Different teeth profile shapes of polymer gears and comparison of their performance

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Abstract
This article presents a lifespan testing analysis of polymer gears manufactured by cutting. Compared to injection molding, machine cutting provides higher accuracy of gear geometry. Two different tooth flank geometries were tested; i.e. involute and $S$-gears. In theory, $S$-gears have several advantages over involute gears due to the convex/concave contact between the matching flanks. The theoretical tooth flank geometry of $S$-gears provides more rolling and less sliding between the matching flanks, compared to involute gears. The convex/concave contact leads to lower contact stress, which in combination with less sliding means lower losses due to sliding friction and consequently less heat generated. The goal of our research was to prove that tooth flank geometry affects the lifetime of polymer gears, and to find the mechanisms and quantitative differences in the performance of both analyzed geometries. The gears were tested on specially designed testing equipment, which allows exact adjustment of the central axis distance. Two different material pairs (POM/POM and POM/PA66) of the drive and driven gears were tested. Each test was done at a constant moment load and a constant rotational speed. Several tests were conducted using the same conditions due to repeatability analysis. All the tests were performed till the failure of the gear pair and without lubrication. In lifespan testing, the polymer $S$-gears showed better performance and longer lifespan than involute polymer gears.

Keywords: Polymer gears, Lifespan testing, Temperature, $S$-gears, Involute

1. Introduction

The involute shape of the tooth flank was suggested by Leonard Euler (1775). Due to its unique properties, the involute shape of the tooth flank prevailed and most of the gears nowadays are involute. The condition that a tooth flank shape should fulfil in order to provide a constant gear ratio is defined by the law of gearing. Many curves, not just the involute one, fulfil it, which has yielded many other tooth flank shapes that are rarely used in practice. The purpose of developing new gear shapes is to improve the disadvantages of involute gears, such as: the convex/convex contact, teeth undercutting, and gears with a small number of teeth have a shorter dedendum, which leads to increased sliding and losses due to friction. Besides involute gearing, cycloidal gearing is the next best known type, mainly used in wrist watch mechanisms. Kapelevich (2013) dedicated much of his research work to gears with asymmetric tooth flanks, where the active tooth flank is still of involute shape. Mohan and Senthivelan (2014) studied the lifespan of asymmetric composite gears. Kim (2006) and Düzcükoğlu (2009) also worked on improving the performance of polymer gears. They used the involute tooth profile and attempted to extend the lifespan of gears by virtue of geometric changes in teeth width. Koide et al. (2017) investigated the impact of the sine-curve gear on the tooth surface temperature and gear power transmission efficiency. Hlebanja et al. (2008) suggested a shape of tooth flank that follows the path of contact in the shape of the letter S and which has several advantages, compared to involute gears. The main advantages include the convex/concave contact, which results in lower contact pressure, higher relative speed, less
sliding at the contact, and thus less loss due to friction. With an identical module, the tooth roots of S-gears are wider, compared to the involute ones, which increases root strength, while at the same time allowing manufacturing gears with a much smaller number of teeth, i.e. a minimum of four teeth. Hlebanja et al. (2001, 2008) verified the advantages of S-gears over the involute ones in several research projects. They compared the lifespan of steel involute and S-gears, while observing the mechanisms of gear defects. Kulovec and Duhovnik (2013) analyzed the effect of specific geometric parameters on the shape of the S-gear tooth flank. Duhovnik et al. (2016) tested the lifespan of injection molded polymer S-gears and compared the test results with polymer involute gears. By means of numerical simulations, their work proved that at an identical moment load, less contact stress will appear on the S-gear flanks, compared to the involute ones. They found that S-gears are sensitive to the quality of manufacturing, which – in the case of injection molding – requires a well-managed technological process.

Since experimental analysis of tooth flank geometry requires very precisely manufactured gears, the polymer gears, used in our research, were manufactured by cutting. This way we can obtain a higher quality grade of the gear geometry than by injection molding. A key advantage of polymer gears over the metal ones is cheap mass production by injection molding. In the case of gear manufacturing by cutting, this advantage no longer exists. The use of injection molding also allows simple use of special gearing shapes. In the case of steel gears, the standardized manufacturing process and manufacturing tools for involute gears are so much cheaper that it often makes the use of special gearing shapes economically unjustified. For the purpose of analyzing different gearing profiles, the use of polymer material requires manufacturing by cutting, as it provides the required precision of geometry. The next step in our work will be the transfer to injection molded gears, where special attention will be paid to both the technological process, and proper tool manufacturing.

