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RESEARCH

Disk on Disk Test of Gear Pair Power Losses

This paper presents the research of friction in gear pairs by using tribometer that provides disk-on-disk contact. When gear teeth mesh, there is a combination of sliding friction and rolling friction, and in that, the values of sliding and rolling velocity are changed, as well as contact pressure. For the simulation of changes of velocities disk of various diameters with controlled tangential velocities are applied, and this enables a broad range of combinations of sliding velocity and rolling velocity. High and variable values of contact pressures that appear in real gear pairs are realized by the combination of variable values of normal load and dimensions (diameter and width) of the disks. Friction coefficient is used as the indicator of friction magnitude. By knowing the changes of contact conditions and friction coefficient during the meshing of one pair of gear teeth it is possible to determine total gear pair power losses. Disk-on-disk contact provides the most realistic simulation of gear teeth meshing.

Keywords: *Gear, Friction, Disk on Disk Contact.*

1. INTRODUCTION

Friction, as the resistance that appears during relative motion of two bodies, also accompanies the operation of gear trains. These undesirable phenomena can, in the design phase, be reduced to the smallest extent by proper choice of influential variables. The friction process is usually monitored by the friction coefficient, energy loss due to friction, and heating of the transmitter.

In gear trains the friction appears in contact of the gears' teeth, in bearing, between the gear and oil, in seals, etc. The largest losses due to friction appear in contact between the gear teeth, and their magnitude can be significantly influenced by the choice of the variables that define the gear pairs. Losses in other transmitters' elements can also be decreased, but constructive variables do not significantly influence their magnitude.

The gear pairs friction is usually monitored by the average values of friction [1,2]. Through that, one obtains only approximate picture of this process, and it is hard to clearly notice the influence of individual parameters on the friction magnitude.

During the gears conjugate action, the relative motion of the gears' flanks occurs, where the sliding and rolling velocities are variable. Gear teeth load is also variable during the contact, so it is obvious that the friction force work and energy losses will also be dependent on the current position of the point of contact [4].

During gear teeth conjugate action process sliding and rolling friction occurs. Mathematical model for the gear pairs power losses calculation, when both types of friction are taken into account, is given in [1].

Experimental determination of the friction coefficient is usually performed on pin-on-disk tribometer type [3 - 7]. More realistic simulation of the gear teeth friction can be obtained using disk-on-disk tribometer type. This approach is shown in papers [8 - 11]. However, that type of simulation requests significantly higher normal forces, and therefore higher motor drive power. Besides, controlled movement of both disks (where sliding percent is set) must be provided.

Numerous factors influence gear pairs efficiency ratio. Paper [12] presents the influence of teeth profile contour, as well as lubrication, on gear transmitters' power losses.

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2. GEAR TEETH CONJUGATE ACTION REALIZATION

Power transmission with the aid of the gear pairs is realized by direct contact of the gear teeth. Each tooth of one gear is, for certain time, in contact (conjugate action) with the conjugate gear tooth, where the contact conditions are variable with time. Sliding velocity varies during the gear pair teeth conjugate action, and its value can, according to the law of conjugate action, be expressed in the following manner [4, 12]:

$$v_{kl} = \omega_1 \cdot r_{w1}\left(\sin\alpha_w - \cos\alpha_w \tan\varphi\right)\cdot\left(1+\frac{1}{u}\right). \qquad (1)$$

Equation (1) gives the dependence of the sliding velocity on angle φ, i.e. line of contact, namely on time, and the graphical representation of this relation is shown in Figure 1.

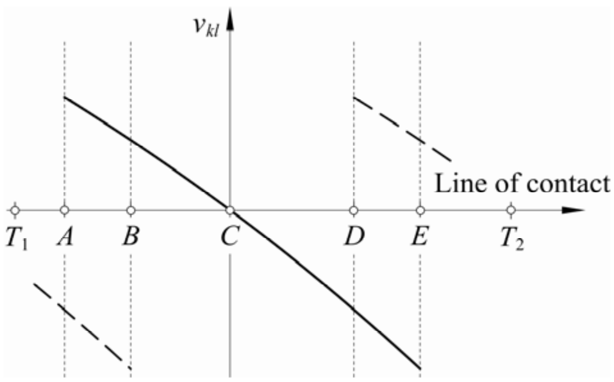

Figure 1. Sliding velocity variation

It is obvious that sliding velocity has maximum values in position of the teeth pair entering the contact (point *A*), and in position of exit of the considered teeth pair from the contact (point *E*). In pitch point *C* sliding velocity is equal to zero, and here appears pure rolling.

