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Tooth-Bending Effects in Plastic Spur Gears Influence on load sharing, stresses and wear

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Abstract—This describes the investigation of a steel-and-plastic gear transmission, using both numerical (FE) and analytical methods. The aim was to study the influence of the stiffness of the gear material on the bending of the gear teeth, and the consequences on contact path, load sharing, stresses and kinematics. As has been shown in previous literature, the tooth bending of plastic gear teeth results in an increase in the contact path length and in a considerable change in load sharing. The root stresses, in principle independent of the modulus, become quite different for plastic gears due to changes in load sharing. Also, on the contact stresses which depend on the modulus the preliminary and prolonged contact path have a significant influence, inducing very high stress peaks. These peaks, due to contact of the sharp tooth tips, result in pressure-velocity (PV) values, which are up to a factor of seven times higher than calculated using conventional theory. Furthermore, the FEA results show that the kinematics of plastic gears change dramatically. In the extended part of the contact path, the tooth tip makes a reciprocating movement on the root of the mating gear. Our hypothesis now is that this high PV, reciprocating movement is the governing mechanism of wear in plastic gears.

Index Terms— Consequences, FEA, Gear teeth, Kinematics, Load sharing, PV Pressure Velocity, Stresses.

1 Introduction

PLASTIC gears are now commonly used in the more demanding automotive applications, such as electronic power steering, electronic throttle control and starter motors. However, the fundamental knowledge of plastic gear design and engineering does not seem to have kept pace with the number of gear applications.

Much research was done in the 1970s and 1980s, especially in Germany, on predominantly polyacetal (POM) and polyamide (PA) gears (Refs. 1–8). These experimental investigations focused on measuring the load capabilities for various gears under varying conditions. These studies typically yielded curves for the normalized force versus the number of cycles to failure. In this approach, the module of the gear and other geometry factors are still important variables. Relatively little attention was paid to stress analysis and wear mechanisms and how these changed under varying conditions. The standardized calculations of ISO 6336 and DIN 3990 were used to calculate indicative stresses, but it was under- stood that a lot of discrepancies existed between "metal" theories and "plastic" practice. In the 1980s, a specific plastic gear standard was developed by VDI (VDI 2545), and although it was withdrawn again in 1996 for unknown reasons (Ref. 9), this standard is still commonly used. In the 1980s, Yelle and Burns (Ref.10) conducted substantial research to develop a more fundamental base to plastic gearing. Their approach succeeded in accounting for the tooth bending of plastic gears and in calculating real contact ratios for this type of transmission.

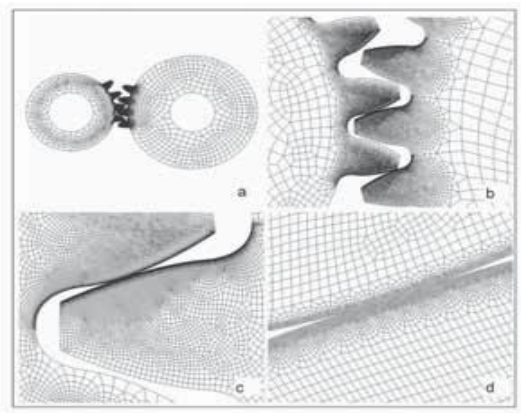

Figure 1—Details of the FE mesh. An overview of the complete model is shown in figure a, while zooming in on the refined mesh in the contact area from figures b to d.

With the ongoing development of finite element software packages and improved algorithms accompanied by sufficient computing power in the 1990s it became possible to solve complex contact problems like meshing (plastic) gears.In the group of Walton at the University of Birmingham, the experimental research on plastic gears was combined with numerical work (Refs.11 and 15–17). They confirmed the analytical findings of Yelle and Burns by FEA and showed that load sharing changes dramatically for plastic gears. Kapelevich and co-workers used FEA to modify tooth shapes and to optimize the tooth geometry specific for plastic gears (Ref. 14).

However, although substantial research has been conducted in the field of plastic gears, this does not seem to have resulted in generally accepted design rules. Today's plastic gear designs still seem to be based on empirics, experience and comparative calculations based on metal gear standards.

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The essential differences between metal and plastic gears and how they affect kinematics, stresses and wear issues—are still rather undefined

In building knowledge in the field of plastic gearing, many fundamental questions were encountered. For instance, what are the differences in kinematics between plastic gears and metal gears? To what extent are the standards developed in the 1970s for metal gears applicable for plastics?

