Surface Life Challenges in Machine Elements

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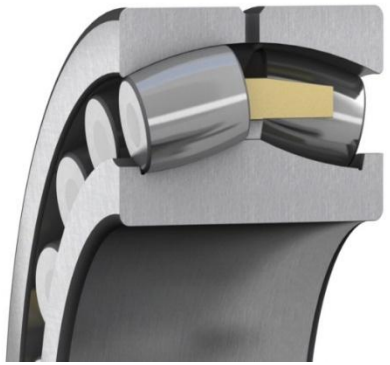
Contents

A Changing World for Tribological Surfaces

A Changing World for Rolling Bearings



Max Pressure ~ 2-4 GPa (weight of 10 cars on a fingernail)

Load cycles ~ 1x 10^{11} $(no.\ stars\ in\ 1000\ Galaxies)$

Film Thickness ~ 100 nm (~ 2 piled up viruses)

Sliding ~ 0-5% (~ 4 m/s)

Max Pressure ~ 1-2 GPa (weight of 1.5 cars on a fingernail)

Load cycles ~ 1x 10^{8} $(no.\ stars\ in\ 1\ Galaxies)$

Film Thickness ~ 300 nm (~ 6 piled up viruses)

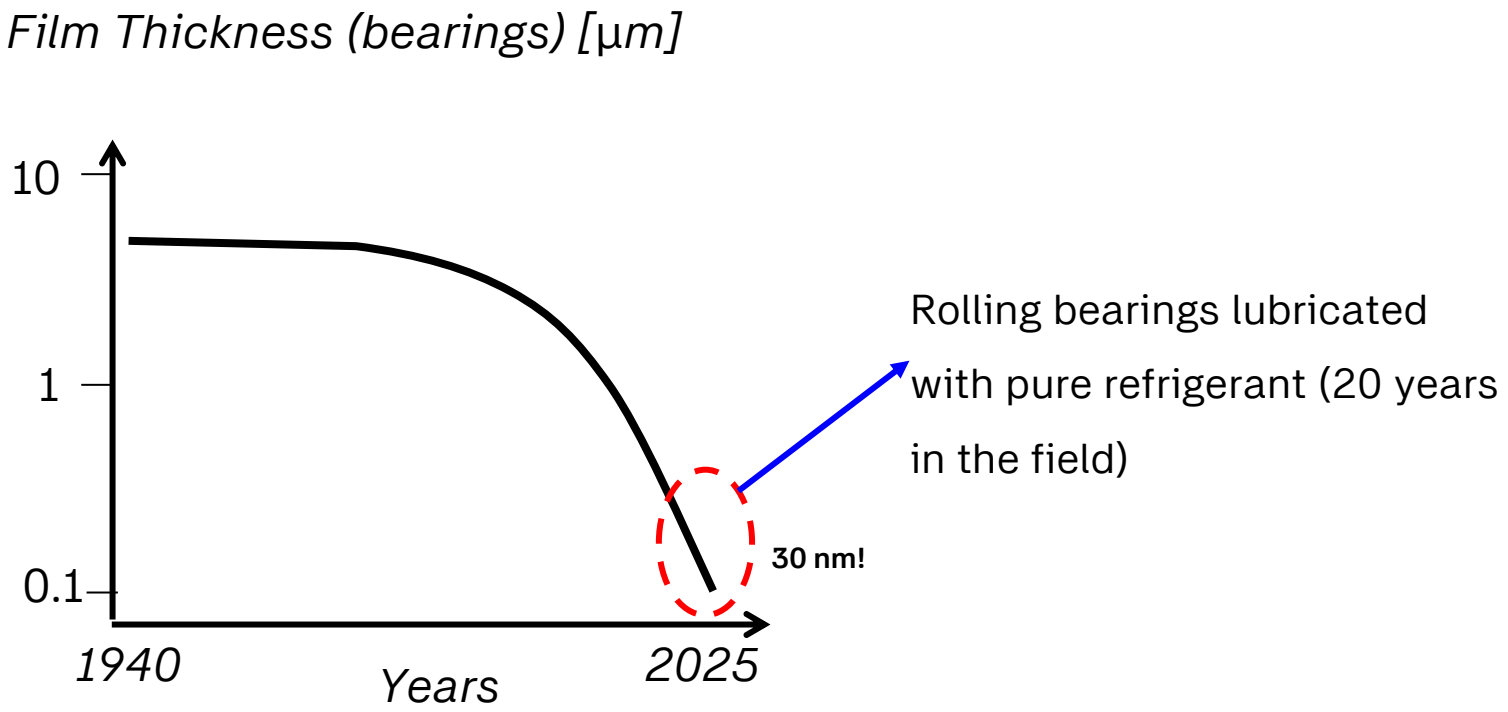
Sliding ~ 0-30% (~ 20 m/s)

Industry Megatrends

Electrification	New lubricants, quiet, faster, hotter, more load, new failure modes
Energy Efficiency	Poorer lubrication, faster, hotter
Speed (Low and High)	Low: Poor lubrication, wear, surface fatigue High: Starvation, adhesive wear

(*) Morales-Espejel, G.E., Surface Life Modelling of Tribological Components – From Surface Roughness to Bearings and Gear Life, Tribology Online, Vol. 18, No. 6 (2023) 255-267.

The Thinning Film Thickness in Rolling Bearings



Extending Dowson's Book*

*Dowson, D, History of Tribology. Logman, New York, NY, USA, 1999

Rolling Bearing Trends: Aero-Engines

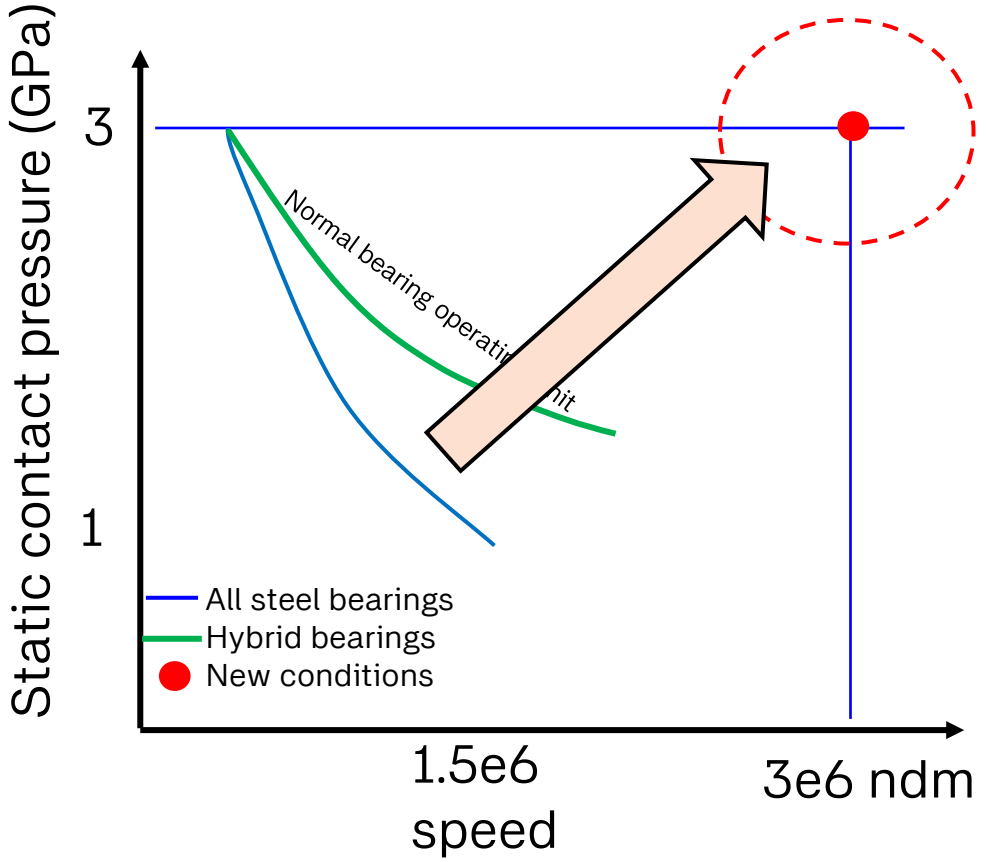
Challenges:

--- High temperatures (~ 300^0C)
--- High loads
--- High speeds ndm ~3 x10^6

n = rpm

dm = bearing mean diameter in mm

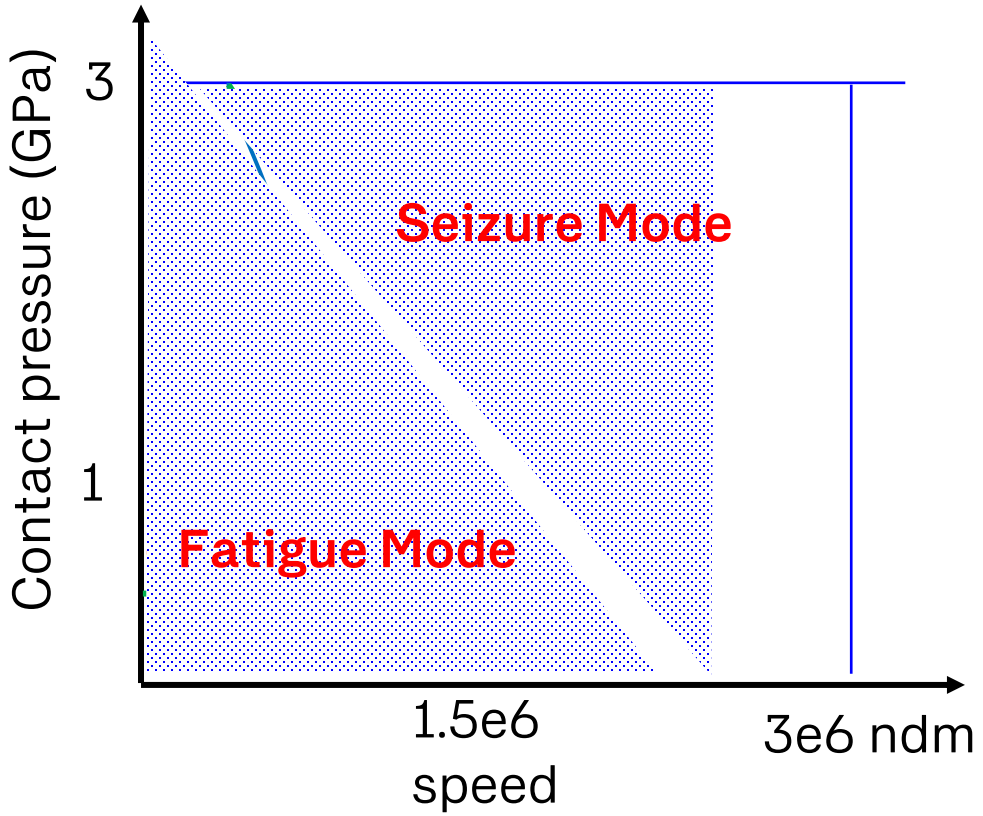
Rolling Bearing Trends: Machine Tool Spindles



Challenges:

--- Stiffness (accuracy)
--- Cleanliness
--- High speeds + High loads

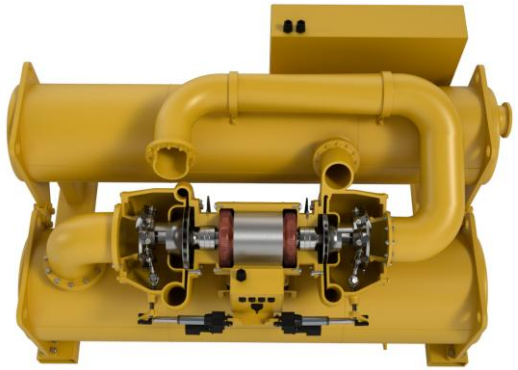
Competition of Fatigue with Frictional Heating



H. Czichos and K. Kirschke. Investigation into Film Failure (Transition Point) of Lubricated Concentrated Contacts. *Wear,* 22, pages 321–336, 1972.

Rolling Bearing Trends: Compressors and Pumps

Refrigeration Compressors



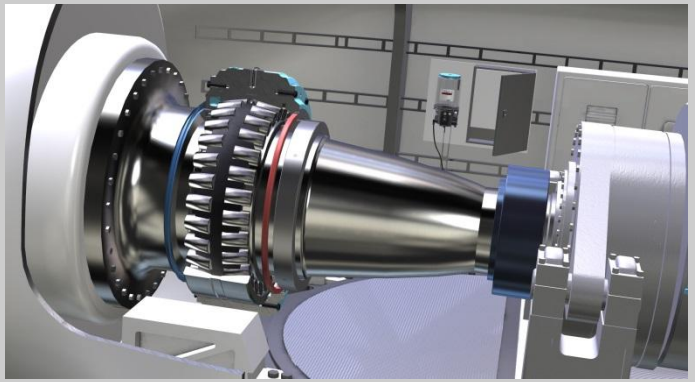
Industrial Heat Pumps

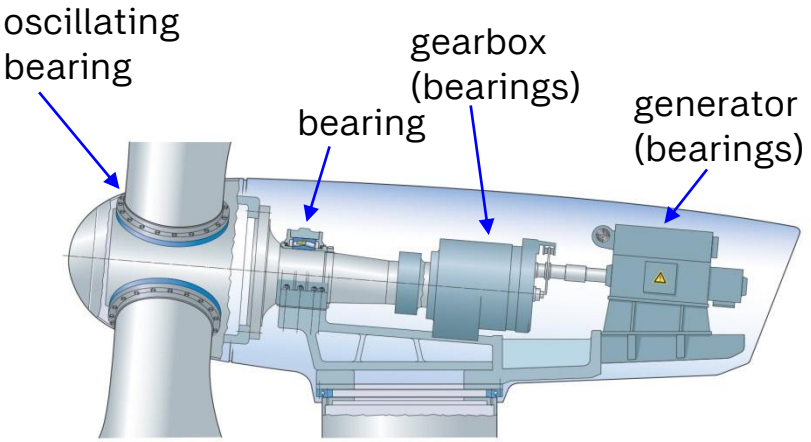
- Temperature: 30^{0}C - 50^{0}C
- Refrigerant dilution: <30 %
- Contact pressures: ~1.2 GPa
- Corrosive refrigerants

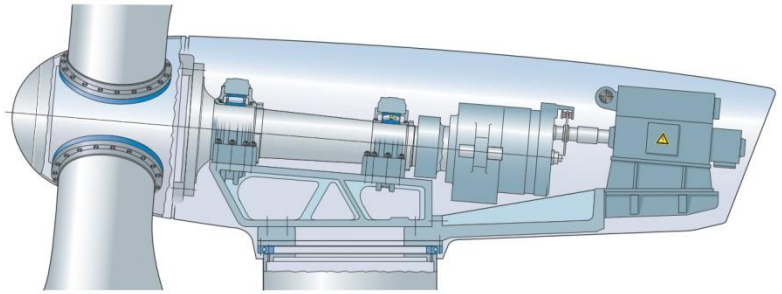
- Temperature: 100^{0}C - 200^{0}C
- Refrigerant dilution: <60 %
- Contact pressures: ~1.5 GPa
- More corrosive refrigerants

Rolling Bearing Trends: Wind Turbines

Wind turbine with gearbox





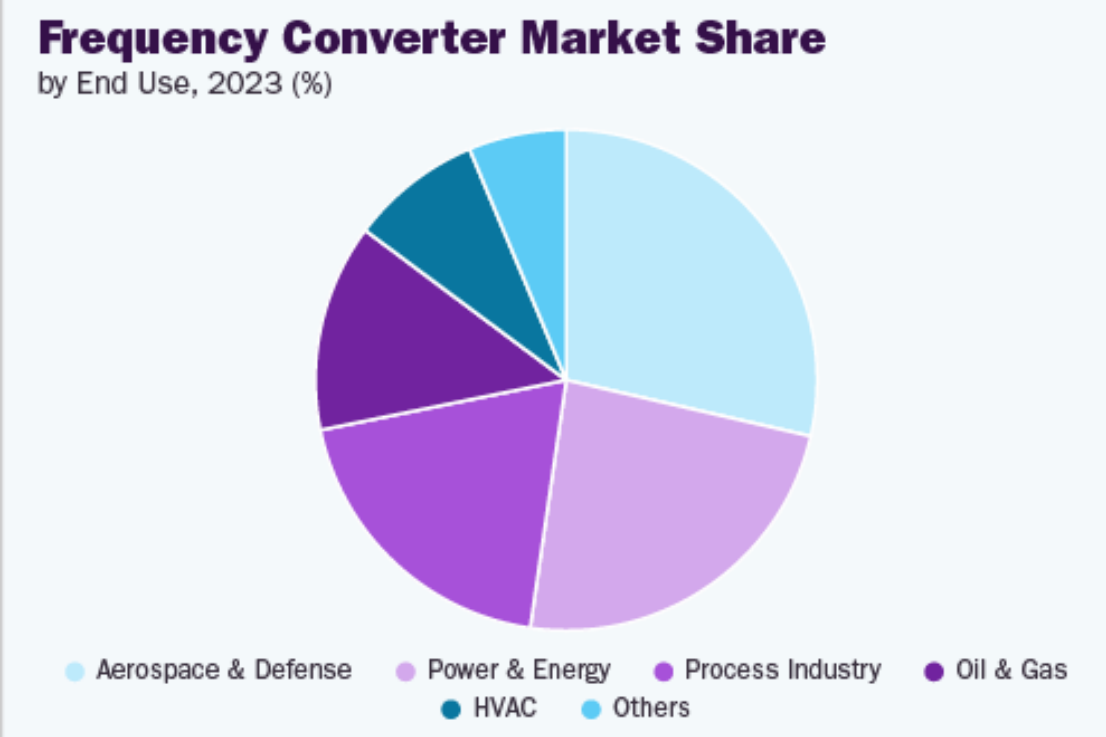


Challenges:

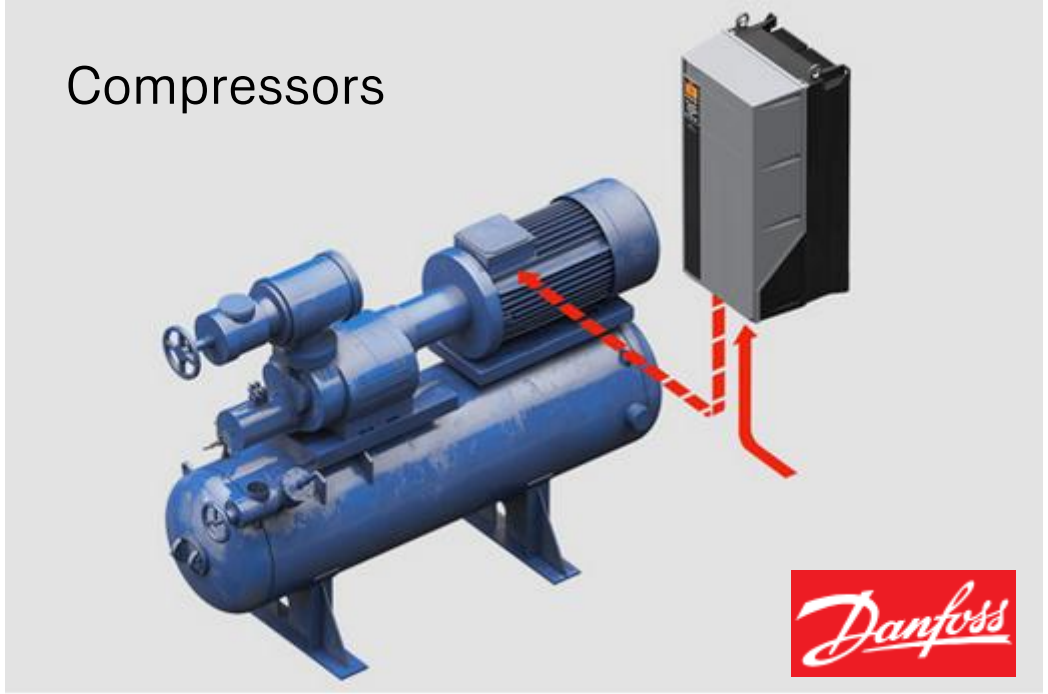
- For gearboxes: Dynamic loads, poor lubrication, particle contamination
- For main shaft: Structural stresses, fretting (creep), poor lubrication (grease)
- Slewing bearings: Wear, fretting

