

Effects of non-Newtonian Lubricants on Surface Roughness in EHL Point Contacts

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Ph.D. Thesis Defense

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Institute of Machine and Industrial Design

Faculty of Mechanical Engineering

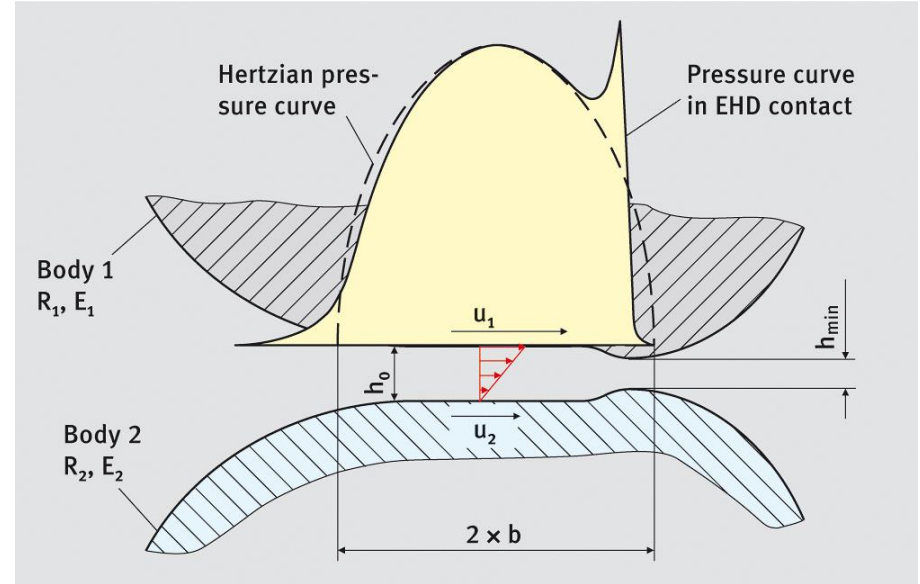
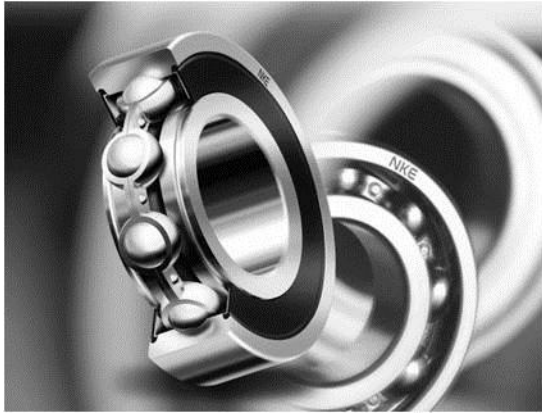
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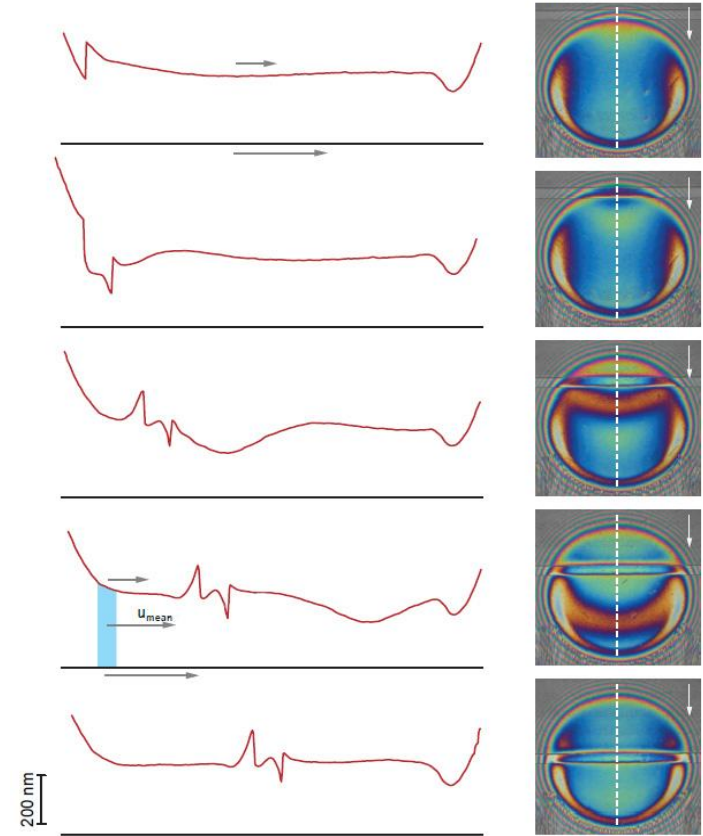
Elastohydrodynamic lubrication (EHL)

- high pressure
- + elastic deformations
- + small contact area
- = **complex problem requiring a detailed solution**

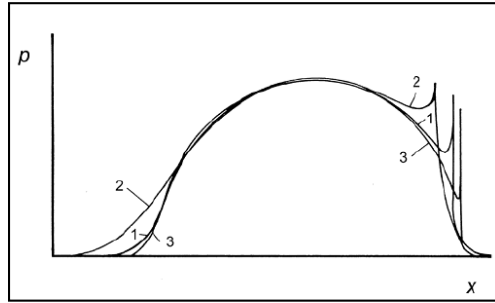


Roughness behavior in EHL contacts

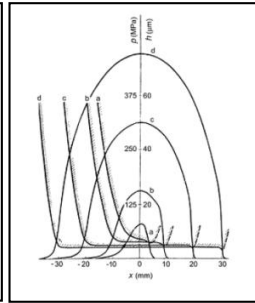
- behavior of deformed roughness inside the contact
- pressure variations which can lead to failures
- numerical simulations – approach to study these effects



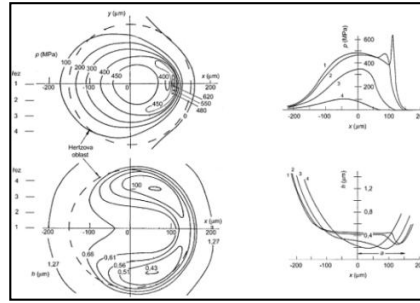
Numerical simulations



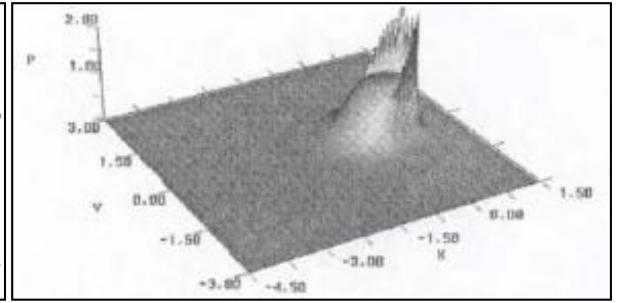
Petrusevich (1951)



Dowson et al. (1959)



Ranger et al. (1974)



Venner et al. (2000)



