

Specification for

Bevel gears (machine cut)

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Foreword

This revision of this British Standard has been prepared under the direction of the Mechanical Engineering Standards Committee. The main changes incorporated are the specification of metric module gears in accordance with ISO 678:1976, published by the International Organization for Standardization (ISO), in place of diametral pitch (D.P.) gears, the introduction of the basic rack form in accordance with ISO 677:1976, revision of the material data and, in view of the need for clear and precise instructions to be given to a supplier or manufacturer when gears are ordered, the addition of a list of ordering instructions.

This standard provides guidance for the smaller or general purpose user on the design of bevel gears for the transmission of normal loads and retains the convenient method of calculation which was a feature of the previous edition. The sophisticated and specialist power train designer or manufacturer has recourse to much more detailed design procedures than those specified in this standard. For high load capacity gears, the detailed design of the teeth has to be developed to suit the deflection characteristics of the gearbox. This may lead to higher load capacities than those predicted by the methods specified in this standard and to meet such circumstances a more comprehensive standard is under consideration within ISO.

The tooth form specified in the 1949 edition of this standard, which had a 20° pressure angle and a working depth of twice the module, has been retained because it represents a well balanced compromise between strength, resistance to wear and quietness of running. Failure of a gear tooth, either by pitting or breakage, is almost invariably the result of fatigue. Therefore, in the determination of both allowable bending loads and allowable surface loads, the probable number of repetitions of stress has to be taken into account and the speed factors embodied in the power capacity formulae, which are based on revolutions per minute, contribute to this end. A means of differentiating between the severity of conditions on the pinion and on the wheel is thus obtained.

This standard supersedes BS 545:1949 which is therefore now withdrawn.

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Summary of pages

This document comprises a front cover, an inside front cover, pages i and ii, pages 1 to 28, an inside back cover and a back cover.

This standard has been updated (see copyright date) and may have had amendments incorporated. This will be indicated in the amendment table on the inside front cover.

1 Scope

This British Standard specifies tooth form, modules, accuracy requirements, methods of determining load capacity and material requirements for machine-cut bevel gears, connecting intersecting shafts which are perpendicular to each other and having teeth with a normal pressure angle of 20° at the pitch cone, whose lengthwise shape may be formed by straight or curved surfaces having forms similar to those shown in Figure 1. A list of information to be supplied when ordering bevel gears is also provided.

Three classes of gear are specified, designated class A, B and C respectively.

- Class A, suitable for precision gears operating at peripheral speeds above 10 m/s when transmitting normal loads.
- Class B, suitable for high class cut gears operating at peripheral speeds below 15 m/s when transmitting normal loads.
- Class C, suitable for commercial cut gears operating at peripheral speeds below 6 m/s when transmitting normal loads.

Class A, the highest accuracy class, corresponds to grade 7 of BS 436-2, whilst classes B and C correspond to grades 9 and 11 respectively of BS 436-2. For the purpose of this standard, it is assumed that gears will be supported on shafts of ample size, provided with thrust and journal bearings, adequately lubricated and have face widths that do not exceed either one-third of the cone distance or three times the pitch. Whilst this standard applies to bevel gears operating on shafts which are perpendicular to each other, bevel gears may be designed for use at other shaft angles, but their load capacity and design characteristics are not within the scope of this standard.

2 References

The titles of the publications referred to in this standard are listed on the inside back cover.

