and load. With the increase of sliding distance (z -axis), the coefficient of friction increases from 0.13 to 0.6, corresponding to the transition from stage I to stage III. The surfaces, A and B, indicate the boundary of different stages. Furthermore, a lubrication behaviour window of the lubricant can be obtained to predict the fully flooded lubrication region in case of lubricated wear (Fig. 6.11). The setting of operation parameters under the breakdown distance surfaces are desired fully flooded lubrication and low coefficient of friction are obtained in this region. In this figure, the condition of high speed (50 mm/s), low pressure (0.25 GPa) and large lubricant amount (0.7 mg/mm^2) gives the longest breakdown distance (18 mm).



(A)



(B)



(C)

Figure 6.9: Comparison of modelling and experiment result of coefficient of friction evolution under various (A) lubricant quantity, (B) load and (C) sliding speed.

6.6 Conclusions

In the present research, an interactive friction model has been developed to characterize the lubricant film breakdown phenomena. The effects of lubricant thickness, sliding speed and contact pressure on the evolution of friction coefficient have been studied and the novel interactive friction model has enabled the prediction of friction coefficient evolution as a function of sliding distance. From the work performed in the chapter, the following conclusions can be drawn:

1. The pin-on-disc test results illustrate that the friction evolution process of a grease lubricated sliding contact comprises of three stages corresponding to different friction mechanisms, namely a stage I with fully flooded lubrication and low coefficient of friction, stage II with sharp increasing coefficient of friction, and stage III, a steady stage with a high coefficient of friction.
2. Lubricant regime transformation plays an important role in the sliding friction process. The gradual decrease of lubricant film thickness causes a transformation from fully flooded lubrication to boundary lubrication and the development of surface damage which marks the beginning of stage II.
3. The interactive friction/film thickness model developed in the present research provides an effective method to predict the film breakdown time and distance at different lubricant amount, load, and sliding speed, and thus allows for modelling of the lubricant film breakdown phenomenon that occurs in the sliding process.

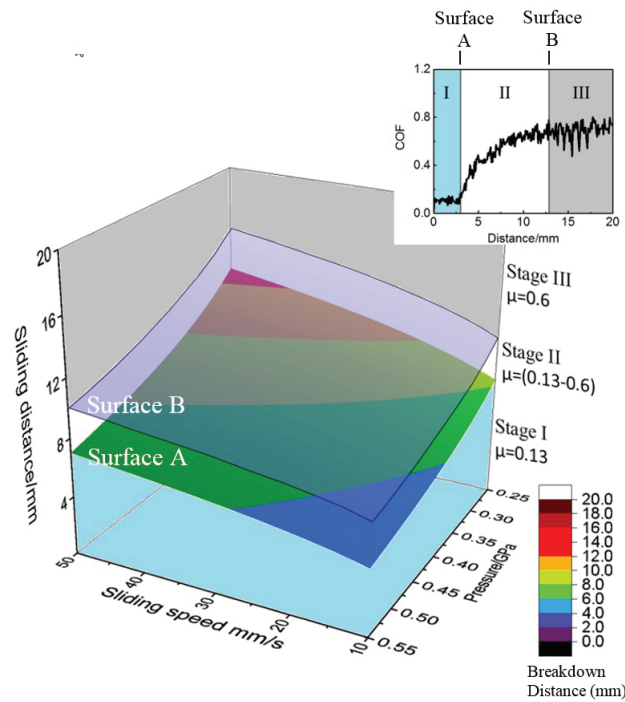


Figure 6.10: Three dimensional representation of the modelling result: lubricant behaviour window at 0.5 mg/mm^2 condition.

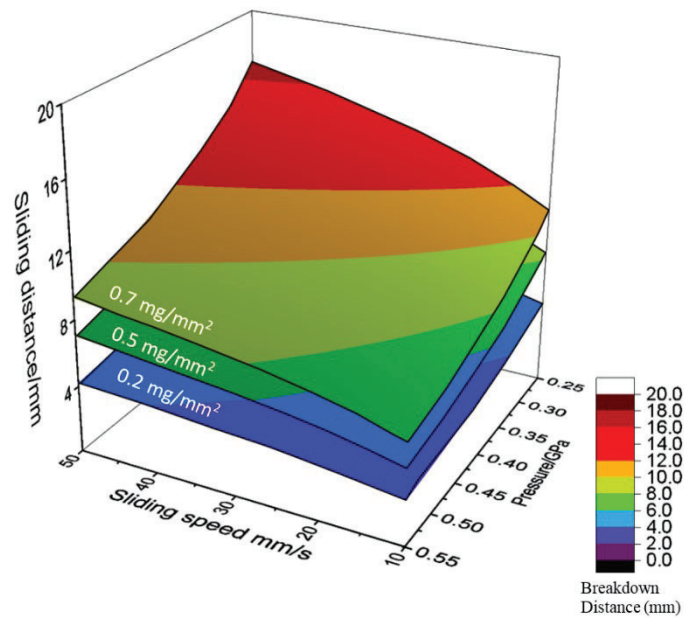


Figure 6.11: Lubricant behaviour window: breakdown distance surfaces of different initial lubricant quantity applied.

Chapter 7

Conclusions

7.1 General conclusions

The novelty of the study of tribological failures, namely galling and lubricant breakdown, presented here can be categorised in two items: firstly, instead of treating the tribological responses (friction and wear) as material constants, their evolutionary phenomena are quantitatively studied by the effect of operation parameters (contact load and sliding distance), and secondly, the interactive mechanisms have been built between these tribological responses, and interactive friction models were developed to predict their frictional evolutions.

It was found that: in galling, the build-up of transfer layer on the tool always leads to an increase of friction due to the high cohesion between the work-piece and the transfer layer, and a decrease of wear rate due to the hard tool asperities covered by relatively soft transfer layer. In lubricant film breakdown, the thinning of lubricant film will lead to the tribo-system to transfer to a dry sliding condition, in which friction increases due to the increased percentage of dry contact and wear rate increases due to the direct contact between contact pairs. The effects of operation parameters were especially studied, including contact pressure, sliding speed, sliding distance and lubricant amount. For galling, the material transfer layer is formed in a certain sliding distance, which can be shortened by increasing of contact pressure. The transfer volume is also increased with contact pressure. For lubricant

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breakdown, high contact load and low sliding speed will lead to this phenomenon. In the transportation of lubricant, the increase of lubricant amount can delay the breakdown. These dependencies of operation parameters and forming history were addressed in the interactive frictional modelling, which showed reasonable agreement when compared to the experimental results. These models could be used in the prediction of tool damage caused by galling and lubricant break down in aluminium metal forming, in which multi-cycle operations and rapid load changes are common.