1st numerical solution

Line contact – direct solution

Line contact – inverse solution

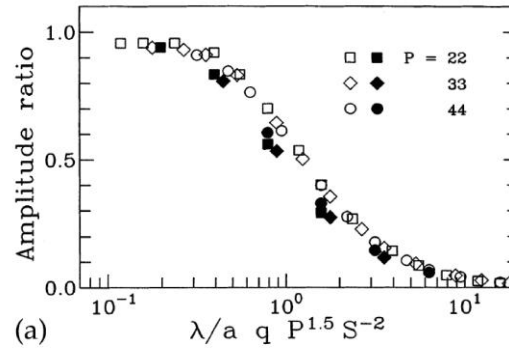
Point contact – inverse solution

Point contact – direct solution

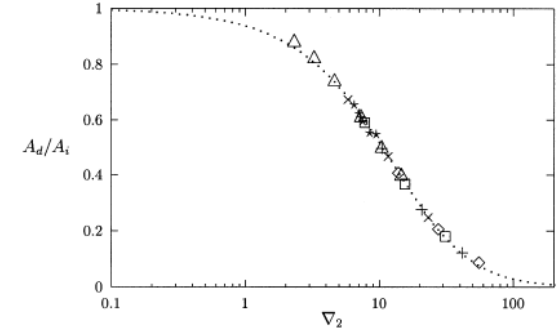
Multigrid methods

Study of roughness deformation

- predictive models – attenuation theory
 - Lubrecht, Venner et al. (1999)
 - Hooke et al. (2000)

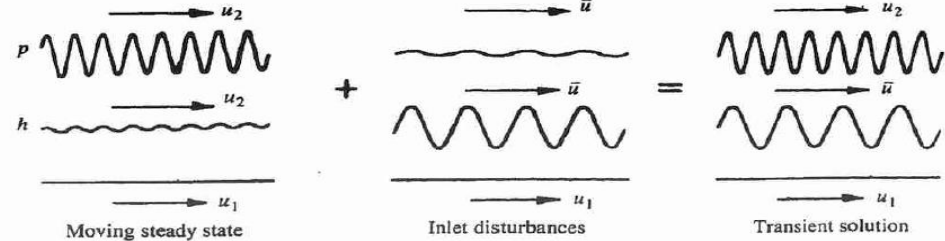


▪ Hooke et al. (2000)



▪ Lubrecht et al. (1999)

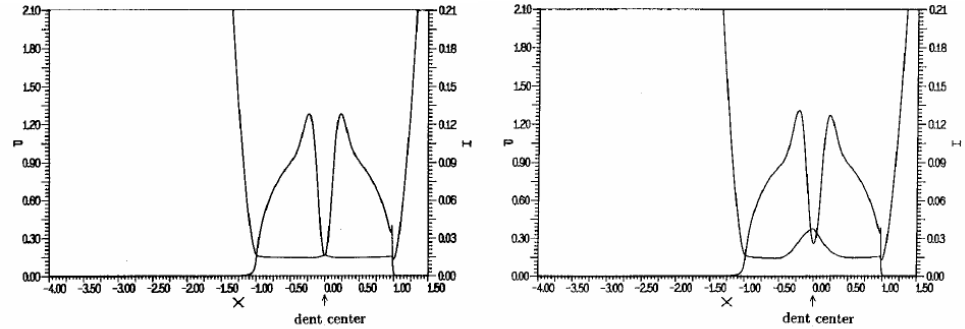
- two components of solution (Greenwood 1994)
 - deformed roughness
 - complementary wave



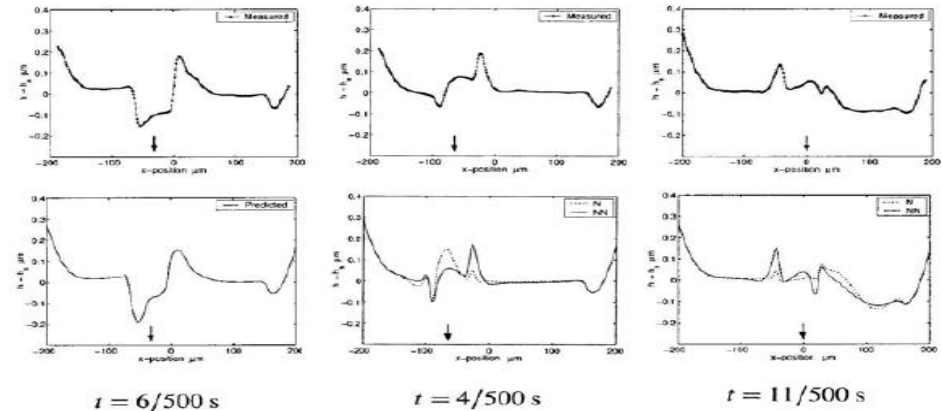
▪ Greenwood et al. (1994)

Full numerical simulations with surface roughness

- time-dependent models (Venner 1991, Ai 1993)
 - importance of the squeeze term in Reynolds equation
 - Newtonian solution
- non-Newtonian simulations (Felix-Quinonez et al. 2004)
 - pure rolling \times rolling-sliding
 - direct comparison with experiments



■ Venner (1991)



■ Felix-Quinonez et al. (2004)

Aims of the Thesis

The aim of the dissertation is to study the effect of non-Newtonian lubricant properties on the behavior of surface roughness inside the contact zone under rolling-sliding conditions by means of numerical simulations.

Partial tasks:

- fast and stable numerical solver
- parametric study of the non-Newtonian lubricant properties and their effect on deformation
- direct comparison with experimental results

Reynolds equation

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{12\eta} \phi_x \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{\rho h^3}{12\eta} \phi_y \frac{\partial p}{\partial y} \right) - u_m \frac{\partial(\rho h)}{\partial x} - \frac{\partial(\rho h)}{\partial t} = 0$$

Film thickness equation

$$h = h_0 + \frac{x^2}{2R_x} + \frac{y^2}{2R_y} - R + \frac{2}{\pi E_r} \iint_{\Omega} \frac{p(x', y') dx' dy'}{\sqrt{(x-x')^2 + (y-y')^2}}$$

Force balance equation

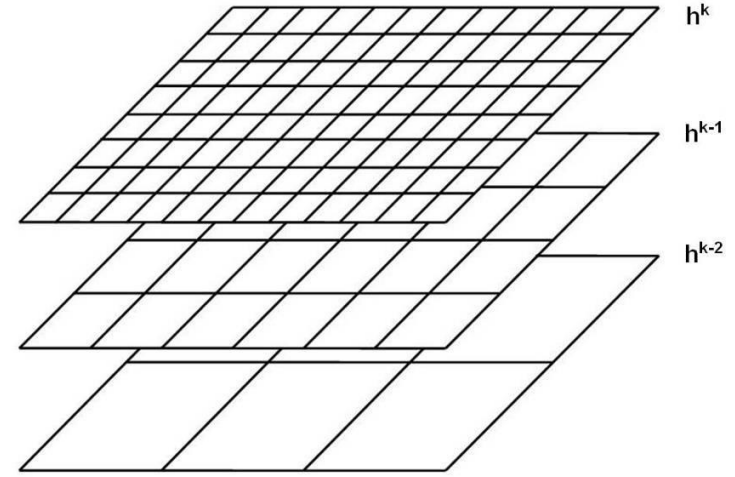
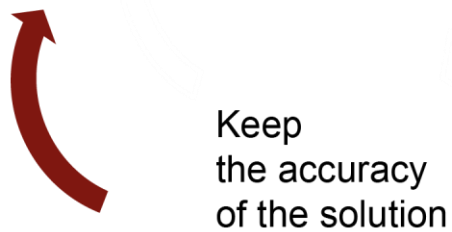
$$w = \iint_{\Omega} p(x, y) dx dy$$

Define:

