# AGMA, ISO, and BS Gear Standards Part I — Pitting Resistance Ratings

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**Summary:**

A study of AGMA 218, the draft ISO standard 6336, and BS 436:1986 methods for rating gear tooth strength and surface durability for metallic spur and helical gears is presented. A comparison of the standards mainly focuses on fundamental formulae and influence factors, such as the load distribution factor, geometry factor, and others. No attempt is made to qualify or judge the standards other than to comment on the facilities or lack of them in each standard reviewed. In Part I a comparison of pitting resistance ratings is made, and in the subsequent issue, Part II will deal with bending stress ratings and comparisons of designs.

### Introduction

Standard spur and helical gears are usually designed to specific standards to meet the requirements of proportions, manufacturing accuracy, and load rating. The load rating is the most important issue discussed in AGMA (American Gear Manufacturers Association), ISO (International Standards Organization), DIN Deutsche Industrie Normen) and BSI (British Standards Institution) gear standards. The standards written by these organizations are widely used for gear design throughout the world and also form the basis of "minority" gear standards. China, for example, issues a gear design standard based on ISO. European gear standards are now becoming very similar. The new BS and the draft ISO standard share much in common with DIN. This paper considers BS 436:1986, the draft ISO standard 6336, and AGMA 218.01. Since this review was written, AGMA introduced AGMA 2001-B88, although this new standard is not considered here. However, the trend toward a universal standard continues, with AGMA 2001 publishing rating factors, some of which are similar to the draft ISO standard.

This article is intended for designers who will appreciate a review of this complex subject. Many designers in the USA will still use AGMA 218 because they are familiar with it, and will, we suspect, continue to do so for some time. (This situation also exists in the UK with respect to the old and new British Standards on gear ratings.) It will take the authors some time before they have enough experience in the use of AGMA 2001 and have been able to validate it against real designs and other rating standards. While there are marked similarities between the old and new AGMA formulas for pitting resistance and bending strength ratings, we have mainly excluded any reference to AGMA 2001.

Although, theoretically, any standard will produce a gear pair which is satisfactory, it is no longer enough to accept any standard when other procedures might produce more competitive designs. On the other hand, if the design calls for special operating conditions, such as shock loads or flexible drives, it may be advantageous for the designer to address a standard which deals more closely with these conditions. In addition, the customer may specify the code to be used. A working knowledge of more than one standard is desirable, particularly if the product is aimed at international markets. An understanding of the differences between gear standards is, therefore, important. It should be pointed out, however, that in a review of gear standards it is impossible to cover every aspect of each code. ISO 6336 Parts 1 to 4, for example, contain over 90 figures and over 20 tables.

### Standards for Spur and Helical Gears

The old British Standard, BS 436:1940,[1] in use for nearly fifty years, was a revision of the original Specification for Machine Cut Gears first issued in 1932. During its long existence, the rating method remained the same with only minor revisions. Though obsolescent, it is still used extensively throughout Britain and elsewhere, mainly because it is easy to use. The standard rates gears on the basis of bending strength and contact stresses, which are referred to as wear (meaning non-abrasive wear). The tooth root bending strength is based on the Lewis equation,[2] considering both tangential and radial loads. The effect of stress concentrations at the root is not taken into account directly, but allowances are made in the use of the allowable bending strength of the gear material, values for which are supplied in the standard. The bending strength is also factored for running speed and life. Contact stresses are based on a modified Hertz equation with allowances for speed, running time, and geometry. The latter item is taken into consideration by a zone factor, which accounts for the influence on the Hertzian stress of tooth flank curvature at the pitch point, and converts the tangential load to a normal force. No serious attempt was made to keep this standard up to date on newer gear materials and processes to predict the higher performances being achieved in practice. Another serious deficiency is that no account was made for surface finish or uneven load distribution.