3 Definitions and symbols

3.1 Definitions. For the purposes of this British Standard, the definitions given in BS 2519-1 apply.

3.2 Symbols. For the purposes of this British Standard, the following symbols apply.

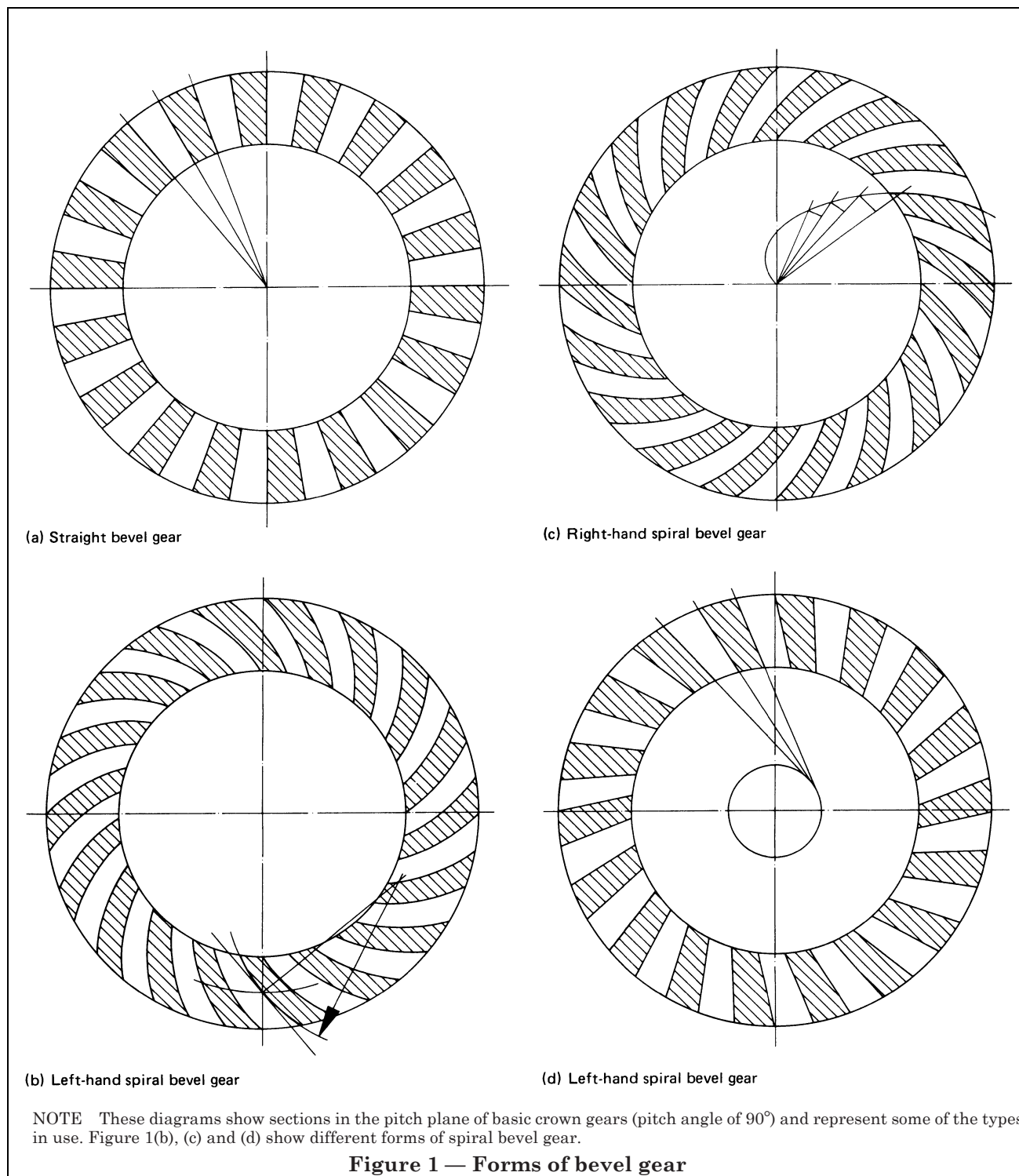
NOTE Where applicable, the symbols are in accordance with BS 2519-2.

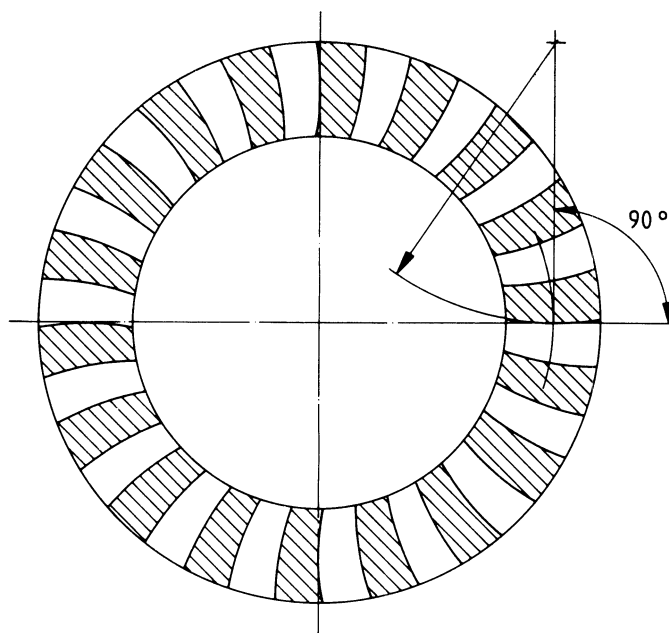
Symbol	Term
d	Reference diameter
h_a	Addendum
R	Cone distance
d'	Reference diameter at mid-facewidth