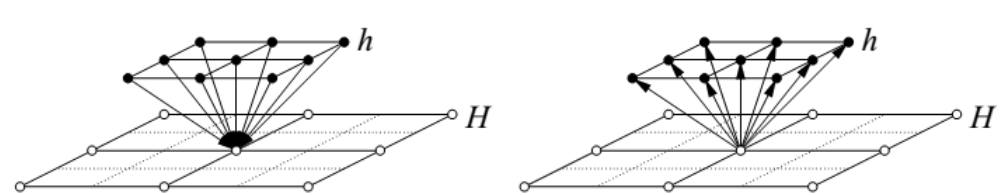
- boundary conditions
- cavitation condition
- lubricant properties

Numerical method

- **multigrid method** – Reynolds equation
- **multilevel multi-integration** – film thickness equation
- accurate definition of the **transfer between the grids**



▪ Urbanec (2007)



▪ Wijnant (1998)

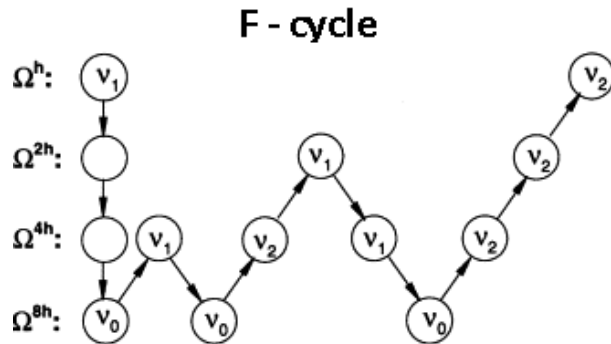
Implementation

Initial approximation

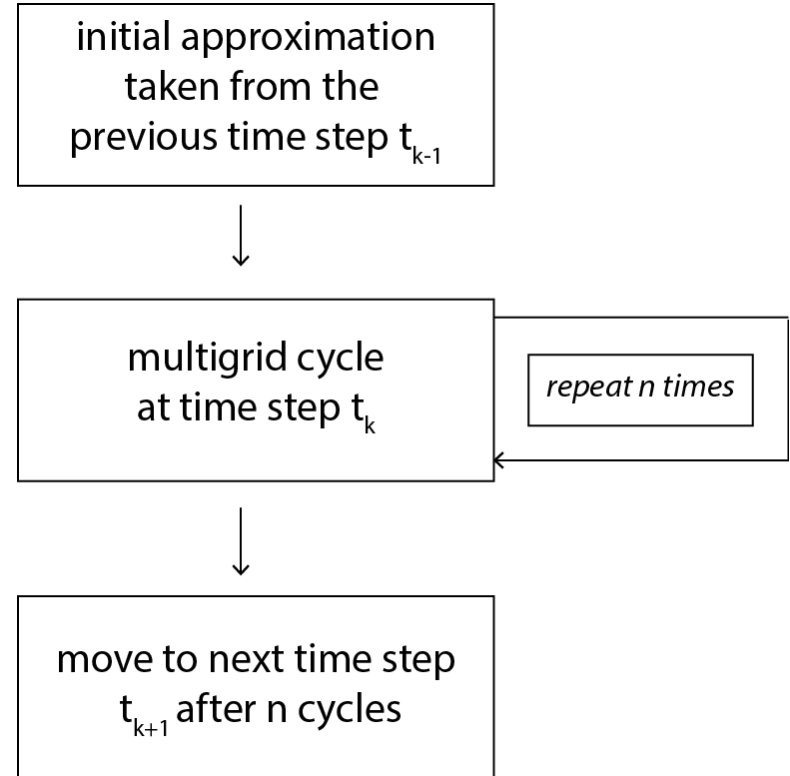
- pressure P
- mutual approach H_0

At each time step

- multigrid F-cycle is applied



▪ Wesseling (1991)

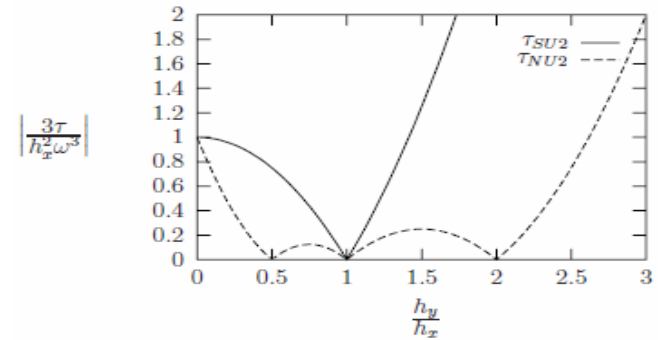
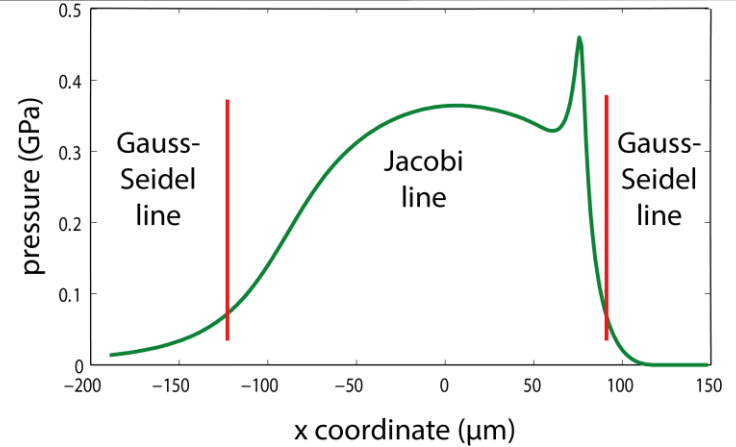


Implementation

- Single grid solution
 - 2 relaxation schemes
 - switch criterion

$$\xi_{\text{lim}} = \frac{\rho H^3}{\eta \lambda}$$

- Discretization and step size
 - 2nd order discretization
 - NU2 scheme



▪ Wijnant (1998)

Newtonian

$$\phi_x = 1 \quad \phi_y = 1$$

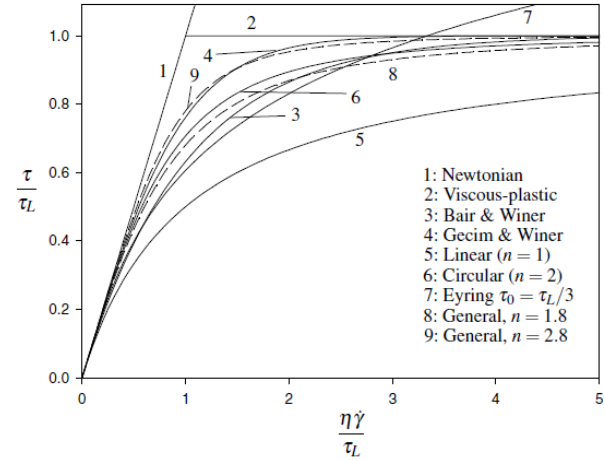
non-Newtonian model

- line × point contacts
- Eyring sinh law × generalized Newtonian

Eyring model

$$f\left(\frac{\tau_m}{\tau_0}\right) = \frac{\tau_0}{\tau_m} \sinh\left(\frac{\tau_m}{\tau_0}\right)$$

$$\phi_x = \cosh\left(\frac{\tau_m}{\tau_0}\right) \quad ! \quad \phi_y = \frac{\tau_0}{\tau_m} \sinh\left(\frac{\tau_m}{\tau_0}\right)$$



▪ Sloetjes (2006)

