Available online at www.sciencedirect.com

ScienceDirect

Procedia Manufacturing 47 (2020) 586–590

Procedia MANUFACTURING

www.elsevier.com/locate/procedia

23rd International Conference on Material Forming (ESAFORM 2020)

Modeling Mixed Lubrication Friction for Sheet Metal Forming Applications

Meghshyam Prabhakar Shisode[a,*], Javad Hazrati[a], Tanmaya Mishra[b], Matthijn de Rooij[b], Ton van den Boogaard[a]

[a]*Nonlinear Solid Mechanics, Faculty of Engineering Technology, University of Twente, Enschede 7522 NB, The Netherlands*
[b]*Surface Technology and Tribology, Faculty of Engineering Technology, University of Twente, Enschede 7522 NB, The Netherlands*

* Corresponding author. Tel.: +31(0)534894335. *E-mail address: m.p.shisode@utwente.nl*

Abstract

Accurate modeling of friction is important in Finite element (FE) analyses of forming processes. Friction behavior depends on various parameters such as local contact load, surface topographies of sheet metal and tool, their material properties and the lubrication condition. In a typical deep drawing process, mixed lubrication condition is common, meaning that a lubricant can influence the coefficient of friction. Friction in the mixed lubrication regime is governed by the direct asperity contact as well as the hydrodynamic pressure developed by the lubricant. Local hydrodynamic pressure is also influenced by surface topography in addition to the lubricant amount and other process parameters. Direct numerical implementation of a measured surface topography in FE simulations is impractical due to the enormous computational effort. In this study, the overall frictional behavior in mixed lubrication regime is determined with the main objective to incorporate real measured surface topography in an efficient manner. An average Reynolds equation is solved on global FE domain of the forming simulation to determine lubricant pressure. A coupled friction model combining the effects of lubricant pressure and direct asperity contact is implemented in the forming simulations.

Keywords: Mixed Lubrication; Average Reynolds Equation; Flow Factor; Friction; Deep Drawing

1. Introduction

In a typical sheet metal forming application, the sheet surface is lubricated. The lubricant can potentially change the contact condition between the tool and the sheet surface from boundary to the mixed lubrication regime. The lubrication condition depends on the lubricant amount, loading conditions and other process parameters. The boundary lubrication regime is the regime where the total contact load between the tool and the sheet surface is completely carried by the direct asperity contact whereas in the mixed lubrication regime, the load is shared by direct asperity contact and the pressurized lubricant in the surface pockets. The Reynolds equation is often solved to determine the hydrodynamic pressure distribution and the viscous shear stresses developed in the lubricant to account for better prediction of frictional behavior.

The major challenge in the numerical simulations of rough surfaces is the smaller scale compared to the global domain. Hence the direct method with the sufficiently fine mesh to account for the surface roughness is almost impossible. Other methods, known as indirect or stochastic methods, are gaining attention. All these methods follow averaging on the solution domain. Patir and Cheng [1] introduced an average form of Reynolds equation (P&C method) applicable to any general surface roughness structure. The proposed P&C method determines the mean fluid pressure using a set of correction factors called “flow factors”. However, this formulation has its limitations especially under the condition of high fractional real

10.1016/j.promfg.2020.04.180

area of contact between the contacting surfaces but has been widely accepted for lightly loaded and full film lubrication regime. Wilson and Marsault [2], recently proposed a similar approach which can be used to determine the flow factors under high fractional real area of contact and presented relations to determine flow factors for Gaussian surfaces.

In deep drawing applications, both boundary and mixed lubrication regimes can exist at the same time at different locations in the die e.g. because of uneven lubricant distribution, tool roughness, loads and relative sliding speed between the tool and the workpiece. In the present study, a friction model applicable to the mixed lubrication regime is proposed. The surface roughness or texture effects are implemented in the formulation by using an average form of Reynolds equation [2]. In the FE analyses, pressure and shear flow factor relations proposed by Wilson and Marsault [2] are used. A boundary friction model proposed by Hol [3] is used to determine the shear stress between contacting asperities at the tool-workpiece interface. The solution from the boundary friction model is coupled with the hydrodynamic friction model. The total shear stress at the tool-workpiece interface is determined from the contribution of direct asperity contact and the solution of average Reynolds equation which accounts for the surface texture effects on the hydrodynamic pressure developed. The formulation is implemented in FE software to perform forming simulations.

