Improving Power-Density and Efficiency of Plastic Gears

Applying non-standard design techniques to plastic gear geometry will significantly improve power density and efficiency

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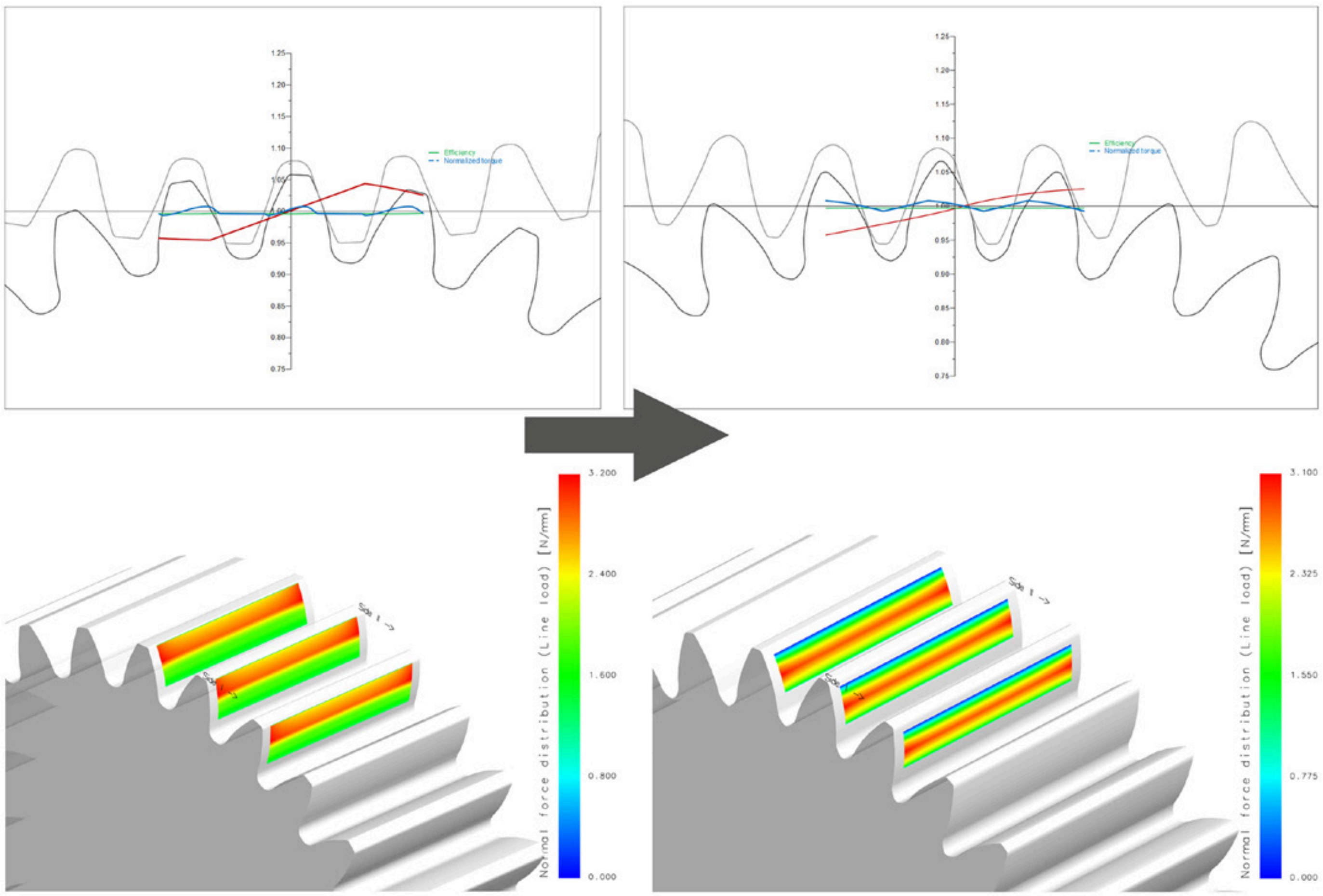

Contact pattern optimization and smoother meshing with the* KISSsoft *Contact Analysis tool.

It has been known that plastic gears have seen a rise in demand due to their inherent quietness, ability to run without lubricants, and low cost in high volume production. Due to the injection molding process, complex shapes and geometries can also be integrated into a single part, and the tooth form can be optimized free of traditional metal gear manufacturing process limitations.