## NOMENCLATURE

**BS 436:1940**

| | |
|---|---|
| $d$ | Pitch diameters of pinion and wheel |
| $E_e$ | Equivalent Young's modulus |
| $F$ | Face width |
| $k$ | Constant in Hertz contact stress formula |
| $K$ | Pitch factor |
| $n, N$ | Pinion and wheel running speed |
| $P$ | Diametral pitch |
| $R$ | Gear ratio |
| $R_r$ | Relative radius of curvature |
| $s$ | Maximum contact stress set in the surface layers of the gear cylinders |
| $S_c$ | Surface stress factor |
| $T$ | Number of teeth on wheel |
| $X_c$ | Speed factor for contact stress |
| $Z$ | Zone factor |
| $\alpha_t$ | Transverse pressure angle at reference cylinder |
| $\alpha_{tw}$ | Transverse pressure angle at pitch cylinder |
| $\beta_b$ | Base helical angle |

**AGMA 218.01**

| | |
|---|---|
| $C_a$ | Application factor for pitting resistance |
| $C_c$ | Curvature factor at pitch line |
| $C_f$ | Surface condition factor |
| $C_H$ | Hardness ratio factor |
| $C_L$ | Life factor for pitting resistance |
| $C_m$ | Load distribution factor for pitting resistance |
| $C_p$ | Elastic coefficient |
| $C_R$ | Reliability factor for pitting resistance |
| $C_s$ | Size factor for pitting resistance |
| $C_T$ | Temperature factor for pitting resistance |
| $C_v$ | Dynamic factor for pitting resistance |
| $C_x$ | Contact height factor |
| $C_\psi$ | Helical overlap factor |
| $d$ | Operating pitch diameter of pinion |
| $F$ | Net facewidth of the narrowest number |
| $I$ | Geometry factor for pitting resistance |
| $m_N$ | Load sharing ratio |
| $n_p$ | Pinion running speed |
| $P_n$ | Diametral pitch |
| $S_{ac}$ | Allowable contact stress number |

**BS 436: 1986 and ISO/DIS 6336**

| | |
|---|---|
| $b$ | Face width |
| $d_1$ | Reference diameter of pinion |
| $K_A$ | Application factor |
| $K_{H\alpha}$ | Transverse load factor for contact stress |
| $K_{H\beta}$ | Face load factor for contact stress |
| $K_V$ | Dynamic factor |
| $n_1$ | Pinion running speed |
| $S_{Hmin}$ | Minimum demanded safety factor on contact stress |
| $Z_M$ | (BS only material quality factor for contact stress) |
| $Z_N$ | Life factor for contact stress |
| $Z_X$ | Size factor for contact stress |
| $Z_E$ | Elasticity factor for contact stress |
| $Z_H$ | Zone factor for contact stress |
| $Z_L, Z_R, Z_V$ | Lubricant influence, roughness, and speed factor for pitting |
| $Z_W$ | Work-hardening factor for contact stress |
| $Z_\epsilon$ | Contact ratio factor for contact stress |
| $Z_\beta$ | Helix angle factor |
| $\alpha_t$ | Transverse pressure angle at reference cylinder |
| $\alpha_{tw}$ | Transverse pressure angle at pitch cylinder |
| $\beta_b$ | Base helix angle |
| $\sigma_{Hlim}$ | Basic endurance limit for contact stress |
| $\sigma_{HP}$ | Permissible contact stress |

Smith[3] described BS 436 as "an average experience method, whereby gear manufacturers and users collaborate to provide extremely empirical rules of thumb based on operating experience. Permissible loads are specified for 'typical' manufacturing accuracies of a given class with 'typical' loading cycles and corrections for speed, etc." Although the standard did not take into account factors known to influence bending and contact stresses, such as application conditions (i.e., the load fluctuations caused by external sources), system dynamics and gear accuracy and the benefits of surface hardening, the standard served its users well.

The original AGMA standard was issued in 1926, and the first draft of AGMA 218.01,[4] used in this review, was drafted in 1973 and approved for publication in 1982. AGMA 218 also rates gears on the basis of bending strength and surface contact stresses, (referred to as surface durability or pitting) but also introduces a number of other factors in the rating equations. For example, influence factors are used to take into account load distribution across the face width, quality of the transmission drive, and transmission accuracy relating to manufacture. Considerable knowledge and judgment is required to determine values for these factors.