2. Formulation of average Reynolds equation

Patir and Cheng [1] proposed an average form of the Reynolds equation to include the roughness effects. However, the P&C method is stable for non-contacting and lightly loaded surfaces. This formulation becomes unstable for higher real areas of contact. Wilson and Marsault [2] proposed a similar formulation to be used for the analysis of mixed lubrication in a rolling process under the high real area of contact. They proposed an alternative form of average Reynolds equation by modifying the existing P&C formulation as follows.

$$\nabla.\left(\frac{h_{avg}^3}{12\eta}\phi_P \nabla\bar{P}\right) = \nabla.\left(\frac{h_{avg}(U_1+U_2)}{2} + \phi_s \frac{S_q(U_1-U_2)}{2}\right) + \frac{\partial h_{avg}}{\partial t} \quad (1)$$

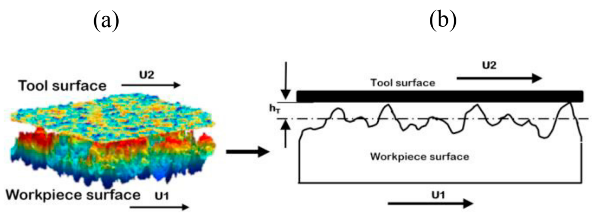

Fig. 1. Tool-workpiece surfaces in contact.

Here, $\nabla = \left(\frac{\partial}{\partial x}, \frac{\partial}{\partial x}\right)$, $\bar{P}\,(x,y)$ is the expected average hydrodynamic fluid pressure, ϕ_P and ϕ_s are pressure and shear flow factor tensors, h_{avg} is the average fluid film thickness, U_1 and U_2 are surface velocities and η is the dynamic viscosity of the fluid. h_{avg} is the area averaged cavity volume between the contacting surfaces which are nominally separated by h_T. S_q is the RMS composite roughness of surfaces. Both, ϕ_P and ϕ_s are 2nd order tensors with off-diagonal terms equal to 0 for isotropic or random surfaces [2, 4]. It should be noted that for smooth parallel surfaces where $h_{avg} \gg S_q$, the average Reynolds equation (1) reduces to the standard form of Reynolds equation for $\phi_P = [I]$ and $\phi_S = [0]$. The pressure flow factors reflect the impedance of flow due to roughness when the flow is driven by a macro-scale pressure gradient. The shear flow factor can be interpreted as the measure of additional fluid carried by rough surface valleys due to the relative sliding of contacting surfaces. In this study, a random surface with gaussian height distribution is used. Therefore, it is assumed that the off-diagonal terms in ϕ_P and ϕ_s are zero and the flow factors are determined from the relations proposed by Wilson and Marsault [2] for a random gaussian surface. For simplicity, it is assumed that the tool surface is smooth. The assumption is valid for the applications such as the deep drawing processes where the tool roughness is much smaller compared to the sheet surface. So, the pair of rough surfaces (see Figure 1a) is reduced to the system shown in Figure 1b. The flow factors used in this study are shown in Figure 2.