Standard Tooth Form Design

Initial gear design was done on *KISSsoft* gear design software, using the Macro Geometry Rough Sizing Module. A 3.13 to 1 ratio was roughly sized using the VDI 2736:2013-modified (YF Method B) calculation method, considering the selected materials, required loads, and required minimum safety factors (amongst other considerations).

Table 1 details the initial design made for this purpose. This design uses a standard tooth profile, which will later be compared to an optimized plastic gear design that does not use a standard tooth form. Other geometry considerations, such as a minimum tooth tip thickness, tolerances, and tip rounding based on the EDM wire diameter, were also considered. Gleason Plastic Gears assumed an ISO 1328:2013 A8 Gear Quality using their proprietary No-Weldline Technology. Table 1 summarizes the gear pair data.

Figure 1 shows the meshing between the initial design of the polymer gear and pinion under load. This gives us a starting point only, with plenty of room for optimization. The dotted line represents the contact path with sharp transitions during tooth engagement and disengagement. This represents shocks that occur during the functioning of the gear pair due to high tooth deflection and unmodified profiles.

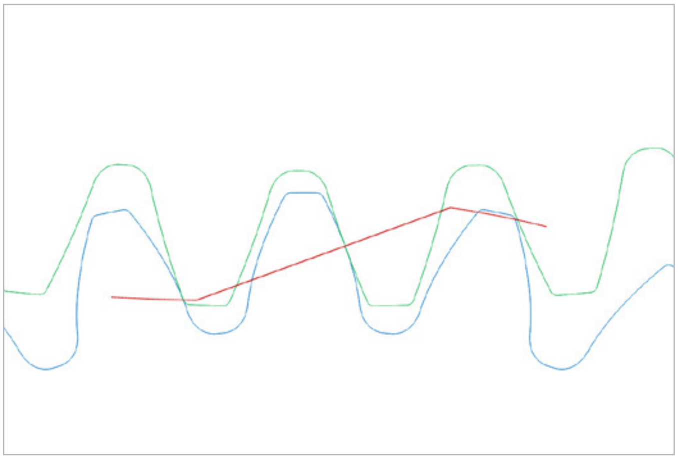

Figure 1—Gear meshing under load, along with the path of contact, shows some contact shocks and higher transmission errors when the gear pair rotates.

Gear data		Gear 1	Gear 2
Number of teeth		31	97
Normal module	mm	0.5	0.5
Facewidth	mm	5	4.5
Normal pressure angle	°	20	20
Addendum coefficient		1	1
Root radius coefficient		0.38	0.38
Dedendum coefficient		1.25	1.25
Profile shift coefficient		0.1946	-0.1946
Reference diameter	mm	15.5	48.5
Base diameter	mm	14.565	45.575
Tip diameter	mm	16.694	49.305
Root diameter	mm	14.444	47.055
Tip form diameter	mm	16.628	49.223
Root form diameter	mm	14.860	47.419
Tooth thickness tolerance	mm	-0.0381/-0.0508	-0.0381/ -0.0508
Quality (ISO 1328:2013)		8	8
Materials		POM	PA 66

Table 1—Gear data of the initial design.

Gear Stress Calculation

The initial design was sized based on the required load and minimum required safety factor under a given temperature using the VDI 2736 guideline and the Macro Geometry Rough Sizing Module in *KISSsoft* 2024. The calculation uses the input geometry and the material properties to determine the load-bearing capacity for a certain safety factor to be achieved on the gear pair. According to VDI 2736, the tooth root stress is calculated as:

$$\sigma_F = K_F \cdot Y_F \cdot Y_S \cdot Y_\varepsilon \cdot Y_\beta \frac{F_t}{b \cdot m_n} \leq \sigma_{FP} = \frac{Y_{St} \cdot \sigma_{FlimN}}{S_{Fmin}}$$

Investigation of the initial design yielded the following results:

Results	Gear 1	Gear 2
Root safety	2.093	1.493
Flank safety	1.832	2.283

Table 2—Root and flank safety as calculated according to VDI 2736 by KISSsoft.

The safety factors meet our minimum requirements, and now we also have some idea of the approximate center distance and face width needed to achieve the ratio and torque transfer required.