Symbol	Term
b	Facewidth
s_t	Spacewidth
$\bar{s}$	Caliper thickness setting
$\bar{h}_a$	Caliper height setting
K	Pitch setting
x	Addendum coefficient
l	Smaller length of arc pitch circle
M	Torque
m_n	Normal module
m	Module (transverse)
n	Speed (r/min)
p	Pitch
p_n	Normal pitch
p_t	Transverse pitch
Q	Interference factor
w	Period in load cycle during which torque changes uniformly
σ_{bm}	Bending stress factor
σ_{cm}	Surface stress factor
z	Number of teeth
H	Time, in h
H_{eb}	Equivalent running time, in h, for strength
H_{ec}	Equivalent running time, in h, for wear
H_b	Equivalent running time, in h, per cycle for strength
H_c	Equivalent running time, in h, per cycle for wear
H_{bw}	Equivalent running time, in h, for period w for strength
H_{cw}	Equivalent running time, in h, for period w for wear
X_b	Speed factor for strength
X_c	Speed factor for wear
Y	Strength factor
Z	Zone factor
τ	Pitch angle
ϵ_β	Overlap ratio
M_u	Cyclic torque ratio
β	Spiral angle
β_w	Spiral angle at mid-face
β_v	Spiral angle at the pitch circle
α_n	Normal pressure angle
β^*	Spiral angle factor
ϵ_β^*	Overlap ratio factor
ϕ_p	Tolerance factor (run-out)
W	Cyclic negative torque ratio

Suffixes. The subscript suffixes 1 or 2, attached to any of the foregoing symbols, indicate that the symbol applies to the pinion and the wheel respectively

NOTE In some mathematical progressions symbols such as $n_{1,1}$ appear (e.g. see 7.11.1) where the first subscript refers to the pinion and the second to the order in the progression.





(e) Left-hand curved tooth bevel gear with zero spiral angle at mid-face

Figure 1 — Forms of bevel gear (concluded)

4 Standard normal modules

The normal modules shall be selected from Table 1. Wherever possible, selection shall be made from the values given in column 1.

NOTE The basic design module of a bevel gear is defined with respect to the reference surface at its intersection with the back cone.

5 Form of gear tooth

5.1 Basic rack. The normal section of the 20° pressure angle basic rack tooth for bevel gears shall be as shown in Figure 2.

NOTE 1 The profile of the basic rack tooth corresponds to a section of the tooth surface of a crown gear of infinitely large diameter on a plane at right angles to the tooth surfaces.

The rack tooth profile shall be straight, although a slight easing of the tip or root is permissible. The amount of this easing shall not exceed that shown in Figure 2.

NOTE 2 Alternatively, the easing may be provided at the root of the tooth, if preferred by the manufacturer.

NOTE 3 A cutting tool of form counterpart to a single flank and root fillet of the basic rack, see Figure 3, may be used to generate single flanks and root fillets of bevel gear teeth complying with the requirements of this standard. In general, neither the amount of the tip easing, nor the fillet radius on the bevel gear tooth will be a constant multiple of the pitch at all distances from the apex.

Table 1 — Standard normal modules

Normal modules m_n	
1st choice	2nd choice
mm	mm
1	
1.25	1.125
1.5	1.375
2	1.75
2.5	2.25
3	2.75
4	3.5
5	4.5
6	5.5
	6.5 ^a
8	7
10	9
12	11
16	14
20	18
25	22
32	28
40	36
50	45

^a The module 6.5 should be avoided.

5.2 Tooth spiral. The form of tooth spiral shall be determined by the manufacturer, as it is dependent on the type of cutter and the method of cutting employed.

NOTE For this reason, no attempt has been made to give a formula for the spiral angle at different positions along the face, although reference is made both to the spiral angle at mid-face (β_w) and to that at the pitch circle (β_v). Some of the types of tooth spiral in use are illustrated in Figure 1.

