



**08FTM06**

# AGMA Technical Paper

# Tooth Fillet Profile Optimization for Gears with Symmetric and Asymmetric Teeth

By A. Kapelevich, AKGears, LLC  
and Y. Shekhtman, One Step  
Molding Solution

# Tooth Fillet Profile Optimization for Gears with Symmetric and Asymmetric Teeth

**Alex Kapelevich, AKGears, LLC and Yuriy Shekhtman, One Step Molding Solution**

[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

## Abstract

The gear tooth fillet is an area of maximum bending stress concentration. However, its profile is typically less specified in the gear drawing and hardly controlled during gear inspection in comparison with the gear tooth flanks. This paper presents the fillet profile optimization technique for gears with symmetric and asymmetric teeth based on the FEA and random search method. It allows achieving substantial bending stress reduction in comparison with traditionally designed gears. This bending stress reduction can be traded for higher load capacity, longer lifetime, lower noise and vibration, and cost reduction.

Copyright © 2008

American Gear Manufacturers Association  
500 Montgomery Street, Suite 350  
Alexandria, Virginia, 22314

October, 2008

ISBN: 978-1-55589-936-3

# Tooth Fillet Profile Optimization for Gears with Symmetric and Asymmetric Teeth

Alex Kapelevich, AKGears, LLC and Yuriy Shekhtman, One Step Molding Solution

## Introduction

Historically, gear geometry improvement efforts were concentrated on the working involute flanks. They are nominally well described and classified by different standard accuracy grades, depending on gear application and defining their tolerance limits for such parameters as runout, profile, lead, pitch variation, and others. Working involute flanks are also modified to localize a bearing contact and provide required performance at different tolerance combinations and possible misalignment as a result of operating conditions (temperature, loads, etc.). Their accuracy is thoroughly controlled by gear inspection machines.

The gear tooth fillet is an area of maximum bending stress concentration. However, its profile and accuracy are marginally defined on the gear drawing by typically very generous root diameter tolerance and, in some cases, by the minimum fillet radius, which is difficult to inspect. In fact, tooth bending strength improvement is usually provided by gear technology (case hardening and shot peening to create compressive residual stress layer, for example) rather than gear geometry.

The gear tooth fillet profile is typically determined by the generating cutting tool (gear hob or shaper

cutter) tooth tip trajectory (Figure 1), also called the trochoid. If the cutter parameters are chosen or designed to generate the involute flank profile, which must work for the specific gear application and satisfy certain operation conditions, the fillet profile is just a byproduct of the cutter motion. The fillet profile and, as a result, bending stress are also dependent on the cutter radial clearance and tip radius. The standard radial clearance usually is  $0.25/P$  or  $0.20/P+0.002''$ , where  $P$  is the standard diametral pitch. The standard cutter tooth radius for the coarse pitch gears is  $0.3/P$ . For the fine pitch gears the standard cutter tooth radius is not standardized and can be as low as zero [1].

Unlike the contact Hertz stress, the bending stress does not define the major dimensions of the gears, such as pitch diameters or center distance. If the calculated bending stress is too high, in many cases, the number of teeth can be reduced and the coarser diametral pitch (larger module) can be applied to keep the same pitch diameters, center distance, and the same (or close) gear ratio. This makes the gear tooth physically larger and reduces bending stress to an acceptable level. Of course, this increases specific sliding and reduces contact ratio and gear mesh efficiency, but this is better than the broken teeth.

