Tribological analysis of engineering plastics/steel friction pairs

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Abstract

This study involves the case of polymer-metal contact, where the deformation behavior of polymer gear and the time dependent strain behavior play a major role. In present work, a section of polymer gear is mated against steel gear and the friction behaviors for different groups of polymer materials of PA 6G, PA 6G Mg, POM-C, PETP/PTFE, PA 66 GF 30 and Bakelite were experimentally studied. From the results obtained, all the materials exhibited a similar tendency of friction characteristics, with coefficient of friction decreased significantly at the pitch point. This was attributed to the pure rolling at the pitch point and tangential traction, which was the

dominant mechanism for resistance to motion. An unstable friction characteristic from the gear flank tooth to the pitch point was evident in the low load conditions.

Keywords: Spur gear, Polymer, Friction, Rolling contact.

1. Introduction

The application of polymer materials and gears in the modern industrial era is increasing and the development of new materials is also attracting a lot of studies to explore many different kinds of material. The use of gears is drastically increasing in the recent technologies. Gear may be of small moving component made of plastics or a giant moving metallic structure. The failure of these moving components is mainly due to friction and wear. Well established gear design procedure is available to make a gear suitable for particular loading conditions, but the understanding of the tribological properties of the gears and their applications in different loading conditions are not yet widely explored. Also, there are a lot of design data and procedures available for metal gears. But, there is a lack of deeper understanding in the area of polymer/steel and polymer/polymer gear combination. The scope for the improvement in terms of design or testing the new kind of polymer materials towards industrial application is not fully available. The advantage of polymer gears over metal is increasing, due to their lightweight and low cost. In addition, the reduction of external lubrication and thermal effect on gears also makes the polymer gear one of the vital components of many engineering systems today.

Furthermore, the need for the compiled behavioral data of gear with different combinations of materials is also the current need for industrial applications. A lot of studies were conducted on wear of various polymer and steel pairs, but majority of these works were concentrated on sliding friction. The tribological research on modern self-lubricating materials is generally performed by means of small-scale specimens, because of the lower costs and the flexibility of testing. The results of these small-scale tests, however, cannot easily be extrapolated to actual industrial applications. Some of the reasons include misalignment, edge effects, material inhomogeneity and wear particle grooves. Still, some of the researchers have attempted to study the frictional behavior of polymer/steel pairs and found a lot of interesting facts.

In addition, the thermoset polymer gears were developed and tested in the last centuries and they were in applications, such as speedometer, wipers, projectors and some automobile components. The development of thermoplastic polymer gears, using injection molding started towards the end of the last century. The study of Targett and Nightingale [1] explained the method of manufacturing the gears, using the injection molding technique with simple and quick-made cavities. The creep and fatigue characterization of various new tooth forms were presented. Tsukamoto et al. [2] initiated the investigation on polymer gears for power transmission applications. This study compared the steel and nylon plastic gear at high loading condition. The test rig developed for the frictional analysis on the gear was also important to continuously monitor the operation of the gear. The design and development of rig for the frictional study of Acetal/steel paired gears was made by Hooke et al. [3], in-situ measurement of wear using the rig showed that there was a sudden increase of wear rate, as the transmitted torque was increased above a critical value. Duzcukoglu [4] reported an experimental analysis of polymer gear sliding against the steel gear. Modification of the polymer gear tooth was made to reduce the thermal damage of the gear at high loading conditions. The cooling holes delayed the gear failure at a high temperature, but breakage occurred due to the stress concentration around the holes.

Moving forward, Gurunathan et al. [5] investigated into the wear characteristic of polyamide 6 and its nanocomposite, using a power absorption type gear test rig. The polymer was chosen as a drive gear against the driven steel gear at different load conditions. The composite gear performed better, due to its high strength. The coating of polymer material using molybdenum disulfide (MoS_2), graphite, polytetrafluoroethylene (PTFE) and other filler materials to reduce friction has been followed for a long time. The low cost of external coating supported the work effectively at frictional places. The coating type or material improved the polymer mechanical and tribological characteristics. Consequently, some researchers also tried the same kind of method on a gear testing. Dearn et al. [6] made an attempt to reduce the friction and wear of polymer gear, using solid lubricants and observed that the PTFE coated polymer gear showed 90% reduction in wear over uncoated gear. Using dry lubricant in polyamide and poly-ether-ether-ketone (PEEK) significantly increased the working life of the gears. Also, wear and acoustic noise generation when running another polyoxymethylene (POM) polymer were also tested against steel. The sound level of POM was inversely proportional to the load, while the other materials such as PEEK and polyamides (PA) showed a proportional increase in sound against load and speed [7].

