Optimal Polymer Gear Design: Metal-to-Plastic Conversion

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Introduction

Nowadays, the progress in polymer materials and injection molding processing has enabled a drastic expansion of plastic gear applications. They are used not only for lightly loaded motion transmissions, but also in moderately loaded power drives in automotive, agriculture, medical, robotics, and many other industries. According to (Ref. 1), "plastic gears may be considered for powertrain applications, like auxiliary drives of vehicle drivetrains, industrial gear units and even the main transmissions of light vehicles." This basically means replacing the currently used metal gears with plastic ones. This metal-to-plastic conversion takes advantage of the benefits of plastic gears — such as low production cost, reduced weight and inertia, low noise and vibration, zero corrosion and electric current conductivity — and the advantages of the injection molding process in producing complicated multifunctional parts. However, exact replication of a metal gear design typically does not work, mainly because of the low strength, wear resistance, thermal resistance, and thermal conductivity of polymers compared to metals. These downsides can be compensated for by the optimal design of plastic gears.

Direct Gear Design Method

Traditional gear design is based on rack generation, imitating the hobbing process of machined metal gears (Fig. 1). The main advantage of traditional gear design is the ability to use the same hob cutter for machining gears with different numbers of teeth and addendum modifications (X-shifts), significantly reducing tooling inventory. Traditional gear design is also well-supported by standards and the availability of standard gear cutters. At the same time, this gear design method has limited options to optimize gear tooth geometry for achieving maximum performance for a particular application. After selecting the basic (or generating) rack parameters — which in most cases are standard — there are only addendum modifications (X-shifts) that can be considered as optimization parameters.

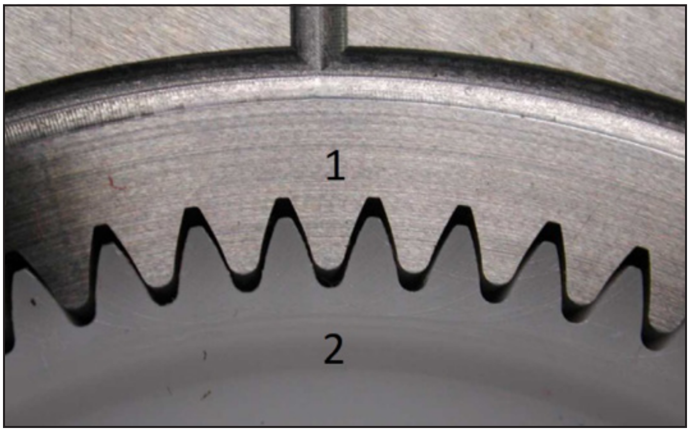

Figure 2 **Plastic gear injection molding; 1 – tooling cavity, 2 – gear.**

Mass-produced plastic gears are formed by the cost-effective injection molding fabrication technology. A gear molding tool is very different than a hob cutter used for machining metal gears. Its cavity has the same profile as the gear, but adjusted for shrinkage and warpage during polymer cooling and crystallization (Fig. 2). Unlike a hob cutter, it is dedicated to producing one particular gear. Nevertheless, plastic gears are designed the same way as machined metal gears, with all the limitations of the standard traditional gear design, based on rack generation, though without the benefit of using one tool for gears with different numbers of teeth. It is not necessary to use this gear design technique for plastic gears.

The alternative *Direct Gear Design* (Ref. 2) defines and optimizes tooth geometry based on gear mesh characteristics without the limitations of specific gear tooling or machining technology. This gear design method utilizes mathematical modeling, finite element analysis (FEA), and CAD software, allowing us to optimize gear tooth geometry for maximum performance in custom gear drives. It is applicable for gears with symmetric and asymmetric teeth, which makes it the preferable design method for plastic gears.