5.3 Addendum and dedendum

5.3.1 Addendum. The pinion addendum and wheel addendum shall be determined from the following equations

$$h_{a,1} = \frac{p_n}{\pi} (1 + x_1) = m_n (1 + x_1)$$

$$h_{a,2} = \frac{p_n}{\pi} (1 + x_2) = m_n (1 + x_2)$$

where

x_1 is taken as the greater of the following:

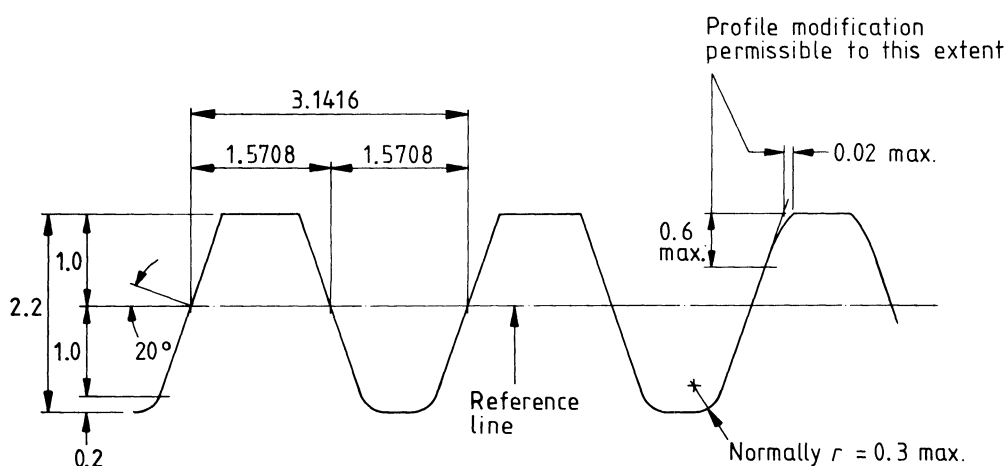
$$a) \quad x_1 = 0.4 \left\{ 1 - \frac{z_1 \sec \tau_1}{z_2 \sec \tau_2} \right\};$$

$$b) \quad x_1 = 1 - \frac{2z_1 \sec \tau_1}{Q};$$

x_2 is calculated in the same way as x_1 , by substituting z_2 for z_1 , τ_2 for τ_1 and vice versa.

Q is obtained from Figure 5.

5.3.2 Dedendum. The dedendum of the tooth shall be equal to the difference between the addendum as determined in 5.3.1 and the whole depth of tooth.



All dimensions are in millimetres.

NOTE 1 The profile of the basic rack refers to bevel gears with involute teeth. This profile has substantially straight sides and the following characteristics:

- a pressure angle of 20° ;
- the height of the tooth equal to $2.20 m$ mm.

NOTE 2 These gears have a constant bottom clearance. In these gear pairs, the tip angle of one gear is equal to the difference between the designed shaft angle and the root angle of the mating gear.

NOTE 3 The value of the radius at the root of the tooth is normally fixed at $0.3 m$ maximum, keeping in mind that this radius should be as large as possible for maximum strength. In certain cases, and as far as permitted by the conditions of meshing, this value may be exceeded up to a maximum of $0.35 m$. The maximum radius is usually limited by the tooth roots at the smaller end of the teeth.

NOTE 4 In practice, the bevel tooth corresponding to the basic rack is that which is conjugate to a tooth of a crown gear with straight flanks. The tooth is developed from the surfaces described by the cutting edge of the tool or of the two cutting tools. The radius of the crown wheel considered is equal to the length of the cone distance of the teeth which are to be cut.

