Design Investigations and Indications for Acoustical Optimized Gear Meshes Using Plastic Gears

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Introduction

When it comes to a steel-gear mesh, there are several common standards and design rules on how to reduce noise emissions in the mesh. But if plastic gears are involved, this is no longer the case. The topic of this presentation is to highlight some of the differences between metal-and-plastic gear meshes, i.e. — which design strategies can be stated as valid for metal as well as plastic — and which are not? This should lead to some basic hints to what a noise-optimized toothing should look like, but also to what might be deleterious effects on other important features, e.g. — strength. The differences will be shown by examples of FEA results demonstrating effects of the lower Young's Modulus and others on specific design strategies. It will also be shown that due to the lower stiffness, some design rules — like aiming for integer numbers of overlap ratios, as it is known, for steel meshes is still not wrong — but the allowable spread around the integer overlap is much higher for yet minor noise emissions. This will be shown with test results as well as FEA. Further, other interesting topics are investigated physically, such as the effects of different tip modifications, etc.

When it comes to noise generation and emission due to gearing movement, the root causes of these generally can be stated as:

- Stiffness variation during the tooth contact
- Tooth meshing impact
- Sliding effects (effects of roughness and relative sliding)
- Geometric errors, such as runout, etc.

Once the structure-borne sound is generated, it has to be transferred through the parts to finally find a surface where it is transferred to airborne sound. Therefore, the damping effects, as well as acoustic impedances of the materials, are main influencing topics.

Regarding different materials in a geartrain, it can be said that there are many investigations regarding the material combination of steel-steel. However, plastic sometimes behaves very differently when it comes to the noise effects of some specific design issues, such as tip modifications and transverse and overlap contact ratios. Calculations and tests indicate that the root causes are still the same, but the specific behavior sometimes changes dramatically, mainly because of the higher deformations caused by lower Young's Module of the plastic material.

Because of the higher deformation of the gear mesh, contact and overlap ratios change dramatically when loads are applied; therefore the optimum of the theoretical, non-deformed calculated contact ratios shifts while under load. It can be seen that there is not a specific optimum at all load conditions if deformation is taken into account.

Also, there are some main effects not driven by the higher deformation itself, but instead the difference between the deformations between both gears if different materials are used. For example, regarding a geartrain where a metal gear is paired with a plastic gear, stiffness modulations can be stated higher, in general — as if the material combination were equal. Even if it is only different plastics that are used, the difference can be particularly high when reinforced and non-reinforced materials are combined. Especially when the plastic gear is the driver, big tip modifications are essential to avoid high pressure and acoustic problems. For an acoustically optimized gear there must also be taken into account some specific plastic behaviors regarding toothing errors. To gain more data and knowledge about the deformation-influenced behavior, several tests were performed in-house.

A Closer Look at Spring Stiffness and Gear Mesh Behavior

In principle, the transmission of circular motion in a perfectly shaped, ideal stiff gear mesh would be perfectly steady. In practice, in a real gear mesh the transmission is not perfectly steady (Refs. 1 and 5). This is caused by changes to the stiffness of the mesh at different meshing positions, as well as other effects like tooth meshing impact and inaccuracies caused by manufacturing [1, 1].

Therefore, the stiffness variability of the gear mesh is an important factor of the noise generation. Because of the different length of the lever arms of a tooth along the path of contact, as well as of other effects like transverse contact ratios (different number of teeth in the mesh at different positions), the stiffness can vary. As stated by FVA investigations regarding steel, an axial overlap ratio given as an integer therefore leads to minimum structure-borne sound emissions [2, 132]. An explanation of this effect is that for an even overlap ratio, every point at the path of contact is utilized as a contact point in a specific transverse section at any angular position of the gear mesh. Put simply, no differences in the stiffness situation occur overall, because at the same time both the less stiff and