Electrification beyond Mobility

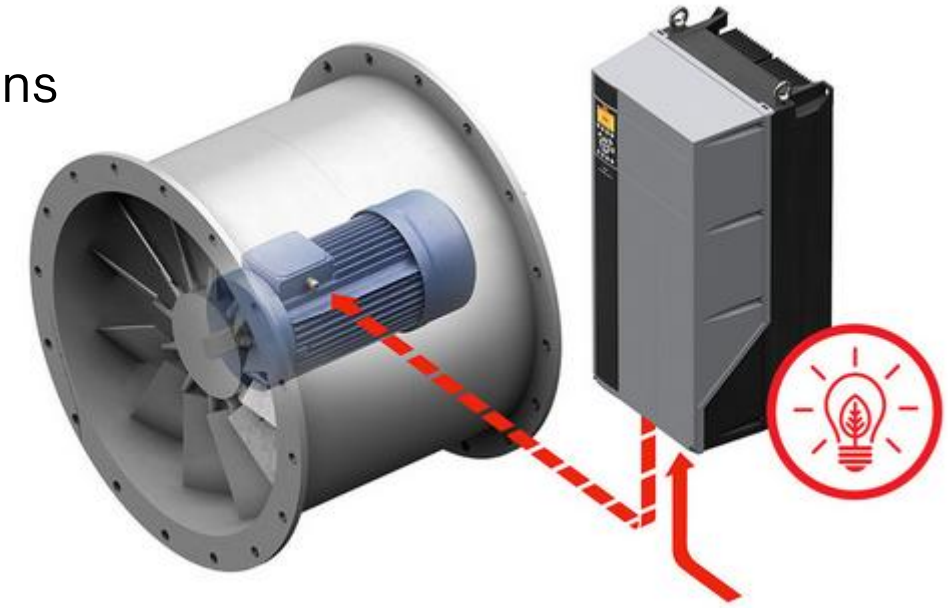
Frequency Converters



Compressors



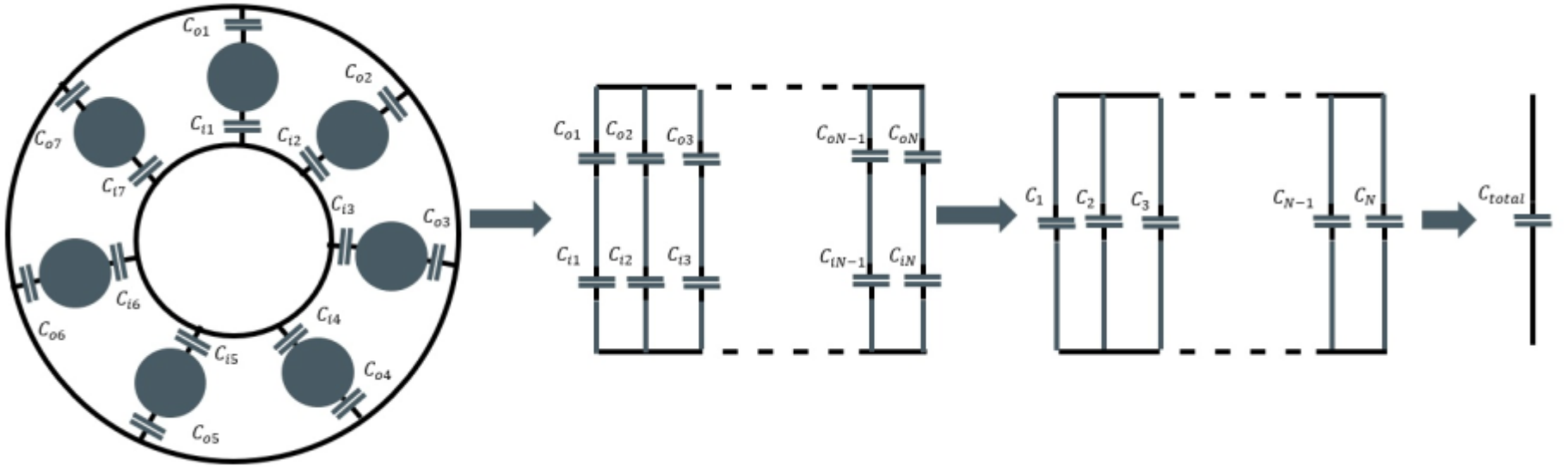
Fans

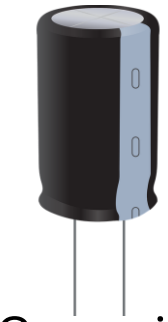


HVAC = Heating, Ventilation and Air Conditioning

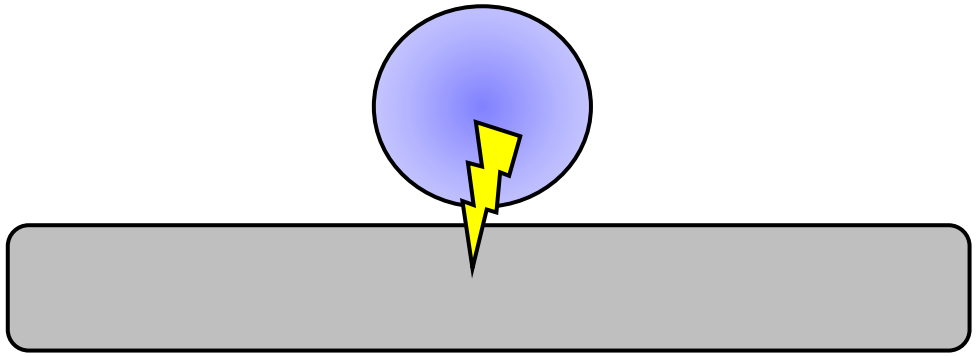
(*) Frequency Converter Market Size, Share & Trends Analysis Report By Type (Static, Rotary), By Phase (Single Phase, Three Phase), By End Use, By Region, And Segment Forecasts, 2024 – 2030
Grand View Report ID: GVR-4-68040-387-5

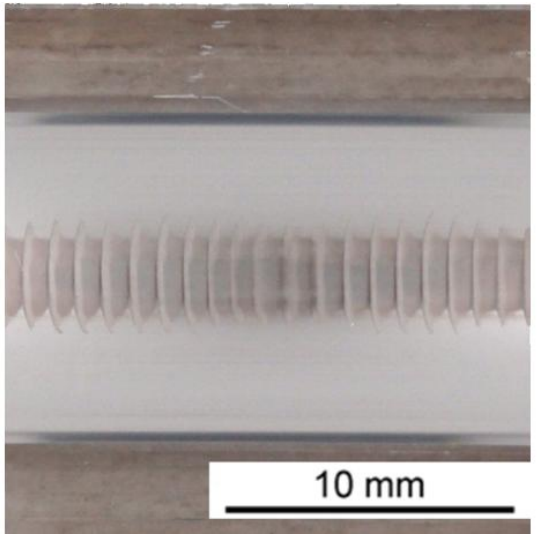
Electrical Damage in Bearings

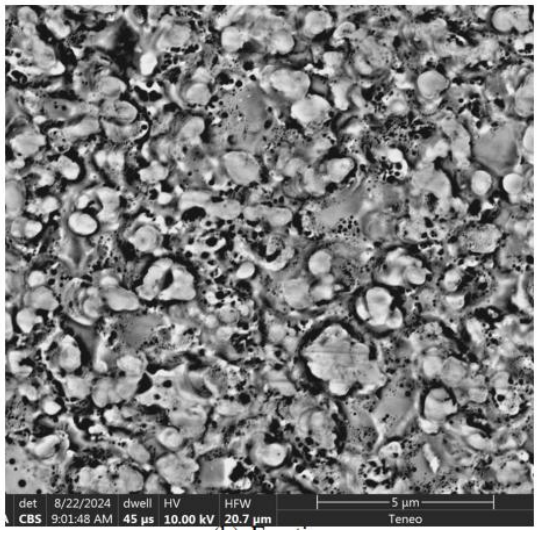




Capacitor

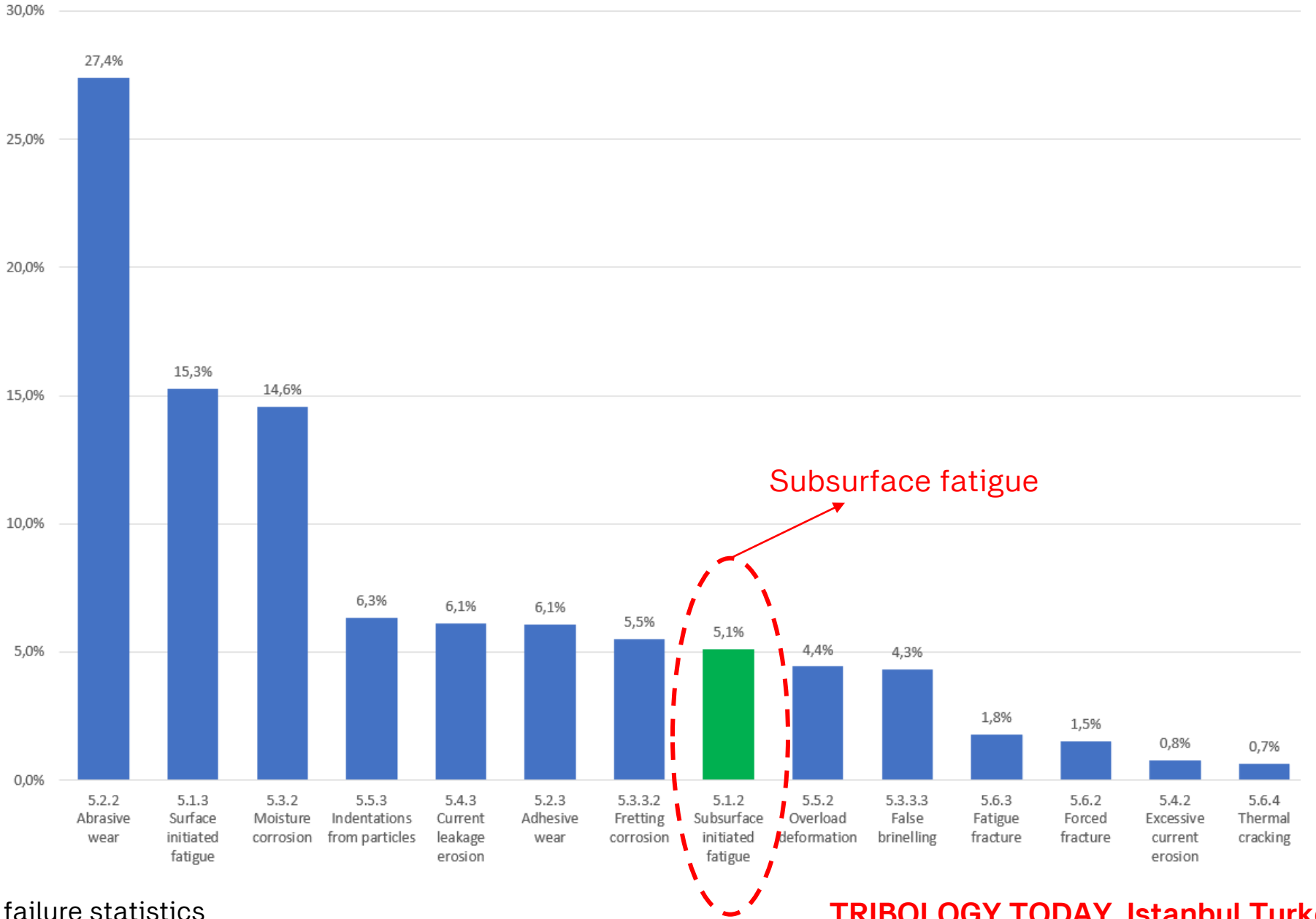






Tribology Modelling of Emerging Failure Modes

SKF Modern Failure Statistics in Bearings



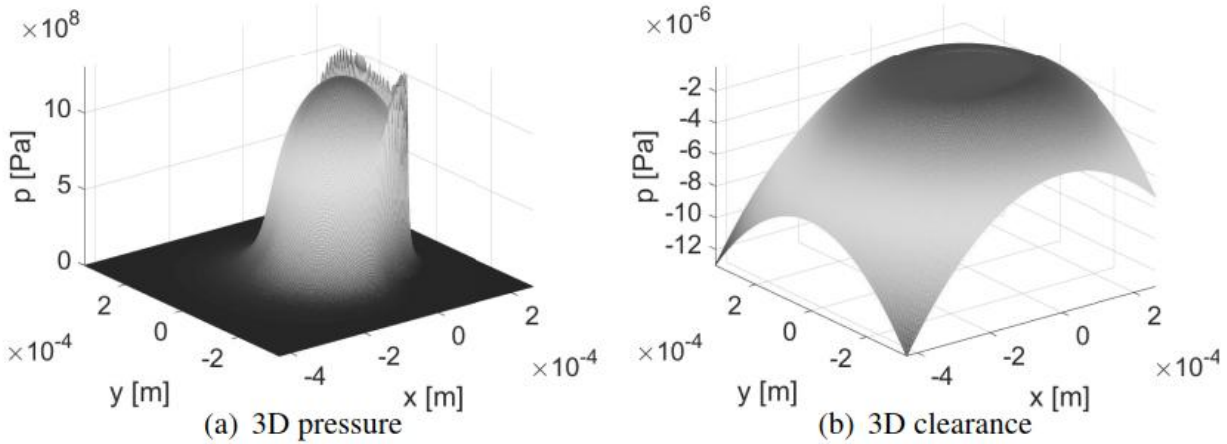
Source: SKF field failure statistics

Lubrication

Numerical Solution of EHL

Classical Solution Methods:

- Multigrid (Lubrecht and Venner)
- FEM Full System (Habchi et al.)
- Others



- Reynolds Equation:

$$\underbrace{\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial y}\right)}_{\text{Poiseuille flow}}=\underbrace{12\bar{u}\frac{\partial(\rho h)}{\partial x}}_{\text{Couette flow}}+\underbrace{12\frac{\partial(\rho h)}{\partial t}}_{\text{squeeze film term}}$$

- Film Thickness Equation:

$$h(x,y)=\underbrace{h_0}_{\text{constant approach}}+\underbrace{\frac{x^2}{2R_x}+\frac{y^2}{2R_y}}_{\text{parabolic}}+\underbrace{\frac{2}{\pi E'}\int_{-\infty}^{\infty}\int_{-\infty}^{\infty}\frac{p(x',y')}{\sqrt{(x-x')^2+(y-y')^2}}}_{\text{elastic deformation}}$$

- Load Equilibrium Equation:

$$Q=\int_{-\infty}^{\infty}\int_{-\infty}^{\infty}p(x,y)dxdy$$

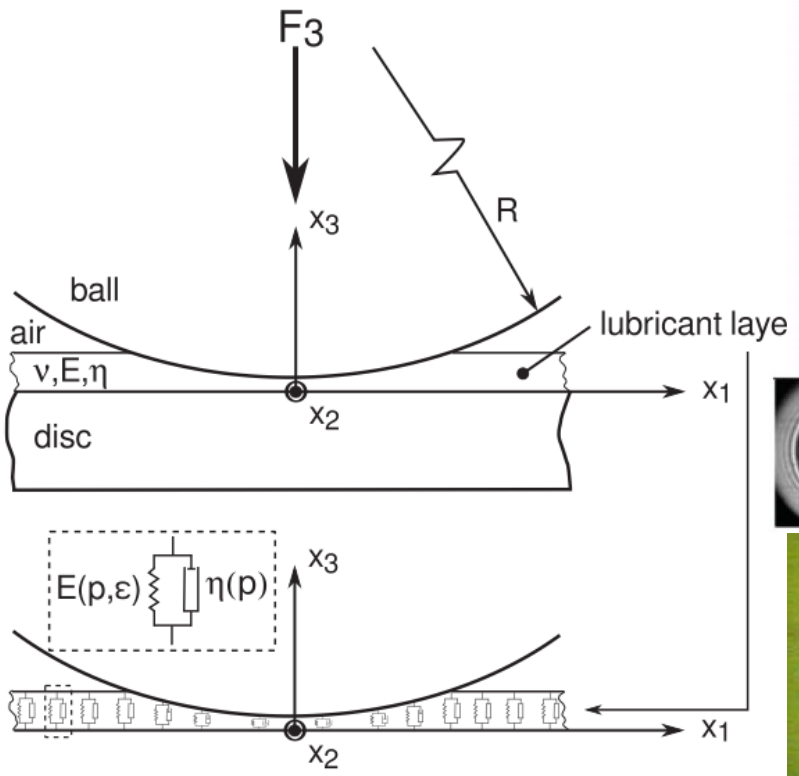
- State Equation for Lubricant

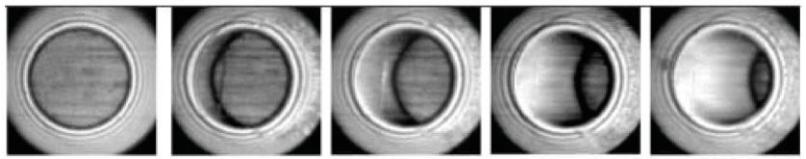
$$\rho(p)=f_\rho(p)\qquad\eta(p)=f_\eta(p)$$

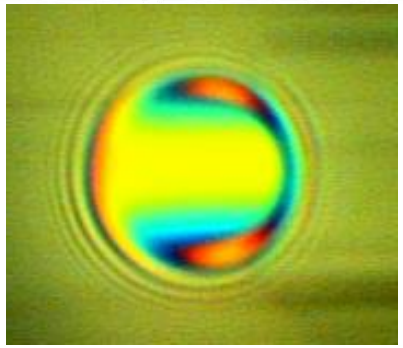
$$G=\alpha E',\quad W=\frac{Q}{(E'R_x^2)},\quad U=\frac{\bar{u}\eta_0}{E'R_x},\quad H_D=h/R_x,\quad P_D=p/E'$$

