EXPERIMENTAL DETERMINATION OF THE FRICTION COEFFICIENT IN A SIMULATED GEAR TRANSMISSION

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ABSTRACT

The authors experimentally determined the friction coefficient in simulated contacts between gear teeth flanks having the slide-to-roll ratio between 0.045 to 0.15 and operating in mixed lubrication regimes. The dependence between friction coefficient in mixed lubrication regime and lubricant parameter λ has been verified according to the others models. Also, the variation of the friction coefficient with the slide-to-roll ratio was obtained.

Keywords: Friction coefficient, mixed lubrication, two rollers machine, slide-to-roll ratio

1. INTRODUCTION

In order to evaluate the friction losses in the spur gear transmissions, a lot of experiments were realized in a simulated two rollers contact. Xiao, Li et al. (Xiao, 2003) investigated the effect of surface topography on friction in roller contact. Using a test rig with two rollers running against each other and a Hertzian pressure between 225 to 350 MPa, Xiao, Li et al. obtained a lot of Stribeck curves considering the variation of the determined friction coefficient with the Hershey parameter, H, defined by relation: 

\[ H = \eta \cdot \frac{v}{p} \]

where \( \eta \) is the lubricant viscosity in Pas, \( v \) is the velocity in the contact of the discs in m/s and \( p \) is Hertzian contact pressure in Pa. Xiao, Li et al. obtained by they experiments the important increases of the friction coefficient when increases the surfaces roughness.

Using some test rigs with two rollers, Xiao et al. (Xiao, 2007) studied the correlations between friction coefficient in slide/rolling contacts operating in mixed lubrication conditions and the lubricant parameter \( \lambda \). They developed following equation:

\[ \mu_{mix} = \mu_{EHD} \cdot f(\lambda) + \mu_{BD} \cdot (1 - f(\lambda)) \] (1)

where \( \mu_{EHD} \) is friction coefficient for \( \lambda \geq 3 \) (fully flooded lubrication conditions), \( \mu_{BD} \) is friction coefficient for boundary lubrication conditions and \( f(\lambda) \) is the load share function establishing the normal load supported by boundary contact between the roughness and by the EHL full film.

One equation for the \( f(\lambda) \) is proposed by Zhou (Castro, 2008), having following form:

\[ f(\lambda) = \frac{1.21 \cdot \lambda^{0.64}}{1 + 0.37 \cdot \lambda^{0.26}} \] (2)

Castro and Seabra (Castro, 2008) based on the experiments realized on gear transmissions, proposed following equation for \( f(\lambda) \):

\[ f(\lambda) = 0.82 \cdot \lambda^{0.28} \] (3)

Must be remarked that Xiao et al. (Xiao, 2007) used in the experiments high values for slide-to-roll ratio: 0.3 to 1.5.

Castro and Seabra (Castro, 2008) analyzed the influence of the pressure, the roughness and effect of the slide-to-roll ratio on the friction coefficient. Was determined friction coefficient for the mean pressure between 600 to 1600 MPa and for high values of the slide-to-roll ratio. Was evidenced that by increasing the pressure increases the friction coefficient and by increasing the slide-to-roll ratio decreases the friction coefficient.
In the rough slide/rolling contact surfaces specified to the gear transmissions, Castro and Seabra (Castro, 2008) proposed a new equation for the friction coefficient:

$$\mu_{mix} = \mu_{EHD} \cdot f(\lambda) + \mu_{BD} \cdot (1 - f(\lambda))$$

In equations (2) and (3) λ is lubrication parameter given by equation:

$$\lambda = \frac{h_{min}}{1.15 \cdot \sqrt{R_{a1}^2 + R_{a2}^2}}$$

where $h_{min}$ is minimum film thickness and $R_{a1}$, $R_{a2}$ are the contact surfaces roughness. The methodology to determine lubrication regimes and minimum film thickness in a gear transmission has been presented by Balan et al. (Balan, 2012).

Martins et al. (Martins, 2006) realized a lot of experiments on a F2G test rig and using an energetic model obtained equations for optimized average friction coefficient between gear teeth both mineral and ester oils. So, for mineral oil, Martins et al. proposed following equation for average friction coefficient between gear teeth:

$$\mu = 0.048 \left( \frac{F_N}{b} \right)^{0.2} \cdot \eta^{-0.05} \cdot R_0^{0.25} \cdot \frac{1}{(F_N/b)^{0.053}}$$

The authors experimentally determined the friction coefficient in simulated contacts between gear teeth flanks having the slide-to-roll ratio between 0.045 to 0.15 and operating in mixed lubrication regimes. The dependence between friction coefficient in mixed lubrication regime and lubricant parameter λ has been verified according to the others models.

2. EXPERIMENTAL INVESTIGATIONS

A two rollers machine was used to simulate the friction generated between the gear tooth surfaces. In Fig. 1 is presented the two rollers machine.