To find answers to these questions, a program to compare the results of semi-analytical methods (e.g., ISO standards and KissSoft) and finite element analysis was begun, in addition to an extensive experimental program on plastic gears. In our approach, these calculations were used to study previously observed phenomena, like changes in contact path length and load sharing, but also to investigate stresses and kinematics. Finally, a hypothesis was formulated on the governing mechanism of wear in plastic gears.

Table 1—Details of gear geometries.

Parameter	Symbol	Pinion	Gear	Un
Module	m	2	2	m
Center	a	53	53	m
Pressure Angle	α	20°	20	°
Widt	b	12	12	m
Number of	z	22	31	-
Pro?le Shift	x	0	0	Mo
Addendum	haP	1.25	1.2	Mo
Dedendum	hfP	1.0	1.0	Mo
Root Radius	ρf	0.38	0.3	Mo

Table 2—Relevant material properties required for FEA.

Material	Young's Modulus [GPa]	Poisson's Ratio [-]
Steel	206	0.3
Stanyl GF30	10	0.4
Stanyl UF 23°C	3	0.4
Stanyl UF 140°C	0.7	0.45

2. METHODS

This paper describes the results of Finite Element Analyses (FEA) per- formed in order to study the effect of modulus and tooth bending on the kinematics and stresses in a gear pair. The FEA were done using a commercial FE software package, MSC.MARC. The gears were modeled as two discs with four teeth each, under plane strain conditions (2-D analysis). Around 80,000 first-order quads were used, with a refined mesh near the contact surfaces, in order to capture accurately the con- tact stresses. Detailed pictures of the finite element mesh of the gears are shown in Figure 1.

Linear elastic deformation behavior was assumed for all materials. Although the unfilled materials do show significantly inelastic behavior, this is expected to have only a minor influence.

The calculations were done using the ISO 6336 standard (similar to DIN3990) and KissSoft machine design program (semi-analytical).

In this study the focus was on a gear transmission, consisting of a steel pinion and gear, which was made of various materials. Both gears have a module of two, a width of 12 mm, and no profile shifts are applied. The tooth profile is standard according to DIN 867, with a pressure angle of 20°. Further details are given in Table 1.

In the FEA, different material properties were assigned to the gears. A steel gear was chosen, with the same properties as the pinion, a glass- fiber (GF)-reinforced grade of Stanyl (PA-46, 30 wt % glass fiber), and an unfilled Stanyl, tested at either room temperature or at 140°C. The relevant elastic properties are given in Table 2.

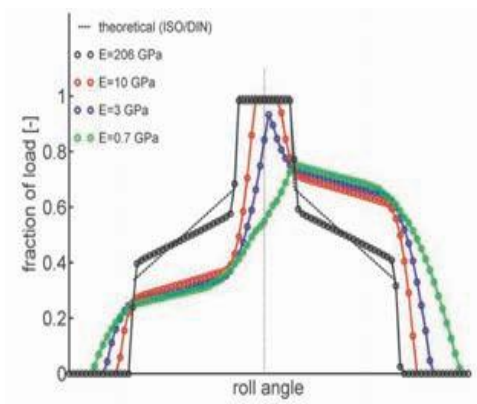

Figure 2—Load sharing between teeth of a steel pinion and a steel gear (black circles), a glass fiber (GF)-reinforced plastic gear (red circles), an unfilled plastic gear (blue circles) and an unfilled gear at elevated temperature (green circles). The dashed black line represents the theory (ISO 6336) and the dotted black line indicates the pitch point.

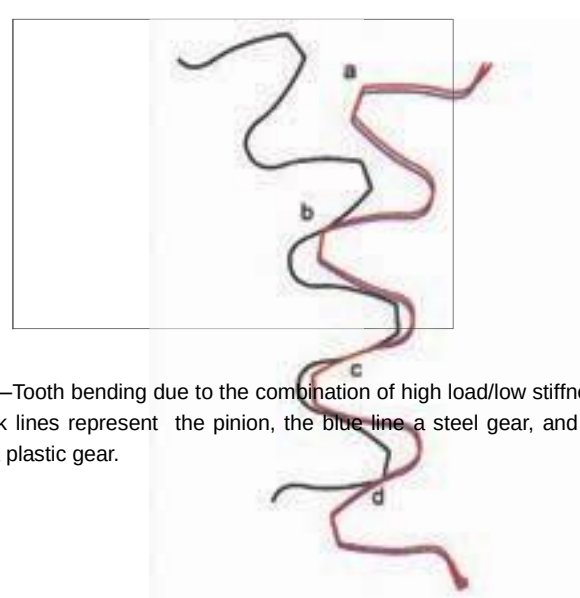

Figure 3—Tooth bending due to the combination of high load/low stiffness. The black lines represent the pinion, the blue line a steel gear, and the red line a plastic gear.