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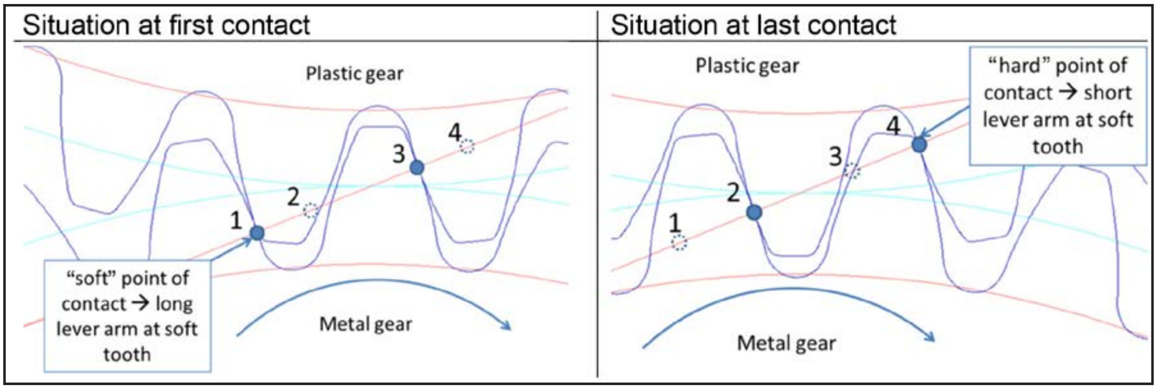

Figure 1 Metal gear as driving gear.

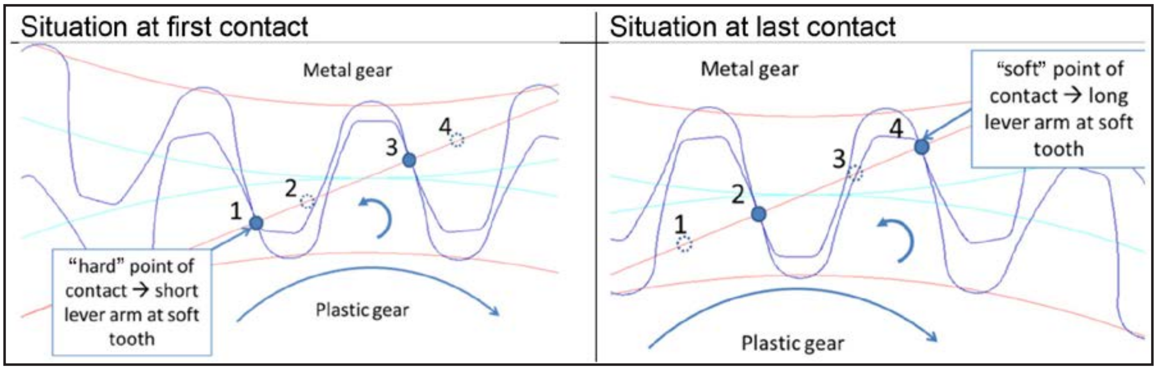

Figure 2 Plastic gear as driving gear.

more stiff contact positions are engaged. This is essentially for a steady transmission; as can be seen (Fig. 1), the lever arm at position 1 for the plastic gear is much longer than at position 4. This means that the stiffness decreases for contact at pos. 1 and increases until the end of contact at pos. 4—if just one pair of teeth should be in contact. In fact, it must be taken into account that the change of the number of teeth in contact means that positions with two teeth in contact are stiffer than positions with one pair of teeth in contact. Therefore the contact situation slightly after point 2 is the softest position, because one tooth is in contact and the soft plastic gear has the high leverage arm. It can be generally stated that the changes in the stiffness are much higher in the metal-and-plastic combination, because in a geartrain there is always a long bending lever arm at one tooth paired with a short one at the opposite. When the same materials are used, this to some degree averages the effects of changing leverage arms. When it comes to an application with paired metal-and-plastic because of the high difference in the Young's Modulus, the metal tooth almost bends not compared to the plastic tooth and because of that the averaging effect does virtually not exist.