After further macro- and microgeometry optimization of the tooth form in later steps, the root stresses will be calculated and compared using the following methods:

- Analytical Method (Stress as calculated by VDI 2736)
- Loaded Tooth Contact Analysis (LTCA)
- FEM

Macrogeometry Optimization

Applying *KISSsoft*'s Macro Geometry Fine Sizing Module, the gear tooth profiles can be further optimized. Parameters such as tooth proportions, module, pressure angle, etc., can be varied and cross-calculated. The idea is to achieve an optimized macrogeometry solution that meets the design targets. In this design, the targets are:

- **High contact ratio tooth profile:** The gear tooth height can be increased by extending the gear tooth addendum and dedendum to target higher transverse contact ratios (2.0 and above). We must consider the effects of the stresses on the gear tooth flank and root, as well as specific sliding with higher contact ratio designs. In addition, tooth tip thickness needs to be considered here, as these types of designs can sometimes lead to pointed teeth.

- **Maximized root fillet:** With an extended dedendum and sufficient clearance to the mating gear tooth, a full root fillet can be used. Further steps to optimize this non-standard root geometry can be taken later.
- **Low/Balanced Specific Sliding (sizing of profile shift coefficients):** Specific sliding is the ratio of sliding velocity to the rolling velocity for a gear tooth. It is recommended by AGMA 917-B97 Ref. 1 to keep this balanced from the start of contact to the end of path to reduce tooth wear.

Table 3 presents the gear information after optimization of the pair. The module is slightly lowered to increase the number of teeth. This is done to increase the contact ratio.

Gear data		Gear 1	Gear 2
Number of teeth		34	109
Normal module	mm	0.45	0.45
Face Width	mm	5	4.5
Normal pressure angle	°	20	20
Addendum coefficient		1.11	1.31
Root radius coefficient		0.31	0.41
Dedendum coefficient		1.56	1.36
Profile shift coefficient		0.2961	-0.6768
Reference diameter	mm	15.300	49.050
Base diameter	mm	14.377	46.091
Tip diameter	mm	16.565	49.619
Root diameter	mm	14.162	47.216
Tip form diameter	mm	16.500	49.539
Root form diameter	mm	14.583	47.662
Tooth thickness tolerance	mm	-0.039 /-0.051	-0.039 /-0.051
Quality (ISO 1328:2013)		8	8
Materials		POM	PA 66

Table 3—Gear data of the optimized design.

Operating Backlash

Plastic gears typically are less accurate than metal gears and are highly sensitive to thermal and moisture effects. *KISSsoft*'s Operating Backlash Module was therefore used to consider not only tooth thickness and center distance tolerances, but also temperature and thermal expansion of the gears and housing, gear runout, manufacturing errors, and any potential misalignment. Tooth thickness tolerances, tip/root diameters, and/or center distance tolerances were then adjusted to ensure adequate backlash and tip clearance at all operating conditions.

Root Optimization

Once the macrogeometry is optimized, there might be a further need to examine the root, as modifications to the profile can often result in undesirable stress concentrations in the tooth fillet. It is an additional opportunity to also increase the strength of the root with non-standard geometry, and often high-contact ratio designs can lead to higher root stresses due to the cantilever effect of a tall tooth. Once again, this is a benefit of designing plastic gears manufactured by the molding process—the tooth and root shape can essentially be made to any shape required, within the limitations of manufacturing and injection molding tooling requirements. Elliptical root modification is one useful option to reduce stress concentration even more, as the stress can be lowered since it increases the material and reduces stress lines through the root. This might need to be followed with a radius at the root of the gear to reduce the sharp corners produced by the elliptical root modification, but not always.

A final verification using LTCA is performed after final microgeometry optimization to analyze the root stresses, contact, sliding, and transmission error.

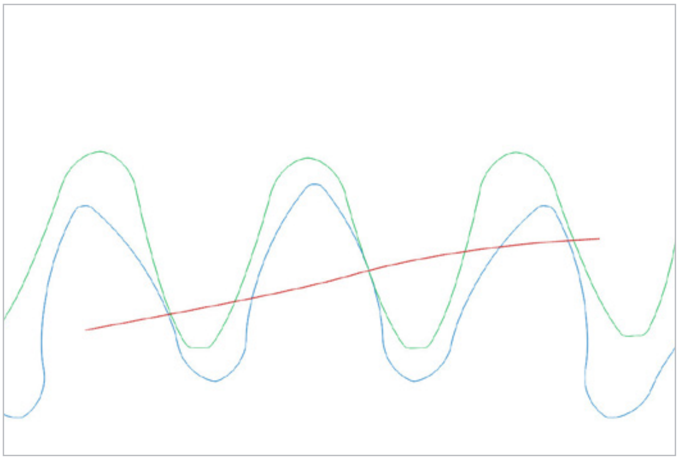

Figure 2—Meshing of the optimized gear pair under load, with the path of contact this time being free of any sharp points, indicating negligible contact shocks and lower transmission error.