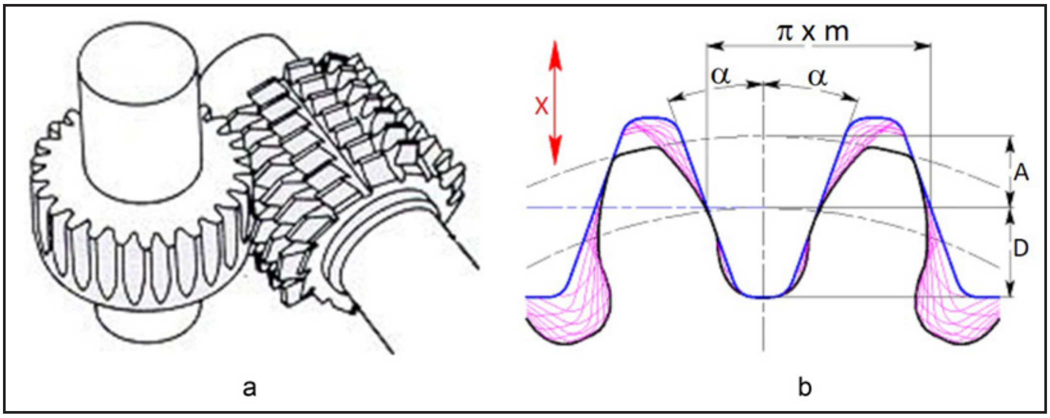

Figure 1 **a – metal gear hobbing; b – definition of the gear profile by rack generation; m – module; α – rack profile (pressure) angle; A – gear tooth addendum; D – gear tooth dedendum; X – addendum modification or X-shift.**

This method describes a symmetric gear tooth (Fig. 3) as formed by two involute flanks (1) of the base circle diameter d_b, an arc distance between them represented by the tooth thickness S at the reference diameter d, the tooth tip diameter d_a and the root fillet (2).

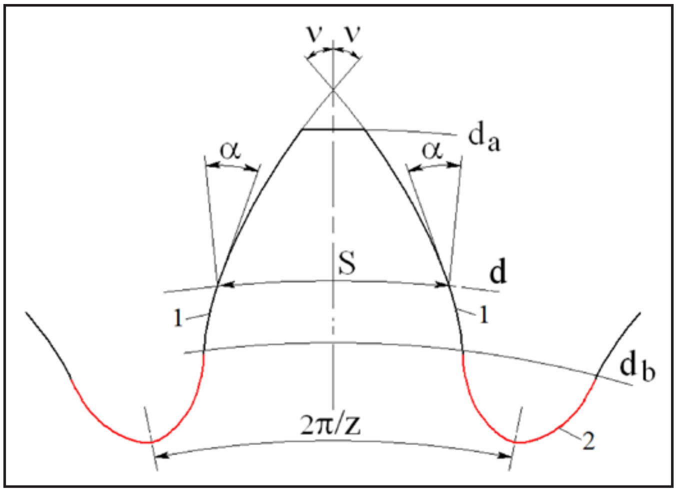

Figure 3 Gear tooth profile construction.

1 – involute tooth flanks
2 – root fillet
d – reference circle diameter
d_b – base circle diameter
d_a – tooth tip circle diameter
S – circular tooth thickness at the reference diameter d
ν – involute intersection profile angle
z – number of teeth

Two gears with equal base circle pitch p_b can be engaged in a gear mesh (Fig. 4).

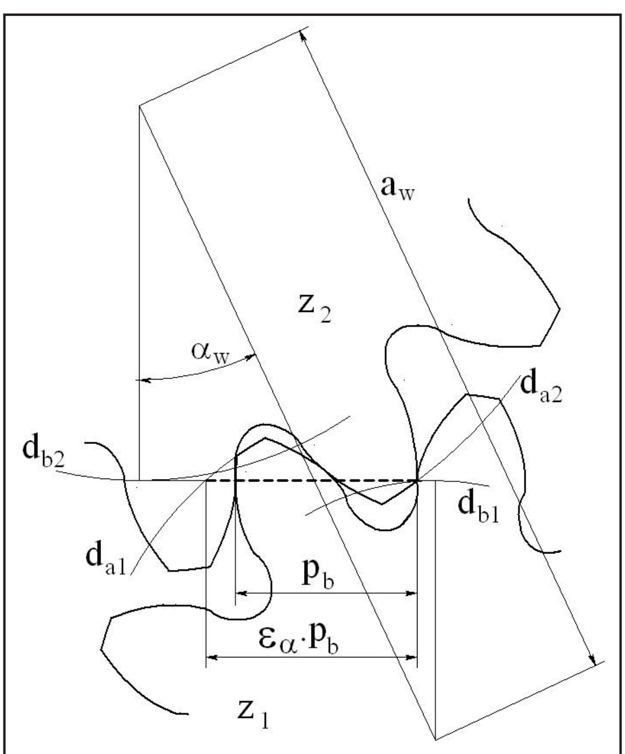

Figure 4 Gear mesh.

