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Bending and Compressive Stress Analysis of External Helical Gearsets of Varying Contact Ratios

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American Gear Manufacturers Association



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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

In an attempt to better understand how the durability of a helical gearset is affected by changes in profile (m_p) and face contact ratio (m_f), an analytical investigation was done in which m_p was varied in levels of 1.1, 1.6, 2.1, and 2.6 and m_f was varied in levels of 0, 0.6, 1.1, 1.6, 2.1, 2.6, 3.1, and 3.6. Thirty-two combinations were studied in total. The gears were modeled using the hybrid finite element computational method Contact Analysis Programming Package (CAPP), of Advanced Numerical Solutions. CAPP was chosen for this analysis since it has the ability to calculate both bending and compressive stresses in gear teeth without the use of a supercomputer.

The results of this analysis suggest that for a given level of face contact ratio, an advantage in bending and compressive stress exists at the $m_p=2.1$ level over all of the others considered. No such optimum level was found for face contact ratio in this study. As the helix angle and m_f were increased, the maximum bending stresses increased due to the increases in normal tooth load and compressive stresses decreased due to the increases in normal face width. Increasing both m_p and m_f had the effect of smoothing out the bending and compressive stresses when viewed as a function of roll angle.

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Introduction

The influence of contact ratio on the generation of noise from a gear mesh is fairly well known and documented throughout the industry. [1,2] In most instances, increasing the profile contact ratio (m_p) of a spur gearset moves the high point of single tooth contact (HPSTC) closer to the pitch point, which reduces the bending moment arm about the base of the tooth. The effect is a smoother action at the extremes of the mesh's arc of action.

Helical gears operate more quietly than spur gears because their line of contact is not parallel to the axis of rotation, but skewed at an oblique angle, extending the length of the contact line onto adjacent tooth pairs. Increasing the length of the line of contact spreads the transmitted load over a greater area, reducing the load intensity of the gear mesh. In addition to the reduced load intensity, each infinitesimally thin slice in the plane of rotation is advanced in roll angle by an amount proportional to the helix angle. Thus, contrary to spur gears, contact across the gear face does not take place at the same roll angle, but rather it is broken up. Neighboring slices contact their mating surfaces at an advanced or retarded roll angle.

What limits the gear designer in producing a tooth form with increased contact ratio is the bending and contact stresses of the tooth. In general, higher profile contact ratio gears have longer and thinner teeth, lower pressure angles and finer pitches. The usual result is a compromise between strength and quietness.

Recently the writer redesigned the complete geartrain of a rear wheel transmission for a torque requirement which was lower than that for which the center distance was selected. Here was an opportunity to design the gear teeth with contact ratios in excess of the usual range ($m_p \approx 1.6$, $m_f \approx 1.6$). Pitches were increased, pressure angles were lowered and tooth depths were increased to design for $m_p > 2.0$. Helix angles were also increased to design for $m_f > 2.0$. The strengths of the gear meshes were still adequate when calculated by conventional methods.

In-vehicle testing of initial prototypes proved that the gears would operate quietly, but this was expected with the elevated contact ratios. Some increased durability of the gearsets was also expected, since work has been done in this area on spur gears [4-8], reporting that high profile contact ratio gearing could allow products to be designed to

greater power densities. However, no work could be found which studied the strength of helical gears designed with high profile and face contact ratios. With the benefits of reduced noise generation now demonstrated, it became necessary to re-evaluate the way the gear teeth are stressed under load and how m_p and m_f effect these stresses.

Since an empirical investigation would be too expensive for a study academic in nature, an analytic study using the hybrid finite element program Contact Analysis Programming Package (CAPP) from Advanced Numerical Solutions was used instead. The gear meshes were modeled and the bending and contact stresses were calculated and the results are presented here.

Description of the Experiment

CAPP is a recently developed analysis tool for the study of all kinds of gears [3]. The analysis of two gears in contact is a problem which is too complicated for general purpose finite element packages. When gears are brought into contact, the width of the contact zone is typically an order of magnitude smaller than the other dimensions of the gears. This gives rise to the need for a very highly refined finite element mesh near the contact zone. But given the fact that the contact zone moves over the surface of the gear as it rolls through its zone of action, one would need a very highly refined mesh all over the contacting surface. Finite element models refined to this extent cannot be accommodated on even the largest of today's computers. Compounding this difficulty is the fact that the contact conditions are very sensitive to the geometry of the contacting surfaces. General purpose finite element models cannot provide the required level of geometric accuracy. Finally, the difficulties of generating an optimal three-dimensional mesh that can accurately model the stress gradients in the critical regions while minimizing the number of degrees of freedom of the

model have kept the finite element method from being widely used to solve the complete gear contact problem.

The approach that CAPP uses is to combine the finite element model with an elastic halfspace model. The finite element model is used to obtain gross deformations of the gear away from the contact zone, while the elastic halfspace model is used to obtain relative deflections within the contact zone. The finite element deformations are then "matched" with local deformations in a least squares sense. What results is an accurate representation of the three-dimensional gear contact problem which is able to be solved on an analysis workstation.

Transverse and axial plane tooth geometry were kept independent in this experiment. Changes in face contact ratio were achieved entirely through a change in the helix angle, with a corresponding change in normal pressure angle such that the transverse pressure angle remained unchanged. Changes in profile contact ratio were achieved entirely through changes in the tooth height. Thus, not only were the profile and face contact ratios kept as independent variables, but also the tooth geometry in the transverse and axial planes as well. Eight levels of face contact ratio were analyzed (spur, 0.6, 1.1, 1.6, ..., 3.6) and four levels of profile contact ratio (1.1, 1.6, 2.1, 2.6).

Since all the gears were loaded to their design torque (290#'), the tooth forms were relieved to account for tooth deflections. The amount of tip relief was arbitrarily selected to be .0012" and parabolic in nature. The starting roll angle of modification was determined by trial and error to minimize contact stresses at entry and exit of mesh. Leads were crowned to .0002" circular modification centered at the midface, without end relief.

Discussion of the Results

Figure 1 is a surface plot of maximum bending stress plotted