3. Results

Load Sharing- During meshing of a gear pair, the total load is shared between different teeth. In standard spur gear transmissions, a contact ratio between one and two is found, meaning that for a certain part of the meshing cycle, a single tooth carries the load; while for the remaining time of the meshing cycle, two teeth share the load. Following the classical theory—for instance the ISO 6336 standard (specific for metal gears)—the load is shared according to the 1/3-2/3-3/3rule, meaning that the load fraction at the initial moment of contact equals 1/3, increases to 2/3 at the moment the preceding tooth is leaving the contact, and is carrying the full load during the period of single-tooth contact around the pitch point. After the pitch point, the load fraction follows the same scheme, but in reverse order. In Figure 2, which shows the load fraction as a function of the roll angle (rotational angle of the pinion), this is represented by the dashed (theoretical) line.

Using FEA, the load sharing can also be studied. Calculations and other studies in literature (Refs. 11 and 12) show that during meshing of two steel gears, load sharing is closer to a 2/5-3/5-5/5 rule, represented in Figure 2 by the black circles (E=206 GPa).

When a steel pinion meshes with a plastic gear, the load sharing changes dramatically. Figure 2 shows the load sharing for combinations of steel and various plastics, and the most striking change is that the load sharing becomes skewed when a steel pinion meshes with a plastic gear, resulting in a load share of approximately 1/3 in the first part of the meshing cycle and a load share of 2/3 in the last part.

Another significant effect is that the contact path increases, resulting in a preliminary contact earlier in the meshing cycle and a prolonged contact at the end, reducing the period of single- tooth contact. This has already been reported in literature in analytical (Ref.10) and numerical studies (Refs. 11 and 12). For a glass fiber-reinforced plastic gear (E=10 GPa), represented by the red circles in Figure 2, the single-tooth contact period is approximately halved, compared to a steel-on-steel combination. For an unfilled plastic gear (E=3GPa) and an unfilled plastic gear at elevated temperature (E=0.7 GPa), respectively, represented by the blue and green circles, single-tooth contact no longer occurs during meshing. With decreasing modulus, the maximum load share decreases to a plateau value of approximately 2/3. The origin of these effects lies in the fact that considerable tooth bending takes place, as is shown in figure 3.

The black steel pinion meshes with the blue steel gear, as expected, while considerable tooth bending for the red plastic gear is observed. At position 'a,' the red gear has a significant lag compared to the blue gear. At position 'b,' a lag can still be seen in that the red gear comes into contact with the black pinion before the theoretical point of initial contact. For position 'c,' the teeth of the red and blue gear are virtually at the same position. However, this tooth is the most deformed, making the rest of the gear and teeth lag. At position 'd,' the load on the tooth reduces and the tooth bends back.

Tooth bending is not solely determined by the modulus of the gear, but also by a combination of stiffness (Young's modulus) and loading. As the ratio of Young's modulus and yield stress (i.e., loading limit) for polymers is roughly an order of magnitude lower than for steel, tooth bending plays a much more prominent role in plastic gearing.

Contact Path. Preliminary and prolonged contact takes place during meshing of plastic gears, resulting in an extended contact path. In theory, the contact path length is defined as the length along the contact line from the initial point of contact until the last point of contact divided by the base circle pitch.

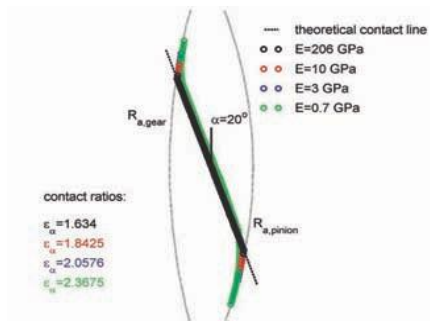

Figure 4—Contact path during meshing of a steel pinion with a gear of various materials.

Table 3—Maximum occurring root stresses and Hertzian contact stresses for a steel pinion meshing with a gear of various materials (Young's moduli).