Exploring New Ways of Modelling Lubrication

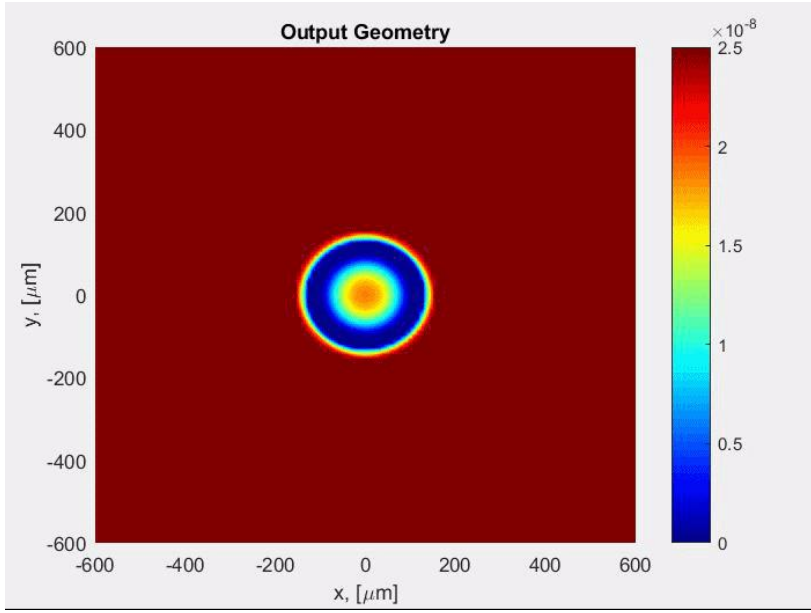
Towards proper modelling of “mixed-Lubrication”

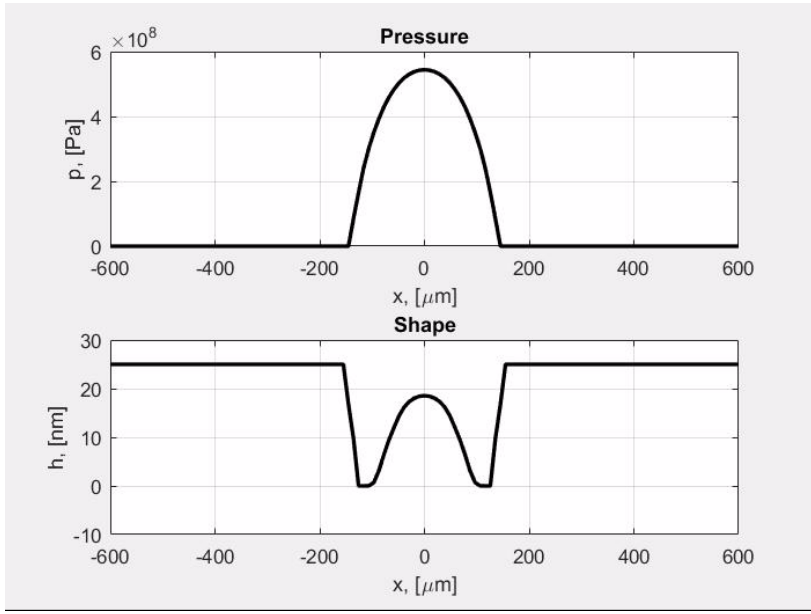






Glovnea & Spikes





* van Emden, E.,Venner, C.H.,Morales-Espejel, G.E., “Investigation into the viscoelastic behaviour of a thin lubricant layer in an EHL contact” , Trib. Inter., vol. 111, pp. 197-210, 2017

* Glovnea, R. P., Spikes, H. A.,, “Elastohydrodynamic film formation at the start-up of the motion” , Proc. IMechE, part J., vol. 215, pp. 125-138, 2001

Inlet Analysis (Ertel-Grubin) - A Reminder

Consider the line contact steady-state Reynolds equation:

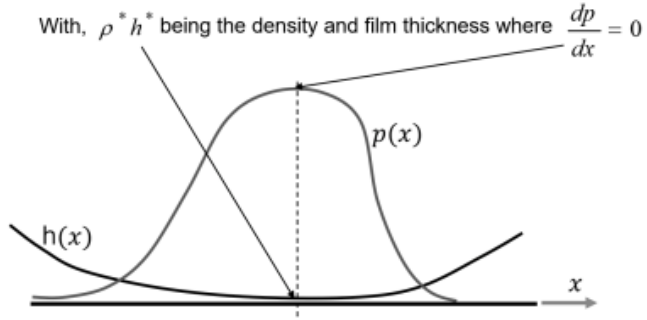
$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) = \bar{u}\frac{\partial(\rho h)}{\partial x}$$

Therefore, integrating one time in x:

$$\int_0^{\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)} \partial\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) = \bar{u}\int_{\rho^* h^*}^{\rho h}\frac{\partial(\rho h)\partial x}{\partial x}$$

$$\frac{dp}{dx} = 12\eta\bar{u}\frac{\rho h - \rho^* h^*}{\rho h^3}$$

with ρ^* and h^* being the values of density and film thickness where $dp/dx = 0$



Barus law: $\eta(p) = \eta_0 \exp(\alpha p)$

introduced, named "reduced pressure", $\hat{q}$:

$$\hat{q} = [1 - \exp(-\alpha p)]/\alpha$$

which is now substituted

$$\frac{d\hat{q}}{dx} = 12\eta_0\bar{u}\frac{\rho h - \rho^* h^*}{\rho h^3}$$

Introducing dimensionless groups:

$$X = x/a, \quad P = p/p_0, \quad \hat{H} = 2R_x h/a^2, \quad \hat{Q} = \hat{q}/\hat{q}_0, \quad K = \frac{48\bar{u}\eta_0 R_x^2}{a^3 p_0}$$

Notice that the dimensionless pressures can be recovered from $P = -\frac{1}{K_1}\ln(1 - K_1\hat{Q})$, with $K_1 = \alpha p_0$.

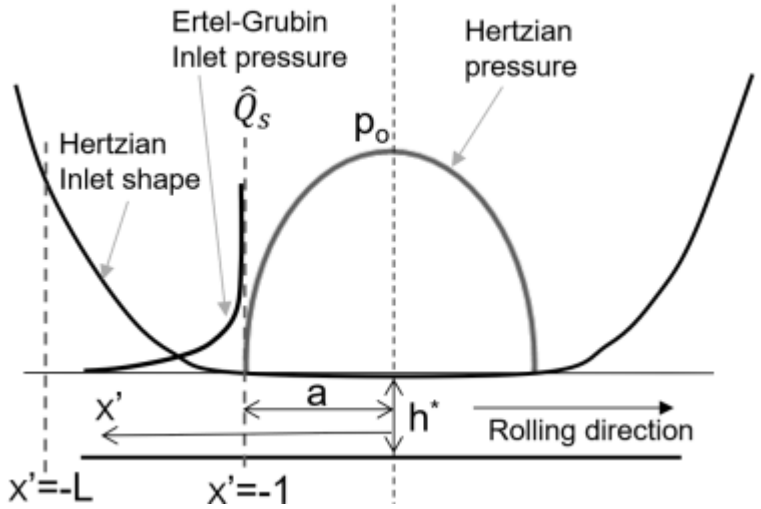
The Reynolds equation can be rewritten:

$$\frac{d\hat{Q}}{dX} = K\frac{\hat{H} - \hat{H}^*}{\hat{H}^3}$$

Integrating .....

Grubin, A. N., "Investigation of the Contact of Machine Components" , (Ed. Kh. F. Ketova), 1949, vol. 30 (Central Scientific Research Institute for Technology and Mechanical Engineering, Moscow) (DSIR translation 337)

Inlet Analysis (Ertel-Grubin) - A Reminder



at the end of the inlet (where $x' = -a$) the pressure is so high that can be assumed as $\hat{Q} = \hat{Q}_s = 1/K_1$ at $X' = -1$

$$\int_{\hat{Q}_s}^{\hat{Q}} d\hat{Q} = -K \int_{X'=-1}^{X'} \frac{\hat{H} - \hat{H}^*}{\hat{H}^3} dX'$$

$$\hat{Q} = \hat{Q}_s - K \int_{X'=-1}^{X'} \frac{\hat{H} - \hat{H}^*}{\hat{H}^3} dX'$$

Then,

$$\hat{Q} = \frac{1}{K_1} - K \int_{X'=-1}^{X'} \frac{\hat{H} - \hat{H}^*}{\hat{H}^3} dX'$$

Finally, applying the second set of boundary conditions, at $X' = -L$ being $L \approx \infty$ then $P = \hat{Q} = 0$. Then,

$$0 = \frac{1}{K_1} - K \int_{X'=-1}^{-L} \frac{\hat{H} - \hat{H}^*}{\hat{H}^3} dX'$$

Introducing the shape of the deformed surface outside the Hertz flat zone:

$$\hat{H} - \hat{H}^* = |X|(X^2 - 1)^{1/2} - \ln\left[|X| + (X^2 - 1)^{1/2}\right]$$

The above Reynolds equation can be solved iteratively for $\hat{H}^*$.

With this solution, Ertel-Grubin obtained a curve-fitted equation in

Terms of Dowson-Higginson parameters:

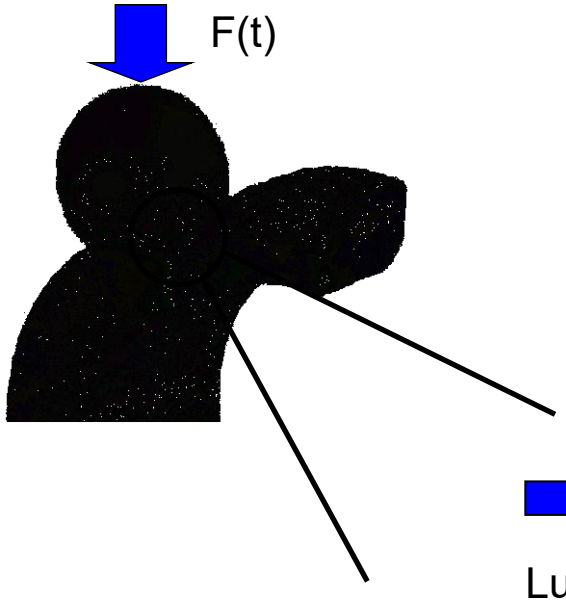
$$H_D^* = 1.95(GU)^{8/11} W^{-1/11} \approx 1.95(GU)^{0.73} W^{-0.091}$$

The inlet pressure can be recovered with:

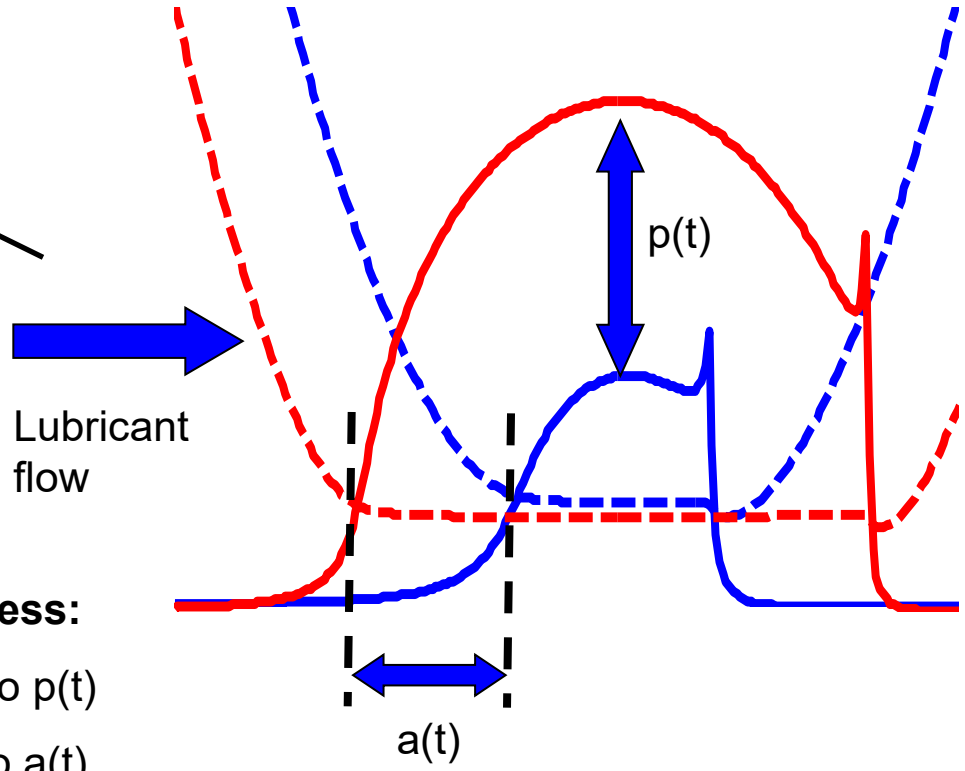
$$P = -\tfrac{1}{K_1} \ln(1 - K_1 \hat{Q})$$

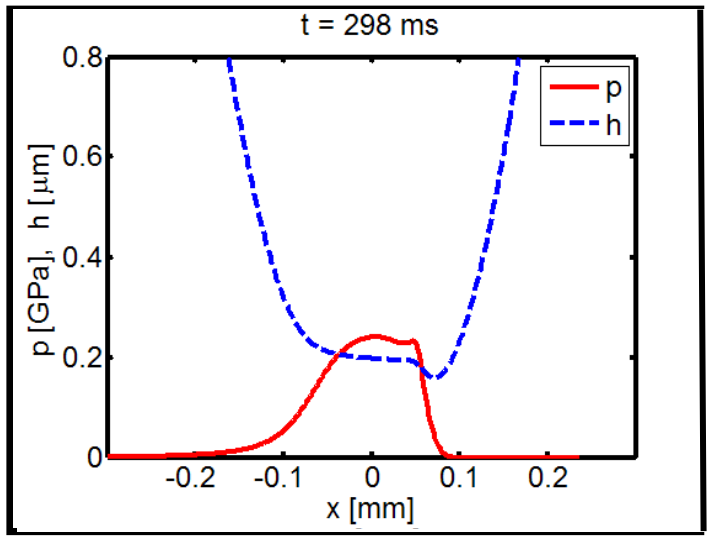
Grubin, A. N., "Investigation of the Contact of Machine Components" , (Ed. Kh. F. Ketova), 1949, vol. 30 (Central Scientific Research Institute for Technology and Mechanical Engineering, Moscow) (DSIR translation 337)

Dynamic Effects in Lubrication – The Inlet



Typical EHL pressure and film thickness profiles

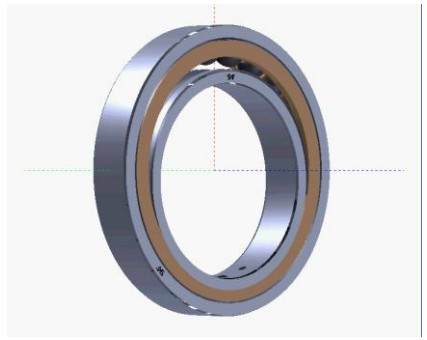




Numerical Simulation with Sinusoidal Normal Load

Effects on the film thickness:

- Global film squeeze due to p(t)
- Inlet film fluctuation due to a(t), i.e. $u = u_0 + da/dt$



1. Morales-Espejel, G.E., "Central film thickness in time-varying normal approach of rolling elastohydrodynamically lubricated contacts", IMechE, part C vol. 222, pp. 1271-1280, 2008

2. Félix-Quiñonez, A., Morales-Espejel, G.E., "Film thickness in time-varying normal loading of rolling elastohydrodynamically lubricated contacts", IMechE, part C vol. 224, pp. 2559-2567, 2010

Transient Ertel-Grubin Model

The Reynolds equation 1D, transient is:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right) = 12\bar{u}\frac{\partial(\rho h)}{\partial x} + 12\frac{\partial(\rho h)}{\partial t}$$

Corrected to consider variation of contact semi-width in time

The first integration respect to x gives:

$$\frac{\partial p}{\partial x} = 12\bar{u}\eta\frac{\rho h - \rho^* h^*}{\rho h^3} + \frac{12\eta}{\rho h^3}\int_{x^*}^{x}\frac{\partial(\rho h)}{\partial t}\partial x$$

Introducing dimensionless groups, gives:

$$X = \frac{x}{a}, \quad \hat{H} = \frac{2R_x h}{a^2}, \quad P = \frac{p}{p_o}$$

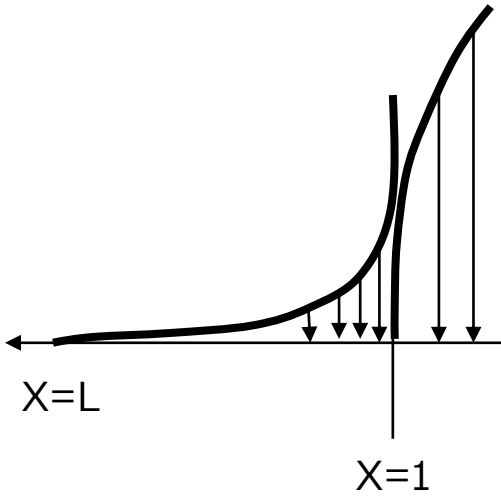
$$Q = \frac{q}{p_o}, \quad K = \frac{48\bar{u}\eta_o R_x^2}{(a^3 p_o)} \qquad K_1 = \propto p_o$$

$$\hat{T} = t\bar{u}/a \qquad q = [1 - e^{-\propto p}]/\propto$$

$$\frac{\partial Q}{\partial X} = \frac{K}{\hat{H}^3}\left(\hat{H} - \hat{H}^* + \int_{X=1}^{X}\frac{\partial \hat{H}}{\partial \hat{T}}\partial X\right)$$

Applying Grubin boundary conditions for X=1, Q=1/K_1 and integrating, then applying X=L≈ ∞, Q=0

$$\frac{1}{K_r} = \int_{X=1}^{L}\frac{\hat{H}' - \hat{H}'^* + (\mathrm{d}\hat{H}^*/\mathrm{d}\hat{T})(X-1)}{\hat{H}'^3}\partial X$$



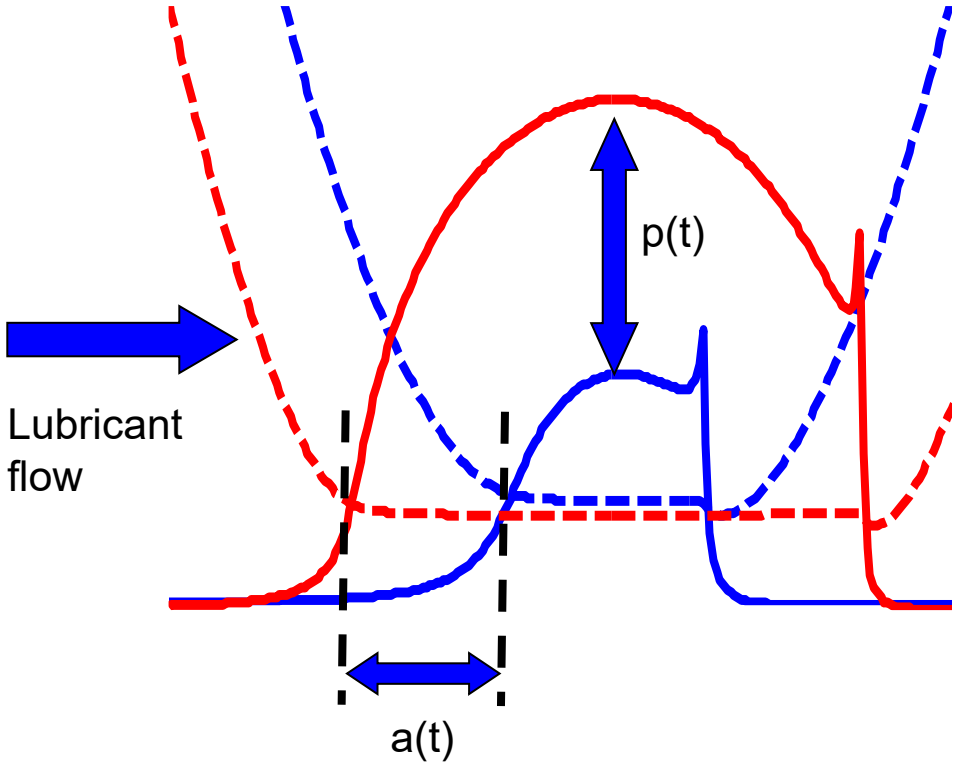
$$K_1 K = K_r = (48\bar{u}\eta_0 R_x^2\alpha)/(a^3)$$