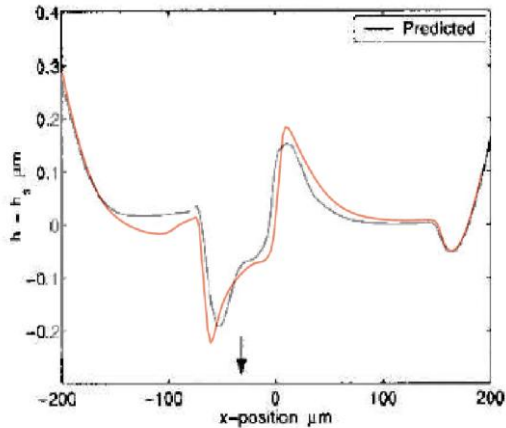
Effective viscosities

Verification and validation of the model

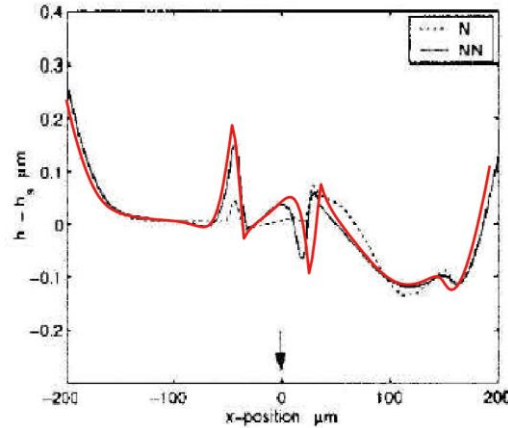
Comparison with previously published work - Felix-Quinonez (2004)

- flat-top transverse ridge - same geometry
- Newtonian vs. Eyring model, comparison with experiments

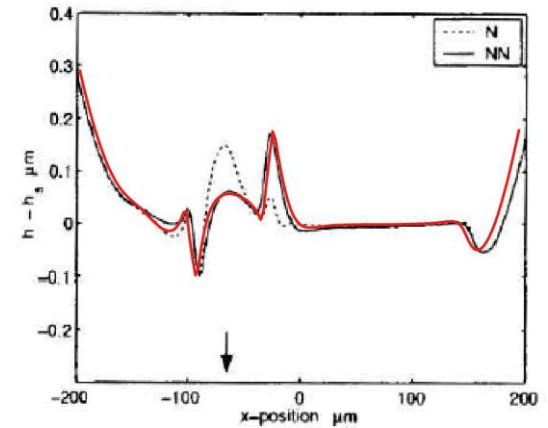
pure rolling



SRR=+100%



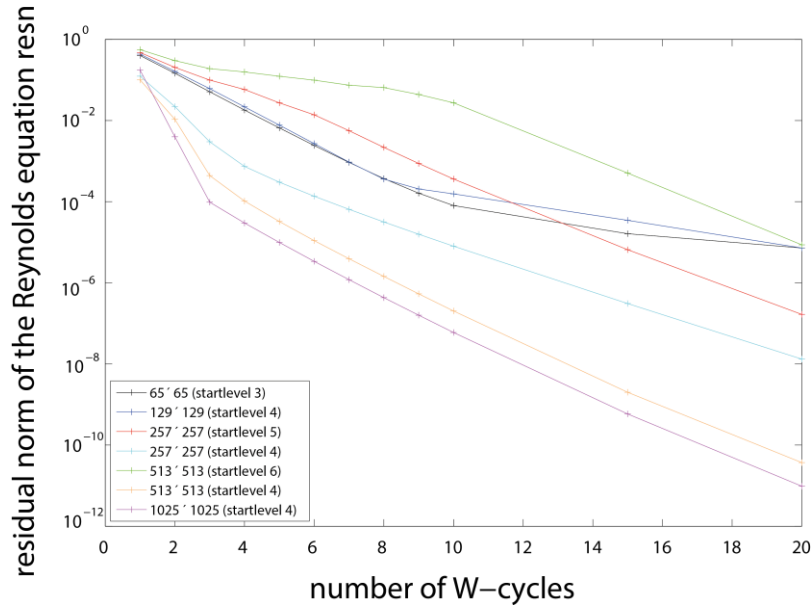
SRR= - 100%



Accuracy of the solver

- Accuracy

- stationary – up to 0 (10^{-10})
- non-stationary ~ 0 ($10^{-3} - 10^{-4}$)



- Effect of time step

Time step	Residual norm Reynolds eqn.	Force balance residual
$\Delta T = 2\Delta X$	$6.3 \cdot 10^{-3}$	$1.8 \cdot 10^{-3}$
$\Delta T = \Delta X$	$3.1 \cdot 10^{-3}$	$6.6 \cdot 10^{-4}$
$\Delta T = 0.5\Delta X$	$2.8 \cdot 10^{-3}$	$4.3 \cdot 10^{-4}$

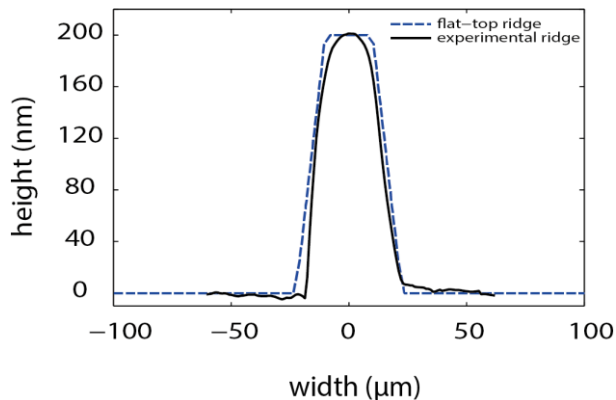
Results - Transient simulations

Input parameters

Value

Load (w)	30 N
Material (steel-glass)	123.8 GPa
Reduced radius (R_x)	0.0127 m
Mean velocity (u_m)	0.02, 0.04, 0.08 m/s
Slide to roll ratio (SRR)	0%, $\pm 50\%$, $\pm 100\%$, $\pm 150\%$

Single flat-top transverse ridge

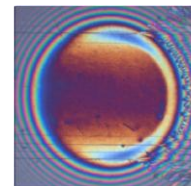
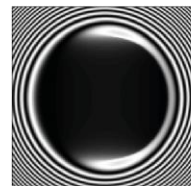
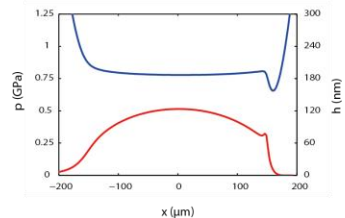


height
200 nm

base width
45 μm

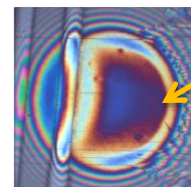
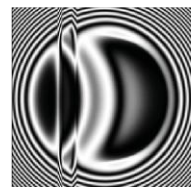
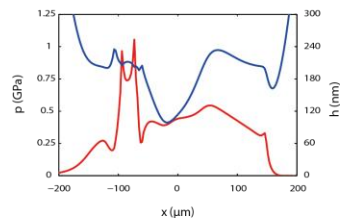
top width
20 μm

SRR +100%, $u_m = 0.08$ m/s, SR 600



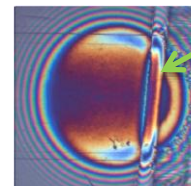
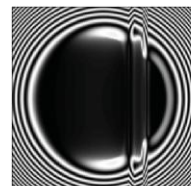
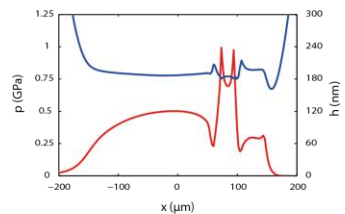
(a) $x_d = -417.5 \mu\text{m}$

