Lubricated Contact Model for Cold Metal Rolling Processes

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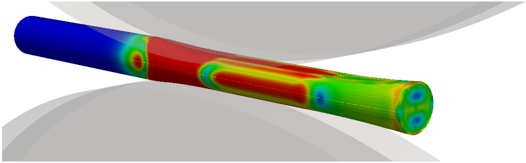
Overview

Motivation

- In wire manufacturing industry placing a new product on the market requires 5 years due to large number of experimental tests.
- In order to reduce the required time, the numerical modelling of manufacturing processes is utilized:
  - Reduces number of required experimental tests
  - Easier planning of the manufacturing process,
  - Insight on the material behaviour,
  - Product optimisation,
  - ...
- Development of a numerical package for metal forming based on `foam-extend` software package, in collaboration between Bekaert, UCD Dublin and UNIZAG FSB.

Research Topic

- Development of a reliable and efficient numerical model for lubricated contact taking into account:
  - Hydrodynamic, mixed and boundary lubrication regimes,
  - Contact surface topology (roughness),
  - Changes of lubricant rheology properties due to temperature and pressure changes,
  - Heat transfer.
- Improving contact detection efficiency
- The contact models should be used as boundary conditions for a large strain hyperelastoplastic deformation solver developed by Cardiff et al. [1]



Lubricated Contact in Metal Forming

- Lubricated contact between surfaces in relative motion can be divided into three regimes [2]:
  1. **"Thick" film hydrodynamic** regime ($z = d/R_q > 10$) where smooth surfaces are valid assumption (R_q is the composite surface roughness); **"Thin" film hydrodynamic** regime ($3 < z < 10$) where there is still no solid-solid contact, but asperities play important role in lubricant flow;
  2. **Mixed lubrication** regime ($1 < z < 3$) where surface pressure is shared between asperities in contact and lubricant;
  3. **Boundary lubrication** regime ($z < 1$) major part of surface pressure is carried by asperities in contact.
- Due to different lubrication regimes, both solid-solid and solid-fluid contact models have to be formulated and implemented, and coupled in an appropriate way in order to give a good representation of mixed lubrication regime.
- Heat generation and heat transfer effects along with the changes of lubricant properties have to be taken into account.

Lubricated Contact in Metal Forming

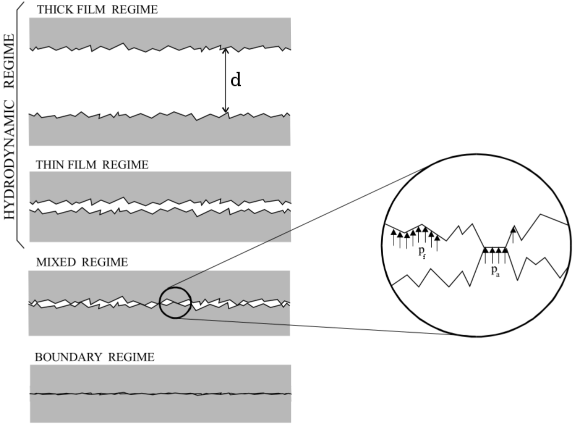


Figure 1 : Lubrication regimes [2]

Contact Coupling

- In mixed and boundary lubrication regimes normal surface contact pressure is shared between asperities in contact and lubricant:

$$P_n = AP_a + (1 - A)\, P_f$$

P_n - normal surface contact pressure
P_a - asperity contact pressure
P_f - lubricant pressure
A - ratio of asperity contact area to the unit nominal area

- Correspondingly, shear traction is defined by:

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

P_t - total surface shear traction
τ_a - asperity traction
τ_f - lubricant shear stress

Greenwood-Williamson Contact Model

- Based on the Hertzian contact theory - elastic deformation of asperities.
- One surface has a large number of asperities, while the other surface is taken as smooth.
- Asperities are considered to be hemispheres.
- Ratio of asperity contact area to the unit nominal area:

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

N - asperity density (approximate number of asperities per square meter)
R - average asperity radius
h - asperity height
d - mean distance between two reference surfaces
$f(h)$ - asperity height distribution, e.g. Gaussian

Greenwood-Williamson Contact Model

- Asperity contact pressure:

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

where reduced modulus E' of two contacting bodies (1 and 2) is equal to:

$$E' = \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2},$$

ν_1, ν_2 - Poissons's ratio
E_1, E_2 - Young's modulus

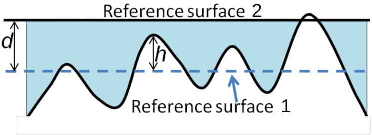


Figure 2 : Schematic of a GW contact [3]

Basic Reynolds Equation

- Partial differential equation governing the pressure distribution of thin viscous fluid films in the Lubrication theory.
- 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.
  - "Thick" lubricant film $\rightarrow z = d/R_q > 10$
- Derived from the Navier-Stokes equations of fluid motion:

$$\nabla_s \bullet \left(\frac{\rho d^3}{12\mu} \nabla_s p \right) = \nabla_s \bullet \left[\frac{\rho d \left(\mathbf{U}_1 + \mathbf{U}_2 \right)}{2} \right] + \frac{\partial \left(\rho d \right)}{\partial t}$$

d - distance between two surfaces
$\mathbf{U}_1$, $\mathbf{U}_2$ - tangential surface velocity
ρ - fluid density
μ - fluid dynamic viscosity

Modified Reynolds Equation

- Basic Reynolds equation is only valid for "thick" films, where surface roughness has very little or no effect on the lubricant flow.
- However, in order to simulate "thin" film, mixed and boundary lubrication regimes, where surface roughness plays important role, modification of the basic Reynolds equation is required.
- For that reason, Patir and Cheng [4] developed the *Modified Reynolds Equation*:

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

h_T - average film thickness
ϕ_{xy} - pressure flow factor [4]
ϕ_s - shear flow factor [4]
R_q - composite surface roughness

Modified Reynolds Equation

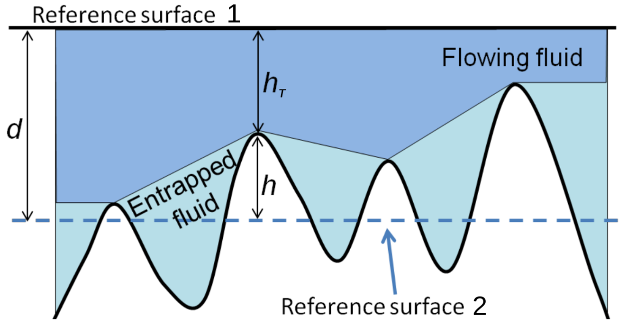


Figure 3 : Schematic of thin film contact [3]

Sheet Rolling Simulation

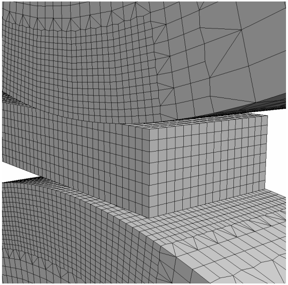


Figure 4 : Computational mesh

Case set-up:

- Sheet
  - $L = 100$ mm, $W = 16$ mm, $H = 8$ mm
  - $\rho = 7800$ kg/m^3, $E = 177$ GPa, $\nu = 0.3$
- Roller
  - $D = 158$ mm, $B = 9.6$ mm
  - Rigid body

- Lubricant viscosity: 0.5, 2, 5 Pas
- Sheet thickness reduction: 10, 20, 30%
- Roller speed: 60, 120, 240 RPM