The following contributions to the state of the art in the tribological responses and modelling of the interactive and evolutionary phenomena are highlighted:

In galling,

1. The aluminium transfer experiences a transition from a running-in state, in which the transfer amount gradually increases until a dynamic equilibrium is achieved, namely the steady state. Meanwhile, coefficient of friction begins at a low initial value and increases to a constant when the tribo-system reaches the steady state; wear rate begins at a high initial value and gradually reduces to a low constant at the steady state.
2. A high contact load will shorten the running-in state and increase the saturated transfer amount as well as wear rate at the steady state.
3. Contact load change led to a rapid change of the formation rate of the aluminium transfer layer and, furthermore, interactively changed the evolutions of coefficient of friction.

In lubricant breakdown:

1. The friction evolution process of a grease lubricated sliding contact comprises of three stages corresponding to different friction mechanisms, namely a stage I with fully flooded lubrication and low coefficient of friction, stage II with sharp increasing coefficient of friction, and stage III, a steady stage with a high coefficient of friction.

2. The gradual decrease of lubricant film thickness causes a transformation from fully flooded lubrication to boundary lubrication and the development of surface damage which marks the beginning of stage II.

7.2 Future work

In the real forming process, coating breakdown, galling and lubricant breakdown phenomena have a combining effect, where the lubricant breakdown directly leads to the surface scratching and build-up of galling and thus causes complex variations of friction and wear. As FE simulations with advanced tribological models are gradually applied, it is expected that the combined effect and interaction between friction and wear will be addressed in the future. Combined with FE technique, this method can overcome the experimental limits and will provide a general approach for the prediction of friction distribution and tool damage together, to further enable the tool-life prediction under multi-cycle forming condition.

In hot stamping process, temperature variation is a key operation parameter and significantly affects the local tribological condition by active chemical and mechanical reactions. These complex tribological processes involve many interactions between friction and wear that stem from the contact morphology variation. Due to the impossibility of recording real time distribution of friction and wear in metal forming, the methodology of modelling calibrated by lab-scale experimental result is applied to study the morphology variation, which requires a deep understanding of the mechanism of dependencies of every processes, such as contact pressure, sliding distance and temperature. Through the methodology of tribological experiment, numerical modelling and FE simulation, these dependencies are studied with the contact mechanisms to develop the interactive friction models, which will be implemented into FE software and realize the prediction of tool-life under multi-cycle condition.

The following future works to increase the accuracy and reliability of the interactive friction models are highlighted:

CONCLUSIONS

1. Develop holistic tribology models to address the combined effect of tribological failures and interaction between friction and wear.
2. Study the temperature effect on the evolutionary tribological phenomena in hot stamping process.
3. Calibrate holistic tribology models by multi-cycle industry forming tests.

In Appendix I, the interactive friction model that was developed to characterize the evolutions of coefficient of friction and galling on the tool surface, is implemented into the FE simulation of a sheet metal forming process for a feasibility study. This simulation addresses the evolution of galling distribution and COF variation under a multi-cycle forming condition and thus enables the prediction of tool-life.

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Appendix A

Calculation of galling area

After tests, surface morphologies were captured by moving samples from the tribometer. The galling area fraction covering the cast iron was calculated from the WLI 3D surface plot by height. As the original surface is a Gaussian surface, after test, asperities with height of $3\sigma \approx 500$ nm above the mean surface were identified as the transfer layer. σ is the standard deviation of the original cast iron surface and the standard of 3σ makes 99.7% of the asperities on the original surface can exclude from the calculation. Particles with the major dimension smaller than 500 nm may be neglected by this standard. However, they are beyond the scope of the present chapter and it will be proved that have very limited effect on the macro-scope galling area. The procedure of galling area calculation are listed below:

In the Veeco WLI software, Vision 4.10

1. Obtained WLI pictures are set as plan view.
2. The minimum limitation of height are set as 500 nm.
3. The color bar is set as grey scale.

In the statistic software, OriginPro 2015

1. The processed WLI picture are analysed and the black area (original area of substrate) can be calculated.
2. Pie charts are plotted as shown in Fig A.1.

As shown in Fig. A.1 example, the black (uncovered) and white (covered) area can be easily calculated. This figure shows the calculation process of WLI pictures obtained under 2N condition in test 8 of Chapter 4. Three covering areas are obtained (20.41%, 27.16% and 36.62%) and give an average of 28.06%.