Complementary
wave



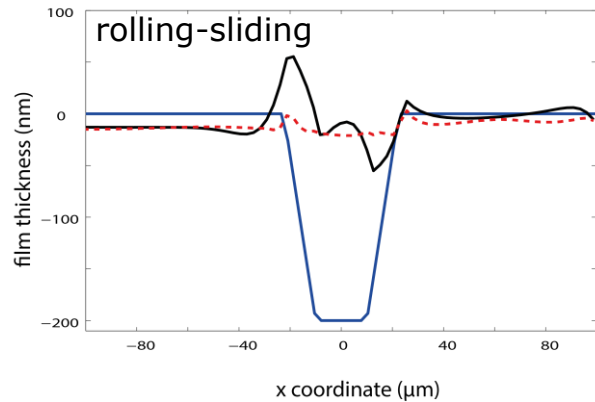
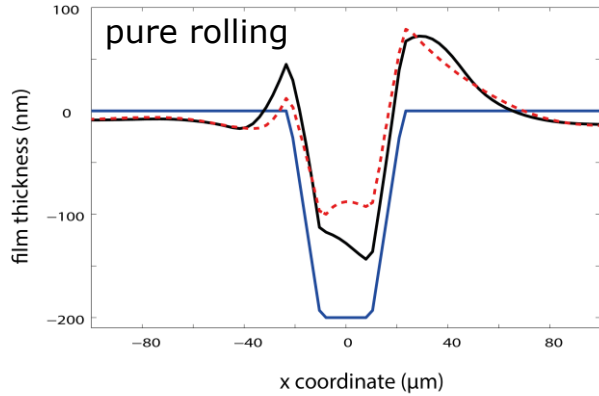
roughness
deformation

