

A Gear Design Optimization Procedure that Identifies Robust, Minimum Stress and Minimum Noise Gear Pair Designs

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American Gear Manufacturers Association



TECHNICAL PAPER

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[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

Typical gear design procedures are based on an iterative process that uses rather basic formulas to predict stresses. Modifications such as tip relief and lead crowning are based on experience and these modifications are usually selected after the design has been developed. In the traditional approach, the selection of the optimal designs is really sub-optimal, since there are always compromises that are made between the many design performance evaluation variables.

The design procedure that is described in this paper starts with micro-topographies in the form of profile and lead modifications that are appropriate for a given design and that are easy to manufacture. We then evaluate the load distribution, bending and contact stresses, transmission error, film thickness, flash temperature, etc., for a large number of designs (sometimes as many as 1 million) in an effort to find the “best” design for the predetermined tooth micro-topographies. The key to the analysis is the rapid evaluation of the load distribution. A unique graphically oriented selection procedure is then used to determine designs that are good in many of the above mentioned design response variables. The best of these designs are then subjected to an evaluation of the effects of manufacturing errors on the performance evaluation variables (stresses, noise excitations, etc.). Designs that are least sensitive to manufacturing errors are then deemed most robust and hence “best” for the given application.

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INTRODUCTION

The design of a gear pair is, indeed, a complicated process. Usually there are many measures of a good design including bending and contact stress balancing and minimization, minimizing scoring features such as flash temperature, and minimizing noise. Noise minimization is seldom considered in great detail at the initiation of a design, but is often considered only when noise becomes a problem. In addition, the designer seeks to find designs that are insensitive to manufacturing and mounting errors. In order to achieve good designs the designer may vary a large number of geometry and manufacturing factors such as number of teeth, pressure angle, helix angle, tooth height, hob shift, etc. Usually designers apply their past experiences to select design parameters, but due to the tedium of the process may not consider very many alternative designs. Also, the tooth surface micro-geometry that is defined by the tooth topography is rarely considered at the beginning of a design, but may be considered after the main geometric parameters are chosen.

This paper presents an optimization methodology that allows the manipulation of a large number of design parameters and allows selecting designs that are superb in many of the measures of goodness that the designer chooses to use. Tooth micro-design and design for minimum noise are considered throughout the design process and finally, the procedure allows one to assess the sensitivity of the design to manufacturing and assembly deviations.

DESIGN OPTIMIZATION

Design optimization has, over the past couple of decades, literally become an engineering specialty. Yet most demonstrations of optimization techniques vary few design variables and use relatively simple objective functions for the optimization of the designs. We will first discuss design optimization issues encountered in gear design and then present a unique gear design optimization approach.

Macro vs. Micro Gear Design

By macro design we mean the varying of design parameters such as numbers of teeth, pressure angle, helix angle, etc. Changing these parameters can have a large effect on evaluation parameters such as root stress or contact stress and allows the application of evaluation equations such as those used by AGMA [1] or that appear in design textbooks.

Micro-design is the establishment of the specification of the shape of the gear tooth's active profile. This procedure includes the application of lead crowning and lead modifications that are often necessary for compensating for misalignment and mounting inaccuracies and profile modifications that are used to reduce contact along the tooth tips and roots and for noise reduction. Micro design is usually applied after one has selected the macro-design parameters. Although the shapes used in micro-design are unlimited, manufacturing considerations usually limit the complexity of the shapes that are practical.

Optimization Background

Most gear designers [2-5] who have used design optimization have applied their methodology using conventional gear stress formulas in a macro-design approach. The approaches used usually have taken into consideration only one or two parameters and have rarely considered such conventional design approaches as the balancing of the strength and durability lives or the balancing of bending strength between the pinion and gear. Each of these criteria is best solved using cutter profile shift in order to adjust tooth thickness of the gear and pinion. The optimization approach applied by these authors requires the selection of an objective function that is usually minimized using some type of steepest ascent method.