Sheet Rolling Simulation

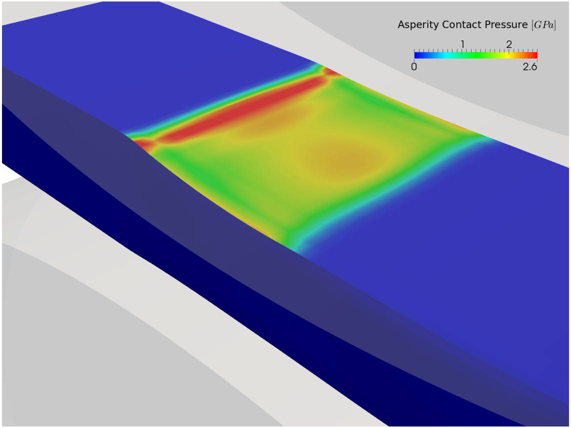


Figure 5 : Asperity contact pressure

Sheet Rolling Simulation

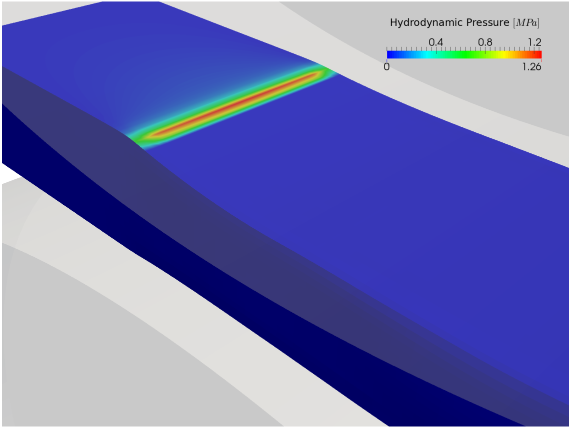


Figure 6 : Hydrodynamic pressure

Sheet Rolling Simulation

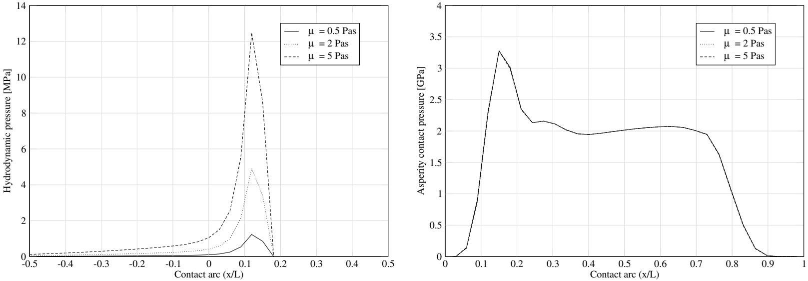


Figure 7 : Influence of lubricant viscosity on contact pressures ($n = 120$ RPM, $\delta = 20$ %)

Sheet Rolling Simulation

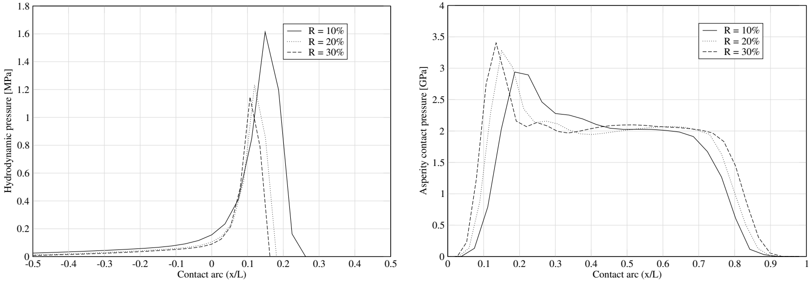


Figure 8 : Influence of sheet thickness reduction on contact pressure ($\mu_l = 0.5$ Pas, $n = 120$ RPM)

Sheet Rolling Simulation

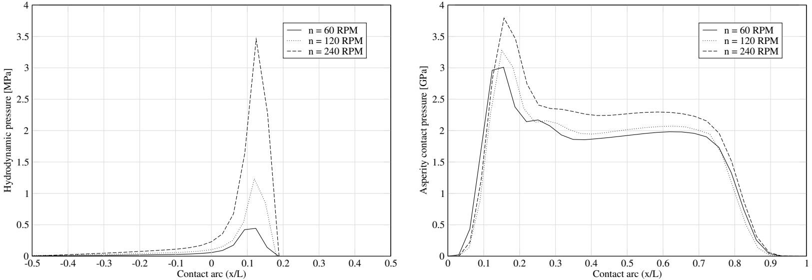


Figure 9 : Influence of rolling speed on contact pressure ($\mu_l = 0.5$ Pas, $\delta = 20$ %)

Wire Rolling Simulation

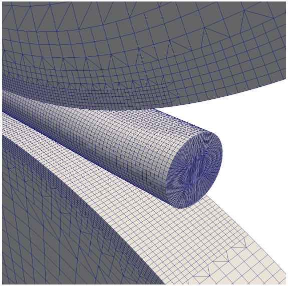


Figure 10 : Computational mesh

Case set-up:

- Wire
  - $L = 100$ mm, $D = 8$ mm
  - $\rho = 7800$ kg/m^3, $E = 177$ GPa, $\nu = 0.3$
- Roller
  - $D = 158$ mm, $B = 12$ mm
  - Rigid body

