Friction of Polymer/Steel Gear Pairs

Róbert KERESZTES, László ZSIDAI, Gábor KALÁCSKA,
Mátyás ANDÓ and Rajmund LEFÁNTI
Department of Maintenance of Machinery, Institute for Mechanical Engineering Technology

Abstract

Engineering plastics are often applied in tribological systems, where moving parts are subjected to severe friction and wear processes. Compared to metals engineering plastics are used because of their good friction and wear properties together with increased corrosion resistance and vibration damping ability. There are many sorts of technical polymers available of which sliding elements can be produced. To choose proper polymers for a given tribological application is not a simple task owing to many different parameters influencing the performance of a polymer sliding element.

1. Introduction

In many cases investigations on rigs in laboratory have to be used to reveal the tribological properties of different polymer/metal pairs. In our institutes also many investigations on the tribological properties of technical polymer/steel pairs were performed using different tribometers. It was stated that the ranking of polymer/steel pairs alters according to the friction and wear measured on different tribometers, therefore it would be difficult to choose the best polymer/steel pair to make a polymer gear for a given application. We started to approach of gear mesh with a real gear tests. We concluded that the change of the coefficient of friction during the meshing and the misalignment of gears made difficult to evaluate the actual friction coefficient between the teeth of polymer/steel gears. This problem was solved with the following test system.

2. Formulations

The common normal to the tooth profile at the point of contact must always pass through a fixed point called the pitch point in order to maintain a constant angular velocity ratio of the two gears. The involute curve satisfied the law of gearing and is most commonly used for gear teeth profiles in the practice. Data of connecting involute profile gears:

tooth number:	$z_1 = z_2 = 12$
module:	$m = 10$ mm
pressure angle:	$\alpha = 20\ °$
face width:	$b = 5$ mm (at polymer segment gear)

We can see from the data, that the used gears are undercut. We chose these gears with large module, because we can measure clearly the changing of force through line of connection. The forces arising from sliding and rolling tooth connection. Therefore we had to modify the addendum circle diameter. The value of maximal addendum circle diameter (d_{fmax}):

$$d_{f\,max} = 2 \cdot \sqrt{(a_w \cdot \sin\alpha)^2 + \left(\frac{d_o}{2} \cdot \cos\alpha\right)^2} \quad [\text{mm}] \tag{1}$$

where:
a_w – applied center distance [mm]
d_o – pitch circle diameter [mm]
α – pressure angle [°]

The calculated engagement factor (ε) with modified addendum circle diameter:

$$\varepsilon = \frac{\overline{AE}}{p_w \cdot \cos\alpha} \tag{2}$$

where:
AE – length of connection section
(A – the first, E - the last connection point) [mm]
p_w – base pitch [mm]

$$\overline{AE} = \rho_{a1} + \rho_{a2} - a_w \cdot \sin\alpha \tag{3}$$

where:
ρ_{a1} – radius of involute curvature of drive gear at E point [mm]
ρ_{a2} – radius of involute curvature of driven gear at A point [mm]
a_w – applied center distance [mm]

$$p_w = \pi \cdot m \tag{4}$$

where:
m – module [mm]

$$\rho_a = \sqrt{{r_f}^2 - {r_a}^2} \tag{5}$$

where:
r_f – addendum radius [mm]
r_a – base radius [mm]

The allowable tangential force F_{tomax} (N) at the pitch circle of polyamide spur gear can be obtained from the Lewis formula. However, the basic equations used are applicable to all other plastic materials if the appropriate values for the factors are applied.

$$F_{to\max} = m \cdot y \cdot b \cdot \sigma_b \cdot K_v \quad [N] \tag{6}$$

where:
m – module [mm]
y – form factor at pitch point
b – face width [mm]
σ_b – allowable bending stress [N/mm^2]
K_v – speed factor

The values of different factors can be found in the tables.

The surface strength using Hertz contact stress, σ_H, is calculated by this equation

$$\sigma_H = \sqrt{\frac{F_{to}}{b \cdot d_o} \cdot \frac{i+1}{i}} \cdot \sqrt{\frac{1.4}{\left(\frac{1}{E_1} + \frac{1}{E_2}\right) \cdot \sin 2\alpha}} \quad [\text{N/mm}^2] \tag{7}$$

where:
F_{to} – tangential force on surface [N]
I – gear ratio
E_1,E_2 – modulus of elasticity of material [N/mm^2]
Α – pressure angle [°]

The arising sliding velocity between contact teeth is difference of the tangential velocities (v_t).

The sliding velocity determine the frictional heating and wear of teeth.