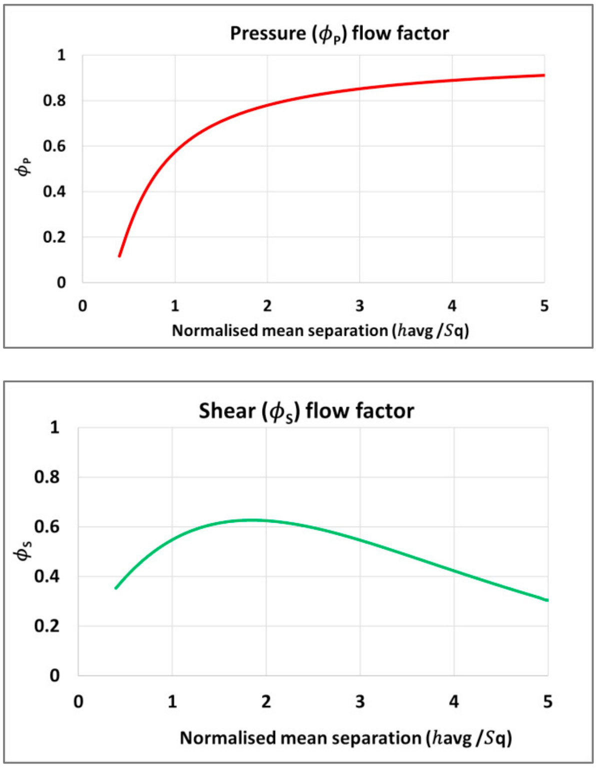

Fig. 2. pressure and shear flow factor [2] for a random gaussian surface.

The calculated flow factors are plotted as a function of normalized mean separation h_{avg}/S_q. The pressure flow factor ϕ_P approaches unity for higher mean separation. This is because for higher mean separation, the flow becomes smoother reducing the influence of roughness on the flow. The shear flow factor ϕ_S approaches rather slowly to zero as

h_{avg}/S_q increases. There is a rise in ϕ_S as h_{avg}/S_q decreases up to a certain value of h_{avg}/S_q and then decreases rapidly. This is due to the increasing number of contacts at lower h_{avg}/S_q. Contact points permit no flow and hence the shear flow factor decreases because of the sudden change in the available flow area.

3. Mixed lubrication friction model

In the mixed lubrication regime, the total friction force depends on the contribution from direct asperity contact and the viscous shear stress developed due to the pressurized lubricant at the tool-workpiece surface. Hol et. al [3] implemented a boundary lubrication friction model applicable to deep drawing processes. Hol [3] uses analytical and semi-analytical flattening models to determine the deformed state of the workpiece for different loading conditions present in the deep drawing application. Single asperity level friction model proposed by Challen and Oxley is adopted at macro-scale to determine the shear force contribution from each contact patch at tool-workpiece interface. The overall coefficient of friction is determined by summing up the contribution from each contact patch. The model is validated by using lab-scale deep drawing experiments. In this study, the boundary friction model proposed by Hol et. al [3] is used to determine the shear force contribution from direct asperity contact. Furthermore, average Reynolds equation (1) is solved in global FE domain to determine pressure developed in lubricant.

The boundary friction model is coupled with the solution of the hydrodynamic pressure to determine the overall coefficient of friction in mixed lubrication regime. The model is implemented in FE to be used for forming simulations. The inputs required to solve the average Reynolds equation in the global domain are, the lubricant properties, velocities of tool and workpiece, h_{avg}, ϕ_P and ϕ_S at each node of the FE mesh. The local contact loads (contact pressure and equivalent strain) at each node are extracted from FE contact algorithm. Using the flattening model [3], a deformed state of the workpiece is determined from which the deformed state of the surface, h_{avg}, ϕ_P and ϕ_S are determined. An offline study is performed to determine the relation between the contact loads and h_{avg} by running the flattening models at different combinations of contact loads (nominal contact pressure and equivalent strain). A look-up table is generated which includes parameters h_{avg}, ϕ_P , ϕ_S, fractional real area of contact and shear stress contribution from direct asperity contact [3] at range of nodal contact loads. This information is used in FE analysis at each node to solve average Reynolds equation in a global domain and to determine the viscous shear stresses. A nodal viscous shear stress is determined using the lubricant pressure field which is then coupled with the shear stress from direct asperity contact to determine the overall coefficient of friction.