a_w – center distance
p_b – base circle pitch
α_w – operating pressure angle
ε_α – contact ratio

The nominal pressure angle α_w and contact ratio ε_α, are defined by the following equations (Ref. 2):

$$\alpha_w = arcinv\left(\frac{inv\nu_1 + u\,inv\nu_2 - \frac{\pi}{z_1}}{1+u}\right) \quad (1)$$

$$\varepsilon_\alpha = \frac{z_1}{2\pi}\left[\tan\alpha_{a1} + u\tan\alpha_{a2} - (1+u)\tan\alpha_w\right] \quad (2)$$

Where
$z_{1,2}$ – numbers of teeth of the pinion and gear;
u – gear ratio, $u = \frac{z_2}{z_1}$,
$\alpha_{a1,2}$ – involute profile angles at the tooth tip diameters,
$\alpha_{a1,2} = \arccos\frac{d_{b1,2}}{d_{a1,2}}$

Tooth flank optimization. In *Direct Gear Design*, practically every parameter of the gear tooth and mesh is a subject for optimization. It allows us to simultaneously increase the nominal (or designed) pressure angle and the contact ratio, which are defined by tooth geometric parameters in Equations 1 and 2. The operating or effective contact ratio can be defined as the ratio of the tooth engagement angle to the angular pitch. The tooth engagement angle is the gear rotation angle from the start of tooth engagement with the mating gear tooth to the end of the engagement. For a spur gear pair, the effective contact ratio is:

$$\varepsilon_{\alpha e} = \frac{\varphi_1}{360/z_1} = \frac{\varphi_2}{360/z_2} \quad (3)$$

where:
φ_1 and φ_2 – pinion and gear engagement angles (Fig. 5)
$360/z_1$ and $360/z_2$ – pinion and gear angular pitches

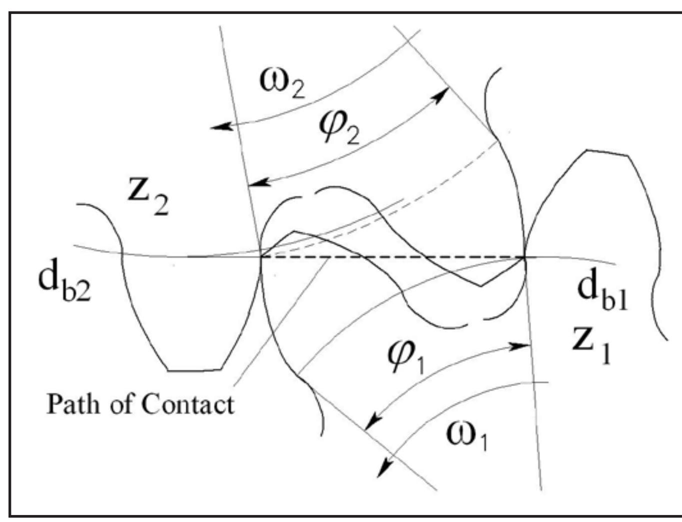

Figure 5 Pinion and gear engagement angles.

The effective contact ratio is affected by manufacturing tolerances and operating conditions, including deflections under the operating load, temperature, etc. of the gears and other gearbox components. In this article, only bending and contact tooth deflections are considered for the definition of the effective contact ratio. Each angular position of the driven gear relative to the driving gear is iteratively defined by equalizing the sum of the tooth contact load moments of each gear to its applied

