LUBRICATED CONTACT MODEL FOR COLD METAL ROLLING PROCESSES

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INTRODUCTION

Numerical model for calculating lubricated contact pressures and friction in cold metal rolling is presented in this study. Lubrication and friction are important factors in metal forming processes, since unoptimised frictional parameters can result in lower productivity of the rolling machinery and deteriorate surface quality of the product. In order to capture frictional effects during lubricated rolling simulations the presented numerical model was implemented as a boundary condition for a large strain hyperelastoplastic deformation solver. The model takes into account surface roughness effects, different lubrication regimes and lubricant properties variations. Simulation results of a 3D sheet rolling case are presented.

MATHEMATICAL MODEL

Lubricated contact between surfaces in relative motion can be divided into three regimes: hydrodynamic, mixed and boundary lubrication regime. In hydrodynamic regime two surfaces are completely separated by the lubricant. In mixed regime contact surface pressure is shared between asperities in contact and lubricant. In boundary lubrication regime major part of surface pressure is carried by asperities in contact. In order to take into account all three lubrication regimes, total contact pressure P_n is divided into two parts: asperity contact pressure P_a and hydrodynamic pressure P_f.

$$P_n = A P_a + (1 - A) P_f,$$

where A is the ratio of asperity contact area to the unit nominal area. Correspondingly, shear traction P_t is defined by:

$$P_t = A \tau_a + (1 - A) \tau_f,$$

where τ_a is asperity traction and τ_f is lubricant shear stress.
Solid-to-solid contact model used here is the **Greenwood-Williamson** (GW) contact model. GW is a statistical model based on the Hertzian contact theory where asperity deformation is considered elastic, and their shape is hemispherical. GW model defines the ratio of asperity contact area to the unit nominal area A:

$$A(d) = \pi N R \int_d^{\infty} (h - d)\, f(h)\, \mathrm{d}h,$$

where N is asperity density, R is average asperity radius, h is asperity height, d is mean distance between two reference surfaces, $f(h)$ is asperity height distribution (e.g. Gaussian). Asperity contact pressure is defined as:

$$P_a(d) = \frac{4}{3A(d)} N E' R^{1/2} \int_d^{\infty} (h - d)^{3/2} f(h)\, \mathrm{d}h,$$

where E' is a reduced modulus of two bodies in contact.
Lubricant hydrodynamic pressure is calculated using the modified **Reynolds equation** which takes into account roughness of surfaces in contact. Reynolds equation uses the following assumptions: fluid viscous forces dominate over fluid body, inertia and surface tension forces, i.e. the latter can be neglected; fluid film curvature can be neglected (thickness of the fluid is much smaller than the width and length of the film); the variation of pressure across the fluid film is negligibly small. Modified Reynolds equation is defined as:

$$\nabla_s \cdot \left(\phi_{xy} \frac{\rho d^3}{12\mu} \nabla_s p \right) = \nabla_s \cdot \left[\frac{\rho h_T (\mathbf{U}_1 + \mathbf{U}_2)}{2} \right] + \frac{\mathbf{U}_1 - \mathbf{U}_2}{2} R_q \nabla_s (\phi_s) + \frac{\partial (\rho d)}{\partial t},$$

where h_T is average film thickness, ϕ_{xy} and ϕ_s are pressure and shear flow factors, R_q is the composite surface roughness.

SHEET ROLLING CASE

A three-dimensional sheet rolling case with lubricated contact is presented. Seven simulations were performed where influences of three different parameters on contact pressures were observed: lubricant viscosity (0.5, 2, 5 Pas), sheet thickness reduction (10%, 20%, 30%) and roller speed (60, 120, 240 RPM). Length of the nondeformed sheet is 100 mm, width 16 mm and thickness 8 mm. Roller diameter is 158 mm and width 19.2 mm. Sheet material properties are: $\rho = 7800\ \mathrm{kg/m^3}$, $E = 177$ GPa, $\nu = 0.3$. Rollers are considered rigid.
Figure 2 depicts hydrodynamic pressure build-up at the inlet of the rolling bite, where the area of maximum total contact pressure is also located. Figure 3 shows that an increase of lubricant viscosity has a great effect on hydrodynamic pressure. In this case a viscosity increase from 0.5 to 2 Pas results in increase of hydrodynamic pressure by 400%, while an increase from 2 to 5 Pas increases hydrodynamic pressure by 140%. Changes of lubricant viscosity have almost no influence on asperity contact pressure, since hydrodynamic pressure is three orders of magnitude smaller than the asperity pressure.