These above-mentioned effects—seen also in Figure 3—were different material combinations we compared. Here the geometry of the spur-toothed test gear was the base that was modified for the tooth thickness correction as well as for the high-toothed gear. It can be seen that from this point of view, the variations are lowest for the stiffer materials. But, damping here is not recommended, and therefore structure-borne sound emissions for the high stiffness combinations—like steel-steel—are usually worse if the same degree

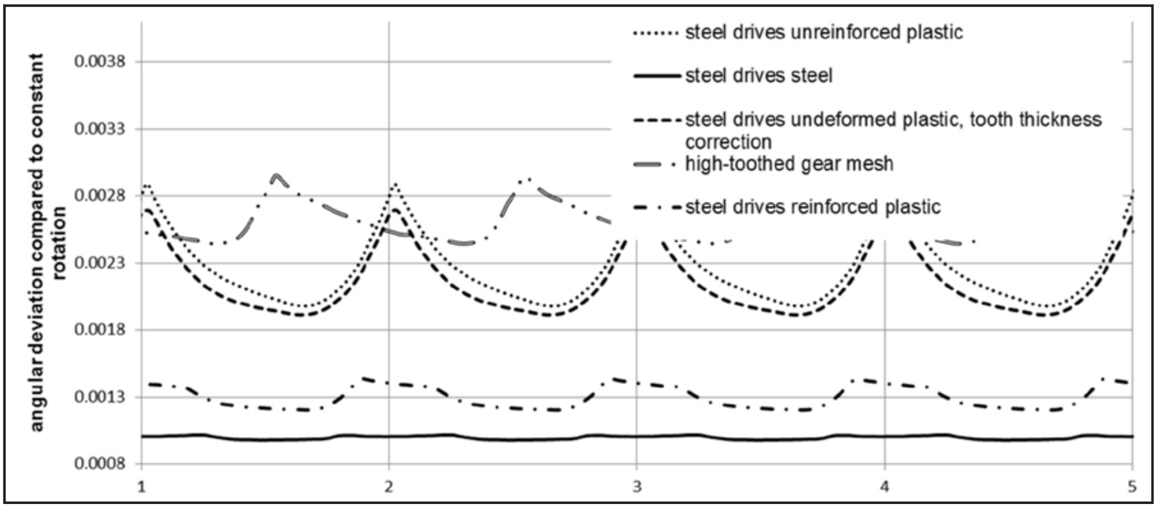

Figure 3 Angular deviations compared to constant rotation for different material combinations and toothings.

of quality level is taken into account.

As can be seen (Figs. 1–2), there is also a significant, principal difference if the metal part is the driving gear or the driver. In a situation where metal is the driving gear, shortly before the next pair of teeth will engage (point 1), the gear mesh is not so soft, because at point 3 there is a short leverage at the soft tooth. Conversely, when plastic is the driver, shortly before the next pair of teeth makes contact there is a low stiffness and high bending in the mesh. That's because here, point 3 is a point of really low stiffness/high leverage at the soft tooth). Therefore for a geartrain in which plastic is the driver, the impact when the next teeth are coming into contact is much higher than when metal is the driver. In this case a tip modification at the metal gear has to be defined relatively large and carefully—to avoid both excessive noise generation and pressure-related failure modes.

Therefore some practical tests were performed regarding the following points:

- Effects of different materials
- Effects of tip modifications
- Effects of changed overlap ratios and loads

Test Rig and Test Gears

For the tests a given test rig was used with given IMS-test gears, but also with specially modified test gears, i.e.—injection-molded parts as well as machined parts were also tested. The injection-molded parts have the benefit of being very close to serial-like production, whereas the machined parts have the benefit of making different geometries possible without too much effort. At the same time, the influence of toothing errors is not so high for the machined parts, since the achievable accuracy is much higher. To avoid misinterpretations, there was also a comparison between machined and injection-molded parts with the same geometries.

Tables 1 and 2 show some of the geometric data of the test gears. For the helical gears, different axial overlap ratios were reached by changing both the tooth width and the helix angle. The examples shown later have changed overlap ratios by modification of the helix angle.

The used test rig is, as shown (Fig. 4), a test setting with three gears in a mesh, whereas the plastic gear represents the intermediate wheel. The powertrain consists of an electric motor (servo synchronous motor), which transfers the power to the input side of the gearbox. A planetary gear transmits the torque. An incremental rotary encoder detects the rotation speed and angle. On the output side of the gearbox you can find a magnetic-powder brake, which applies the load; a torque gauge measures the torque (Ref. 1).