The two rollers running against each other with different rotational speed, having the ratio $k = n_1/n_2 = 1.104$, where $n_1$ and $n_2$ are rotational speed of the lower and upper roller respectively. The slide-to-roll ratio $\xi$ is determined by equation:

$$\xi = \frac{2 \left( v_1 - v_2 \right)}{v_1 + v_2} = \frac{2 \left( 1 - k \cdot D_2/D_1 \right)}{1 - k \cdot D_2/D_1}$$

where $D_1$ and $D_2$ are the lower and upper roller diameters. The rotational speed $n_1$ varied between 38 and 380 rpm.

Two steel rollers with diameters of 56 mm and 59 mm were used in combinations as $D_1$ and $D_2$ to realize following slide-to-roll ratios: 4.65 %, 9.86 % and 15.06%. The roller with diameter of 56 mm was cylindrical and roller with diameter of 59 mm was crowned with radius of 30 mm.

The roughness of the rollers was $R_a = 0.23 \mu m$. The normal load varied between 40 and 120 N and realizes Hertzian pressure in the contact of rollers with values between 630 to 920 MPa. Two lubricant oils were used: SAE90 EP with a dynamic viscosity of 0.35 Pa·s at 25 °C and H42 with a dynamic viscosity of 0.1 Pa·s at 25 °C.

The friction torque generated at the lower roller $D_1$ was measured by a strain-gauge system with a Vishay P3 data acquisition system. The data acquisition system was calibrated to indicate directly the friction coefficient by using equation $\mu = 2 \cdot M_f/(Q \cdot D_1)$, where $M_f$ is friction torque and $Q$ is normal load acting on the rollers.

3. RESULTS

3.1. Variations of the Friction Coefficient with the Time

Fig. 2 presents a typical diagram including the variation with the time of the friction coefficient experimentally determined by the above-described equipment.

It can be observed that during the test period some small variations of the friction coefficient were obtained. The maximum deviation of the friction coefficient form the average value not exceeds 12 %. Also, during the short period of the tests no variation of the friction coefficient with the time was observed. The measurements of the rollers roughness before and after the tests evidence that was not produced important modifications.

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Fig. 3. 2D profiles for cylindrical and crowned roller before (A) and after (B) the tests.
3.2. The Stribeck curves

To correlate the obtained results with the Xiao et al. (Xiao, 2003) results were realized the Stribeck curves when friction coefficient was plotted as function of the Hershey parameter $H$.

In Figs. 4, 5 and 6 are presented the Stribeck curves obtained with the experimental results for the three Hertzian contact pressures: 630 MPa, 866 MPa and 920 MPa. The curves presented in Figs. 4 and 6 were realized for viscosity of 0.35 Pas and in Fig. 5 are presented the curves for oils with 0.35 Pas and 0.1 Pas.

The Stribeck curves obtained suggests mixed lubrication regimes for all the experiments. Xiao et al. (Xiao, 2003) obtained similar Stribeck curves having friction coefficient values of 0.02 to 0.12 for Hershey parameter values between $10^{-11}$ m to $10^{-9}$ m.

3.2. Correlation between Friction Coefficient and Lubricant Parameter $\lambda$

In Figs. 7, 8 and 9 are presented the variation of the friction coefficient with the lubricant parameter $\lambda$.

By using the Hamrock maps of lubrications (Hamrock, 1994) and the computer program developed by Balan et al. (Balan, 2012), was established that all the experiments was realized in EHL lubrication conditions and the minimum film thickness $h_{min}$ was computed with adequate equation (Balan, 2012).
was plotted on the same diagram the experimental results, the analytical values obtained with eq (1) by considering the function $f(\text{Zhou})$ given by eq. (2) and the analytical values obtained with eq (1) by considering the function $f(\text{Castro})$ given by eq. (3).

In fig. 10 are presented the experimental and analytical values for Hertzian contact pressure of 866 MPa. So, in eq. (1) was used for $\mu_{\text{EHD}}$ value 0.045 (when $\lambda \geq 3$) experimentally determined and for $\mu_{\text{BD}}$ was used value of 0.12.

In Fig. 10, it can be observed that equation (1) with Zhou expression for function $f(\lambda)$ is not in good correlation with the experiments (red curve). In the same time, if in equation (1) is used the Castro expression for the function $f(\lambda)$ a good correlation between experiments and analytical model is obtained.

The equation (4) developed by Castro and Seabra with the function $f(\lambda)$ given by equation (2) or (3) leads to the values of the friction coefficients lower that the experimental values.

3.3. Correlation between Friction Coefficient and Slide-to-Roll ratio $\xi$

In the researchers of Xiao et al. (Xiao 2007) was obtained, for high values of slide-to-roll ratio $\xi$ (35% -150%) decreasing of the friction coefficient when increase slide – to roll ratio. Our experiments were realized with slide – to roll ratio having values between 4.65% and 15.05%. In Fig. 11 are presented the variations of the friction coefficient with the slide- to - roll ratio and pressure for lubricant parameter $\lambda=0.6$ (boundary lubrication conditions).
and in Fig. 12 are presented the variations of the friction coefficient with the slide - to - roll ratio and pressure for lubricant parameter $\lambda=3$ (fully flooded lubrication conditions).