Compared to the old British Standard, AGMA 218 is considerably more comprehensive. Ratings for pitting resistance are based on the Hertzian equation for contact pressure between curved surfaces, which is modified for the effect of load sharing between adjacent teeth. The Lewis equation has been modified to account for effects, such as stress concentrations at the tooth root, compressive stresses resulting from the radial component of the gear load, load distribution due to misalignment between meshing teeth, and load sharing. The critical bending stress is assumed to occur at the tooth fillet but, as in the old British Standard, the effect of blank geometry (e.g., rim and web size and how the relative size of these effects stresses at the tooth root) is not considered.

The ISO standard, ISO 6336,[5] was issued in 1980, though it is still a draft. The standard covers a wide range of designs and applications, and is the most detailed document among the gear rating standards considered here. It contains a vast amount of collected knowledge and the options to calculate factors at various levels of complexity. It gives procedures for determining gear capacity as limited by pitting and tooth breakage, as in other standards, and also considers

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scuffing. The basic equations are modified by applying influence factors as in AGMA. These procedures demand a realistic appraisal of all influence factors, particularly those for the allowable stress, the probability of failure, and the appropriate safety factor. ISO also offers three different methods for determining bending and contact stresses and their influence factors, depending on the application and accuracy required. Since the latest German standard, DIN 3990: 1987,[6] is substantially similar to ISO 6336, a detailed discussion of DIN is excluded.

The new British standard, BS 436:1986,[7] is similar to ISO/DIS 6336 and is a complete revision of the old standard. It is, however, much more user friendly than ISO. Like the other standards, the new British standard uses modified Lewis and Hertz equations, using correction factors, such as the dynamic factor to account for load fluctuations arising from manufacturing errors, and a load distribution factor to take into account the increase in local load due to non-uniform loading arising from conditions such as shaft stiffness and, in the case of helical gears, helix error. The correction factors are the same or are similar to those used by ISO. Geometry factors in the new BS 436 are similar to those in ISO (method B) while other methods in ISO use different approaches for geometry factors. The new British standard, however, draws on a considerable amount of previously published research and uses additional parameters, like material quality factors, not allowed for by ISO or AGMA. This standard does not work on "typical" figures for each rating factor as in the old standard, but uses researched data to predict load increases caused by deflections, alignment tolerances, and helix modifications. Throughout this review, the term BS refers to the new standard, except where stated.

In the stress analysis procedures of BS and ISO, bending and contact stresses are classified into three groups: 1) Nominal or basic stresses, which are calculated for geometrically perfect gears meshing with perfect load distribution, 2) Actual stresses, calculated from the nominal stresses, but allowing for manufacturing and mounting errors, and 3) Permissible stresses, calculated for the gear material taking into account the required life, gear finish, lubrication, and the minimum specified factors of safety. The BS differs significantly from ISO and DIN in the determination of permissible stresses.

### Literature Survey

Comparisons have been made between AGMA and ISO covering basic theories and results for applications. Those comparisons that were published were mainly based on old versions of AGMA (AGMA 215.01 and AGMA 225.01)[8,9] and the older, approved version of ISO/DP 6336. No detailed comparisons have been made between BS 436 and other standards. More recently some comparisons have been made between the latest British and German standards by Hofmann,[10] who described the theoretical basis of the latest BS and DIN and the differences in determining permissible stresses.

Imwalle and Labath[11] made a design survey of different gear sets for the purpose of comparing AGMA (AGMA 215 and AGMA 225) and ISO/DP 6336. The results were summarized for a comparison of dynamic load distribution and geometry factors and allowable stress. In the comparison of geometry factors, all the factors which are linked with gear tooth geometry were combined to form a "total geometry factor". Comparisons showed that ISO usually gives a higher factor of safety and higher calculated bending and contact stresses for case-hardened gears compared to AGMA, but lower values for through-hardened examples. It may be noted that ISO provides data on the latest and most advanced gear materials and treatments. In another paper[12] by the same authors the concept of a basic stress was used, defined as the stress which is calculated if all the modifying factors are set at unity. The results showed that ISO usually gives a higher basic bending stress, but a lower basic contact stress compared to AGMA. Again, comparisons were made of geometry, dynamic load distribution factors, life factors, and allowable stresses.