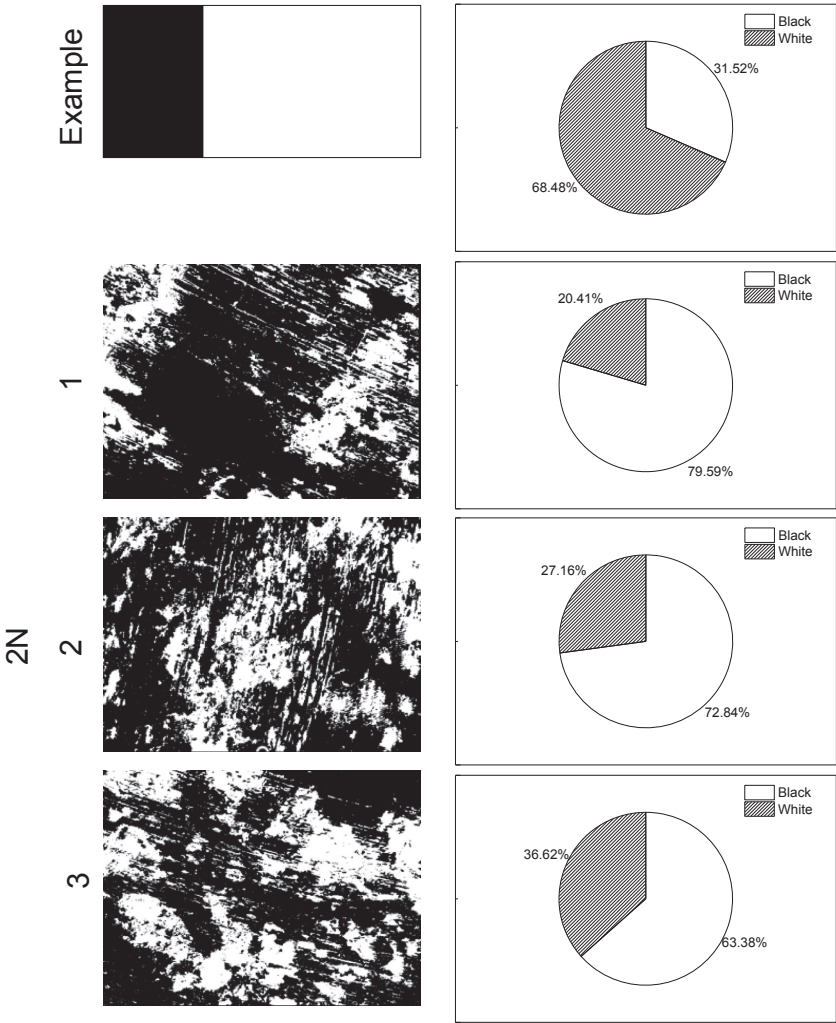


Figure A.1: Calculation of galling area under 2N.

Appendix B

Relation between the density of entrapped third body and galling area

De Rooij et al. (2013) developed an advanced method of modelling the material transfer on a single asperity scale. In this model, it is assumed that a transfer lump grows on a hexagon base (Fig. B1). The area of a hexagon base, A_{base} , can be calculated by Eq. B1. In this equation, l is the length of the base of the asperity before the top of the asperity in sliding direction, l_b is the effective length of the base of the asperity behind the top of the asperity; w_I , w_{II} and w_{III} are the widths at each region of the hexagon base. The volume of transferred aluminium will be deposited on this model of asperity. It was assumed that the hexagonal base is covered with a layer having a constant thickness. The width w_{II} remains the same during lump growth but w_I and w_{III} increase. Therefore, the volume of the asperity base, V_{base} , is calculated by Eq. B2 utilising the height of the asperity, h . The volume of aluminium during one cycle of transfer process can be calculated by Eq. B2. By calculating the transferred volume ($\dot{V}_{lump}$), a new asperity geometry can be calculated by solving scaling factor c_s and subsequently the dimension of h , l , w_I and w_{II} can be scaled accordingly.

$$A_{base} = (l + l_b) \left(\frac{1}{2} w_I + w_{II} + \frac{1}{2} w_{III} \right) \quad (B1)$$

$$V_{base} = \frac{hl}{6}(w_I + 3w_{II} + w_{III}) \quad (B2)$$

$$\dot{V}_{lump} = \frac{h}{l}((w_I + w_{II})c_s^2 + 2w_{II}c_s^2 - (w_I + 3w_{II} + w_{III})) \quad (B3)$$

These equations provide a method of calculating the geometry variation on a

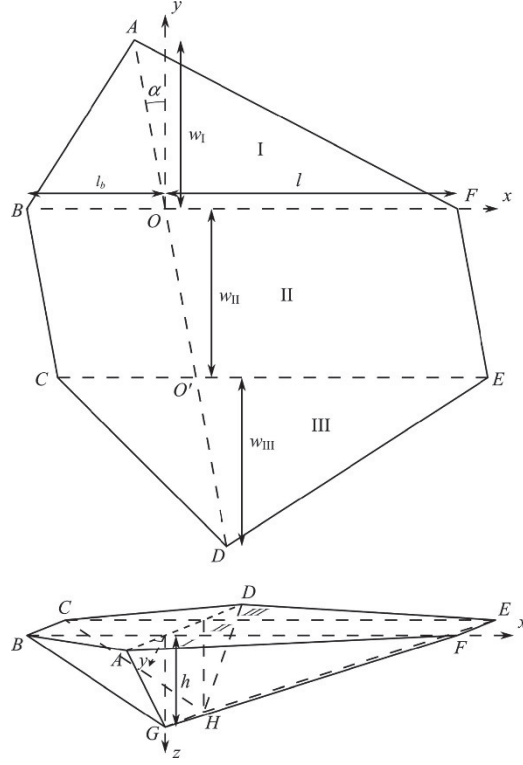


Figure B1: Dimensions of the hexagon base (de Rooij et al., 2013). A-H: outline of the asperity.

single aluminium lump during aluminium transfer that triggered by an asperity. A non-linear relation between $V_{lump}(t)$ and $f_{lump}(t)$ is presented in Eq. B1 to B3. To simplify this relation, Eq. B4 is constructed, where a and λ_1 are model parameters and $V_{lump,avg}(t)$ and $f_{lump,avg}(t)$ are the average volume and galling area of a single aluminium lump. This model describes the galling area increases from zero with the increase of the volume of a single aluminium transfer lump. By accumulating the volume of aluminium lumps at the contact interface, a macro-scale equation of galling area can be written by Eq. B5. Combining the definition of density of third body, Eq. B5 can be written as Eq.B6, where the galling area increases with the

density of third body increase at the interface.

$$f_{lump,avg}(t) = a(V_{lump,avg}(t))^{\lambda_1} \quad (\text{B4})$$

$$f(t) = \sum_{n=1}^N f_{lump,avg}(t) = a(NV_{lump,avg}(t))^{\lambda_1} \quad (\text{B5})$$

$$f(t) = a(V_{avg}A\rho_i(t))^{\lambda_1} = b(\rho_i(t))^{\lambda_1} \quad (\text{B6})$$