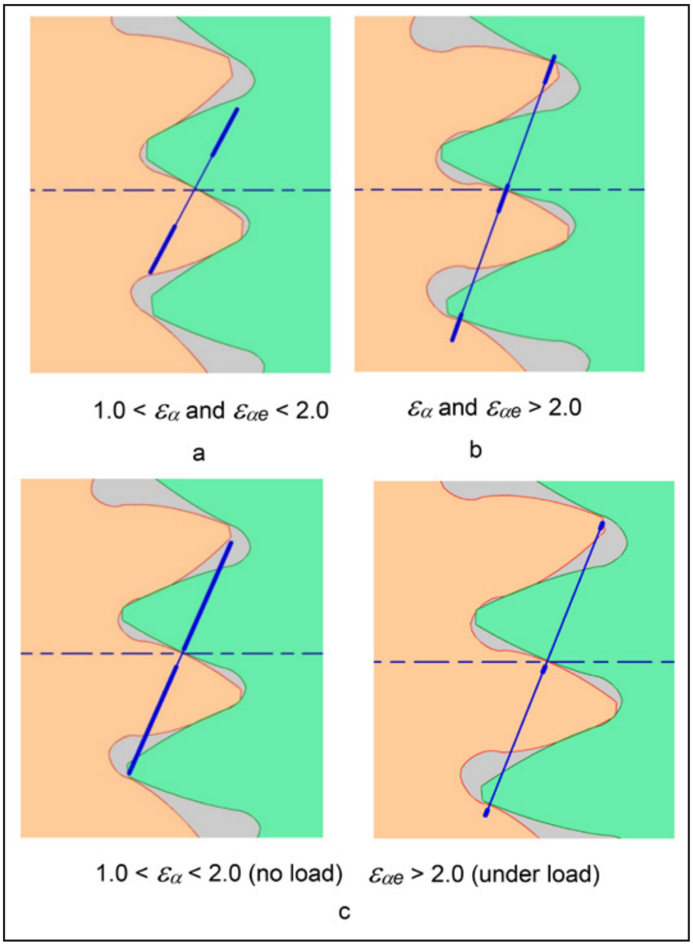

Figure 6 Gears with medium (a), high (b), and transitional (c) contact ratios.

torque. The corresponding tooth contact loads are also iteratively defined to conform to tooth bending and contact deflections, where the tooth bending deflection in each contact point is determined based on the FEA-calculated flexibility and the tooth contact deflection is calculated by the Hertz equation. Under the operating load the effective contact ratio $\varepsilon_{\alpha e}$ is greater than the nominal contact ratio ε_α — mostly because of bending tooth deflections.

Conventional spur gears have medium nominal and effective contact ratios $1.0 < \varepsilon_\alpha$ and $\varepsilon_{\alpha e} < 2.0$ (Fig. 6a). High contact ratio (HCR) spur gears have both nominal and effective contact ratios ε_α and $\varepsilon_{\alpha e} > 2.0$, sharing the transmitted load between at least two tooth pairs and significantly reducing tooth flank and root stresses (Fig. 6b).

Considering that the flexural modulus of gear polymers is dozens of times lower compared to gear steels, the goal of plastic gear flank optimization is to achieve a high pressure angle (≥25°) and an effective contact ratio under the operating load slightly greater than 2.0 (in a range of $\varepsilon_{\alpha e} = 2.05–2.10$) while having a medium nominal contact ratio $1.0 < \varepsilon_\alpha < 2.0$. Such transitional contact ratio gears perform as HCR gears under the operating load (Fig. 6c).

Tooth root optimization. The tooth root fillet is designed after completing the definition of the involute flank parameters. The optimized root fillet profile provides an even distribution for the maximum bending stress along a large portion of the fillet and minimizes bending stress concentration. The initial fillet profile traces the trajectory of the mating gear tooth tip in a zero-backlash mesh (Fig. 7). This prevents interference with the mating gear tooth tip.