Mathematical means were provided by Castellani and Castelli[13] to compute the parameters for calculating the tooth form factor and the stress correction factor (allowing for stress concentrations at the tooth fillet) which are used in AGMA and ISO. Comparisons made between these two factors in the gear strength ratings gave the following two differences:

1) A different choice of reference points on the tooth root profile for the tooth form and stress correction factors is made. ISO chooses the same critical point for both tooth form and stress correction factors relating to the point of the fillet whose tangent forms a 30° angle with the tooth axis. The critical point for the tooth form factor depends on the gear type (spur or helical) and its accuracy. AGMA considers the minimum curvature radius for the stress correction factor relating to the point where the fillet connects to the root circle.

2) Both standards assume that the load application side of the tooth flank is critical with respect to bending failure. AGMA takes this assumption into consideration by subtracting the radial, compressive stress component from the bending stress, while ISO only considers the tangential bending stress. ISO compensates for this by making allowances on the values of the stress correction factor.

### Comparison of Pitting Resistance Ratings

Comparing gear standards can present difficulties for the following reasons:

1) There are a number of influence factors included in each standard, but the number and the numerical values of these factors differ. Taking the power rating formula for pitting resistance as an example, BS 436:1940 has four influence factors, while there are twelve in AGMA 218, compared with sixteen in both ISO and the BS. These are discussed later.

2) The determination of influence factors usually requires a knowledge of additional parameters, some of which are not always readily available. For example, in order to use the analytical method to calculate the AGMA geometry factor for pitting resistance, four additional data items (a curvature factor at the pitch line to determine the radius of curvature between the two contact surfaces; a contact height factor to adjust the location of the height of the tooth profile where the stress is calculated; a helical factor to account for the effect of helix angle on contact stresses; and a load sharing ratio) have to be employed. To determine these four factors more infor-

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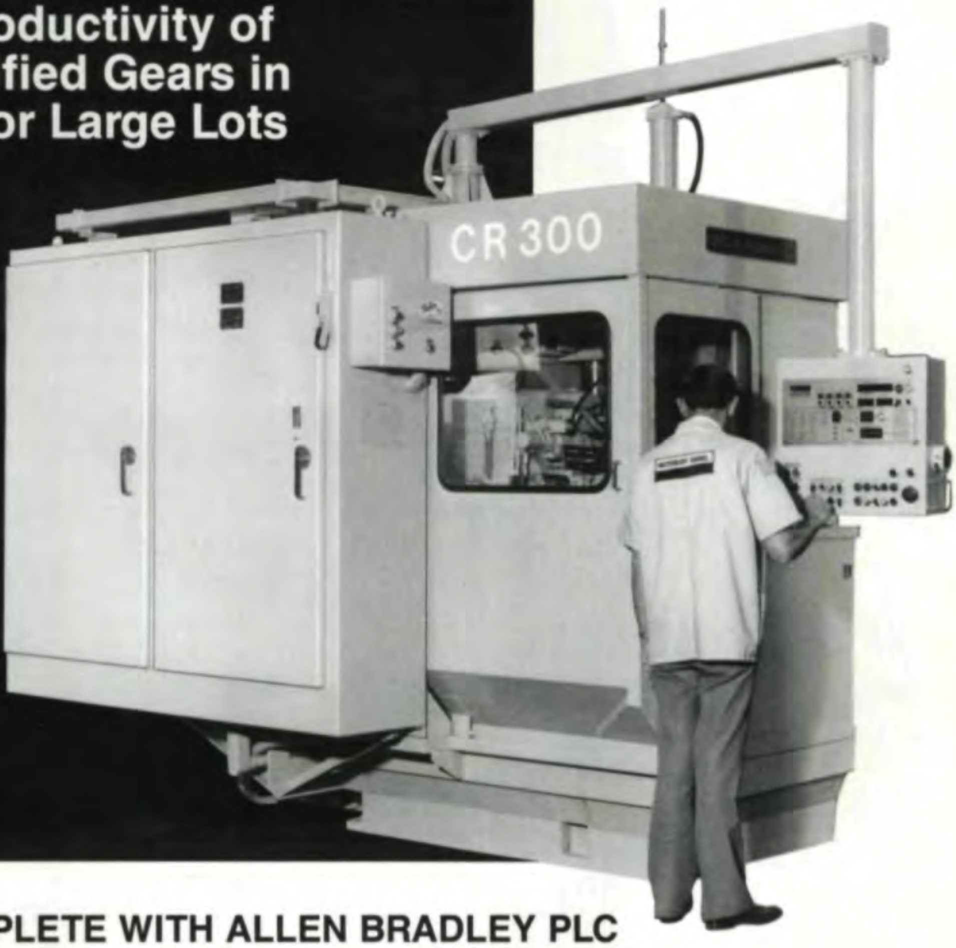


