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ANALYSIS OF THE ELASTO-HYDRODYNAMIC LUBRICATION IN COATED FINITE LENGTH LINE CONTACTS

CATEGORY: LUBRICATION FUNDAMENTALS – EHL MODELLING AND EVALUATION

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INTRODUCTION

Due to the finite lengths of components, such as cam-roller followers pairs, rolling element bearings, gear teeth etc., high stresses are typically generated towards the extremities of the contact, often referred to as edge loading. Therefore axial surface profiling of the components is often utilized to minimize edge loading.
Depending on the type of surface profiling, the pressure and film thickness distribution may deviate significantly from that predicted using the infinitely long line contact assumption, i.e. the film shapes near the extremities are very different from those at the central plane. In fact. the absolute minimum film thickness and maximum pressure, which are crucial design parameters, always occur near the position where axial surface profiling starts. This has also been proven both experimentally and theoretically by a number of researchers [1] [2] [3] [4].
Further, nowadays there is an increasing trend in the use of surface coatings in lubricated contacts. It has been observed from past studies (see for instance [5], [6] and [7]) that the use of surface coatings can significantly enhance lubrication performance if the stiffness and thickness of the coating is designed properly. However, past studies concerning lubricated coated contacts, are mostly limited to infinite line contacts and circular contacts with and without coatings.
Therefore, in this work a finite element method (FEM)-based finite line contact EHL model is developed, that includes the possibility of having a coating on both surfaces.
As case study, the lubricated conjunction of an axially surface profiled coated roller on a coated plate is analyzed. The analysis is simplified by assuming that the two substrates and coatings are made of the same material, i.e. the coating and the bulk material of both surfaces have the Young's moduli and Poisson's ratios, although the coating can have different properties than the substrate.
Here, the influence of operating conditions such as lubricant entrainment speed, load, lubricant properties, axial surface parameters and coating material properties on the film thickness are analyzed.

MODEL DESCRIPTION

The model presented herein is similar to the fully coupled circular coated contact model presented by Habchi [7], but simulates finite line contacts. Furthermore, isothermal conditions are assumed to simplify the analysis. The model relies on a full system finite element discretization of the EHL governing equations, linear elasticity and the load balance equations. A straight cylindrical roller with axial dub-off profiling was considered (see Figure 1). Taking this model geometry gives us the opportunity to investigate the influence of two axial surface parameters, namely the straight roller length $\boldsymbol{l_\mathrm{s}}$ and dub-off radius $\boldsymbol{R_\mathrm{d}}$.

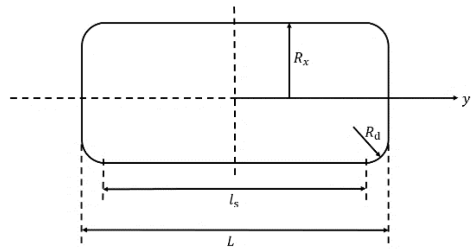
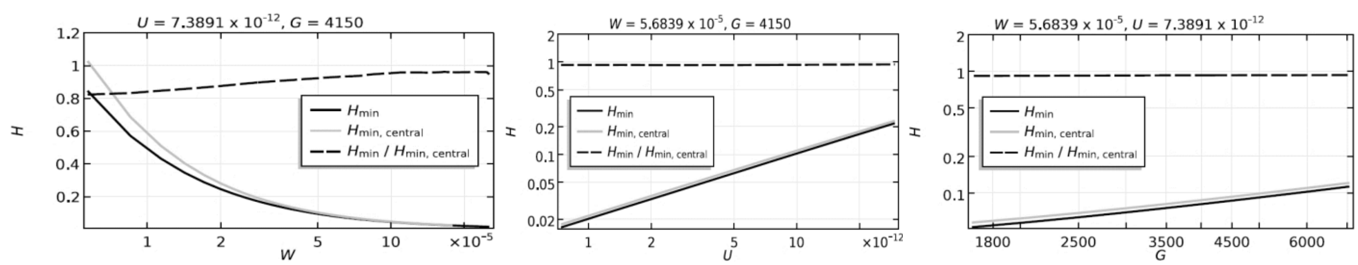


Figure 1: Schematic of considered roller axial profile.

SIMULATION RESULTS

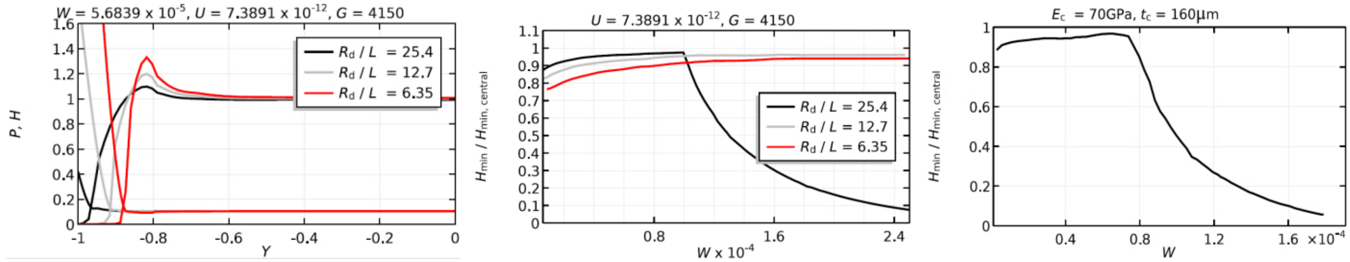
Figure 2 plots the variation of ratio $H_\mathrm{min}/H_\mathrm{min,central}$ as function of the dimensionless load W, speed U and material property G parameters. H_min is the absolute minimum film thickness, occurring at the extremities of the contact (usually where axial surface profiling starts) and $H_\mathrm{min,central}$ is the minimum film thickness occurring at the central plane. The behavior of $H_\mathrm{min,central}$ is much more explainable using traditional EHL solutions for infinite line contacts [8]. It is therefore much more interesting to study the behavior of ratio $H_\mathrm{min}/H_\mathrm{min,central}$ from a designers perspective. In practice one would like to maximize the value of $H_\mathrm{min}/H_\mathrm{min,central}$ as $H_\mathrm{min,central}$ can fairly be estimated using the infinite line contact assumption (and can thus be roughly estimated using the popular Dowson-Higginson film thickness formula).
From Figure 2a it is clear that H_min is highly affected by variations in load. Especially from low to moderate loads this phenomenon is much more visible. For variations in speed and material properties the ratio $H_\mathrm{min}/H_\mathrm{min,central}$ seems to remain constant.



a) b) c)

Figure 2: Variation of H_min, $H_\mathrm{min,central}$ and ratio $H_\mathrm{min}/H_\mathrm{min,central}$ with **a)** dimensionless speed, **b)** material and **c)** load parameters while keeping two fixed at a time. The results correspond to the case when both interacting components are uncoated.

Apart from varying operating conditions it is also interesting to take a look at the influence of geometrical parameters on the pressure and film thickness distributions. In fact, for the axial profile of the roller one may vary the straight roller length and dub-off radius to optimize the pressure distribution, i.e. to make it more uniform by reducing edge stress concentrations and consequently increase $H_\mathrm{min}/H_\mathrm{min,central}$. Note that the H_min is restricted by the local pressure gradient as per flow continuity demand.
From Figure 3a it is clear that a higher relief radius smears out the pressure peak at the sides of the contact, resulting to a larger contact area. Furthermore, from Figure 3b on can observe that the ratio $H_\mathrm{min}/H_\mathrm{min,central}$ seems to increase with increasing R_d. The aforementioned is amplified with increasing loads. It would then be expected that that choosing a larger R_d results in a more uniform pressure profile and thus a better design.
However, there seems to be an optimum range for minimum film thickness vs dub-off radius mapped against the range of loads. In fact, for a too large R_d the ratio $H_\mathrm{min}/H_\mathrm{min,central}$ starts to decrease after a certain applied load. This is mainly due to the fact that there is no space available for the pressure profile to extend as a zero boundary condition (fully flooded conditions) is imposed at the extremities. Consequently, the pressure gradient $\frac{dP}{dY}$ at the extremities increases and thus H_min decreases (see Figure 3b).



a) b) c)

Figure 3 : a) Influence of dub-off radius radius R_d on pressure and film thickness distribution (symmetry at the $Y = 0$ plane, uncoated contact). **b)** Influence of dub-off radius R_d on minimum film thickness (mapped against load, uncoated contact). **b)** Influence of load on roller with soft and thick coatings.