During the conjugate gear motion, one or two pairs of teeth are alternatively in conjugate action. During unilateral conjugate action, the total normal force F_{bn} is being transmitted by one teeth pair, while during the double conjugate action it is divided between two teeth pairs. For analysis, we shall adopt that during the double conjugate action the normal force is uniformly distributed to both teeth pairs, namely that each pair is loaded with the normal force $F_{bn}/2$.

Figure 2 presents variation of the normal force during the conjugate action. The dashed lines present the variations of the normal forces of the preceding and following gear pairs.

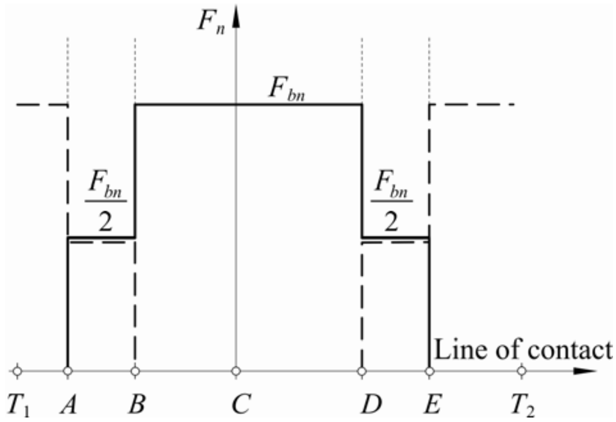

Figure 2. Variation of the normal force during the conjugate action

Value of the contact pressure can be determined according to the Hertz's formula:

$$\sigma_H = \sqrt{\frac{F_n}{b\cdot\rho}\cdot\frac{1}{\pi\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}}. \qquad (2)$$

It is obvious that the contact pressure on the teeth sides can change during the conjugate action, due to variation of the normal force value on the side (Figure 2) and variation of the equivalent radius [12]. Figure 3 presents the variation of the contact pressure on the teeth sides during the conjugate action.

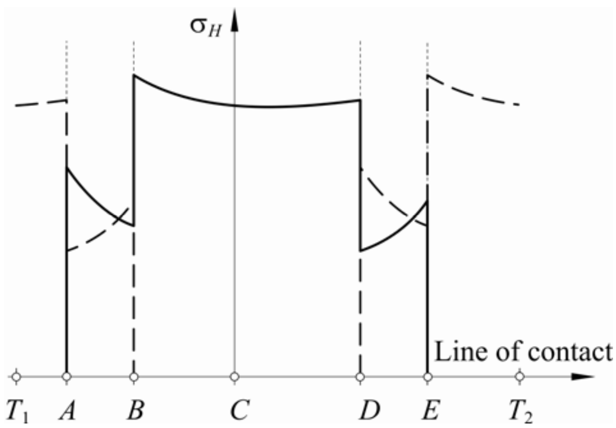

Figure 3. Variation of the contact pressure during the conjugate action

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

$$\mu = \frac{F_{tr}}{F_n}. \qquad (3)$$

During the conjugate action of gear teeth sides, the sliding velocity and the contact pressure are greatly changed, and this was analyzed earlier. The friction coefficient, which depends on these variables, will also have variable values during the conjugate action. To establish the dependence of the friction coefficient on the sliding velocity and the contact pressure, the tribometric tests were performed.

3. THE PRELIMINARY EXPERIMENTAL PROGRAM

Depending on the size of the meshing gears and loads (magnitude of the torque) the contact pressure is usually within range 500 to 1500 (3000) MPa. Considering that the contact between teeth of the two gears is realized along the line, the normal force in the contact zone, which acts perpendicularly to the teeth flanks, usually ranges between 500 and 5000 N.

During the power and motion transfer, friction appears in the contact zone of the gear teeth. The rolling friction appears only in the contact zone, which is around the base circles of the gears that are in conjugate action. In the contact zones below and above the base circles both rolling and sliding friction appear. The consequence of appearance of this friction in the contact zones below and above the base circles is the increased wear of teeth in those zones (Figure 4).

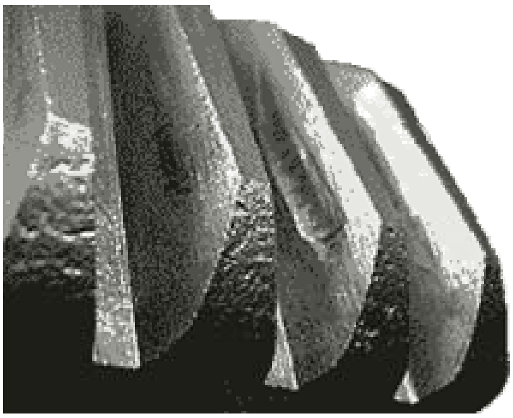

Figure 4. Worn teeth of a cylindrical gear

The preliminary experimental program in which the tribological characteristics of the two types of the gear oils were investigated was realized on the UT-07 tribometer. Its basic configuration (Disk-on-Disk) [9] (Figure 5) enables investigation in contact conditions where both the sliding and the rolling friction appear simultaneously (Figure 6).