(b) $x_d = -83.5 \mu\text{m}$



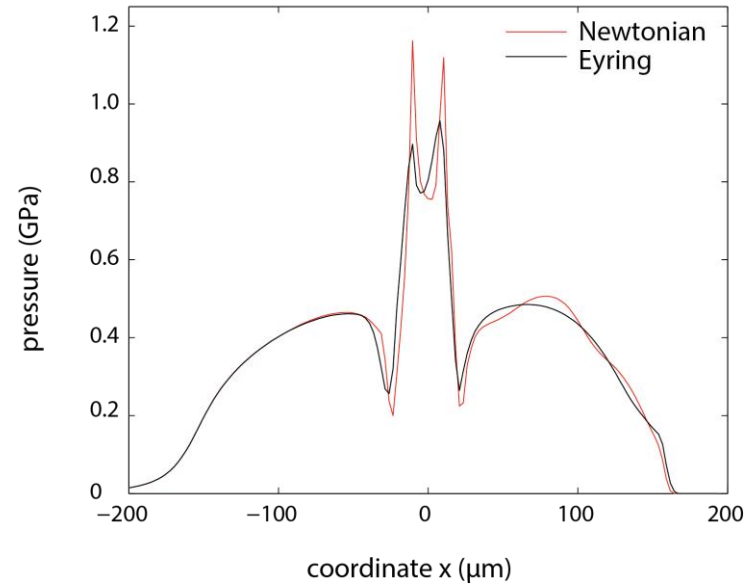
(c) $x_d = 83.5 \mu\text{m}$

Results - Newtonian vs. non-Newtonian model

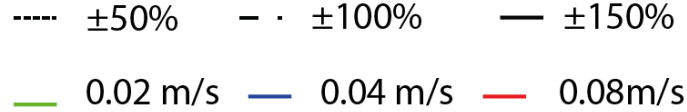


rolling-sliding

SRR +100%, $u_m=0.08$ m/s, SR 600

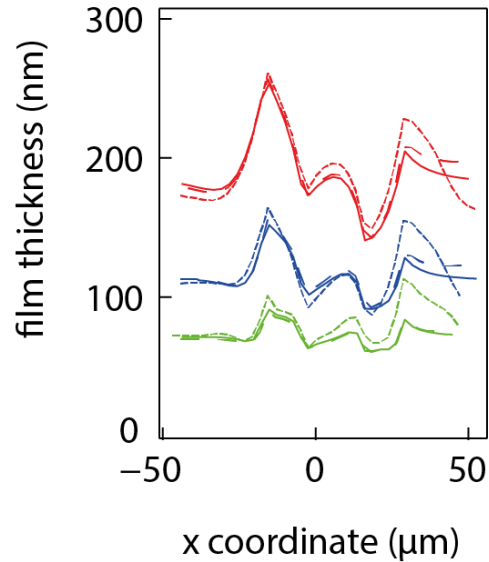


Results - Effect of mean speed and SRR



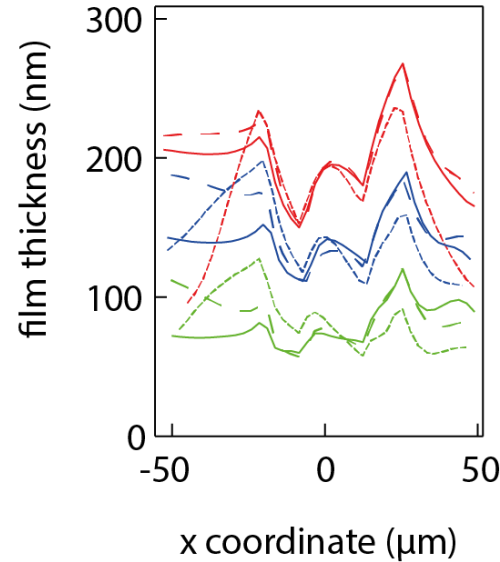
Positive SRRs

slower
roughness

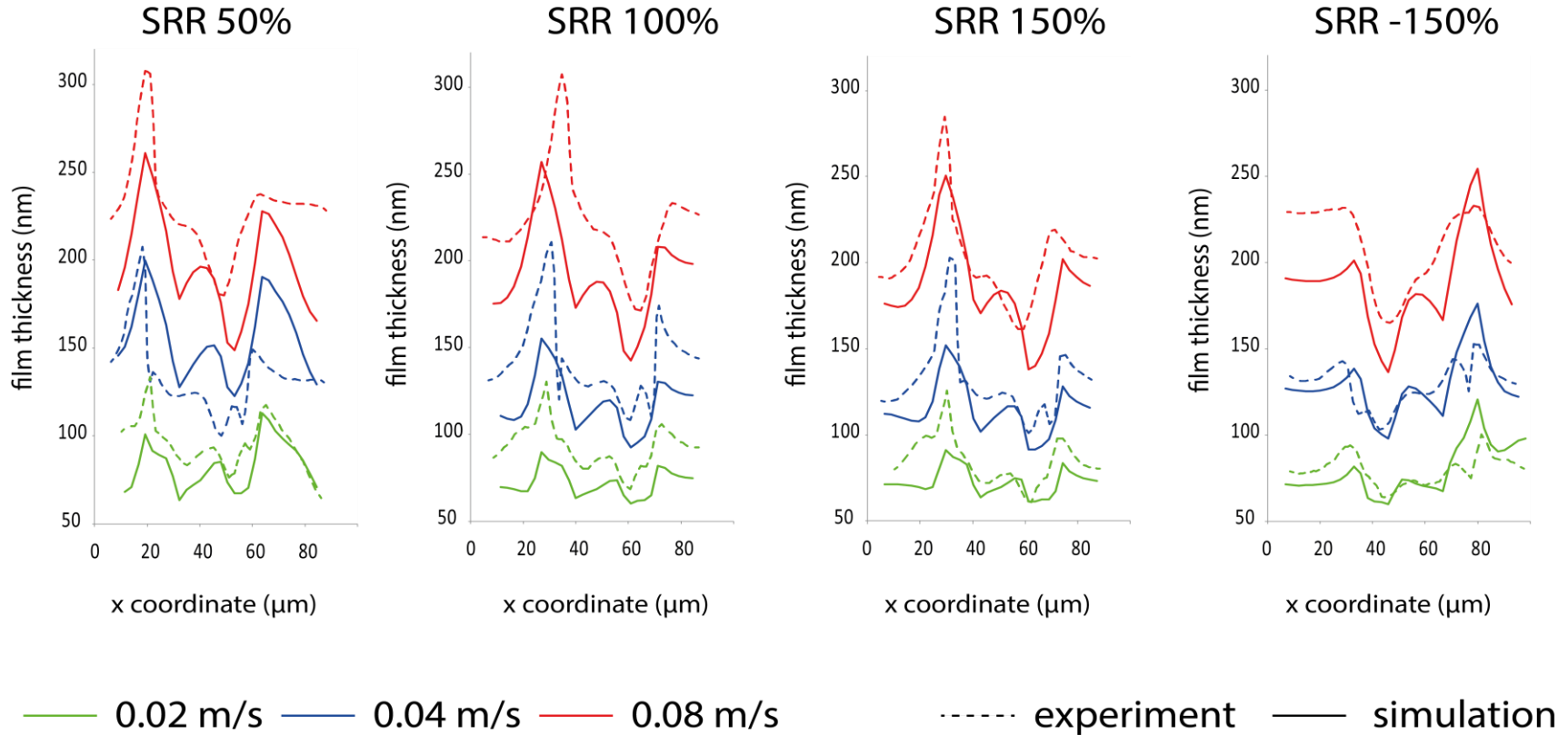


Negative SRRs

faster
roughness



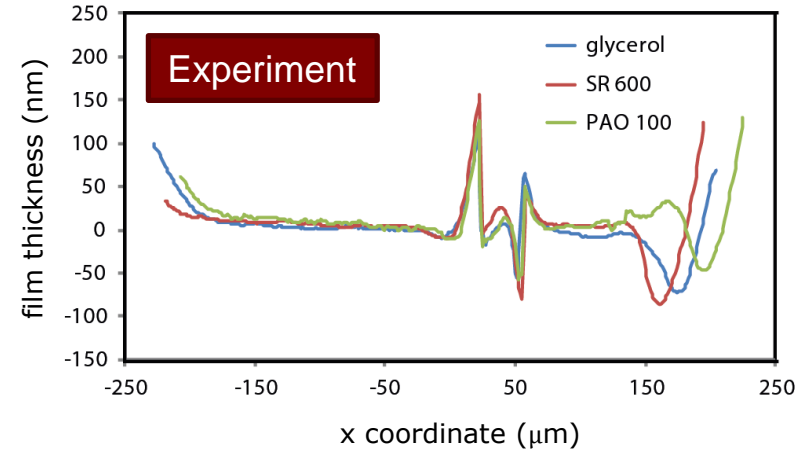
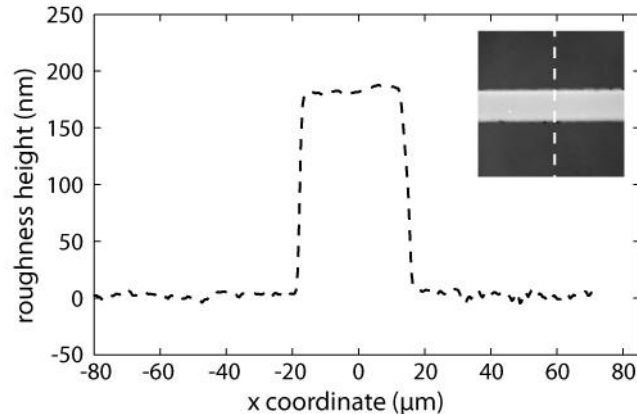
Results - Comparison with experiments



Results - Effect of lubricant properties

Oil	SR 600	Glycerol	PAO 100
Ambient viscosity η_0 (Pa s)	0.22	0.46	1.8
Pressure-viscosity coefficient α (GPa ⁻¹)	24	5	20
Eyring shear stress τ_0 (MPa)	5	2.5	1

- SRR +120%
- load 55 N

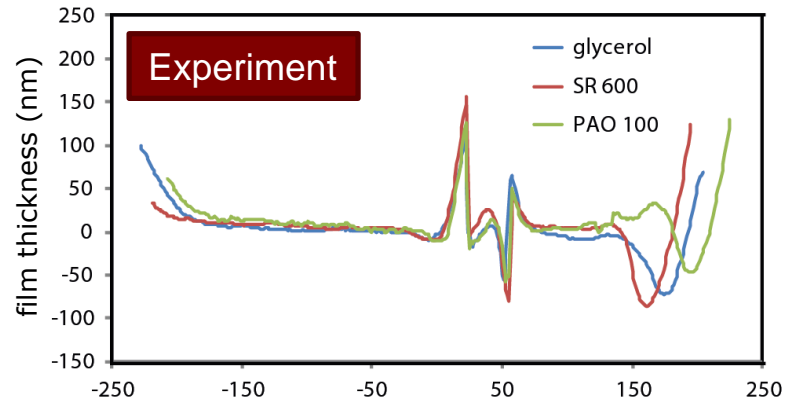


height
190 nm

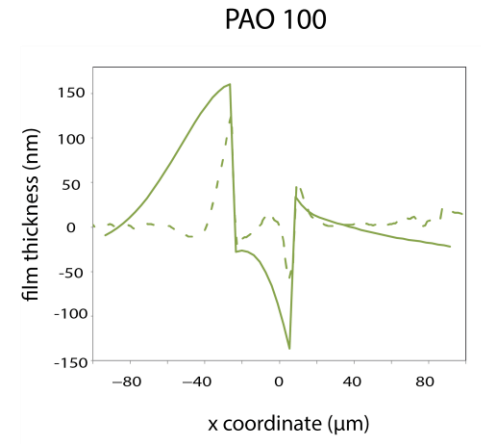
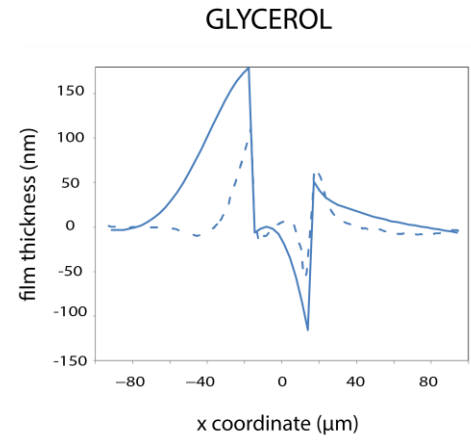
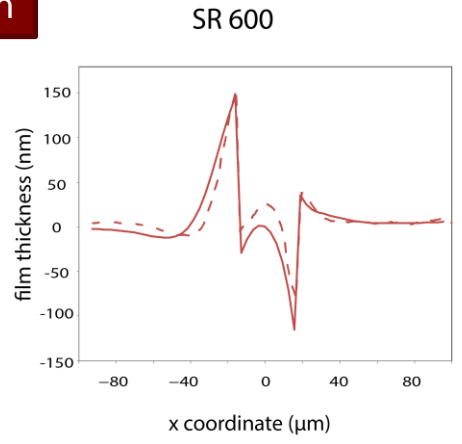
width
40 μm