Microgeometry Modifications

Profile modifications were sized and optimized in *KISSsoft* using the Modification Sizing Module. Flank line modifications were not considered due to the injection molding process. This can often be the most time-consuming part of the entire design process. Here we initially size some profile modifications such as tip relief, pressure angle modification, and/or profile crowning for a single load case. Then we vary the value and roll angle of the modifications on the active tooth profile across a range of operating loads for a single temperature case. Using Loaded Tooth Contact Analysis (LTCA), we can verify gear excitation, stresses, contact, and sliding in a loaded state, considering the microgeometry (which is not considered in the initial sizing using VDI 2736). Figure 3 shows the profile diagrams; no lead modifications were added due to the moldability of the part. Figure 4 compares the contact patches with the standard tooth profiles and optimized tooth and the presence of the highest contact on the center of the active profile and gradual changes are desirable outcomes of microgeometry addition.

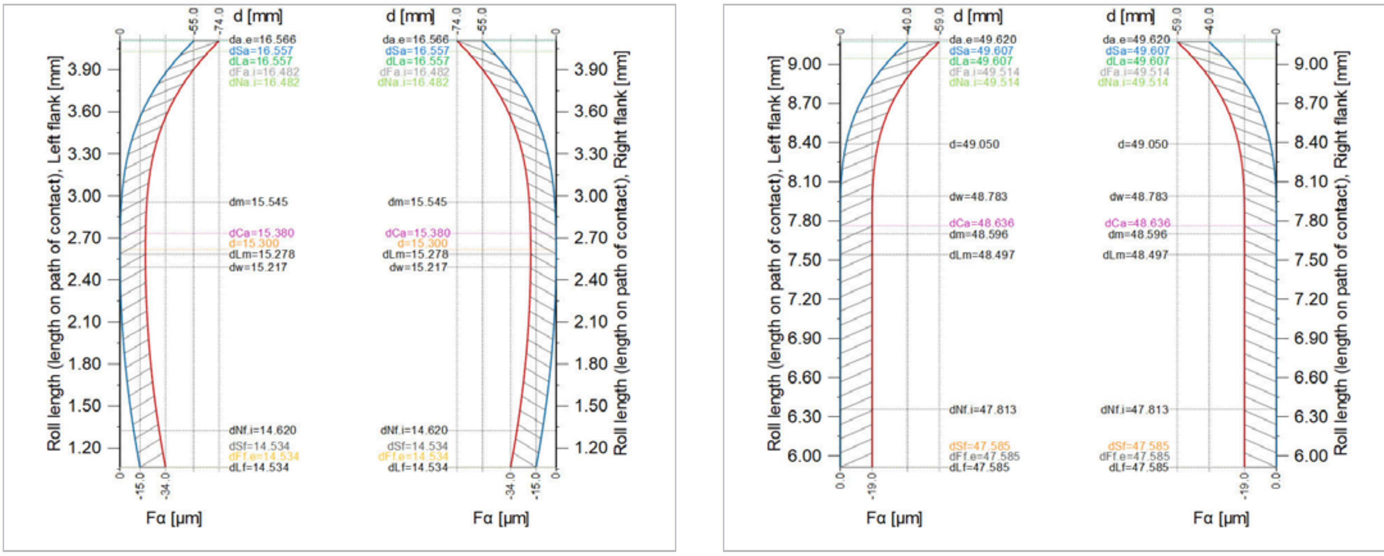

Figure 3—Profile diagrams for Gear 1 (left) and Gear 2 (right) with applied microgeometry corrections.

