Journal of the Balkan Tribological Association *Vol. 16, No 4, 517–526 (2010)*

Friction

EXPERIMENTAL DETERMINATION OF FRICTION COEFFICIENT AT GEAR DRIVES

N. MARJANOVIC, B. IVKOVIC, M. BLAGOJEVIC, B. STOJANOVIC

Faculty of Mechanical Engineering, 6 Sestre Janjic Street, 34 000 Kragujevac, Serbia
E-mail: *nesam@kg.ac.rs*

ABSTRACT

Procedure of tribological investigation for identification of the dependence between coefficient of friction from sliding velocity and contact pressure is defined in this paper. Sliding speed and contact pressure are changing during the gear pair teeth conjugate action. Thereby and coefficient of friction has variable values.

Experimental determination of the friction coefficient is performed on pin-on-disk tribometer. Thanks to regular election pins and discs, there is implemented contact on line what corresponds to the real conditions in the gear pair teeth conjugate action. Dependencies of the friction coefficient on sliding speed and contact pressure were predicted, based on the data obtained by measurement.

Keywords: gear pair, coefficient of friction, tribometer.

AIMS AND BACKGROUND

Friction at gear drives is very often subject of scientific investigations in the world because it causes relevant dissipation of energy, wear and damaging of main elements in gear trains[1]. Designers more and more take care about friction in projecting process of machines and devices[2,3]. Tribology is not only a simple sum of friction, wear and lubrication in independent tribo-pair. The most efficient way of transferring tribology knowledge to industry is to carry out tribology design at the start of a design process[4]. Load, sliding speed and rolling speed are changeable during the gear pair teeth conjugate action. If we follow these changes, we can assess the influence of some gear trains parameters on friction.

During the conjugate gear motion, one or two pairs of teeth alternatively are in conjugate action and it directly influences the value of normal force on teeth[5,6].

* For correspondence.

Wink and Serpa treated load distribution problem of conjugated gear teeth along the line of action as elastic contact problem[7]. The values of coefficient of friction are experimentally determined by pin-on-disc and disc-on-disc tribometers[8]. Conditions and way of lubrication (thickness of oil film, pressure and temperature in conjugated gear pair contact) have significant influence on the coefficient of friction. It is presented in details in Refs 9–12.

The phenomenon of wear in both static and dynamic conditions is tested and presented in papers (Refs 13-16).

Pin-on-disc tribometer is used for tribological investigations in this paper. This tribometer is developed in the Center for Revitalisation of Industrial Systems at the Faculty of Mechanical Engineering in Kragujevac.

GEAR PAIR TEETH CONJUGATE ACTION

Power transmission by means of the gear pairs is realised by direct contact of the gear teeth. Each tooth of one gear is, for a certain time, in contact (conjugate action) with the conjugated gear tooth, and the contact conditions are variable with time[17].

The unfolding of the gear teeth conjugate action can be monitored through several characteristic points (Fig. 1): point A represents the entering of the observed pair of teeth into the contact; point B represents the beginning of the unilateral conjugate action; the projections of the both teeth speeds onto the common tangential line, equal to each other, are at the pitch point (point C), meaning there is no sliding at point C; when the contact reaches point D, the new teeth pair enters the contact; the considered teeth pair finishes contact at point E.

The conjugate action process is convenient to be monitored through the variation of the angle φ which defines the position of the instantaneous point of contact P with respect to the interference point T_1. If we assume that $\omega = \frac{d\varphi}{dt} = \text{const}$, it can be concluded that $d\varphi = \omega dt$, namely $\varphi = \omega t$. Thus, by monitoring the variation of