Table 1 Geometry data. spur toothed test gear (Ref. 1)

	Plastic gear	Metal gear
Number of teeth [-]	38	39
Normal module [mm]	1.46	
Normal pressure angle [mm]	20	
Tooth width [mm]	11.5	11.5

Table 2 Geometry data. helical toothed test gear (Ref. 1)

	Plastic gear	Metal gear
Number of teeth [-]	36	36
Normal module [mm]	1.46	
Normal pressure angle [mm]	20	
Helix angle [mm]	23	
Tooth width [mm]	11.5	11.5

The software of the modified wear-test stand allows an operation on a defined rotation speed and torque with rotation speeds up to 300 rev/min used. For the acoustic tests, additionally a ramp-up of the speed was implemented to start the measurement at zero speed, perform a ramp up and then measure also at a steady state speed (Ref. 1).

Acceleration sensors of a mobile acoustic measurement station register the structure-borne sound at significant places. The incremental rotary encoder delivers the necessary speed information needed for the analysis. The analysis leads to a statement about the noise generation caused by the gear in case of different geometric and system parameters. It is also a statement about the equality of the movement translation given.

The test assembly includes a multitude of noise-generating machine elements. A frequency analysis is carried out to the time signal of the acceleration sensors to differ the part of the signal that is really generated by the tested gear set.

Because the additional effort was marginal, sensors are placed at several locations. The best results have been delivered by the bearing point sensor; it's the closest possible point to the test gearing (sensor position shown in Fig. 4).

As shown in the subsequent figures, the toothing entering frequency is clearly visible, although the gear box is really solid. The frequency is also not in the range of interference frequencies caused by test stand components (for example the planetary gear).

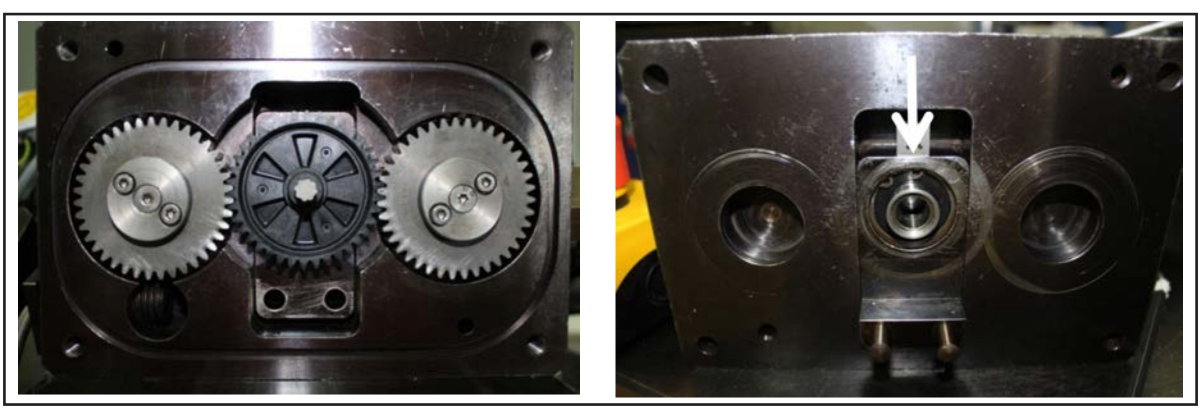

Figure 4 Test setup (left) and position of the solid borne sound sensor (Y-direction) (Ref.).

