Theoretical & Applied Mechanics Letters 8 (2018) 57-61

Contents lists available at ScienceDirect

Theoretical & Applied Mechanics Letters

journal homepage: www.elsevier.com/locate/taml

Letter

A first order friction model for lubricated sheet metal forming

Hans B. Löfgren*

School of Business, Engineering and Science, Halmstad University, P. O. Box 823, SE - 301 18 Halmstad, Sweden

HIGHLIGHTS

- A first order friction model for sheet metal forming simulations is derived.
- The friction model is based on local contact conditions.

ARTICLE INFO

Article history:
Received 10 October 2017
Received in revised form 28 November 2017
Accepted 29 November 2017
Available online 12 December 2017
*This article belongs to the Biomechanics and Interdiscipline

Keywords:
Friction model
Lubrication
Sheet metal forming

ABSTRACT

This paper presents the derivation of a first order friction model for lubricated sheet metal forming. Assuming purely plastic real contacts, Newton's law of viscosity, and a square root behavior of the hydrodynamic coefficient of friction with respect to the hydrodynamic Hersey parameter an analytic model is found. The model predicts the coefficient of friction as a function of the relative pressure, the relative Hersey parameter and the real contact coefficient of friction. Questions about local and global friction are raised in the validation of the model against flat tool sheet experiments. For some flat tool sheet experiments reasonable agreements are obtained assuming a rigid punch pressure distribution. The restricted number of user inputs makes the model useful in early tool design simulations.

The need for variable friction models is greater than ever due to the increasing demand from industry to predict manufacturing processes through simulations. A general purpose friction model is however only tractable for processes where the friction can be determined by local conditions. Since lubricants like oil are able to transmit and build up pressure in lubricated contacts over large regions, local friction models will ultimately fail unless the contact is nominally flat or is in a state of starvation.

An application that benefits from a local friction model is the sheet metal forming process [1-10]. Emmens [1] investigates both theoretically and experimentally the variable friction of the deep drawing process. Combining a roughness parameter with the Hersey parameter, a linear friction model for mixed lubrication is derived for low loads. For high load boundary lubrication an equally simple and widely used model exists called the shear-cap friction model [4]. Both these models are central to the study presented in this paper.

In deriving a model we have to assume, despite the experimental difficulties, that there is something simple about lubricated friction and make approximations accordingly. The difficulties in finding a "stable" experimental technique could be explained by noting that friction is a global phenomenon. The global phenomenon of friction arises from the fact that most contacts generate pressure distributions that deviate considerably from the mean pressure approximation. Since different pressure distributions can give the same mean pressure we understand that friction can appear "unstable" when measured.

In the following analysis we will apply a continuum mechanical approach assuming that there exists a local mean pressure, p, on a scale much smaller than the nominal contact area but much larger than the microscopic scale of the roughness. At this local scale of area A, we may construct the normal and tangential load balances as

$$pA = p_c A_c + p_h (A - A_c)\,, \tag{1}$$

$$\mu p A = \tau_c A_c + \tau_h (A - A_c)\,, \tag{2}$$

* Corresponding author.
E-mail address: hans.lofgren@hh.se (H.B. Löfgren) .

http://dx.doi.org/10.1016/j.taml.2018.01.002

where μ is the local coefficient of friction, p_c and τ_c are the real contact mean pressure and shear stress, p_h and τ_h are the hydrodynamic mean pressure and shear stress, and A_c is the real contact area (Fig. 1).

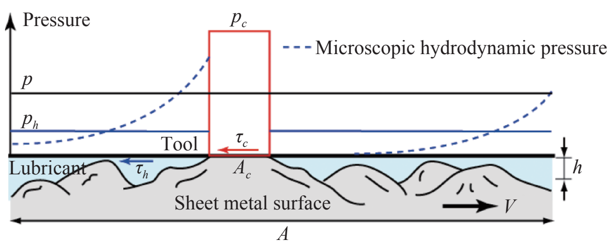

Fig. 1. Schematic representation of the lubricated contact between the tool and the sheet metal.