We have observed several difficulties with conventional gear design optimization techniques:

1. The number of design variables in even the simplest gear design is of the nature of from 6-10. With this number of variables we find that there are many combinations of design variables that provide near optimum solutions. These multiple optima lead to major shortcomings of hill climbing optimization methods that will reach the peak of one of these "hills" but may miss analyzing any of the other hills.
2. When multiple term objective functions are used, there is no rational means of weighting each of the variables. For instance, a variable set might include minimizing bending stress, contact stress and flash temperature. If one chooses to weight them equally, just the definition of "equal" is difficult, since we are dealing with quantities with totally different units.
3. In addition, multiple term objective functions tend to provide a single optimum design solution. Simply changing one of the weighting factors by a small amount will likely provide a totally different "optimum" design. Since we are uncertain what weighting factors to use, we find that there are likely to be a huge number of satisfactory designs that fit within the range of weightings that we deem acceptable.
4. Since micro-topographies have rarely been used in gear design optimization schemes, it has been impossible to include such factors as noise or sensitivity to manufacturing errors or assembly variations into the design procedure.

Robustness Considerations

In this paper, the word "robustness" refers to the insensitivity of a design to manufacturing and assembly errors. Assembly errors include such factors as center distance variability and shaft or gear mounting misalignment. Manufacturing errors are best defined as deviations in the tooth profiles, leads, and spacing from

their design goals. In this case both first order deviations (profile and lead (slope errors) and second order deviations (curvature errors) are used. Modern gear inspection machines may be used to obtain statistics on these errors for different parts and manufacturing processes. Spacing errors [6] that are important in evaluating the effect of transverse load sharing on bending and contact stresses are not considered in this paper.

DESIGN OPTIMIZATION APPROACH APPLIED IN THIS PAPER

Our approach to gear design optimization that is applied in this paper addresses many of the issues presented in the forgoing paragraphs. First, tooth surface optimization is applied to the design, then an analysis of the load distribution is performed and the numerous design evaluation parameters are calculated based on the predicted load distribution. The load distribution analysis is performed on a large number of designs (sometimes over one million) and two different schemes may be used for the selection of the best of these many designs. Finally, the procedure allows the evaluation of robustness of the optimum designs. An alternative approach that is not shown in this paper would be to perform robustness analysis on each of the designs prior to selection of the best designs.

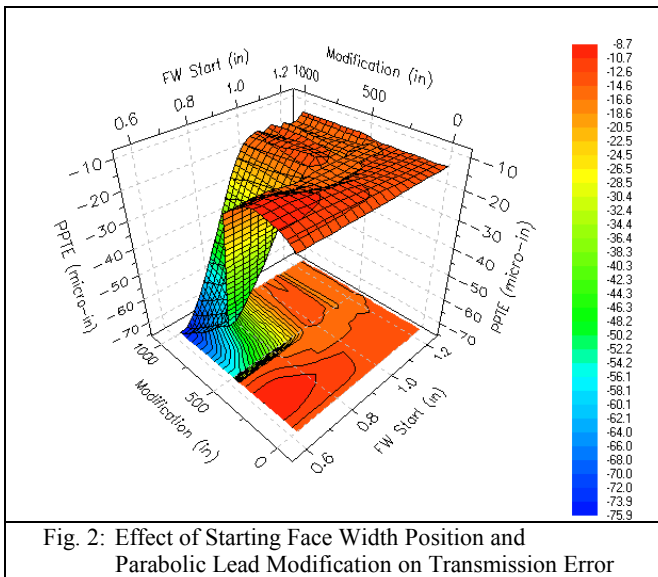
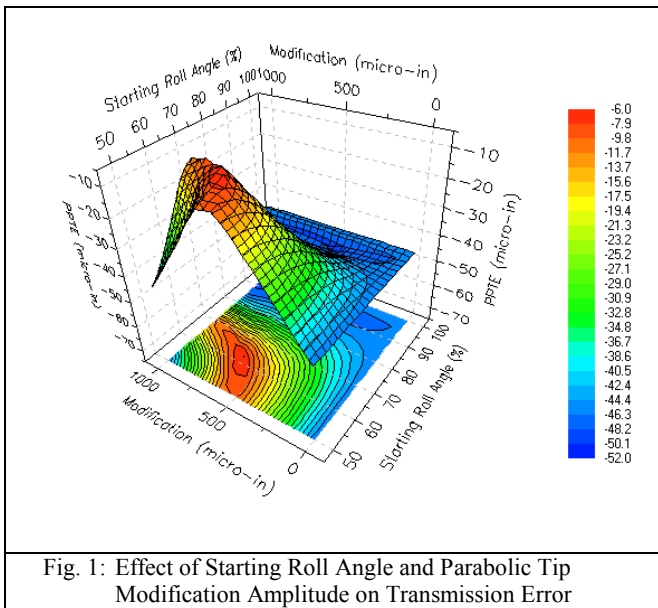
Tooth Surface Optimization

In an earlier work, Regalado, et. al. [7-8] showed that it was possible to control tooth stresses while at the same time determining the tooth micro-topography for minimizing transmission error. Unfortunately, the topographical shape that results is very complex and would be literally impossible to manufacture in an accurate and economical manner. Therefore, our new approach is to start with easy to manufacture profile and lead modification shapes and then optimize the shape coefficients (amplitudes) as outlined below.