Stress	Modulus	ISO 6336	VDI2545	KissSoft	MSC.MARC
Root Stress	E=206 GPa	74.8	42	77.4	73.4
	E=10 GPa	74.8	42	77.4	70.7
	E=3 GPa	74.8	42	77.4	65.1
	E=0.7 GPa	74.8	42	77.4	51.7
Contact Stress	E=206 GPa	609	609	609	800
	E=10 GPa	193	193	193	230
	E=3 GPa	107	107	107	120
	E=0.7 GPa	54	54	54	50

As can be seen in Figure 4, the theoretical line of contact—typical for involute gearing—is well captured by two meshing steel gears (black circles). A contact ratio, of 1.634, derived by FEA, also agrees well with theory (1.63). For a GF-reinforced plastic (red circles), the contact ratio increases to 1.84, and for

an unfilled plastic, a contact ratio of two is just exceeded, which corresponds well with the fact that single-tooth contact was not seen in Figure 2. For an unfilled plastic gear at elevated temperature, the contact ratio goes to 2.37.

As can be seen in Figure 4, for plastic gears the contact path increases at the beginning and end. However, the extension of the contact path does not coincide with the straight contact line, but instead bends off. In fact, the extension of the contact path lies exactly on two circles—the tip circles of both gears.

Stresses Table 3 shows the values for the root (bending) stress and the (Hertzian) contact stresses as calculated following ISO 6336 (or DIN 3990), VDI 2545, using KissSoft and MSC MARC (maximum around pitch point). The values for steel gears calculated by ISO/DIN, KissSoft and FEA agree very well. The VDI standard gives much lower values for the root stress because the stress correction factor, which is taken into account by ISO/DIN and KissSoft, is ignored. For the contact stresses, the agreement between the standards is excellent, given that they are using virtually the same equations.

Furthermore, Table 3 shows that both the root stresses and the Hertzian contact stresses decrease with decreasing Young's modulus of the mating gear. For the Hertzian contact stresses, this perfectly agrees with theory, as the contact stresses are dependent on the contact stiffness. For the root stresses, this is less evident, as the stress level in theory only depends on geometry and load. Nevertheless a decrease is seen for plastic gears, which is the subject of further investigation in the next sections.

Root Stresses. By using FEA, the root stresses of a single tooth can be monitored during meshing. Figure 5 shows the root stress as a function of the roll angle for a steel pinion meshing with gears of various materials. Similar regions can be recognized, as were seen in the load-sharing graph of figure.2.

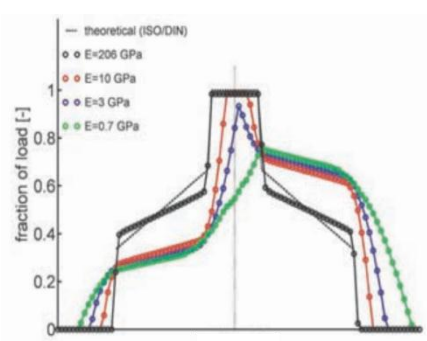

Figure 5—Root stress as a function of the roll angle for a steel pinion meshing with gears of various materials (moduli).

For a steel pinion and gear, the root stress is around 45 MPa in the first part of the meshing cycle. At the first moment of single-tooth contact, a jump to 74 MPa is seen. With increasing roll angle, the stress level lowers again as the contact point moves down along the tooth flank of the gear, reducing the lever arm. At the end of single-tooth contact, a large decrease is seen—to some 35 MPa—reducing even further towards the end of the meshing cycle.

As the load fraction carried by a single tooth determines the level of the root stress, the fact that the load sharing becomes skewed for plastic gears is expressed in the root stresses. The root stresses are considerably lower in the first stage of the meshing cycle compared to steel gears, and are considerably higher at the last part of the meshing cycle. Depending on the con- tact ratio and thus modulus, a period of single-tooth contact is achieved during which the root stresses become high. As can be seen in Figure 5, the level of root stresses reached in the single- tooth contact period follows the same trend as those found in steel gears.

However, due to the changes in load sharing, the time span of single-tooth contact becomes shorter, and the high stress level of the initial moment of contact for a steel gear is not reached. For an unfilled plastic gear at elevated temperature, the peak stresses around the pitch have completely disappeared, and the maximum values tend to go to a value of some 2/3 of the theoretical maximum value. It is expected that a further decrease of the modulus will not result in a further decrease of the root stress level.

An implication of this lowered root stress (compared to standardized calculations) is that the gears can be loaded to higher torques than expected based on the comparison of the theoretical root stress and yield stress of the material. This does not mean that the standardized approach is fundamentally wrong, but rather that it should be modified by incorporating the altered load sharing due to tooth bending.