It can be observed that by increasing of the slide – to – roll ratio was obtained increasing of the friction coefficient, more accentuated in boundary lubrication conditions. By comparing our results with the results of Xiao et al. (Xiao 2007) following comments can explain these different. In a traction curve realized in the presence of the lubricant (variation of the friction coefficient with the slide – to - roll ratio $\xi$) it can observe two important components: the first component with a quasi linear increasing of the friction coefficient with $\xi$ to a maximum value, when $\xi$ increases from zero to $(10 – 15)\%$ and a second component with a decreasing of the friction coefficient by increasing of the $\xi$, caused by increasing of the thermal effects. Or, all the experiments realized by Xiao et al. (Xiao 2007) was realized with high values of the $\xi$ and our opinion is that their results was influenced by thermal effects.

3.4. Correlation between Friction Coefficient and Contact Pressure

Based on the experimental results in Figs 13 and 14 are presented the variation of the friction coefficient with the Hertzian contact pressure both in mixed and in fully flooded lubrication conditions.

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Based on the experimental results in Figs 13 and 14 are presented the variation of the friction coefficient with the Hertzian contact pressure both in mixed and in fully flooded lubrication conditions.

Fig.11. Variation of the friction coefficient with slide – to – roll ratio $\xi$ and contact pressure for lubricant parameter $\lambda = 0.6$

Fig.12. Variation of the friction coefficient with slide – to – roll ratio $\xi$ and contact pressure for lubricant parameter $\lambda = 3$

Fig.13. Variation of the friction coefficient with contact pressure an lubricant parameter $\lambda$ for a slide – to – roll ratio $\xi = 4.65\%$

When the slide – to – roll ratio has small value ($\xi = 4.65\%$) the increasing of the contact pressure leads to a small increases of the friction coefficient when $\lambda = 3$ and decreases of the friction coefficients for $\lambda <3$.

For high value of the slide – to – roll ratio ($\xi = 15.06\%$), the increases of the pressure leads to decreases of the friction coefficient both for $\lambda = 3$ and for $\lambda < 3$.

Xiao et al. (Xiao 2007) obtained increasing of the friction coefficient with the mean contact pressure between $p_m$ of 600MPa to 1100 MPa but using high values for slide – to – roll ratio ($\xi = 35\% - 150\%$) and lubricant parameter $\lambda$ not exceeded 0.5 in all experiments.

Also, Xiao et al. (Xiao 2007) proposes following equation for friction coefficient in a slide/rolling contact operating in mixed lubrication conditions if $p_m\leq 1110\text{MPa}$: 
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\[ \mu_{\text{mixt}} = 0.078 + 0.22 \cdot \frac{p_m}{1110} - 0.055 \cdot \frac{p_m}{1110} \cdot \xi \]  
(8)

If we consider that \( p_m = 2/3 \cdot p \) results \( p_m = 613 \text{MPa} \) for Hertzian contact pressure \( p = 920 \text{MPa} \). Imposing our maximum slide-to-roll value of \( \xi = 0.1506 \) (\( \xi = 15.06\% \)) in eq. (8) it can obtain \( \mu_{\text{mixt}} = 0.198 \), value that not corresponds with our experimental value. Supplementary, eq. (8) not indicates the influence of lubrication parameter \( \lambda \).

Fig. 14. Variation of the friction coefficient with contact pressure an lubricant parameter \( \lambda \) for a slide – to – roll ratio \( \xi = 15.06\% \)

By comparing our experimental friction coefficient values with the results of other authors it can be observed that all the variation of the friction coefficients in slide/rolling contacts depends on geometrical and operating parameters (roughness, contact pressure, lubricant viscosity, speed, slide-to-roll ratio). As consequence it is difficult to extrapolate the given equations for all conditions.

4. CONCLUSIONS

The authors realized a lot of experiments and determined the variation of the friction coefficient in sliding/rolling contacts specific to the spur gears. Was realized simulated contacts between gear teeth flanks with a two rollers machine and friction coefficient was obtained by data acquisition with a Vishay P3 system.

Based on the experimental results following conclusions was obtained:

1. All the experiments was realized in mixed lubrication conditions with lubrication parameter \( \lambda \) having values between 0.6 to 3 and the results of friction coefficients varied between 0.04 to 0.08.

2. The Stribeck curves obtained with the experimental results are in good correlation with the results obtained by other authors.

3. The variation of the friction coefficient with lubricant parameter \( \lambda \) was validated with the Castro an Seabra equations.

4. Was obtained small increases of the friction coefficient with the slide–to-roll ratio \( \xi \) when \( \xi \) varied between 4.65\% and 15.05\%.

5. Was obtained decreasing of the friction coefficient by increasing the Hertzian contact pressure between 630 MPa to 920 MPa, more accentuated when slide-to-roll ratio was 15.06\%.

6. Was demonstrated that all the variation of the friction coefficients in slide/rolling contacts depends on geometrical and operating parameters (roughness, contact pressure, lubricant viscosity, speed, slide-to-roll ratio).

REFERENCES
