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Multi-scale friction modeling for sheet metal forming: The mixed lubrication regime

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ABSTRACT

A mixed lubrication friction model is presented to accurately account for friction in sheet metal forming FE simulations. The advanced friction model comprises a coupling between a hydrodynamic friction model and a boundary lubrication friction model, based on the lubricant film thickness. Mixed lubrication interface elements are introduced to solve the governing differential equations. The interface elements have been implemented in FE forming software. Two deep-drawing applications are discussed to demonstrate the performance of the friction model. Results show friction coefficients that vary in space and time, and depend on external process settings like the amount and type of lubricant. A comparison with experimentally obtained punch-force displacement diagrams is made to prove the enhanced predictive capabilities of FE forming simulations.

1. Introduction

Finite element (FE) formability analyses are everyday practice in the metal-forming industry. Metal forming processes are controlled by restraining the workpiece material through friction between the workpiece and dies. This interaction makes both tribological as mechanical knowledge necessary to optimize forming processes. In the majority of FE simulations a simple Coulomb friction model is however still used. The Coulomb friction model does not include the influence of important parameters such as pressure, punch speed or the type of lubricant used [1–4].

Four main lubrication regimes can be defined to describe lubricant flow under metal forming conditions [5]. The lubrication regime is called the boundary lubrication regime if the normal load is totally carried by contacting surface asperities. Hydrodynamic stresses within the lubricant do not affect friction in this regime. In the remaining regimes, the load is either totally carried by the lubricant (thick film and thin film lubrication regimes) or partly carried by the lubricant and partly carried by contacting surface asperities (mixed lubrication regime). These lubrication regimes are distinguished by the ratio between the fluid film thickness h and the composite RMS surface roughness S_q [5]. The lubrication regime is called the thick film lubrication regime for $h/S_q > 10$, the thin film lubrication regime for $3 < h/S_q < 10$ and the mixed lubrication regime for $h/S_q < 3$.

In the boundary lubrication regime friction is mainly described by adhesion and ploughing between contacting asperities. Wilson [6] and Challen and Oxley [7,8] developed models to account for these effects theoretically. Wilson [6] described the effect of adhesion and ploughing separately, while Challen and Oxley take the combined effect of ploughing and adhesion into account. The real area of contact, playing an important role in characterizing friction, relies on the roughness of both the tool and the workpiece surface. The roughness of the workpiece is liable to changes due to flattening and roughening mechanisms. The main asperity flattening mechanisms during sheet metal forming, which tend to increase the real area of contact, are flattening due to normal loading, flattening due to sliding and flattening due to combined normal loading and deformation of the underlying bulk material. Roughening of the surface, observed during deformation of the bulk material without applying a normal load [9,10], tends to decrease the real area of contact. Most of the theoretical models describing the flattening behavior of asperities continue the pioneering work of Greenwood and Williamson [11], who proposed an elastic contact model that accounts for a stochastic description of rough surfaces. Over recent years, modifications have been made to this model to account for arbitrarily shaped asperities [12], plastically deforming asperities [13,14], the interaction between asperities [12,15] and the influence of stretching the underlying bulk material [16,17]. Hol et al. [4,18] adopted a multi-scale approach in which the afore mentioned friction mechanisms have been combined.

In the thin film and thick film lubrication regime the contact load is carried completely by the lubricant. Friction in the thick

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film lubrication regime relies on the rheological properties of the lubricant, and follows from viscous shear stresses at the fluid–solid interfaces. The hydrodynamic pressure distribution within the lubricant can be calculated from the Reynolds equation. The basic differential equations governed by the Reynolds equation was first derived by Reynolds in [19] for incompressible fluids, and formed the foundation for calculations of thick film lubricated systems. The Reynolds equation can be derived from the Navier–Stokes equations and continuity equations if assumed that the lubricant flow is laminar, the film thickness h is thin compared to the contact zone, the lubricant is Newtonian, zero-slip occurs at the fluid–solid interfaces and that surfaces are smooth. In the thin film lubrication regime, the Reynolds equation loses its applicability due to the influence of surface roughness on fluid flow. Patir and Cheng [20,21] proposed an averaged form of the Reynolds equation which considers the combined roughness of mating surfaces. The averaged Reynolds equation describes the average pressure and shear driven flow between rough surfaces by using the so-called flow factors. Flow factors can be obtained from numerical flow simulations [20,22] which inspired many researchers to derive flow factor expressions applicable in the thin film and the mixed lubrication regime.

In the mixed lubrication regime, part of the load is carried by contacting surface asperities and part of the load is carried by the lubricant. Friction mechanisms active in both the boundary lubrication regime and thin film lubrication regime act simultaneously in this regime [5]. Surface changes due to asperity deformations influence the load-carrying capacity of the lubricant [6,5]. As for the thin film lubrication regime, the averaged Reynolds equation can be used to calculate the load-carrying capacity of the lubricant [23–25].