Contact Stresses- The contact stresses occurring during meshing of two gear teeth can be estimated using Hertzian theory. The contact stresses can be expressed as a function of the load, contact stiffness, reduced radius of the contact and the width of the contact, according to:

$$\sigma H2=\sqrt{FEr/2\pi bRr} \qquad (1)$$

where F is the force (load), E_r reduced modulus or contact stifness (see equ 2), b is the width of the gear and R*r* is the reduced radius of the contact (see equ 3)

$$Er=\frac{1}{(1-Vg2)/Eg+(1-Vp2)/Ep} \qquad (2)$$

Where Eg & Rp are the young's modulus, V_g & V_p are the poisson's ratios of the gear and pinion, respectively,

$$Rr = \frac{1}{1+1} \qquad (3)$$

Where R_g & R_p are the local radii of the tooth flanks of the gear and pinion, respectively.

The FEA results of the contact stresses as a function of the roll angle are represented in Figure 6. The contact stresses for a steel pinion and gear, rep- resented by black circles, have a maxi- mum in the region of single-tooth con- tact (around 800 MPa). This agrees with the values that were calculated semi-analytically. Furthermore, the local radii of curvature and the load fraction deter- mined the course of the contact stress, as a function of roll angle.

According to Equations 1 and 2, the maximum contact stress occurring around the pitch point will reduce markedly with decreasing modulus. For a GF-reinforced plastic gear, the maxi- mum contact stress around the pitch point reduces from 800 MPa to 250 MPa. For an unfilled plastic gear, this reduces to 120 MPa, and at elevated temperature to 50 MPa. The values found using FEA correlate very well with the values using standardized calculations or KissSoft (Table 4).

However, as can be clearly seen in Figure 6, very high peaks appear at the beginning and end of the meshing cycle. Due to the tooth bending and the increased contact path, preliminary and prolonged contact occurs for plastic gears. At preliminary contact, the tip of the gear meshes with the root of the pinion, while at the prolonged contact, the tip of the steel pinion meshes with the root of the gear. This implies that it is not the involute parts of the flanks that are meshing, but that the tip of the tooth, which is determined by tip radius (tip radius equals 0.1 mm in the calculations), is in contact with the other tooth flank. For a GF-reinforced plastic gear (red circles), the contact of the sharp tip radius of the steel pinion and the tooth root of the gear (end of mesh- ing cycle) results in a contact stress of 940 MPa, which exceeds the values found for the steel pinion and gear. For an unfilled plastic gear, the value of the contact stress reaches 500 MPa at room temperature and 220 MPa at elevated temperature.

Using Equation 1 and taking the load sharing into account, the contact stress at the prolonged contact can be estimated using Equation 4.

$$\sigma H_2 = \sqrt{\propto IsFEr/2bRtip} \qquad (4)$$

Where αls the load sharing factor and Rtip is the tip radius. Figure 2 shows that the load share carried by a single tooth at the moment the maximum contact is observed is around 50% and hence αls is chosen to be equal to 0.5.The maximum values found by FEA and those obtained using Equation 4 compare very well (Table 4, Column 3). This equation can be used to estimate contact stresses, based on geometry and loading, without going through the elaborate process of FEA.

Pressure-Velocity Values The large difference in theoretical contact stress and maximum contact stress due to the extended contact path has significant implications. In the conventional approach, the maximum contact stress occurs around the pitch point, at which the sliding velocity is low or even zero. The maximum presuure-velocity (PV) value, crucial in wear prediction, was found at the beginning and end of the meshing cycle, combined with a larger sliding velocity (upto 20%), PV values are found which are much higher than calculated based on conventional theory.

Figure 7 shows the PVvalues for a steel pinion meshing with a gear of various materials. For a GF-reinforced gear, the PV value is expected to drop by a factor of 4; instead, a value is found which exceeds the PV value of a steel-on-steel combination by 30%. For the unfilled materials, the PV values are about a factor of seven times higher than those calculated by conventional theory. The PV values calculated by conventional theory and FEA are summarized in Table 5.