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mation is required and some, like generating the tooth layout to give the necessary geometry data, may be difficult to obtain.

3) Each standard has its own definitions for the influence factors, but factors bearing the same names do not necessarily have the same effects. This makes direct comparisons of influence factors difficult. For example, although AGMA and ISO both introduce a service factor, the values cannot be compared directly.

The terms used in each standard are listed in the nomenclature.

The power rating for contact stresses given by BS436:1940 is

$$\frac{X_c\, S_c\, Z\, F\, N\, T}{126\,000\, K\, P} \tag{1}$$

Equation 1 can be rewritten as

$$\frac{n\, F\, d^2}{126\,000}\ \frac{R_r^{\,0.8}}{d}\, X_c\ \frac{s^2}{k\, E_e} \tag{2}$$

For AGMA 218.01 the transmissible power based on pitting is

$$\frac{n_p\, F\, d^2}{126\,000}\ \frac{I\, C_v}{C_s\, C_m\, C_f\, C_a}\ \left(\frac{S_{ac}\, C_L\, C_H}{C_p\, C_T\, C_R}\right)^2 \tag{3}$$

*The BS 436:1986 power rating is*

$$\frac{b\, d_1^{\,2}\, n_1}{126\,000}\ \frac{u}{u+1}\ \frac{1}{Z_H^{\,2}\, Z_\varepsilon^2}\ \frac{1}{K_A\, K_v\, K_{H\alpha}\, K_{H\beta}}\ \left(\frac{\sigma_{HP}}{Z_E}\right)^2 \tag{4}$$

where $\sigma_{HP} = \sigma_{Hlim}\, Z_L\, Z_V\, Z_R\, Z_M\, Z_W\, Z_X\, Z_N / S_{Hmin}$ (5)

and for ISO/DIS 6336 the power is

$$\frac{b\, d_1^{\,2}\, n_1}{126\,000}\ \frac{u}{u+1}\ \frac{1}{Z_H^{\,2}\, Z_\varepsilon^2\, Z_\beta^2}\ \frac{1}{K_A\, K_v\, K_{H\alpha}\, K_{H\beta}}\ \left(\frac{\sigma_{HP}}{Z_E}\right)^2 \tag{6}$$

where

$$\sigma_{HP} = \left(\frac{\sigma_{Hlim}\, Z_N}{S_{Hmin}}\right) Z_L\, Z_R\, Z_V\, Z_W\, Z_X \tag{7}$$

From Equations 2-7 it may be seen that for a given gear ratio the transmissible power is proportional to the square of the pinion pitch circle diameter and permissible contact stresses. Therefore, to increase gear power capacity in terms of surface durabilty, it is more effective to increase the pinion PCD or permissible surface stresses than to increase the gear pair facewidth.