In line with previous findings (see for example [7]), for coated contacts we also see that for more elastic coatings (softer coatings) the lubricated contact area is increased due to increased deformation. This effect is further amplified with increasing coating thickness. The same also applies for increasing contact force, i.e. the lubricated contact area is also expanded with increasing load. This means that one should be careful when dealing with elastic and thick coatings and high loads, in terms of minimum film thickness H_min, as all three aforementioned factors lead to an increase in contact area. From Figure 3c it is clear that at relatively high loads the ratio $H_\mathrm{min}/H_\mathrm{min,central}$ dramatically decreases. This is mainly due to the fact that at the rear of the contact the pressure distribution does not have sufficient space to expand, and thus the pressure peak at the extremities again grows in magnitude. Consequently, pressure gradients at the extremities increase and thus will the minimum film thickness H_min decrease.
All these findings make it really hard to develop a robust correlation between absolute minimum film thickness, maximum pressure and operating conditions, as coating and axial profile geometrical parameters play an equally important role. The present results certainly contribute to a better understanding of lubricated and coated finite line contacts. This model can effectively be used for improved designs of finite line contact applications in terms of film thickness and pressure distributions, and thus ultimately, contributing to longer service life of the components.

ACKNOWLEDGMENTS

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KEYWORDS
Elastohydrodynamic lubrication, finite line contacts, coatings

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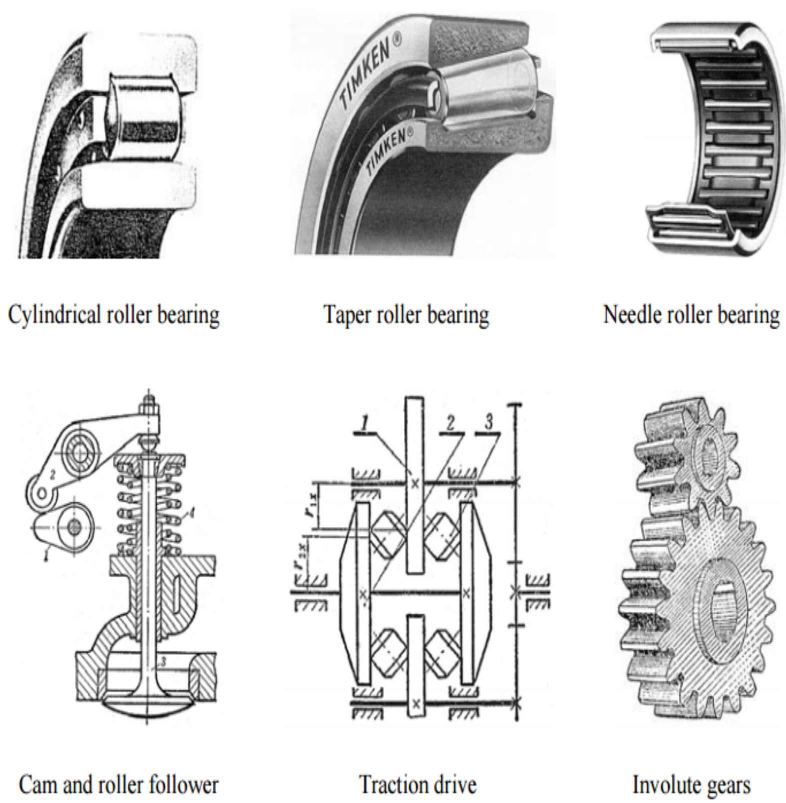
Outline

- Problem definition
- EHL modelling
- Results
  - Uncoated case
    - Influence of operating conditions, such as speed, load and material elasticity, on pressure and film thickness distribution
    - Influence of varying axial geometry on film thickness and pressure distribution
  - Coated case
    - Influence of coating elasticity and thickness on pressure and film thickness distribution
- Conclusions

Problem definition II



These elements are normally profiled at their axial ends to overcome the problems of edge effects caused by their finite

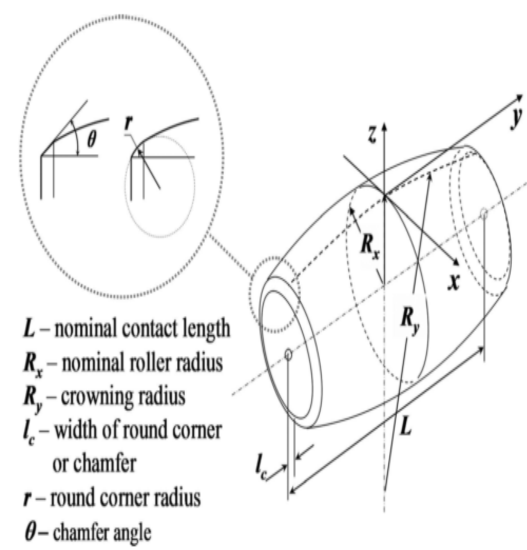
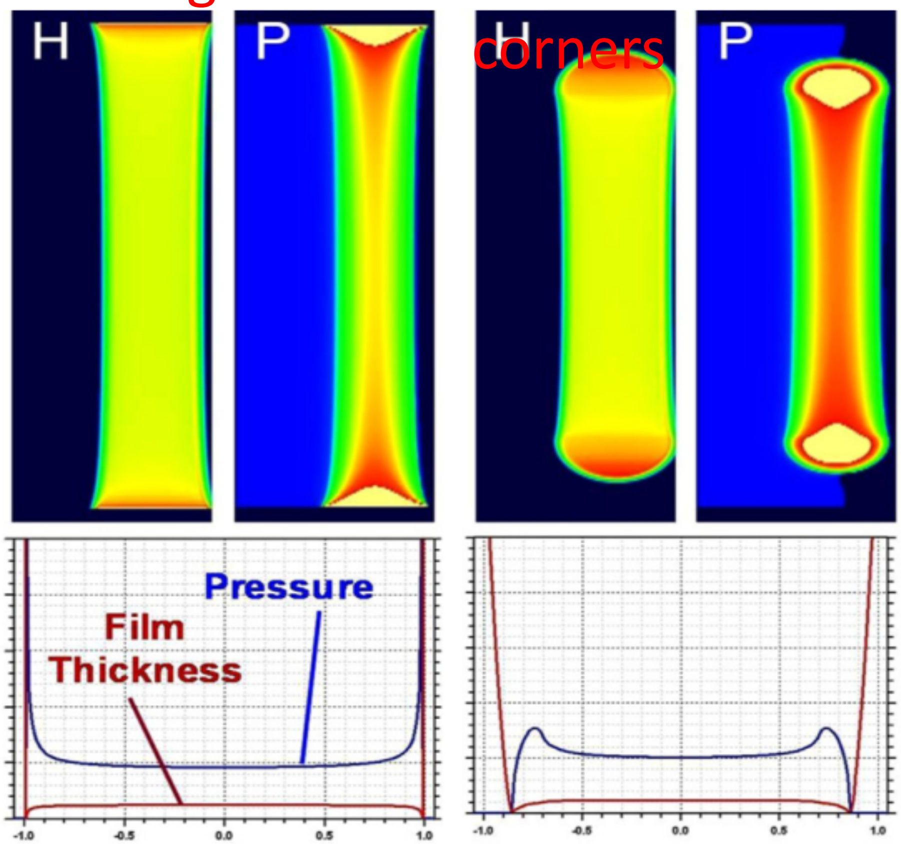


Figure reproduced from Chen et al[1] (2006)

[1] Chen, X., Sun, H., & Shen, X. (2006). Review and prospects for the development of EHL of finite line contacts. In *IUTAM Symposium on Elastohydrodynamics and Micro-elastohydrodynamics* (pp. 95-106). Springer Netherlands.

Problem definition I: Roller on plate

Straight roller Roller with round corners



Reproduced from Zhu et al (2012) [1]

For axially straight rollers, with finite differences the pressure would reach an infinite value with a large number of mesh points at the edge discontinuities and the lubricant film

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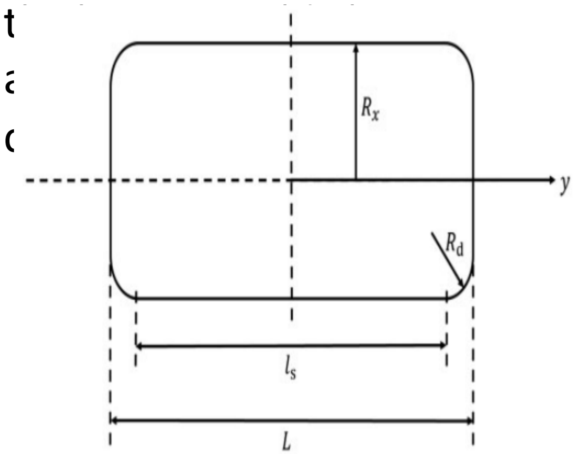


[1]Zhu, D., Wang, J., Ren, N., & Wang, Q. J. (2012). Mixed elastohydrodynamic lubrication in finite roller contacts involving realistic geometry and surface roughness. *Journal of Tribology*, *134*(1), 011504.