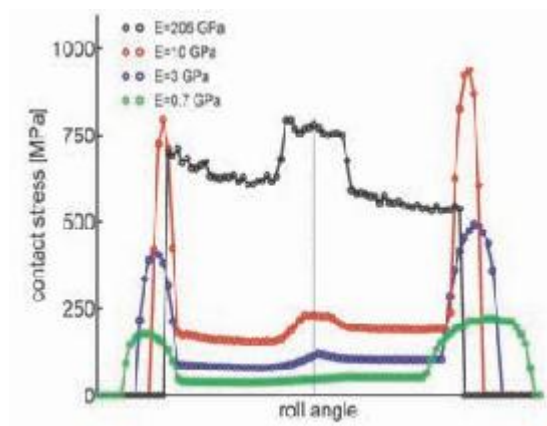

Figure 6—(Hertzian) contact stress as a function of the roll angle for a steel pinion meshing with a gear of various materials.

Table 4—Contact stresses determined by various methods.

Modulus	KissSoft/Analytical	FEA, Maximum
GPa	10^5 MPa.mm/s	
206	8.7	8.7
10	2.1	11.8
3	1.1	6.6
0.7	0.55	3.5

It must be noted that the calculations are performed using a tip-rounding radius of 0.1mm. If a larger tip radius is used, contact stresses will drop according to equation 3. In a steel-plastic combination, the peak in contact streeses and PV value at the beginning of the meshing cycle is less important than the peaks at the end. At the beginning of the meshing cycle, the tip of the poastic gear is in contact with the root of the steel gear. After running in, the tip of the plastic gear will be worn in

such a way that its contact radius has become larger and the contact stresses have reduced. Apart from a lowered contact ratio, this will have no significant negative side effects.

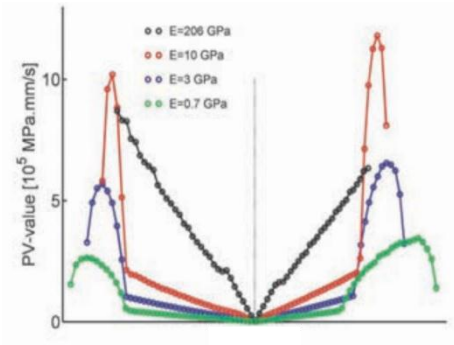

Figure 7—PV values found for a steel pinion meshing with gears of various materials.

Table 5—PV values determined analytically and by FEA.

Modulus	KissSoft/a na-	FEA, Around	FEA, Maxim	Analytic al
GPa	MP			
206	686	800	N/	N/
10	217	230	940	983
3	121	120	495	548
0.7	60	50	220	273

Implications of Tooth Bending Wear. It has been shown that due to tooth bending, many secondary effects occur in plastic gearing. Apart from an increasing contact and lowering of the root stresses, the effects on the kinematics and contact stresses are negative. As shown, the PV values can become much higher due to the extended contact path, which puts the bad correlation between tribological tests (e.g., pin-on-disk measurements) and experimental wear data on gears into a completely different context.

A closer look at Figure 4 reveals that the extension of the contact path is perpendicular to the contact line and the rest of the contact path. Combined with the bending of the teeth, the question arises whether the involute flanks are still meshing properly. Therefore the kinematics of the meshing teeth were investigated in more detail. Figure8 shows four subsequent images of the gear flank on four stages during the meshing cycle.

Figure 8a shows that, at the initial contact stage, the tip of the gear flank is in contact with the root of the pinion. Although not visible in this picture, the tip (and thus tip rounding) remain in contact for quite awhile, meaning that the rolling component of the velocity is very low during this stage. Therefore this stage can be regarded as if a plastic pin (or cylinder) is sliding on a steel surface.

With increasing roll angle, the con- tact point slides along the flank, as is expected based on theory. The involute flanks mesh properly and the contact point moves with fluid motion from the tip to the root of the gear (Fig. 8b).

Figure 8c visualizes the last moment of the meshing cycle when the contact point has moved along the contact line and reached the root of the gear. The contact point then enters the stage of prolonged contact.

Figure 8d reveals what is actually happening in the extension of the contact path the contact point is moving back along the flank towards the pitch point. Hence in the prolonged and also preliminary contact, a reciprocating movement takes place at the root of the gear with a high sliding velocity, and very high contact pressure (and high PV).

In the case of a steel pinion meshing with a plastic gear, it is as if a sharp cylindrical, steel tip (radius of curvature equals the tip-rounding radius) is digging into the root of the gear. Depending on the length of the contact path (and thus tooth bending and Young's modulus), the end of contact approaches the pitch point.

Our hypothesis now is that this high-PV, reciprocating movement at the beginning and end of the contact is the governing mechanism of wear in plastic gears.