We define the Hersey parameters and the hydrodynamic coefficient of friction as

$$He = \frac{\eta V}{p}, \tag{3}$$

$$He_h = \frac{\eta V}{p_h}, \tag{4}$$

$$\mu_h = \frac{\tau_h}{p_h}, \tag{5}$$

where η is the dynamic viscosity and V the sliding velocity.

We also make the following assumptions

$$\tau_h = \frac{\eta V}{h}, \tag{6}$$

$$\mu_h \propto \sqrt{He_h}, \tag{7}$$

where Eq. (6) is Newton's law of viscosity with h as the mean separation between the lubricated surfaces. This assumption should be seen as a measure of the mean hydrodynamic shear stress within the fluid cavities of the local scale A. By Eq. (7), we assume that the behaviour of the hydrodynamic coefficient of friction found in hydrodynamic lubrication [1] is valid in all lubrication regimes.

The most important consequences of Eqs. (3)-(7) are

$$\frac{He_h}{h^2} = \frac{\mu_h}{h} = c_E, \tag{8}$$

$$\frac{p_h}{p} = \frac{He}{He_0}\left(\frac{h_0}{h}\right)^2, \tag{9}$$

$$\frac{\tau_h}{p} = \mu_0 \frac{He}{He_0}\frac{h_0}{h}, \tag{10}$$

where $c_E \approx 1\times10^3\ \mathrm{m}^{-1}$ is Emmens constant [1]. $He_0 = c_E h_0^2$ and $\mu_0 = c_E h_0$ are respectively the Hersey parameter and the hydrodynamic coefficient of friction at the separation h_0 (the peak height of the surface roughness).

We define

$$\alpha = \frac{A_c}{A}, \tag{11}$$

$$\mu_c = \frac{\tau_c}{p_c}, \tag{12}$$

where α is the real contact area ratio and μ_c is the real contact coefficient of friction.

We assume purely plastic real contacts based on the assumptions

$$p_c = H_s, \tag{13}$$

$$\beta = \frac{1}{2}\left(\frac{h_0 - z}{h_0}\right)^2 \quad \text{for} \quad 0 \le z \le h_0, \tag{14}$$

$$\alpha = 2\beta \quad \text{at} \quad z = h, \tag{15}$$

where H_s is the surface hardness and β the Abbott–Firestone curve of the sheet metal surface. The assumption of purely plastic real contacts is commonly applied for rough contacts between a soft and a hard surface. This means that the asperities of the soft surface will deform plastically at some pressure, here called the surface hardness H_s. The actual value of this hardness may depend on sliding speed, temperature, and deformation (strain hardening), which ultimately are given by experiments. The Abbot–Firestone curve describes the material distribution along a coordinate normal to the surface. Equation (14) is the simplest approximation to the upper half of the Abbott–Firestone curve valid for a typical sheet metal. The tool surface is assumed to be rigid and flat. During plastic flattening of the roughness the displaced material is forced to flow in some way. Equation (15) is found assuming that the displaced volume fills the fluid cavity according to Fig. 2.

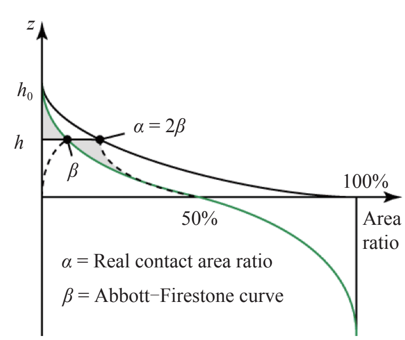

Fig. 2. The real contact area determined by mirror imaging volume displacements against the separation line h.

The most important consequences of Eqs. (11)-(15) are

$$h_0 = \frac{1}{2}S_k + \frac{3}{2}S_{pk}, \tag{16}$$

$$\alpha = \left(\frac{h_0 - h}{h_0}\right)^2 \quad \text{for} \quad 0 \le h \le h_0, \tag{17}$$

$$\tau_c = \mu_c H_s, \tag{18}$$

where the statistical surface parameters S_k and S_{pk} are the core