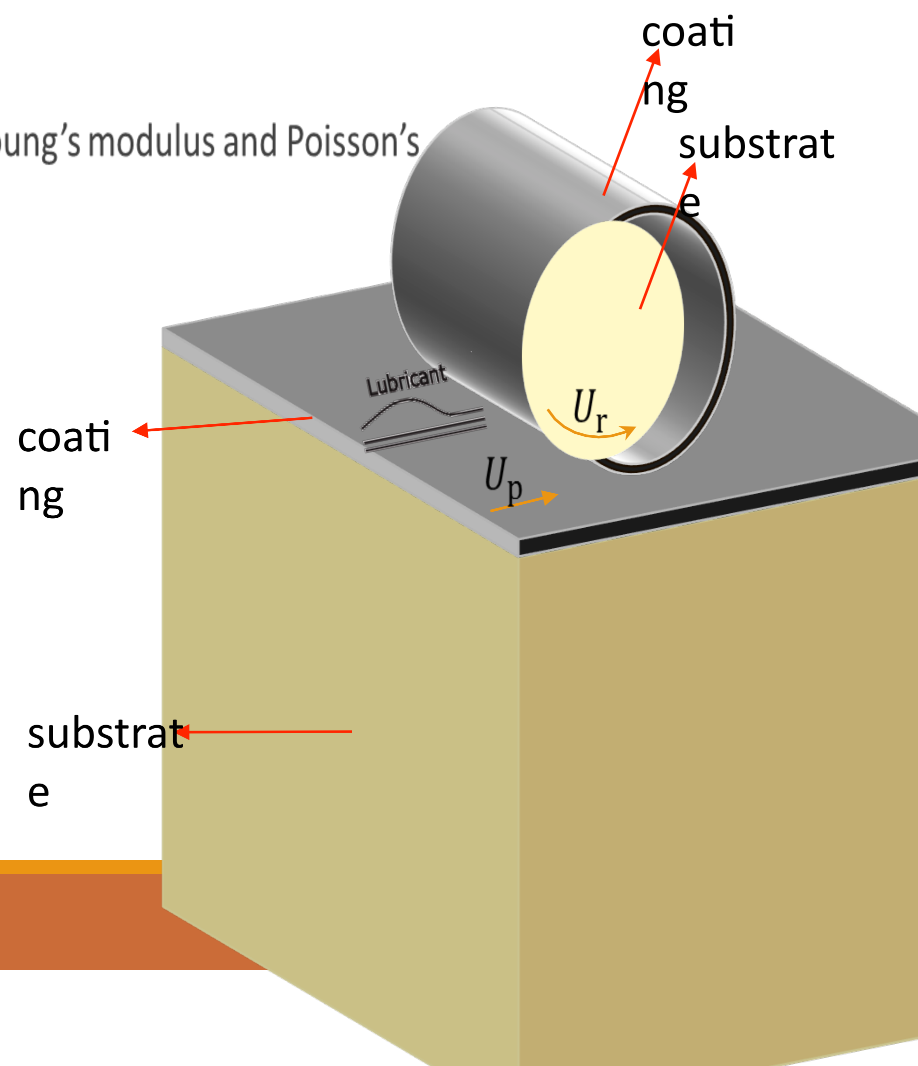
Modelling I: assumptions

- Smooth surfaces, i.e. roughness effects are ignored.
- Isothermal analysis and Newtonian lubricant
- Both contacting solids are coated
- Substrates of interacting bodies share similar mechanical properties (Young's modulus and Poisson's ratio), i.e.:

$E_{s,1} = E_{s,2}$ & $\nu_{s,1} = \nu_{s,2}$

- Coatings of interacting bodies share similar mechanical properties, i.e.:

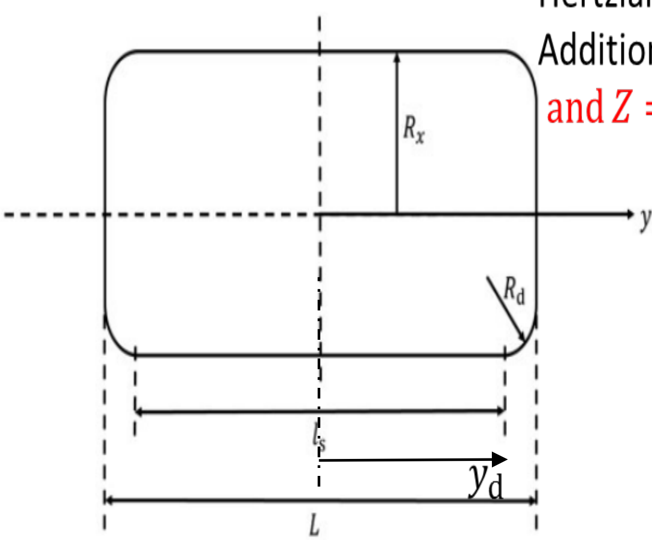
$E_{c,1} = E_{c,2}$ & $\nu_{c,1} = \nu_{c,2}$



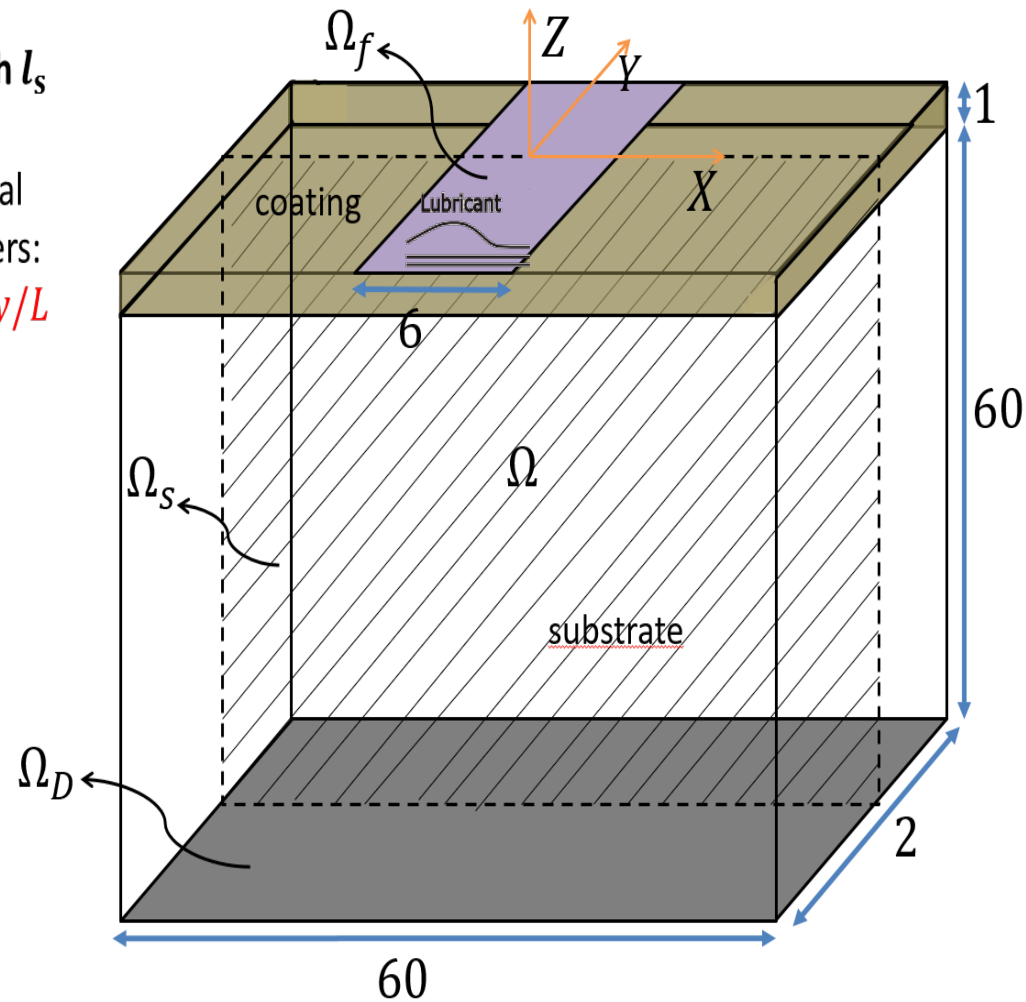
Modelling II: computational domain

- Considered case: A coated roller with straight length l_s and rounded corners (with dub-off radius R_d) on a coated plate

Scaling using typical Hertzian parameters: Additionally, $Y = y/L$ and $Z = z/t_c$



$$h = h_0 + \frac{x^2}{2R_x} + \frac{(y - y_d)^2}{2R_d}(y > y_d) - \delta$$



Modelling II: model features

- Hydrodynamic problem:
  - Reynolds equation → pressure distribution
    - Fully flooded conditions
    - Free boundary problem→ penalty formulation
    - Lubricant piezo viscous behavior & compressibility → Roelands and Dowson-Higginson rheological expressions
    - Suitable numerical stabilization techniques employed at high loads → SUPG+ isotropic diffusion

- Elastic problem:
  - 3D linear elasticity equations→ displacement field
    - Equivalent elasticity problem: Avoids calculating the deformation for the contacting bodies twice
    - Automatic continuity of stress at common interface of coating and substrate

- The developed model is solved using the Finite element method (FEM) in a fully coupled framework; The developed model here is similar to the one presented by Habchi et al[1], but then modified to consider a finite length line contacts.

[1]Habchi, Wasim, et al. "A full-system approach of the elastohydrodynamic line/point contact problem." *Journal of Tribology* 130.2 (2008): 021501.

Results: Uncoated case

Results: reference operating conditions
Uncoated case

- Note that the particular case when the Young's moduli of coating and substrate are identical, i.e. $E_c = E_s = E$, corresponds to an uncoated contact.