Figure 1: Sheet rolling mesh and film thickness ($\mu = 0.5$ Pas, $R = 20\%$, $n = 120$ RPM)

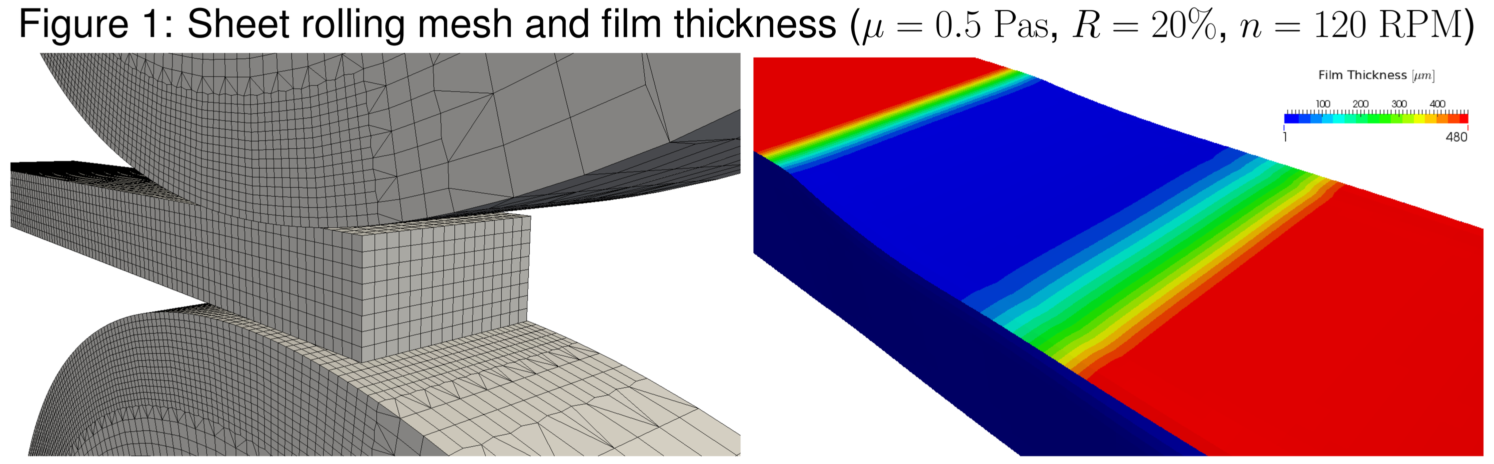


Figure 2: Hydrodynamic and total contact pressure ($\mu = 0.5$ Pas, $R = 20\%$, $n = 120$ RPM)

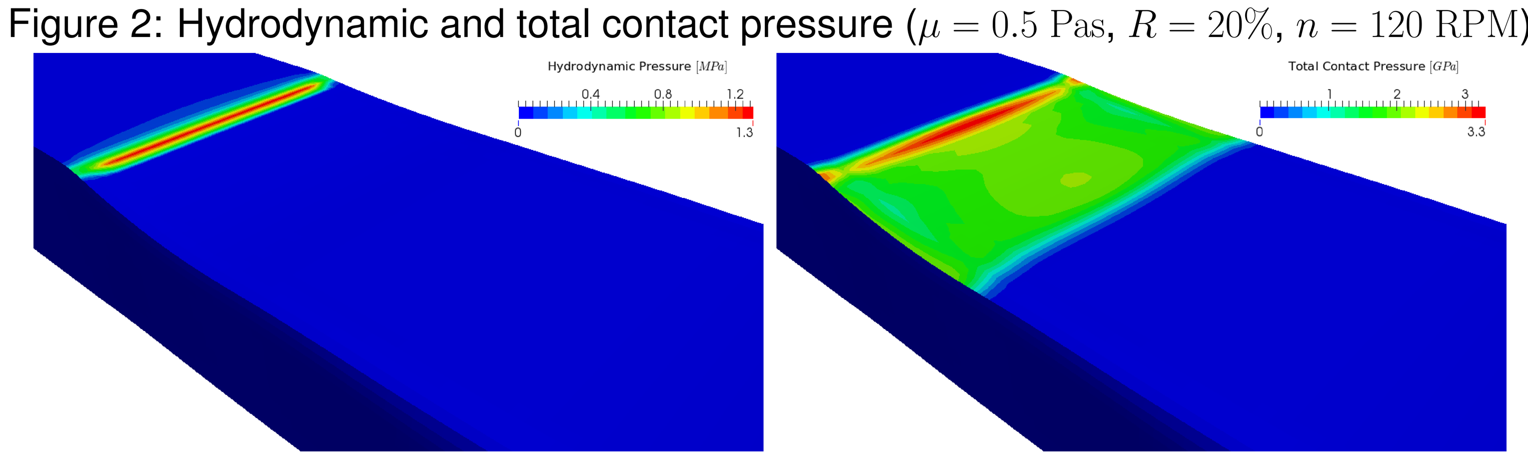


Figure 3: Influence of lubricant viscosity on hydrodynamic and asperity contact pressure ($R = 20\%$, $n = 120$ RPM)