The failure mechanisms of polymer gears are diverse and depend on load conditions. Polymer gear defects and the loads where they appear were dealt with in the works of Senthivelan and Gnanamoorthy (2004, 2007). Tsukamoto et al. (1981, 1991) investigated about the strength of plastic gears and proposed strength design methods for plastic gears. Tsukamoto et al. (1982) have also investigated the effect of tooth profile deformation on the noise and transmission efficiency of nylon gears. Terashima et al. (1984) proposed measures for increasing the load carrying capacity of polymer gears. Besides increasing the module and tooth width, using gears with a contact ratio above 2, and tooth profile modifications was proposed. Mao et al. (2009, 2010) studied the wear of polymer gears. Different combinations with POM and PA materials were tested. The conclusion was that the POM/PA combination proves the most effective, with the drive gear made of POM. Wright and Kukureka (2001) also examined wear. Their goal was to find the wear correlation between a simple tribologic pin on disk tests and polymer gears. They found out that there is no correlation between the two applications, and that no conclusion on the wear of polymer gears can be made on the basis of simple tribologic tests. Reliable use of polymer gears in practice requires manufacturing and testing under operational, or near operational conditions. Traditional lifespan testing is time consuming, which led Pogačnik and Tavčar (2015) to develop a method for accelerated polymer gear testing.

Most of polymer gears defects are heat-related, which makes it sensible to measure gear temperature during testing. However, attention should be paid to the position of the thermal camera and the emissivity of the material whose temperature is being measured, (Letzelter, 2010). There are several models available for the calculation of the temperature of polymer gears. However, in practice they yield inaccurate results or their use is too complex (Hachmann and Strickle, 1966, Takanashi and Shoiji, 1980, Gauvin et al., 1980, Koffi et al., 1985, Mao, 2007, VDI 2736, 2014). The most widely used is the Hachmann Strickle (1966) model that is also the basis for heat calculations of gears in VDI 2736 (2014) recommendation. Also, this model also yields inaccurate results in practice, and suggestions for modifications have already been submitted, (Pogačnik and Tavčar, 2015).

The paper presents the results of experimental comparison between polymer involute gears and S-gears. The objective was to verify the higher load capacity of S-gears, which has already been proved in theory and experimentally verified on metal gears, (Hlebanja et al., 2001, 2008).
2. Methodology

2.1 Gear geometry and materials

The shape of the S-gear rack is defined by Eq. (1):

\[ y_{Pi} = a_P(1 - (1 - x_{Pi})^n) \]  

(1)

where \((x_{Pi}, y_{Pi})\) are Cartesian coordinates with the origin in the kinematic point C, while \(a_P\) is the size factor and \(n\) the exponent, (Hlebanja, 2008). Both \(a_P\) and \(n\) parameters affect the shape of the S-gear flank, and thus the gear’s expected load capacity. The effects of individual parameters on the shape of the tooth and S-gears properties were theoretically analyzed in the work of Kulovec and Duhovnik (2013).

The parameters, defining the geometry of the tested gears, are presented in Table 1. The difference between the two tooth profile shapes that were tested in our work is shown in Fig. 1. As we can see, the path of contact of the involute gears is a straight line, in the case of S-gears the path of contact is a curved line.

The gears were manufactured from the following materials:

- TECAFORM AH natural (POM-C), manufactured by Ensinger
- TECAMID 66 natural (PA 66), manufactured by Ensinger

The prescribed gear tooth thickness tolerance according to DIN 3967 was e25, with tooth thickness allowances of 

\[-0.030/-0.060 \text{ mm}\]. The circumferential backlash allowances were 0.128/0.052 mm, and the normal backlash allowances were 0.120/0.049 mm. After manufacturing, the gear geometry was measured on a Wenzel 3D measuring machine. According to the obtained values, the gears were classified as quality grade \(Q = 10\), according to ISO 1328 (1999).

Table 1. Specification of test gear parameters.