The pitting resistance related factors above may be grouped into common and non-common factors. Common factors are those which have the same meanings in all the standards, (not all these factors appear in the standards) and their values can be compared directly. For example, the dynamic factor which allows for internally generated gear tooth loads induced by non-conjugate meshing action of the gear teeth, appears in all the standards except the old BS, and values can be compared directly. Non-common factors, such as the geometry factor, are those which are only equivalent to each other in the sense of having the same effects on the rating results, although specific values cannot be compared.

Table 1 classifies all the contact stress influence factors into either common or non-common groups where it can be seen that the old British Standard had few parameters for comparison with other standards. The similarity between BS and ISO *is also apparent. A comparison between the influence* factors given in each standard is made in the following paragraphs.

1) Application factors. An application factor is used in AGMA, ISO, and BS to evaluate external influences tending to apply a greater load to the gear teeth than that based on steady running conditions. Typical external influences are the drive characteristics (e.g., smoothness and load fluctuations) of the prime mover and of the driven machine. Values suggested in ISO correspond to those for the overload factor in AGMA 215. While no data appears in AGMA 218, these factors may be found in related AGMA application standards. BS gives more detailed conditions than ISO, although values are similar.

2) Dynamic factors. As discussed above, the application factor is used to handle dynamic loads unrelated to tooth accuracy. The effect of dynamic load related gear tooth accuracy is then evaluated by the inclusion of a dynamic factor which accounts for effects of gear set mass elastic effects and transmission errors. AGMA modified the experimental work of Wellauer[14] to obtain dynamic factors as a function of transmission error. These accuracy levels can under certain manufacturing conditions be the same as the gear quality numbers given in AGMA 390.[15]

ISO dynamic factors are based on Buckingham's incremental load method[16] and work by Weber and Banaschek.[17] ISO (analytical) method B presents a calculation procedure for the main resonance speed and divides the running speed into three sectors. The dynamic factor corresponds to each of these speed sectors and may help designers to adjust the operating speed or alter the design to avoid critical speeds. ISO method C is only applicable to gears with accuracy numbers of 3 to 10 and cannot be used for gears operating at or near the main resonance speed. The dynamic factor in the BS is very close to ISO method C. In the old BS, dynamic effects were not considered. The speed factor used in the old BS is not to be confused with dynamic loads, but was intended to allow for load reversals and their effect on fatigue during the life of the gear.

It has been customary for AGMA to put the dynamic factor in the denominator of the rating formula, whereas ISO and BS apply it to the numerator. Nevertheless, the dynamic factor is defined as a multiplier of the transmitted load in all the standards, although some believe that the effect should be additive.[18]

3) Load distribution factors. A load distribution factor is used in the rating equations to reflect the non-uniform load distribution along the contact lines caused by deflections, alignment, and helix modifications (including crowning and end relief), and profile and pitch deviations. The evaluation procedure for this factor is rather complex, since many variables are involved, and some of them, such as the component of the gear system alignment and manufacturing errors, are difficult to determine.

| TABLE 1 COMPARISON OF PITTING RESISTANCE INFLUENCE FACTORS | | | | |
|---|---|---|---|---|
| | BS 436:1940 | AGMA 218 | BS 436:1986 | ISO/DIS 6336 |
| Geometry Factors* | $\frac{R_r^{0.8}}{d}$ | I | $\frac{u}{u+1} \frac{1}{Z_H^2 Z_\epsilon^2}$ | $\frac{u}{u+1} \frac{1}{Z_H^2 Z_\epsilon^2 Z_\beta^2}$ |
| Elasticity Factors* | $(kE_e)^{0.5}$ | $C_p$ | $Z_E$ | $Z_E$ |
| Size Factors* | – | $C_s$ | $\frac{1}{Z_x^2}$ | $\frac{1}{Z_x^2}$ |
| Lubrication Film Factors* | – | $C_f^{0.5} C_T$ | $\frac{1}{Z_L Z_V Z_R}$ | $\frac{1}{Z_L Z_V Z_R}$ |
| Application Factors† | – | $C_a$ | $K_A$ | $K_A$ |
| Dynamic Factors† | – | $C_V$ | $\frac{1}{K_V}$ | $\frac{1}{K_V}$ |
| Load Distribution Factors† | – | $C_m$ | $\frac{1}{K_{H\alpha} K_{H\beta}}$ | $\frac{1}{K_{H\alpha} K_{H\beta}}$ |
| Work Hardening Factors† | – | $C_H$ | $Z_W$ | $Z_W$ |
| Life Factors† | – | $C_L$ | $Z_N$ | $Z_N$ |
| Reliability Factors† | – | $C_R$ | $S_{Hlim}$ | $S_{Hmin}$ |
| Material Quality Factor† | – | – | $Z_M$ | – |
| Speed Factor† | $X_C$ | – | – | – |