A coupling approach proposed by Hol et. al [5] in the FE software code is adopted. In mixed lubrication regime, the total contact pressure is shared between solid to solid direct asperity contact (P_{solid}) and the lubricant pressure (P_{lub}) developed at tool-workpiece interface. P_{lub} at each node in each iteration is determined by solving the average Reynolds equation. For this purpose, an interface contact element [5] with additional pressure degrees of freedom is used. P_{solid} at node is determined by subtracting P_{lub} from total contact pressure P_{nom} determined from the contact algorithm in the FE software.

$$P_{solid} = P_{nom} - P_{lub} \tag{2}$$

It should be noted that P_{lub} is used from the previous increment to determine P_{solid} for the current increment. The nodal contact pressure P_{nom} and equivalent strain ε_{eq} are read from the FE results. The local shear stress, τ_{solid} at a node due to the direct asperity contact is determined from boundary friction model at nodal P_{solid} and ε_{eq}. The viscous shear stress τ_{lub} at each node is obtained from the lubricant pressure field determined by solving the average Reynolds equation. The Newtonian viscous shear stress developed at workpiece-fluid interface $\tau_{lub}\ (\tau_{xz}, \tau_{yz})$ is determined as,

$$\tau_{lub} = \eta \frac{(\nu_2 - \nu_1)}{h_{avg}} - \frac{h_{avg}}{2} \nabla P_{lub} \tag{3}$$

The resultant viscous shear stress τ_{LUB} at workpiece-fluid interface is,

$$\tau_{LUB} = \sqrt{{\tau_{lub_{xz}}}^2 + {\tau_{lub_{yz}}}^2} \tag{4}$$

Where, $\tau_{lub_{xz}}$ and $\tau_{lub_{yz}}$ are the in-plane nodal viscous shear stresses at workpiece-fluid interface. To determine the overall local shear stress τ_{tot} at FE node, the resultant viscous shear stress τ_{LUB} at workpiece-fluid interface and shear stress from direct asperity contact τ_{solid} are used.

$$\tau_{tot} = \alpha_{sol}\tau_{sol} + \alpha_{lub}\tau_{LUB} \tag{5}$$

where, $\alpha_{lub} = 1 - \alpha_{sol}$, is the nodal fractional fluid contact area and α_{sol} is the nodal fractional real contact area which is read from the look-up table at a given contact loads. Finally, the local nodal coefficient of friction μ_{tot} combining the effects of direct asperity contact and lubricant pressure is determined using τ_{tot} and P_{nom}.

$$\mu_{tot} = \frac{\tau_{tot}}{P_{nom}} \tag{6}$$

4. Application to deep drawing processes

A deep drawing simulation of a top hat is chosen to demonstrate the applicability of the developed mixed lubrication model. The geometry of the product and the tools are shown in Figure 3. A topography of the workpiece and the tool surfaces measured using confocal microscopy is used. Here, a measured tool surface at punch corner radius is used for the demonstration. Figure 4 shows the 3D surface height data and its distribution for a workpiece and tool surface. The measured tool roughness (S_q=0.59 µm) is much less than the workpiece (S_q=1.8 µm). Due to symmetry, only a quarter of the geometry is modeled. The tooling is modeled as rigid contours. The workpiece blank is meshed with triangular

Kirchhoff shell elements using 3 integration points in the plane and 11 integration points in the thickness direction. Interface contact elements are attached on the top and bottom of the shell elements. Interface elements are used to simulate the structural contact behavior between the tool and workpiece and to solve the average Reynolds equation. The squeeze term $\frac{\partial h_{avg}}{\partial t}$ in the Reynolds equation is neglected in this study because a quasi-static situation is assumed during the top hat forming. Details of the simulation and process parameters can be found in Table 1. The Vegter model [6] is used to describe the yield surface of the workpiece material along with Bergström–Van Liempt hardening relation [7]. Vegter yield parameters and Bergström–Van Liempt hardening parameters for steel DC06 can be found in [3].

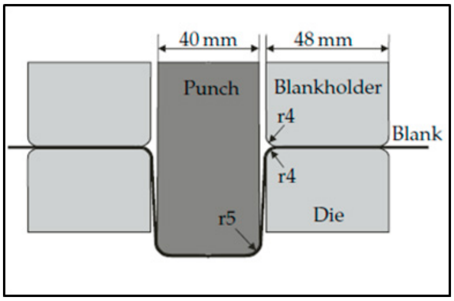

Fig. 3. Top hat geometry for FE simulation.