Parameter	value	unit	
η_0	0.013	Pa·s	
U_m	1	m/s	
R_x	0.008	m	
L	0.01	m	
R_d	0.127	m	
l_s	0.0085	m	
U	7.3891E−12	–	speed parameter
W	1.7904E−4	–	load parameter
G	4150	–	material parameter
p_h	1.17	GPa	
Y_d	0.85	-	
R_d/L	12.7	-	

Parameters to be varied:

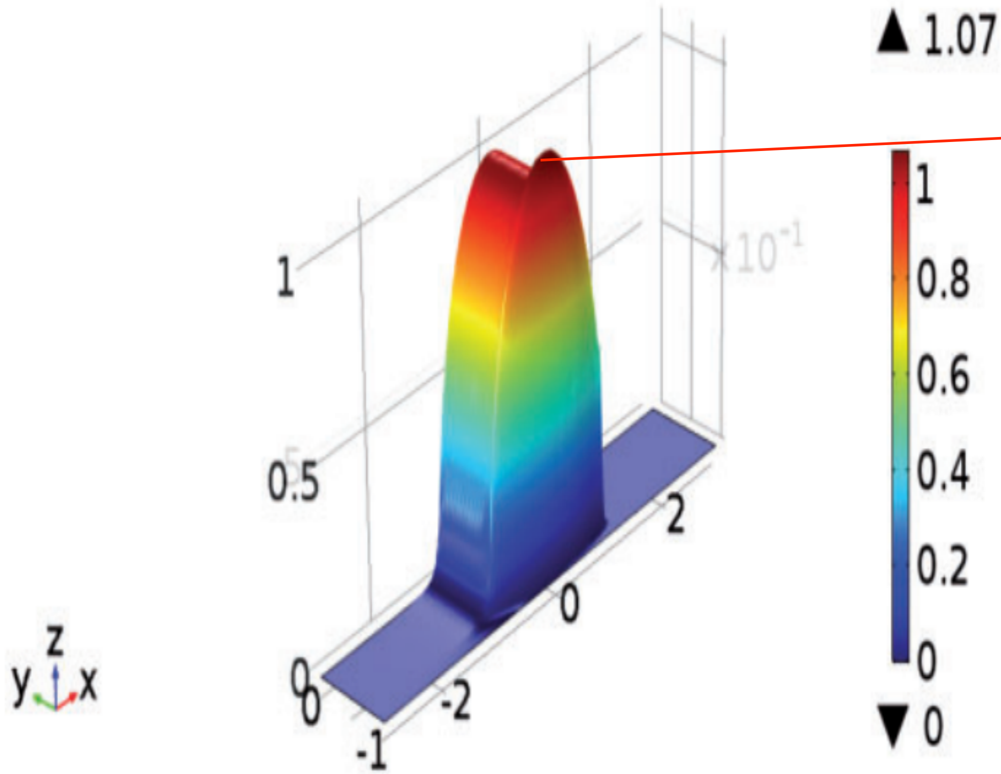
$$U = \frac{2\eta_0 U_m}{E' R_x}$$

$$W = \frac{F}{E' R_x L}$$

$$G = E' \alpha$$

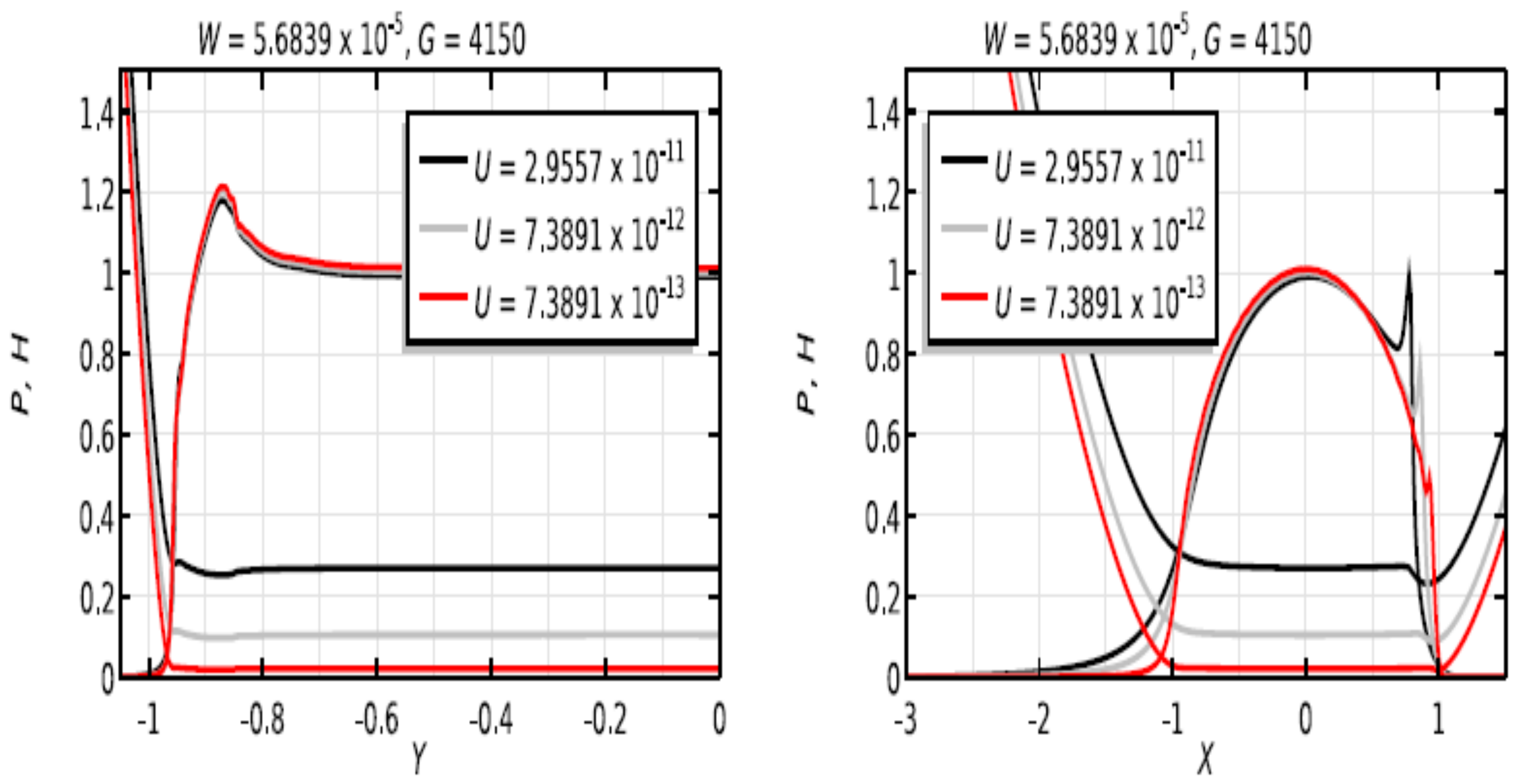
$$Y_d = \frac{l_s}{L}$$

Results: reference case: pressure distribution

