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The Effect of Manufacturing Microgeometry Variations on the Load Distribution Factor and on Gear Contact and Root Stresses

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Introduction

Gear rating formulas have numerous design factors that are intended to create realistic evaluations of the stress levels encountered by a gear pair. The dynamic factor, however, is the only factor that has manufacturing accuracy directly included in its evaluation. This paper discusses most of the other factors in the AGMA rating procedure [1] that might be influenced by manufacturing accuracy and then uses load distribution analysis to assess the effects of profile, lead and spacing deviations on root stresses, contact stresses and load distribution factors. The dynamic factor has received ample study in the past [2–7], so it will not be further investigated here. As part of the presented analysis, a procedure is provided for obtaining an acceptable microgeometry design that is relatively insensitive to manufacturing deviations and misalignment.

Manufacturing accuracy definitions

Prior to looking at the factors that affect tooth stresses, what is meant in this paper as manufacturing accuracy will be defined. In this context, manufacturing accuracy encompasses all factors in manufacturing or assembly that change the microgeometries and hence the load sharing of a tooth pair. The AGMA accuracy classification standard [8] uses its quality number system to define quality levels for profiles, leads, runout and spacing. The accuracy of housings, bearings and support shafting, geometry changes to the surface and root geometries and variables such as center distance, backlash, outside diameter, tooth thickness are not included, but are occasionally discussed. Also not included are deviations that are measured through composite and single flank tests. However, these deviations result from the elemental variations that are considered.

Provided below is a brief discussion of factors affecting the tooth microgeometry that are studied in this paper.

Lead deviations and misalignment: These are discussed together since they have similar effects on the load distribution across the tooth face width. The effects of lead deviations, which essentially shift the load to one end of the tooth, have been discussed in many papers [9–14]. Misalignment that is at right angles to the normal contacting plane is additive to lead slope deviation so in this paper these effects for both the gear and pinion will be lumped together into a single variation. The AGMA accuracy standard [8] recognizes that there are potentially two types of lead deviations, one of the linear type and the second being a curvature deviation. The linear slope deviation may be added to misalignment, while the curvature deviation is treated as a deviation in the specified lead crown.

Profile deviations: Profile deviations are often thought of as deviations of the tooth form from a true involute, but, in loaded teeth operating at the gear pair's rated load, profile modifications in the form of tip and root relief are desirable variations from a perfect involute. Hence, profile deviations are thought of as deviations from the specified profile shape. Again, the AGMA accuracy standard allows one to specify deviations in terms of slope and curvature. Profile deviations tend to affect tooth-to-tooth load sharing across the profile of the tooth pair.

Bias deviation: Although not spelled out in AGMA standards, this type of deviation, which is essentially a twisting of the tooth form, is identified by performing multiple profile and/or lead measurements on each measured tooth. This type of deviation, which commonly occurs when gears are finished by screw type generation grinding, also affects load sharing.

Spacing deviations: Tooth to tooth spacing deviations may affect dynamics, but have a greater effect on tooth to tooth load sharing [15]. In this paper, AGMA quality number values are used and these load sharing effects are analyzed.

Runout deviations: Runout results from eccentricities both in the manufacture and the assembly of gears. The most common form of runout is radial runout, which manifests itself in terms of cyclic spacing deviations, cyclic changes in the profile slope, and cyclic changes in the operating center distance and/or the effective outside diameter. The latter two effects slightly change the profile contact ratio of the gear pair. In the analysis of this paper, only the spacing deviation effect of profile runout is considered. Another form of runout commonly referred to as lead runout or lead wobble occurs in the face width direction. It is assumed that lead wobble effects are already included in the tolerance used for lead deviations so no special analysis of lead wobble is performed in this paper.

AGMA rating equations

Since this study concentrates on the effects of manufacturing deviations on stresses, we first look at the current AGMA method for computing these stresses and discuss in general how each factor in these stress equations is affected by manufacturing deviations.

The AGMA stress formulas [1] for bending and durability are respectively given below:

Contact stress equation:

$$s_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m C_f}{d F I}}$$

Tensile bending stress equation:

$$s_t = W_t K_o K_v K_s \frac{P_d K_m K_B}{F J}$$

where,

- s_c contact stress number, lb/in²;
- C_p elastic coefficient, [lb/in²]^{0.5};
- W_t transmitted tangential load, lb;
- K_m load distribution factor;
- K_o overload factor;
- K_v dynamic factor;
- K_s size factor;
- C_f surface condition factor for pitting resistance;
- F net face width of narrowest member, in;
- I geometry factor for pitting resistance;
- d operating pitch diameter of pinion, in;

- s_t tensile stress number, lb/in²;
- K_B rim thickness factor;
- J geometry factor for bending strength;
- P_d transverse diametral pitch, in⁻¹.