The v_s sliding velocity:

$$v_s = v_{t1} - v_{t2} \text{ or } v_s = s \cdot (\omega_1 + \omega_2) \tag{8}$$

where:
s – distance between pitch point and contact point on line of action [mm]
ω_1, ω_2 – angular velocity [rad/s]

Kozma earlier studied the friction phenomena between the gears and found that the forces and torque during the connection changed due to the friction. He distinguished two cases: the change of the teeth-force and torque in case of constant drive torque, and the change of teeth-force and torque in case of constant driven torque. In his theoretical studies he took constant friction between the surfaces, however we know from our previous research projects and from the literature that the friction between a polymer and steel surfaces was nearly never constant.

3. The test systems

We made a new gear connection test rig. The new test method (*Fig. 1.*) using large teeth was developed to measure the influence of friction on the tooth forces. In this method the rotation is limited; the variation of forces is measured during only one meshing cycle of a tooth pair. The three-teethed segments were made of the investigated polymers, but the mating steel gear must be prepared in full size due to balancing reasons.

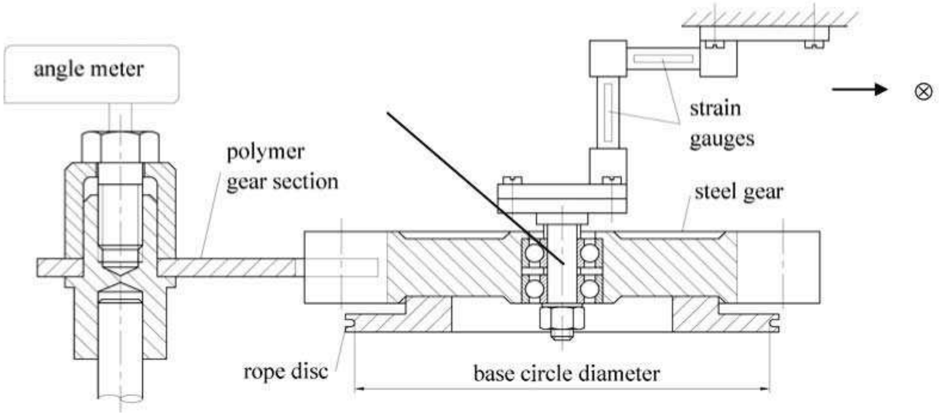

Figure 1. Drawing of test rig

3.1. Measured and calculated forces

The forces on the axe l of steel gear (F_x, F_y) were measured with strain gauges, as it is shown in *Fig. 1.* The normal force (F_N) was equal the applied load force (F_g), because each line of forces were same.

If we consider the effect of friction along the connection section, it's clear that the direction of friction force changes at the pitch point. In this article we focus on this change limiting to the one tooth connection phase.

There are two different section under one tooth connection. The pitch point (C) is in the center position. The value of normal force is equal to load force F_n=F_g [N]. There is a relationship between the friction force and the measured axle force (*Fig. 2.*). The axle force increasing until pitch point and decreasing after it. We can evaluate their relationship using the following equations:

in BD connection section:

$$F_x = F_s \cdot \cos\alpha \tag{9}$$

where:

F_x – measured axel force [N]

α – pressure angle [°]

F_s – friction force [N]

$$F_s = \frac{F_x}{\cos\alpha} \quad [N] \tag{10}$$

In this system we define a coefficient of friction between contact teeth. It is calculated by this equation.

$$\mu = \frac{F_s}{F_n} = \frac{F_x}{F_n \cdot \cos\alpha} \tag{11}$$

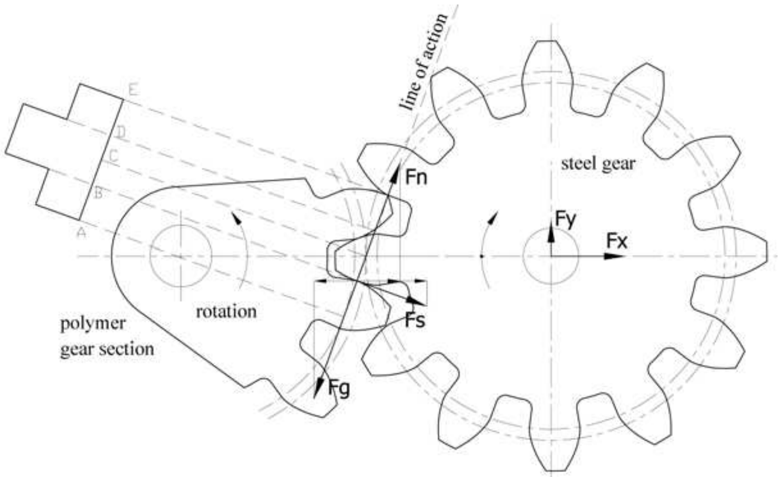

Figure 2. The arising forces at one tooth connection

At the *table 1.* we show the testing conditions.