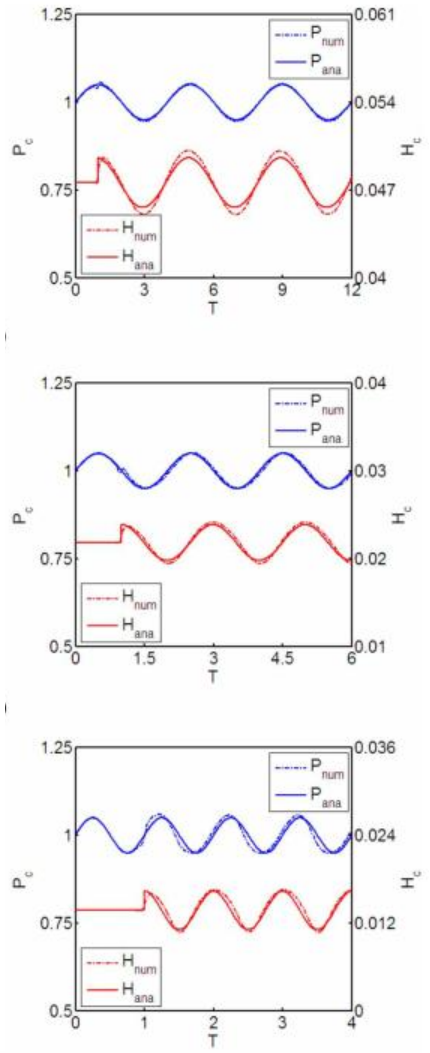
Crook: $\hat{H}(X) = C(X-1)^{3/2}$; $C = \frac{4}{3}\sqrt{2}$

$$\frac{\partial \hat{H}^*}{\partial \hat{T}} = \frac{C^{4/3}(\hat{H}^*)^{5/3}}{0.2687}\left[\frac{1}{K_r} - \frac{0.2687}{C^{2/3}(\hat{H}^*)^{4/3}}\right]$$

Model Comparison

Typical EHL pressure and film thickness profiles



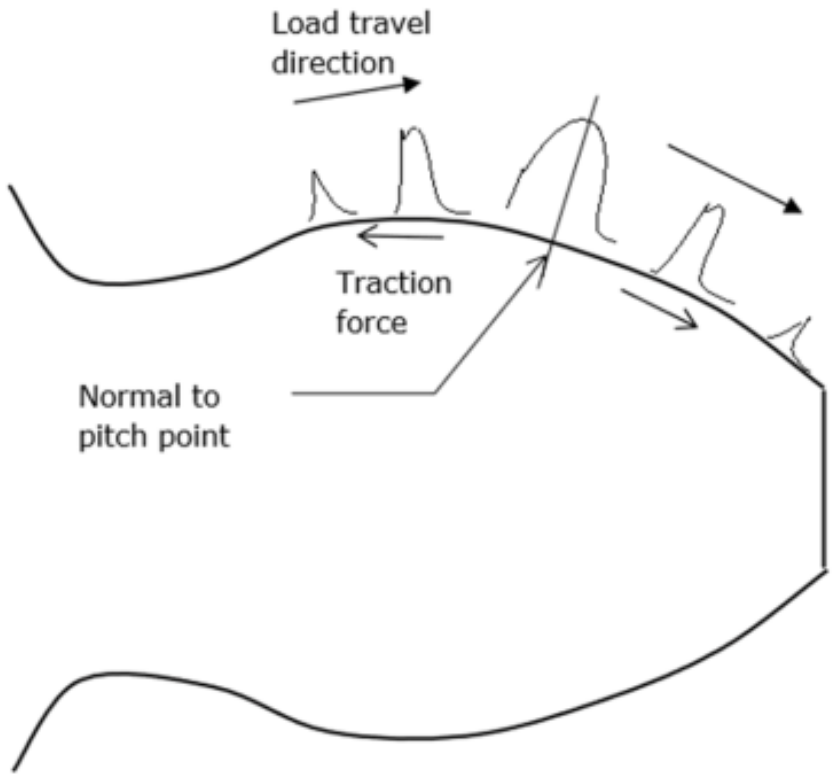


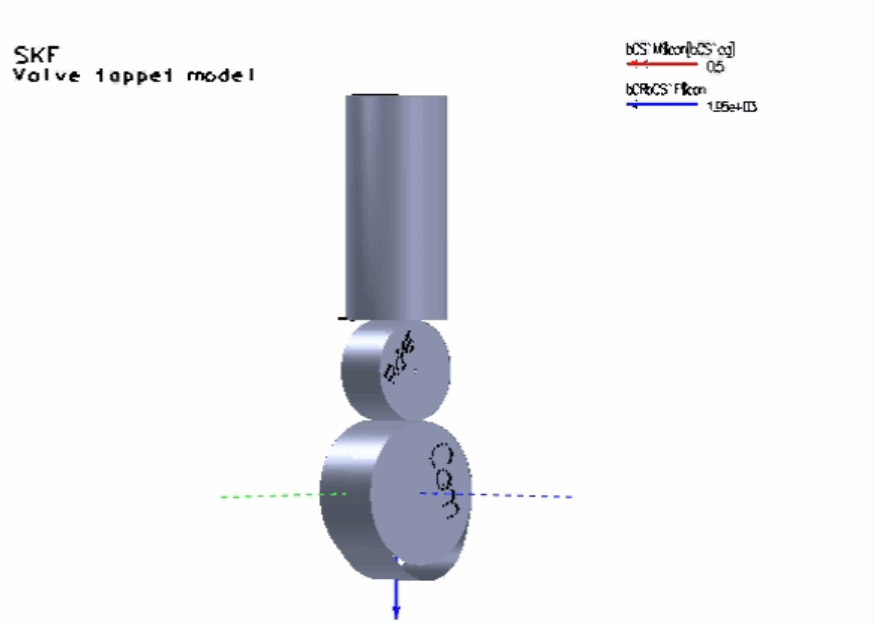
(c)

Comparison of numerical and analytical central film thickness and pressures as a function of time. The analytical results on the left-hand side figures correspond to the original method. The results including the proposed entrainment speed correction are given on the right-hand side plots: (a) $M = 40$, $L = 10$, $\Omega = 0.5\pi$, $A_w = 0.1$; (b) $M = 80$, $L = 10$, $\Omega = \pi$, $A_w = 0.1$; and (c) $M = 120$, $L = 10$, $\Omega = 2\pi$, $A_w = 0.1$

Félix-Quiñonez, A., Morales-Espejel, G.E., "Film thickness in time-varying normal loading of rolling elastohydrodynamically lubricated contacts", IMechE, part C vol. 224, pp. 2559-2567, 2010

Variable Speed





Variable Speed

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right) = 12\bar{u}\frac{\partial(\rho h)}{\partial x} + 12\frac{\partial(\rho h)}{\partial t} \quad \Rightarrow \quad \frac{\partial Q}{\partial X} = \frac{K}{\hat{H}^3}\left(\hat{H}' - \hat{H}'^* + \frac{K_2}{\Delta\hat{T}}\int_{X=1}^{X}(\hat{H}' - \hat{H})\partial X\right)$$

With H' for the dimensionless clearance for the time $t + \Delta t$ and H at time t

The term $\left(\hat{H}' - \hat{H}\right)$ is independent of X, thus the final equation to solve is:

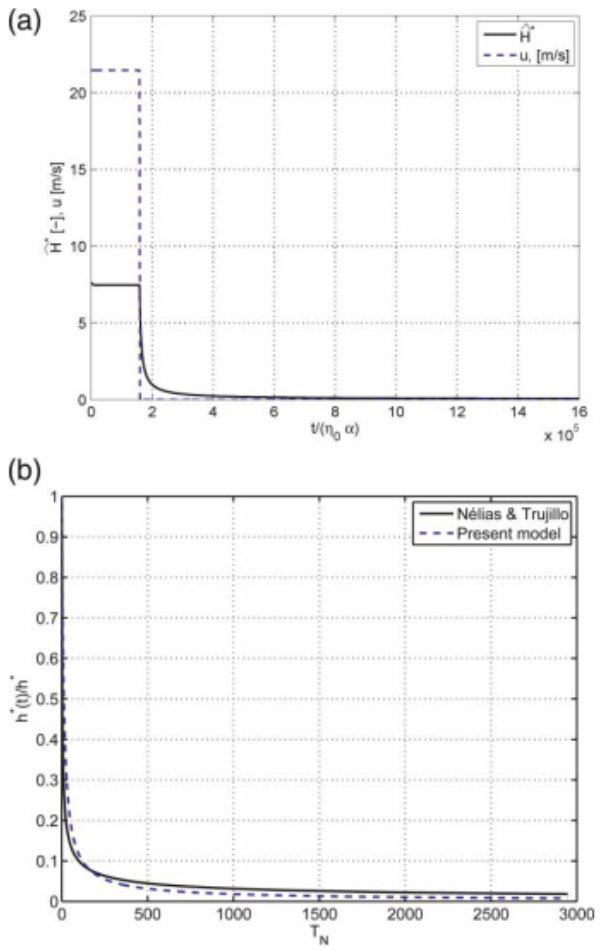
$$\frac{\partial Q}{\partial X} = \frac{K}{\hat{H}^3}\left(\hat{H}' - \hat{H}'^* + K_2\frac{\hat{H}' - \hat{H}}{\Delta\hat{T}}(X-1)\right)$$

With:

$$\hat{T} = t/(\eta_0\alpha). \quad K_2 = a/\eta_o\bar{u}\alpha$$

Table 1 Steady-state data used in the sudden halting example

F (N)	α (GPa)$^{-1}$	E' (GPa)	η_o (Pa s)	R_x (m)	$\bar{u}$ (m/s)	p_o (GPa)	K (−)	K_1 (−)	K_2 (−)	H^* (−)
1022	22.81	226	0.4247	0.0087	21.45	1.17	5.0	26.67	854.8	7.63

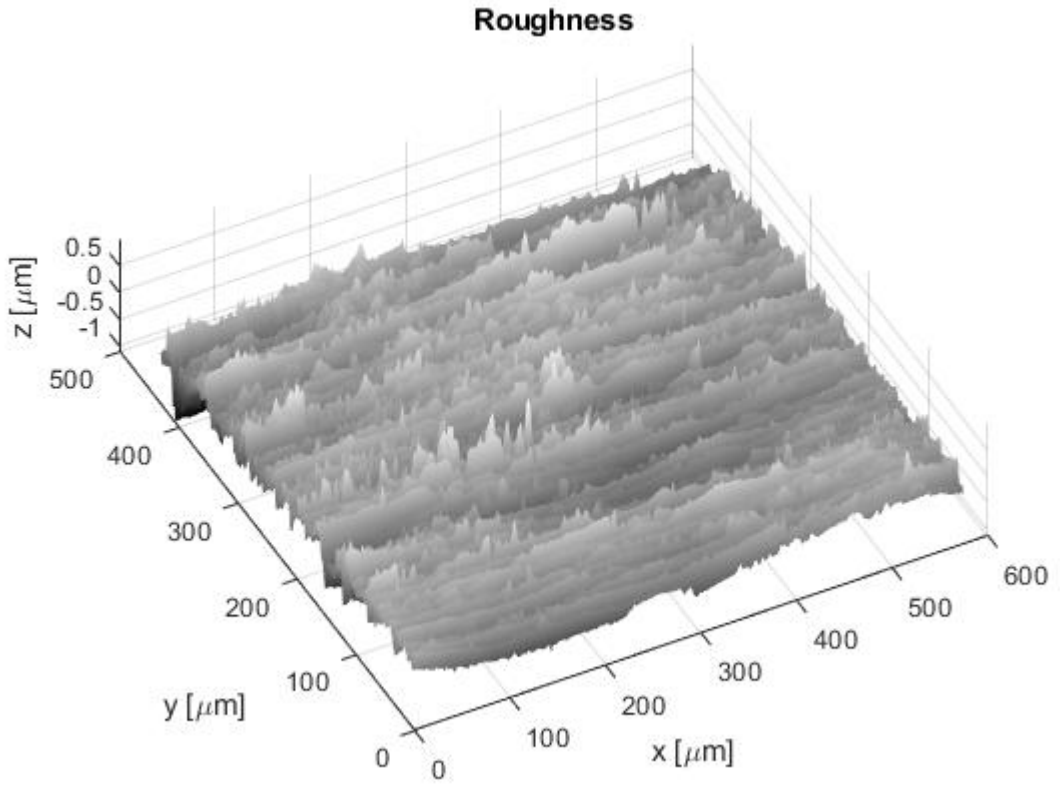


1 Sudden halting motion for example 1 of Table 1, initial values $K = 5.0$, $K_1 = 26.67$, and $K_2 = 854.8$. (a) variation of $\bar{u}$ and $\hat{H}_c$ in time, $\hat{t}$ and (b) comparison with model of reference [17]

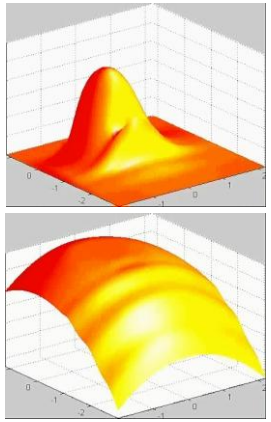
Morales-Espejel, G.E., Wemekamp, A.W. "Ertel-Grubin Methods in Elastohydrodynamic Lubrication – A Review", IMechE, part J vol. 222, pp. 15-34, 2008

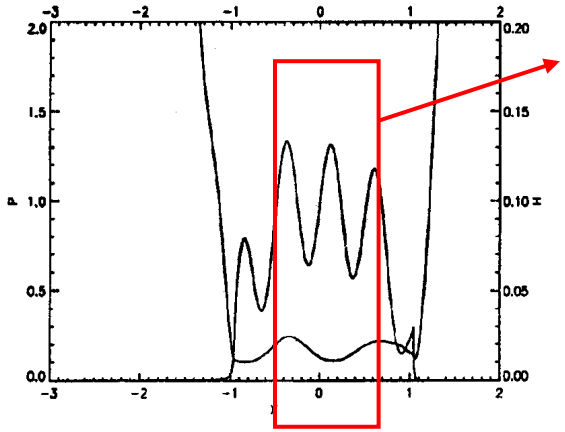
* Nélias, D., Trujillo Jiménez, A., "A Simplified Model to Study EHL Film Collapse During Rapid Halting Motion. Tribol. Trans., 2002, 45(4), 512–520.

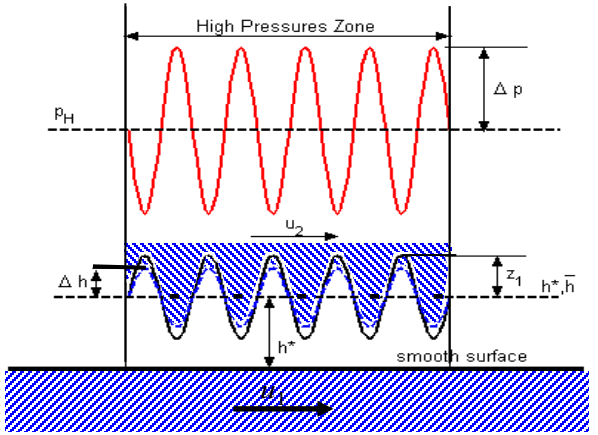
Modelling EHL with Roughness (Deterministic)



Rapid Methods for Micro-EHL



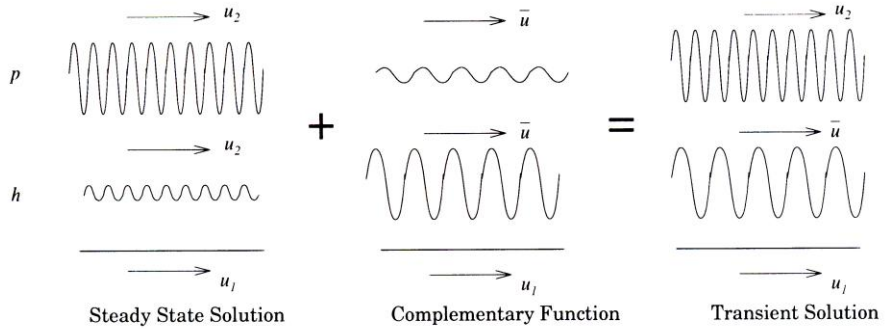




$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) = \bar{u}\frac{\partial \rho h}{\partial x} + \frac{\partial \rho h}{\partial t}$$

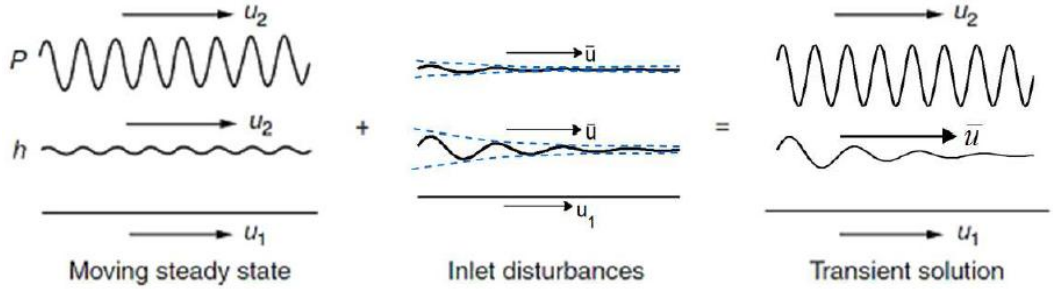
- ***Linearised Reynolds Equation***
- ***Use powerful Fourier techniques***

Newtonian Behaviour



Deformed waviness | Inlet propagated wave | Full solution

Non-Newtonian Behaviour



* Greenwood, J.A., Morales-Espejel, G.E., "The Behaviour of Transverse Roughness in EHL Contacts", IMechE, part J, vol. 208, pp. 121-132, 1994

* Hooke, C.J., Li, K.Y., Morales-Espejel, G.E., "Rapid calculation of the pressures and clearances in Rough rolling-sliding elastohydrodynamically lubricated contacts. Part1: Low-amplitude sinusoidal roughness", IMechE, part J, vol. 224, pp. 621-637, 2010