Appendix C

Calculation of $\dot{\bar{\rho}}_i$ under load change condition

In load change condition, the entrapped third body at time t is determined by the load change history. In a sliding process experiences n times of load change, the density of entrapped third body can be calculated by the accumulation of density of third body entrapped in each duration under a certain contact load (Eq. C.1). In this equation, the density of entrapped third body at time t_n can be calculated by the summation of $\rho_i(t_n) - \rho_i(t_{n-1})$, which indicates the amount of third body entrapped in the time duration $t_n - t_{n-1}$. The value of this term can be calculated by linear interpolation method in Eq. C.2, where the term, $\bar{\rho}_i(t_n) - \bar{\rho}_i(t_{n-1})$, presents the increase of normalised density of entrapped third body in this time duration. The ratio, $\frac{\rho_i(t_n) - \rho_i(t_{n-1})}{\bar{\rho}_i(t_n) - \bar{\rho}_i(t_{n-1})}$, is equals to fraction when the system reaches the steady state, which can be derived by using saturated values ($\rho_i(t_{sat,n})$ and $\bar{\rho}_i(t_{sat})$). Applying the saturated value, Eq. C.2 can be reformed into Eq. C.3.

$$\rho_i(t_n) = \sum_{i=1}^n (\rho_i(t_n) - \rho_i(t_{n-1})) \quad (C.1)$$

$$\frac{\rho_i(t_n) - \rho_i(t_{n-1})}{\bar{\rho}_i(t_n) - \bar{\rho}_i(t_{n-1})} = \frac{\rho_i(t_{sat,n}) - \rho_i(t_{n-1})}{\bar{\rho}_i(t_{sat}) - \bar{\rho}_i(t_{n-1})} \quad (C.2)$$
$$\bar{\rho}_i(t_{sat}) = 1$$

$$\rho_i(t_n) - \rho_i(t_{n-1}) = [\bar{\rho}_i(t_n) - \bar{\rho}_i(t_{n-1})] \left(\frac{\rho_i(t_{sat,n}) - \rho_i(t_{n-1})}{1 - \bar{\rho}_i(t_{n-1})} \right) \quad (C.3)$$

By calculating the limits of Eq. C.3, the $\dot{\rho}_i(t)$ can be calculated (Eq. C.4) and the final expression of $\dot{\rho}_i(t)$ can be written as Eq. C.5. In this equation, if the saturated entrapped third body is larger than the present density of entrapped third body ($\rho_i(t_{sat}) > \rho_i(t)$), the tribo-system will give a positive rate of $\dot{\rho}_i(t)$ and accumulate more third body at the contact. Otherwise, if the saturated entrapped third body is smaller than the already entrapped third body ($\rho_i(t_{sat}) < \rho_i(t)$) due to contact pressure reduction, the tribo-system is under an ‘oversaturation’ state and the entrapped third body will be ejected from the interface. When $\rho_i(t_{sat}) = \rho_i(t)$, a steady state of the tribo-system is reached that a zero change rate, $\dot{\rho}_i(t) = 0$ can be calculated.

$$\begin{aligned}\dot{\rho}_i(t) &= \lim_{t_n \rightarrow t_{n-1}} \frac{\rho_i(t_n) - \rho_i(t_{n-1})}{t_n - t_{n-1}} \\ &= \lim_{t_n \rightarrow t_{n-1}} \left(\frac{\bar{\rho}_i(t_n) - \bar{\rho}_i(t_{n-1})}{t_n - t_{n-1}} \right) \left(\frac{\rho_i(t_{sat,n}) - \rho_i(t_{n-1})}{1 - \bar{\rho}_i(t_{n-1})} \right)\end{aligned}\tag{C.4}$$

$$\dot{\rho}_i(t) = \dot{\bar{\rho}}_i(t) \left(\frac{\rho_i(t_{sat}) - \rho_i(t)}{1 - \bar{\rho}_i(t)} \right)\tag{C.5}$$

Appendix D

Expression of wear volume

The wear volume between the solid aluminium and cast iron stems from two origins corresponding to two different mechanisms. $V_{s,Al-CI}(t)$ indicates the wear volume generated between the solid aluminium and clean cast iron surface, whereas $V_{s,Al-Al}(t)$ indicates the wear volume generated between the solid aluminium and aluminium third body layer. $V_{s,Al-CI}(t)$ can be referred to the wear volume generated at the beginning of running-in state, where direct contact between solid aluminium and cast iron occurs at the interface. $V_{s,Al-Al}(t)$ can be referred to the wear volume generated at the steady state generated between the solid aluminium and the aluminium third body. Therefore, the transition of the contact mechanisms can be modelled by the formation of the aluminium third body and $V_s(t)$ can be written as a function of the variation of normalised density of third body, $\bar{\rho}_i(t)$ (Eq. D.1).

$$V_s(t) = V_{s,Al-CI}(t)(1 - \bar{\rho}_i(t)) + V_{s,Al-Al}(t)\bar{\rho}_i(t) \quad (D.1)$$

The expressions of $V_{s,Al-CI}(t)$ and $V_{s,Al-Al}(t)$ can be written in the form of Holm-Archard equation (Eq. D.2 and D.3), where K_{Al-CI} and K_{Al-Al} are the specific wear coefficient of aluminium-cast iron and aluminium-aluminium contact, respectively, P is the contact pressure, A is the contact area and x is the sliding distance. This equation provides: (i) a feasible form for time integration and (ii) the relationship of steady state wear between different parameters. These parameters include: (i) operation parameters (contact pressure/load and sliding displacement), (ii) nature of material and (iii) other system variables, e.g. surface roughness, micro-material

structures, combining the effects of adhesive and abrasive wear and tribo-chemical reaction, which all contribute to the specific wear coefficient, K (Kato, 2002). As of yet, this coefficient has not been well described by a physical model.

$$V_{s,Al-CI}(t) = K_{Al-CI}AP(t)x(t) \quad (D.2)$$

$$V_{s,Al-Al}(t) = K_{Al-Al}AP(t)x(t) \quad (D.3)$$

Substituting Eq. D.2 and D.3 in Eq. D.1, $V_s(t)$ can be written as Eq. D.4.

$$V_s(t) = K_{Al-CI}AP(t)x(t)(1 - \bar{\rho}_i(t)) + K_{Al-Al}AP(t)x(t)\bar{\rho}_i(t) \quad (D.4)$$