Table 1. Parameters of the tests

Parameters	Tooth connection model tests
load, M [Nm]	1.1 / 5.5
angular velocity, ω [1/s]	0.1
test time, cycle	1 / 100 / 500 / 1000 / 2000 cycles
ambient temperature, T [°C]	24 °C
Relative humidity, RH [%]	50 %

3.2. Tested materials

The properties of the investigated polymers are presented in the *Table 2.* The gear mating with polymer gears was made of structural steel S355 with a surface finish (CLA) R_a 2,5 μm.

Table 2. Properties of investigated polymers

Polymer	Elongation at rupture A (%)	Young modulus E (MPa)	Rockwell M hardness	Tensile strength R_m (MPa)
PA 6G-Mg	40	3000	86	85
PA 6G-Na	25	3300	88	80
PA 66 GF30	7	5200	98	185
POM C	30	3000	86	70
PETP /PTFE	8	3200	94	75
Bakelite		7000	98	80

4. Results and discussion

Because of the large number of experiments and continuous monitoring of gear mesh friction many different graphs can be drawn about teeth surface processes.

As we have explained earlier in the test system the motion of the measurements were reciprocating (towards – upload, backwards –down load) between the mating teeth and from each cycle the upload process under „one-tooth" connection was grabbed to evaluate.

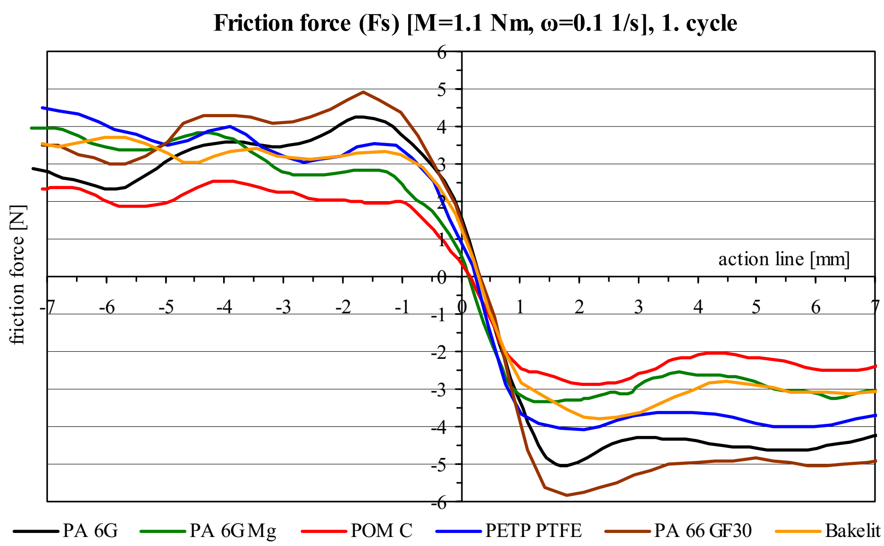

Figure 3. Friction forces during the first test cycle

Fig.3. shows a summary about the measured friction force during the first cycle of drive in case of six different polymer materials mashing with S355 structural steels. The role of pitch point is very spectacular.