To substantiate this hypothesis, the results of experimental studies were consulted. The first results investigated are experimental studies conducted by Walton and Weale at the University of Birmingham, in which they performed experiments on polyacetal (POM) gears to investigate their wear behavior (Ref.17). Although our simulations are done on steel-plastic combinations, the same phenomena will occur in plastic gear pairs.

At the tooth tip, indicated by 'a' in Figure 9, high wear is observed. This can be explained by the fact that the tip of the gear is coming into preliminary contact with the root of the pinion, experiencing a high PV (Fig. 7; i.e., negative roll angles, left-hand side).

T h e addend u m s h o w s a much smoother wear pattern, indicated by 'b' in Figure 9. In light of the simulations, in which the involute tooth flanks of the addendum are meshing properly (although still at a considerable PV), this is also in agreement.

In the region indicated by 'd,' the most severe wear occurs, and the surface shows surface pits and scoops. This is the region in which the high PV, reciprocating movement takes place, and indeed the most severe wear is expected in this area.

The last striking feature is the formation of a ridge near the pitch point. This can be explained by the fact that, due to the reciprocating movement, the tip of the pinion returns at a high PV towards the pitch point. The high contact pressure digs out and deforms material, and pushes it toward the pitch point. At the end of the meshing cycle the pressure reduces,

leaving the deformed material on a ridge. The severe effects seen in Figure 9 are the result of many repeated cycles.

If the mechanism described above is the governing mechanism in the wear of plastic gears, it would imply that the wear scars on the pinion (driver gear) and driven gear must be different. The reason for this is that, for the driven gear, the contact point moves from the tip to the root and then moves back towards the pitch point, as shown in Figure 8. For the drive gear, the opposite is true; the tip of the gear contacts the pinion near the pitch point and moves towards the root of the pinion, pushing the material down. The direction of the movement changes, and the contact point moves along the contact line from the root to the tip of the pin-ion. The experimental study on polyacetal gears (Ref. 17) shows that the wear patterns at the dedendum of the pinion (drive gear) are very different and that a valley or depression without ridges is seen near the pitch point.

Breeds, et al., (Ref. 17) explained the observed differences by the fact that the surface stresses resulting from the friction forces are compressive for the driven gear, while the driver gear experiences tensile stresses. The differences in wear of the addendum and dedendum were explained by the fact that the directions of the rolling and sliding velocities are different above and below the pitch point. Thus the rolling component of the tip of the gears in the preliminary and prolonged contact is only of minor influence.

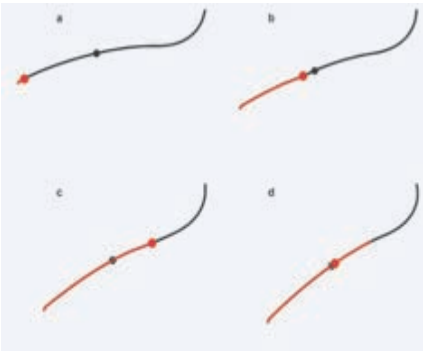

Figure 8—Kinematics of two meshing tooth flanks (only gear flank is drawn). The solid black line represents the tooth flank of the gear, the black dot represents the pitch point, the red dot rep- resents the moving contact point and the solid red line represents the part of the flank that has already been in contact.

Further support for our hypothesis is found in an experimental study of White, et al. (Ref. 18), in which they investigated the effect of tip relief on wear in plastic spur gears. Their general observation is that tip relief has very beneficial effects on the wear performance of plastic gears and that doubling the amount of required tip relief (based on calculations) gives the lowest wear rates. They state further that the shape of tip relief is important, as it determines the level of contact stress. These observations complement our simulations, as tip relief reduces the effects of preliminary and prolonged contact. Moreover, the level of contact stress can also be considerably lowered. For example, applying a tip rounding radius of 0.3 mm instead of 0.1 mm reduces the PV value by 30%.

4. Conclusions

It has been shown that the load sharing of a steel-plastic gear pair changes dramatically compared to the conventional theory of steel gears. Not only does the length of the contact path increase, so also does the shape of load sharing change due to significant tooth bending. For steel-to-steel and plastic-to-plastic gear pairs, the load sharing is symmetrical around the pitch, while for a steel-plastic gear pair it becomes skewed, and the teeth face the most severe loading in the last part of the meshing cycle (roughly 2/3 of the load). It was also shown that the extension of the contact path lies perpendicular to the theoretical contact line of involute gearing.