Therefore, considering the insufficient available comparative studies on wear and friction of various polymer gears with metal or alloys as evident from the aforementioned reports among others, this work focused on frictional and wear behaviors of engineering polymer/steel; mainly on various plastics/steel gear pairs. The current tribological study adopted the Czichos approach to address the problem. The main focus of this study was on present role of polymer/steel gear drive at unlubricated conditions. The test results obtained from the tribological study allowed comparison of different test systems and clarify the correlation between friction results of small-scale, dynamic, large-scale tests as well as real gear friction results.

2. Materials and method

Table 1 represents the list of tested polymer gear materials. Semi-finished engineering plastic products were used in all cases, as a starting base material. The test samples were produced for the small and large-scale samples as well as the gear samples by machining processes. The following conditions were chosen to conduct the frictional study:

1. Initially, the small-scale test was conducted for all the materials, using conventional pin-on-disk (POD) tribometer to obtain friction and wear data. The dynamic modelling and testing were also performed for the same materials.
2. The large-scale tribological testing was conducted with dynamic condition for the selected material gear pairs.
3. A new method was employed to measure the friction during gear mesh along the line of action of involute gear pairs. The friction value was obtained along the line of action during the gear drive running, as a function of sliding distance.

Small-scale tribotests were carried out on engineering plastic samples, using POD on static, dynamic conditions and real gear teeth connection. The main purpose of the tests was to understand the basic friction and wear behaviors of different material pairs, under various conditions and to study the role of dynamic effects in various load and speed conditions.

Similarly, large-scale tribotests were carried out at Laboratory Soete, University Gent, Belgium. The role of the test measurements was to study the friction and wear processes in the function of strongly differing dimensions of the tested samples.

2.1 Measurements with small-scale specimens

The main objective of the measurement was to determine the sliding friction characteristics of the selected polymer samples on a ground steel surface. Initially, the laboratory model tests in the traditional POD systems were performed. On the basis of the unidirectional sliding friction and the surface load arising from the contact along the surface, the friction and wear characteristics of the polymers were properly ranked and the effects of additives were well recognized. In practice, majority of parts are subjected to dynamic effects. For testing the effects of dynamic stress on friction, a dynamic motion path was programmed on which the plastic sample was supposed to travel at a variable speed and load. Thus, the effects of static and dynamic stress were analyzed in the friction and wear behaviors of the polymers.

The static and dynamic testing accommodated a sample size (diameter) of 6 mm and length of 15 mm. Table 2 represents the different testing conditions deployed for the small-scale test. The counterface material was S355 structural steel, with an elasticity modulus of 210 GPa. The diameter and thickness of the steel disk were 350 mm and 13 mm, respectively.

2.2 Measurements with large-scale specimens

From the perceptive of study on tribology, the sizes of the contact zone and the test specimen are of great importance. Earlier experiments have proven that the heat conduction, deformation and inhomogeneous stress distribution in natural and composite polymers can greatly influence friction and wear. Therefore, an important supplementary part of this work was planned on a large-scale research system, including dynamic effects and evaluation of measurement results. The measurements were obtained by using special instrument developed in the Laboratory Soete at University Gent, Belgium.

The steel counter surface (block) material in a friction contact with the engineering plastic was a steel alloy of 40CrMnNiMo8, frequently used in European engineering practice. Its overall (inclusive) dimensions were 90 x 106 x 420 mm. In the small-scale sample testing systems, the contact zone has a size of 28.2 mm^2, while the measurement system of large-scale samples was 24000 mm^2. This represented an increase by a factor of 851, nearly three orders of magnitude.

Moreover, the measurements were performed along with alternating motion, at several different load levels. Each measurement was performed with dual repetition, that is, three-fold data recording. Fig. 1 shows the average values of measurement series. By changing the sliding directions and acceleration, the static and dynamic friction values were generated. During gear mesh, the contacting teeth surfaces were subjected to a complex tribological effect. This was due to rolling and sliding phenomena with changing loads. In comparison with the stress levels applied at the polymer machine elements, the difference between static and dynamic friction could be accurately established, due to large surface loads. Accuracy of the data can be compared with the phenomenon observed at the surface of gear teeth. In the vicinity of the pitch point, the transitory rolling zone and the start/end of sliding created on the surfaces performed similar phenomenon.

Importantly, the friction measured on large-scale test equipment was compared with the POD measurements performed on small-scale samples, at constant speeds.

