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Numerical simulation of meshing polymer and composite involute gears

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1. Introduction

The use of polymer gears is rapidly increasing due to their advantages compared to the gears made of metallic materials. Mass production of plastic gears is cheaper, they can carry loads without lubrication, which makes them very attractive for applications where the lubricant is not desirable (e.g. printers, household appliances). Their noise level during operation is significantly lower and they dampen vibrations better. Many of the available polymeric materials allow different variations of material pairs, which can be used for the manufacturing of gears. For better mechanical properties of the materials used they are added various fillers, usually glass and carbon fibres, whose positive impact has been recognized [1].

Despite an increase in the application of polymer gears there is no applicable standard yet for their conversion; there are only some guidelines, issued by engineering associations, such as VDI 2736 [2] (summarizing the DIN 3990 standard [3]), AGMA 920-A01 [4], ANSI/AGMA 1106-A97 [5], and by individual producers (Kissoft [6] POLYPENCO [7]). Using the finite elements method, there were some comparisons made between different standards/recommendations and numerical calculations. They revealed significant differences between the calculations based on the recommendations, and numerical results [8]. VDI 2736 [2] guidelines have collected conversion data for only few materials. Currently, there is a large variety of possible materials available, which makes a lack of data on these materials a major shortcoming. Proposed were also other approaches to plastic gears dimensioning, currently not yet in general use [9]. Therefore, for the purpose of dimensioning and using plastic gears in real applications, gears should be tested in operating or similar conditions [8]. Because testing is time-consuming and expensive, research is going in the direction of accelerated testing, where a gradual increase in load can predict the lifetime of gears at a smaller load [10].

The article will present the results obtained from numerical simulations of meshing polymer and composite gear pairs. The results of the numerical model will be analysed and compared to the results, published in Pogačnik and Tačvar's work [10] and the values, calculated according to VDI 2736 [2]. A numerical model was developed and used to simulate testing examples and to define the stress-strain state of the gears under testing conditions. The mechanisms of defects in polymer and composite gears are well-known [1]. The stress in the tooth root affects material fatigue and causes cracks in the root. Stress on the tooth flank is the main cause of wear and pitting, and cracked flank surfaces. Increased contact stress increases friction between meshing flanks, which results in higher gear temperature. When polymer gears heat up beyond the temperature of glass transition, the material softens, causing severe plastic deformation of the teeth and the gear pair fails. Analysing the tooth deformations, it was found that due to the effect of large teeth deflection, the actual kinematics of a polymer or composite gear pair is significantly different from the theoretical one, based on the assumption that tooth deformation is negligible [8, 11]. The actual contact ratio is significantly larger than the theoretically calculated one [11], which affects load transmission and product lifetime. Due to greater teeth deflections, compared to gears made of metal, transmission error is also greater, which was analysed for a variety of material pairs.

Polymers exhibit viscoelastic behaviour, which poses a question how to model material properties. There are several approaches in the literature. To model material properties, van Melick [8] used the linear elastic modulus. For conversions, he used Young's modulus at both room temperature and at a higher temperature. Letzelter et al. developed a new model that takes account of viscoelastic properties of a material [12]. There are examples in the literature, where a polymer material was modelled as hyperelastic, which is usually used for rubber materials [13]. Viscoelastic behaviour can be modelled with one of the existing models, already implemented in commercially available software, however, it requires experimentally obtained data. To obtain the data, the shear creep test or volumetric relaxation test can be used. Both require material samples and a testing device. On the assumption that a gear polymer material is in glassy state just before failure, its mechanical properties can be modelled as isotropic linear elastic, without making a major error. At the same time, no large amounts of basic data are required as we only need to know Young's modulus and Poisson's ratio, whose values are given by material manufacturers.

In the literature, there are several examples of dealing with polymer gears, using the finite element method [8, 13-15]. These models mostly simulate meshing in a single static point rather than along the entire tooth flank. There are fewer results published on the numerical modelling of composite gears [16, 17]. A variety of models is also available for modelling the material properties of composites [18]. Due to the orientation of fibres, which was defined on the basis of the numerical simulation of injection moulding and the results in Senthilvelan and Gnanamoorthy's work [19], we used the transversely isotropic model for modelling the composite material.

2. Methodology

The article objective is to compare the results of experimental work with the results of numerical simulations. An experimental research, known from the literature [10], provides some information on friction measurement and the effect of friction and low elastic modulus on force transmission. In this article, the primary objective is to present the possibility of numerical modelling of friction on tooth flanks at high friction coefficients, and the possibility of numerical simulation of this phenomenon on gears. It should be pointed out that compared to ordinary metallic gear pairs, two key parameters are different for polymer gears. They are: (1) small elastic modulus, which causes severe teeth deflection and different actual force transmission between meshing gears, (2) higher coefficient of friction.

A small elastic modulus prevents full kinematics at force transmission, so the tooth profile should be modified accordingly. However, a high coefficient of friction causes higher temperature at meshing, which directly calls for improved heat dissipation. It is these two parameters that will be the key for the development of polymer gears.

Pogačnik and Tavčar's work [10] analyses the results of lifetime testing of polymer and composite involute gear pairs. Various material pairs were tested at different rotational speeds and torques. Presented are the temperatures during operation, and root stress at testing loads, calculated according to VDI 2736 [2] standard. Using statistical methods, the authors developed a model to predict gear pair's life.