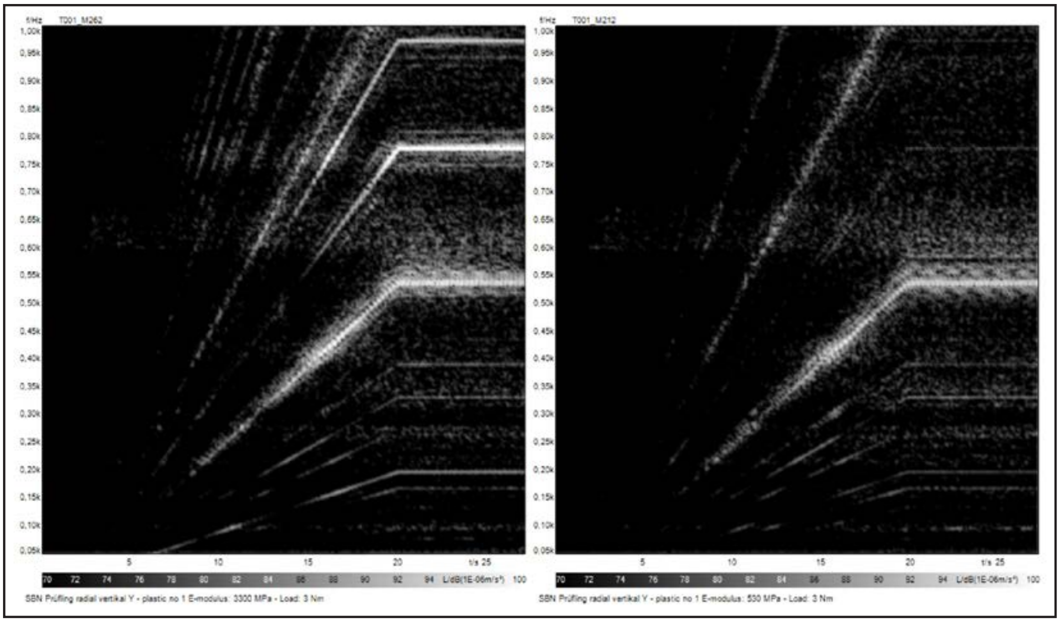

Figure 5 Plastic with higher Young's Modulus (left, 3,300 MPa) vs. plastic with low Young's Modulus (right, 530 MPa) in a steel-plastic-steel gear mesh.

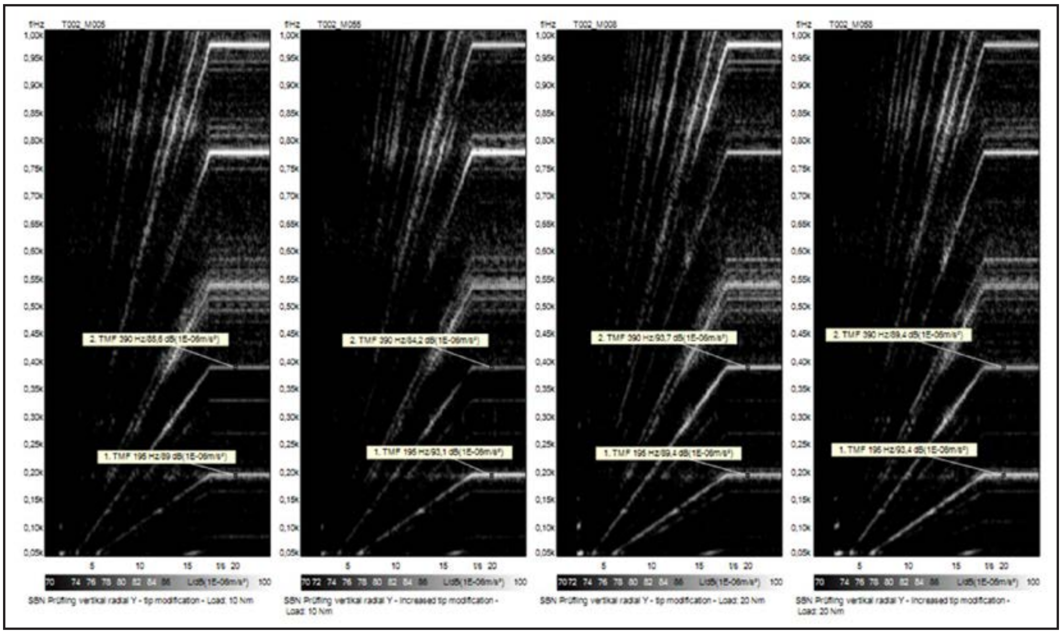

Figure 6 Comparison of parts with smaller modification (chart 1 and 3 from left) and bigger tip modification (chart 2 and 4 from left) at different torques (left: 10 Nm, right: 20 Nm).