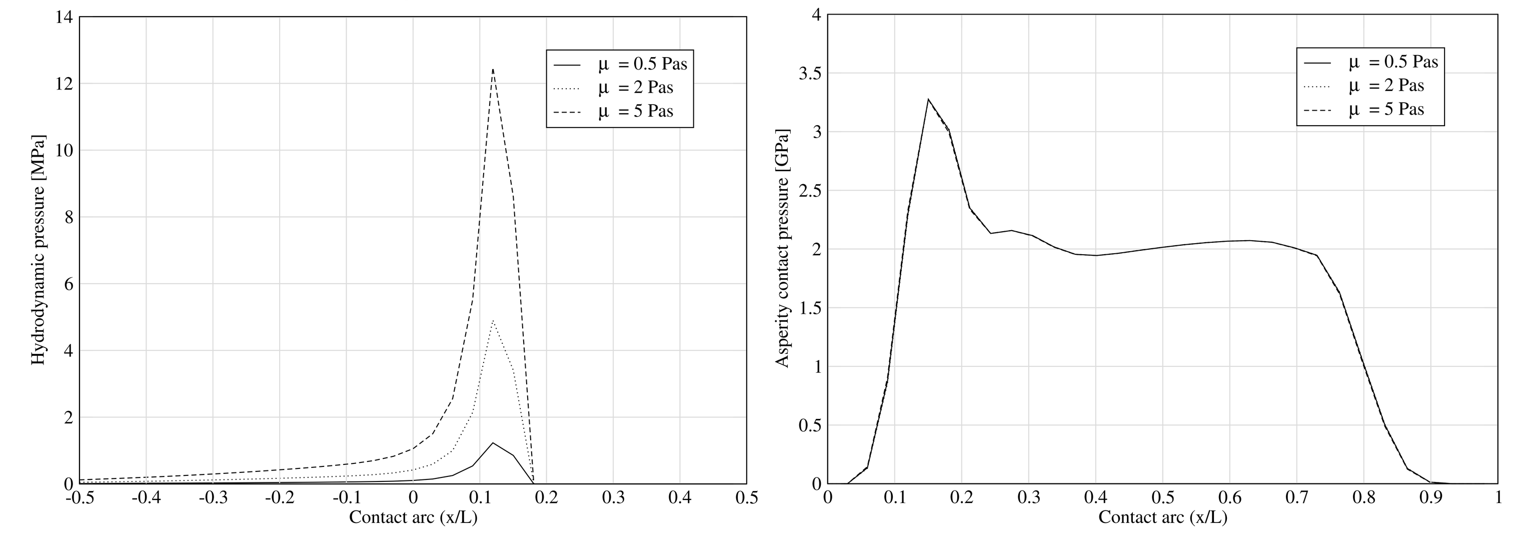


Figure 4: Influence of sheet thickness reduction on hydrodynamic and asperity contact pressure ($\mu = 0.5$ Pas, $n = 120$ RPM)