Results - Effect of lubricant properties

Oil	SR 600	Glycerol	PAO 100
Ambient viscosity η_0 (Pa s)	0.22	0.46	1.8
Pressure-viscosity coefficient α (GPa ⁻¹)	24	5	20
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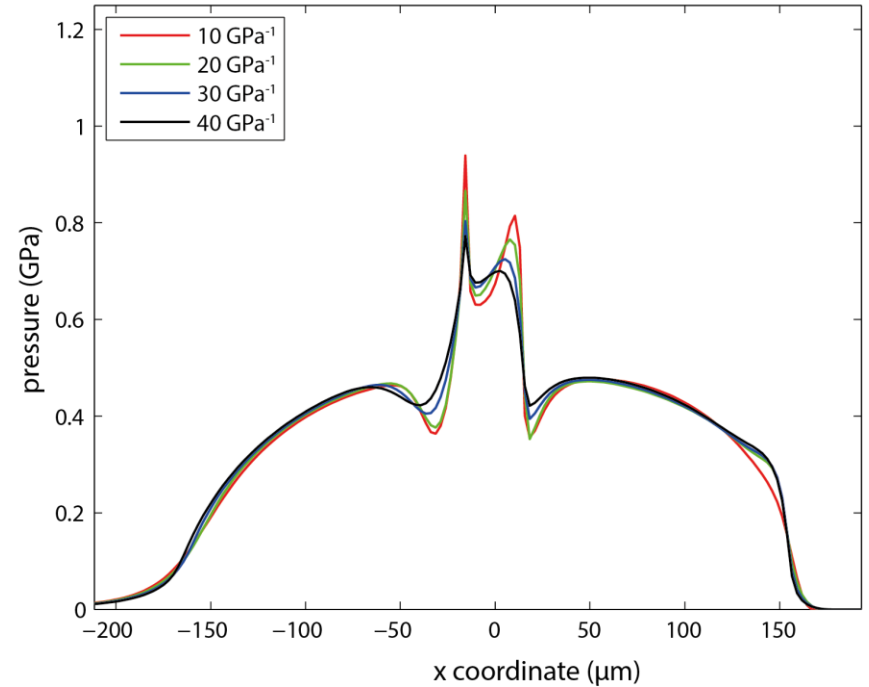
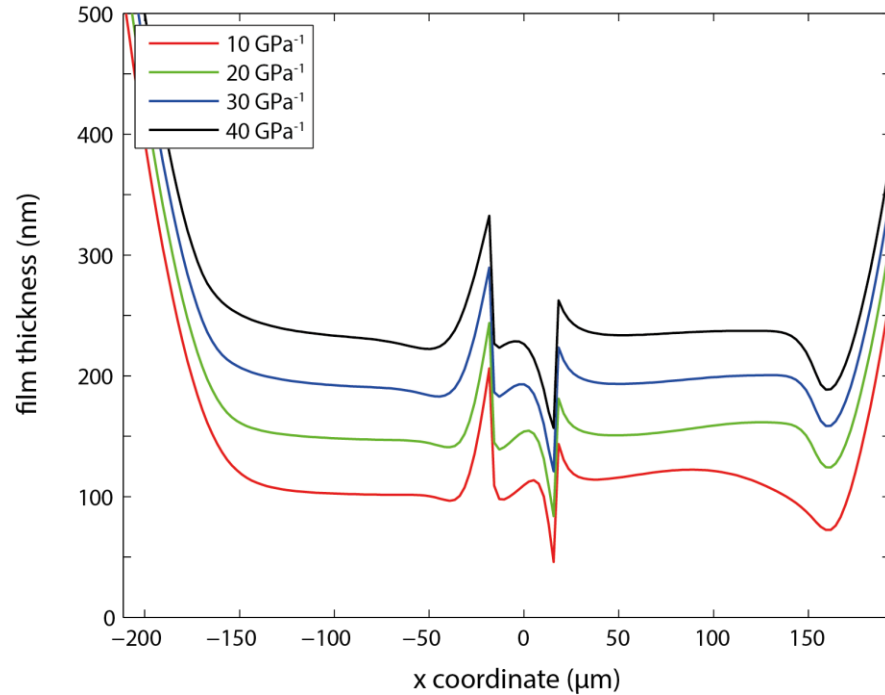
Comparison



Results - Effect of lubricant properties

Pressure-viscosity coefficient α

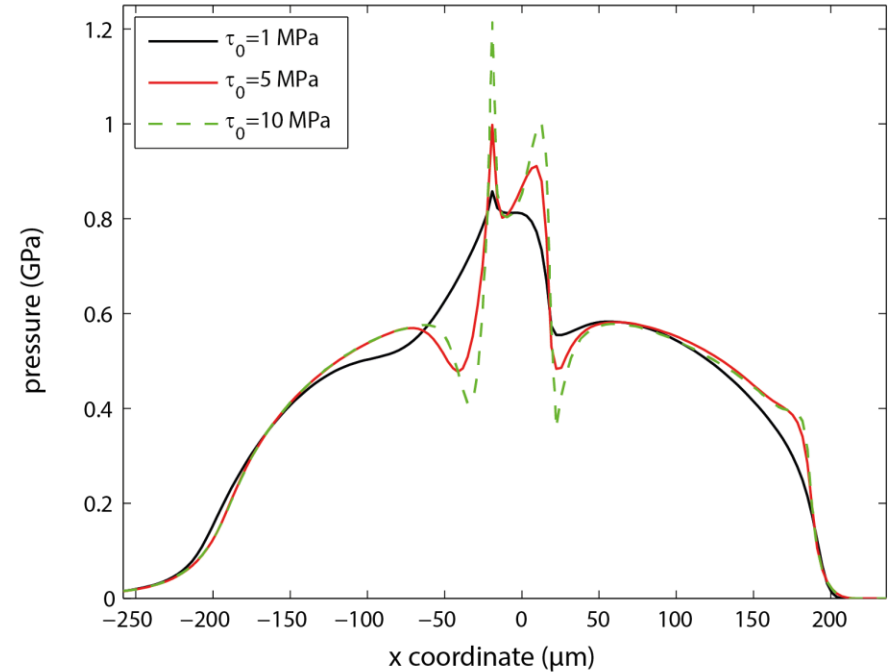
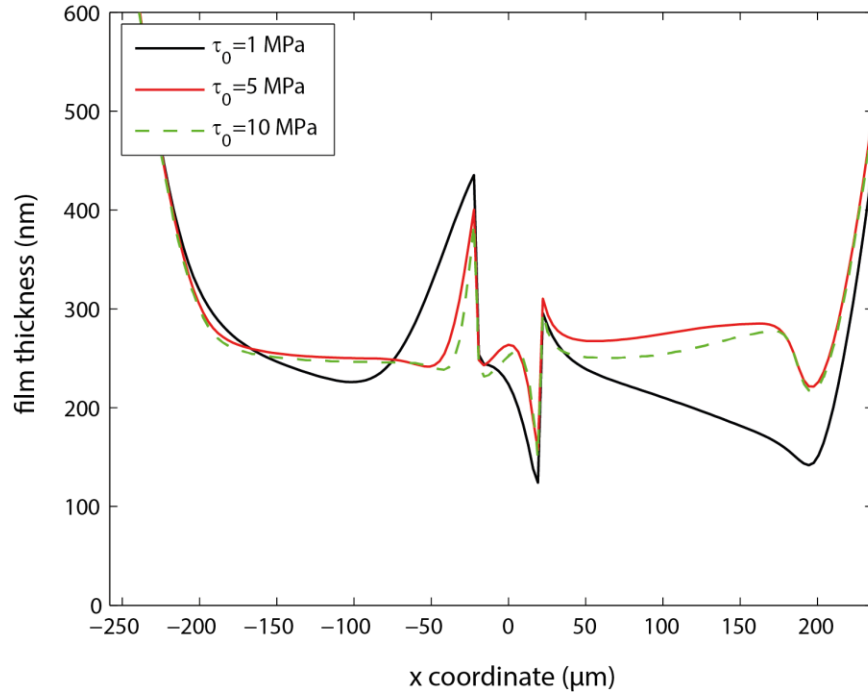
SRR +120%, $u_m=0.133$ m/s, 55 N, $\tau_0=5$ MPa