Newtonian Model

- *Steady State Solution*: Steady state solution for pure sliding

$$h = \bar{h} + z + v$$

$$h(x) = \bar{h} + h_1 \sin\left(\frac{2\pi x}{\lambda}\right) \qquad z(x) = z_1 \sin\left(\frac{2\pi x}{\lambda}\right) \qquad p(x) = p_o + p_1 \sin\left(\frac{2\pi x}{\lambda}\right) \qquad v(x) = v_1 \sin\left(\frac{2\pi x}{\lambda}\right)$$

$$v_1 = \frac{2\lambda p_1}{\pi E'} \qquad \rho h = \bar{\rho}\bar{h} \quad \text{with} \quad \frac{\rho(p)}{\rho(p_H)} \approx 1 + C(p - p_H)/\bar{h} \quad \text{from which:}$$

$$p_1 = \frac{-z_1}{C + A}\ ,\ h_1 = \frac{z_1 C}{C + A}$$

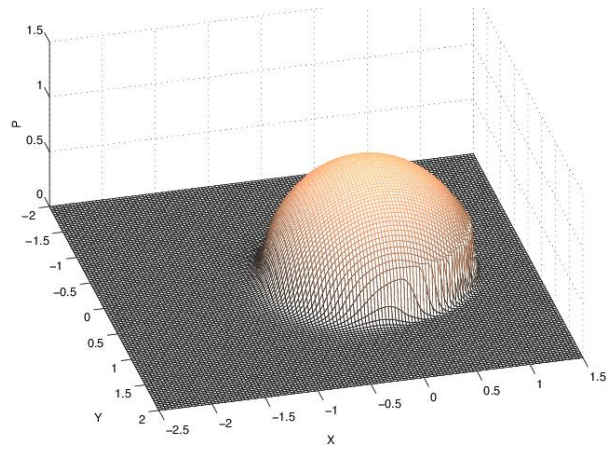
$$A = \frac{2\lambda}{\pi E'} \qquad C = \frac{(\gamma - \beta)\ \bar{h}}{(1 + \gamma\ p_H)(1 + \beta\ p_H)}$$

- *Complementary Function*: Inlet excitation from a partly deformed roughness

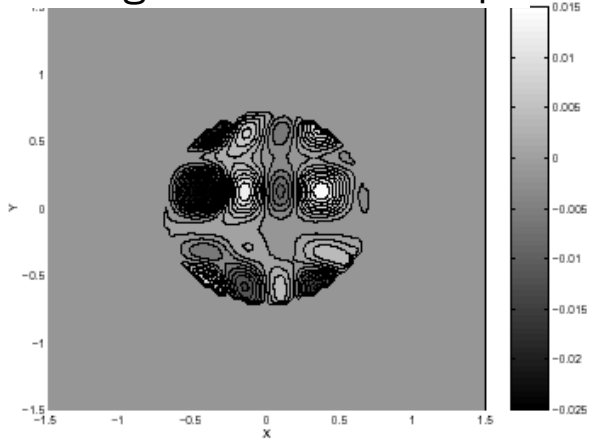
$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) = \bar{u}\frac{\partial \rho h}{\partial x} + \frac{\partial \rho h}{\partial t} \qquad (0)$$

$$\bar{u}\frac{\partial r}{\partial x} + \frac{\partial r}{\partial t} = 0 \qquad r = \frac{\rho h}{\bar{\rho}\bar{h}} \quad \text{(oil flow)}$$

Comparison with Numerical Models

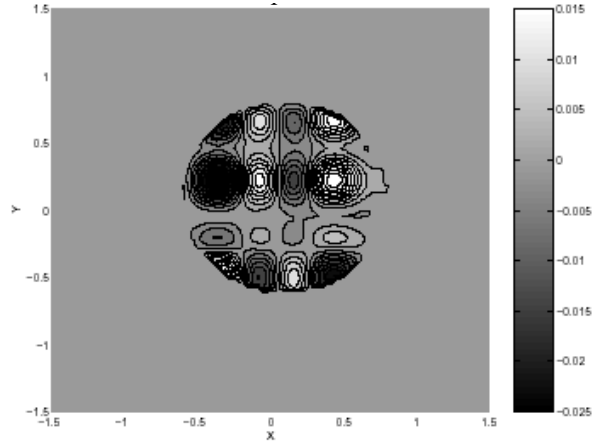


Multigrid Deformed Shape

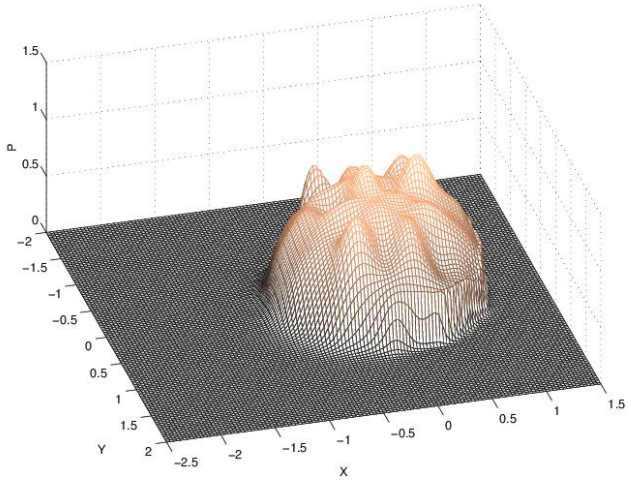


(a) Deformed roughness, numerical solution

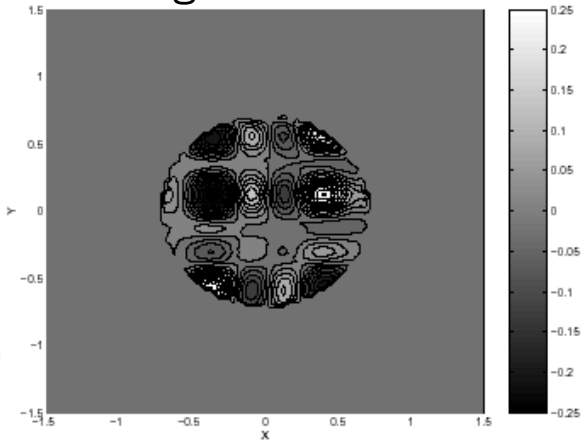
Rapid Method Deformed Shape



(b) Deformed roughness, present scheme

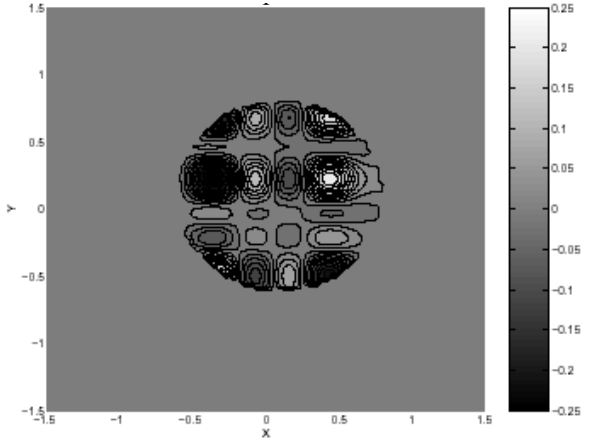


Multigrid Pressures



(c) Pressure ripples, numerical solution

Rapid Method Pressures

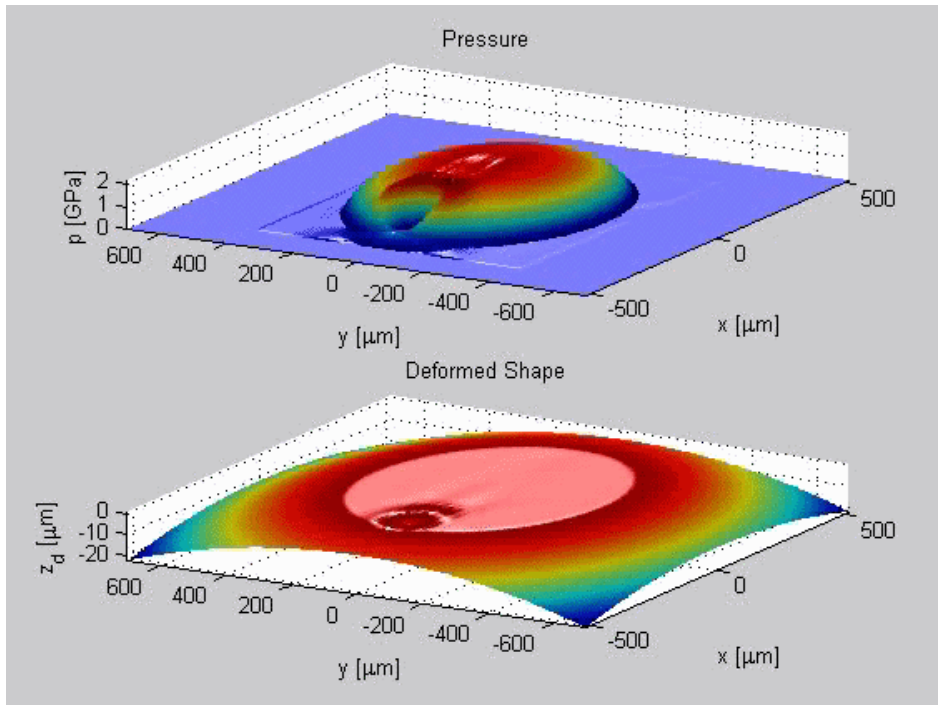


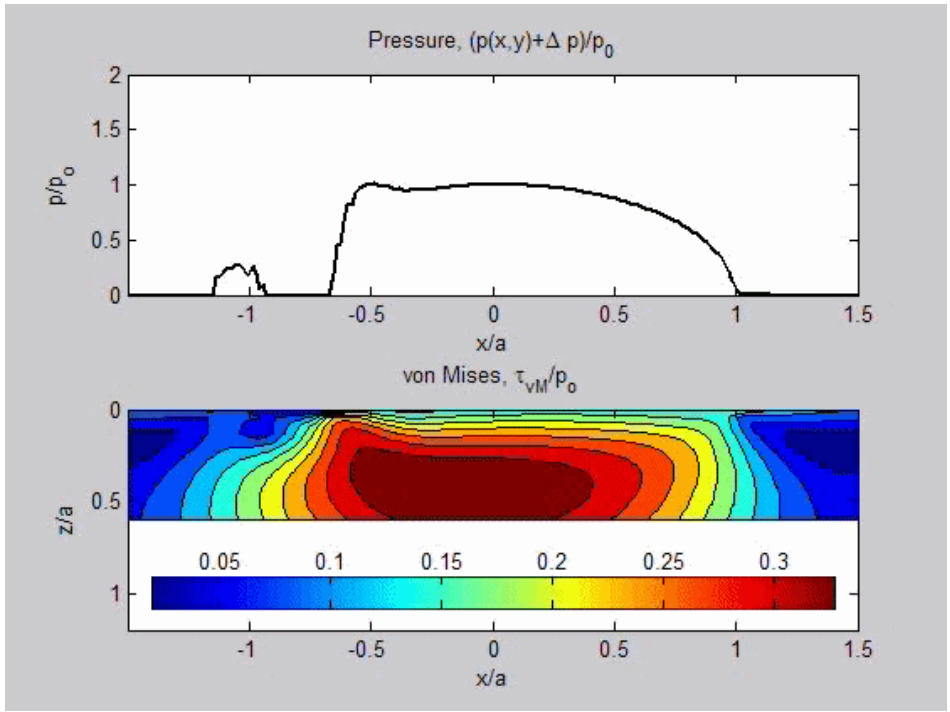
(d) Pressure ripples, present scheme

Morales-Espejel, G.E., Lugt, P.M., Kuilenburg, van, J., Tripp, J., **Effects of Surface Micro-Geometry on the Pressures and Internal Stresses of Pure Rolling EHL Contacts,** Trib. Trans. Vol. 46(2), pp. 260-262, 2003

Case of Pure Rolling (Zero Sliding)

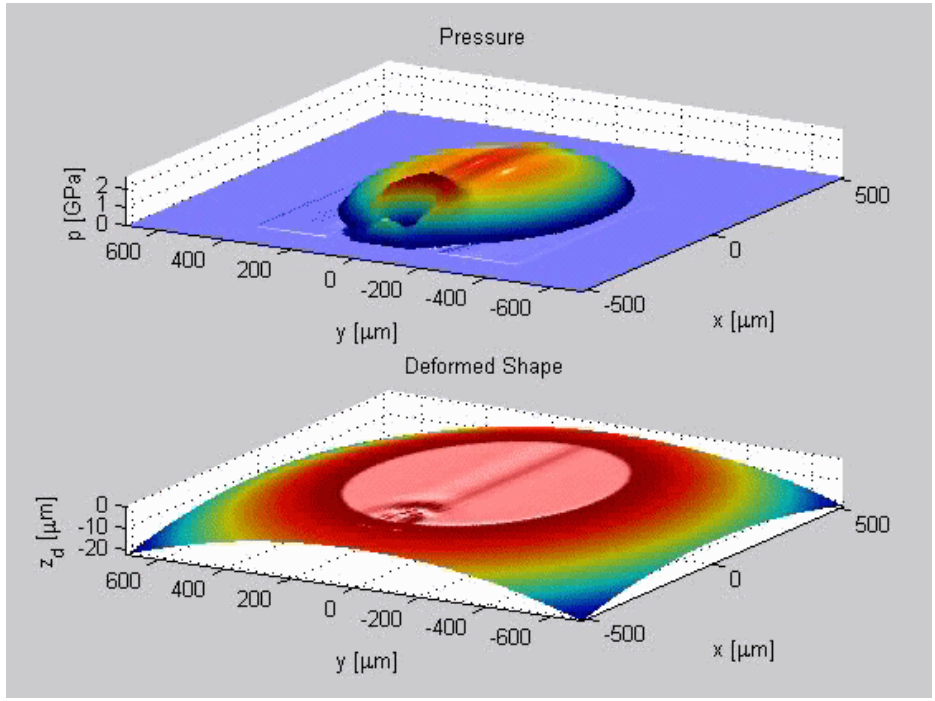
Non-Newtonian Fluid

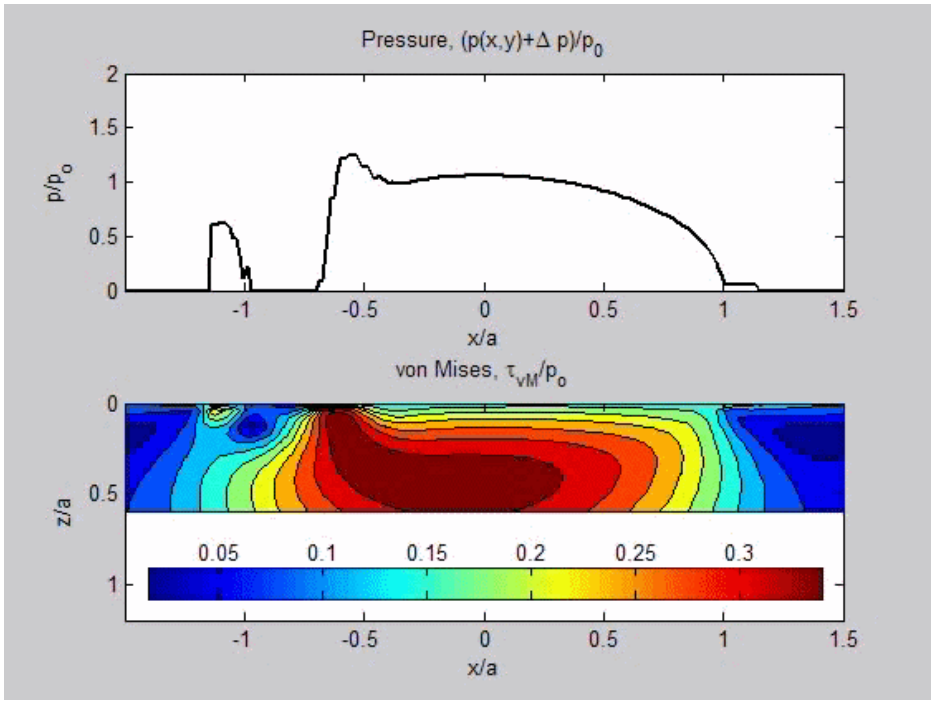




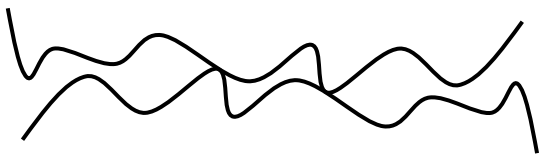
Case of Sliding, Indented Surface Moving 30% Faster

Non-Newtonian Fluid





Conversion to Mixed Lubrication



From which the frequency response matrix is

$$w(i,j) = \frac{1}{\sqrt{(i-1)^2 + [(j-1)l]^2}},$$
$$\forall i = 1,\ldots,n/2 \quad \text{and} \quad j = 1,\ldots,m/2$$
$$w(i,j) = \frac{1}{\sqrt{(i-1)^2 + [(m-j+1)l]^2}},$$
$$\forall i = 1,\ldots,n/2 \quad \text{and} \quad j = m/2+1,\ldots,m$$
$$w(i,j) = \frac{1}{\sqrt{(n-i+1)^2 + [(j-1)l]^2}},$$
$$\forall i = n/2+1,\ldots,n/2 \quad \text{and} \quad j = 1,\ldots,m/2$$
$$w(i,j) = \frac{1}{\sqrt{(n-i+1)^2 + [(m-j+1)l]^2}},$$
$$\forall i = n/2+1,\ldots,n/2 \quad \text{and} \quad j = m/2+1,\ldots,m$$

for $w(1,1) = 0$, where $l = L_x/L_y$.