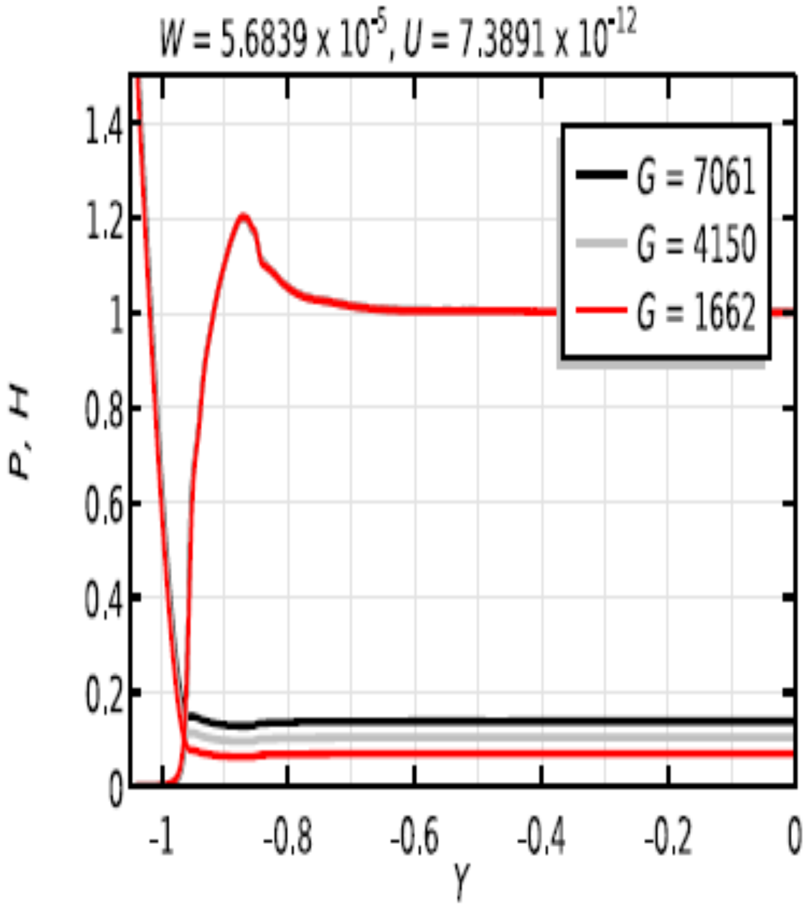


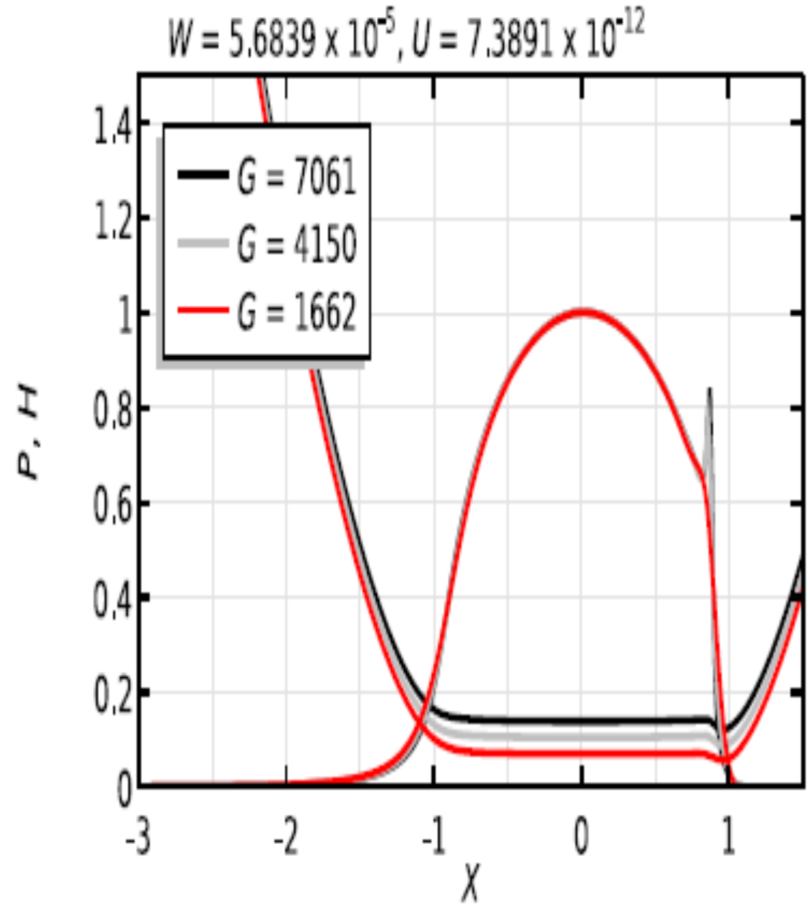
Secondary pressure peak

Results: Variation of Velocity U_m Uncoated case

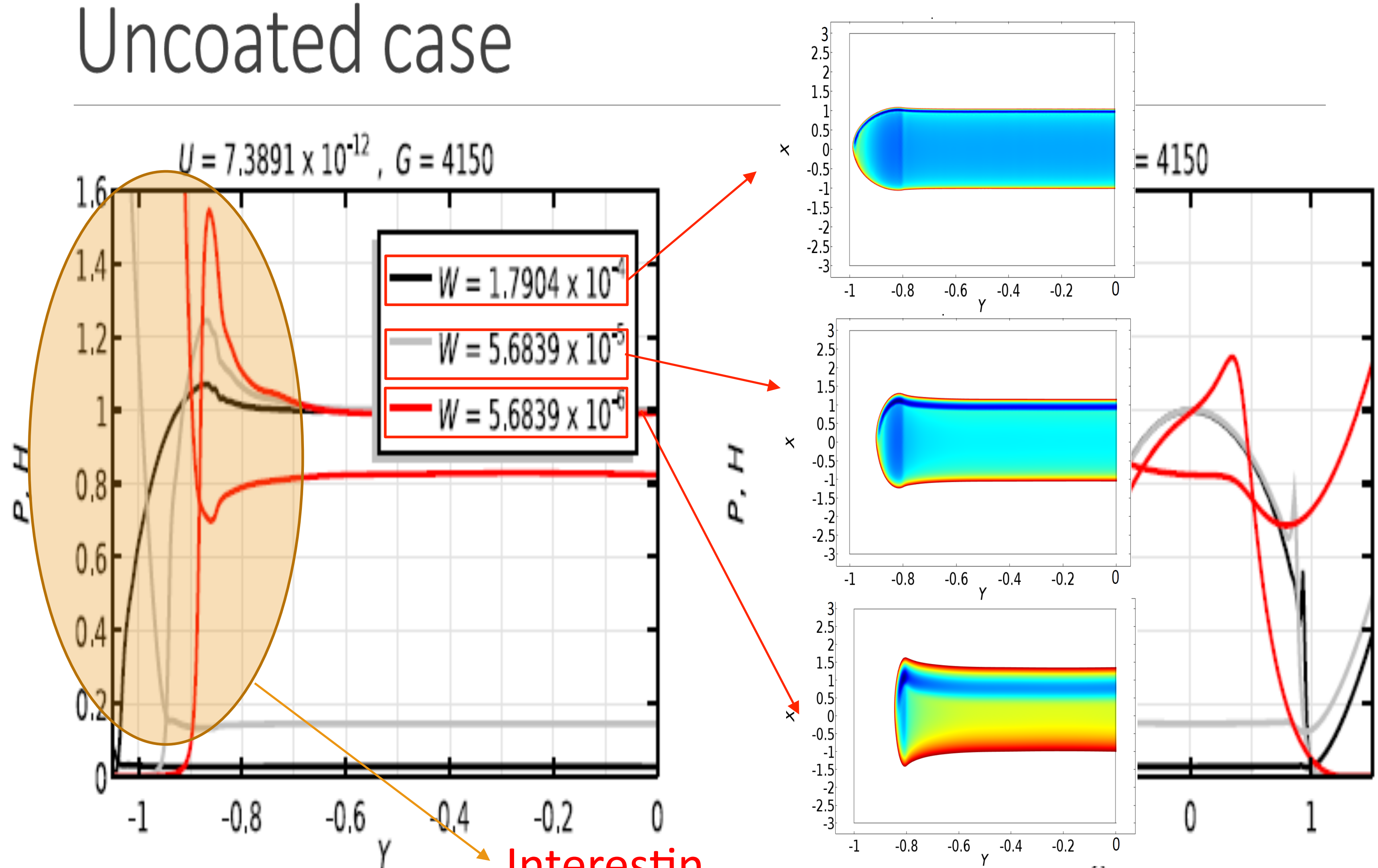


Results: Variation of elasticity modulus E
Uncoated case





Results: Variation of Load F
Uncoated case



Interestin
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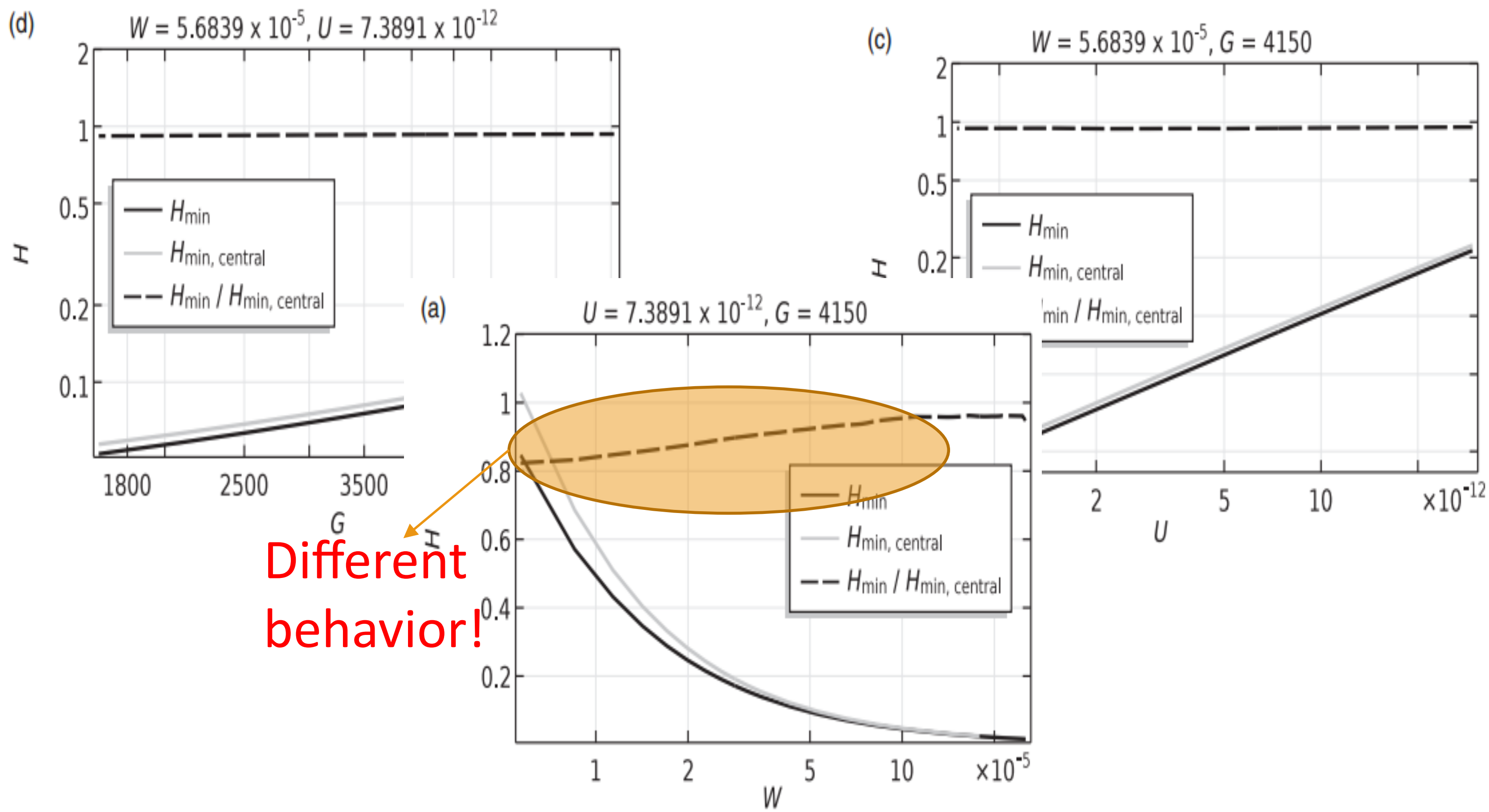
Remarks: Variation of operating conditions
Uncoated case