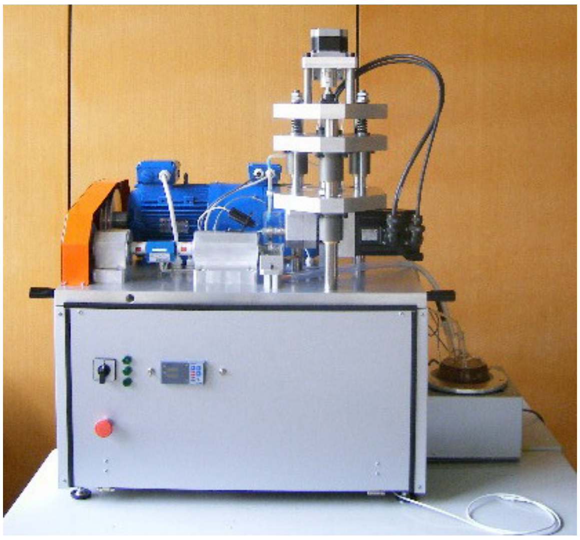

Figure 5. Tribometer UT-07

The preliminary testing was conducted in the following conditions:

Number of rpm on both disks: 500 -3000 rpm
Torque: 1 – 10 Nm
Normal force: 100 – 3000N
Disks diameters: 0 – 50 mm
Lubrication: High viscosity oil
Temperature: Room temperature

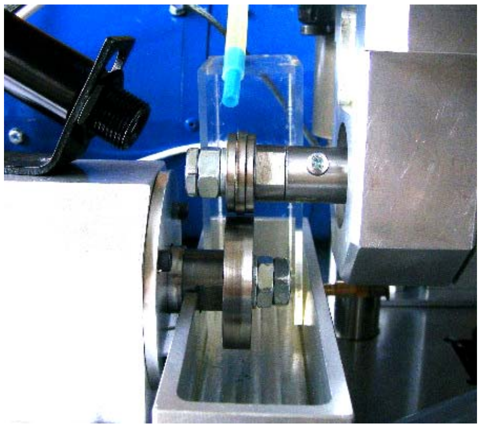

Figure 6. Disk-on-Disk contact

The numbers of rpm of both disks can be varied continuously from 0 to 4000 rpm by independent motors. The ratio of rolling to sliding friction in the contact zone can also be varied continuously from 0 to 100 depending on the selected numbers of rpm and diameters of disks. When the upper disk does not rotate (n=0) only the sliding friction appears in the contact zone. When both disks are rotating with equal numbers of rpm only the rolling friction appears in the contact zone. By selecting the disks' diameters and numbers of rpm, it is possible to realize different ratios of the rolling to sliding friction.

The selection of parameters, that define conditions under which the contact between the two disks will be realized during the experiment, is done by entering data into PC equipped with appropriate software.

The following data that are entered are about:

- Type of contact (disk on disk),
- diameters of disks, in mm,
- numbers of rpm,
- normal load in N, and
- duration of experiments in min.

During the experiment, the value of normal force is measured and adjusted, so that it is as close to the required value as possible. Friction force is determined through moment of friction on the lower disk, which is precisely measured. Friction coefficient is determined in the software as the ratio of currently measured (determined) values of normal force and coefficient force.

During the experiment, the following data are monitored on the PC screen:

- value of the normal load in N
- pressure in the contact zone in GPa
- value of the friction force in N
- value of the friction moment in Nm
- the friction coefficient
- value of wear in mm
- temperature in the contact zone
- temperature of the gear oil.

Figure 7 shows the appearance of the PC screen (Program Notebook) where the entered data and the variations of certain variables are being monitored.

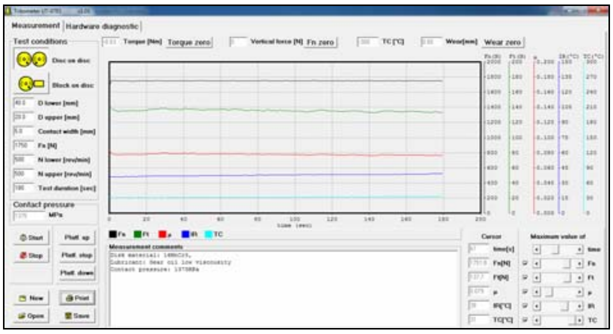

Figure 7. Notebook Program

After the finished experiment, the necessary data on experiment are entered in the "comment" part of the screen, such as:

- materials of both disks
- type of gear oil
- sum of the peripheral velocities
- sliding velocity (difference of the peripheral velocities)
- and other data relevant to the experiment.