Results - Effect of lubricant properties

Eyring shear stress τ_0

SRR +120%, $u_m=0.133$ m/s, 55 N, $\alpha=24$ GPa⁻¹



Conclusions

- Effect of slide to roll ratio
 - in agreement with experiments the deformation is independent of the sliding magnitude (for $|SRR| \geq 0.5$)
- Effect of mean velocity
 - in agreement with experiments with decreasing mean velocity the deformation is increasing
- Effect of lubricant rheological properties
 - simulations with different lubricant properties exhibit discrepancy against experiments
 - pressure-viscosity coefficient α – minor influence
 - Eyring shear stress τ_0 – significant influence
- **Aims of the thesis were fulfilled.**

Thank you for your attention

I. Ficza

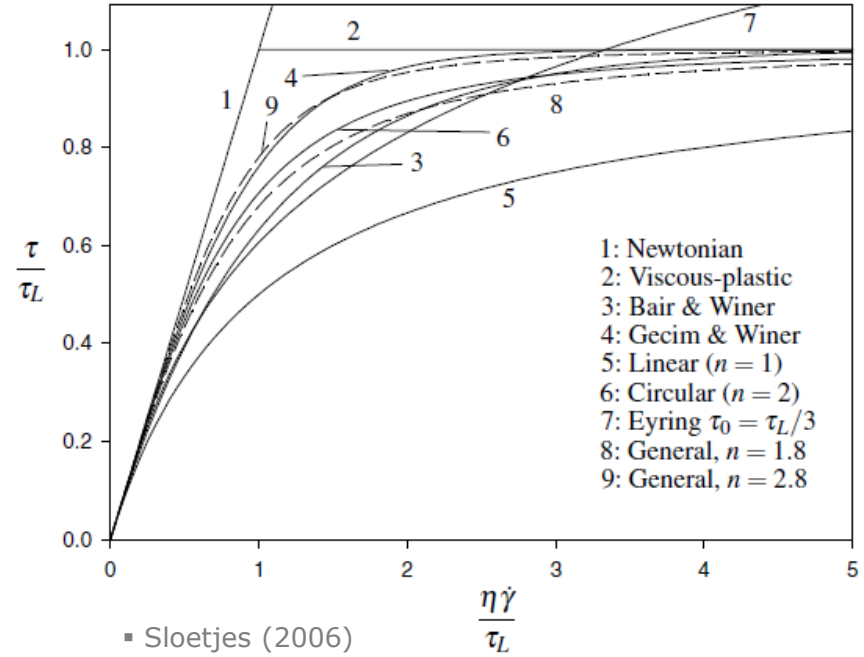


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Reviewer's question

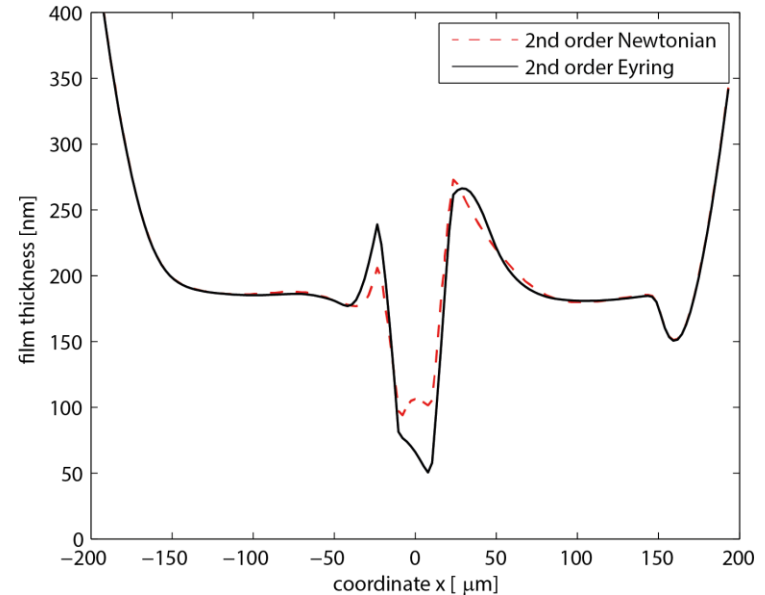
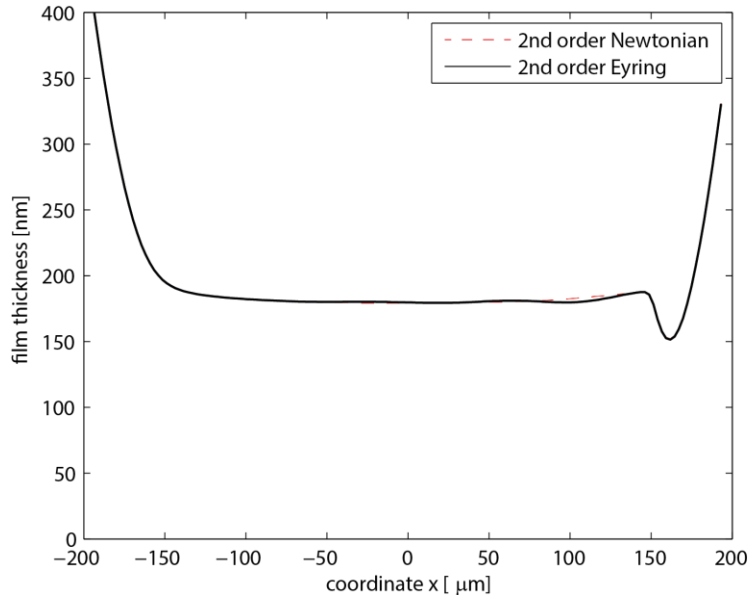
#1) As described by the author, there are numerous rheology models. Why did the author adopt the Eyring shear thinning model?

- Eyring sinh law:
 - implementation – effective viscosities
 - shear stress can be obtained without additional integrations across the film thickness



Reviewer's question

#2) Fig. 6.16 shows the film profiles obtained in pure rolling. The film thickness is slightly larger for the Eyring model than for the Newtonian model. Could the author explain the reason?



Reviewer's question

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