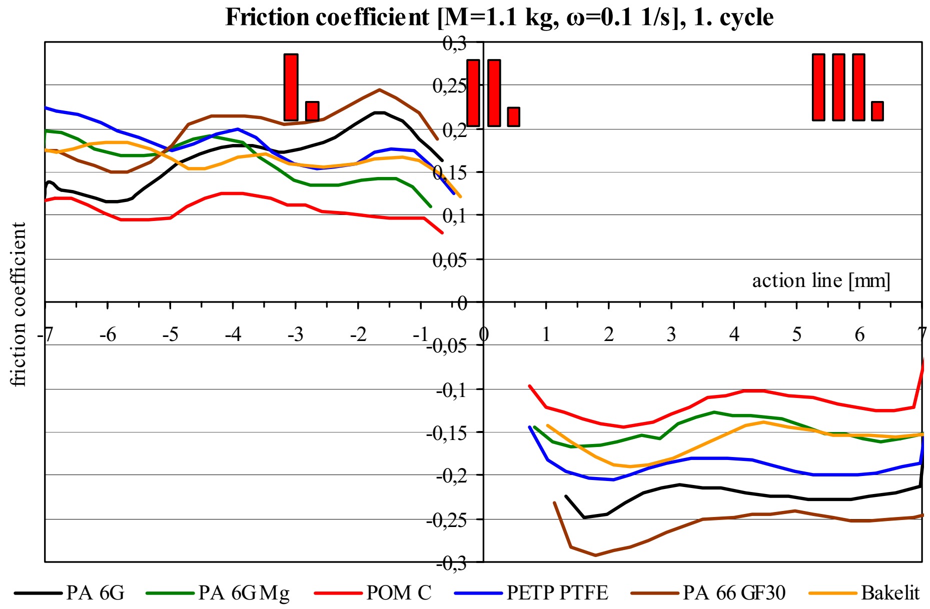

Figure 4. The calculated friction coefficients during the first test cycle

At the beginning of the measurements the POM-C performed the lowest friction and PA 66 GF30 gave relatively high friction force. If we compare the two cast polyamide versions - Na and Mg catalytic types – essential difference can be realized. The Mg catalytic cast polyamide 6 material is better performing much lower friction force.

In Fig.4. the calculated friction coefficients are shown. The main difference between the curves plotted in Fig 3. and Fig.4. is the domain. While friction forces are drawn as continuous functions along action line, curves of friction coefficients are interrupted at pitch point and its transition zone. The reason is the origin of friction movement. At pitch point and its transition zone the dominant form of friction is rolling instead of sliding. Before and after pitch point zone the friction is generated mainly from sliding, the rolling friction effect is much lower.

That means along the action line the friction form van be split to:

I. sliding zone before pitch point with decreasing sliding speed to reach pitch zone
II. pitch transition zone with mainly rolling movement
III. sliding zone after pitch point with increasing sliding speed leaving pitch zone

For further evaluation of gear mesh friction coefficient curves we introduce more values according to Fig. 5.

As an example of the evaluations Fig.6. shows the change with cycle numbers of local maximum friction coefficient values for the different polymer materials.

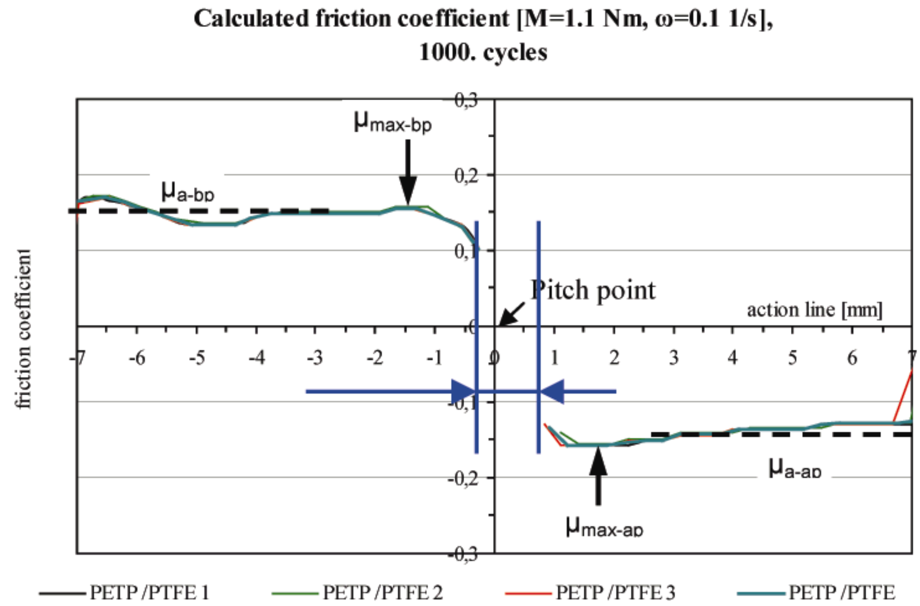

Figure 5. The calculated friction coefficients during the first test cycle
($\mu_{a\text{-}bp}$: average friction coefficient before pitch zone,
$\mu_{max\text{-}bp}$: local maximum of friction before pitch zone,
$\mu_{max\text{-}ap}$: local maximum of friction after pitch zone,
$\mu_{a\text{-}ap}$: average friction coefficient after pitch zone)