<table>
<thead>
<tr>
<th></th>
<th>Involute gears</th>
<th>S-gears</th>
</tr>
</thead>
<tbody>
<tr>
<td>module - (m) [mm]</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>number of teeth - (z)</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>pressure angle - (\alpha) [°]</td>
<td>20</td>
<td>18*</td>
</tr>
<tr>
<td>exponent - (n)</td>
<td>/</td>
<td>2.05</td>
</tr>
<tr>
<td>size factor - (a_P)</td>
<td>/</td>
<td>1.5</td>
</tr>
<tr>
<td>gear width - (b) [mm]</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>reference diameter – (d) [mm]</td>
<td>20.00</td>
<td></td>
</tr>
<tr>
<td>tip diameter – (d_t) [mm]</td>
<td>22.00</td>
<td></td>
</tr>
<tr>
<td>root diameter – (d_r) [mm]</td>
<td>17.50</td>
<td></td>
</tr>
<tr>
<td>center distance – (a) [mm]</td>
<td>20.00</td>
<td></td>
</tr>
<tr>
<td>contact ratio - (e_a)</td>
<td>1.557</td>
<td>1.425</td>
</tr>
<tr>
<td>thinning for backlash - (f_n) [mm]</td>
<td>0.045</td>
<td>0.045</td>
</tr>
<tr>
<td>backlash – (j) [mm]</td>
<td>0.090</td>
<td></td>
</tr>
<tr>
<td>pin diameter - (d_p) [mm]</td>
<td>2.00</td>
<td></td>
</tr>
<tr>
<td>ODB (without clearance) – MdK [mm]</td>
<td>23.276</td>
<td>23.313</td>
</tr>
<tr>
<td>ODB (actual) – MdK.e/I [mm]</td>
<td>23.185</td>
<td>23.198</td>
</tr>
</tbody>
</table>

*For the S-gears this is the initial pressure angle at the kinematic point C. In the case of S-gears the pressure angle is variable along the path of contact.
If we compare the tooth profile geometries from Fig. 1, we can see that the distances $A_1C$ and $E_2C$ are shorter than the distances $A_2C$ and $E_1C$ for both profile geometries. There is a larger difference between these distances in the case of involute gears, which indicates that there is more relative sliding of the meshing flanks in the case of involute gears, i.e. S-gears have a more favorable rolling/sliding ratio. The calculated sliding velocities of tested gear geometries are presented in Fig. 2. According the calculations, there are lower sliding velocities in the case of S-gears, which in combination with a more favorable rolling/sliding ratio has a beneficial impact on the heat generation.

2.2. Gear test rig

Tests were performed on the in-house designed test rig. The rig configuration allows free access to the gears for the purpose of temperature measurement with a thermographic camera. The test rig allows a continuous moment setting by using drive and braking electric motors, controlled via frequency regulators. Power is transmitted from the drive motor to the gear drive shaft via a toothed belt, which has the beneficial impact of dampening vibrations. A schematic representation of the test rig with indicated vital component parts is shown in Fig. 3. For testing purposes, the axis distance between the tested gears should be precisely adjustable. For this purpose, a positioning table was used that allows setting the axis distance to within 0.02 mm. It is a known fact that involute gears are insensitive to the changes in axis distance, which is not the case with S-gears (Hlebanja, 2013). Deflection from the calculated theoretical axis distance can occur due to manufacturing errors, axis distance setting tolerance on the test rig, deformation of shafts and bearing housings, as well as thermal dilatation of gears due to heat build-up. With polyamide gears, the geometry is also affected by moisture absorption. With polymer gears, the main reason for failure is increased gear temperature and the accompanying changes in the mechanical properties of the material. Letzelter et al. (2010) found that a slight increase in axis distance, and the resulting increased backlash, significantly increases the lifespan of polymer gears. With increased backlash the gear pair captures more air in the gaps between the teeth, which has the beneficial effect of heat dissipation away from the gear. Conversely, even a slight reduction in axis distance, and thus in backlash, will reduce the gear’s lifespan.
2.3. Testing conditions

