

Comparing Standards

The keys to understanding ISO 6336-1 gear rating

Don McVittie

This is the second of a series of articles introducing the new ISO 6336 gear rating standard and its methods of calculation. The opinions expressed herein are those of the author as an individual and not necessarily those of any organization of which he is a member.

One of the best ways to learn the ISO 6336 gear rating system is to recalculate the capacity of a few existing designs and to compare the ISO 6336 calculated capacity to your experience with those designs and to other rating methods. For these articles, I'll assume that you have a copy of ISO 6336, you have chosen a design for which you have manufacturing drawings and an existing gear capacity calculation according to AGMA 2001 or another method. I'll also assume that you have converted dimensions, loads, etc. into the SI system of measurement.

ISO 6336 looks into more details of your design, so it requires about 80 input values compared to about 60 for AGMA 2001. The additional information required includes details about things like blank construction, root fillet finish, lubricant viscosity, arrangement of the gear set with respect to the bearings and location of light load contact pattern with respect to the bearings. We'll cover these differences as they apply, section by section.

Input Data

Most of the input data is similar to other rating standards, like AGMA 2001. The basic gear

geometry values of numbers of teeth, center distance, outside diameters and face widths are the same. ISO 6336 requires the input of minimum safety factors for surface contact (pitting) and root bending stresses. The safety factors are used in the calculation of allowable working stresses in parts 2 and 3. While safety factors as low as 1.0 are permitted, most users will want to select a higher value for root bending, such as 1.2 or more. If the required safety factor is not achieved with the input values of materials, geometry and load, one or more of the inputs should be changed until the required safety factor is reached.

Module, normal pressure angle and helix angle are determined at the reference (generating) diameter, based on a carefully defined basic rack geometry per ISO 53. The basic rack is a gear with an infinite number of teeth whose reference plane is located where the tooth thickness and the space width are equal. The basic rack is not the cutting tool. It is conjugate to a hypothetical rack shaped tool which defines the tooth shape.

Tooth thickness is defined in terms of the profile shift factor X of a "zero-backlash" gear pair derived from the basic rack geometry. (See the article in July/August 1993 and a correction to one of the formulas in January/February 1994 for more information on basic racks and profile shift.) The X factor and the tool geometry—tool addendum, tip radius, protuberance and finish stock allowance—are critical to the calculated result. Figs. 1 & 2, from the AGMA ISO 6336 program user's manual, illustrate the principle. It is important to note that the definition of tool addendum used in ISO 6336 may not be the same as you are using in your present gear rating program, so that value may have to be adjusted for input to the ISO 6336 calculation. Fig. 3 illustrates the ISO definition.

The input power or torque must be specified, since the ISO dynamic and load distribution factors are load dependent. Although the ISO standard lists numerical application factor values which are identical to AGMA's, those values should be used with caution. The ISO definition of application factor is broader than AGMA's, including many "dynamic" effects, such as the effect of tooth spacing deviations which are part of the AGMA dynamic factor. The differences are most noticeable in the calculat-

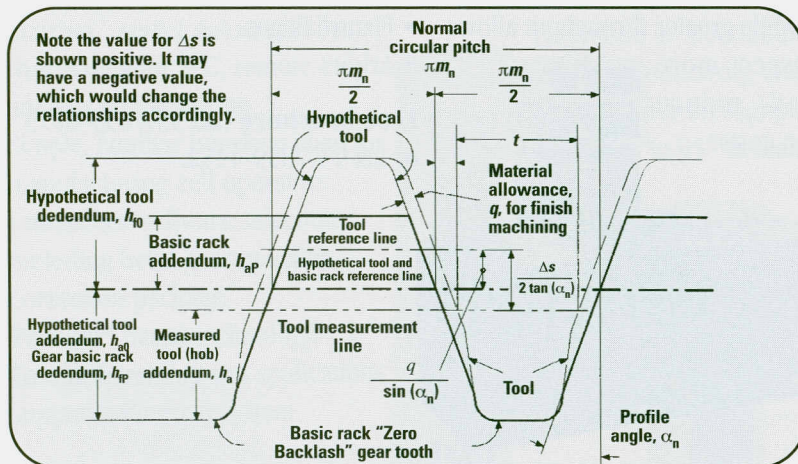


Fig. 1 — Basic rack and hypothetical tool represented in rack form (©AGMA)