This equation can be written as Eq. D.5 where the term $[K_{Al-CI}(1 - \bar{\rho}_i(t)) + K_{Al-Al}\bar{\rho}_i(t)]$, represents the specific wear coefficient. When the tribo-system reaches the steady state, $\bar{f}(t) = 1$ and the expression of $V_s(t) = K_{Al-Al}APx(t)$, which becomes the form of classic Holm-Archard equation at the steady state. In the classic Holm-Archard equation, it can only model the wear volume variation at the steady state and the wear coefficient is a constant (Yang, 2003). Eq. D.6 provides a form of the wear coefficient in galling phenomenon to model the wear volume variation in both the running-in and steady state.

$$V_s(t) = [K_{Al-CI}(1 - \bar{\rho}_i(t)) + K_{Al-Al}\bar{\rho}_i(t)]AP(t)x(t) \quad (D.5)$$

$$K = K_{Al-CI}(1 - \bar{\rho}_i(t)) + K_{Al-Al}\bar{\rho}_i(t) \quad (D.6)$$

According to the definition of the density of third body, $\rho_s(t)$ can be written as Eq. D.7

$$\rho_s(t) = \frac{V_s(t)}{AV_{avg}} \quad (D.7)$$

Substituting Eq. eqD.5 into Eq. D.7, $\rho_s(t)$ can be expressed by Eq. D.8.

$$\rho_s(t) = \frac{1}{V_{avg}}[K_{Al-CI}P(t)x(t)(1 - \bar{\rho}_i(t)) + K_{Al-Al}P(t)x(t)\bar{\rho}_i(t)] \quad (D.8)$$

Under a constant contact load condition, the derivative of Eq. D.8 with respect to time, t , can be written as Eq. D.9.

$$\begin{aligned} \dot{\rho}_s(t) = & \left[\frac{(K_{Al-Al} - K_{Al-CI})}{V_{avg}} Px(t) \right] \dot{\bar{\rho}}_i(t) \\ & + \left[\frac{(K_{Al-Al} - K_{Al-CI})}{V_{avg}} Pv(t) \right] \bar{\rho}_i(t) \\ & + \frac{K_{Al-CI}Pv(t)}{V_{avg}} \end{aligned} \quad (D.9)$$

Under a change load condition, the derivative of Eq. D.8 with respect to time, t , can be written as Eq. D.10.

$$\begin{aligned}
 \dot{\rho}_s(t) &= \left[\frac{(K_{Al-Al} - K_{Al-CI})}{V_{avg}} P(t)x(t) \right] \dot{\rho}_i(t) \\
 &+ \left[\frac{(K_{Al-Al}P(t)v(t) + K_{Al-Al}\dot{P}(t)x(t) - K_{Al-CI}P(t)v(t) - K_{Al-CI}\dot{P}(t)x(t))}{V_{avg}} \right] \bar{\rho}_i(t) \\
 &+ \frac{K_{Al-CI}}{V_{avg}} (Pv(t) + \dot{P}(t)x(t))
 \end{aligned} \tag{D.10}$$

Based on the discussion, the evolution of $\dot{\rho}_s(t)$ decreases from a maximum value at the beginning of running-in state to a stable value at the steady state. In Eq. D.9, as $K_{Al-Al} - K_{Al-CI} < 0$, the decreasing trend is presented by the first and second terms. The first term $\left[\frac{(K_{Al-Al} - K_{Al-CI})}{V_{avg}} Px(t) \right] \dot{\rho}_i(t)$, represents the decrease of source rate due to the decrease of formation rate of the third body layer. In this process, $\dot{\rho}_i(t)$ gradually reduces to zero when reaches the steady state. The second term, $\left[\frac{(K_{Al-Al} - K_{Al-CI})}{V_{avg}} Pv(t) \right] \bar{\rho}_i(t)$, represents a decrease of source rate due to the increase of entrapped third body, where this term reduces from zero at the beginning of running-in and reaches the minimum at the steady state. To simplify the equation, the second term can be treated as a constant, which equals to its saturated value. Eq. D.9 can be rewritten in the form of Eq. D.10, where c and d are model constants. At the beginning of the running-in state, the term, $c\dot{\rho}_i(t)$, combines the effect of the first and second term and plays the important role on influencing the evolution. When the tribo-system is close to the steady state, the constant term, d , which is the combination of the second and third term in Eq. D.9 dominates the evolution.

$$\dot{\rho}_s(t) = c\dot{\rho}_i(t) + d \tag{D.11}$$

The model parameters in Eq. D.10 can be calculated by applying boundary condition. When the tribo-system reaches the equilibrium ($t = t_{sat}$), $\dot{\rho}_i(t_{sat}) = 0$ and d equals to $\dot{\rho}_s(t_{sat})$ (Eq. D.11). Substituting in D.10, $\dot{\rho}_s(t)$ can be written as Eq. D.12. A constant n was applied to replace the ratio, $\frac{c}{\dot{\rho}_s(t_{sat})}$ (Eq. D.14), which represents the

magnitude relation between the initial wear rate and its saturated value. The value of n can be determined from further experimental measurement and the expression of source rate can be written as Eq. D.15.

$$\dot{\rho}_s(t_{sat}) = c\dot{\rho}_i(t_{sat}) + d = d \quad (\text{D.12})$$

$$\dot{\rho}_s(t) = \dot{\rho}_s(t_{sat}) \left[\frac{c}{\dot{\rho}_s(t_{sat})} \dot{\rho}_i(t) + 1 \right] \quad (\text{D.13})$$

$$n = \frac{c}{\dot{\rho}_s(t_{sat})} \quad (\text{D.14})$$

$$\dot{\rho}_s(t) = \dot{\rho}_s(t_{sat}) [n\dot{\rho}_i(t) + 1] \quad (\text{D.15})$$

Appendix E

Expression of ejection rate

The expression of the change rate of density of ejected third body can be written as Eq. E.1. The model parameter e can be calculated by applying the boundary condition to this equation when the tribo-system reaches the equilibrium. When $t = t_{sat}$, the value of $\dot{\rho}_e(t_{sat}) = \dot{\rho}_s(t_{sat}) = \text{constant}$ and $\bar{\rho}_i(t_{sat}) = 1$ (Eq. E.2). Thus $e = \dot{\rho}_s(t_{sat})$ (Eq. E.3). Substituting in Eq. E.1, the equation of ejection rate can be rewritten as Eq. E.4.