After the completed experiment, the data on experiment are being stored in the selected directory by the "save" command and the test results are obtained in printed form by the "print" command. Printed form is shown in Figure 8.

Based on given values of disks dimensions, their number of revolutions and normal force, the software calculates the values of contact pressure, tangential velocities of upper and lower disks, as well as sliding and rolling velocities.

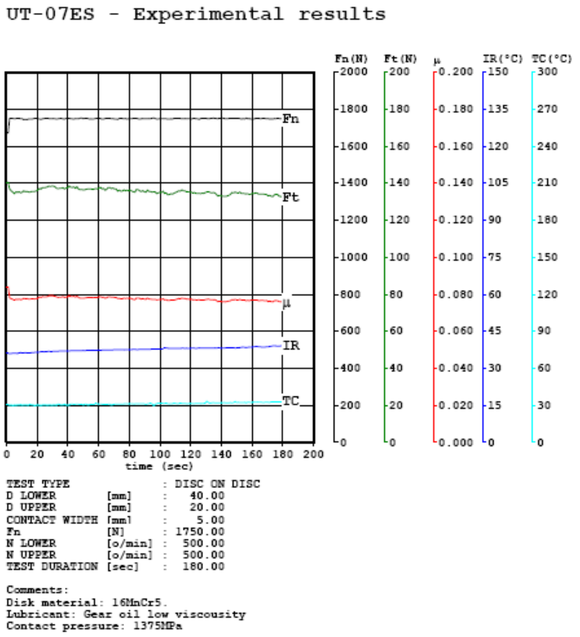

Figure 8. Example of testing results in printed form (single test)

Apart from monitoring the values of the normal load, the friction force and the friction coefficient, the contact zone temperature (IR) and the gear oil in the bath below the lower disk (TC) are also monitored during the test.

4. THE EXPERIMENTAL TESTINGS RESULTS

The preliminary tests were conducted with disks made of alloyed steel 16MnCr5, heat treated (tempered) and ground along the peripheral surface.

By the preliminary experimental program, the intention was to investigate the influence of the sliding velocity and the normal load on the friction force and coefficient.

In conducting the experimental program, the variation of the numbers of rpm of both disks was performed, in the range from 500 to 4000 rpm and variation of the normal load of the contact zone from 500 to 3000N.

The disks of diameters 40 and 20 and 40 and 10 mm were used in the majority of the tests. The realized contact pressure in the contact zone was most frequently within range 200 MPa and 1800 MPa, depending on the disks' diameters, disks' widths and normal loads.

Depending on the number of rpm and disks' diameters the sliding velocity was realized in the contact zone in the range from 0.1 to 4.5 m/s.

Figure 9 shows the influence of the sliding velocity in the contact zone on the value of the friction coefficient.

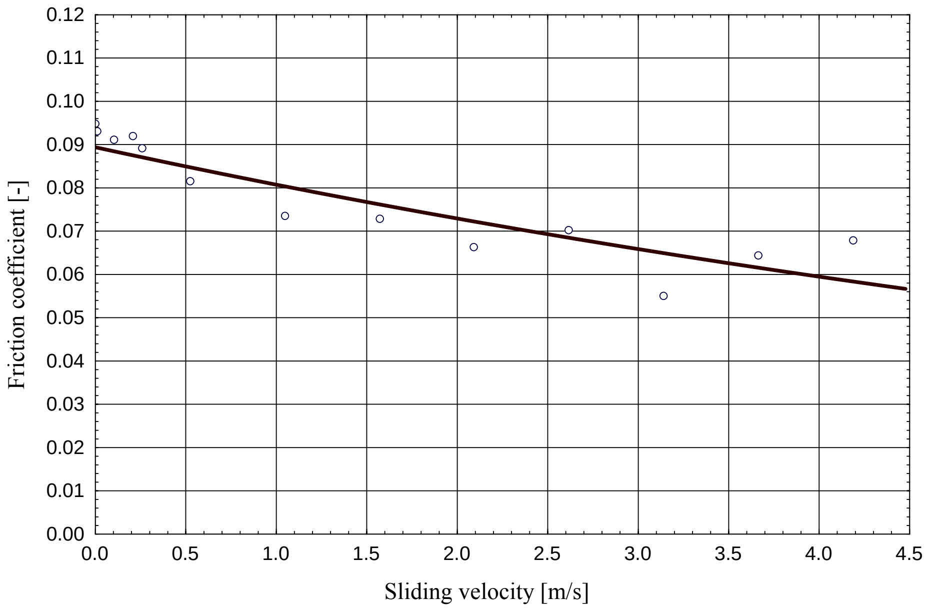

Figure 9. The friction coefficient as a function of sliding velocity, F_n=500 N.