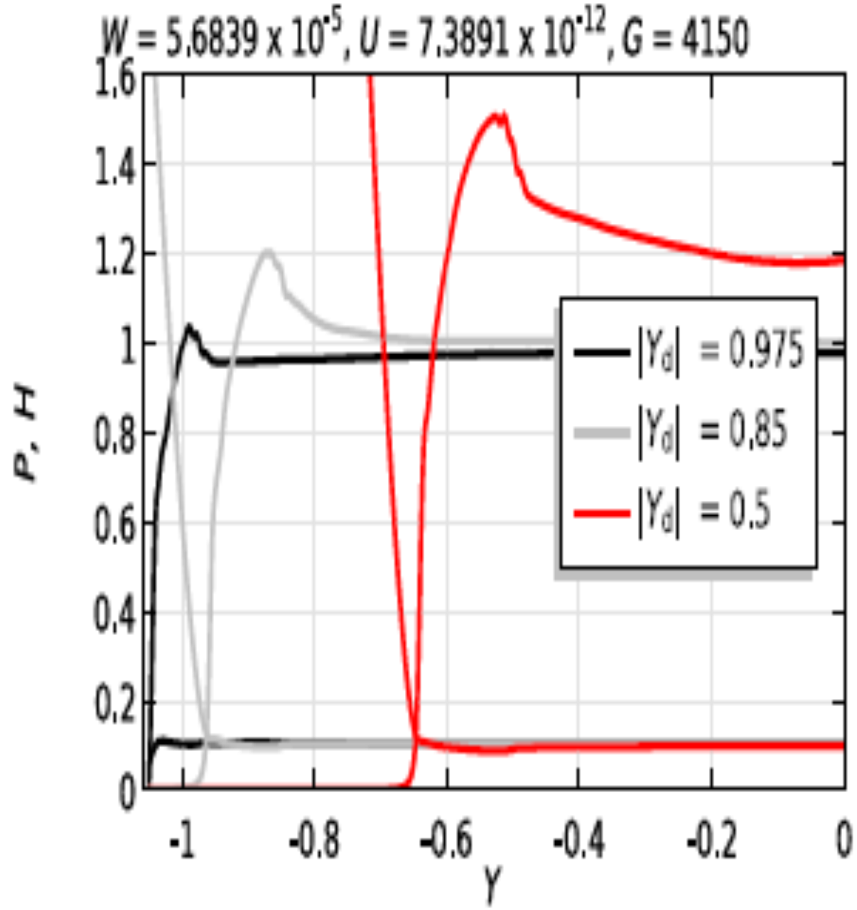


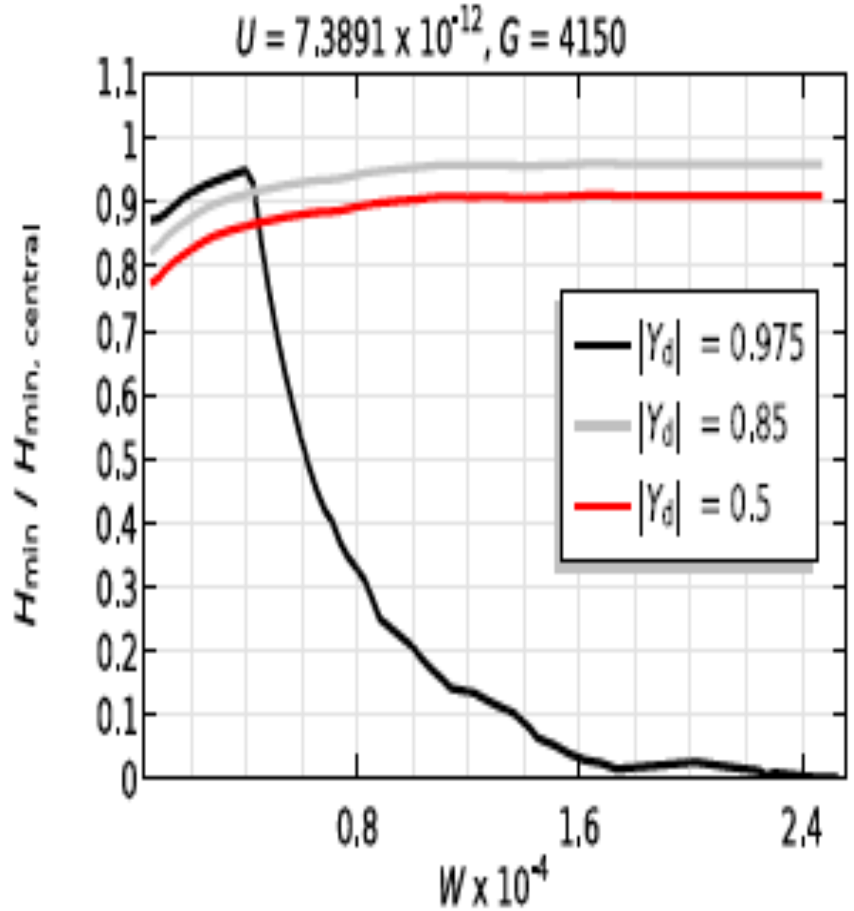
Remarks: Variation of Load
Uncoated case

- We see that the finite line contact behavior is highly affected by variations in load, unlike traditional well described contacts such as elliptical and circular contacts!!

- Notice that the contact area increases with increasing load→ Secondary pressure peak decreases!

Results: Variation of roller straight length l_s
Uncoated case





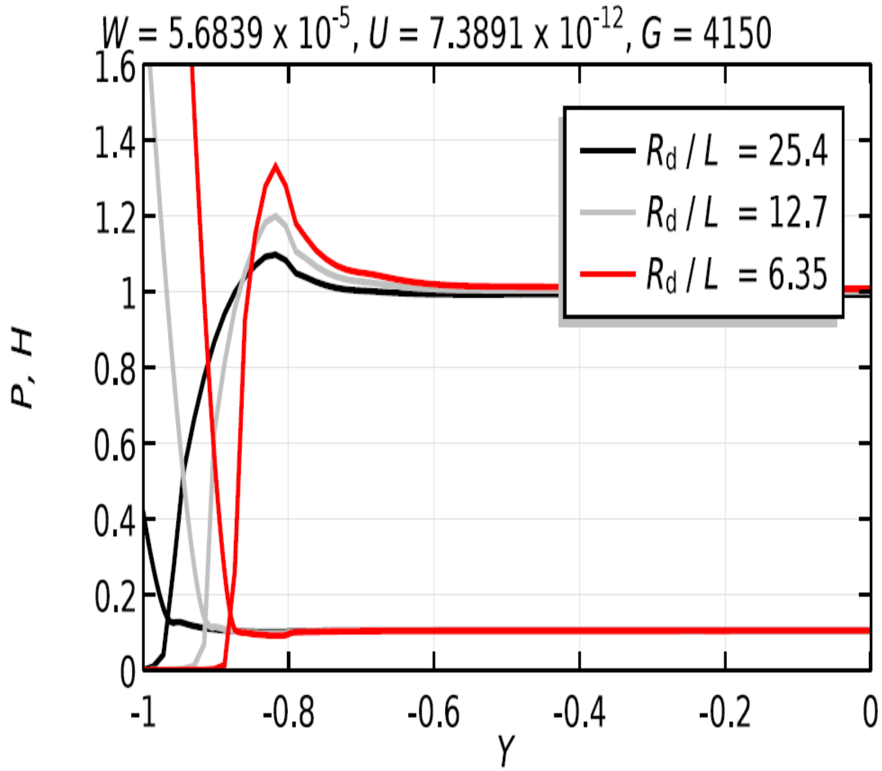
Remarks: Variation of Load Uncoated case

