IMPROVING THE RELIABILITY OF THE GEAR TRANSMISSION ACCORDING TO THE TRIBOLOGICAL PROCESSES

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Abstract: *The authors developed a complex program to evaluate the scuffing risk in gear transmissions. The program include evaluation of the film thickness between the flanks of the teeth, evaluation of the real pressure according to the roughness and determination of the local gear scuffing criterion (SL). The program was validated by experiments on a two disk machine.*

Keywords: *gear lubrication, map of lubrication, contact pressure, scuffing failure risk.*

1. Introduction

In the gear transmissions tribological processes are considered as important criteria in increasing the reliability. Friction between gear tooth surfaces is the most important source of power loss in the gear transmissions and is depending of the lubrication regime. Lubrication regimes are depending on the speed, lubricant viscosity and quality, power transmited by gears, the roughnesses of the teeth's flanks. Xiao et al. [1], Castro and Seabra [2], Castro et al. [3], evaluated the friction coefficient between the flanks of the teeth and included as lubricant parameter Λ as an important parameter in evaluating of the friction coefficient. Variable rolling/sliding ratios, variable normal loads and mixed lubrication regimes characterize the contacts between the teeth of spur gear transmissions. In these operating conditions the local lubricant film breakdown can usually appears and local scuffing can be developed. Castro and Seabra [4] defined both a local and a global criterion to evaluate the gear scuffing failure. The local gear scuffing criterion is based only on the energetic phenomena obtained in the contacts of the roughness asperities and can be evaluated using the following equation, [4]:

$$SL = \frac{\left(\tau^{MIX} \cdot V_S\right)_{\max}}{C_{SL}} \qquad (1)$$

where τ^{MIX} is the tangential shear stress in the mixed lubrication contacts between the teeth surfaces, V_S is the local sliding speed and C_{SL} is a complex parameter depending on lubricant, gear materials, heat transfer.

When the local scuffing criterion $SL \geq 1$, the probability of scuffing development in gear is very high.

In the last time, Balan et al. [5,6,7] realized a lot of experiments on two disc machine and obtained variation of the friction coefficient with the slide/roll ratios for various lubricant viscosity and surface roughness. Also was evaluated the risk of the scuffing by using the local scuffing criterion defined by Eq.(1) in the slide/roll contacts of two discs.

In this paper the authors present a general program to evaluate the scuffing risk for a gear transmission having imposed geometry and operating conditions by considering tribological processes.

2. Lubrication film thickness

To evaluate the correct values of the film thickness in the contact surfaces for a gear transmission it is necessary to establishes in the first step the lubrication regime according to the geometry, lubricant, temperature and operation conditions (speed and power). Using the methodology presented by Hamrock, [8] and Olaru [9], Balan et al.[10] developed a computer program to realize the map of lubrication for a given spur gear. In a spur gear transmission, the curvature radii of the teeth flanks R_1 and R_2 as is presented in the figure 1 can be determined by equations:

$$R_1 = m \cdot (z_1 / 2)\sin\alpha + s \tag{2}$$

$$R_2 = m \cdot (z_2 / 2)\sin\alpha - s \tag{3}$$

where m is module of the gear, z_1 and z_2 are the number of the teeth, α is the gear angle. The distance s has a variable value depending on the position of the contact point on the teeth surface.

For the top of the pinion flank the distance s is determined by equation:

$$s = m\left[\sqrt{(z_1/2+1)^2 - (z_1/2\cos\alpha)^2} - \frac{z_1}{2}\sin\alpha\right]$$

(4)

For the end of pinion flank the distance s is determined with equation:

$$s = m\left[\sqrt{(z_2/2+1)^2 - (z_2/2\cos\alpha)^2} - \frac{z_2}{2}\sin\alpha\right]$$

(5)

The tangential speeds on the flanks surfaces in the contact point are determined by equations:

$$v_1 = \omega_1\left[m \cdot (z_1/2)\sin\alpha + s\right] \tag{6}$$

$$v_2 = \omega_2\left[m \cdot (z_2/2)\sin\alpha - s\right] \tag{7}$$

where ω_1 and ω_2 are the angular speed of the pinion and wheel, respectively.

