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

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Ph.D. Thesis Defense
July 24th, 2015

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Introduction

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
Point contact – inverse solution
Point contact – direct solution
Multigrid methods

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Study of roughness deformation

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

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

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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 × 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
Mathematical model

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

Change the number of grid points
Increase computational speed
Keep the accuracy of the solution

▪ Urbanec (2007)
▪ Wijnant (1998)
Implementation

Initial approximation
- pressure $P$
- mutual approach $H_0$

At each time step
- multigrid F-cycle is applied

\[ \text{initial approximation taken from the previous time step } t_{k-1} \]

\[ \text{multigrid cycle at time step } t_k \]

\[ \text{move to next time step } t_{k+1} \text{ after } n \text{ cycles} \]

- Wesseling (1991)

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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)
Implementation

Newtonian

\[ \phi_x = 1 \quad \phi_y = 1 \]

non-Newtonian model

• line × point contacts

• Eyring sinh law × generalized Newtonian

Eyring model

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

Effective viscosities

\[
\phi_x = \cosh\left(\frac{\tau_m}{\tau_0}\right)
\]

\[
\phi_y = \frac{\tau_0}{\tau_m} \sinh\left(\frac{\tau_m}{\tau_0}\right)
\]

Sloetjes (2006)
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

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Accuracy of the solver

- Accuracy
  - stationary – up to $0 \left(10^{-10}\right)$
  - non-stationary $\sim 0 \left(10^{-3} - 10^{-4}\right)$

- Effect of time step

<table>
<thead>
<tr>
<th>Time step</th>
<th>Residual norm Reynolds eqn.</th>
<th>Force balance residual</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta T = 2\Delta X$</td>
<td>$6.3 \cdot 10^{-3}$</td>
<td>$1.8 \cdot 10^{-3}$</td>
</tr>
<tr>
<td>$\Delta T = \Delta X$</td>
<td>$3.1 \cdot 10^{-3}$</td>
<td>$6.6 \cdot 10^{-4}$</td>
</tr>
<tr>
<td>$\Delta T = 0.5\Delta X$</td>
<td>$2.8 \cdot 10^{-3}$</td>
<td>$4.3 \cdot 10^{-4}$</td>
</tr>
</tbody>
</table>
## Results - Transient simulations

<table>
<thead>
<tr>
<th>Input parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load (w)</td>
<td>30 N</td>
</tr>
<tr>
<td>Material (steel-glass)</td>
<td>123.8 GPa</td>
</tr>
<tr>
<td>Reduced radius (Rx)</td>
<td>0.0127 m</td>
</tr>
<tr>
<td>Mean velocity ((u_m))</td>
<td>0.02, 0.04, 0.08 m/s</td>
</tr>
<tr>
<td>Slide to roll ratio (SRR)</td>
<td>0%, ±50%, ±100%, ±150%</td>
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</table>

### SRR +100\%, \(u_m=0.08\) m/s, SR 600

**Complementary wave**

**Roughness deformation**

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**Single flat-top transverse ridge**

- Height: 200 nm
- Base width: 45 μm
- Top width: 20 μm

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Results - Newtonian vs. non-Newtonian model

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Results - Effect of mean speed and SRR

Positive SRRs
slower roughness

Negative SRRs
faster roughness

- - - ±50%  - - ±100%  - - - ±150%

0.02 m/s  0.04 m/s  0.08 m/s
Results - Comparison with experiments

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## Results - Effect of lubricant properties

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<th>PAO 100</th>
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<td>Ambient viscosity $\eta_0$ (Pa s)</td>
<td>0.22</td>
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<td>Pressure-viscosity coefficient $\alpha$ (GPa$^{-1}$)</td>
<td>24</td>
<td>5</td>
<td>20</td>
</tr>
<tr>
<td>Eyring shear stress $\tau_0$ (MPa)</td>
<td>5</td>
<td>2.5</td>
<td>1</td>
</tr>
</tbody>
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- SRR +120%
- Load 55 N

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**Experiment**

- Height 190 nm
- Width 40 $\mu$m
### Results - Effect of lubricant properties

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#### Comparison

**SR 600**

**Glycerol**

**PAO 100**
Results - Effect of lubricant properties

Pressure-viscosity coefficient $\alpha$

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

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 \(|\text{SRR}| \geq 0.5\))

• 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 \(\alpha\) – minor influence
  • Eyring shear stress \(\tau_0\) – significant influence

• Aims of the thesis were fulfilled.
Thank you for your attention
#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
#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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