Testing Results and Comparison with FEA

All the shown measurements were done as a speed-up test and then, afterwards ran with a constant frequency for some seconds. Here it can be measured the intensity of the structure borne noise at the meshing frequency and also their harmonics.

As shown (Fig. 5), two non-reinforced plastic materials were tested and shown as FFT versus time in a Campbell diagram. At the left side, a material with a Young's Modulus of app. 3,300 MPa was tested, at the right side a material with app. 530 MPa. The load was relatively low with 3 Nm. As can be seen, the less stiffer material didn't create so much noise. Additionally, for the low stiffness material, the missing high sound levels at the harmonic frequencies are noticeable compared to the stiffer material. This can probably be caused by better damping and a less stiff impact, which generally leads to less acoustic excitation of the higher harmonic frequencies.

As shown, there were also some tests performed with different tooth tip modifications. The result was that for both types of modification the torque level had just a small impact on the sound generation at tooth meshing frequency, whereas the 1st harmonic was strongly affected by the torque level. Generally, the smaller modification generated less noise at tooth meshing frequency but more noise at the 1st harmonic. A possible explanation of this behavior is that the modification was made at the plastic gear tooth tip. Therefore, it has especially a benefit at the gear mesh, were steel is the driver (plastic tip is in touch for the tooth meshing impact), which is even more a benefit at higher loads and therefore higher deformations. Since the test setting is a three gear mesh, the plastic gear at the same time is the driver for the output steel gear. In this stage, the modification is not a benefit, but a disadvantage, since here at first the tip of the steel gear meets with the root of the plastic gear, causing the consequences written in the paragraph above. Here, the stiffness right before the tip impact is reduced and therefore the impact is more energetic. Generally, at this mesh the tip modification is causing smaller transverse contact ratios. The hypothesis is that at this specific gear set, the mesh with the output metal gear is causing more noise because of the worse tip impact situation. Therefore the modification of the plastic tip causes more noise at this frequency. The input metal gear mesh with the plastic gear is softer and has the lower impact stiffness, causing less noise. The influence of this stage can be seen more at the 1st harmonic, since the tip impacts of both stages don't happen at the same time (there is a slightly triangular setup of the gears). Therefore the 1st harmonic is positive influenced by the tip modification with increasing benefit for higher torques. However, regarding this topic, there have to be more tests done for further investigation.

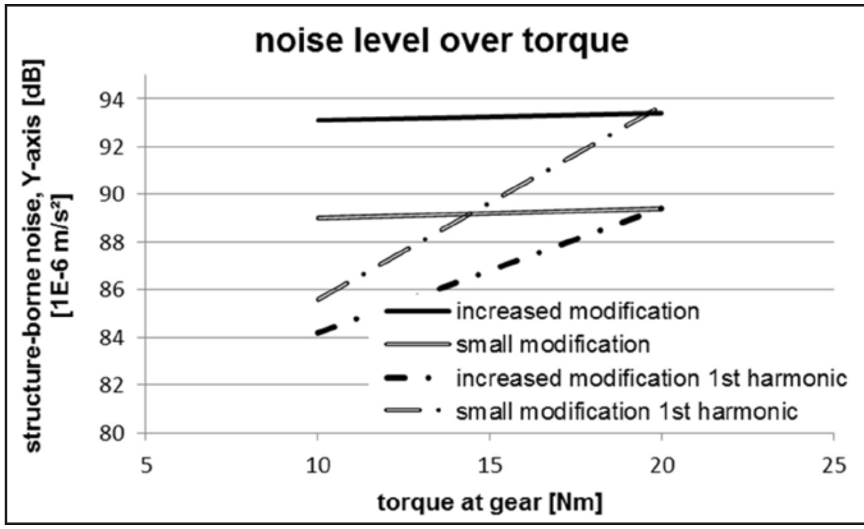

Figure 7 Results of Figure 6 as chart of the structure-borne sound of the tooth meshing frequency and the 1st harmonic.