2.3 Developed new test equipment

The modified instrument developed from the small-scale testing model was used, as shown in Figs. 2 and 3. The motor rotated the polymer gear segment via a worm-gear drive. This segment was contacted by the steel gear, which has bearings on the axis knuckle of the holder head equipped with strain gauges. The constant weight load exerted its effect on the action line of gears via the rope pulley, corresponding to the base circle diameter of the wheels. Regarding the plastic gear segment, the angular positions of the start and end points of the tooth contact along the action line were accurately determined by calculations. These positions were indicated by two micro-switches. The motor rotated the segment between two terminal positions by alternating rotating directions. The angular signal transmitter was located on the axis of the polymer gear segment. The type used was HEDS-5701 G00 incremental signal transmitter, with an accuracy of 0.25°. The test conditions were determined according to the polymer gear design methods. The geometrical data of the tested gears are presented in Table 3.

From the gear data, it was obvious that the wheels were undercut. For the purpose of testing, large module gears were chosen, because they needed well-defined and unambiguous friction force changes, arising from sliding and rolling in the course of tooth contact. However, for the purpose of correct contact, outer diameter of the gears has to be modified. The material of the mating structural steel gear was S355 type.

The steel gear was manufactured by wire-spark erosion, the manufacturing accuracy was ±0.01 mm. Three tooth polymer gear segment with thickness of 5 mm was prepared, using a CNC milling center. The tested polymers were identical to the materials described at the small-scale polymer test specimen trials (Table 4).

2.4 Teeth frictional force and coefficient

During the course of measurements, the value of the force, F_y in accordance with a conscious planning of the measurement system ended up being approximately zero. Thus, it can be neglected in case of further calculation. The calculation of the friction force values in the single tooth contact section is described in the following section.

According to Fig. 4, Eq. (1) can be specified for the balance of the following forces:

$$F_x = F_s \cdot \cos\alpha \tag{1}$$

Friction force can be calculated, using Eq. (2),

$$F_s = \frac{F_x}{\cos\alpha} \quad [N] \tag{2}$$

Where: F_x represented measured shaft force [N] and α denoted connecting angle [°].

In the tooth contact testing system, static and dynamic friction coefficients between the contacting tooth according to the motion characteristics was defined and it was determined as a quotient of the friction force and the normal force. Directly before and after the pitch point, where the sliding speed was near zero as well as in the pitch point, where the teeth roll on one another, the friction coefficient originating from sliding in a classical sense cannot be defined, due to the elastic deformation of the polymer tooth.

The calculated friction coefficients were obtained from Eqs. (3) and (4):

$$\mu = \frac{F_s}{F_n} \tag{3}$$

$$\mu = \frac{F_x}{F_n \cdot \cos\alpha} \tag{4}$$

Fig. 5 depicts the change of the friction coefficient along the action line, determined by calculations on the basis of measurements. In the pitch point, contacting tooth rolled on one another, thus only rolling friction occurred in this point. Due to deformation and elasticity of polymers, in the direct vicinity of the pitch point where sliding was near zero, contacting surfaces might experience a mutual adhesion. Based on this phenomenon, displacement was determined by elastic deformation within the material. The interruption location was 45^o contact point of the curves. Thereby, it separated the measurement results to two plus one transitory sections (according to Fig. 5, Sections I, III and the transitory Section II). In the section prior to the pitch point, the sliding value continuously approximated zero, whereas, it continuously increased from zero after the pitch point. This change has an effect on the value of the friction coefficient.

Also, the notation of the local maximum friction coefficient measured in a single tooth-pair contact section prior to changing of directions (concerning the friction coefficient related to rolling through the pitch point) was $\mu_{fe\text{-}max}$. After frictional direction change, a local $\mu_{fu\text{-}max}$ was measured in terms of an absolute value, which has a negative sign due to directional change at the pitch point, according to the scaling on Fig. 5.

Prior to and after rolling through the pitch point, average friction coefficients at the dynamic sliding friction section was defined, with notation of $\mu_{fe\text{-}átl}$, $\mu_{fu\text{-}átl}$. In summary, $\mu_{fe\text{-}max}$ and $\mu_{fe\text{-}avr}$ represented the local maximum and average values of friction coefficients before reaching pitch point, respectively. While, $\mu_{fu\text{-}max}$ and $\mu_{fu\text{-}avr}$ denoted the local maximum and average values of friction coefficient after leaving pitch point, respectively.

3. Results and discussion

*3.1 **Transmitted torque of** 1.1 Nm and angular velocity of 0.11 /s*

Fig. 6 depicts the first tested cycle, the values of the friction coefficient occurred along the action line between the tested engineering polymer gear tooth and the steel gear tooth, whereas

Fig. 7 depicts the friction coefficient values measured at the 500 mating cycles. The load torque, M and the angular velocity, ω of the rotation were identical in all cases.