For a deeper understanding of the reasons behind defects and failure of polymer and composite gears, we simulated the meshing of tested gear pairs. For each material pair, two examples at different loads were studied, i.e. the load where the gear pair failed, and a maximum load where none of the pairs failed in the number of cycles tested. The simulated examples are shown in Table 1. The highest load with no failure in any of the material pairs was 0.4 Nm. The simulations included friction between the meshing flanks; the size of the coefficient of friction was determined according to the said article [10].

Table 1. Simulated examples

Material pair (driving/driven)	Load [Nm]	Load at gear failure [Nm]	Coefficient of friction
PA6/PA6	0.4	0.45	0.48
POM/PA6	0.4	0.75	0.29
POM/POM	0.4	0.55	0.29
PA6/PA6-30	0.4	0.55	0.4
POM/PA6-30	0.4	0.75	0.27
PA6-30/PA6-30	0.4	0.65	0.4

2.1. Model presentation

2.1.1. Geometry

A geometric model of the analysed gears was modelled in the Siemens NX 9.0 software and imported to ANSYS Workbench 15.0 with the use of step format to be analysed with the finite element method. Fig. 1 shows the geometry of the model, used in the simulations. The values, defining the geometry of involute gears, used in the simulations, are shown in Table 2.

The geometric model consists of five teeth, as the path of contact is the same for all teeth, subject to ideal geometry. To calculate the stress-strain state it is enough to analyse the path of contact for one tooth along the entire active length of the tooth flank. It means that the tooth in question goes through all of the characteristic points of contact:

- A, the starting point of contact,
- B, the inside point of the single tooth contact of the driving gear, i.e. the outside point of the single tooth contact of the driven gear,
- C, kinematic point,
- D, the outside point of the single tooth contact of the driving gear, i.e. the inside point of the single tooth contact of the driven gear,
- E, the end point of contact.

Point B is important for the calculation of the maximum stress in the root of the driven gear, and point D for the calculation of the maximum stress in the root of the driving gear. The middle (i.e. third) tooth passes all three characteristic points. The analysis of the results focuses on the third tooth for both the driving and the driven gear.

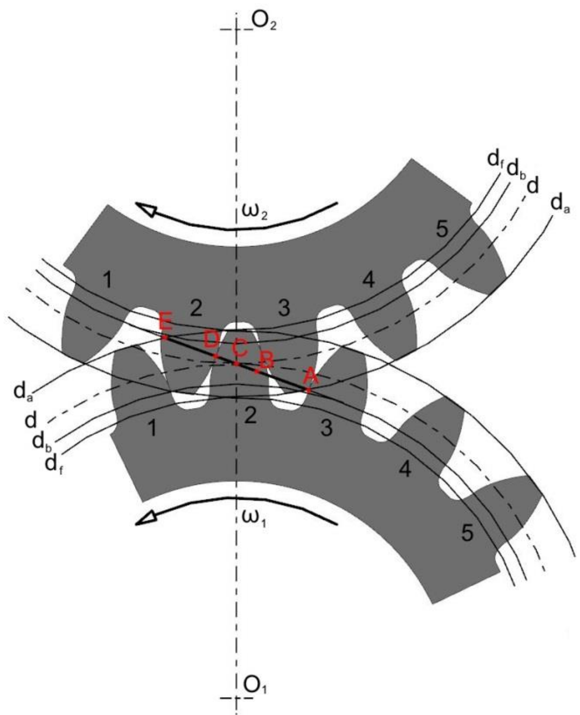


Fig. 1 Geometric model

Table 2. Gear values

Module - m [mm]	1
Gear width - b [mm]	6
Number of teeth - z	20
Pitch diameter - d [mm]	20
Pressure angle - α_n [°]	20
Contact ratio - ε_α	1.557

With the meshing gears with a profile overlap of $1.1 < \varepsilon_\alpha < 2$, the load is transferred via one tooth for one part of the meshing, and via two teeth for the other part – load sharing. To describe load transmission for the meshing of one tooth through all the characteristic points, at least three teeth are required. In order to consider the effect of other aspects of geometry on tooth stiffness and load transfer, one more tooth was added on each side. The geometric design of the model thus consists of five teeth. Because meshing along the entire active length of the tooth flank is simulated, it is possible to follow the stress-strain state in all the meshing points along the path of contact.

2.1.2. Materials properties

According to the tested material pairs in Pogačnik and Tavčar's work [10], which we refer to, the following was selected as the gear material:

- Polyamide 6 (PA6, Ultramid B3S, BASF)
- Polyamide 6 + 30% glass fibres (PA6-30, Zytel 73G30HSL NC010, DuPont)
- Polyacetal (POM, Delrin 500P NC010, DuPont)

The gear material without reinforcing fibres was modelled as isotropic linear elastic, using the elastic module and Poisson's ratio, whose values are given by material manufacturers. Modelling isotropic elasticity, material constants E_{PA6} = 3400 MPa and ν_{PA6} = 0.4. were used for PA6. As for POM material constants E_{POM} = 3100 and ν_{POM} = 0.35 were used. Polymer materials behave viscoelasticly, however, our assumption was that modelling polymer with isotropic elastic model means no major error in our simulation. Viscoelastic properties of a material play a more significant role when a material heats up beyond the temperature of glass transition. It means that at least for the initial stage of operation, when the gears are not yet heated, no major error is being made. It takes time for the viscous properties to become noticeable. Due to the short-term load of an individual tooth, it is assumed that no major error is being made if the linear elastic modulus is used.