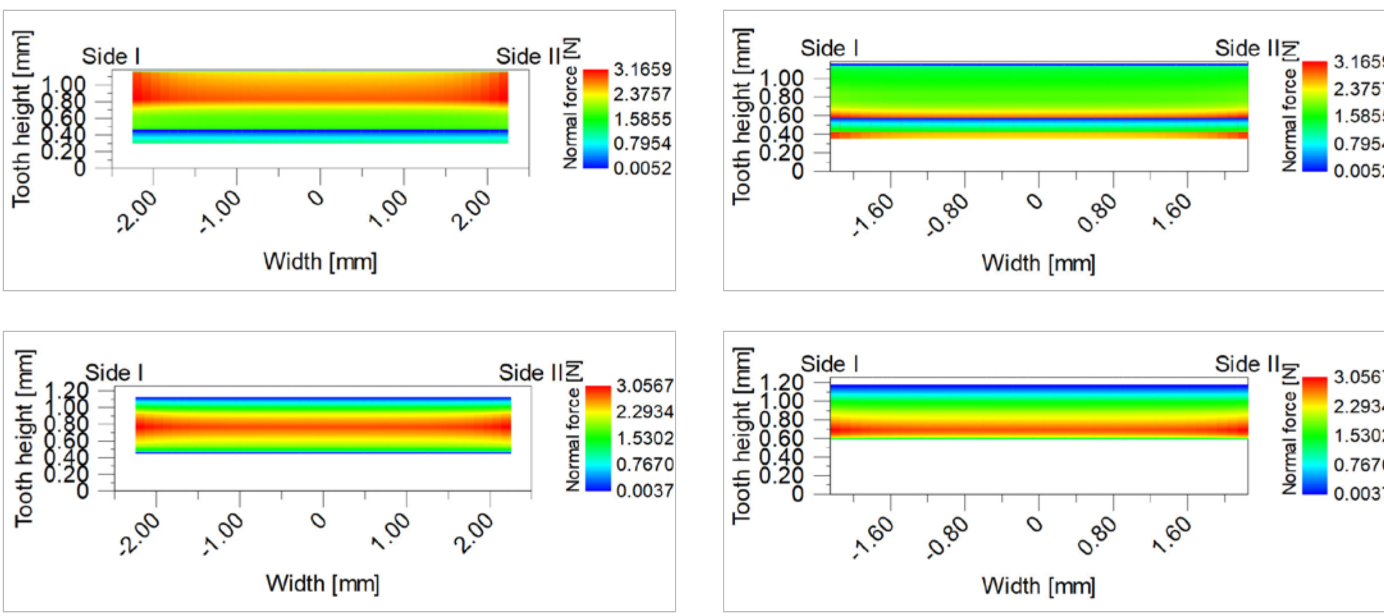

Figure 4—Contact pattern under load for Pinion (left) and Gear (right) shows contrasting pattern and non-uniform patches for standard geometry (top) and central contact zone and uniform transition for the optimized gear pair (bottom).

Stress Calculations After Optimization

Final stress calculations are shown in Table 4. It could be seen that the standard calculation results in ~42 percent and ~30 percent higher root stresses in Gear 1 and Gear 2, respectively. This calculation, according to VDI 2736 Method B, does not account for tooth deformation and load sharing. Thus, additional tools such as loaded tooth contact analysis show ~20 percent lower root stress for Gear 1 and ~13 percent higher for Gear 2, but overall, much better contact and lower transmission error, with sufficient safety factors. Figure 5 shows the FEM results for the gear pair stresses under load. In this, the Loaded Tooth Contact Analysis (LTCA) is used to simulate and does not show any major differences in the root stress.