The theoretical straight lines of the reported tooth friction referred to more complex surface processes, which can be observed when the results of Fig. 6 were examined. In case of tooth pairs given according to the same Fig., along with a constant load torque, the friction coefficient along with the action line was not constant. This phenomenon was more apparent in the initial stage of the friction (running-in), where significant differences occurred between individual material pairs.

In the event around the vicinity of the pitch point, the local maximum of friction was clearly increased in comparison with the friction values of the sliding sections, the stick-slip characteristics of the process can be recognized. If this local maximum – the friction related to the smallest sliding speeds was only slightly or not different from the friction values of the sliding phases, the stick-slip tendency was also small, thus the noise level arising from the run of the gear pair was also lower. This measurement result was in accordance with subjective observations.

Fig. 7 shows the measured results during the 500 cycles. It was observed that the PA 6G *transitory phase* was the greatest around the pitch point. This cannot be clearly described adhesion phenomena, because PA 6G did not possess the greatest adhesion tendency among the tested materials, but not even deformations alone could result in this phenomenon, since PA 6G has an average elasticity module. However, the most likely scenario was an overall effect, where adhesion, the formation of a transfer film, material elasticity and surface geometry together resulted in a relatively noisy run prone to stick-slip effects. The friction of the textile Bakelite started to become unstable. The friction order before and after the pitch point were not identical, the curves crossed over one another, showing slightly dissimilar trend.

After the pitch point, PA 6G and PA 66 GF30 exhibited a definitely larger degree of friction, when compared with the other polymers, with exception of the textile Bakelite. The difference between POM C and PETP/PTFE was not significant. PETP/PTFE increasingly approximated to the favorable friction characteristics of POM C and the addition of PTFE increasingly exerted its positive effects.

From Table 5, it can be evidently observed that the friction order has fundamentally changed in comparison with the starting phase of running-in. The separable trends can be formulated for friction processes before and after the pitch point.

3.2 Transmitted torque of 5.5 Nm and angular velocity of 0.11 /s

When the load moment was increased in five-fold in comparison with the previous testing system, various comparisons were drawn. Firstly, comparison was possible with the friction results and trends of the lower load level. On the other hand, the behavior of individual polymers can be mutually compared. Therefore, Fig. 8 depicts the friction results of the first running-in cycle in case of a load torque of 5.5 Nm.

When compared with the results of the lower load level, the curves exhibited a different trend. The PA 6G Mg responded to the increased surface pressure with a greater degree of surface deformation and adhesion, with a significant high friction value. The rolling-adhesion zone around the pitch point was significantly large, when compared with other plastics. Also, after the pitch point, the start of sliding was clearly moved to the pitch point, which can be explained by a greater degree of deformation arising from a larger load. The positions of local maxima after the pitch point exhibited a spread. The greater adhesion tendency of polyamides was clearly present in comparison with POM C and PETP/PTFE.

Additionally, the textile Bakelite/steel gear wheel pair resulted in an unexpected low friction values. The friction of POM C has deteriorated in comparison with results obtained at smaller load levels, whereas an improvement was detected in case of PETP/PTFE. The interval of friction coefficients was fundamentally identical at the two load levels. But, clear rank orders cannot be established at higher load levels. This was possible at lower load levels, because the friction curves of individual materials were not identical in their slopes or characteristics, as some curves intersected one another.

More also, upon reaching the 500 cycles (Fig. 9), a clear trend of friction was observed, with no fundamental difference occurred before and after the pitch point. The absolute values of friction were only different. The PETP/PTFE, PA 66 GF 30, POM C, PA 6G Mg, PA 6G and textile Bakelite orders were clear. It was observed that the textile Bakelite violated all known characteristics and trends, especially its friction after the pitch point and hence it exhibited a highly intensive adhesion tendency. Comparatively, the PETP/PTFE composite recorded the best properties. Its absolute value of the friction coefficient was further decreased in comparison with the 100 cycles, the PTFE additive exerted relevant effect in the course of sliding. With polyamides, it was significantly observed that the friction of the glass fiber filled PA 66 was smaller than the natural cast polyamide 6 versions. However, there was an insignificant difference occurred with both POM C and PA 6G Mg. Although, the local maximum after the pitch point was reduced in case of PA 6G Mg, when compared with that of 100 cycles.

On the other hand, the Na-catalyzed PA 6G exhibited a much more unfavorable friction characteristic than Mg-catalyzed PA 6G Mg. This can be related to an earlier phenomenon concerning PA 6G Mg, the transfer film occurred earlier and it was thicker on the metal surface, which played an important role in retaining a dynamic friction balance.