Modelling the properties of composite gears material, reinforced with glass fibres, the transversely isotropic model was used [18]. The following material constants were used for modelling material properties of glass fibre-reinforced gears:

$E_{x,\ PA6\text{-}30}$ = 8776 MPa
$E_{y,\ PA6\text{-}30} = E_{z,\ PA6\text{-}30}$ = 5364 MPa
$\nu_{12,\ PA6\text{-}30} = \nu_{13,\ PA6\text{-}30}$ = 0.4057
$\nu_{13,\ PA6\text{-}30}$ = 0.4496
$G_{xy} = G_{xz}$ = 2282 MPa
G_{yz} = 1850 MPa

The Zytel (PA6 + 30% GF) material properties were obtained from the materials properties data sheet, issued by a commercial injection moulding simulation software Autodesk Simulation Moldflow Insight 2015, where the injection moulding simulation was performed, Fig. 2.

2.1.3. Boundary conditions

The driving gear is linked via a rotational joint to fixed point O_1 on the central axis, Fig. 1. The driven gear is linked to point O_2 in the same manner. The sides of the elements, linked to the fixed points via rotational joints, are allowed rotation only around the z-axis of the corresponding local coordinate system, Fig. 3. The use of a local rotational joint makes gear rotation possible. To simulate gear meshing, the rotational joints should be assigned torque and rotation, Fig. 3. The rotational joint of the driving gear was assigned rotation, and the joint of the driven gear was assigned torque in the direction opposite to rotation. Rotation and torque are transmitted to the gear geometry via the linkages.

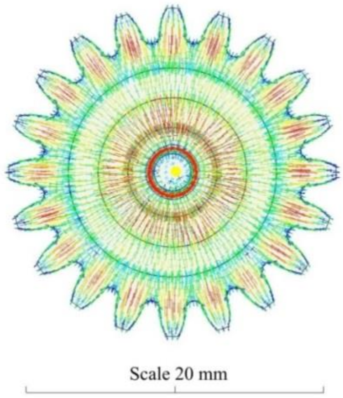


Fig. 2 Fibre orientation after injection moulding

2.1.4. Discretisation of the area

For discretisation of the area, we used a generic meshing algorithm, included in the software. The geometry of tooth flanks was divided into smaller sub-areas. They were assigned smaller elements for a denser mesh in order to provide a more detailed description of stress in the material. The elements used were hexahedrons with midside nodes. Tetrahedrons with midside nodes were used in the places where the software was unable to describe geometry with hexahedrons. The prescribed length of the elements on the active flank of the middle tooth is 0.05 mm, and 0.1 mm on the other teeth in contact.

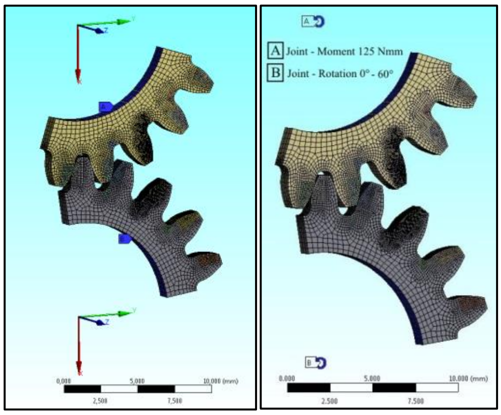


Fig. 3 Discretisation of the area, local coordinate systems of rotational joints and boundary conditions

2.1.5. Friction modelling

In the simulation of meshing gears, friction between the tooth flanks in contact was modelled. As contact surfaces, selected were the flank areas the profile of which is of involute shape. The prescribed coefficients of friction for individual material pairs were prescribed as shown in Table 1 [10]. Used was the Augmented Lagrangian formulation of contact between the elements. Modelling the contact with friction results in the simulation becoming non-linear, which requires good skills in numerical modelling in order to achieve the convergence of a solution. Taking account of friction yields a more accurate solution as friction has a significant effect on shear stress that occurs on the surface and under it.

3. Results and discussion

Using a numerical model, we calculated stresses and deformations that occur at tested loads. The analysis of results focused on maximum values of von Mises equivalent stress values in the tooth root and flank. Gear stress analysis, tested at a load of 0.4 Nm, will be presented first, followed by the calculated stress at the failure point load of a material pair. Analysed will be the effect of large teeth deflection on the meshing kinematics and transmission error. On the assumption of ideal geometry, the stress-deformation pattern is the same for each tooth along its path of contact, which makes it sufficient to direct our focus on the middle, i.e. third, tooth.

3.1. Root stress

In the presented results of Pogačnik and Tavčar's work [10], the highest torque where none of the material pair failed was 0.4 Nm. The calculated values in the tooth root of the driving and driven gears are shown in Fig. 4. It is logical that the course of the highest stress in the tooth roots of the driven gear runs in the direction exactly opposite to the one of the driving gear.