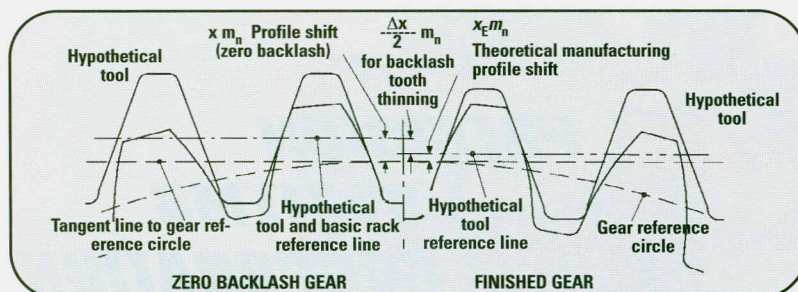


Fig. 2 — Hypothetical tool with zero backlash and finished gear (©AGMA)

ed capacities of large, low speed, relatively less accurate gears, such as those found in bridges, mill drives, train positioners and antennas. The ISO dynamic factor estimates gear pair resonance as an equivalent single mass system with the mesh stiffness as a spring. The ISO dynamic factor is strongly influenced by the ratio between the mesh pass frequency and the calculated resonant frequency of the single mass model. High speed, lightly constructed, lightly loaded gear sets have high ISO dynamic factors and low speed, heavily constructed, heavily loaded gear sets have very low ISO dynamic factors. Many of the "dynamic" effects which AGMA includes in its dynamic factor are shifted to the application factor by ISO. You will need to be careful in your evaluation of the differences and in the application factor you choose to evaluate your gear sets. The product of $K_A \cdot K_V$ should be nearly the same in both systems if rated gear capacities are to be comparable.

Calculating the Dynamic Factor

The ISO dynamic factor calculation estimates the rotational moments of inertia of the pinion and wheel from the inside diameters of their rims and their calculated root diameters with an adjustment for webs, if any. The gear blank geometry must be provided for this calculation. The mesh stiffness and root diameters are calculated from the X factors and tool geometry, so accuracy of these input values affects the dynamic factor. A smaller portion of the dynamic factor is a function of tooth accuracy, expressed as an ISO 1328 quality number. There is no consistent way to estimate ISO 1328 quality from AGMA 2000 quality numbers, so the actual tolerances must be known or calculated, then ISO 1328 quality numbers recalculated from the tolerances. The fundamental equation in ISO 6336-1 for dynamic factor is:

$$K_v = (N \cdot K) + 1$$

Where:

N is the ratio of mesh pass frequency to resonant frequency.

K represents the effect of gear tooth accuracy.

Typical values of K_v for industrial enclosed drive gears are 1.05 or less.

The basic ISO dynamic factor is Method B of ISO 6336-1. Methods C and D are simplified versions of method B with applicability restricted by their underlying simplifying assumptions.

The AGMA dynamic factor is included in ISO 6336-1 as alternative method E for those who prefer it. If method E is used, it might be appropriate to use a lower (AGMA) value for application factor.

Evaluating Load Distribution

Much of ISO 6336-1 is devoted to various methods of evaluating load distribution across