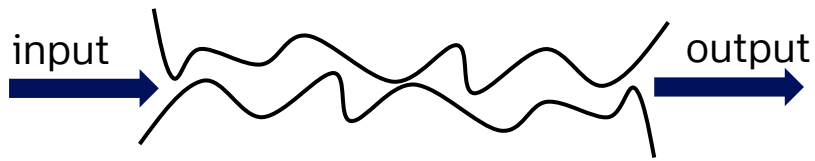
Following Stanley and Kato, assuming periodicity in roughness, in dry contact, the elastic displacements are:

$$\mathbf{u}(\mathbf{p}) = \text{IFFT}\left\{\mathbf{w} \cdot FFT(\mathbf{p})\right\}$$

The contact problem is solved by finding p (pressures):

$$\min(f) = \frac{1}{2}\int_S pu(p)\mathrm{d}S + \int_S pg\,\mathrm{d}S$$
$$\frac{1}{A}\int_S p\,\mathrm{d}S = p_{\text{target}} \quad p \geqslant 0$$

The mixed lubrication algorithm relies on the load sharing principle, until the flow is balanced

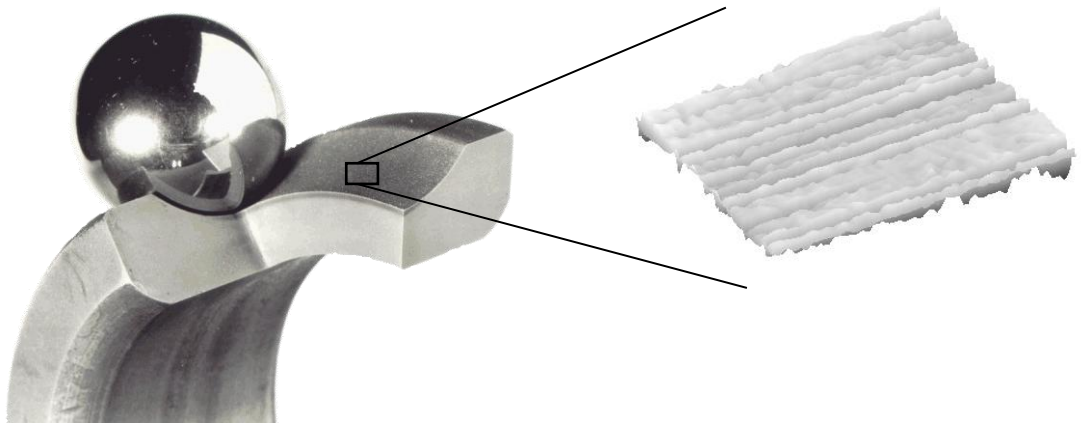


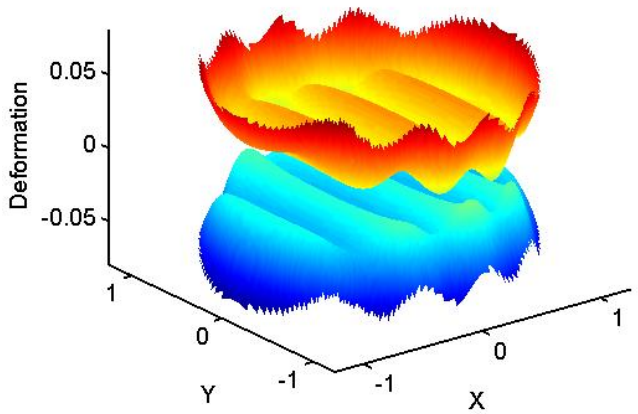
Input = output

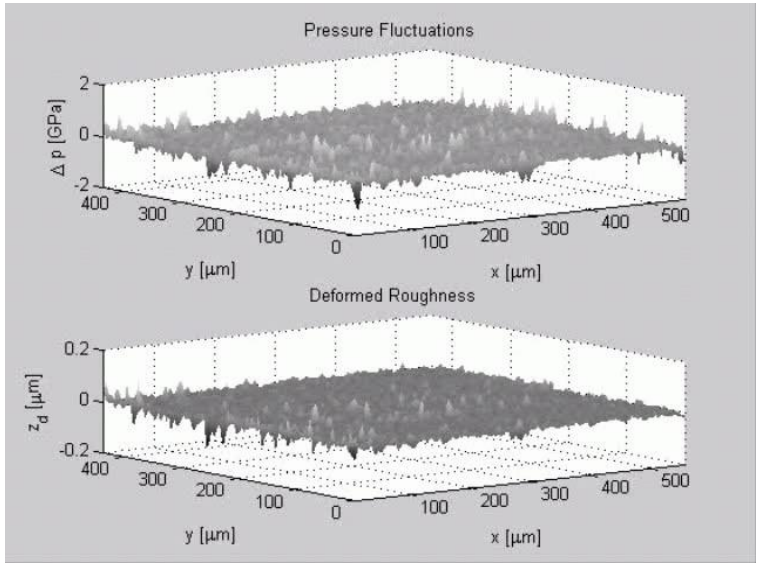
* Morales-Espejel, G.E., Wemekamp, A.W., Félix-Quiñonez, A. Micro-geometry effects on the sliding friction transition in elastohydrodynamic lubrication IMechE, part J, J. of Eng. Trib., vol. 224, pp. 621-637, 2010

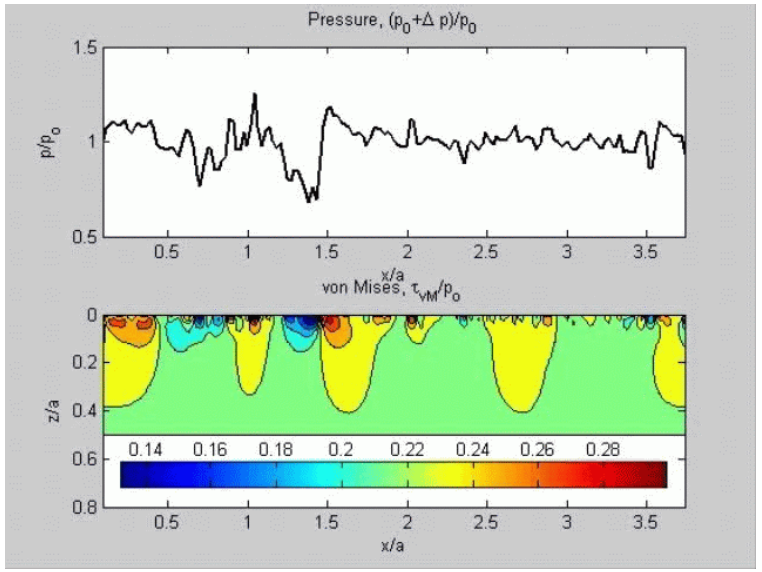
Stanley, H. M. and **Kato, T.** A FFT-based method for rough surface contact. *ASME, J. Tribol.*, 1997, **119**, 481–485.

Modelling Lubrication in Real Time*









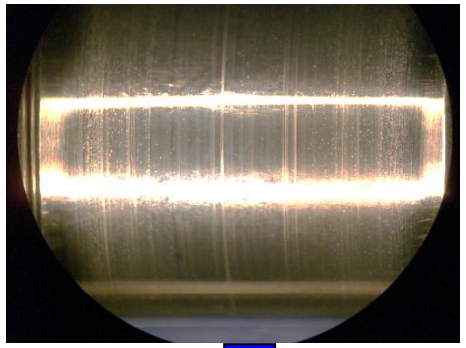
Includes:

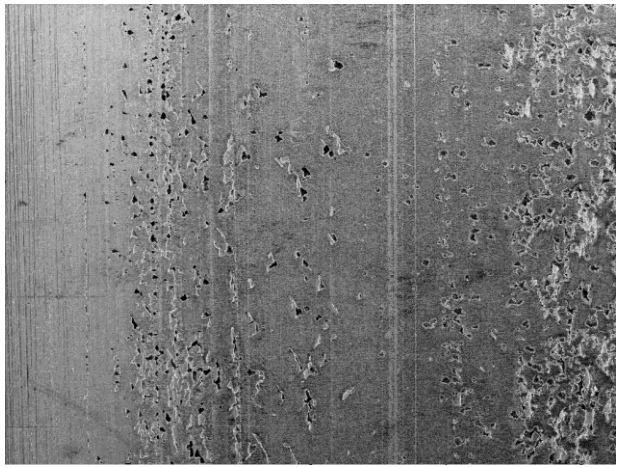
- Roughness
- Time dependent
- Mixed-Lubrication
- Non-Newtonian
- Rolling/Sliding
- Thermal effects
- Runs as fast as the video

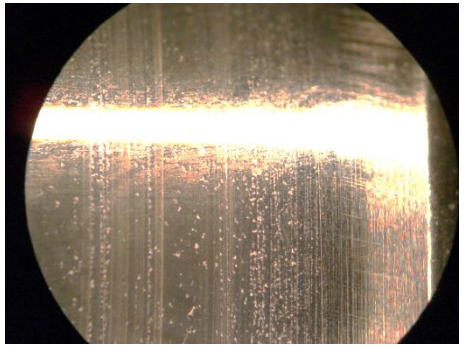
* Morales-Espejel, G.E., Wemekamp, A.W., Félix-Quiñonez, A.
Micro-geometry effects on the sliding friction transition in elastohydrodynamic lubrication
IMechE, part J, J. of Eng. Trib., vol. 224, pp. 621-637, 2010

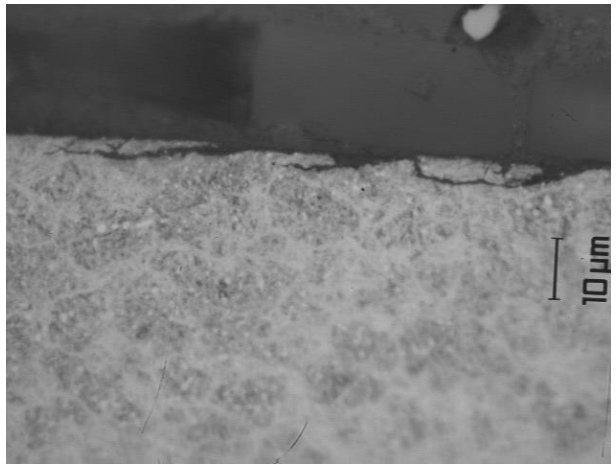
Surface Distress

Modelling Surface Distress









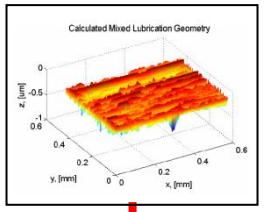
Definition Micropitting:

Tiny surface spalls, pits & cracks which sometimes appear on the surface of rolling-sliding contacts in poor lubrication conditions, ISO 15243 calls it *surface distress*

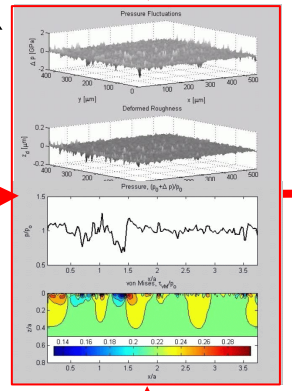
Micropitting is in fact surface fatigue at asperity level. It competes with mild wear which removes the fatigued layers of the surface.

Surface Distress – Micropitting

Surface topography of both surfaces



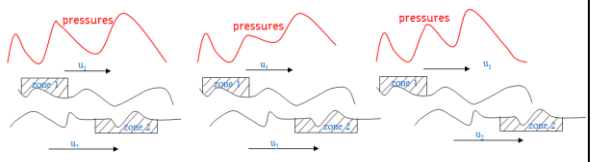
EHL Mixed-Lubrication solver



Lubricant Rheology

Operating Conditions, Material and Lubricant Properties

Asperity Contacts and Pressure History



Asperity Fatigue Stress (Dan Vang)

Damage Accumulation (Palmgren-Miner)

t=t_f?

yes

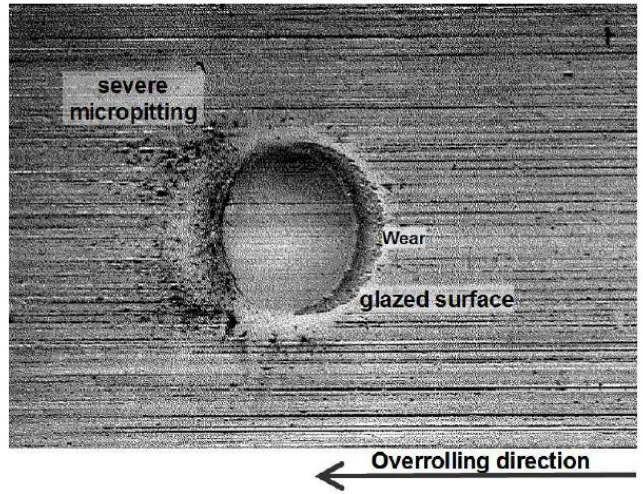
no

Surface Distress Prediction

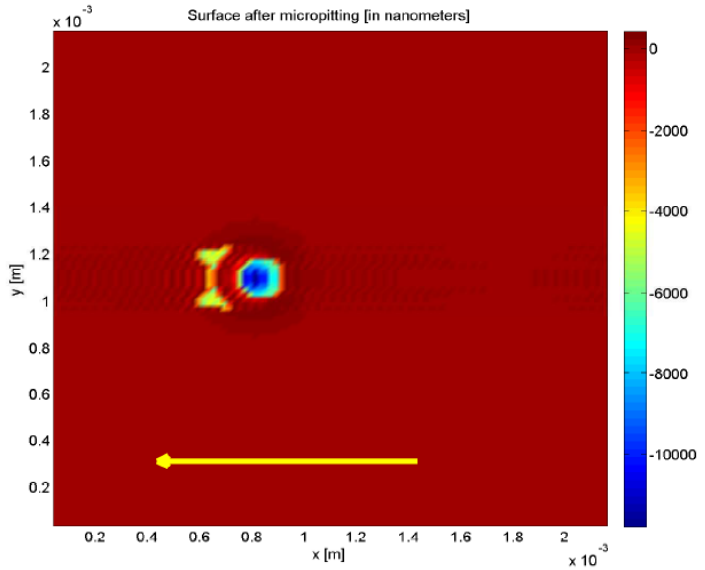
local mild wear

$$\Delta h_w(x,y) = k(x,y)\frac{p(x,y)}{H}u_s\left(\frac{l}{u_1}\right)$$

$t_{i+1} = t_i + \Delta t$



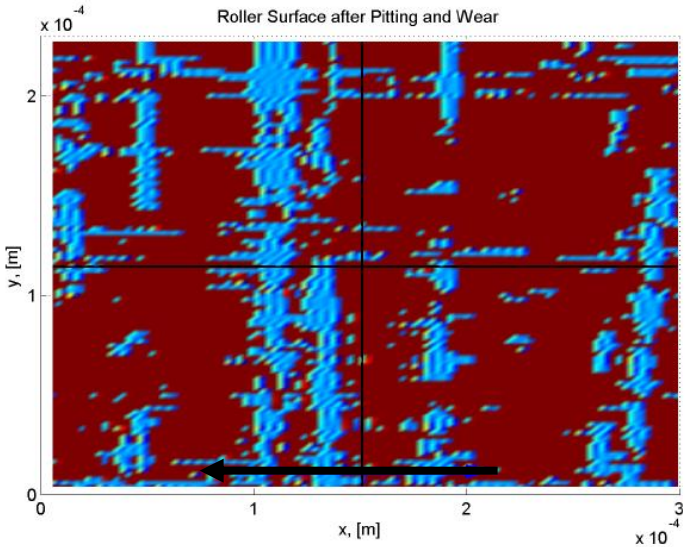
(a) Experimental 2250X10^6 over-rollings



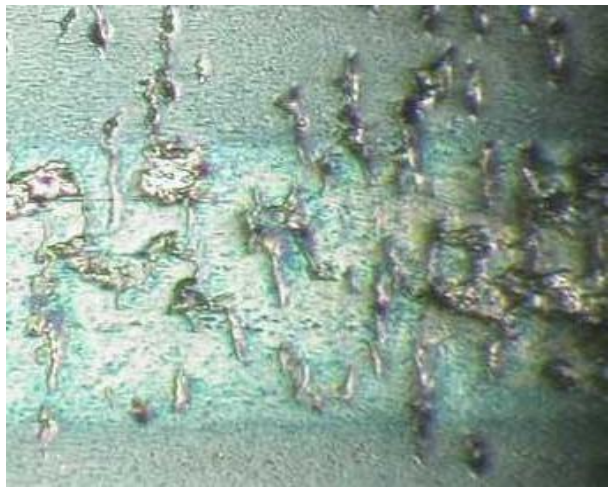
(b) Model 2250X10^6 over-rollings

* Morales-Espejel, G.E., Brizmer, V., "Micropitting Modelling in Rolling–Sliding Contacts: Application to Rolling Bearings", Tribology Transactions, 54, pp. 625-643, 2011.

Roughness Lay Effect

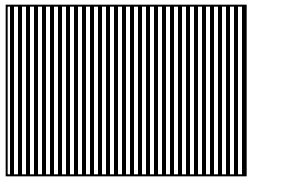


Model – Transverse Roughness

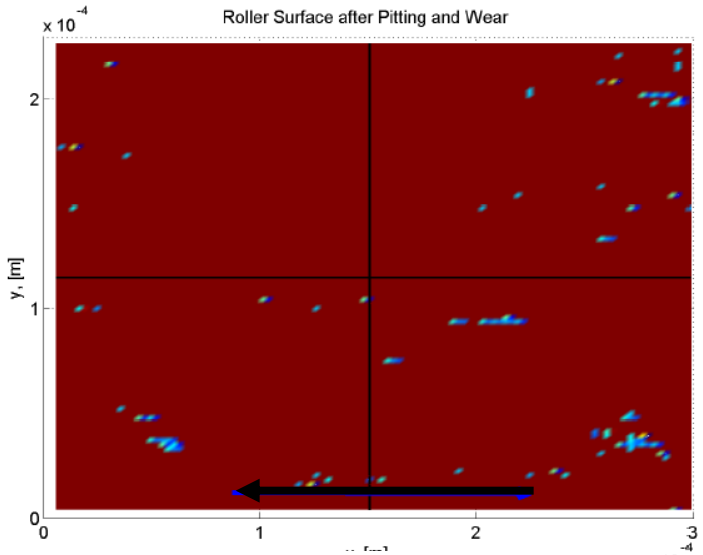


Experiment – Transverse Roughness

Λ=0.1
720000 cycles

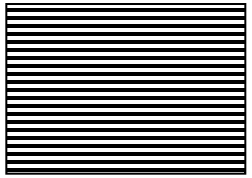


rolling



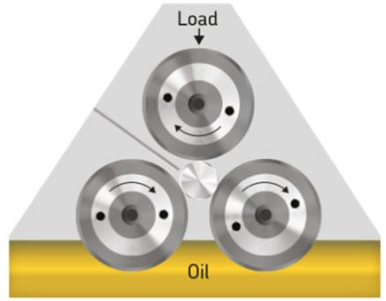
Model – Longitudinal Roughness

Experiment – Longitudinal Roughness



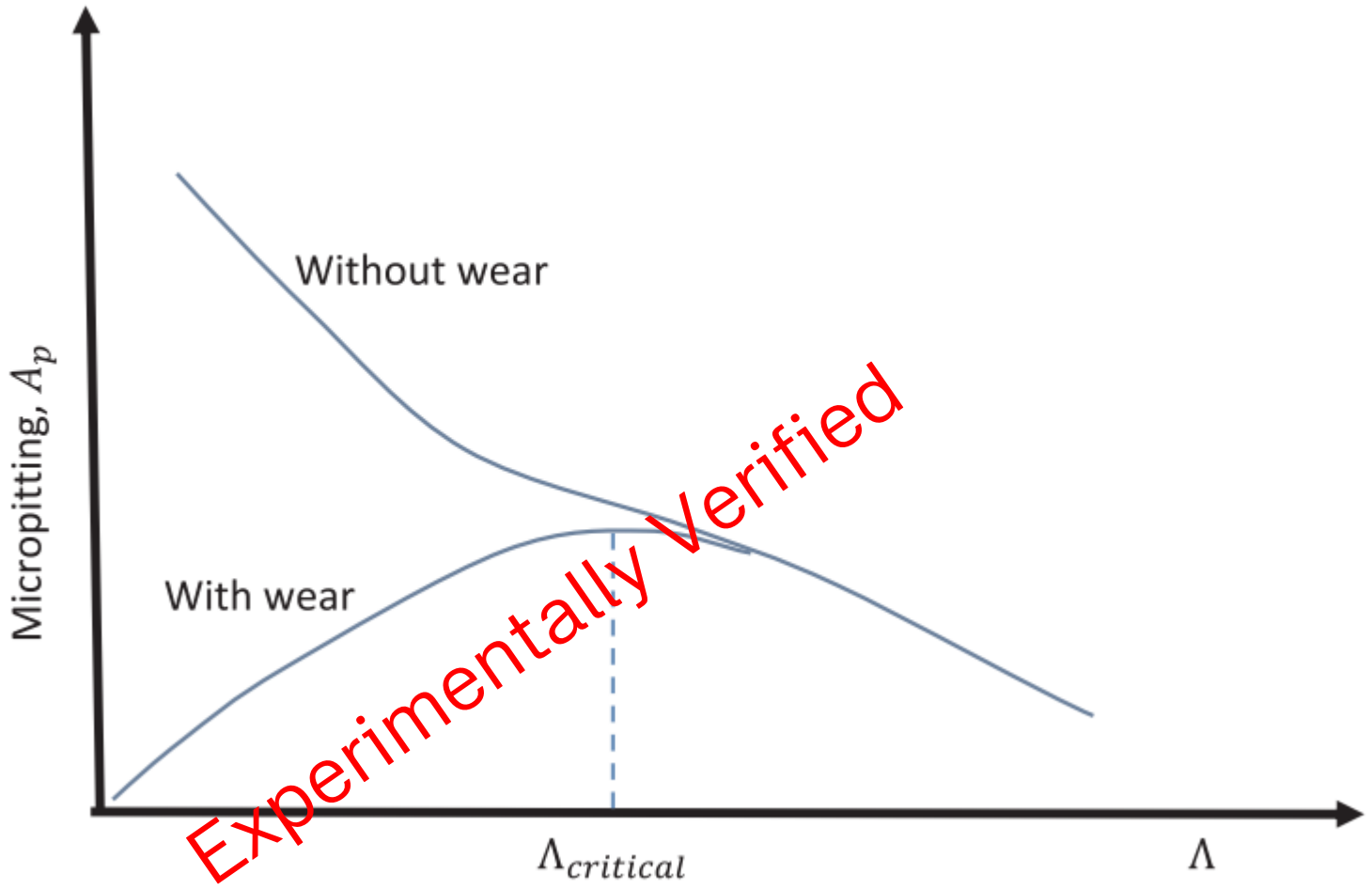
rolling

3-Disc Machine



* Morales-Espejel, G.E., Brizmer, V., "Micropitting modelling in rolling-sliding contacts: Application to rolling bearings", Trib. Trans., 54, pp. 625-643, 2011