From Table 6, it was observed that the friction order has changed comparing to the lower load level, different trends can be defined before and after the pitch point.

3.3 Comparison of the different test systems

Studied friction characteristics of material pairs by experiments, with small-scale samples under static circumstances using *pin-on-disc* and dynamic *pin-on-plate* model testing systems as well as with large-scale samples, in dynamic (plate-on-plate) systems require elucidation. In the definition domains of testing system, relative friction orders with respect to the running-in and the steady-state were established. Also, comparison of the results of static and dynamic systems with the results obtained in the tooth contact testing system and established partial, material-dependent, limited correlations between individual testing systems were essential.

Therefore, Tables 7 and 8 present a brief summary of the systems and correlations, regardless of the system-dependent absolute values of friction coefficients. In the definition domains of testing systems, relative teeth friction ranking can be described for running-in state based on the (a) dynamic pin-on-plate test method in relation of POM C and PA 6G, PA 6G Mg mating with S355 steel as well as in relation of PETP/PTFE and PA 66 GF30 mating with S355 steel and (b) large-scale block-on-plate test method in relation of POM C and PA 6G Mg mating with S355 steel as well as in relation of PETP/PTFE and PA 6G Mg and PA 66 GF30 mating with the used steels.

Similarly, in the definition domains of testing systems, relative teeth friction ranking can be described for steady-state based on the (c) pin-on-disc test method in relation of Bakelite and the tested thermoplastics mating with S355 steel, PA 66 GF 30, PA 6G and PA 6G Mg mating with S355 steel as well as in relation of PETP/PTFE and the other tested materials mating with S355 steel and (d) large-scale block-on-plate test method in relation of all the tested polymers of PETP/PTFE, POM C, PA 66 GF 30, PA 6G and PA 6G Mg mating with the used structural steels.

4. Conclusions

The friction force was measured between the single teeth pair contact section of involute gear tooth profile. In the line of action, the friction force changed with drive given to the gear and the transition range was visible in the vicinity of the pitch. The friction coefficient between polymer/steel tooth pairs was not constant in case of constant load torque and number of gear rotation, along the action line. When running, a friction reduction of 5-10% occurred in the steady-state, irrespective of the load, in case of PETP/PTFE and steel gear pairing, which can be attributed to an increase in the driving efficiency. At minimum load condition, the POM C friction was stabilized and prior to the pitch point, the friction coefficient slightly increased.

Increasing load showed an unfavorable result. The friction coefficient of PA 66 GF30 exceeded that of natural cast polyamide 6 types of material by 12% in the running-in phase. At a greater load level, the friction loss of natural cast polyamide materials increased by 50% on average. The friction of PA 66 GF 30 stabilized at a 10% lower level, when comparison within the running-in. The friction coefficient of the textile Bakelite was decreased when use, irrespective of applied load. In a system without lubrication, the friction characteristics are fundamentally different from thermoplastic engineering materials, sometimes caused friction imbalance, as formed in case of textile Bakelite.

Summarily, engineering optimal tribological application of the different studied polymer gears should depend on their various responses to wear, especially when mating with structural steel S355.

References

[1] J. Targett, E.J. Nightingale, Thermoplastic gears. Archive: Proceedings of The Institution of Mechanical Engineers, Conference Proceedings, 178-184 (1969) 1964-1970.

[2] N. Tsukamoto, K. Terashima, Development of plastic gears for power transmission: Various methods of lengthening the life of plastic gears and their effect, Bull. Japan Soc. Mech. Eng. 29 (247) (1986) 249–255.

[3] C.J. Hooke, K. Mao, D. Walton, A.R. Breeds, S.N. Kukureka, Measurement and prediction of the surface temperature in polymer gears and its relationship to gear wear, J. Tribol. Trans. ASME. 115 (1993) 119–124.

[4] H. Düzcükoğlu, Study on development of polyamide gears for improvement of load-carrying capacity, Tribol. Int. 42 (8) (2009) 1146–1153.

[5] C. Gurunathan, S. Kirupasankar, R. Gnanamoorthy, Wear characteristics of polyamide nanocomposite spur gears. Proc. Inst. Mech. Eng. Part J: J. Eng. Manuf. 225 (2011) 299–306.

[6] K.D. Dearn, T.J. Hoskins, D.G. Petrov, S.C. Reynolds, R. Banks, Applications of dry film lubricants for polymer gears, Wear 298–299 (2013) 99–108.

[7] T.J. Hoskins, K.D. Dearn, S.N. Kukureka, et al. Acoustic noise from polymer gears – A tribological investigation. Mater. Des. 32 (2011) 3509–3515.