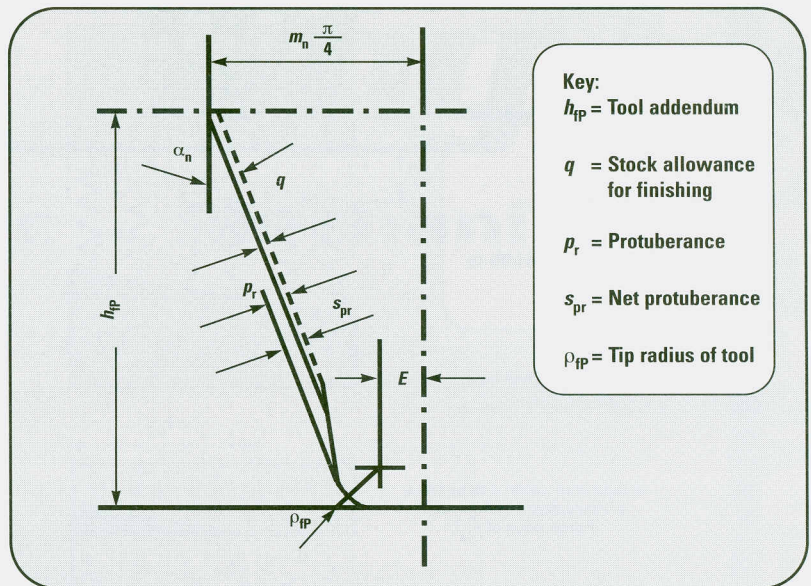


Fig. 3 — Dimensions and basic rack profile of the teeth (finished profile with undercut).

and between the teeth. Although different influence factors are defined for load distribution in root bending stress and contact stress in the face and transverse directions, the principal influence on the load capacity of the gear set is $K_{H\beta}$, the face load distribution factor for contact stress. $K_{F\beta}$, the face load distribution factor for root bending stress has a similar value and the transverse load distribution factors $K_{H\alpha}$ and $K_{F\alpha}$ are usually nearly 1.0.

The correct evaluation of the load distribution factor is critical to obtaining satisfactory results with ISO 6336, just as it is with other gear rating systems, such as AGMA 2001. I suggest that you try the various optional methods in ISO 6336 while making comparative capacity calculations with your own gear designs, to develop experience with them. Since the mathematical definition of face load distribution factor is the same in ISO 6336 as in AGMA 2001, you should expect similar values of this factor from either method. If the values are not within 10% of each other, you should investigate further and resolve the difference. It may be necessary to make an experimental investigation or a detailed calculation of deflections under load to get the "right" answer. It is important to consider the effects of manufacturing variations as well as the average meshing conditions in this analysis. Method A is commonly used to evaluate $K_{H\beta}$ —assuming that a mathematical model gives satisfactory results. Recent investigations by an ad hoc group sponsored by ISO/TC60/SC2/WG6 in which the same gear sets are calculated by different "correct" mathematical models have shown that the value of $K_{H\beta}$ can vary more than 10% depending on small variations in the calculation method and the underlying assumptions. Future editions of ISO 6336-1 are

- Key:
- h_{IP} = Tool addendum
 - q = Stock allowance for finishing
 - p_r = Protuberance
 - s_{pr} = Net protuberance
 - ρ_{IP} = Tip radius of tool

Don McVittie

is one of Gear Technology's technical editors. He is president of Gear Engineers, Inc., Seattle, WA and a former president of AGMA. McVittie is a licensed professional engineer in the state of Washington and has been involved with gear standards development for more than 25 years.