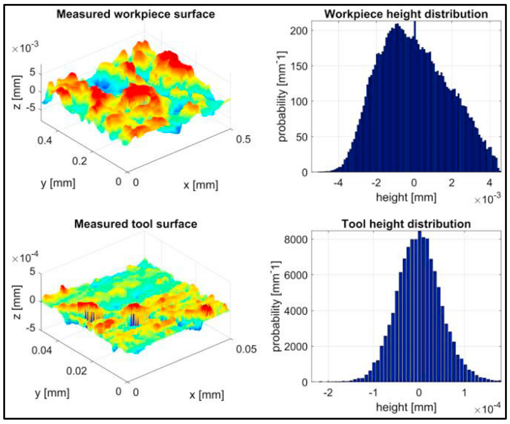

Fig. 4. Measured workpiece and tool surface height distribution.

FE simulations are performed at 0.6g/m^2 and 2.0g/m^2 lubricant amount. The lubricant amount and the type Reynolds equation used in the simulations are shown in Table 2. All other simulation parameters are same for all the simulations as shown in Table 1. For each simulation, a punch-force displacement curve is extracted for comparison. 0.6g/m^2 lubricant amount is too low to fill up the cavities of rough workpiece for the available loading conditions. Hence, there is no lubricant pressure developed for this case as validated from FE simulation results. Therefore, the hydrodynamic part is irrelevant and μ_{tot} is completely governed by boundary lubrication. For the simulation at lubricant amount of 2.0g/m^2, a lubricant pressure is developed as seen in FE results in Figure 5. The simulation with 2.0g/m^2 is performed with and without the roughness effects to compare the effect of roughness on the simulation results such as punch force-displacement curve and lubricant pressure. For the simulation without roughness effects, the flow factors, $\phi_P = [I]$ and $\phi_S = [0]$ are set which reduce the average Reynolds equation to standard Reynolds equation (1). Figure 5 shows the total nominal pressure P_{nom}, lubricant pressure developed P_{lub} and overall coefficient of friction μ on the workpiece-die side.

Table 1. Parameters for FE simulations.

Parameter	
Blank material	Steel DC06
Lubricant and amount	Quaker FERROCOAT N6130
Blank geometry	300 x 25 x 0.8 mm
Lubricant viscosity	23mPas (@ 40 deg)
Punch velocity	50 mm/s
Blank holder force	25 KN
Punch stroke	75mm

Table 2. FE simulations.

Simulation #	Std. Reynolds (BL)	Std. Reynolds (ML)	Avg. Reynolds (ML)
Lubricant amount	0.6g/m^2	2.0g/m^2	2.0g/m^2
Reynolds equation	Standard, $\phi_P = [I]$, $\phi_S = [0]$ in equation (1)	Standard, $\phi_P = [I]$, $\phi_S = [0]$ in equation (1)	Standard, $\phi_P \neq [I]$, $\phi_S \neq [0]$ in equation (1)

As expected, P_{lub} is higher for the case of average Reynolds equation compared to standard Reynolds equation at a constant lubricant amount of 2.0g/m^2. This demonstrates the effect of roughness on hydrodynamic pressure and the overall coefficient of friction. As per equation (2), the higher the P_{lub}, the lower is the direct asperity to asperity contact pressure P_{sol}. Hence, the overall coefficient of friction reduces. The difference in μ is clearly visible in the die corner region (see Figure 5). Figure 6 shows the punch force-displacement curves extracted from the simulation results. The punch force decreases with increase in lubricant amount. This is because of the shift in lubrication regime from boundary to mixed lubrication. It is also important to note that the punch force is lower for average Reynolds case compared to standard Reynolds case. This is because the roughness effects accounted in the solution of average Reynolds equation result a higher lubricant pressure. This decreases the overall coefficient of friction and punch force.