Stress in the tooth root according to VDI 2736 [2] standard can be calculated using equation 1. It shows that the calculation of root stress is not dependent on the elastic modulus but on the modulus and gear width, and the influencing factors (K_F, Y_{Fa} – tooth shape coefficient, Y_{Sa} – stress correction factor, Y_ε – root contact ratio factor, Y_β – helix angle factor).

$$\sigma_F = K_F \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_\varepsilon \cdot Y_\beta \cdot \frac{F_t}{b \cdot m_n} \qquad (1)$$

$$K_F = K_A \cdot K_v \cdot K_{F\beta} \cdot K_{F\alpha} \qquad (2)$$

Coefficient K_F is the product of several coefficients, equation 2 (K_A – application factor, K_v – dynamic factor, $K_{f\beta}$ – face load factor, $K_{f\alpha}$ – transverse load factor). For thermoplastic gears, fulfilling the condition $b/m_n \leq 12$, VDI 2736 guidelines [2] suggest this simplification: $K_F \approx K_A$; $K_A = 1 \ldots 1.25$.

In contrast to the standard conversion, some differences occur in the tooth root stress values. Van Melick [8] arrived at a similar conclusion. Table 3 shows the highest calculated stress values in the tooth root of the driving and driven gears. The biggest difference in the driving gears occurs between the material pair POM/POM and PA6-30/PA6-30, and equals 4.24 MPa. With driven gears, the biggest difference occurs between the material pair PA6/PA6 and PA6-30/PA6-30, equalling 4.20 MPa. With the numerical model, the calculated stress values are a little lower, compared to those calculated according to VDI 2736 [2] guidelines, if using the application factor K_A = 1.25, or similar if using the application factor K_A = 1.

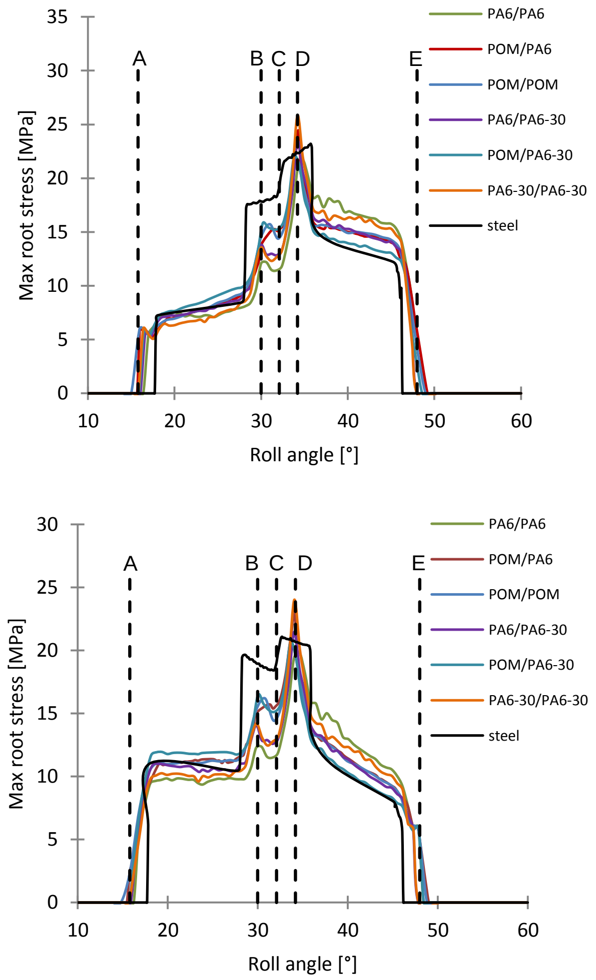


Fig. 4 Maximum root stress in driving (top) and driven (bottom) gear

Table 3. Maximum stress values in the roots of the driving and driven gears

Material pair (driving/ driven)	Max. root stress in driving gear [MPa]	Max. root stress in driven gear [MPa]	Calculation according to VDI 2736 K_F = 1.25 [MPa]	Calculation according to VDI 2736 K_F = 1 [MPa]
POM/POM	21.59	20.33	28.23	22.58
POM/PA6	24.10	22.87	28.23	22.58
PA6/PA6	21.73	19.83	28.23	22.58
PA6-30/PA6-30	25.83	24.03	28.23	22.58
POM/PA6-30	21.99	20.65	28.23	22.58
PA6/PA6-30	23.13	21.31	28.23	22.58

3.2. Flank stress

VDI 2736 [2] guidelines also say that calculated flank stress depends on the properties of the material of which the gear is made. The effect of material is expressed by the coefficient of elasticity Z_E. In the numerical calculation, the difference in the calculated stress values between different material pairs at the same load is very noticeable.

With increased material stiffness, contact stress between two meshing flanks is expectedly higher. Flank stress values, obtained by the numerical model, are shown in Fig. 5.

The lowest calculated flank stress occurs in the material pair POM/PA6 which proved the most suitable in testing [10, 20]. Lower flank stress means lower friction and consequently less heat build-up. It can be noticed that together with a favourable coefficient of friction the material pair POM/PA6 also exhibits the best contact conditions, i.e. the lowest contact stress. It shows that the results of the numerical simulation can suggest a suitable material pair for a gear pair. This is very useful when deciding on new materials, when no experimental data are available yet.