Competition with Mild Wear

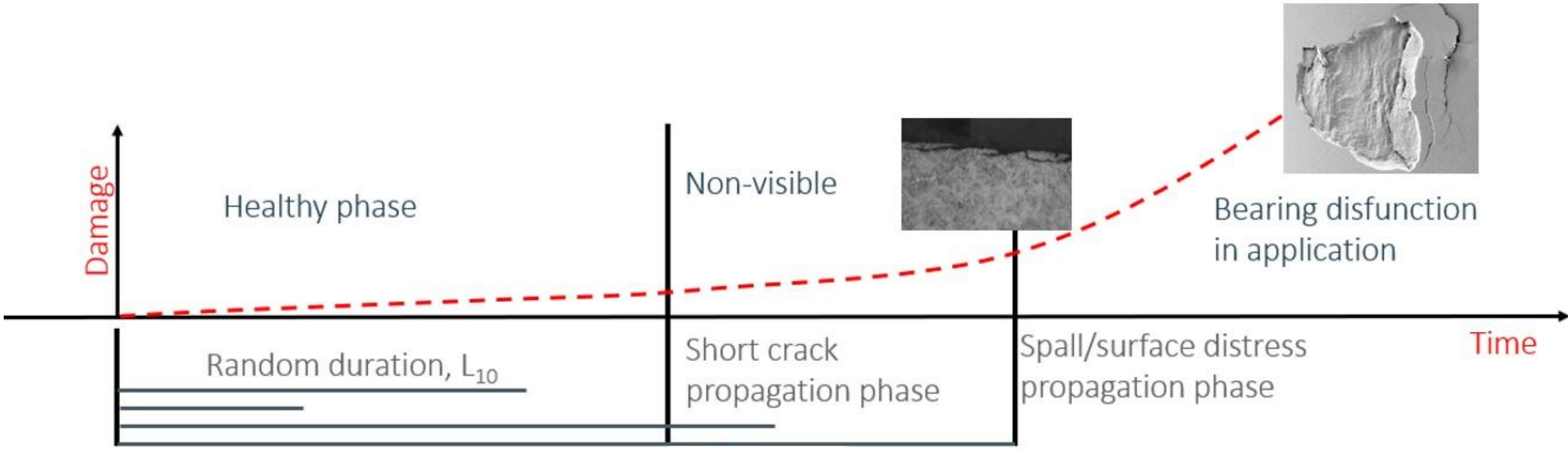


G.E. Morales-Espejel, P. Rycerz, A. Kadiric, Prediction of micropitting damage in gear teeth contacts considering the concurrent effects of surface fatigue and mild wear, in press, Wear 398–399 (2018) pp. 99–115.

New Concepts in the Life of Machine Elements

Generalized Bearing Life Model (GBLM)

The Life of Machine Elements



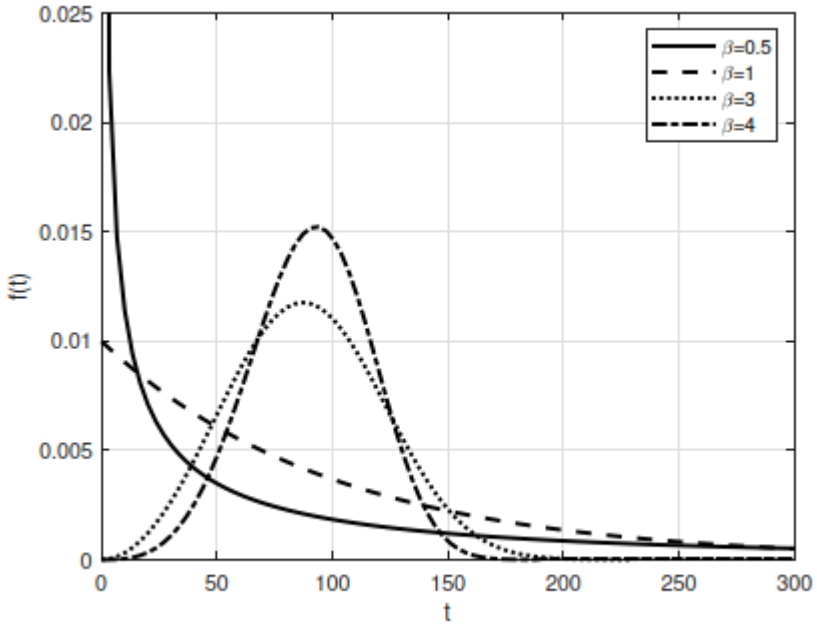
Randomness of Bearing Life

The life of rolling bearings (and other machine elements) is a random number, which follows a certain probability distribution. Therefore, it can not be calculated, it can only be estimated for a population of seemly equal bearings running at seemly equal operating conditions with a certain reliability (or probability of survival). Normally 90 % survival probability is used, thus the L_{10} is calculated.

Rolling Bearing Life in general follows a Weibull Probability Distribution

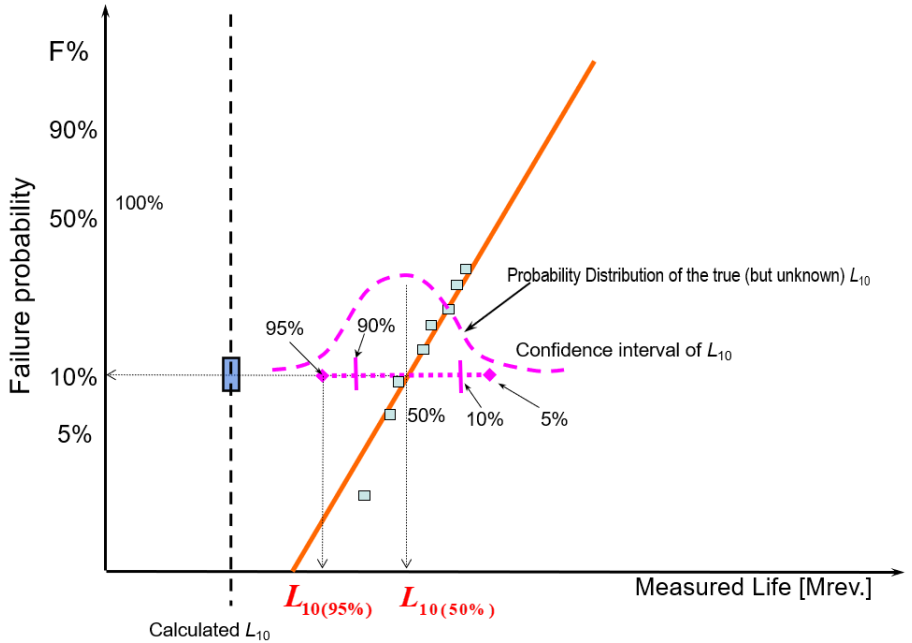
Weibull Density Probability Function:

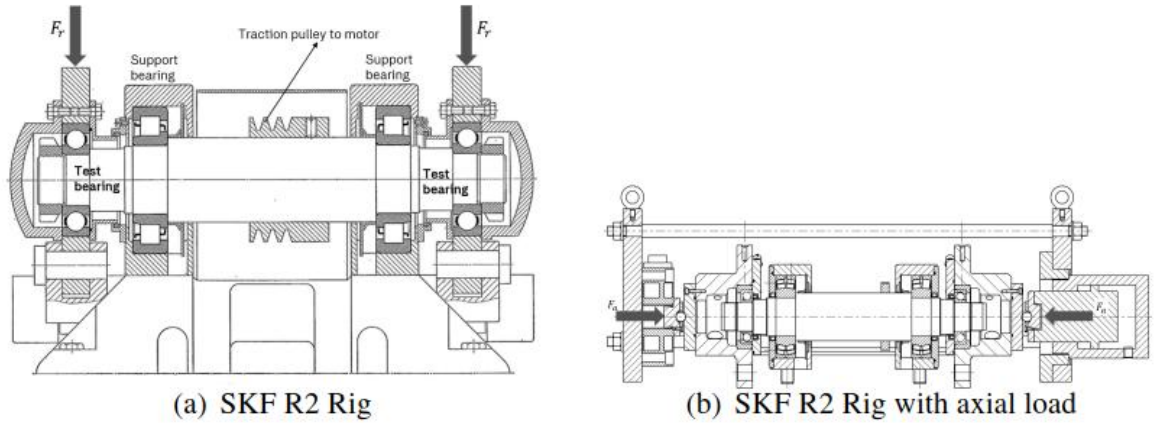
$$f(t) = \frac{dF(t)}{dt} = \frac{\beta}{\tilde{\eta}} \left(\frac{t - L_0}{\tilde{\eta}} \right)^{\beta - 1} \exp \left[- \left(\frac{t - L_0}{\tilde{\eta}} \right)^{\beta} \right], \forall \quad t \geq L_0$$



(a) Density, $f(t)$

Weibull Paper And Endurance Testing





(a) SKF R2 Rig (b) SKF R2 Rig with axial load

Figure 2.6 SKF R2 test rigs for small rolling bearings. (Courtesy of SKF.)

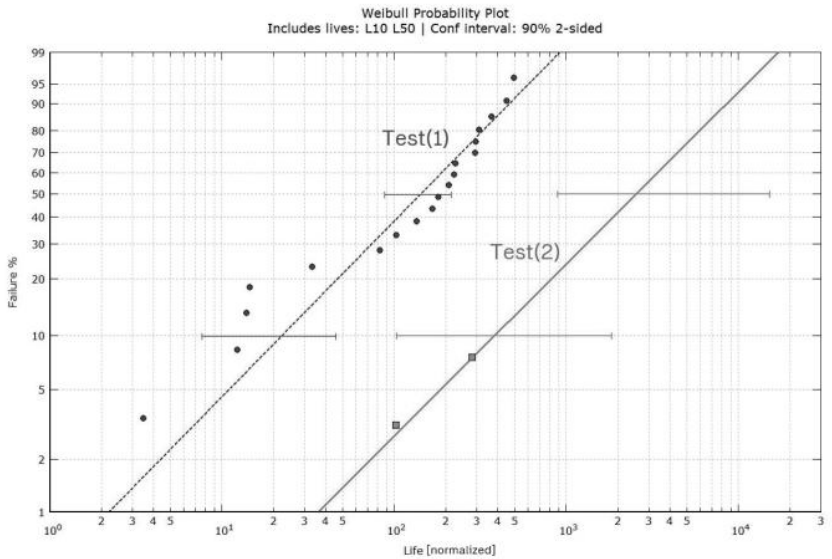
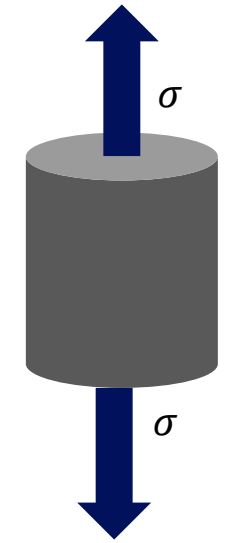


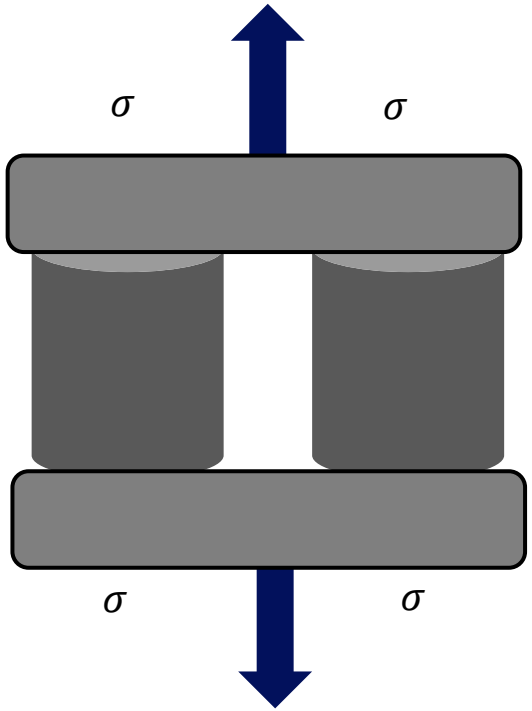
Figure 2.5 Real Weibull plot from endurance testing showing two distinct tests, including the 90% confidence intervals for L_{10} and L_{50}.

Weibull Theory of Strength

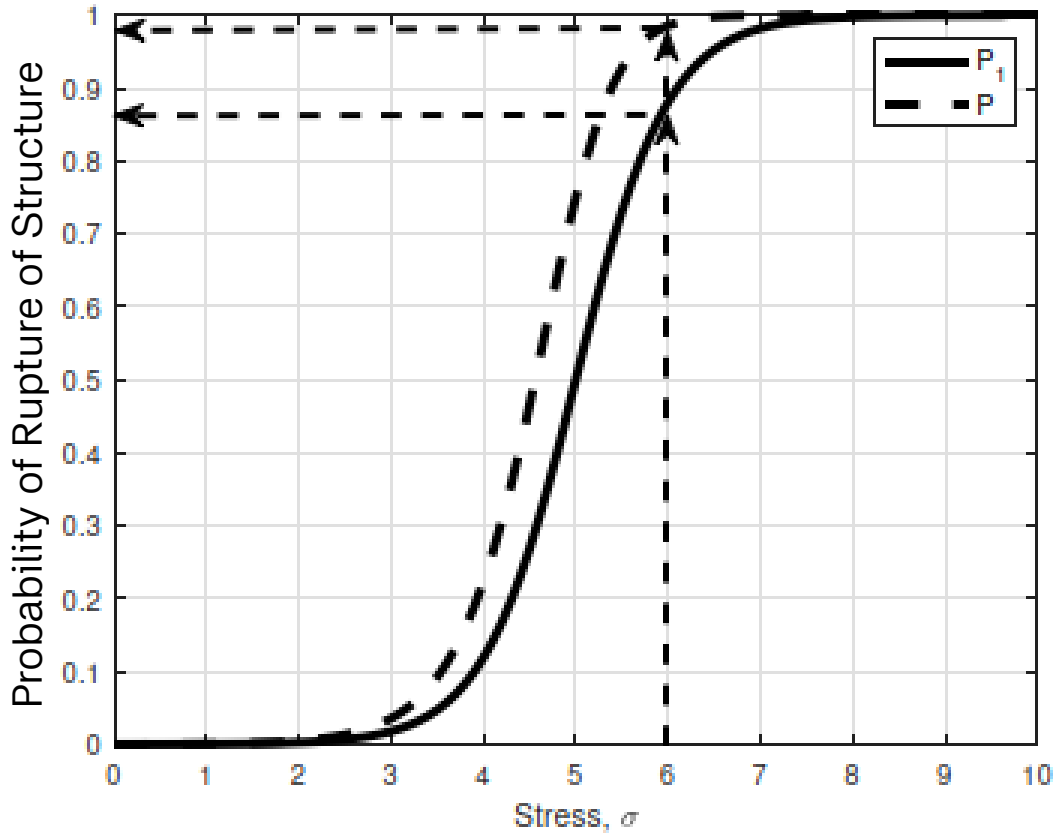
Probability of rupture under stress: $P_1 = f(\sigma)$



Component



Structure



Probability of survival under stress:

$$S = 1 - P_1$$

$$S = S_1 S_2$$

$$P = 1 - S$$

W. Weibull. A Statistical Theory of Strength of Materials. *Royal Swedish Academy of Engineering Sciences, 151*, pages 1–45, 1939.

Ioannides and Harris Model, Bases of current ISO 281

Starting with :

$$\ln\frac{1}{S} \propto \frac{N^e \tau_0^c V}{z_0^h} \qquad \ln\frac{1}{S} = \bar{A} N^e \int_V \frac{\langle \tau - \tau_u \rangle^c}{z'^h} dV \qquad z'(x,y) = \frac{\int_0^{z_{max}} (\tau - \tau_u) z dz}{\int_0^{z_{max}} (\tau - \tau_u) dz}$$

$$\langle \tau - \tau_u \rangle = \begin{cases} 0, & \forall \quad (\tau - \tau_u) < 0 \\ (\tau - \tau_u), & \forall \quad (\tau - \tau_u) \geq 0 \end{cases}$$

Introduction of the Fatigue Limit: τ_u

with $L = u\,N$, L = Life in Mrevs and N =Life in no. load cycles

Finalizing with :

$$L_{1-S} = \frac{1}{u}\left[\frac{\ln\frac{1}{S}}{\bar{A}\int_V \frac{\langle \tau - \tau_u \rangle^c}{z'^h} dV}\right]^{1/e}$$

simplification

$$L_{10} = a_{SLF}\left(\frac{C}{P}\right)^p$$

E. Ioannides and T.A. Harris. A New Fatigue Life Model for Rolling Bearings. *Trans. of ASME, J. of Tribology, 107*, pages 367–377, 1985.

E. Ioannides, G. Bergling, and A. Gabelli. An Analytical Formulation for the Life of Rolling Bearings. *Acta Polytechnica Scandinavica, Mechanical Engineering Series, 137*, pages 1–80, 2012.