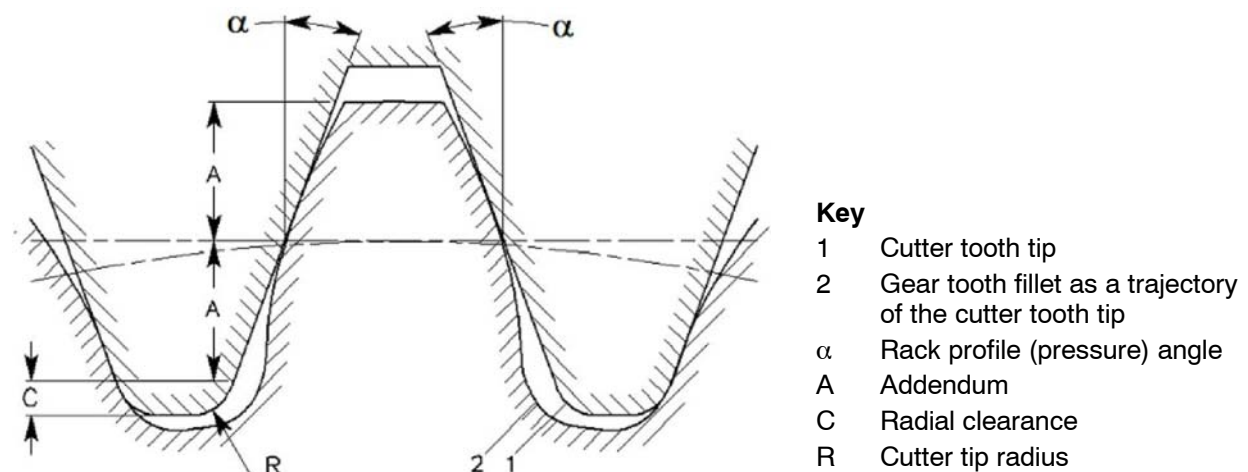


Figure 1. Gear tooth fillet generation by the rack cutter (gear hob)

There are two general approaches to reducing bending stress for the given tooth size. One of them is to alter the generating cutter tooth tip – most common application of this approach is the rack with the full tip radius. Another approach is to alter the gear tooth fillet profile – the most common application here is the circular (instead of trochoid) fillet. Further development of both these approaches is based on a mathematical function fitting technique where the cutter tip radius or the gear tooth trochoid fillet profile is replaced by a parabola, ellipsis, chain curve, or other curve, reducing the bending stress (see for example [2, 3]). Bending stress reduction achieved by such fillet profile improvement is varied and greatly depends on the cutter or gear tooth parameters. The resulting tooth fillet profile must be checked for interference with the mating gear at various gear (and center distance) tolerance combinations.

This paper presents the Direct Gear Design fillet profile optimization technique, which allows for a substantial bending stress reduction in comparison to traditionally designed gears. It also describes how bending stress reduction can produce other gear performance benefits.

## Optimization method

Direct Gear Design [4] defines all gear geometry parameters without using the pre-selected basic or generating rack. It is applied for custom gears and allows for the separation of the active involute flank and tooth fillet design.

The flank profiles are designed first to satisfy primary performance requirements, such as maximum load capacity with acceptable contact stress level, maximum gear mesh efficiency (minimum specific sliding), etc.

The tooth fillet design is based on completely defined involute flank parameters. The initial fillet profile is a trajectory of the mating gear tooth tip in the tight (zero backlash) mesh. For practical purposes, this trajectory is defined at the minimum center distance (including both gears' runout), maximum tooth thickness, and maximum outer diameter of the external mating gear (for an internal mating gear the minimum inner diameter is used). This allows the exclusion of interference with the mating gear tooth.

The fillet optimization consists of three major components [5]:

- trigonometric functions for fillet profile approximation;
- FEA for stress calculation;
- a random search method to define the optimal set of the trigonometric functions' coefficients, which allows them to reach the minimum bending stress.

The trigonometric functions are selected in such a way that the first and the last FE nodes of the initial fillet profile are placed on the form diameter circle (Figure 2) and cannot be moved during the optimization process. The rest of the initial fillet FE nodes are moved along the beams that pass through the fillet center. The center of the fillet is the center of the best-fitted circle. The bending stresses are calculated for every new fillet profile configuration. The adjustment of the optimizing variable parameters is defined based on the successful (leading to stress reduction) and unsuccessful (leading to stress increase) iteration steps and some random vector. The number of iteration steps (or optimization time) and minimal iteration steps are limited. The random nature of this method does not yield absolutely identical results for the same set of gear parameters and number of iteration steps. The program was adjusted so that the maximum bending stress difference between repeated calculations does not exceed 2%. The fillet shapes for these cases are also slightly different.

## Optimization Results

As an example of the fillet profile optimization, different fillets were constructed for the gear pair with the standard involute tooth profile and the following parameters (see Figure 3):

- Number of teeth of both mating gears – 24;
- Diametral pitch – 12;
- Generating rack profile (pressure) angle – 20°;
- Addendum coefficient (also known as normalized addendum coefficient) – 1.0;
- Face width of both mating gears – 1.0";
- Operating torque – 200 in-lb.

Table 1 presents the FEA of the different fillet profiles. It also indicates that the optimized fillet has the largest curvature radius at the maximum stress