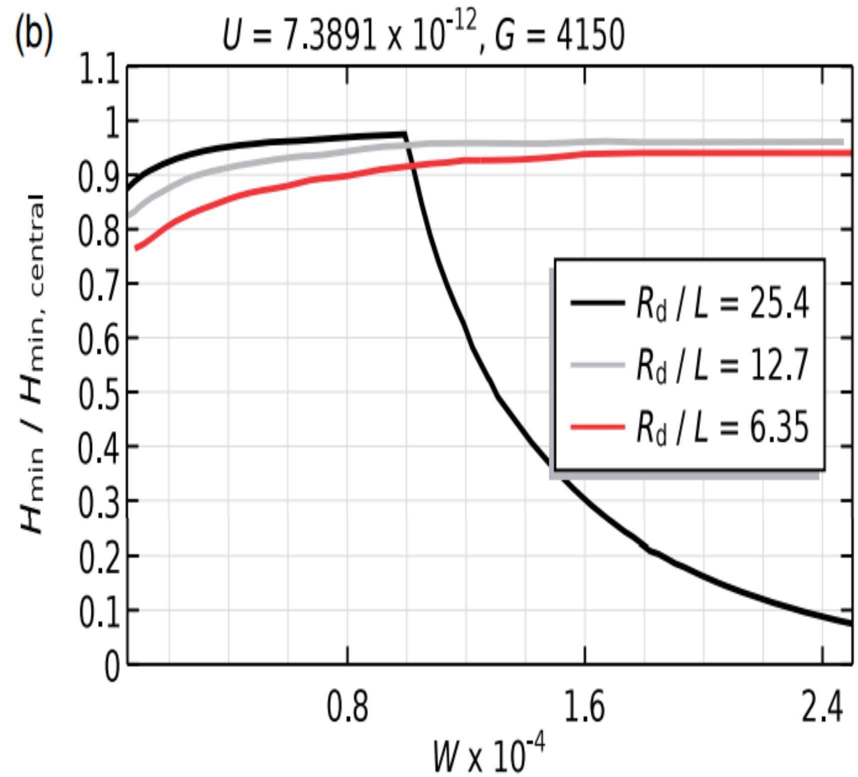
- Notice that the contact area increases with increasing load→ Secondary pressure peak decreases!

- …..Uptill a certain point where there no space available to allow for further increase in contact area→ Secondary pressure peak will increase again to compensate for increasing load!!! This will result to decrease in absolute minimum film thickness!