The gears were tested at different loads and an ambient temperature of 23 °C +/- 5 °C, without lubrication. Two different material pairs (POM/POM and POM/PA66) of the drive and driven gears were tested. With the POM/POM combination, several tests were performed for each load value, while with the POM/PA66 combination there was only one test for each load value. The coefficient of friction in operation without lubrication equals $\mu = 0.28$ for the POM/POM material pair and $\mu = 0.18$ for POM/PA66 (VDI 2736, 2014). The POM/POM gear pairs were tested at moment load of 0.5 Nm – 0.8 Nm, and POM/PA66 gear pairs at a moment of 1 Nm – 1.6 Nm. All tests were performed until gear failure. During the test, surface temperature was measured using FLIR T 420 thermal camera. Emissivity of the used materials is $\varepsilon = 0.95$ (Letzelter et al., 2010). All gears were tested at an axis distance of 20.00 mm, which was measured prior to each test with a caliper with an accuracy of 0.05 mm. Appropriate tolerance for tooth thickness (DIN 3967 e25), was prescribed prior to manufacturing in order to prevent thermal elongation causing teeth jamming and a rise in temperature due to decreased lateral backlash.

![Figure 3: Representation of the test rig: 1 – drive EM, 2 – drive shaft, 3 – drive gear, 4 – thermal camera, 5 – driven gear, 6 – needle bearing, 7 – driven shaft, 8 – braking EM, 9 – positioning table, 10 - belt transmission from the driven shaft to the braking EM, 11 – ball bearing, 12 – belt transmission from drive EM to driven shaft.](image)

3. Results

The tests objective was to determine the lifespan of gears under specific loads, the tribological compatibility of the used materials, the types of defects, and gear surface temperature during operation. Such testing is time consuming, however, it yields a knowledge basis, necessary for the use of polymer gears in real applications.

Three tests were performed for POM/POM gear pairs at each load level. In defining the lifespan of a machine element, you want to know the probability that it will last a certain number of load cycles. Using the Weibull distribution, we determined the number of cycles up to which 90% of the test items would last. The Weibull distribution was used as it is suitable for a small number of samples (Abernethy, 2006). The estimated Weibull distribution parameters $\beta$ and $\eta$ were defined using MINITAB software. Figure 4 shows the lifespan curves, drawn with a 90% probability that the test item would survive. It can be observed that S-gears display longer lifespan at all tested loads.

The POM/POM material pair has been previously recognized as tribologically incompatible (Mao et al., 2009). It was assumed at the beginning of the research that for the purpose of a mere qualitative comparison between different gear geometries the choice of this material pair should be appropriate. Testing showed differences between both geometries, however, a rather small number of operational cycles was achieved at tested loads of 0.8 – 0.6 Nm. In the transition to 0.5 Nm load, there is a leap in the lifespan curve, and the lifespan improves by several classes. With a larger number of operational cycles, wear is the main failure mechanism. In this case, the yield point, i.e. when a tooth pair is no longer usable, should be defined differently. A worn gear will still transfer moment and thus fulfil its main function, however, noise and vibration levels are too high for use in a real-life application, and moment transfer via worn tooth flanks is no longer constant.
Wear results in the loss of tooth profile shape, the study of which was our goal. For this reason a POM/PA66 material combination for the drive and driven gear was used in the subsequent work. For testing purposes, all the drive gears were made of POM, and the driven ones of PA66. Figure 5 shows the results of lifespan tests for this material pair. Only one test for each load value was performed, which makes it impossible to discuss the probability of survival at a specific load level. The results can lead to a presumption of a higher load capacity of S-gears, as the gears with this flank profile geometry yielded better results at all tested loads.

In all tests, temperature was also measured during the operation of a gear pair. The surface temperature in the area of meshing teeth was measured, Fig. 3. The measured temperature is the gear’s bulk temperature, not the contact temperature, as the latter is short-lived and difficult to measure. Temperature measurements can reveal typical differences between S-gears and involute gears. Figure 6 presents gear surface temperature measurements for the POM/POM gear pairs. When S-gears are manufactured precisely, temperature increases more slowly, compared to involute gears, which – in the case of heat-related defects – results in longer lifetime. The main reason is better contact between tooth flanks, which is convex/concave at the start and end of the meshing in the case of S-gears, and convex/convex in the case of involute gears. This results in lower contact stresses in S-gears (Duhovnik et al., 2016), which in combination with less sliding and smaller sliding velocities means lower heat generation as a result of losses due to friction between tooth flanks.
4. Discussion