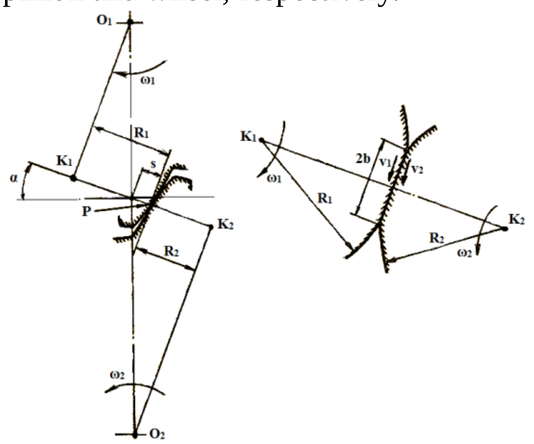

Figure 1. *The geometry of a contact between the gear teeth*

If the teeth of the gear have curvature radii in the axial direction as in figure 2, R_{b1} and R_{b2}, the contact between the teeth surfaces is initially a point contact.

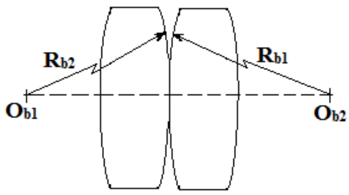

Figure 2. *The curvature radii for the teeth flanks in axial direction*

In a lubricated point contact it can be developed four lubrication regimes as function of the two parameter: viscosity parameter g_v and elasticity parameter g_e given by equations:[10]

$$g_v = G \cdot W^3 \cdot U^{-2} \tag{8}$$

$$g_e = W^{8/3} \cdot U^{-2} \tag{9}$$

The dimensionless parameters G, U and W are the materials, the speed and the load parameters. Balan et al. [10] developed the expressions for the parameters G,U and W applied to a spur gear transmission having following geometrical characteristics: modulus $m=5$; transmission ratio $i=5$, teeth numbers $z_1=25$, $z_2=125$, $\alpha=20^o$; $R_{b1}=1000$ *mm*, $R_{b2}=1000$ *mm*. Following operation conditions was imposed: rotational speed of the pinion $n_1=3000$ *rot/min*, the power at the pinion $P_1=10$ *KW* and the operating temperature was 80^{o}C. The gears was realized from steel having elastically modulus $E_1 = E_2 = 2.1\cdot 10^{11} Pa$ and $\nu_1=\nu_2=0.3$. The lubrication was realised with mineral oil SAE 140 having the viscosity at *80^{o}C* $\eta_0 = 0.07 Pa\cdot s$.

In figure 3 is presented the map of lubrication regimes for the top of the pinion flank.

It can be observed that for given gear transmission, at the top of the pinion flank the lubrication regime is PVE (Elastohydrodynamic).

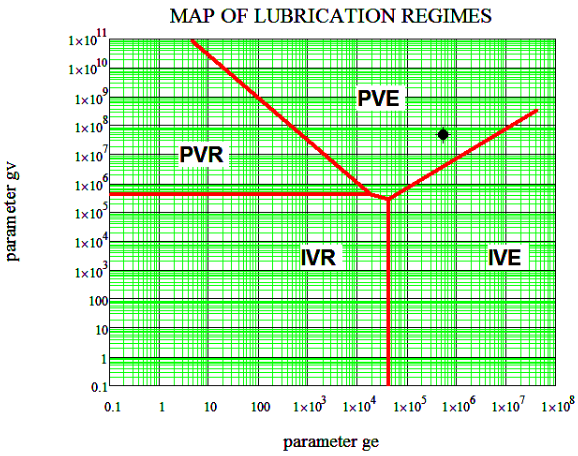

Figure 3. *Map of lubrication for the top of the pinion*

For PVE regime, the minimum film thickness between the contact surfaces h_{min} is given by following equations:

$$h_{min} = 3{,}42 \cdot g_v{}^{0{,}49} \cdot g_e{}^{0{,}17}\left(1 - e^{-0{,}68 \cdot k}\right) \cdot R_x \cdot \left(\frac{U}{W}\right)^2 \quad (10)$$

where $R_x = (1/R_1 + 1/R_2)^{-1}$, $R_y = (1/R_{b1} + 1/R_{b2})^{-1}$ and $k = 1{,}03 \cdot (R_y / R_x)^{0.64}$ [10].

For above mentioned gear transmission the minimum film thickness results: $h_{min} = 2.04\mu m$. In the pitch position the minimum film thickness decreases to $h_{min} = 1.761\mu m$.

The minimum thickness lubrication film can completely or partially separate the flanks surfaces depending of the surfaces roughness. The lubricant parameter Λ includes both the minimum film thickness and surfaces roughness and can be estimated by equation:

$$\Lambda = h_{min} \Big/ (R_{q1}^2 + R_{q2}^2)^{0.5} \quad (11)$$

where Rq_1 and Rq_2 are the root mean squared roughness of the contact surfaces. For $\Lambda > 3$, the surfaces are completely separated by the lubricant film, for $\Lambda = (1...3)$, the surfaces are partially separated by lubricant film and for $\Lambda < 1$, the lubrication is boundary with severe risk for scuffing failure. For the considered transmission, the lubricant parameter Λ have values between 1.3 to 1.6 that means a mixed lubrication regime.