- Lubricant viscosity: 0.5, 2, 5 Pas
- Sheet thickness reduction: 10, 20, 30%
- Roller speed: 120, 240 RPM

Wire Rolling Simulation

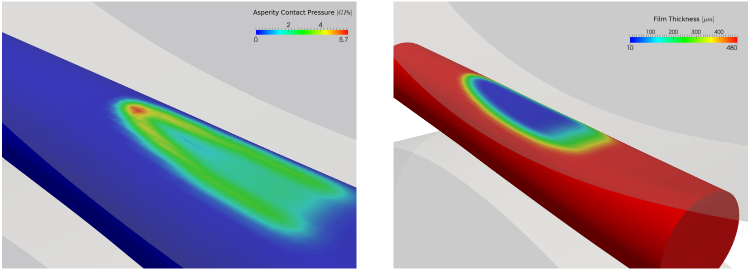


Figure 11 : Wire asperity contact pressure and film thickness

Wire Rolling Simulation

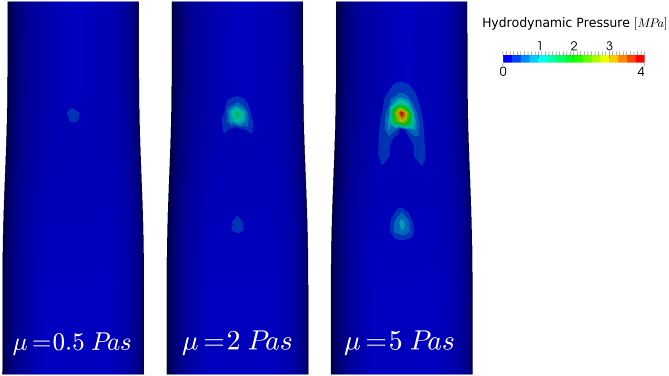


Figure 12 : Influence of lubricant viscosity on hydrodynamic pressure ($n = 120$ RPM, $\delta = 20$ %)

Wire Rolling Simulation

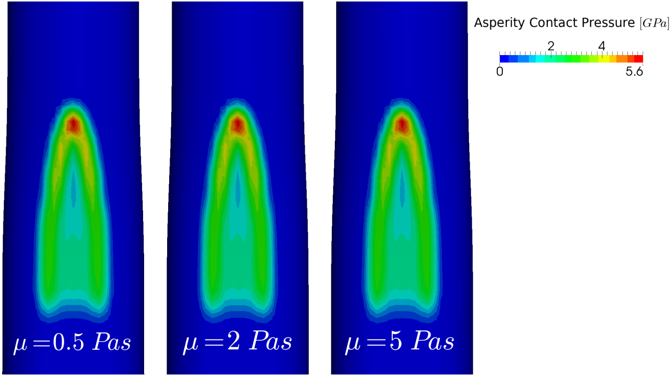


Figure 13 : Influence of lubricant viscosity on asperity contact pressure ($n = 120$ RPM, $\delta = 20$ %)

Wire Rolling Simulation

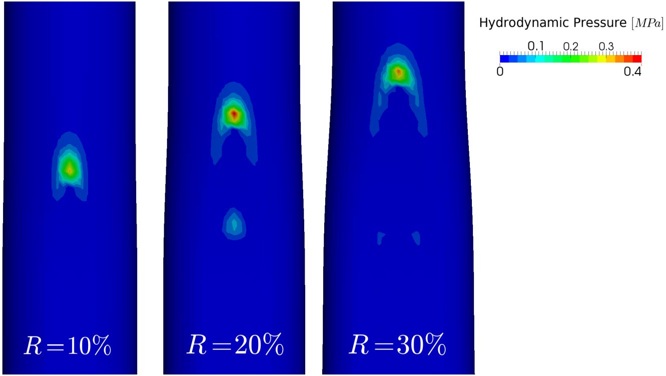


Figure 14 : Influence of wire thickness reduction on hydrodynamic pressure ($\mu_l = 0.5$ Pas, $n = 120$ RPM)