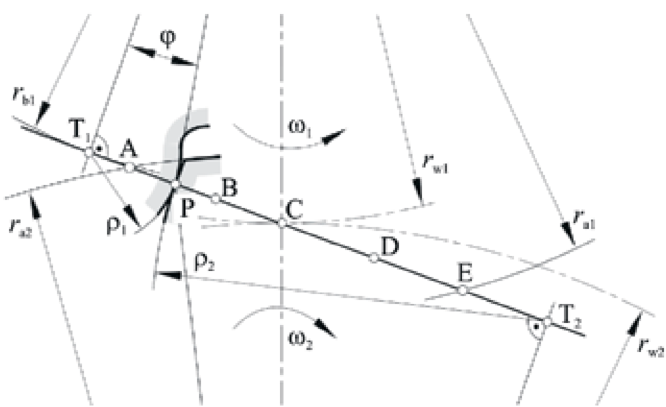

Fig. 1. Kinematics of the gear pair conjugate action

variables as the function of angle φ, we also obtain their variation in time. The assumption that ω = const means that we are considering the stationary motion of gears, which is usual. The following consideration will refer to the spur gears, but it can also be extended to other gears.

EXPERIMENTAL DETERMINATION OF COEFFICIENT OF FRICTION

The friction coefficient represents the ratio of the friction force and the normal load, namely:

$$\mu = \frac{F_{\mathrm{fr}}}{F_{\mathrm{n}}} \tag{1}$$

Definition of the coefficient values at each contact point, during the conjugate action of the gears teeth sides, is a very complex task, due to large number of the influential factors variable over time. The most important factors that influence the friction coefficient are: the sliding speed, the contact pressure, the gears materials, technological inheritance, etc. The influence only of the first two factors shall be considered in this paper, since their values can be influenced by selection of the gear transmitter parameters (module, teeth number, gear width, etc.) during the design process.

During the conjugate action of gear teeth sides, extensive change of the sliding speed and the contact pressure, occurs, what was analysed before. The friction coefficient that depends on these variables will also have variable values during the conjugate action. To establish the dependence of the friction coefficient on the sliding speed and contact pressure, tribometric investigations were performed.

Gears usually operate under high pressure conditions on the teeth sides. In design of gear pairs one of the criteria is also the teeth sides strength. Consequently, if the gear is correctly designed and loaded by a nominal load, the stress on sides will be close to dynamic strength of sides. Since the power transmitter gears are usually made of high quality, thermally treated steels, with high dynamic strength of sides ($1000 \div 1500$ MPa), it means that the contact pressures on the teeth sides also will be of the same order of magnitude. It is, therefore, necessary to consider relative motion of two cylindrical bodies at different sliding speeds and different contact pressures, of relatively high magnitudes.

Tribometer TPD93 (Fig. 2) used for tribological investigations[12] was developed in the Center for Revitalisation of Industrial Systems at the Faculty of Mechanical Engineering in Kragujevac. By applying the pin and disc, shown in Figs 3 and 4, the contact was realised along a line, that corresponds to real contact conditions of gear teeth. Tribometer is recording the normal load and the friction force during the contact by two-component dynamometer. Dynamometer is connected to a personal computer through the AD converter. Calculation

Fig. 2. Tribometer

Fig. 3. Disc and pins used for investigations

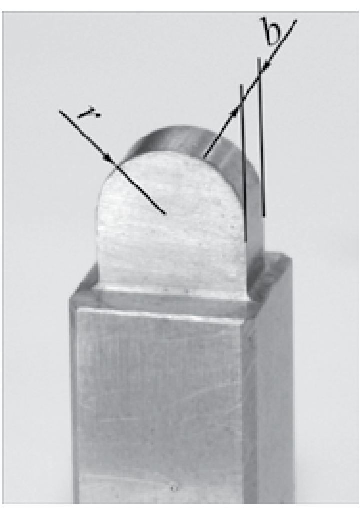

Fig. 4. Pins used for investigations

of the friction coefficient as well as the graphical representation of variations of all the variables during contact is done by personal computer. Then, measured and calculated values are recorded in the form of data files (ASCII), which enables further processing.