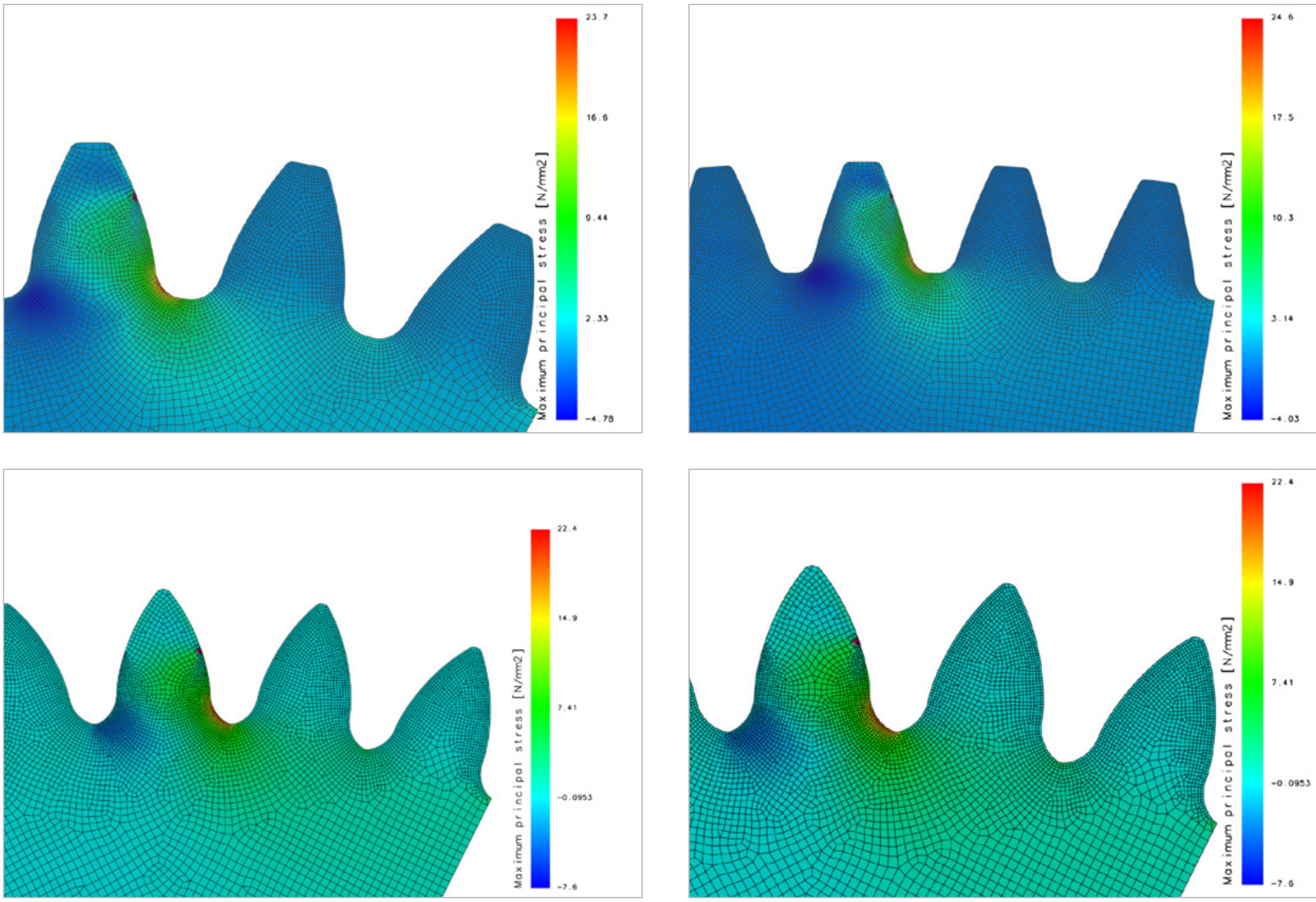

Figure 5—FEM results of the gear pair under load.

Standard Tooth Profile													
Gear #	*Teeth*	*Module*	*Face Width (mm)*	*Temp (deg C)*	*RPM*	*Torque (Nm)*	*Material*	*KISSsoft Tooth Root Stress (MPa)*	*LTCA Root Stress (MPa)*	*FEA Max Root Stress (MPa)*	*Allowable Bending Stress for Material (MPa)*	*Safety According to VDI2736*	*LTCA Root Safety Factor*	*FEA Safety Factor*	*Δ Transmission Error*
Z1	31	0.50	5.0	25	2500	0.1450	POM	21.69	18.19	24.80	45.40	2.09	2.50	1.83	7.78
Z2	97		4.5	25	799	0.4520	PA66	24.78	12.73	23.40	37.00	1.49	2.91	1.58	
Optimized Tooth Profile															
Z1	34	0.45	5.0	25	2561	0.1410	POM	32.23	14.57	24.30	45.40	1.41	3.12	1.87	3.14
Z2	109		4.5	25	799	0.4520	PA66	35.60	14.48	23.40	37.00	1.04	2.55	1.58	

Table 4—Summary of the results before and after optimization.

Conclusion

In this article, we explored plastic gear geometry optimizations that could be carried out using *KISSsoft*. It is encouraged to not stick with standard root shapes for polymer gears unless they are being hobbed which is a less desired process for plastic gearing. Injection molded plastic gears with optimized tooth shape were studied and compared to standard tooth shapes. It is concluded that even with a smaller tooth size, stresses are comparable or even lower on non-standard gear pairs.

The higher contact ratio achieved due to increased tooth count helps with load distribution and reduces transmission error, which leads to better acoustics of the gearset. While there may be more solutions and different designs that may satisfy the particular load case, this article emphasizes the need to use non-standard design techniques for plastic gearing, making polymer gears more power-dense and highly efficient.

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