$$\dot{\rho}_e(t) = e\bar{\rho}_i(t) \quad (\text{E.1})$$

$$\dot{\rho}_e(t_{sat}) = e\bar{\rho}_i(t_{sat}) = \dot{\rho}_s(t_{sat}) \quad (\text{E.2})$$

$$e = \dot{\rho}_s(t_{sat}) \quad (\text{E.3})$$

$$\dot{\rho}_e(t) = \dot{\rho}_s(t_{sat}) \bar{\rho}_i(t) \quad (\text{E.4})$$

Appendix F

Expression of saturated source rate

The term, saturated source rate ($\dot{\rho}_s(t_{sat})$) can be modelled by building the relation between $\dot{V}_s(t_{sat})$ and $\dot{\rho}_s(t_{sat})$. At the steady state, the wear volume, $V_s(t)$, can be modelled by Holm-Archard equation (Eq. F.1) (Fillot et al., 2007), where K is the specific wear coefficient, P is the contact pressure, A is the contact area and x is the sliding distance. The instantaneous form of Holm-Archard equation can be calculated by the derivation of Eq. E.1 with respect to time, t (Eq.F.2).

$$V_s(t) = KAPx(t) \quad (F.1)$$

$$\dot{V}_s(t) = KA[P(t)v(t) + \dot{P}(t)x(t)] \quad (F.2)$$

By the definition of density of third body (Eq. 6), the source rate can be expressed as Eq. F.3. Substituting Eq. F.2 in Eq. F.3 and combining the constants, K , V_{avg} into B , the expression of the saturated source rate under the sliding speed, $v(t)$, can be modelled by Eq. F.4.

$$\dot{\rho}_s(t_{sat}) = \frac{1}{A} \left(\frac{\dot{V}_s(t_{sat})}{V_{avg}} \right) \quad (F.3)$$

$$\dot{\rho}_s(t_{sat}) = B[P(t)v(t) + \dot{P}(t)x(t)] \quad (F.4)$$

Appendix G

Derivatives of contact pressure

In the present experiment, contact pressure (MPa) is generated by two different dead weight (2 and 10 N) and can be numerically expressed by piecewise functions (load increase: Eq. G.1; load decrease: Eq. G.2).

$$P(t) = \begin{cases} 0.64, & 0 < t < 125.6 \\ 3.18, & x \geq 125.6 \end{cases} \quad (\text{G.1})$$

$$P(t) = \begin{cases} 3.18, & 0 < t < 62.8 \\ 0.64, & x \geq 62.8 \end{cases} \quad (\text{G.2})$$

In the load increase condition, the derivative process can be separated into 3 cases: $0 < t < 125.6$, $125.6 < t$ and $t = 125.6$. For the first two cases, the function $P(t)$ is defined by constants and the derivatives is shown in Eq. G.3. At $t = 125.6$, it is clearly that the left-hand and right-hand limits are not equal and thus $P(t)$ is not differentiable for $t = 125.6$. Similar to the derivative process of load increase condition, the derivative of load decrease condition can be expressed by Eq. G.4.

$$\dot{P}(t) = \begin{cases} 0, & 0 < t < 125.6 \\ 0, & x > 125.6 \end{cases} \quad (\text{G.3})$$

$$\dot{P}(t) = \begin{cases} 0, & 0 < t < 62.8 \\ 0, & x > 62.8 \end{cases} \quad (\text{G.4})$$

Appendix H

Expression of the saturated density of third body at the interface

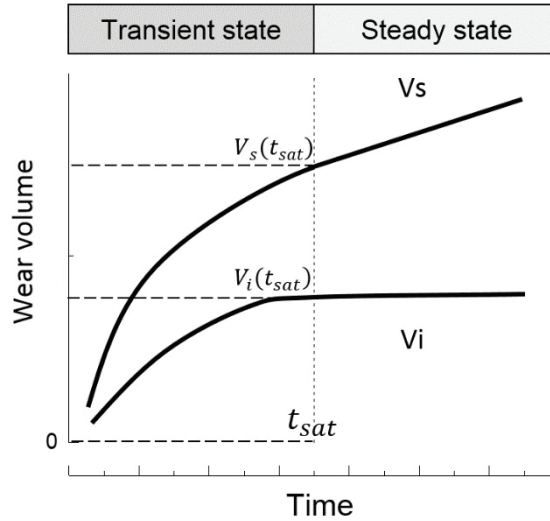


Figure H.1: Fractional relationship between $V_i(t_{sat})$ and $V_s(t_{sat})$.

When $t = t_{sat}$, the volume of third body that is entrapped at the interface can be calculated by Eq. H.1.1, where the entrapped third body, $V_i(t_{sat})$, is a fraction of the wear volume from solid aluminium, $V_s(t_{sat})$ (as shown in Fig. H.1), and E is the fraction factor. By the definition of the density of third body, the saturated

density of third body at the interface, $\rho_i(t_{sat})$, can be written as Eq.H.2. As $V_s(t_{sat})$ can be expressed as the Holm-Archard equation, the expression of $\rho_i(t_{sat})$ can be transformed into Eq. H.3.

$$V_i(t_{sat}) = EV_s(t_{sat}) \quad (\text{H.1})$$

$$\rho_i(t_{sat}) = \frac{1}{AV_{avg}} V_i(t_{sat}) = \frac{1}{AV_{avg}} EV_s(t_{sat}) \quad (\text{H.2})$$

$$\rho_i(t_{sat}) = \frac{1}{AV_{avg}} E(KAP(t_{sat})x(t_{sat})) \quad (\text{H.3})$$

Based on the experimental result, the sliding distance of running-in state, $x(t_{sat})$, is a function of the contact pressure, where the increase of contact pressure will shorten the sliding distance at the running-in state. Thus, the expression of $x(t_{sat})$ can be written as Eq. H.4, where K' and λ_2 are model parameters and $1 > \lambda_2 > 0$. By substituting Eq. H.4 in Eq. H.3, the equation of $\rho_i(t_{sat})$ can be expressed as Eq. H.5. Combining the model constants of V_{avg} , E , K and K' , the equation of $\rho_i(t_{sat})$ can be rewritten as Eq. H.6.