Pins dimensions are given in Table 1. The pin and disc material was the cemented and thermally

Table 1. Pins dimensions

Number	*r*	*b*
1	∝	9
2	3	4
3	4	4
4	5	4
5	4	3
6	5	3

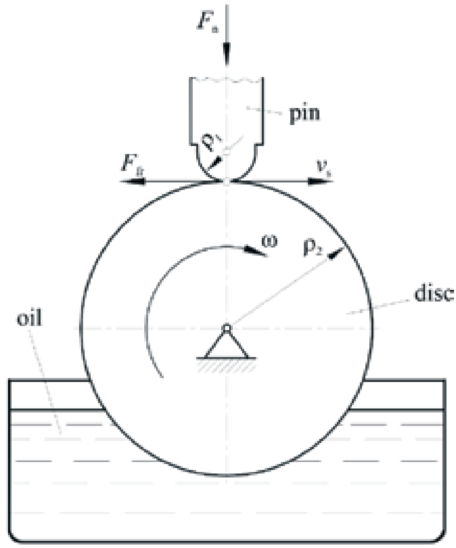

Fig. 5. Schematic representation of tribological investigations

treated 15CrNi6 steel. Monitoring of hardness provided values of about 62 HRC. Average value of the roughness height, measured at three referent lengths was $R_z \approx$ 1.4 μm or pins and $R_z \approx 0.92$ μm for disc. Lubrication was done with gear trains oil (GALAX HIPOL B, viscosity code: SAE 90).

The schematic representation of tribological investigations is presented in Fig. 5.

The shape and dimensions of pins and disc enabled realisation of high values of contact pressures.

RESULTS

Five sliding speeds (v_s = 0.56; 1.09; 1.71; 2.18 and 2.72 m/s) and 5 different normal loads (approximately from 100 to 450 N) were varied during tests.

Three-dimensional view of the friction coefficient dependence on the sliding speed and contact pressure is given in Fig. 6.

It is obvious that the friction coefficient increases with the increase of the contact pressure. With the increase of the sliding speed, the friction coefficient first decreases rapidly, and very slightly changes at higher speeds. This can be explained by the change of the contact and lubrication conditions with change of the sliding speed and contact pressure.

Dependencies of the friction coefficient on sliding speed and contact pressure were predicted based on data obtained by measurement. The three forms of the empirical relations were applied as follows[1]:

$$\mu = c\,\frac{\sigma_H^{\alpha}}{v_s^{\beta}} \tag{2}$$

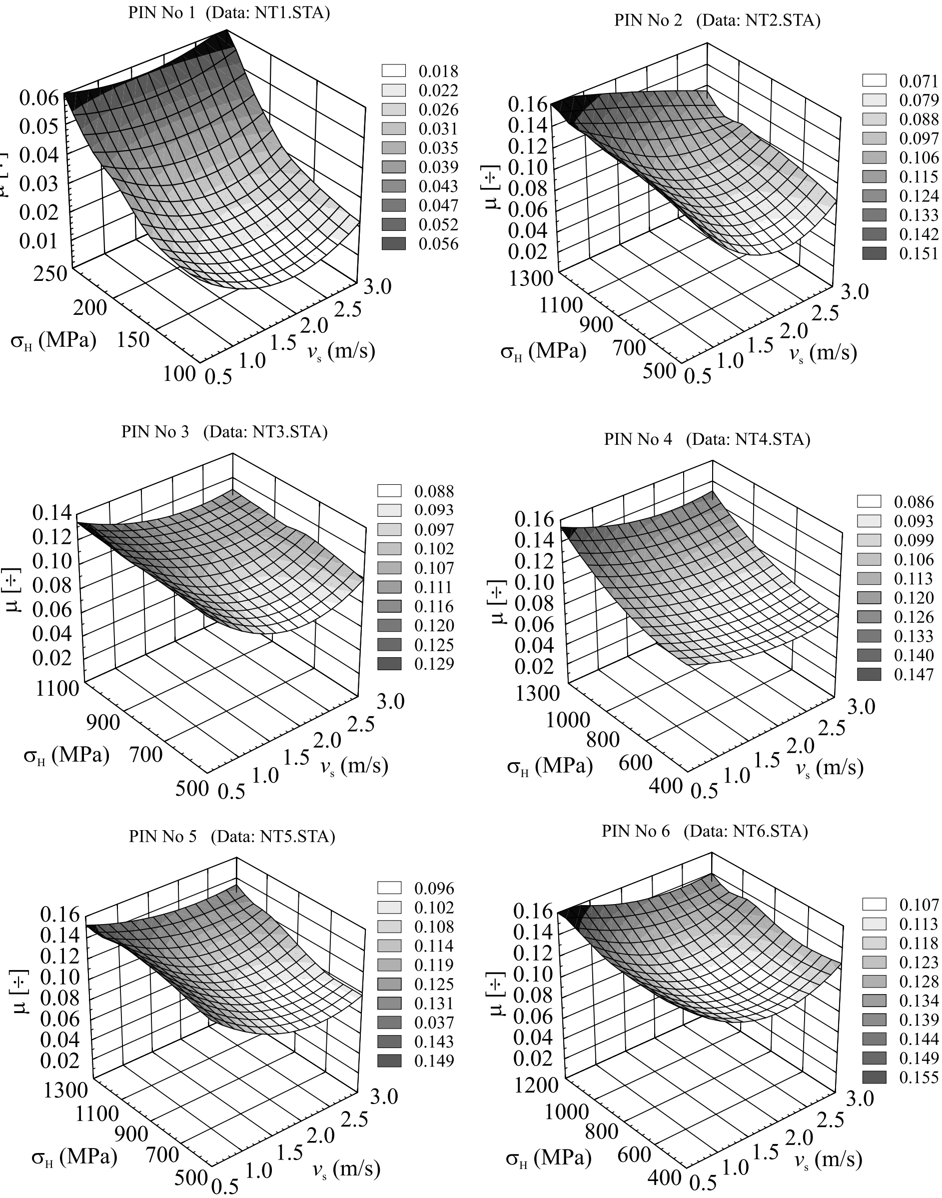

Fig. 6. Friction coefficient as a function of the sliding speed and the contact pressure (Pin No 1 to No 6)

$$\mu = c_0 + c_1 \frac{\sigma_H^{\alpha}}{v_s^{\beta}} \tag{3}$$

$$\mu = c_0 + c_1 v_s + c_2 \sigma_H + c_{11} v_s^2 + c_{12} v_s \sigma_H + c_{22} \sigma_H^2 . \tag{4}$$

In Tables 2, 3 and 4 are given the calculated by this way values of constants for each combination pin-on-disc investigations for case when all data of measur-

Table 2. Values of coefficients

Empirical relation: $\mu = c \frac{\sigma_H^{\alpha}}{v_s^{\beta}}$

Pin number	Constants			Index R
	c	α	β	
1	0.000028	1.369	0.180	0.924
2	0.005349	0.458	0.350	0.921
3	0.02220	0.245	0.191	0.921
4	0.02921	0.589	0.145	0.950
5	0.01953	0.277	0.236	0.930
6	0.02470	0.249	0.160	0.862
Σ	0.00234	0.584	0.231	0.937

Table 3. Values of coefficients

Empirical relation: $\mu = c_0 + c_1 \frac{\sigma_H^{\alpha}}{v_s^{\beta}}$

Pin number	Constants				Index R
	c_0	c_1	α	β	
1	0.14	0.29×10^{-8}	2.99	0.241	0.930
2	–0.224	0.1199	0.156	0.118	0.925
3	–0.374	0.0448	0.182	0.142	0.921
4	–0.128	0.0042	0.514	0.130	0.949
5	–1.27	1.1850	0.245	0.207	0.934
6	0.0742	0.0009	0.601	0.395	0.872
Σ	–0.322	0.194	0.1227	0.056	0.955

Table 4. Values of coefficients

Empirical relation: $\mu = c_0 + c_1 v_{kl} + c_2 \sigma_H + c_{11} v^2_{kl} + c_{12} v_{kl} \sigma_H + c_{22} \sigma^2_H$