It was observed during testing that different load levels result in different types of failure modes. The failure mode also depends on the gear material and gear geometry. When overloaded, heat build-up in gears causes material softening and teeth deformation. With POM/POM gear pairs, such failure has appeared at a moment load of 0.8 – 0.6 Nm, regardless of the shape of tooth profile. In these types of test loads, S-gears lasted more operational cycles, however, the type of failures was the same, i.e. material softening due to overheating, as seen in Fig. 7.a. A considerable leap in lifespan was observed at the transition to the moment load 0.5 Nm, shown in Fig. 4. The reason for a major leap in lifespan lies in a change in the failure mechanism, as the 0.5 Nm load no longer causes gear overload. Gears heat up to a temperature of around 50°C, after which it stabilizes. The use of identical materials causes strong adhesion between the parts in contact, resulting in increased wear of the tooth flank. The failure mechanism of the gear pair at this load is tooth flank wear. Increased wear also increases noise and vibration during the operation of this tooth pair. At the end of the test, when a critical amount of tooth is worn away and the rest of the tooth can no longer transfer moment, teeth break, Fig. 7.b. With some experience, gear wear, tested at 0.5 Nm, can be detected by sound signals. After around 2 million cycles, noise level during the operation of the tested gears increased significantly, which can lead to a conclusion that gear wear is considerable. Some gear pairs were also stopped before complete failure in order to be able to check the amount of wear after a certain number of cycles. Figure 7.c shows wear of an involute gear after $9.826 \times 10^6$ cycles. Such a gear can still transfer moment, however, the gear ratio is no longer constant, vibrations and noise levels are high, which makes such a gear unsuitable for operation.

When POM/PA66 pairs were tested, different failure mechanisms were observed. At a load of 1.6-1.4 Nm thermal failure occurs. At a load of 1.3-1.0 Nm it is the combination of temperature and fatigue related defects. Figure 8.a shows an S-gear pair that yielded after $0.386 \times 10^6$ cycles due to thermal overload while transferring a 1.5 Nm moment. Figure 8.b shows an S-gear pair that failed under a load of 1.3 Nm after $2.3 \times 10^6$ cycles. Cracks are visible on the teeth of the driven gear, and deformed teeth on the drive gear as a result of overheated material. Figure 9.a shows an involute gear pair that failed at a moment of 1.3 Nm due to thermal overload after $0.44 \times 10^6$ cycles. Figure 9.b shows the defects on an involute gear pair after transferring 1.2 Nm and failing after $0.588 \times 10^6$ cycles. Figure 10 shows the result of measured surface temperature of an involute and an S-gear pair POM/PA66 at a load of 1.5 Nm. It can be observed that the temperature increase gradient is similar, however, in the initial operation phase, S-gears heat up less than the involute ones. With more suitable material combinations, differences between involute and S-gears are more substantial. It is also the case that in both gear types different kinds of defects appear under identical loads.

Fig. 6 Gear surface temperature measurements, POM/POM gear pairs.

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Fig. 7 Failure mechanisms of tested POM/POM gear pairs: a) material softening and plastic deformation of the teeth, b) wear and complete tooth breakage, c) severe wear.

Fig. 8 S-gear POM/PA66 gear pairs failure modes: a) thermal failure, b) combination of temperature and fatigue related defects.

Fig. 9 Involute POM/PA66 gear pairs failure modes: a) thermal failure, b) combination of temperature and fatigue related defects.

Fig. 10 Surface temperature at the moment load of 1.5 Nm and 1298 rpm, POM/PA66 gear pair.

5. Conclusion

This work has experimentally verified better performance of S-gears, as well as researched and identified the reasons for differences between both geometries, which supports our hypothesis, stated in the beginning. It can be claimed that the shape of S-gears tooth profile improves contact conditions, which results in lower contact stress and the related heat generation. Based on temperature measurement, it has been found that the course of temperature increase in S-gears is different from that in involute gears. Temperature increase in involute gears is faster, which
means higher heat generation as a result of losses due to friction in contact. A more significant difference between the load capacities of both gear geometries was detected when testing the POM/PA66 material combination. Because the POM/POM material combination is tribologically incompatible, gears wear too quickly, tooth profile shape is no longer right and all of the advantages of the S-gear profile shape are lost. This time, tests were conducted under high loads in order to obtain results in a reasonable time. Overloading triggers the overheating failure mechanism. Practical applications often require material data after 10 or even 100 million load cycles. A different failure mechanism is expected – fatigue. For this reason, future tests will be carried out under lower load levels. Priority will be given to material pairs that are tribologically compatible and interesting for real applications (such as POM/PA).

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