3. Hertzian contact pressure

By considering the theoretical contact surfaces (neglecting the roughness) the Hertzian contact pressure distribution and semi axes of contact ellipse were determined. The calculus was based on the methodology presented by Creţu [11,12]. Was developed a program for determine the semi major and semi minor contact ellipse axis *a* and *b*, respectively and, finally to determine the contact pressure distribution for the above mentioned spur gear transmission. In Fig. 4 are presented the dimensions of the contact ellipse between the teeth flanks in the pitch point and in the Fig. 5 is presented the 3D pressure distribution.

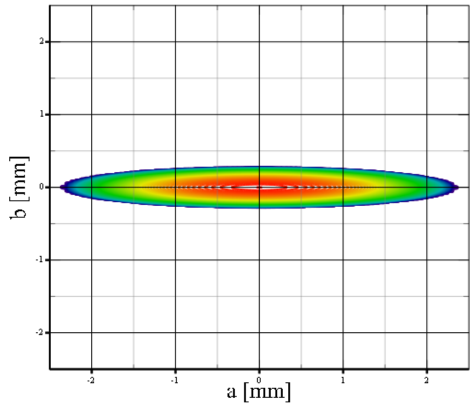

Figure 4. *The contact ellipse between the flanks of the teeths*

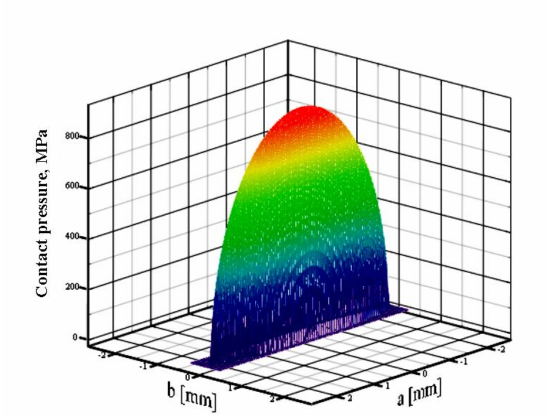

Figure 5. *The contact pressure distribution between the flanks of the teeths*

Folowing values were obtained: normal contact load *Q =1260N*, semi major axis *a = 2.402mm*, semi minor axis *b = 0.276mm* and the maximum Hertzian contact pressure $\sigma_0 = 907{,}4MPa$.

5. Real distribution pressure by considering dry roughness contacts

Based on the methodology developed by Cretu [11,12], Balan et al. [5,7] determined the real distribution of the contact pressure between two rollers by considering the equivalent roughness on the two surfaces. The contact between rollers simulate the real contact between the flanks of the teeth from the above mentioned gear transmission. In fig. 6 is presented the real pressure distribution in the rolling direction obtained for a equivalent roughness $R_a=0.35\mu m$.

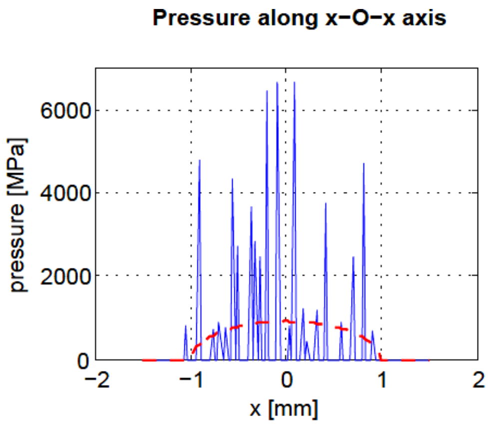

Figure 6. *Real pressure distribution in dry conditions for* $R_a = 0.23\mu m$

In figure 7 is presented the real pressure distribution for an equivalent roughness $R_a=0.035\mu m$.

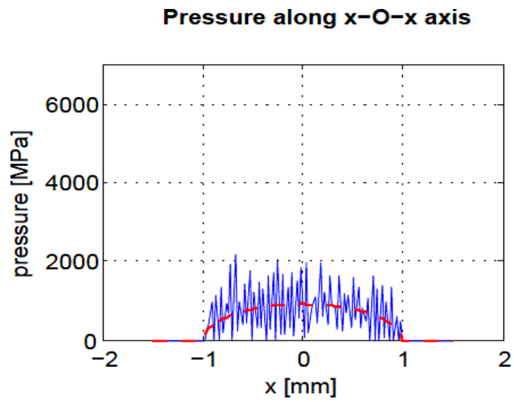

Figure 7. *Real pressure distribution in dry conditions for* $R_a = 0.023\mu m$

With blue color is indicated the real picks of pressure and with red color is indicated the Hertzian pressure distribution. It can be observed that by decreasing of the roughness the pressure distribution has a tendency to be one pure Hertzian.