$$x(t_{sat}) = K'P^{-\lambda_2} \quad (\text{H.4})$$

$$\rho_i(t_{sat}) = \frac{1}{AV_{avg}} [E(KK'AP^{1-\lambda_2})] \quad (\text{H.5})$$

$$\rho_i(t_{sat}) = CP^{1-\lambda_2} \quad (\text{H.6})$$

Appendix I

FE simulation of aluminium galling induced forming tool failures under multi-cycle loading conditions: a feasibility study

I.1 Introduction

¹Unlike many other transient tribological phenomena that occur in a short period, galling process has a long running-in state, which may go through hundreds of forming cycles before reaching the steady state (Olsson et al., 2002). This long duration makes the tribological variables, such as galling area, wear rate and coefficient of friction, variate between each forming cycle (Heinrichs and Jacobson, 2011). In this work, a multi-cycle stamping modelling is conducted to address these variations. The evolution of galling area and coefficient of friction between workpiece and the die material as well as the effects of key operation parameters have been studied.

¹This chapter is based on paper work ‘Y. Hu, Y. Zheng, D. J. Politis, L. Wang, *FE simulation of aluminium galling induced forming tool failures under multi-cycle loading conditions: a feasibility study*, accepted by The 4th International Conference on Advanced High Strength Steel and Press Hardening, 2018.’

This quantitative model not only deepens the understanding of the evolutionary tribological conditions at the tool-workpiece interface but also enables a tool-life prediction in terms of critical galling area and coefficient of friction.

I.2 Interactive friction model

During the dry sliding between an aluminium alloy and cast iron, the aluminium transfer process is found to be dependent on sliding distance and contact load. Furthermore, it is revealed that this transfer process has induced the transition in the frictional mechanisms and thus affects the evolution of the coefficient of friction. These dependencies and interactions can be drawn as follows. i) Sliding distance dependency: the aluminium transfer experiences a transition from a running-in state, in which the transfer amount gradually increases until to a dynamic equilibrium, namely, the steady state. Meanwhile, the coefficient of friction starts off at a low initial value and increases to a constant when the tribo-system reaches the steady state. ii) Contact load dependency: a high contact load will shorten the running-in state and increase the saturated transfer amount as well as wear rate at the steady state.

Based on these analysis, an interactive friction model is developed to enable the prediction of evolutions of the coefficient of friction and the galling area. The origins of the coefficient of friction can be classified as i) aluminium-cast iron contact and ii) aluminium-aluminium contact (Eq. 4.2). To represent the variation of galling area in the formation progression of transfer layer, a normalised galling area, $\bar{f}(t)$ is introduced, which is defined as the galling area at a certain sliding time ($f(t)$) by the saturated galling area at the steady state ($f(t_{sat})$) (Eq. 4.3). The coefficient of friction, as a function of the transfer progress, can be predicted, where the coefficient of friction generated at the clean surface is gradually replaced by the saturated value with the increase of normalised galling area.

At the equilibrium state, although wear particles are continually generated, the morphology of the interface is relatively stable (de Rooij et al., 2013, Österle et al.,

2012). An important conclusion is drawn from these observations where the wear particle generation and ejection mechanisms play a critical role in balancing the wear particles and lumps (third body) at the contact. To describe this dynamic balance, the concept of third body density, ρ , is introduced, where it is a measurement of the number of third body elements (N) in a unit contact area (A) (Eq. 4.4). The variation of $f(t)$ is the main consequence of the accumulation of third body at the interface, which can be expressed by Eq. 4.7, where b and λ_1 are model parameters.

In the running-in state, there is a competition between the rate of generation ($\dot{\rho}_s$) and ejection ($\dot{\rho}_e$) at the contact. This competition results in the accumulation rate of entrapped third body at the interface, $\dot{\rho}_i$ (Eq. 4.10). The calculation of $\rho_i(t)$ is shown by Eq. 4.11. The expression of $\dot{\rho}_s$, $\dot{\rho}_e$ and $\rho_i(t)$ can be written by Eq. I.1, Eq. I.2 and Eq. I.3, where m , n and B are model parameters. In simulation, the blank temperature varies in a narrow temperature range between 400-500 °C (Liu et al., 2017), where the effect of temperature on galling is not sensitive. Therefore, the temperature parameter, C_T , can be treated as a constant in this study.

$$\dot{\rho}_s(t) = \mathbf{C_T} B P v(t_{sat}) [n \dot{\bar{\rho}}_i(t) + 1] \quad (\text{I.1})$$

$$\dot{\rho}_e(t) = \mathbf{C_T} B P v(t_{sat}) \bar{\rho}_i(t) \quad (\text{I.2})$$

$$\rho_i(t_{sat}) = B/mP^{1-\lambda_2} \quad (\text{I.3})$$

Table A1: Model constant of interactive friction model

Parameters	m	n	B	C_T	b	μ_{Al-Al}	μ_{Al-CI}	λ_1	λ_2
Values	0.1	15	0.09	10.5	0.43	0.20	0.55	1.6	0.7

I.3 Numerical simulation of hot stamping

Finite element (FE) simulations are conducted using the commercial software PAM-STAMP. As shown in Fig. I.1, the simulation model comprises of a punch, die, AA6082 tailor welded blanks (TWBs) and blank holder pairs. The tools are modelled

using rigid elements and a stepped surface is made on the punch facing towards the TWB. A friction coefficient of 0.2 is chosen to account for surface interaction between the sliding sheet and the die assembly (Liu et al., 2017). The data obtained from

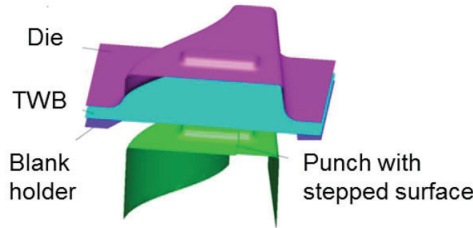


Figure I.1: PAM-STAMP simulation model.