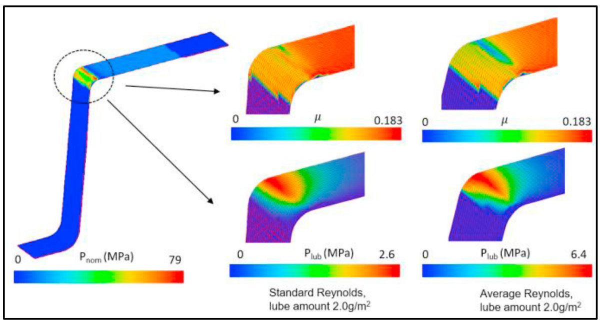

Fig. 5. Simulation results: total coefficient of friction (μ) and lubricant pressure (P_{lub}) on the workpiece-die side at 75mm punch displacement.

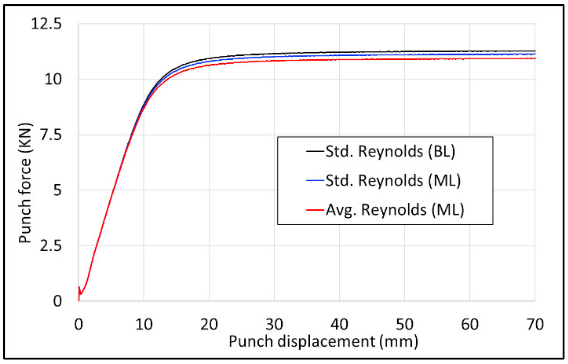

Fig. 6. Simulation results: punch force-displacement curve at different lubricant amount.

5. Conclusions

A mixed lubrication friction model allows to determine the local coefficient of friction based on local contact conditions and its resulting tribological system. For that purpose, a coupling is made between boundary friction and the hydrodynamic friction model. The boundary friction model determines the contribution of direct asperity contact between the measured tool and workpiece surface. The roughness effects are accounted for by using an average form of the Reynolds equation. The coupled friction model is implemented in FE software to be used for industrial scale forming simulations. A simple deep drawing simulation is performed using the proposed model. The simulation results clearly show the influence of the lubricant amount and surface roughness on the coefficient of friction. The future research will aim at validating the reliability of the model using lab-scale friction and deep drawing experiments.

Acknowledgements

This research was carried out under project number S22.1.14520b in the framework of the Partnership Program of the Materials innovation institute M2i *(www.m2i.nl)* and the Technology Foundation TTW *(www.stw.nl)*, which is part of the Netherlands Organization for Scientific Research (*www.nwo.nl)*.

References

[1] Patir N, Cheng HS. An Average Flow Model for Determining Effects of Three-Dimensional Roughness on Partial Hydrodynamic Lubrication. Journal of Lubrication Technology, 1978. 100(1): p. 12-17.

[2] Wilson W, Marsault N. Partial hydrodynamic lubrication with large fractional contact areas. Journal of tribology, 1998. 120(1): p. 16-20.

[3] Hol J, Meinders VT, de Rooij MB and van den Boogaard AH. Multi-scale friction modeling for sheet metal forming: The boundary lubrication regime. Tribology International, 2015. 81: p. 112-128.

[4] Tripp JH. Surface Roughness Effects in Hydrodynamic Lubrication: The Flow Factor Method. Journal of Lubrication Technology, 1983. 105(3): p. 458-463.

[5] Hol J, Meinders VT, Geijselaers HJM and van den Boogaard AH. Multi-scale friction modeling for sheet metal forming: The mixed lubrication regime. Tribology international, 2015. 85: p. 10-25.

[6] Vegter H, van den Boogaard AH. A plane stress yield function for anisotropic sheet material by interpolation of biaxial stress states. International journal of plasticity, 2006. 22(3): p. 557-580.

[7] van den Boogaard AH, Huétink J. Simulation of aluminium sheet forming at elevated temperatures. Computer methods in applied mechanics and engineering, 2006. 195(48-49): p. 6691-6709.