Figure 2 — Basic rack tooth profile for unit normal module

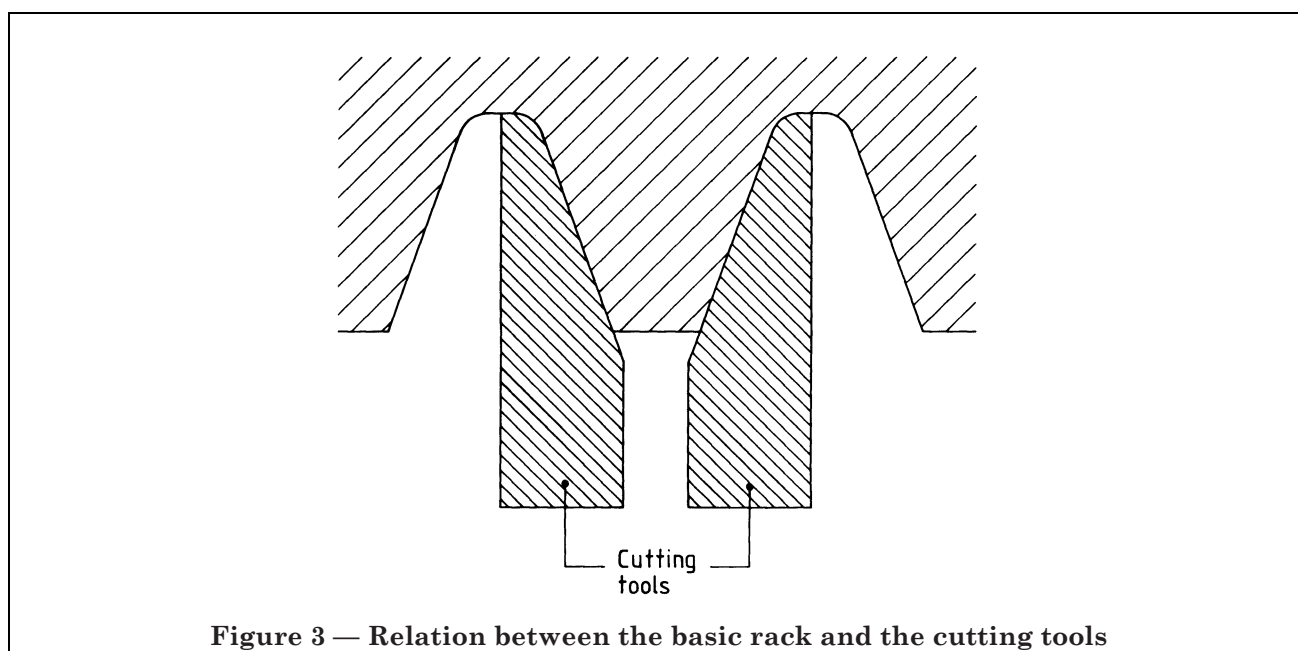


Figure 3 — Relation between the basic rack and the cutting tools

5.3.3 Addendum and dedendum modification. For straight bevel gears, $z_1 \sec \tau_1 + z_2 \sec \tau_2$ shall not be less than 34.2 and this excludes the combination of gears which fall within the small areas cut off by the dotted lines A on Figure 7 and Figure 10. for bevel gears with a spiral angle of 30° at mid-face, $z_1 \sec \tau_1 + z_2 \sec \tau_2$ shall not be less than 26.5 and this excludes the combination of gears which fall within the small areas cut off by the dotted lines B on Figure 7 and Figure 8.

NOTE Normally $x_1 = -x_2$. Where the number of teeth in the two gears is small, i.e. when $z_1 \sec \tau_1 + z_2 \sec \tau_2$ is less than Q , interference or undercutting will be present and the use of such gears should be avoided.

The addendum and dedendum shall each be reduced by $0.4 - 0.0344 z_1 \sec^2 \beta_v \sec \tau_1$ (for unit normal module) unless this quantity exceeds 0.065, in which case the gears are outside the scope of this standard, if z_1 is less than the value given by the following expression:

$$\frac{11.63}{\sec^2 \beta_v \sec \tau_1}$$

NOTE 1 Since the shafts are at right angles to following relationship will be applicable:

$$\frac{z_1 \sec \tau_1}{z_2 \sec \tau_2} = \left(\frac{z_1}{z_2} \right)^2$$

Example. To find the addendum of a pair of spiral bevel gears connecting perpendicular shafts and having the following particulars:

$$z_1 = 10;$$

$$z_2 = 45;$$

$$\alpha_n = 20^\circ;$$

$$\beta_v = 30^\circ;$$

$$m_h = 2;$$

$$\tan \tau_1 = \frac{10}{45} \text{ and so } \tau_1 = 12^\circ 32'.$$

x_1 is equal to the greater of the following values:

$$\text{a) } 0.4 \left[1 - \left(\frac{z_1}{z_2} \right)^2 \right] = 0.4 \left[1 - \left(\frac{10}{45} \right)^2 \right] = 0.38;$$

$$\text{b) } \left[1 - \frac{2z_1 \sec \tau_1}{Q} \right] = \left[1 - \frac{20 \times 1.0244}{26.5} \right] = 0.22,$$

i.e. $x_1 = 0.38$.