PAM-STAMP is imported into a MATLAB programme, in which the multi-cycle forming calculation is carried out by running the friction model cyclically. Galling area and coefficient of friction are set as outputs. After the calculation, the result from the MATLAB programme is imported back to PAM-STAMP and thus the distribution of results relating to each parameter can be visually analyzed.

I.4 Results and discussion

I.4.1 Distribution characteristics of tribological conditions

The tribological conditions of sheet metal forming have the following characteristics : i) uneven in spatial distribution and ii) uneven in historical distribution, which indicates that these influential parameters will vary with location on the tool/work-piece surface as well as forming phases/number of cycles. In this simulation, different die areas present clear variation of tribological conditions. As shown in Fig. I.2, a medium contact pressure (0-3MPa) and a high sliding distance (17-60mm) are applied on the blank holder areas. On the contrary, the stepped feature on the bottom of die generates a high contact pressure (10-122MPa) but short sliding distance (≤ 10 mm) due to die closure. The entry and exit of radius areas experience a medium contact pressure (3-6MPa) as well as a high sliding distance (25-52mm).

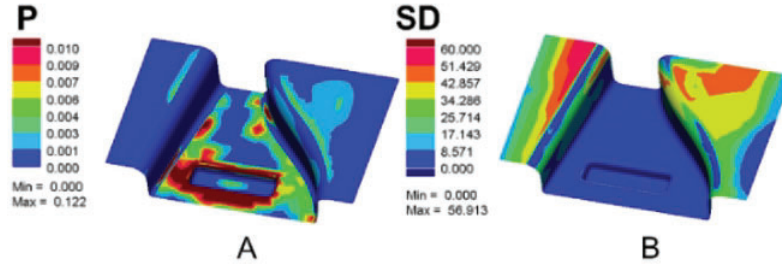


Figure I.2: Distribution characteristics of pressure, P (A) and sliding distance, SD (B) on the die surface at the end of one forming cycle.

I.4.2 Effect of forming cycles

At the same location with same contact pressure, the galling area increases with the increase of forming cycle. Based on the general observations described by Holm-Archard equation, there is a linear relationship between the wear volume and the amount of entrapped wear particles, which is the source of aluminium transfer layer. Thus, the increase of entrapped wear particles results in an increase of the galling area.

The different tribological conditions also result in a variation of accumulation of galling area. It can be seen in Fig. I.3 that the contact pressure and sliding distance have a combining effect on material transfer. Due to the combining of high contact pressure and long sliding distance, initial galling is generated on the entry of radius area at 5 cycles. Then the blank holder area starts to generate galling layer due to high sliding distance (10 cycle). Although the bottom area is subject to short sliding distance, the stepped feature generates high contact pressure, which strengthens the galling accumulation. At the end (300 cycles), both the blank holder areas and the stepped feature area accumulate a severe galling area that close to 100%.

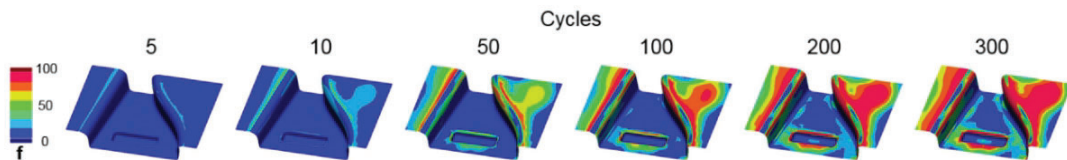


Figure I.3: Distribution of galling area under different forming cycles.

I.4.3 Effect of contact pressure

Fig. I.4 shows the evolutions of galling area and coefficient of friction of two points on the die with increasing forming cycles. As shown in Fig. I.4A, point A and B are two elements, which locate at entry and exit of the radius area respectively with large difference of contact pressures (A: 0.83 MPa; B: 3.93MPa) but similar sliding distances (A: 36.06mm; B: 36.08mm). Under the high contact pressure at the entry, point B has a larger generation rate of galling area with forming cycles increasing (Fig. I.4B). At this point, the contact reaches the steady state of galling area earlier than Point A (B: no. of cycle=136, A: no. of cycle=250) and has a high saturation galling area (B: 96%; A: 92%). The galling area increase will simultaneously increase the friction at the contact and change the distribution of COF. Coefficients of friction of point A and B increase in a similar pattern as galling area. At the beginning clean contact, the coefficients of friction start at a constant value of 0.2. When they reach the steady state, a pressure-independent coefficient of friction of 0.55 is obtained.

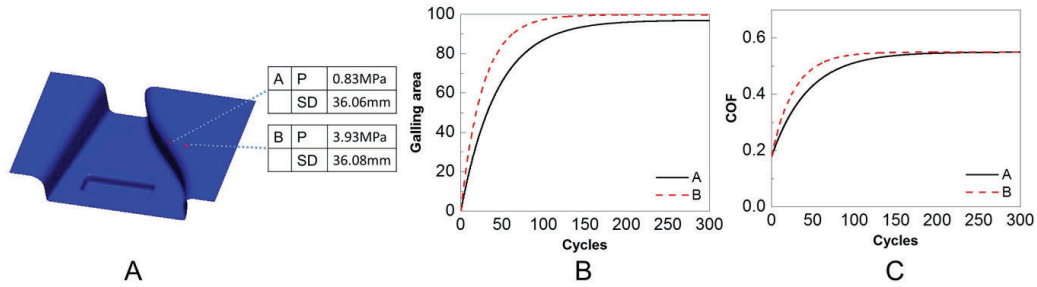


Figure I.4: Effect of contact pressure on galling area and coefficient of friction. (A) the location of point A and B; (B) galling area evolution of point A and B; (C) coefficient of friction evolution of point A and B.

I.5 Conclusions

In the present research, an interactive friction model is implemented into a stamping simulation and enables a tool-life prediction. The dependency of pressure and sliding distance (forming cycles) can be drawn as follows: i) the aluminium galling area

gradually increases with increasing forming cycles until a steady state and ii) high contact pressure will shorten the running-in state and increase the saturated transfer amount.