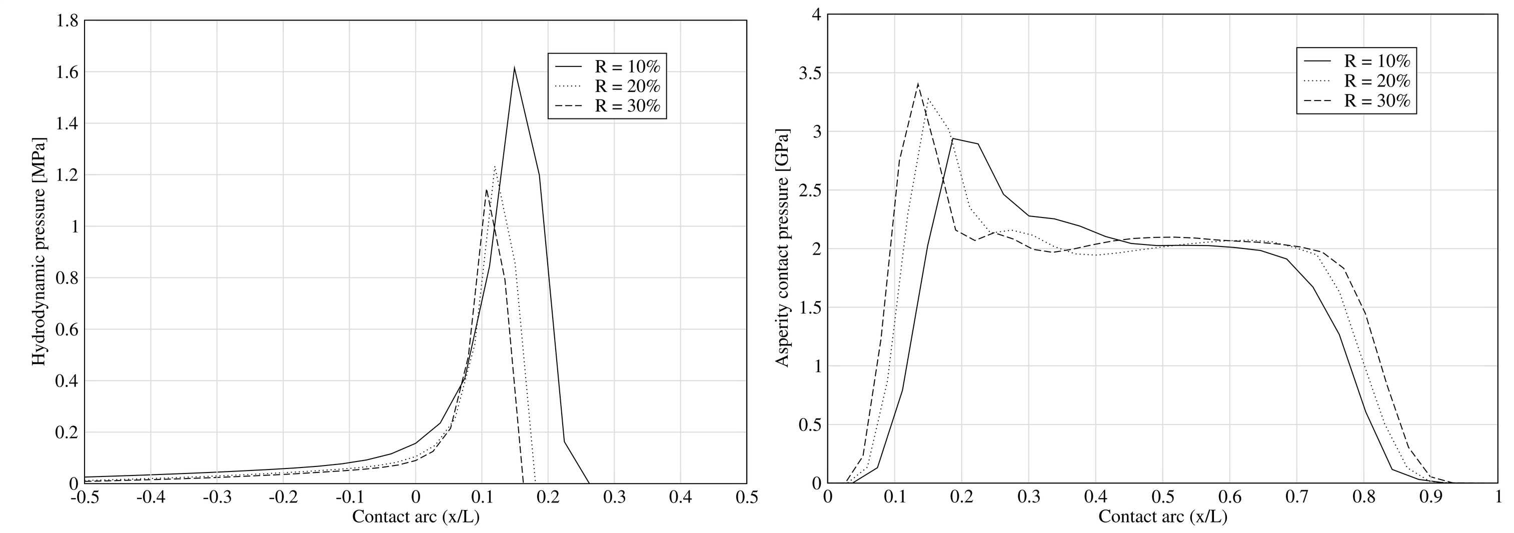
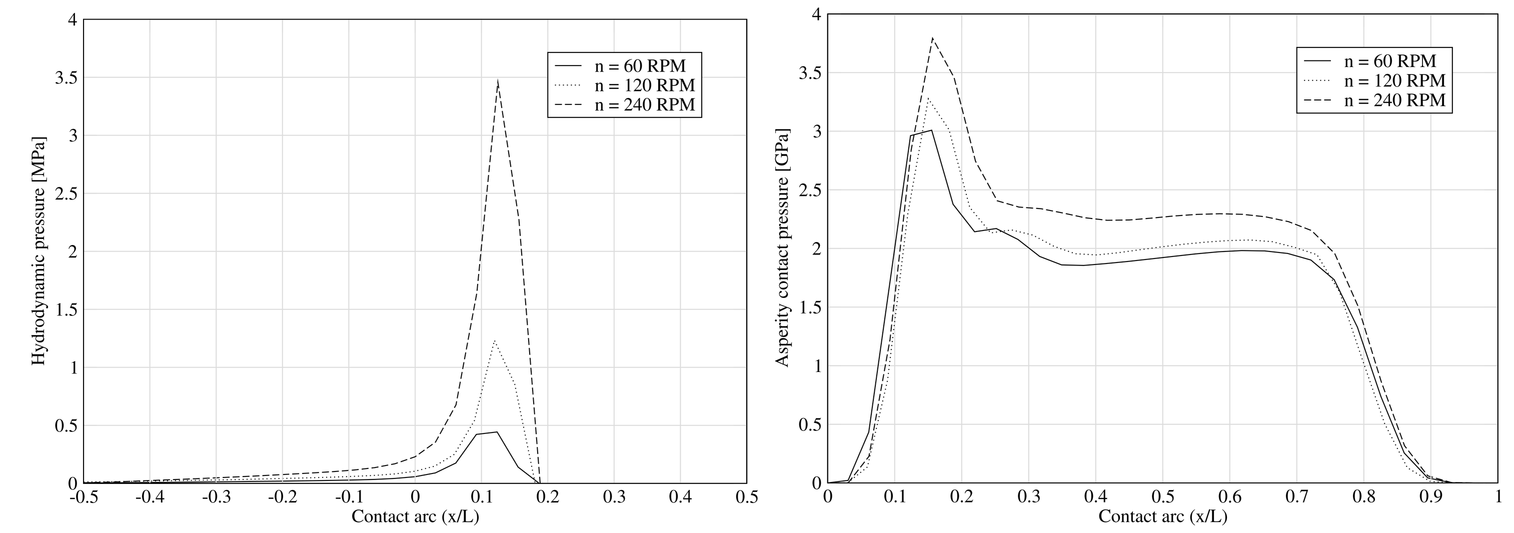


Figure 5: Influence of roller speed on hydrodynamic and asperity contact pressure ($\mu = 0.5$ Pas, $R = 20\%$)



By increasing sheet thickness reduction, Figure 4, maximum value of hydrodynamic pressure increases and shifts more into the rolling bite, while asperity contact area expands and its maximum value increases. Increse of rolling speed, Figure 5, by 100% increases hydrodynamic pressure by almost 300%. Asperity contact pressure is also increased in the whole area of contact.

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