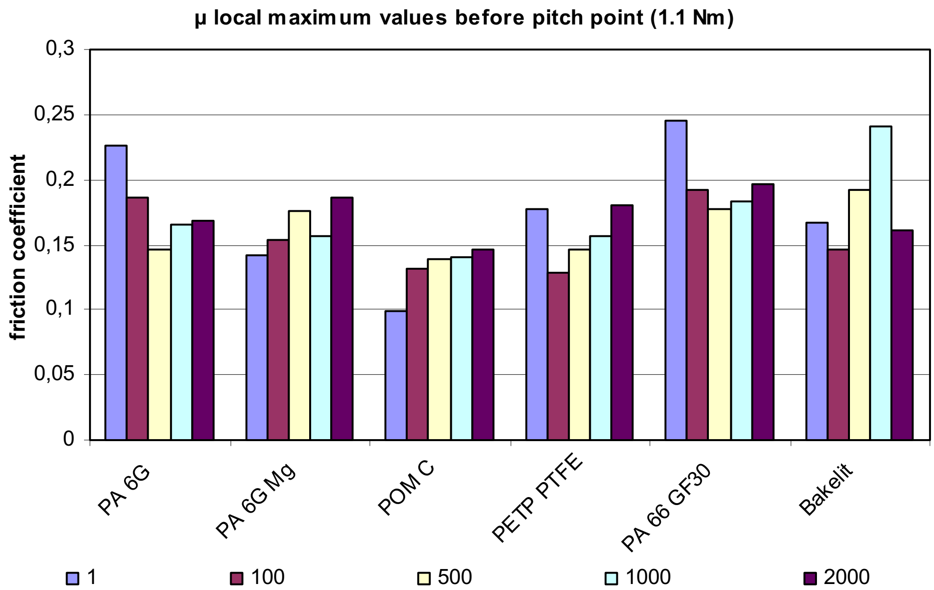

Figure 6. Local maximum values of friction before pitch zone
under different cycle numbers

If we follow the different behaviour of friction values in the function running distance (number test cycles) we can distinguish four different groups.

1. Materials performing basin-like curves or columns (PA 6G, PETP/PTFE, PA 66 GF30, in Fig 6.): with increasing meshing time the friction first decreasing but after a certain running cycle it starts to elevate.
2. Material with continuously increasing friction with increasing meshing time (POM C in Fig 6.): according to the test condition (load and speed) the function of friction increasing with different slopes during the gear mesh.
3. Material with low difference increasing tendency (PA 6G Mg)
4. Stochastic-like friction results with large differences in friction coefficients (bakelite)

Regarding the absolute values of friction coefficient in *Fig 9.* we can find POM C and PETP/PTFE the best ones. Comparing the two cast polyamide types we can state that the Na catalytic cast Pa (PA 6 G) performed higher friction than PA 6 G Mg.

5. Conclusions

- The developed test method is useful to study the friction process along the action line under „one-tooth” connection phase between polymer/steel gears.
- The theoretical friction can be discovered more in depth and described with a given friction pairs.
- The friction changes along the action line.
- The trends also change in the function of load and meshing time (test cycles)
- Different changing trends of friction means different change of efficiency of polymer/gear drive. In our database we set these trends.
- Pitch point rolling effect in the practice means pitch zone. The width of the pitch zone is different with different friction pairs due to the different adhesion and deformation.
- The comparison of local maximum friction and average sliding friction values gives information about „even” or „un-even” running of mesh, the sensitivity for „stick-slip” behaviour of gear drive. Where we find local maximum friction to be much higher to average friction values, that means potential „stick-slip” danger. That is typical for PA 6G under light load.
- Based on our new research method we discovered more material- and system-specific phenomena with polymer/steel gear pair friction and our new database can help to design and maintain such a kind of gear drives.

Acknowledgements

The research was supported by OTKA T42511, OTKA T37244, OTKA NI 62729 and TéT B-1/04 research funds.

References

Antal – Fledrich – Kalácska – Kozma: Műszaki műanyagok gépészeti alapjai, Műszaki műanyagok gépészeti alapjai, Minerva-Sop Bt. Sopron, 1997

Benedict, G.H., and Kelley, B.W.: Instantaneous Coefficients of Gear Tooth Friction. ASLE Transactions, Vol. 4, No. 1, 1961. p. 59–70

Brian Rebbechi, Fred B Oswald, Dennis P. Townsend: Measurement of Gear Tooth Dynamic Friction. NASA Technical memorandum 107279, Army Research Laboratory, 1996

Kozma Mihály: A fogaskerekek súrlódási vesztesége. Gép okt. – nov., 2004

www.quattroplast.hu

Yamaguchi Yukisaburo: Tribology of plastic materials. Amsterdam: Tribology series 16,