5. Real distribution pressure by considering mixed lubricated contacts

In a mixed lubrication conditions the normal load supported by the top of the roughness (Q_a) can be determined by equation, [4]:

$$\frac{Q_a}{Q} = \exp\left(-1.8 \cdot \Lambda^{1.2}\right) \qquad (12)$$

Where Q is total normal force applied on the contact and λ is lubricant parameter.

Using the methodology developed in [5] for dry contacts, was simulated the real pressure distribution in mixed lubrication conditions. So, according to the equation (12) was determined the normal load acting on the asperities Q_a. This normal load was distributed on the equivalent roughness and the pressure distribution p_j on contact ellipse caused only by asperities contacts was determined (blue picks). In the first situation was considered a plastic deformation limit at a level of the contact asperities.

Supplementary, a normal load ($Q - Q_a$) is supported by the lubricant film and the pressure distribution is approximate by a Hertzian pressure distribution (red distribution). Finally, the real pressure distribution on the contact ellipse is a result of a sum of the dry and film pressure. In Fig. 8 is presented the real pressure distribution for equivalent roughness $R_a = 0.35\mu m$ and lubricant parameter $\Lambda = 1.9$

By decreasing of the lubricant parameter Λ to 0.92 the real pressure distribution includes an increasing of the dry contact picks between the top of the asperities as in Figure 9.

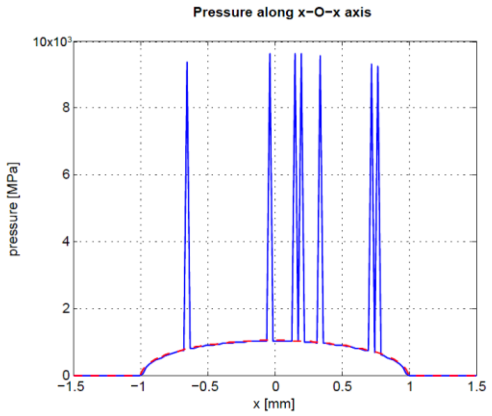

Figure 8 *Pressure distribution in mixed lubrication conditions for Λ= 1.9*

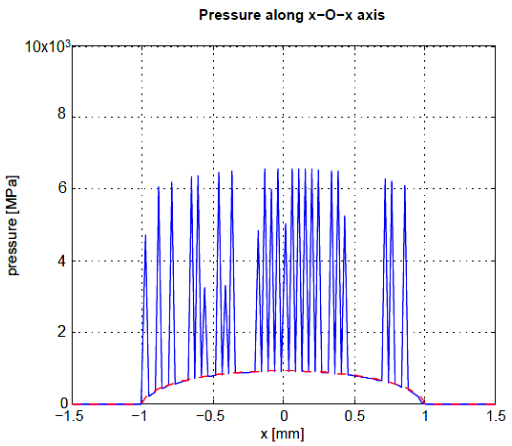

Figure 9. *Pressure distribution in mixed lubrication conditions for Λ= 0.92*

It can be observed that by decreasing the lubricant parameter *Λ*, the number of the pressure picks is increasing and the normal load is dominant supported by the roughness.

6. Evaluation of the scuffing risk

Having solved the problems reefers to the lubricant parameter Λ, relative sliding speed and local pressure on contact ellipse can be evaluated the scuffing risk according to the equation (1).

So, a complex program was elaborated to include all the tribological parameters on contact ellipse between the flanks of a given gear transmission.

So, was realized on contact ellipse a discredited mesh and for every point was determined the tangential shear stress τ^{MIX} by using following equation Castro & Seabra [4]:

$$\tau^{MIX} = \tau^{EHD} f^{1.2}(\Lambda) + \tau^{BDR} \cdot \left(1 - f(\Lambda)\right) \qquad (13)$$

where *f(Λ)* is the normal load shear function to determine the partition of the load supported by the top of the roughness and by the full EHD film in mixed lubrication. For *f(Λ)* was used following equation proposed by Castro & Seabra [4]:

$$f(\Lambda) = 0.82 \cdot \Lambda^{0.28} \qquad (14)$$

For every point *(i)* of the discredited mesh was determined τ_i^{BDR} with equation $\tau_i^{BDR}=\mu^{BDR}\cdot p_i$, where μ^{BDR} is friction coefficient in boundary lubrication conditions (was considered $\mu^{BDR}=0.12$) and p_i is local pressure on the top of the roughness.