Impact shocks [6] appear at the beginning and end of contact path. According to the results in the literature [8] they have a significant impact on teeth wear. The highest flank stress values are calculated at the beginning and end of meshing, where sliding velocity is the highest. Impact shocks can be reduced by geometrical modifications – tip relief [6].

Table 4 shows the calculated stress values on the tooth flank, using a numerical model according to VDI 2736 [2] guidelines, equation 3. The meaning of the coefficients in equation 3: Z_E – coefficient of elasticity, Z_H – tooth shape coefficient, Z_ε – flank contact ratio factor, Z_β – flank helix angle factor. Similarly to the calculations of root stress, VDI 2736 guidelines [2] suggest the following simplification for the calculation of flank stress in thermoplastic gears: $K_H \approx K_A$; $K_A = 1 \ldots 1.25$.

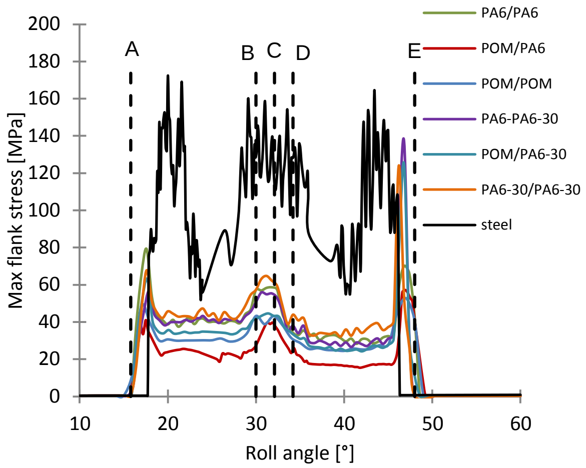
$$\sigma_H = Z_E \cdot Z_H \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t \cdot K_H}{b_w \cdot d_1} \cdot \frac{u+1}{u}} \qquad (3)$$

$$K_H = K_A \cdot K_v \cdot K_{H\beta} \cdot K_{H\alpha} \qquad (4)$$

Table 4. Maximum flank stress values for the driving and driven gears

Material pair (driving/ driven)	Max. flank stress in driving gear [MPa]	Max. flank stress in driven gear [MPa]	Calculation according to VDI 2736 $K_F = 1.25$ [MPa]	Calculation according to VDI 2736 $K_F = 1$ [MPa]
POM/POM	43.25	46.20	48.70	43.56
POM/PA6	43.49	41.52	49.81	44.56
PA6/PA6	59.85	61.70	51	45.62
PA6-30/PA6-30	64.77	65.73	70.52	63.08
POM/PA6-30	44.63	49.34	56.68	50.69
PA6/PA6-30	57.37	61.64	58.45	52.28

Comparing the calculated stress values in Table 4, it can be noticed that there are no major differences between the stress values, calculated according to VDI 2736 [2] guidelines and the FEM ones, i.e. they mostly depend on the value of the K_F coefficient. It should be noted that the stress values (Table 4), calculated by the numerical model, do not include the starting and ending shocks. Instead, they include the highest stress value that occurs in the middle of the path of contact around the kinematic point because also the conversion according to VDI 2736 [2] is made for this area. The shocks at the beginning and end of contact are considerably higher.



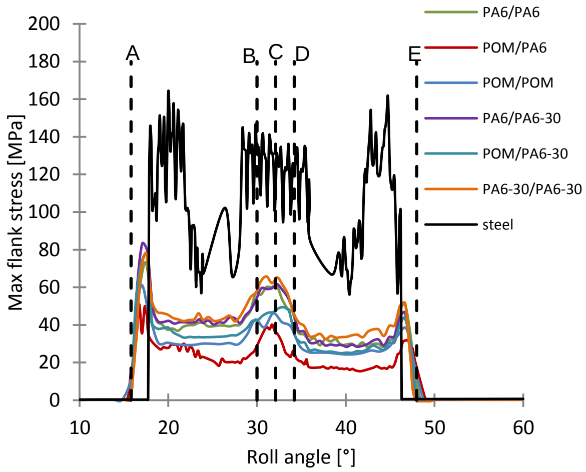


Fig. 5 Maximum flank stress in driving (top) and driven (bottom) gear

3.3. Load at failure point

It is interesting to look at the stress-deformation state of gears at failure point loads. The calculated stress values in the tooth root of the driving and driven gear are the highest in the POM/PA6 material pair, Fig. 6. The calculated stress values on the tooth flank of the driving and driven gear are shown in Fig. 7. It turns out that flank stress values are not the highest in the POM/PA6 material pair, regardless of the fact that the highest load was simulated on this gear pair. It shows that materials properties have a considerably lower impact on root stress (which is why the standard does not consider them) compared to the stress that occurs on the flank. In the POM/PA6-30 material pair the same load was simulated as on the POM/PA6; however the calculated stress values in the root are slightly lower, which can lead to the conclusion that materials properties have a low impact also on the size of stress values in the root. The same was found in studying the identically loaded examples. The guideline does not consider this impact, as it is not as significant as for the stress values on tooth flank.