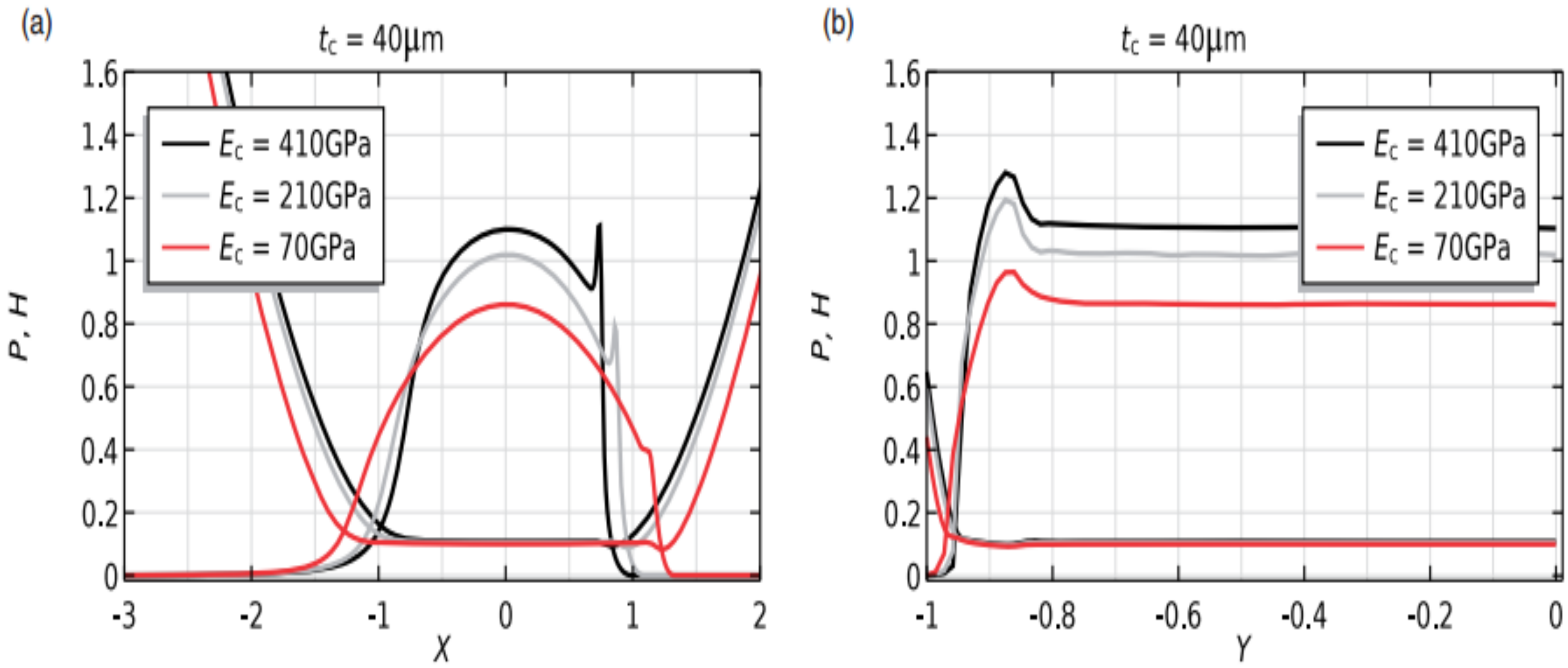
Results: Coated case

Results: Variation of roller dub-off radius R_d Uncoated case

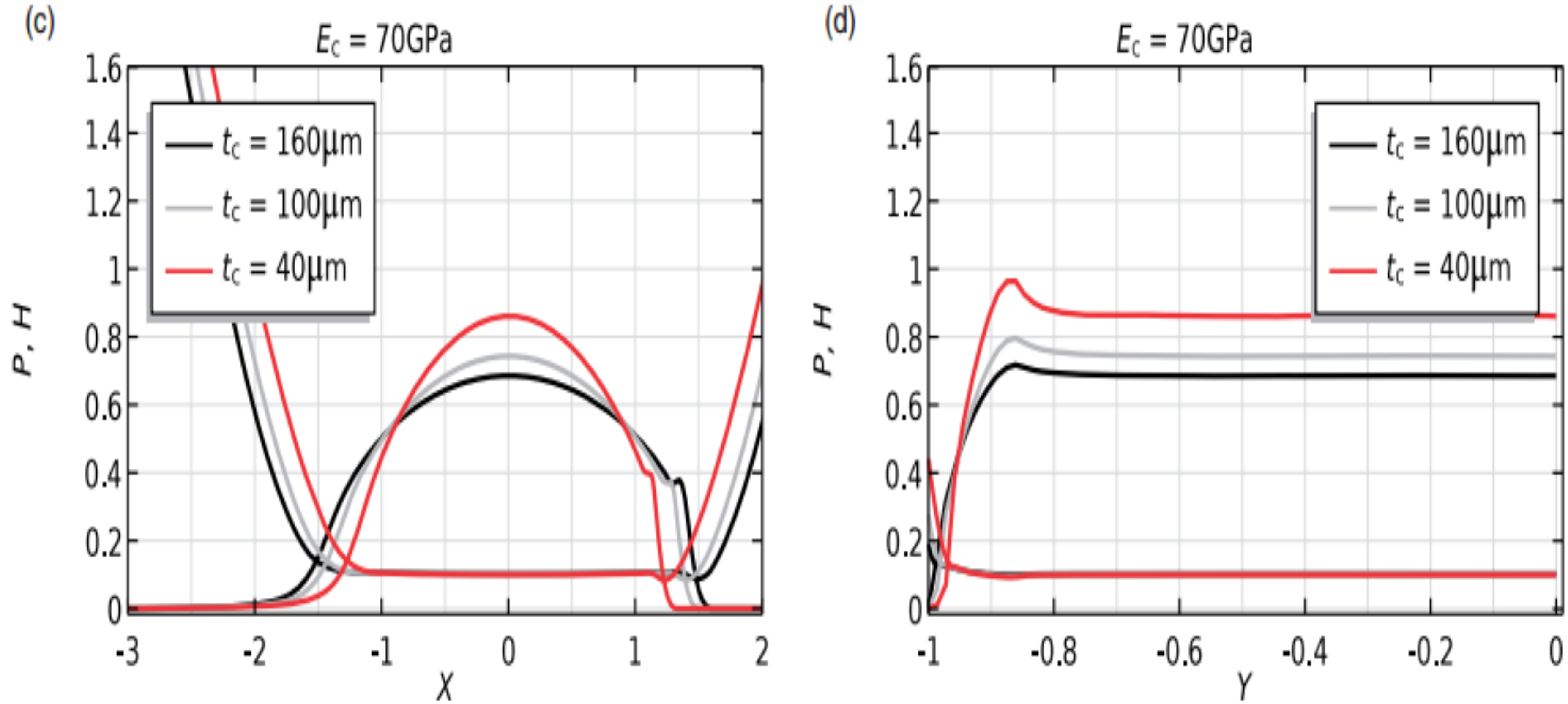




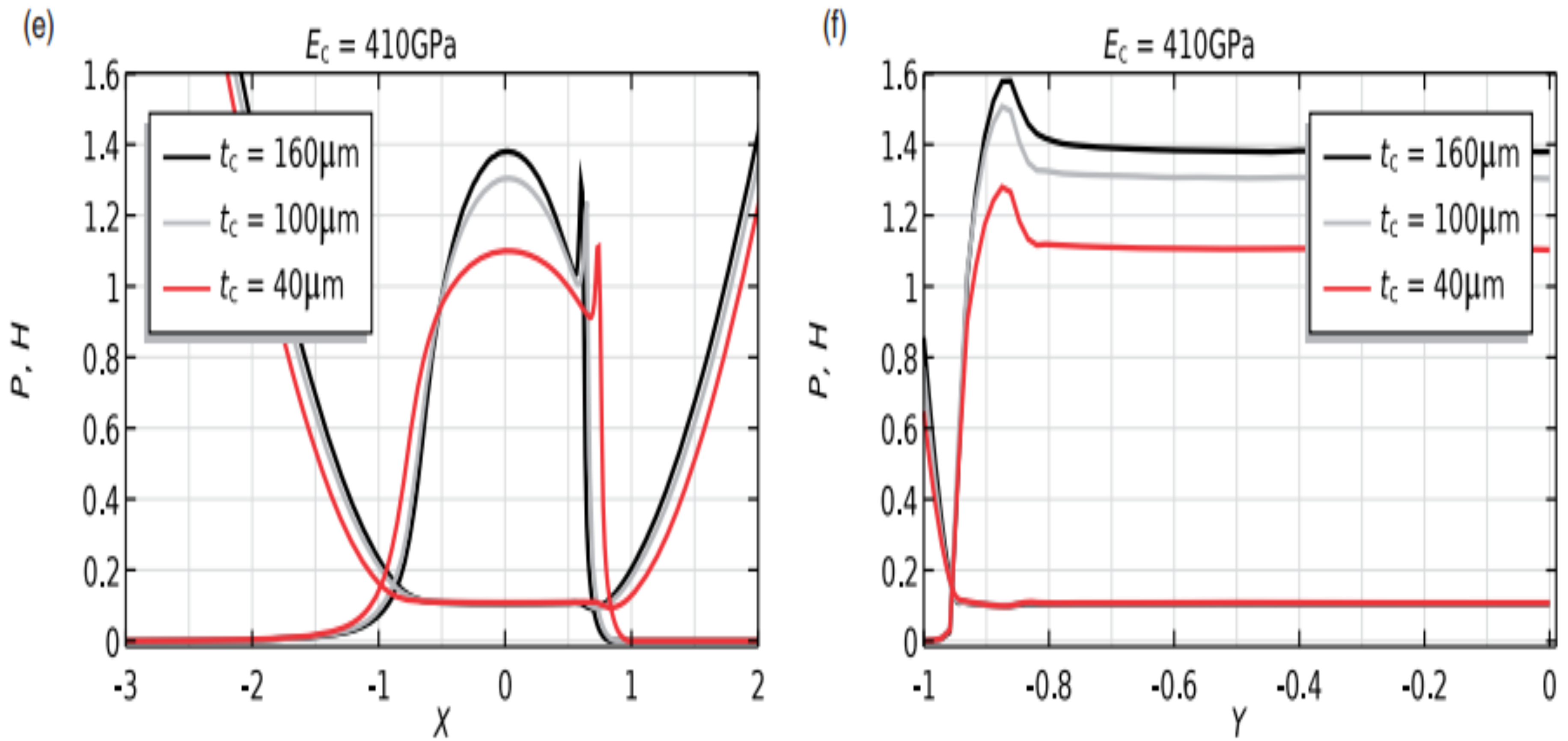
Results:
Variation of coating elasticity modulus E_c



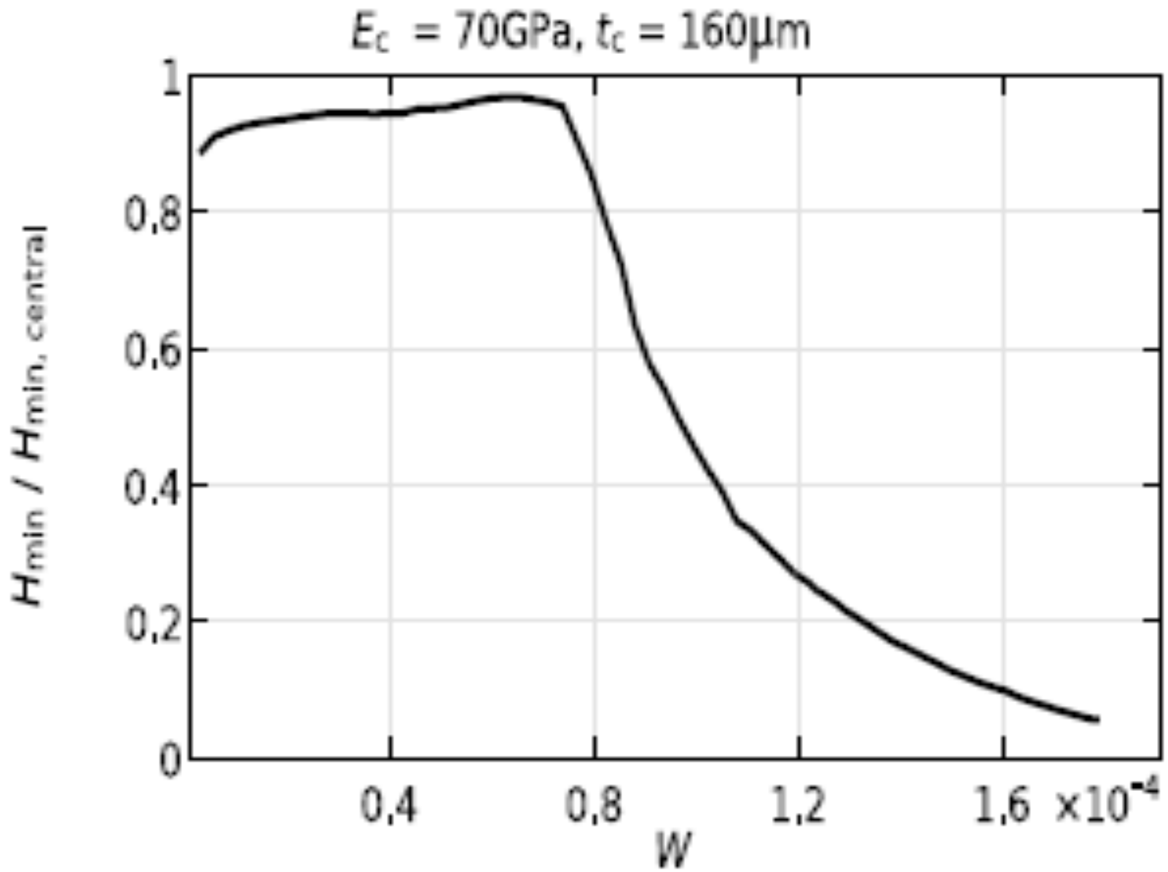
Results: Flexible coatings
Variation of coating thickness t_c



Results: Stiff coatings
Variation of coating thickness t_c



Results: Soft & thick coatings
Variation of with load F



Conclusing remarks

- In line with previously reported findings, it is shown that the pressure and film thickness distributions for finite line contacts vary significantly different with applied load as compared to infinite line contact models.
  - At increasing loads, the pressure distribution becomes more uniform in axial direction as long as there is space available for contact area expansion. When no space is left to compensate for higher loads, the secondary pressure peak at the extremities grows again and hence the absolute minimum film thickness decreases.

- Large round corner radii, large straight roller lengths, too elastic and thick coatings, all amplify the effect of increasing loads.

- All these findings make it really hard to develop a robust correlation between absolute minimum film thickness, maximum pressure and operating conditions, as coating and axial profile geometrical parameters play an equally important role. Numerical analysis becomes inevitable…..

Conclusing remarks

Interested readers are welcome to read our recently (online) published paper regarding this topic:

Alakhramsing, S. S., de Rooij, M. B., Schipper, D. J., & van Drogen, M. (2017). Elastohydrodynamic lubrication of coated finite line contacts. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, https://doi.org/10.1177/1350650117705037.