Most of the existing methods to solve the (averaged) Reynolds equation are derived for a particular forming process. For these cases, external process parameters like e.g. tooling radii, sheet thickness and size of the lubrication zone appear explicitly in the lubrication equations. Moreover, the lubrication analysis is generally split into an inlet zone, controlling the formation of the lubricant film, and a working zone, describing lubricant flow underneath the tooling of the forming process. This approach was first introduced by Wilson and Wang [26]. Wilson and Wang demonstrated their approach by a hydrodynamically lubricated stretch forming process. The same approach has been adopted to describe lubricated strip-rolling processes in Wilson and Sheu [27], hemispherical punch stretching in Martinet and Chabrand [28] and axisymmetric cup drawing in Karupannasamy et al. [29]. For 2D forming processes, this approach has proved its applicability, however, the applicability is bound by the process for which the method is developed and becomes extremely complicated for 3D forming processes.

In order to describe lubricant flow during 3D non-stationary forming processes, a more general formulation is required. A reliable approach can be obtained by making a coupling between the (already existing) FE mesh of a forming simulation and a discretized form of the (averaged) Reynolds equation. Solving the Reynolds equation by adopting an FE approach finds its first application in bearing mechanics, see for example Booker and Huebner [30]. Hu and Liu [31] first adopted this approach to describe a thick film lubricated steady-state strip rolling process. They solved the Reynolds equation by using an arbitrary Lagrangian Eulerian (ALE) formulation of the Reynolds Equation. Yang and Lo [32] applied the FE method to solve thin film lubrication problems during axisymmetric cup stretching. Boman and Ponthot [33] proposed a more general form of the (ALE) FE formulation of the averaged Reynolds equation, making the description of arbitrary forming processes possible. As for the other cases, Boman and Ponthot considered contact conditions occurring in the thick film and/or thin film lubrication regime, excluding possible mixed lubrication friction during forming.

In this paper a friction model is proposed that can describe friction in the mixed lubrication regime. A coupling between a hydrodynamic friction model and a boundary lubrication friction model has been made for this purpose. The hydrodynamic friction model comprises the averaged Reynolds equation and viscous shear terms to equate hydrodynamic stresses at the solid–fluid interfaces. To solve the averaged Reynolds equation a FE interface element has been developed including pressure degrees of freedom. The interface element describes fluid flow between the deformed sheet material (described by a Lagrangian FE formulation) and the dies. The amount of asperity deformation of contacting surface asperities has been used to obtain the required fluid film thickness h. In this respect, the fluid film thickness h describes the key component in making the coupling between the hydrodynamic and boundary lubrication friction model. To obtain shear stresses between contacting asperities and the amount of asperity deformation the boundary lubrication friction model proposed by Hol et al. [4,18] has been used. By following this approach, friction in the boundary lubrication regime and mixed lubrication regime can be described. No explicit friction coefficients have to be specified. The friction force solely relies on physical based parameters of the material and the lubricant. The final part of this paper demonstrates the predicting capabilities of the friction model by two forming simulations.

2. Boundary lubrication friction model

The boundary lubrication friction model as described by Hol et al. in [4,18] comprises three stages. In the first stage, the input step, surface characteristics and material properties are defined. 3D surface textures of both tool and workpiece are read-in to characterize surface properties and to determine stochastic variables. Stage 2, the flattening step, the effect of surface changes due to normal loading, straining the underlying bulk material and sliding is accounted for. The last stage, the friction step, the influence of ploughing and adhesion on the shear stresses is calculated. A brief description of the boundary lubrication friction model will be provided in this section, for a detailed description of the friction model the reader is referred to [4,18].

2.1. Modeling the deformation behavior of rough surfaces

The models described in [4,18] provide expressions for the fractional real area of contact, and are based on a stochastic description of a rough workpiece surface in contact with a flat tool surface. This is considered a valid assumption as the tool surface in sheet metal forming processes is in general much harder and smoother than the workpiece surface. A non-linear work-hardening normal loading model is adopted which is based on energy and volume conservation laws. Asperity flattening due to combined normal loading and deformation of the underlying bulk material has been described by the flattening model proposed by Westeneng [12]. The increase in real contact area due to sliding is captured by adopting the junction growth theory as proposed by Tabor [34].

The asperities of the rough surface are described by bars that can represent arbitrarily shaped asperities. Stochastic parameters have been introduced to make an efficient translation from micro to macro contact modeling, i.e. the normalized surface height distribution function of the rough surface $\phi(z)$, the uniform raise of the non-contacting surface U (based on volume conservation) and the separation between the tool surface and the mean plane of the asperities of the rough surface d, see Fig. 1.