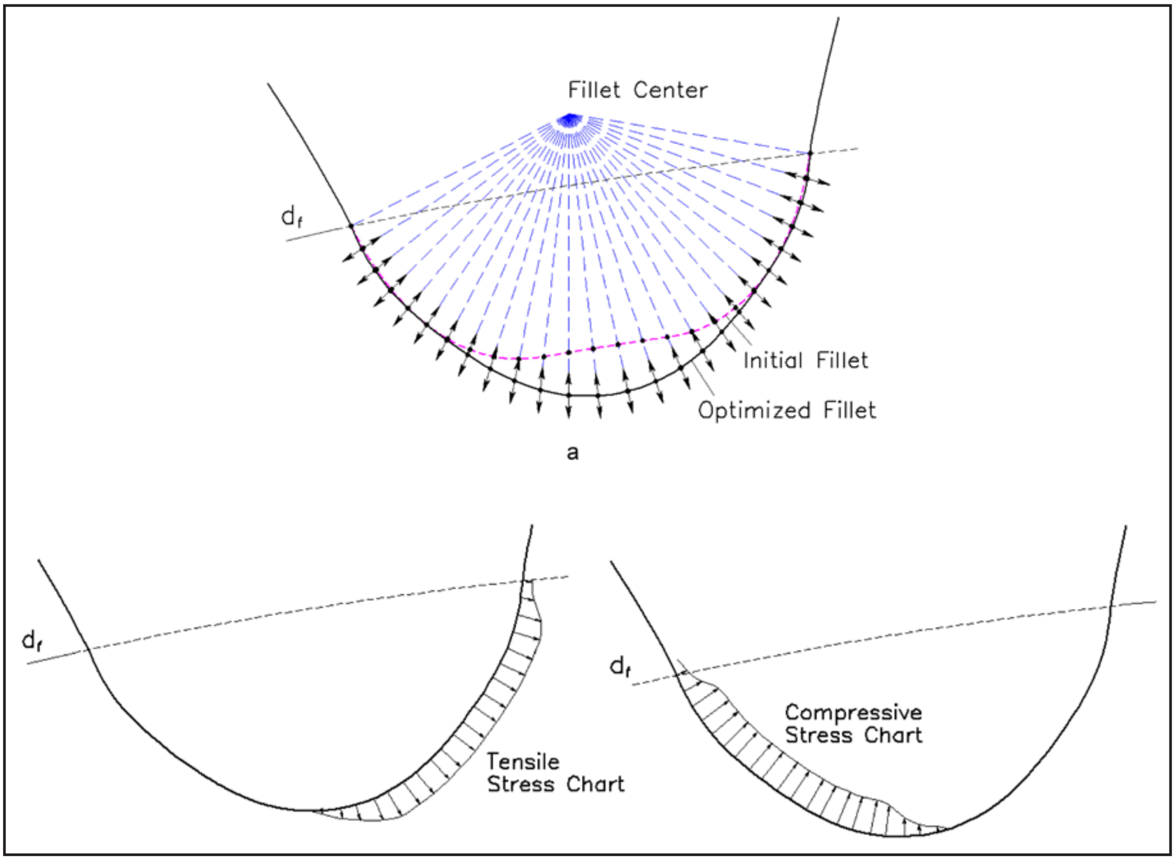

Figure 7 Root fillet profile optimization (a), tensile (b) and compressive (b) stress charts.

Fillet profile optimization utilizes the following calculation processes:

- Definition of a set of mathematical functions that are used to describe the optimized fillet profile. This set may contain trigonometric, polynomial, hyperbolic, exponential, and other functions and their combinations. The parameters in these functions are defined during the optimization process.
- Two-dimensional finite element analysis (FEA) that achieves satisfactory optimization results within a reasonable time period.
- Random search method that defines the next step in the multi-parametric iteration process of the fillet profile optimization.

Detailed description of the tooth root optimization is in (Ref. 2).

Table 1 Metal-to-plastic symmetric tooth gear conversion

		Metal Gear Pair		Plastic Gear Pair	
Design Method		Traditional (Standard 20° Pressure Angle Tool)		Direct Gear Design	
Gear		Driving	Driven	Driving	Driven
Number of Teeth		32	56	24	42
Normal Module, mm		1.500		2.000	
Pressure Angle		20°		26°	
Generating Rack Coefficients	Addendum	1.0		N/A	
	Dedendum	1.25		N/A	
	Tip Radius	0.3		N/A	
	Radial Clearance	0.25		N/A	
X-shift Coefficient		0.0	0.0	N/A	N/A
Tooth Tip Thickness Coefficient		0.73	0.76	0.25	0.25
Pitch Diameter (PD), mm		48.000	84.000	48.000	84.000
Base Diameter, mm		45.105	78.934	43.142	75.499
Tooth Tip Diameter, mm		50.924	72.427	53.195	88.163
Root Diameter, mm		44.162	86.925	43.468	78.344
Root Fillet		Trochoidal	Trochoidal	Optimized	Optimized
Tooth Thickness at PD, mm		2.330	2.330	3.487	2.740
Normal Backlash, mm		0.050		0.050	
Center Distance, mm		66.000		66.000	
Face Width, mm		13.0	12.0	13.0	12.0
Nominal Contact Ratio		1.66		1.65	
Mesh Efficiency, %		98.7		98.3	
Operating Temperature, °C		80		80	
Maximum Driving Torque, Nm		10.0	-	10.0	-
Gear Material		Steel AIS1-1144, annealed		Victrex HPG 140 GRA [5]	
Flexural Modulus, MPa		200,000		3,500	
Poisson Ratio		0.29		0.4	
Effective Contact Ratio		1.76		2.08	
Yield Tensile Strength, MPa		345		70	
Root Stress (FEA), MPa		56.6	61.6	27.8	27.8
Root Safety Factor		6.1:1	5.6:1	2.5:1	2.5:1
Compressive Strength, MPa		794		100	
Contact Stress (Hertz), MPa		504		54.4	
Flank Safety Factor		1.6:1		1.8:1	