The above Figures 8, 9 and 10 show the results of some tests as well as the FEA results regarding this topic. As it can be seen, for the parts with the overlap ratio of 1 (theoretically, un-deformed) the noise level is getting higher with higher torque levels. For the 0.8 overlap ratio parts, there is just a very minor increase until mid-level torques and then even a small drop. If you compare this results with the additionally performed FEA, it can be shown that overlap ratios for the parts starting at an un-deformed ratio of 0.8 increase up to 1.8 if deformation is taken into account. If you think of the

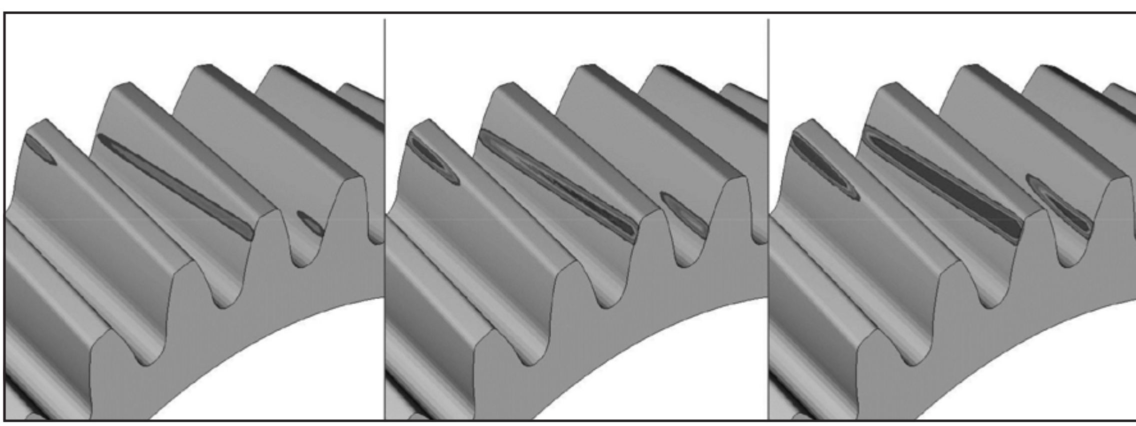

Figure 8 FEA results at different torque levels showing the deformation-induced increase of axial overlap ratio from 0.8 (un-deformed up to 1.8 at max. load).

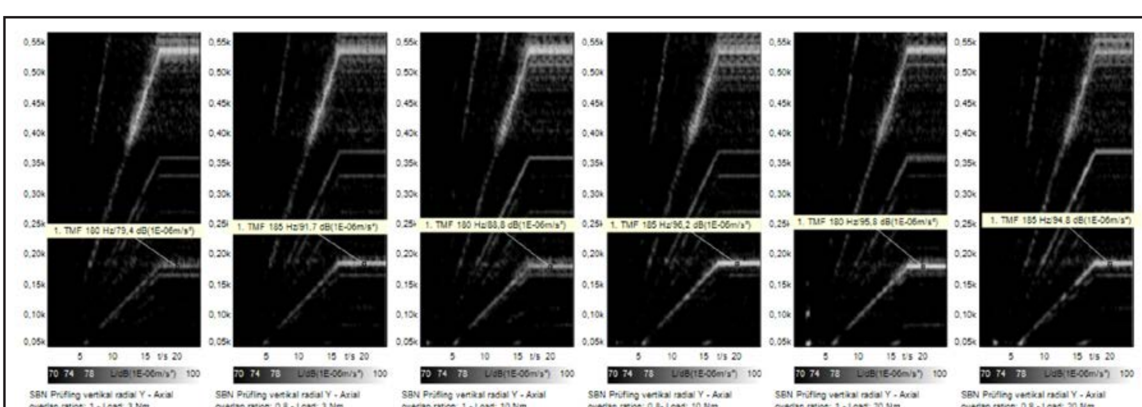

Figure 9 Comparison of parts with un-deformed overlap ratio 1 (chart 1, 3, 5 from left) un-deformed overlap ratio 0.8 (chart 2, 4, 6 from left) at different torques (from left to right: 3 Nm, 10 Nm, 20 Nm).