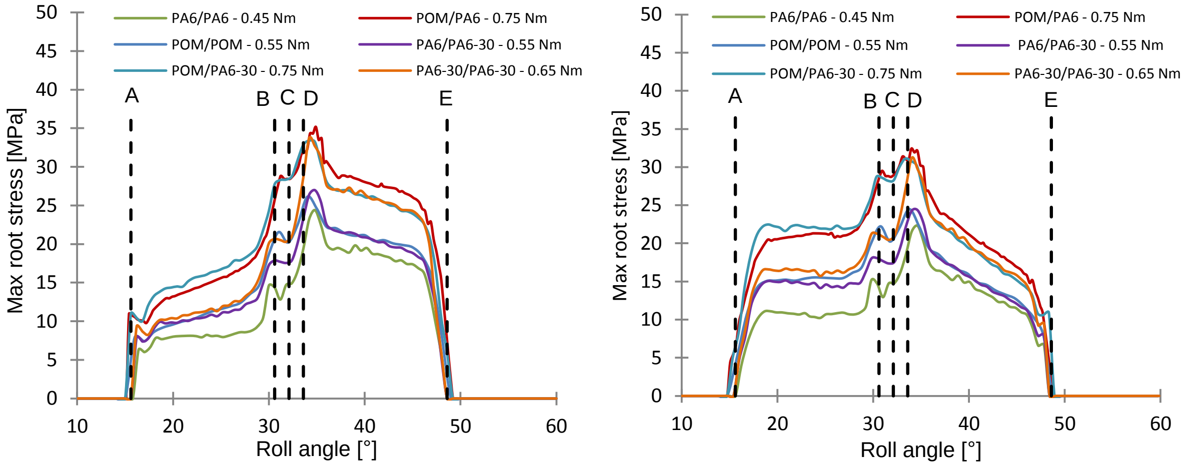


Fig. 6 Maximum root stress in driving (left) and driven (right) gears

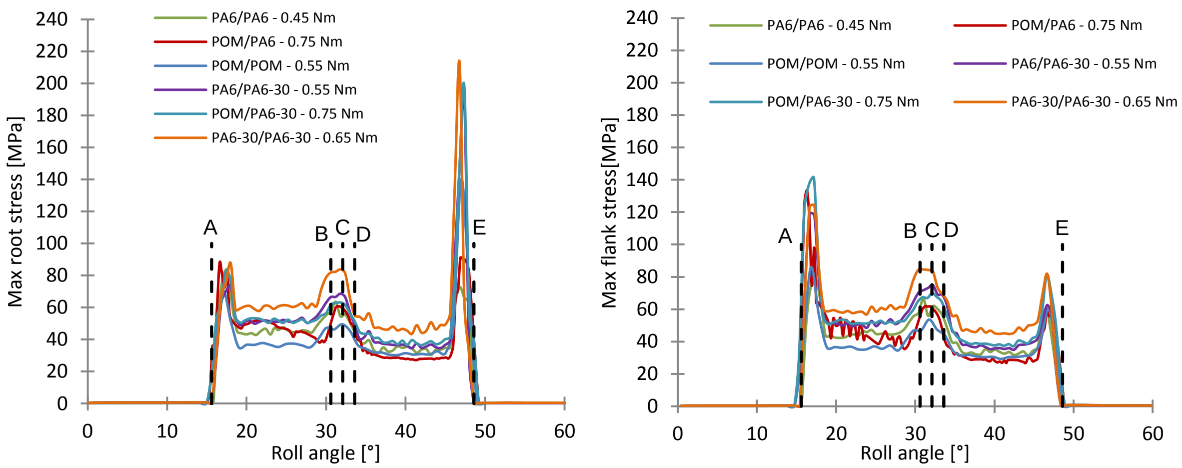


Fig. 7 Maximum flank stress in driving (left) and driven (right) gears

3.4. Meshing kinematics

Simulating polymer and composite gear meshing, it was found that the actual shape of the path of contact is different from the theoretical one, otherwise characteristic for involute gears. The same findings have been already published in the literature [11]. Due to severe teeth deflection, the driving tooth does not begin meshing with the driven in the tooth root. The driven gear's tooth touches the driving tooth with its tip further up from the root, close to the kinematic circle, and continues to mesh down the tooth of the driving gear, Fig. 8 (reciprocal meshing [8]). At some point the meshing direction changes and the tooth of the driven gear normally meshes up, towards the tip of the driving gear's tooth. Such meshing results in the path of contact with extensions at the beginning and end of meshing [8, 11].

In theory, when gears are considered as rigid bodies, the meshing of one tooth (of analysed gear geometry) along the entire path of contact requires a gear rotation of 24.5°. In simulated cases, gears are modelled as deformable bodies. As a result, and due to large teeth deflection, the rotational angle increases by more than 30%. Table 5 shows the required rotations of gear for one tooth meshing along the entire A-E path of contact, and the increased rotational angles, compared to theoretical values. Similar results have been published also by other authors [11].

Together with teeth deformation, theoretical distances between characteristic points of meshing change too. If the path of contact A-E is divided according to theoretical calculations, the distance between points A and B is 35.77 % of the path of contact's length, the distance between points B and D is 28.47% and between D and E 35.77 %. Fig. 9 shows the calculated share of double (A - B, D - E) and single (B - D) pair tooth contacts for the simulated examples. It can be noticed that the proportion of double pair tooth contact as a result of large teeth deflections (and thus longer path of contact) increases. As a result, the proportion of single pair tooth contact decreases, Table 6. It can be concluded that a lower share of single pair tooth contact has a favourable effect on the gear's service life.