Pin number	Constants						Index R
	c_0	c_1	c_2	c_{11}	c_{12}	c_{22}	
1	0.0615	–0.0243	–0.00043	0.00569	1.40×10^{-5}	2.00×10^{-6}	0.930
2	0.2095	–0.1034	-8.65×10^{-5}	0.01996	1.36×10^{-5}	6.99×10^{-8}	0.954
3	0.1283	–0.0626	4.80×10^{-5}	0.01360	$8{,}71 \times 10^{-6}$	-1.74×10^{-8}	0.960
4	0.1434	–0.0331	–0.00012	0.00737	-2.00×10^{-6}	1.49×10^{-7}	0.992
5	0.1940	–0.0808	-4.95×10^{-5}	0.01418	1.72×10^{-5}	3.38×10^{-8}	0.969
6	0.2050	–0.0714	–0.00010	0.01764	2.06×10^{-6}	8.27×10^{-8}	0.971
Σ	0.0380	–0.0488	0.00023	0.01244	-1.40×10^{-5}	8.98×10^{-7}	0.948

ing are taken into account. In the last column are given values of the indices of the curvilinear correlation as indicator of agreement suggested empiric delivery with experimental data.

The constants in these expressions were determined by the least squares method. After all data are taken into account, the indices of the curvilinear correlation that differ only slightly, are obtained, with values greater than 0.9 (0.937, 0.955, 0.954, respectively). Considering these values, one can conclude that each of the proposed distributions has good agreement with experimental data.

In all the cases, the high value is obtained for the index of the curvilinear correlation (R>0.9) that points to the good agreement of experimental data with the proposed empirical distribution. The highest value of index R is obtained for the second order polynomial distribution (4), due to the large number of constants and ability of this polynomial to adapt itself to experimental data.

For further work the distribution given by equation (2) was adopted due to its simplicity and clear physical meaning. The constants of this distribution are given in Table 2. Based on distribution analysis, it is obvious that the friction coefficient increases with increase of the contact pressure, and decreases with increase of the sliding speed up to a certain value. The values of c, α and β coefficients, given in Table 2, show that the influence of the sliding speed and contact pressure on the friction coefficient is more distinguished at small speeds that agrees with conclusions based on experimental data. It is possible to determine the variation of the friction coefficient during the conjugate action, since the empirical dependence of the friction coefficient on the sliding speed and contact pressure, given by equation (2) is accepted. Graphical representation of this dependence is given in Fig. 7, and its maximum value (at point C) would correspond to static friction coefficient at certain value of the contact pressure. The dashed lines represent the friction coefficient variations for the preceding and the following teeth pairs.

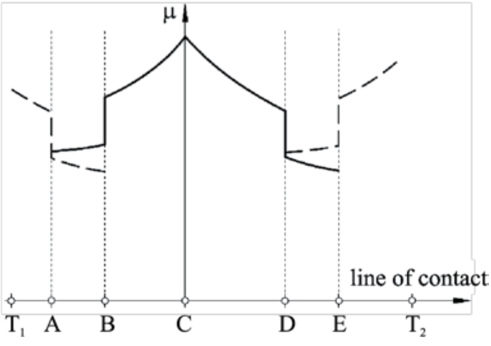

Fig. 7. Variation of the friction coefficient during the conjugate action

CONCLUSIONS

Procedure for experimental determination of coefficient of friction at gear pairs is presented in this paper. Investigations are performed on tribometer pin-on-disc. On the basis of these experimental results, dependencies between coefficient of friction, on the one hand, and sliding velocity and contact pressure, on the other, for all pins are defined.

The friction coefficient decreases with decrease of the contact pressure, and with the increase of the sliding speed over certain limit, it only slightly increases. Consequently, if the outside nominal load of the gear pair remains unchanged, the power losses due to friction will decrease with increase of the teeth dimensions. Investigation of the friction at gear pairs can also be realised by other tribometer types (e.g. disc/disc, four-balls, etc.). The specific equipment and methods such as: IEA, UK, Ryder, USA, or FZG, Germany or other EU countries[5] are frequently used for investigation of lubrication.

Three empiric dependences between coefficient of friction, on the one hand, and sliding velocity and contact pressure, on the other, are recommended in this paper. In regard to fine coincidence between empiric and experimental values ($R>0.9$), the first empiric dependence is taken as relevant. This dependence is very simple and has clear physical meaning.