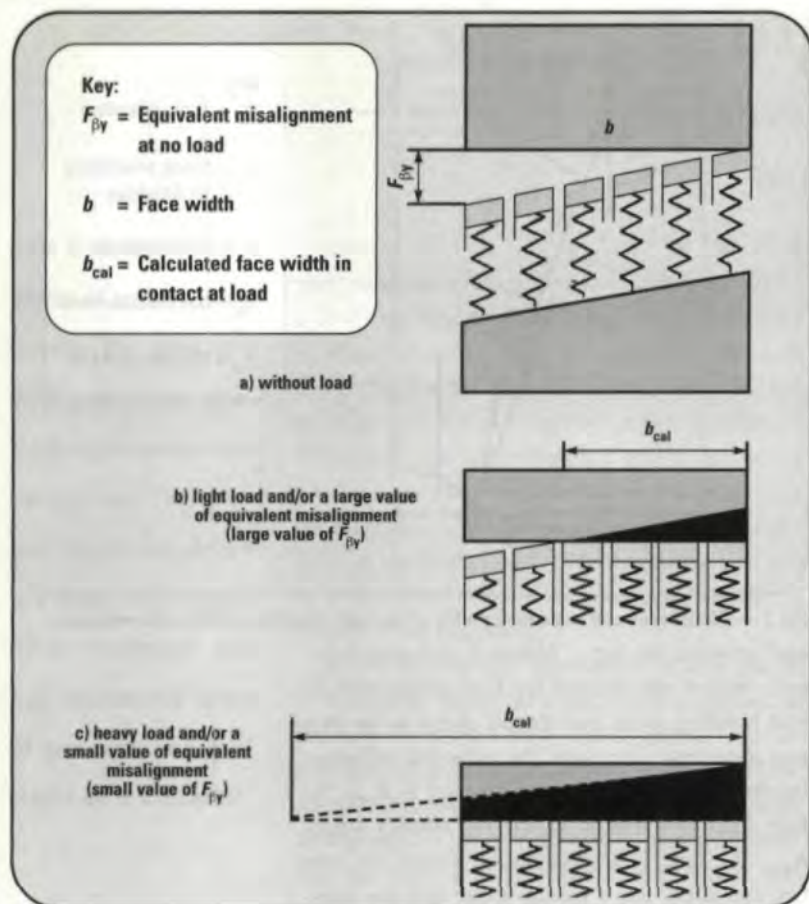


Fig. 4 — Distribution of the load along face width with linear equivalent misalignment.

expected to specify a more detailed calculation method as method B and to restrict method A to experimentally verified values of K_{HB} . In general, values of C_m calculated by AGMA 2001's empirical method are somewhat more conservative than the values of K_{HB} commonly used as method A by European manufacturers of industrial enclosed drives.

Methods B, C and D are based on a simple mathematical model of mesh deflection under load. The mesh stiffness constant is the same as the spring constant used in the method B dynamic factor. That constant is about 1/3 higher (more stiff) than the factor which was used in the AGMA standards, so load distribution factors calculated by methods B, C and D tend to be higher than AGMA values.

Method B assumes a linear load distribution model with constant mesh stiffness, similar to the analytical method which was used in pre-1995 editions of AGMA 2001 (See Fig. 4). The fundamental input is the total loaded mesh misalignment including the effects of manufacturing variations, and elastic deflections due to load. The deflection of the wheel shaft can optionally be included in the total mesh misalignment. Method B doesn't implicitly recognize the benefits of tooth crowning, but it does recognize the benefits of running in and helix modification to compen-

sate for torsional and bending deflections of the pinion. The evaluation of total misalignment is very important, since it has a large influence on K_{HB} and the calculated capacity of the gear set. If you elect to use method B, be sure to read all of the detailed instructions and limitations which are included in section 7 of ISO 6336-1.

The AGMA computer program for ISO 6336 allows the input of a K_{HB} value (method A), the input of a mesh misalignment (method B) or the input of detailed information about the mesh to calculate K_{HB} according to method C. Method C is limited to the case where the center of the pinion is within 30% of the center of the bearing span. Gear sets which do not meet this limit must be calculated by methods A or B. A subset of method C (C1) is a further simplification for symmetrically located pinions, but the same inputs are required. Method C considers the gear set arrangement, pinion shaft deflection, type of crowning or lead correction and the location of the light load contact pattern, so inputs for all of those factors are required. Method C is a bit complicated to use in hand calculations but, once programmed, allows the user to see the effects of changing pinion proportions and crown or lead corrections. Method C can give results which are similar to the empirical method of AGMA 2001 for heavily loaded narrow face width pinions. Lightly loaded wider faced pinions can be heavily penalized. Method D is a further simplification of method C1 for symmetrically located pinions.

This concludes our exploration of ISO 6336-1. It contains the most important influence factors, which are also the most difficult to evaluate and which have the most potential to be different from AGMA or other calculation methods.

The calculation methods for surface contact (pitting) stress and root bending stress in ISO 6336 are very similar to AGMA 2001. The nominal stresses, which depend only on load and geometry, have similar values. The combined effects of the general influence factors from ISO 6336-1 modify the nominal stresses to "calculated" values which have essentially the same definition as AGMA's "stress numbers." Most of the differences between gear ratings by ISO 6336 and AGMA 2001 are explained by the general influence factors of ISO 6336-1. These general influence factors offer the greatest opportunity for further research and improvement of the standard. Users of the standard will need to exercise good judgment in picking the calculation methods to be used and the values of input variables in order to be satisfied with the calculated results. ☉

We will be back on more familiar ground in the next article, which looks at the details of calculating surface contact and root bending stresses in ISO 6336-2 and -3.

Tell Us What You Think . . .

If you found this article of interest and/or useful, please circle 202.