† denotes common and * non-common factors

In AGMA the load distribution factor is the product of the face and transverse load distribution factors. The face or longitudinal (as described in the ISO and BS) load distribution factor accounts for the non-uniform load across the face of the gear, while the transverse load factor reflects the effect of non-uniform distribution of load down the tooth flank due to profile, pitch deviations, and tooth modifications. Although AGMA uses this factor to allow for the effect of the non-uniform distribution of load among the teeth which share the total load, no specific information is given in the standard. The AGMA standard assumes that if the gears are accurately manufactured, the value of the transverse load distribution factor can be taken as unity. AGMA provides both empirical and analytical methods to determine the face load distribution factor. The empirical method is recommended for normal, relatively stiff gear assemblies, and only a minimum amount of information is required. The second method is based on elastic and non-elastic lead mismatch and needs information about design, manufacture, and mounting and is, theoretically, suitable for any gear design.

The ISO load distribution factor is also the product of the transverse and longitudinal load factors. Three different approaches have been made by ISO to determine the longitudinal load factor. Method B is a final proof rating calculation method based on known manufacturing errors. Method C is a preliminary rating method and uses assumed values of manufacturing errors within limits of prescribed tolerances. Method D is even more simplified than method C. The transverse load factor is a function of longitudinal load factor, contact ratio, pitch tolerance, and mean load intensity. Procedures for calculating the load distribution factors in ISO are the most complex and are still under revision. Load distribution factors in the BS employ virtually the same procedure as method C in ISO, except for a difference in determining total misalignment. ISO gives five approximation methods for this, while the BS only gives one.

4) Life factors. Life factors take into account the effects of increments in permissible stresses if a limited number of load cycles is demanded. Among AGMA, ISO, and BS, the most distinct difference lies in the definition of endurance limits.

AGMA 218 sets $10^7$ load cycles as the endurance limit for both bending and pitting, while ISO and BS define limits of $2x10^6$, $5x10^7$, and $10^9$ cycles for contact stresses. Although there is no life factor in the old BS, a procedure to calculate variable duty cycles by determining an equivalent running time was provided. BS 436:1986 also has a procedure to deal with variable duty cycles, while this aspect of gear running is not considered by ISO.

5) Material quality factor. Among the four standards, only the BS introduces material factors in its bending and contact stress ratings in an attempt to allow for the higher permissible stresses to be obtained from using higher quality materials.

6) Size factors. Size factors are used in all, except the old BS, to take into account the influence of tooth size on surface fatigue strength. Values are usually taken as unity because no further information is provided in any of the standards.

7) Work-hardening factors. When the pinion material is substantially harder than the wheel, the effects of cold work hardening and internal stress changes in the softer wheel material may occur, in which case the surface contact stresses will be reduced. These effects have been considered by AGMA, ISO, and BS by introducing a hardness ratio or work-hardening factor. In AGMA, the hardness ratio factor is a function of the gear ratio and pinion and wheel hardness, but AGMA only applies this factor to the wheel rating. A guidance diagram is given by BS for determining the work-hardening factor, based on surface roughness and hardness. The ISO work-hardening factor is only related to wheel hardness.

8) Permissible stresses. Permissible bending and contact stresses are given in the old BS for a limited number of materials listed in the standard. It is usually agreed that the values are generally too pessimistic for surface-hardened gears. Allowable bending and contact stresses, based on laboratory and field experience for each material and heat treatment condition, are provided in AGMA. For most of the steels the allowable bending and contact stress numbers are functions only of material hardness.