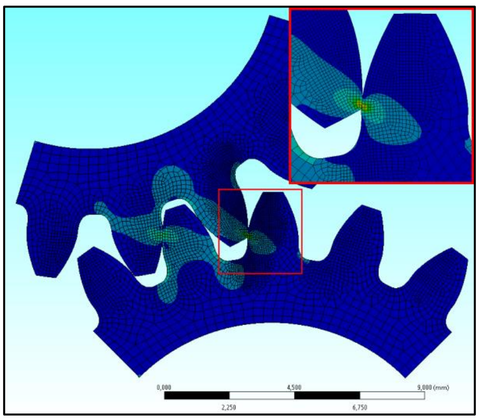


Fig. 8 The beginning of meshing (reciprocal meshing [8])

Table 5. Required rotation of gear for one tooth meshing along the A-E distance

	PA6/PA6	POM/PA6	POM/POM	PA6 / PA6-30	POM / PA6-30	PA6-30 / PA6-30
0.4 Nm	32.8° (33.88%)	33.5° (36.73%)	34° (38.78%)	32.03° (30.73%)	32.71° (33.51%)	32.2° (31.43%)
failure point	32.8° (33.88%)	34.1° (39.18%)	34° (38.78%)	33° (34.69%)	33° (34.69%)	33° (34.69%)

Table 6. Proportion of single pair B-D tooth contact [%]

	PA6/PA6	POM/PA6	POM/POM	PA6 / PA6-30	POM / PA6-30	PA6-30 / PA6-30
0.4 Nm	10.98	7.46	11.76	11.24	11.24	13.04
failure point	10.96	6.45	5.88	9.10	5.26	9.10

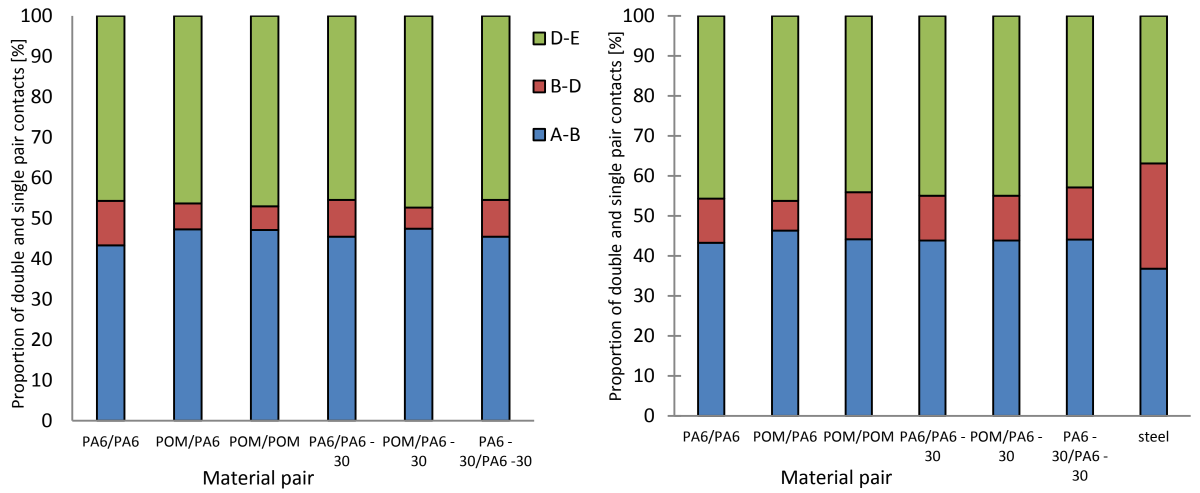


Fig. 9 Comparing the proportion of double and single pair tooth contacts; failure point load (left), identical load (right)

3.5. Transmission error

Transmission error of the meshing tooth pair is the difference between the actual rotation of the driven gear (driven shaft) and the theoretical rotation of the driven gear (driven shaft). Theoretically, the driving and driven gears are rigid bodies and the rotations of the driving and driven shafts are identical in size.

The primary reasons for transmission error include:

- geometric discrepancies in gears (eccentricity, center distance errors, uneven teeth thickness, pitch diameter…)
- local tooth deflection

In our case, geometric discrepancies in gears are not taken into account. The calculated transmission errors are only down to teeth deformation. It turns out that tooth deflection in polymer gears has a more significant effect on transmission error than geometric discrepancies [21].

The driving and driven shafts in the model are represented as rotational joints. The prescribed rotation of the driving gear's rotational joint equals the rotation of the driving gear. Setting up a model of meshing gears in this way allows calculating rotation on the rotational joint of the driven gear, which represents the rotation of the driven shaft. Due to teeth deflection, differences appear between rotations on both the driving and the driven gear's rottional joint. The driven shaft slips a little behind the driving one. Transmission error can be calculated by deducting the driving shaft rotation from the driven shaft rotation.