The factors that are unlikely to be affected much by manufacturing (deviations in materials are not considered in this paper) include the elastic coefficient, the overload factor, and the size factor. As mentioned earlier, the dynamic factor has been studied extensively [2-7] and is the only factor used in the above formulas that is currently based on gear quality. The load distribution factor certainly is affected by misalignment and profile and lead modifications. In a way, the current use of this factor does depend upon accuracy, but this use is not quantified in terms of the accuracy numbers. In the simulations of this paper, the load distribution factor is evaluated based on quality numbers. The surface condition factor is certainly affected by manufacturing, but also is not considered by AGMA in its accuracy definitions and will not be studied in this paper. Both geometry factors are subject to manufacturing tolerances, but again they are not quantified in the AGMA tolerance numbers so will only be briefly discussed. The bending strength geometry factor is subject to changes in surface finish and shape imperfections in the root fillet region that could affect root stresses. For instance, this author once had some gears made in which the hob feed rate was varied and the teeth with the coarser feed rate were found to have lower lives based on single tooth fatigue testing [16]. With regard to the surface durability geometry factor, it has been shown that the contact stress may increase significantly at locations on the tooth flank where there are abrupt changes in the tooth form, an example being the radius of curvature change in the tip relief "break" region [17]. The rim thickness factor is subject to manufacturing to the extent that there are tolerances on the thickness value that could affect root stresses.

Baseline microgeometry selection

When studying microgeometry variations, one must first start with a baseline microgeometry. This, in essence, means to define the baseline profiles and leads of the design. Every gear designer has their own approach to establishing these parameters. Some designers might choose their profile so as to

avoid corner tooth contact and tip interference; others might minimize transmission error and others may wish to minimize the effects of spacing deviations. When selecting lead variation, some designers use no lead crown, others select a standard amount of lead crown based on experience and still others might prefer end relief rather than lead crown. In this study, we shall use a load distribution simulation [18–20] to select both the profile and lead modifications that will avoid corner contact and tip interference and at the same time provide reasonable insensitivity to misalignment.

The basic gear geometry to be used in this paper is given in Table 1: The geometry is similar to one used in a previous study [21] except that the root diameters have been adjusted slightly.

Table 1. Helical gear geometry

	Pinion	Gear
Number of teeth	25	31
Normal diametral pitch (1/in)	8.598	
Normal pressure angle (deg)	23.45	
Helix angle (deg)	21.50	
Center distance (in)	3.50	
Face width (in)	1.25	1.25
Outside diameter (in)	3.360	4.110
Root diameter (in)	2.811	3.561
Standard pitch diameter, d_p (in)	3.125	3.875
Transverse tooth thickness at d_p (in)	0.1934	0.1934
Profile / face contact ratio	1.383 / 1.254	
Total contact ratio	2.637	
Pinion torque, lb-in	5000	

The procedure for coming up with an acceptable microgeometry is as follows:

Step 1. Identify the rough torque rating for the gear pair. For the sample gear set, we used an AGMA rating formula with approximate constants to come up with a torque rating of roughly 5000 lb-in.

Step 2. Using this rating, a load distribution analysis with perfect involutes (shafts not included in the analysis) is performed. Figure 1 shows the load dis-

tribution at one contact position for the perfect involute analysis. One observes that there is significant contact at the tooth tips (tip interference) and at the entering corner of the tooth (corner contact). Figure 2 shows a composite plot of contact stresses that result from the analysis of many positions of contact. The stresses at the tip and root and at the corners are abnormally high due to the tip interference and corner contact and due to the fact that the radius used to compute the contact stress at the tip is much smaller than that along the tooth flank. Peak contact stresses are about 240 ksi in the corners, 200 ksi on the tip edge and 170–180 ksi in the tooth center. From Figure 3, which shows root stresses at 5 equally spaced locations along the root of the tooth, the peak pinion root stress is about 44 ksi. The root stresses of the gear were quite similar, so in all subsequent analyses, we shall only observe the pinion root stresses.

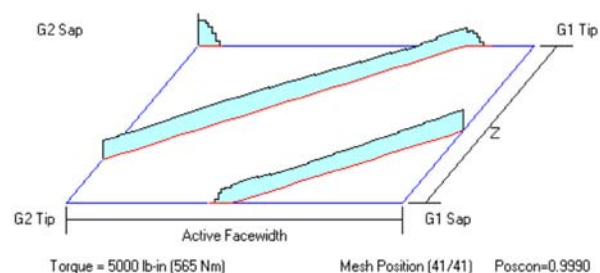


Figure 1. Load distribution of a perfect involute with 5000 lb-in torque

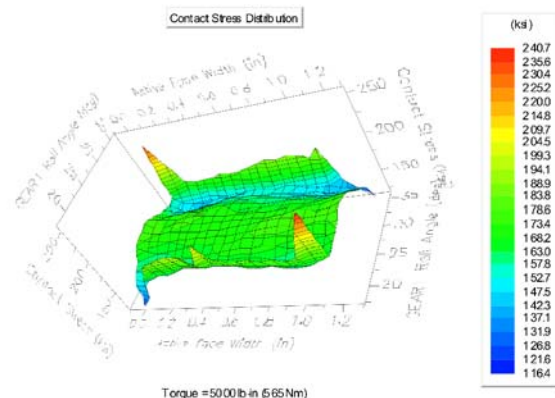


Figure 2. Contact stress distribution of a perfect involute with 5000 lb-in torque