In both BS and ISO the permissible bending/contact stress is based on the bending/surface fatigue endurance limit for the material, taking into account the required life and running conditions. According to the BS, for most gear materials the bending/contact endurance limit depends only on hardness without differentiating between materials and heat treatments. In ISO, bending/contact endurance limit is determined either based on experimental data for test gears of the same material or on prepared, polished specimens. Values are provided in the ISO standard for a wide range of steels and heat treatments.

For surface-hardened gears, the BS bending endurance limit is based on residual stresses and the ultimate tensile strength of the gear material. Determining the residual stresses and tensile strength of surface-hardened gears is, however, difficult, casting some doubt as to the ease with which this method can be used.

9) Factors of safety and reliability. So far there is no accepted method of relating gear reliability to safety factors considering the effect of material quality and gear accuracy. The AGMA reliability factor accounts for the effect of the normal statistical distribution of failures from the allowable stresses and can be chosen according to the reliability required. BS and ISO leave the user to specify a value for this factor. Minimum demanded safety factors for bending strength and contact stress are recommended by both ISO and BS to reflect the confidence in the actual operating conditions and material properties, but the values for these factors are different. The safety factor for bending strength in the old BS is defined as the ratio of ultimate tensile strength to the product of the speed factor and bending stress factor.

10) Non-common geometry factors. Geometry factors account for the influence of the helix angle, contact ratio, and tooth flank curvature at the pitch point on gear load capacity. Ignoring the experimental exponent of 0.8, the geometry factor for the old BS can be written

$$\frac{R}{R+1}\,\frac{\cos\alpha_t \cos\alpha_{tw}}{2\cos\beta_b} \tag{8}$$

For the BS and ISO the geometry factor is

$$\frac{1}{Z_H^{\,2}} = \frac{\cos\alpha_t \sin\alpha_{tw}}{2\cos\beta_b \cos\alpha_{tw}} \tag{9}$$

The similarity between Equations 8 and 9 is not apparent when expressed in the way given in the standards. (See Geometry Factors, Table 1.)

The AGMA geometry factor, I, for contact stress is

$$\frac{C_c\, C_x\, C_\psi^2}{m_N} \tag{10}$$

where $C_c$ is the curvature factor at the pitch line and is a function of the gear ratio and pressure angle. $C_x$ is a contact height factor adjusting the location of the tooth profile where the stress is calculated. The helical factor $C_\psi$ accounts for the helical effect in low contact ratio helical gears, and $m_N$ is the load sharing ratio which depends on the transverse and face contact ratios. Similarly, ISO uses a helix angle factor to account for the helix effect on contact stresses. Both ISO and BS include a contact ratio factor to allow for the influence of transverse contact ratio and overlap ratio on contact stress based ratings.

11) Non-common elasticity factors. Elasticity factors account for the influence of material mechanical properties on the Hertzian stress. Those used in AGMA, ISO, and BS are identical. The only difference between the old BS and the others is that the equation for calculating this factor has been simplified by assuming that Poisson's ratios for the pinion and wheel are the same.

12) Non-common lubrication film factors. BS and ISO account for minimal film thickness between contacting teeth on surface load capacity. In their rating procedures, oil viscosity, surface hardness, and pitch line velocity are considered to be the main factors influencing film thickness. There are some differences between the calculation methods used by BS and ISO. BS gives two diagrams: one for roughness and the other for the product of a lubricant and speed factor. ISO provides three equations and corresponding diagrams to determine these factors. Although AGMA

218 does not consider lubrication, it does take tooth surface roughness and temperature effects into account by introducing a surface condition and a temperature factor. In the old BS, lubrication was ignored altogether. Tooth scuffing, which is covered by ISO and DIN in separate parts, attempts to predict the temperature at which scuffing will occur. This is not dealt with by any of the other standards and, therefore, no comparisons can be made, although scuffing does appear in the new AGMA standard.

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