For the contact ellipse was determined the shear stress in film thickness τ^{EHD} by equation $\tau^{EHD}=\mu^{EHD}\cdot p_0$, where μ^{EHD} is friction coefficient for full film lubrication conditions ($\Lambda \geq 3$) and p_0 is maximum Hertzian pressure. Based on the experiments [5,6,7] it can be adopted for μ^{EHD} value of 0.04. For the lubricant complex parameter C_{SL} can be adapted values according to the viscosity of the used oil, [4].

The sliding speed V_S is determined as a difference between the tangential speeds on the teeth's flanks $V_S = v_1 - v_2$.

7. Results of the simulating of the scuffing risk

Based on the developed program some simulating of the scuffing risk was realized. So, for given spur gear geometry and maximum Hertzian contact pressure was realized simulations to determine the scuffing risk for various speed, lubrication regime and surface roughness.

In the Table 1 are presented the results obtained for two disc with sliding/rolling motion having following characteristics: maximum Hertz contact pressure $\sigma_0 = 908\ MP$, surfaces roughness $Ra1 = Ra2 = 0.23\mu m$, sliding speed between *(0.025 – 0.13)m/s* and mineral oil SAE 90 with dynamic viscosity of *0.35 Pas* at 27^0C.

Table 1: Results of the local scuffing risk for low speed and low roughness surfaces

V_s . *m/s*	0.025	0.05	0.13
λ	0.45	1.3	2.5
$f(\lambda)$	0.65	0.88	1.05
τ^{EHD} , *Pa*	$2.89 \cdot 10^7$	$3.53 \cdot 10^7$	$3.63 \cdot 10^7$
$(\tau^{BDR})_{max}$,*Pa*	$8.85 \cdot 10^8$	$1.24 \cdot 10^9$	$2.64 \cdot 10^9$
$(\tau^{MIX})_{max}$,*Pa*	$3.22 \cdot 10^8$	$1.76 \cdot 10^8$	$3.90 \cdot 10^7$
$C_{SL}, 10^9 N/m^3s$	1.1	1.1	1.1
SL	**0.0073**	**0.0080**	**0.0046**

As a result of very low sliding speed the local gearing scuffing criterion SL has very low values and the risk of scuffing is null.

In the Table 2 are presented the simulating results obtained for two disc with sliding/rolling motion having following characteristics: maximum Hertz contact pressure $\sigma_0 = 908$ *MP*, surfaces roughness $Ra1 = 0.5\ \mu m$, $Ra2 = 0.23 \mu m$, sliding speed between *(0.57 – 1.03) m/s*, mineral oil SAE 90 with dynamic viscosity of *0.35 Pas* at 27^0C and hydraulic oil H46 with dynamic viscosity of *0.08 Pas* at 27^0C.

Table 2: Results of the local scuffing risk for moderate speed and high roughness surfaces

	Oil H46	**Oil SAE 90**
V_s . *m/s*	1.03	0.57
λ	0.92	1.9
$f(\lambda)$	0.80	0.99
τ^{EHD} , *Pa*	$1.83 \cdot 10^7$	$4.17 \cdot 10^7$
$(\tau^{BDR})_{max}$,*Pa*	$7.24 \cdot 10^8$	$1.24 \cdot 10^9$
$(\tau^{MIX})_{max}$,*Pa*	$1.58 \cdot 10^8$	$5.02 \cdot 10^7$
C_{SL} , $10^9 N/m^3s$	0.7	1.1
SL	**0.23**	**0.03**

By increasing of the sliding speed and of the roughness, it can be obtained important increases of the scuffing criterion SL. That means increasing of the scuffing risk.

8. Conclusions

A complex program to evaluate the scuffing risk in gear transmissions has been developed by the authors.

The program include evaluation of the film thickness between the flanks of the teeth, evaluation of the real pressure according to the roughness and determination of the local gear scuffing criterion SL.

The program is based only on the geometrical and operating conditions of the gear transmission.

With developed program was simulated some conditions and was evidenced that the sliding speed, the contact surfaces roughness, the contact pressure and the lubricant parameter Λ are the most important parameters which influence the scuffing risk.

The proposed program can be used as an important methodology to verified the scuffing risk for given spur gear transmission.

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