Wire Rolling Simulation

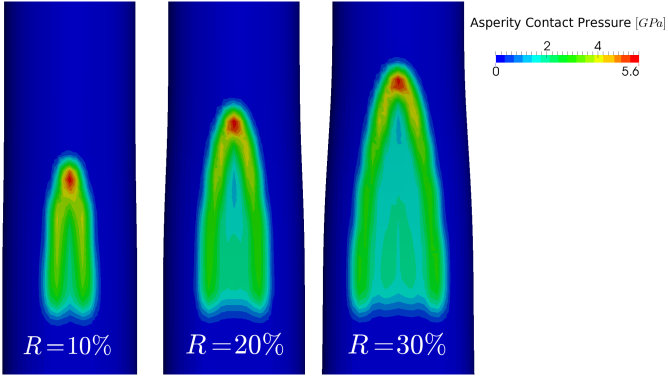


Figure 15 : Influence of wire thickness reduction on asperity contact pressure ($\mu_l = 0.5$ Pas, $n = 120$ RPM)

Wire Rolling Simulation

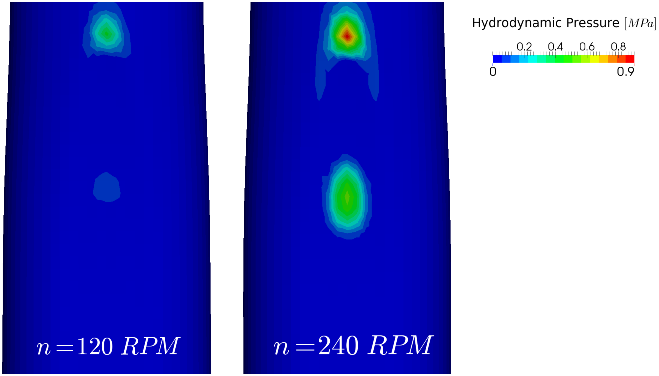


Figure 16 : Influence of rolling speed on hydrodynamic pressure ($\mu_l = 0.5$ Pas, $\delta = 20$ %)

Wire Rolling Simulation

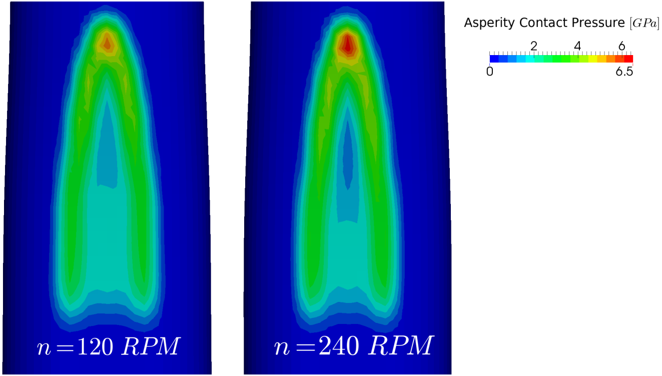


Figure 17 : Influence of rolling speed on asperity contact pressure ($\mu_l = 0.5$ Pas, $\delta = 20$ %)

Future Work and Implementation

- **Solid-solid contact model with elastoplastic asperity deformation**
  - Surface topology evolution.
  - Ability to input surface roughness measurements into the model.
- **Contact heat transfer**
  - *Solid-solid* and *solid-lubricant-solid* heat transfer.
  - Usual assumptions for hydrodynamic contact:
    1. Velocity and temperature gradients are only significant across the lubricant film and are negligible along the flow direction.
    2. Convection is neglected across the lubricant film, and conduction along the flow direction.
    3. Thermal conductivity is constant across the film.
- **Validation and verification of the models.**

References

[1] P. Cardiff, Ž. Tuković, P. D. Jaeger, and A. Ivanković, "A lagrangian cell-centred finite volume method for metal forming simulation," *International Journal for Numerical Methods in Engineering*, In preparation.

[2] R. Boman and J. Ponthot, "Finite element simulation of lubricated contact in rolling using the arbitrary lagrangian-eulerian formulation," *Comput. Methods Appl. Mech. Engrg.*, vol. 193, pp. 4323–4353, 2004.

[3] M. Khan, H. Ruan, L. Zhang, X. Zhao, and X. Zhang, "A new approach to the investigation of mixed lubrication in metal strip rolling," in *Proc. 7 Australasian Congress on Applied Mechanics, ACAM 7*, Adelaide, Australia, 2012.

[4] N. Patir and H. Cheng, "An average flow model for determining effects of three-dimensional roughness on partial hydrodynamic lubrication," *J. of Lubrication Tech.*, vol. 100(1), pp. 12–17, 1978.

Thank you for you attention.
Questions, please?