Gear Polymers

The selection of gear polymers is driven mostly by the required gear drive load capacity, life, and operating conditions (temperature and humidity, for example). Other considerations may include the cost of the gear drive, weight, noise and vibration, prohibition of external lubrication, etc.

The main polymer gear materials are acetals (POM) and nylons, polyesters, and polycarbonates. They can be used with operating temperatures up to 150°C. For elevated temperatures (<170°C) suitable gear polymers are polyphthalamide (PPA), nylon 46, and similar, high-temperature (<200°C) plastic materials including polyetherimide (PEI), polyetheretherketone (PEEK), and liquid crystal polymers (LCPs).

Some drawbacks of gear plastics properties can be mitigated by additives to the polymer composition. Additives for higher flexural strength include glass, carbon, and aramid (Kevlar) fibers. Tooth flank wear resistance of non-lubricated plastic gears can be increased by anti-wear and anti-friction additives: silicone, polytetrafluoroethylene (PTFE), graphite powders, molybdenum disulfide (MoS2), etc.

It is typically recommended to use dissimilar polymers for mating gears to avoid squeaking noise.

Metal-to-Plastic Conversion

An example of the metal-to-plastic symmetric tooth gear conversion is presented in the Table 1. In this case, a moderately loaded standard gear pair made out of the annealed steel AISI 1144 is supposed to be replaced with a plastic gear pair made of Victrex HPG 140 GRA. Previously, Victrex PEEK polymers were used for metal gear replacement in an internal combustion engine mass balance system (Refs. 3–4).

Overlays of the metal and symmetric plastic gear tooth profiles are shown (Fig. 8).

Despite the fact that the yield tensile strength and also the compressive strength of AISI-1144 steel is significantly greater than that of the Victrex HPG140 GRA polymer, the plastic gear's tooth size increase and flank and root optimization result in acceptable root bending stress and flank contact stress safety factors.

For uni-directionally loaded gear drives undergoing

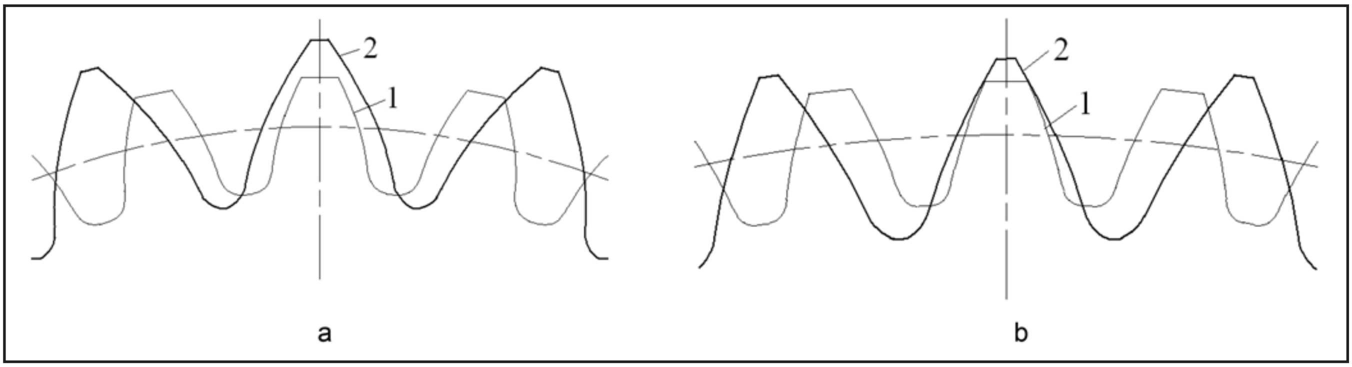

Figure 8 Metal (1) and plastic (2) gear tooth profile comparison; a – pinion teeth, b – gear teeth.