Separation of Surface from Subsurface Survival

- Different failure modes can be considered
- Comparison of damage between surface and subsurface
- Differentiation of surface and subsurface design features can be highlighted
- Flexible scheme where new knowledge could be easily added

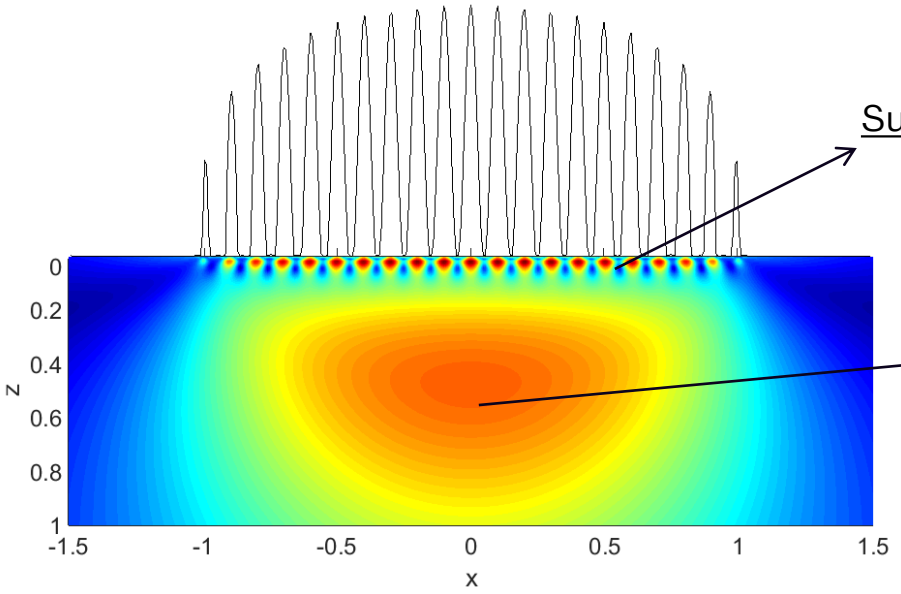
Derivation: Weibull Material Strength Theory

A STATISTICAL THEORY OF THE STRENGTH OF MATERIALS

BY

W. WEIBULL
Professor, Royal Technical University, Stockholm

A pseudo- Hertzian Contact



Surface: Tribology models

Sub-Surface: Hertzian Rolling Contact Fatigue

that the probability of rupture of each rod is S The probability of each rod withstanding the load is $S_i = 1—S$. According to the theory of probabilitiy, the probability S_{12} that two events having the probabilities

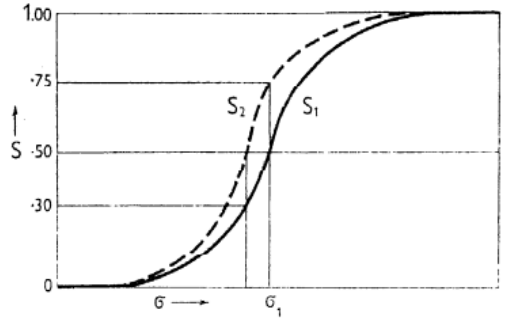


Fig. 1.

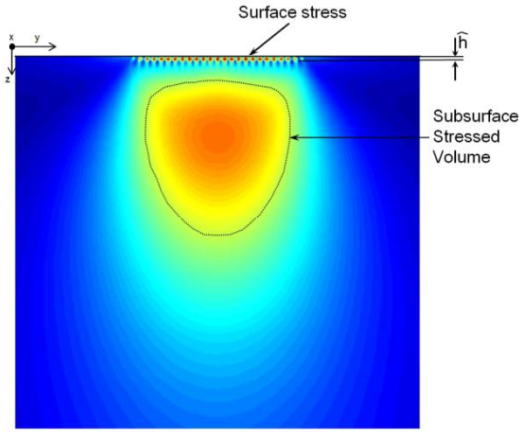
S_1 and S_2 respectively occur simultaneously is obtained by multiplying the two individual probabilities, or

$$S_{12} = S_1 \cdot S_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 1)$$

which may, of course, also be expressed in the equivalent form

$$\log S_{12} = \log S_1 + \log S_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 2)$$

Generalized Bearing Life Model (GBLM)



The probability of survival for different regions:

$$S_1^n = S_1 \cdot S_2 \cdots S_n = \prod_{i=1}^{n} S_i$$

$$\ln\left[\frac{1}{S(N)}\right] = \ln\left[\frac{1}{\Delta S_1(N)}\right] + \ln\left[\frac{1}{\Delta S_2(N)}\right] + \cdots + \ln\left[\frac{1}{\Delta S_n(N)}\right]$$

Introducing the material degradation functions $G_{v.n}$:

$$\ln\left[\frac{1}{S(N)}\right] = \int_{V_{v.1}} G_{v.1}(N)\, dV_{v.1} + \int_{V_{v.2}} G_{v.2}(N)\, dV_{v.2} + \ldots + \int_{V_{v.n}} G_{v.n}(N)\, dV_{v.n}$$

Reducing only to Subsurface (V) and Surface (A):

$$\ln\left[\frac{1}{S(N)}\right] = \int_{V_v} G_v(N)\, dV_v + \hat{h} \int_A G_s(N)\, dA$$

GBLM

For the volume (as before) G_v:

$$G_v = \overline{A}\, N^e \frac{\langle \sigma - \sigma_u \rangle^c}{z^h}$$

For the area G_S:

$$G_s = \overline{B} N^e \langle \sigma - \sigma_u \rangle^c$$

Substituting back in the integrals:

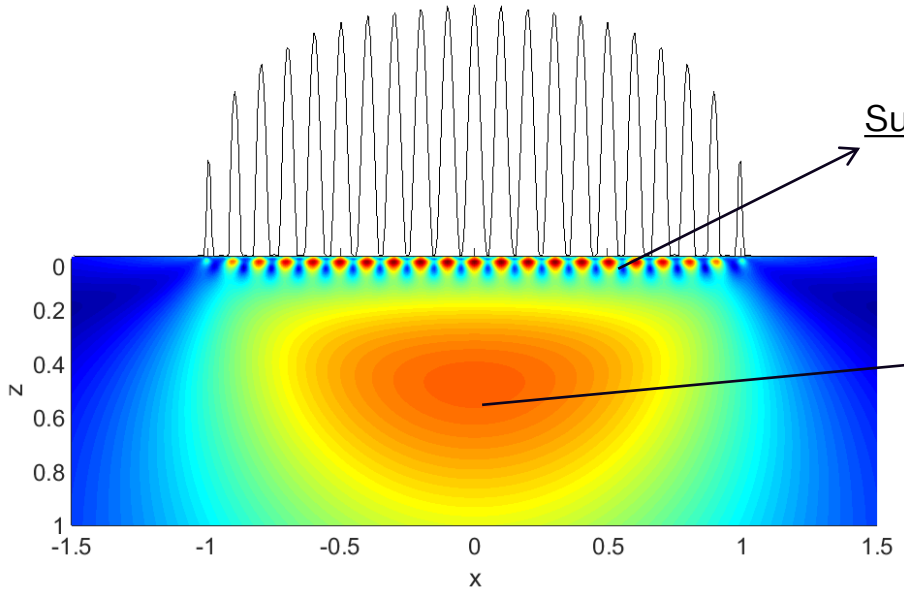
For the volume (as before) G_v:

with $L = u\,N$, L = Life in Mrevs and N =Life in no. load cycles

Solving for L_{1-S}:

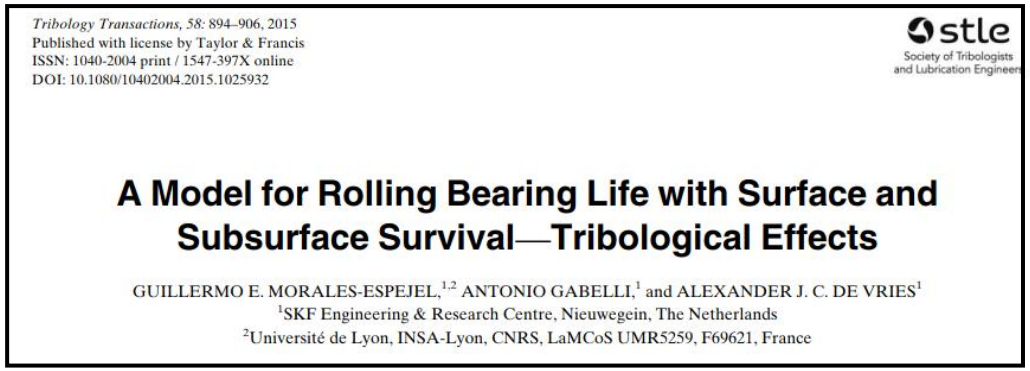
$$L_{1-S}^e = \frac{\ln\left(\frac{1}{S}\right)}{u^e \left[\overline{A} \int_{V_v} \frac{\langle \sigma_v - \sigma_{u.v} \rangle^c}{z^h} dV_v + (uL)^{(m-e)} \overline{B} \int_A \langle \sigma_s - \sigma_{u.s} \rangle^c dA\right]}$$

Generalized Bearing/Gear Life Model



$$L_{10} = \frac{\left[ln\left(\frac{1}{0.9}\right)\right]^{1/e}}{u}\left[\bar{A}\int_{V_n}\frac{\langle\sigma_v - \sigma_{u.v}\rangle^c}{z^h}dV_v + \bar{B}\int_A \langle\sigma_s - \sigma_{u.s}\rangle^c dA\right]^{-1/e}$$

subsurface surface


Tribology Transactions, 58: 894–906, 2015
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DOI: 10.1080/10402004.2015.1025932

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Society of Tribologists and Lubrication Engineers

A Model for Rolling Bearing Life with Surface and Subsurface Survival—Tribological Effects

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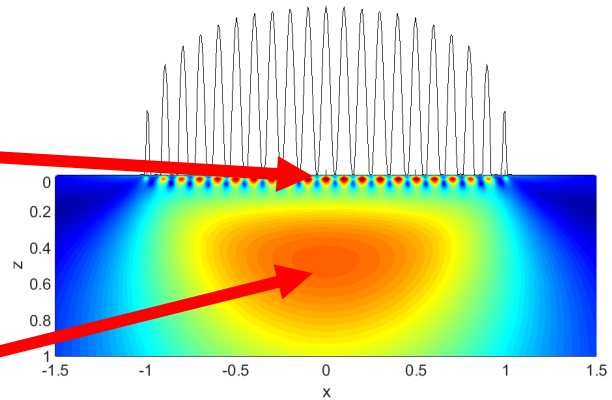
[2]Université de Lyon, INSA-Lyon, CNRS, LaMCoS UMR5259, F69621, France

Relative Surface Fatigue Index

$$L_{10} = \frac{\left[ln\left(\frac{1}{0.9}\right)\right]^{1/e}}{u}\left[\bar{A}\int_{V_v}\frac{\langle\sigma_v - \sigma_{u.v}\rangle^c}{z^h}dV_v + \bar{B}\int_A \langle\sigma_s - \sigma_{u.s}\rangle^c dA\right]^{-1/e}$$

$$I_{ss} = \bar{A}\int_{V_v}\frac{\langle\sigma_v - \sigma_{u.v}\rangle^c}{z^h}dV_v \qquad I_s = \bar{B}\int_A \langle\sigma_s - \sigma_{u.s}\rangle^c dA$$

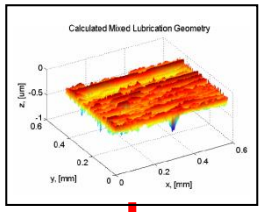
$$S_R = \frac{I_s}{I_s + I_{ss}} \begin{cases} 1 \\ 0 \end{cases}$$



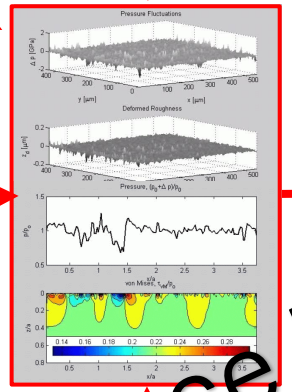
Morales-Espejel, G.E., Gabelli, A., de Vries, A.J.C., A Model for Rolling Bearing Life with Surface and Subsurface Survival – Tribological Effects, Tribology Transactions, 58, pp. 894-906, 2015

Application to Surface Fatigue

Topography of both surfaces



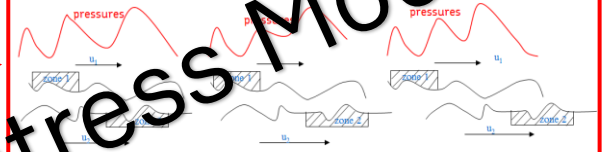
EHL Mixed-Lubrication solver



Lubricant Rheology

Surface Distress Model

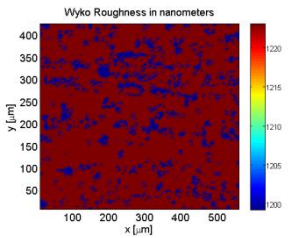
Asperity Contacts and Pressure History



Asperity Fatigue Stress (Dang Van)

Damage Accumulation (Palmgren-Miner)

Operating Conditions and Material Properties



Surface Distress Prediction

yes

$t=t_f$

no

$t_{i+1} = t_i + \Delta t$

local mild wear

$$\Delta h_w(x,y) = k(x,y)\frac{p(x,y)}{H}u_s\left(\frac{l}{u_1}\right)$$

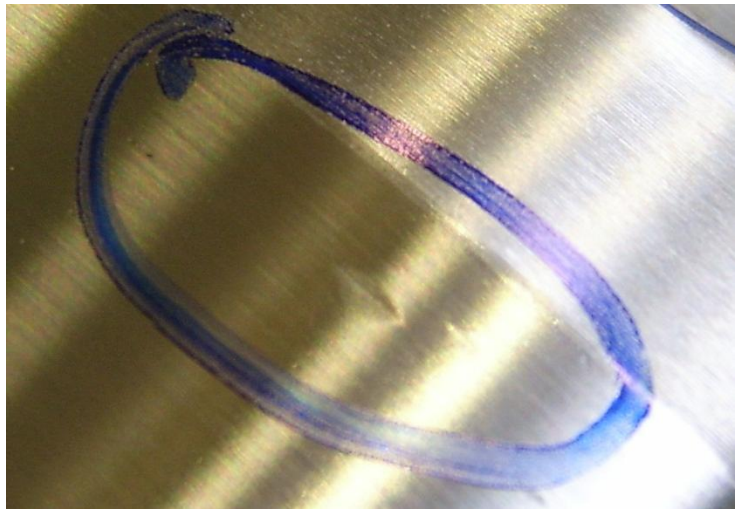
$$I_S = \int_A \langle \sigma_s - \sigma_{u.s} \rangle^c dA$$

Using the advanced surface distress model It is possible to curve fit:

$$R_s = u^m I_s \approx f_1 exp\left[\frac{f_2}{(P/P_u)^{f_3}} + \frac{f_4}{(P/P_u)^{f_5}} + \frac{f_6}{(P/P_u)^{f_7}}\right]$$

Morales-Espejel, G.E., Brizmer, V., "Micropitting Modelling in Rolling–Sliding Contacts: Application to Rolling Bearings", Tribology Transactions, 54(4), pp 625–643, 2011.

Indentation Damage

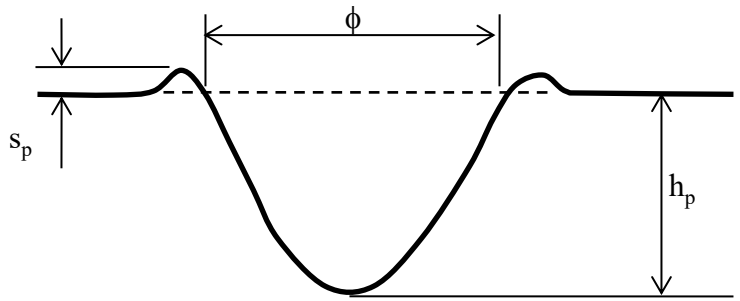


Keep a volume integral at the Surface:

$$L_{10} = \frac{\left[ln\left(\frac{1}{0.9}\right)\right]^{1/e}}{u}\left[\bar{A}\int_{V_v}\frac{\langle\sigma_v - \sigma_{u.v}\rangle^c}{z^h}dV_v + \bar{B}\int_A \langle\sigma_s - \sigma_{u.s}\rangle^c dA + \bar{C}\sum_{i=1}^{n}\int_{V_v}\frac{\langle\sigma_s - \sigma_{u.s}\rangle^c}{z^h}dV_v\right]^{-1/e}$$

subsurface surface indentations

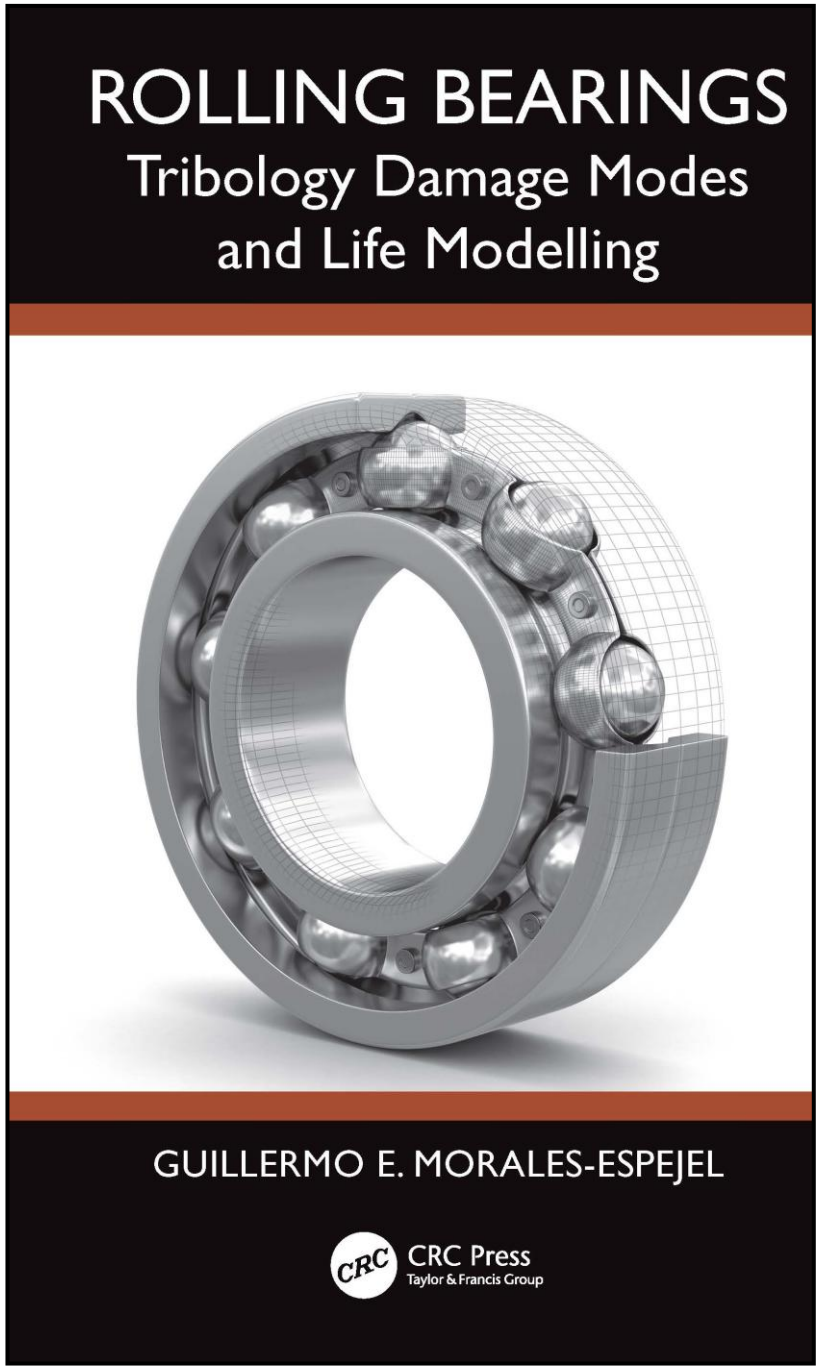
Idealised indentation shape:



Morales-Espejel, G.E., Gabelli, A., A model for rolling bearing life with surface and subsurface survival: Sporadic Surface Damage from Deterministic Indentations, Tribology International 96, pp. 279–288, 2016

Conclusions

- Understanding Tribology is crucial to model modern machine component life.
- Deterministic models are being developed for other failure modes than just subsurface fatigue and they can be incorporated in L10 models.
- Deterministic models for failure modes must consider the competition of the different damaging mechanisms to properly capture a realistic behaviour.




STOP