Due to teeth deflection, the driven gear always slips a little behind the driving gear. In the simulation, both gears were rotated for 60°, which makes it possible to monitor the slipping of the driven gear during the meshing of several teeth. It can be noticed that the slipping pattern repeats cyclically. The driven gear slips behind more in the area of single pair tooth contact and less in the area of the double one. With single pair tooth contact, the entire load is transmitted via a single tooth, which causes larger tooth deflection as a result and the difference between the rotations increases there. The slipping of the driven shaft behind the driving one is not consistent; it fluctuates between the highest and the lowest differences in rotation. Such fluctuations in slipping cause torsional excitation of the shaft and can affect the gear's lifetime if torsional excitation in close to the shaft's resonant frequencies. The calculated transmission error at a load of 0.4 Nm and the failure point loads are shown in Fig. 10.

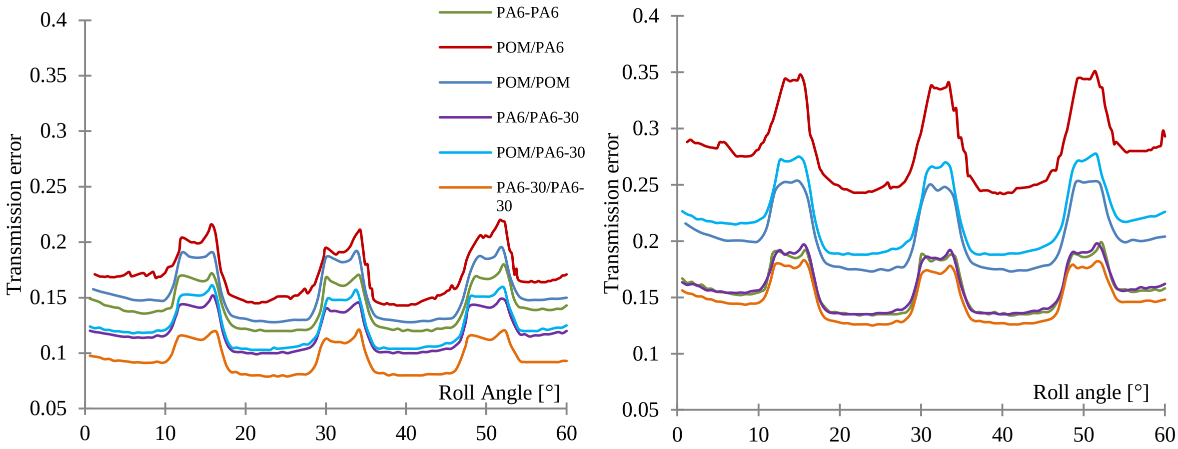


Fig. 1 Transmission error at identical loads (left) and failure point loads (right)

4. Conclusion

It has been shown that numerical simulations can be of assistance in dimensioning gear pairs, made of polymer and composite materials. It has been found that there is no significant difference between the values of root and flank stress, calculated according to VDI 2736 and those, calculated according to FEM. It should be pointed out here that flank impact shocks occur at the beginning and end of contact path, in the area of highest sliding velocity. Higher contact stress results in higher friction, which – in combination with the highest sliding velocities at the beginning and end of contact path – leads to increased wear of gear pairs. The guidelines do not consider impact shocks.

It has been found that numerical calculations can lead to a conclusion on suitable pairing of materials. If a simulation is to take account of friction, it is necessary to know the coefficient of friction of the analysed material pair. It can be obtained from a simple tribological test (pin-on-disk). For a number of material pairs, coefficients of friction are also available in the literature. Polymer gear pairs' service life usually depends on operating temperature. During operation, temperature raises due to friction in contact areas and the hysteresis loss in the material. It is a known fact that hysteresis losses have a much smaller impact on heat generation, compared to friction between teeth. In testing, the POM/PA6 material pair proved to be the most suitable. Calculation results show that this material pair has the best contact conditions, as the lowest flank stress was calculated for it. In combination with the second lowest coefficient of friction it naturally means the lowest friction and thus the lowest temperature generation.

By simulating meshing of polymer and composite gears, we discovered an extended path of contact as a result of large teeth deflections. A longer path of contact triggers an increase in contact ratio and a decrease in the area of single pair tooth contact. By reducing the area of single pair tooth contact, load distribution between the teeth improves. In other words, the load transfer duration via single pair tooth contact is reduced. The calculated results, compared to the results of lifetime tests, have led us to the conclusion that a longer path of contact and consequently a smaller area of single pair tooth contact have a favourable effect on the lifetime of polymer and composite gears.

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A b s t r a c t

The results of a numerical simulation of meshing polymer and composite gears are presented in the paper. The numerical model used for simulations was built in Ansys Workbench 15.0, using the finite element method to obtain the solution. The results are related to the gear tests, performed by the authors and presented in the literature. A detailed stress and strain analysis of meshing gears is presented. On the basis of which gears of different polymer and composite materials can be rated and appropriate material combinations can be chosen. With the presented numerical model, root and flank stresses in each meshing point along the path of contact can be calculated. Stresses (root and flank) obtained by simulations will be compared with the stresses calculated according to VDI 2736 guideline. The numerical model takes into account friction between the tooth flanks in contact, which significantly contributes to improving the accuracy of the numerical solution.

Keywords: polymer and composite gears, stress analysis, deformation analysis, non-linear finite element method