Changing of coefficient of friction along the line of action is in total coincidence with starting assumptions. The obtained results are agreeable with experimental result[6], not taking into account pitch point (point C).

The more realistic picture about the teeth pairs conjugate action can be obtained by application of the disc/disc tribometer[8]. Such a design can provide significantly higher normal load forces, due to larger disc radii, and accordingly, the higher driving engine power. The controlled motion needs to be ensured for both discs (where the percent of sliding is set in advance), etc. This kind of tribometer is used for testing the quality of lubricating oils.

The rolling friction was neglected, within this study, since it is negligible compared to sliding friction, and significantly complicates the procedure. The obtained results show good agreement with real values, based on application of the relatively simple investigation procedure at available equipment.

REFERENCES

1. B. IVKOVIC, A. RAC: Tribology. Yugoslav Society for Tribology, Kragujevac, 1995 (in Serbian).
2. S. TANASIJEVIC: Tribologically Correct Design. Faculty of Mechanical Engineering from Kragujevac, Kragujevac, 2004 (in Serbian).
3. Y. MICHLIN, V. MYUNSTER: Determination of Power Losses in Gear Transmissions with Rolling and Sliding Friction Incorporated. Mechanism and Machine Theory, **37** (2), 167 (2002).

4. Y. B. XIE: On the Tribology Design. Tribology International, **32** (7), 351 (1999).
5. N. MARJANOVIC: Optimization of Gear Trains. Monography. Faculty of Mechanical Engineering from Kragujevac, Kragujevac, Cad Lab, 2007 (in Serbian).
6. J. KLEEMOLA, A. LEHTOVAARA: Experimental Simulation of Gear Contact along the Line of Action. Tribology International, **42** (10), 1453(2009).
7. C. H. WINK, A. L. SERPA: Performance Assessment of Solution Methods for Load Distribution Problem of Gear Teeth. Mechanism and Machine Theory, **43** (1), 80(2008).
8. B. IVKOVIC, N. MARJANOVIC, B. FERNANDEZ: Ecotribology – Disk-on-disk Test of Gear Lubricants Properties. J. of the Balkan Tribological Association, **15** (3), 447 (2009).
9. Y. WANG, H. LI, J. TONG, P. YANG: Transient Thermoelastohydrodynamic Lubrication Analysis of an Involute Spur Gear. Tribology International, **37** (10), 773 (2004).
10. M. AMARNATH, C. SUJATHA, S. SWARNAMANI: Experimental Studies on the Effects of Reduction in Gear Tooth Stiffness and Lubricant Film Thickness in a Spur Geared System. Tribology International, **42** (2), 340 (2009).
11. J. CASTRO, J. SEABRA: Global and Local Analysis of Gear Scuffing Tests Using a Mixed Film Lubrication Model. Tribology International, **41** (4), 244 (2008).
12. R. B. HOEHN, K. MICHAELIS: Test Methods for Gear Lubrication in the FZG Gear Test Rig. In: Proc. Balkantrib 96, 2nd International Conference on Tribology, Thessaloniki, Greece, 1996.
13. V. ONISHCHENKO: Tooth Wear Modelling and Prognostication Parameters of Engagement of Spur Gear Power Transmission. Mechanism and Machine Theory, **43** (12), 1639 (2008).
14. T. OSMAN, Ph. VELEX: Static and Dynamic Simulations of Mild Abrasive Wear in Wide-faced Solid Spur and Helical Gears. Mechanism and Machine Theory, **45** (6), 911(2010).
15. G. LIU, R. PARKER: Impact of Tooth Friction and Its Bending Effect on Gear Dynamics. J. of Sound and Vibration, **320** (4–5), 1039 (2009).
16. B. STOJANOVIC, S. TANASIJEVIC, N. MILORADOVIC: Tribomechanical Systems in Timing Belt Drives. J. of the Balkan Tribological Association, **15** (4), 465 (2009).
17. V. NIKOLIC: Machine Elements – Theory and Examples. Faculty of Mechanical Engineering from Kragujevac, Kragujevac, 2004 (in Serbian).

Received 25 May 2010
Revised 17 July 2010