The changes in load sharing also change the stresses. Although the root (bending) stresses are only dependent on load and geometry, due to the changes in load sharing, the bending stresses decrease for plastic gears. Here a plateau value of roughly 2/3 of the theoretical root stress is predicted. With decreasing modulus the contact stresses around the pitch point become lower. But the extended contact path tip of one gear comes into contact with the other gear, resulting in very high con- tact stress peaks. As a result the PV value can be up to seven times higher than predicted in theory.

In the preliminary and prolonged contact, the involute tooth flanks do not mesh properly, but the tooth tips make a reciprocating movement on the root of the other tooth. We state that this high-PV, reciprocating movement of the tooth tip is the governing mechanism of wear in plastic gears. Experimental observations in the literature substantiate our hypothesis.

5. References

1. Klein, G. "Untersuchungen zur Tragfähigkeit thermoplastischer Kunststoffzahnräder." Konstruktion 20 (1968) pp. 385–390.

2. Cornelius, E.A. and W. Budich. "Untersuchungen an Zahnrädern aus Acetalharzen Konstruktion 22 (1970) pp. 103–116.

3. Budich, W. "Ergebnisse von Versuchen an Geschmierten Thermoplastischen Zahnrädern. Konstruktion 22 (1970) pp.96–401.

4. Beitz, W. and E. Siedke. "Untersuchungen an Zahnrädern aus Polyamid 12." Kunststoffe 62 (1972) pp. 390-393.

5 Severin, D. and H. Lütkebohle. "Wälzreibung zylindrischer Räder aus Kunststoff." Konstruktion 38 (1986) pp. 173–179.

6. Severin, D. and B. Kühlken. "Tragfähigkeit von Kunststoffrädern unter Berücksichtigung der Eigenerwärmung." Konstruktion 43 (1991) pp. 153 160.

7. Baumgart, J. "Tragfähigkeit modifizierter Kunststoff/Stahl-Zahnradpaarungen." Konstruktion 43 (1991) pp. 378 384

8. Chen, Q.T. "Flankenbeanspruchung bei Stirnzahnrädern aus Kunststoffen." Konstruktion 43 (1991) pp. 225 232

9. Homann, H. VDI, Abteilung Richtlinien. Personal Communication, October 2004.

10. Yelle, H. and D.J. Burns. "Calculation of Contact Ratios for Plastic/Plastic and Plastic/Steel Spur Gear Pairs." Journal of Mechanical Design 103 (1981) pp. 528–542.

11. Walton, D., A.A. Tessema, C.J. Hooke and J.M. Shippen. "Load Sharing in Metallic and Non-Metallic Gears." Proc. Inst. Mech. E, Part C: Journal of Mechanical Engineering Science 208 (1994) pp. 81–87.

12. Arafa, M.H. and M.M. Megahed. "Evaluation of Spur Gear Mesh Compliance Using Finite Element Method." Proc. Inst. Mech. E, Part C: Journal of Mechanical Engineering Science 213 (1999) pp. 569–579.

13. Tsai, M. H. and Y.C. Tsai. "A Method for Calculating Static Transmission Errors of Plastic Spur Gears Using FEM Evaluation." Finite Elements in Analysis and Design 27 (1999) pp. 345–357.

14. Litvin, F.L., L. Qiming and A.L. Kapelevich. "Asymmetric Modified Spur Gear Drives: Reduction of Noise Localization of Contact, Simulation of Meshing and Stress Analysis."

15. Comput. Methods Appl. Mech. Engrg. 188 (2000) pp. 363–390.

16. Walton, D., A.A. Tessema, C.J. Hooke and J.M. Shippen. "A Note on Tip Relief and Backlash Allowances in Non-Metallic Gears." Proc. Inst. Mech. E, Part C: Journal of Mechanical Engineering Science 209 (1994) pp.383–388

17. Mao, K., C.J. Hooke and D.Walton. "The Wear Behavior of Polymer Composite Gears." Synthetic Lubrication 12 (1996) pp. 337–342.

18. Breeds, A.R., S.N. Kukureka, K. Mao, D. Walton and C.J. Hooke. "Wear behavior of Acetal Gear Pairs" Wear 166 (1993) pp. 85–91.

19. White, J., D. Walton and D.J. Weale. "The Beneficial Effect of Tip Relief On Plastic Spur Gears." ANTEC Conf. Proc. Society Plastics Engineers 3 (1998) pp. 3013–3107