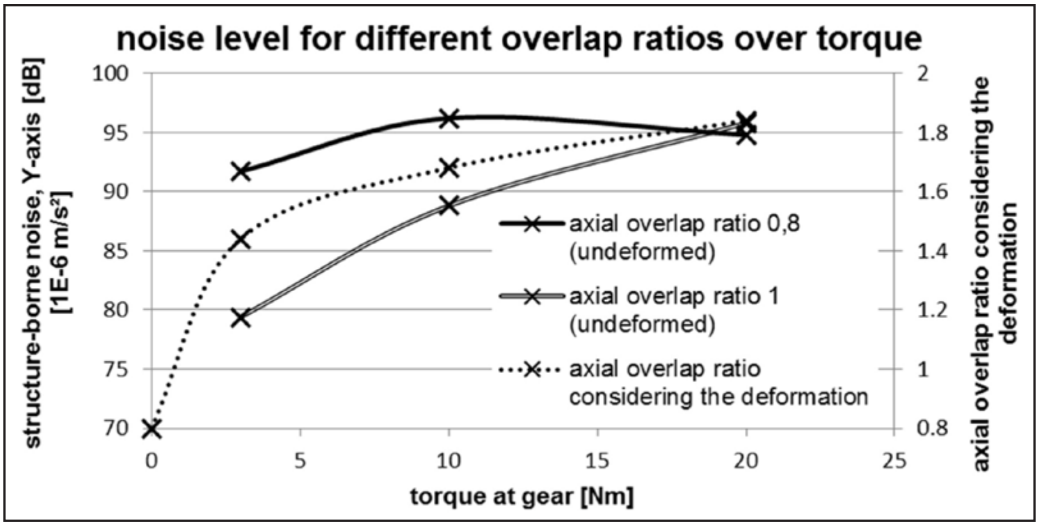

Figure 10 Results of Figure 9 as chart of the structure-borne sound of the tooth meshing frequency over the torque combined with the FEA results for axial overlap ratios considering the deformation.

parts starting with an overlap ratio of 0.8, this means that they were pushed to overlap levels which can be theoretically seen as optimum regarding stiffness variations of the mesh. The parts starting with an un-deformed overlap ratio of 1 will be pushed away from the optimum. As mentioned before, for steel, the optimum would be an axial overlap ratio given as an integer. However, for the plastic gears, both the measurement as well as the calculations show that you have to take into account the deformation-driven change of the overlap ratio to find your optimum. It is likely that the optimum will not be exactly at the even number for the deformed axial overlap ratio but near this number. The reason for this might be that the prolongation of the path of contact happens mainly at the start and the end of it. Therefore in this situation some positions of the teeth representing spring stiffness will be more often in contact or longer in contact than others.

Conclusion

If it comes to a gear mesh which includes plastic, deformation effects have to be taken into account carefully. Some indications regarding the design for an acoustical optimized gear train which are valid for steel can change apparently a lot if you try to transfer them to a gear mesh including plastic. However, at least some of the observed differences are no longer principle ones if you take into account the deformation while checking basic parameters of the mesh like overlap ratios. Therefore there are some basic indications:

- Generally take into account the high deformations at the teeth while evaluating the mesh
- For a plastic — metal or plastic — plastic mesh the deformation under load has to be taken into account while defining optimized tip modifications.
- Under load transverse contact ratios tend to increase. This effect gets bigger when it comes to softer materials, higher loads or higher temperatures.
- Because of this, the theoretical optimized overlap ratio for the un-deformed mesh has to be lower for optimizations regarding higher loads, temperatures or softer materials.
- For plastic, the even number is not always the best value to start with for an un-deformed overlap ratio, since ratios increase under working conditions.
- Plastic materials with smaller Young's Modulus may have better damping, but deformation could be a problem
- Especially for driving/driven situations for a metal gear mesh (high stiffness variation) balancing out the tip modifications is crucial **PTE**

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