metal-to-plastic gear conversion, asymmetric tooth plastic gears can be considered. They allow for an additional performance enhancement by improving the load capacity of the primary drive tooth flanks at the expense of the opposite coast tooth flanks, which are unloaded or lightly loaded during a relatively short work period (Ref. 6).

Direct Gear Design describes an asymmetric gear tooth construction similarly to a symmetric one, but in this case, it is formed by the involute drive and coast flanks unwound from two different base circle diameters d_{bd} and d_{bc} (Fig. 9).

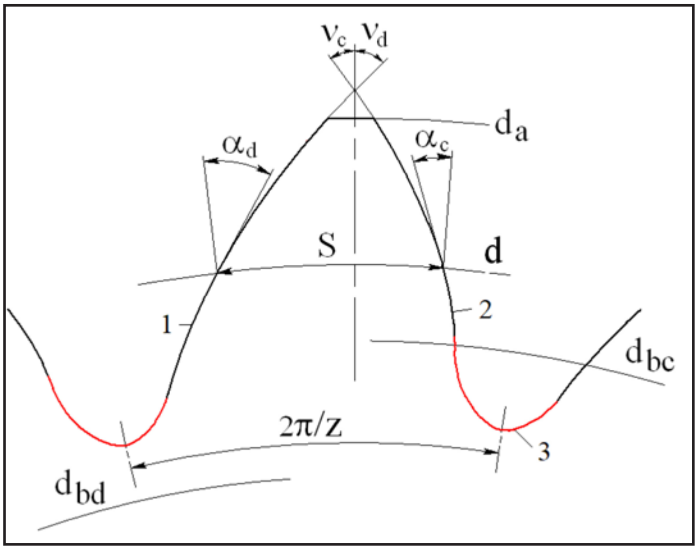

Figure 9 Asymmetric gear tooth profile construction.

1 – drive tooth flank
2 – coast tooth flank
3 – root fillet
d_{bd} – drive flank base circle diameter
d_{bc} – coast flank base circle diameter
ν_d – drive flank intersection profile angle
ν_c – coast flank intersection profile angle

Table 2 presents a metal-to-plastic symmetric tooth gear conversion similar to the one shown in Table 1 with additional columns for the asymmetric plastic gear's data.

Overlays of the metal and asymmetric plastic gear tooth profiles are shown (Fig. 10).

Using optimized asymmetric tooth plastic gears for uni-directionally loaded gear drives can additionally reduce the contact stress and increase drive tooth flank durability.

Conclusion

The goal of using the optimal plastic gear design for metal-to-plastic conversion is to utilize the benefits of polymer materials and injection molding technology, and simultaneously to compensate for relatively low load capacity.

Design guidelines for optimal plastic gear design include:

Increasing tooth size (larger module or coarser diametral pitch) to reduce root bending stress and reducing numbers of teeth to keep the required gear ratio and center distance. This also reduces tolerance sensitivity — especially for fine pitch gears.

Optimizing tooth flanks to achieve an effective (under load) contact ratio ≥ 2.0 and at the same time higher operating pressure angle, which is possible considering the low flexural module of polymers compared to steels. This allows for the distribution of the transmitted load between at least two tooth pairs, significantly reducing tooth flank and root stresses and increasing tooth flank wear resistance.

Optimizing the root fillet profile for root bending stress reduction.

Applying an asymmetric tooth profile for uni-directionally loaded gear drives.

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For more information.

Questions or comments regarding this paper? Contact Alex Kapelevich at *ak@akgears.com*.