The current approach to tooth surface optimization is two-fold:

- Provide adequate lead crowning for low misalignment sensitivity.
- Apply appropriate tip relief to minimize transmission error and also to minimize peak contact stresses due to edge contact.

In this analysis the starting roll angle and amplitude of tip relief are varied and the condition of minimum transmission error is found. Fig. 1 shows a typical result for the baseline helical gear pair to be analyzed later in this paper. In this case, parabolically shaped relief was applied and the best shape occurs when the relief starts near the center of the tooth height. Since this parabolic shape is nearly circular as a function of roll angle, all subsequent analyses will use circular profile modifications.



The next stage of topographical optimization is to weigh the amount of tip relief against the desirable lead modification. Fig. 2 shows the relative amplitude of circular tip relief and circular lead modification on transmission error. We see that a nearly linear relationship occurs where the sum of the profile modification and the lead modification are constant.

The above information is used as a guide for choosing mean modifications that are then used for all of the gear design cases that are run. In essence, we are selecting an easy to manufacture tooth topography and then finding gear geometries for which these modifications provide optimal results for the evaluation parameters.

Run-Many-Cases Approach

In this methodology, we seek to run as many sets of design parameters as possible and then use a creative means to

select the best designs. The methodology borrows from the Seeker – Filter – Viewer approach developed by Chandrasekaran, et. al. [9-10] and was applied to gears by Houser, et. al. [11-12].

A block diagram of the method is shown in Fig. 3. It consists of generating possible candidate designs based on the range of design parameters that are selected. Next, gear micro-geometries are selected based on the methods discussed earlier. These data are then input to a load distribution solver [13] that not only solves for the gear pair's load distribution, but also computes evaluation parameters such as root stresses, contact stresses, flash temperature, transmission error, etc. Two methods of reducing the number of design cases to a tractable number have been developed (Range and Power methods). These methods are used in conjunction with a graphic processor that allows the designer to visualize the selected designs in perspective with the results of all designs that are evaluated. Finally, a robustness analysis may be performed to study the sensitivity of the selected designs to manufacturing error.

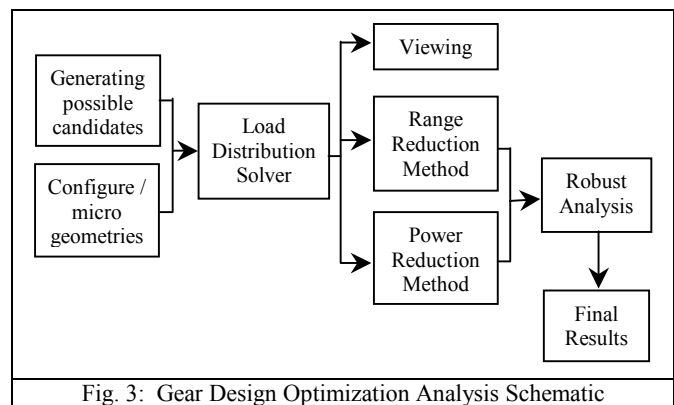


Table 1: NASA Baseline Gear Geometry

	Pinion	Gear
Number of teeth	25	31
Diametral pitch (1/in)	8.598	
Pressure angle (degree)	23.45	
Helix angle (degree)	21.50	
Center distance (in)	3.5	
Outside diameter (in)	3.360	4.110
Root diameter (in)	2.727	3.457
Face width (in)	1.25	1.25
Torque (lb-in)	2000 lb-in	

Through the use of an example design problem whose base geometry is given in Table 1, each block of the schematic of Fig. 3 are explained in detail in the next section of this paper. The design data are taken from a set of gear pairs that were designed and built for noise testing at NASA-Glenn Research Center (Drago, et. al. [14]).