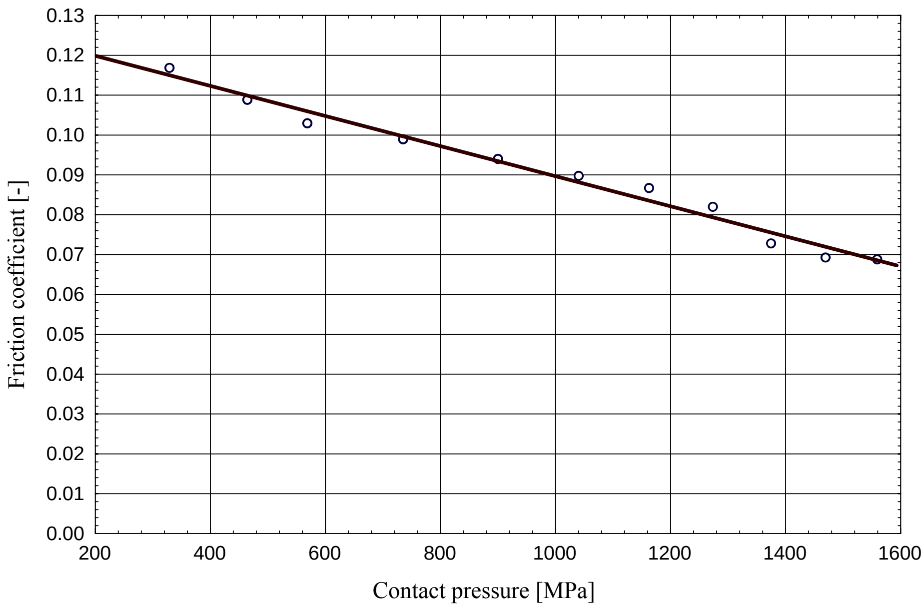

Figure 10. Influence of the value of contact pressure on the value of the friction coefficient. The number of rpm on both disks is 500 rpm.

The results of the testing shown in the previous diagram indicate that with the increase of sliding velocity (with unchanged contact pressure) decreases the value of friction coefficient.

This conclusion s in accordance with other research [12]. The values of friction coefficients, in this testing, have the values similar to the values in testing of friction by pin-on-disk contact.

Dependence of friction coefficient on the contact pressure is shown in figure 10. With the increase of pressure, there is the decrease of friction coefficient in tribomechanical system. Obtained values show almost linear dependence of these two quantities. Given diagram is obtained for one value of sliding velocity. For other values of sliding velocity, the friction coefficient also decreases with the increase of contact pressure. All tests were done for disks of 40 and 20 mm in diameter, the same contact width and the same viscosity oil.

5. CONCLUSION

This paper shows the testing of friction in gear pairs by using tribometer with disk-on-disk contact.

During the meshing of gear teeth sides, there is a combination of rolling and sliding, and in that, the rolling and sliding velocities are variable. Contact pressure is also variable during the meshing.

Disk-on-disk contact provides the simulation of the combination of rolling and sliding friction, which is much more realistic than other types of similar research (pin on disk or block on ring). Besides, this provides a very broad range of sliding velocities (0 – 4.5 m/s), as well as the values of contact pressures (even up to 1600 MPa), which is at the teeth sides endurance limit, for heat treated steel. Very stable and repeatable results were obtained during the testing.

Having the values of friction coefficients for various combinations of sliding velocities and contact pressures provides the possibility of determination of power losses during the meshing of one pair of gear teeth, and the whole gear pair, as well as the research of the influence of individual parameters of gear pairs to tribological characteristics.

The testing results show that with the increase of the sliding velocity, friction coefficient is slightly decreased. This conclusion, as well as the measured values of the friction coefficient, is in accordance with other research studies.

Second part of the research is related to the influence of contact pressure to the value of friction coefficient. The results show that with broad change of value of contact pressure (from 200 to 1600 MPa) the friction coefficient is decreased.

Further research could concentrate on the influence of rolling velocity to friction coefficient or on testing of friction with variable sliding and rolling velocities and contact pressures.

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