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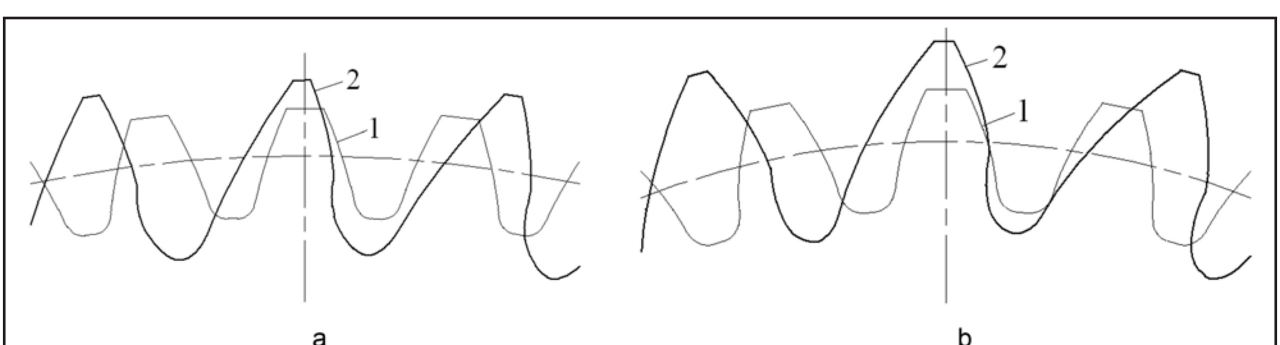

Figure 10 Metal (1) and asymmetric plastic (2) gear tooth profile comparison; a – pinion teeth, b – gear teeth.

Table 2 Metal-to-plastic symmetric tooth gear conversion with asymmetric plastic gear data

		Metal Gear Pair		Plastic Gear Pair		Plastic Gear Pair	
Design Method		Traditional Gear Design (20° Pressure Angle Rack)		Direct Gear Design			
				(Symmetric Teeth)		(Asymmetric Teeth)	
Gear		Driving	Driven	Driving	Driven	Driving	Driven
Number of Teeth		32	56	24	42	24	42
Normal Module, mm		1.500		2.000		2.000	
Pressure Angle		20°		26°		32°/10°*	
Generating Rack Coefficients	Addendum	1.0		N/A		N/A	
	Dedendum	1.25		N/A		N/A	
	Tip Radius	0.3		N/A		N/A	
	Radial Clearance	0.25		N/A		N/A	
X-shift Coefficient		0.0	0.0	N/A	N/A	N/A	N/A
Tooth Tip Thickness Coefficient		0.73	0.76	0.25	0.25	0.25	0.25
Pitch Diameter (PD), mm		48.000	84.000	48.000	84.000	48.000	84.000
Base Diameter, mm		45.105	78.934	43.142	75.499	40.706/ 47.271*	71.236/ 82.724*
Tooth Tip Diameter, mm		50.924	72.427	53.195	88.163	53.655	88.681
Root Diameter, mm		44.162	86.925	43.468	78.344	42.936	77.871
Root Fillet		Trochoidal	Trochoidal	Optimized	Optimized	Optimized	Optimized
Tooth Thickness at PD, mm		2.330	2.330	3.487	2.740	3.490	2.742
Normal Backlash, mm		0.050		0.050		0.050	
Center Distance, mm		66.000		66.000		66.000	
Face Width, mm		13.0	12.0	13.0	12.0	13.0	12.0
Nominal Contact Ratio		1.66		1.65		1.66**	
Mesh Efficiency, %		98.7		98.3		98.3	
Operating Temperature, °C		80		80		80	
Maximum Driving Torque, Nm		10.0	-	10.0	-	10.0	-
Gear Material		Steel AIS1-1144		Victrex HPG 140 GRA		Victrex HPG 140 GRA	
Flexural Modulus, MPa		200,000		3,500		3,500	
Poisson Ratio		0.29		0.4		0.4	
Effective Contact Ratio		1.76		2.08		2.10**	
Yield Tensile Strength, MPa		345		70		70	
Root Stress (FEA), MPa		56.6	61.6	27.8	27.8	31.7	29.6
Root Safety Factor		6.1:1	5.6:1	2.5:1	2.5:1	2.2:1	2.4:1
Compressive Strength, MPa		794		100		100	
Contact Stress (Hertz), MPa		504		54.4		51.1	
Flank Safety Factor		1.6:1		1.8:1		2.0:1	

* drive/coast flank
** drive flank

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