Addendum of pinion = $2(1 + 0.38) = 2.76$

Addendum of wheel = $2(1 - 0.38) = 1.24$

5.4 Caliper settings. Caliper settings shall be calculated as follows:

$$S_{t,1} = \pi m - S_{t,2}$$

$$S_{t,2} = \frac{\pi m}{2} + 2 \tan \alpha_n \left(\frac{h_{a,2}}{\cos \beta_v} - m \right)$$

Caliper thickness setting for pinion, $\bar{s}_1$, is given by the following expression:

$$= \frac{S_{t,1}}{\sec [\beta_v + (S_{t,1}/2R)]}$$

Caliper thickness setting for wheel, $\bar{s}_2$, is given by the following expression:

$$= \frac{S_{t,2}}{\sec [\beta_v + (S_{t,2}/2R)]}$$

Caliper height setting for pinion, $\bar{h}_{a,1}$

$$= h_{a,1} + \frac{\bar{s}_1^2}{4d_1' \sec^2 \beta_2 \sec \tau_1}$$

Caliper height setting for wheel, $\bar{h}_{a,2}$

$$= h_{a,2} + \frac{\bar{s}_2^2}{4d_2' \sec^2 \beta \sec \tau_2}$$

6 Accuracy of manufacture

6.1 General

6.1.1 Properties subject to control. Although elemental checks on the tooth geometry used for parallel shaft spur gearing cannot in general be applied to bevel gears, the pitch accuracy, run-out and, on some gears, contact pattern shall be subject to control.

NOTE The contact pattern depends on the manufacturing system used.

6.1.2 Class of accuracy. A bevel gear shall be classified as belonging to a particular class if its transverse pitch errors do not exceed the values given in Table 2.

NOTE Other quantities may be subject to agreement between the purchaser and the manufacturer.

6.1.3 Tolerance. The tolerances for pitch accuracy for spur and helical gears specified in BS 436-2 shall be applicable to bevel gears, provided that a diameter is specified at which the relevant arc length can be defined.

Since, for convenience, pitch measurements or run-out will normally be taken at mid-facewidth, the relevant diameter shall be defined as the reference cone diameter at mid-facewidth (d').

NOTE These tolerances are concerned with the deviations of the teeth relative to a cone axis. It is therefore important to define the areas of the component which define this axis, i.e. two bearing diameters or a bore and face.

6.2 Transverse pitch tolerance. The tolerances on transverse pitch, both adjacent and cumulative, shall be as given in Table 2, which is based on l , where l is calculated as follows:

$$l = \frac{\pi dt}{z}$$

where t is one for adjacent pitch or any whole number up to $z/2$ for cumulative pitch limits.

Table 2 — Transverse pitch tolerance

Class	Transverse pitch tolerance
	μm
A	$3.55 \sqrt{l} + 9.0$
B	$7.1 \sqrt{l} + 18.0$
C	$14.0 \sqrt{l} + 35.5$

6.3 Run-out of teeth tolerance. The run-out of teeth tolerance, measured normal to the reference cone shall be as given in Table 3 which is based on the factor ϕ_p , calculated as follows:

$$\phi_p = \frac{d' \cos \beta_w}{z} + 0.25 \sqrt{d'}$$

where

β_w is the spiral angle at mid-face, in degrees;

z is the number of teeth.

Table 3 — Run-out of teeth tolerance

Class	Run-out of teeth tolerance
	μm
A	$3.15 \phi_p + 40.0$
B	$5.0 \phi_p + 63.0$
C	$8.0 \phi_p + 100.0$

6.4 Tooth thickness tolerance. The tooth thickness shall, in general, be less than the nominal thickness.

The tolerances on tooth thickness shall be as given in Table 4, where d and p_t are in mm.

NOTE In certain cases, special consideration may be necessary. Two common instances of special tolerances, e.g. engine timing gears and differential or planetary gears, are given in Table 4.

6.5 Tooth contact patterns. Since the load capacity of gear teeth depends on the load distribution across the facewidth, it shall be ensured that the gears operate with a satisfactory tooth contact marking.

NOTE 1 It is found, particularly with gears made of high strength materials, that the tooth contact loads deflect the gear axes and tend to cause uneven load distribution. Manufacturing processes permitting, tooth shapes should be modified to secure balanced load distribution under working conditions.

To ensure an accurate reproducibility of the tooth shape, tooth contact patterns shall be used with specified displacements of the gears between the two or more checks.

NOTE 2 Ideally, pattern reference gears should be retained to maintain a comparison standard for further batches.

Where gears are lapped together to achieve a pattern consistency, the gears shall be kept together as pairs and interchangeability of individual parts cannot be guaranteed.

NOTE 3 Tooth contact patterns can be dependent on the axial positions of the mating gears and the backlash between them. Whilst a nominal mounting distance should be specified, it is difficult to control this dimension accurately to achieve consistent in-service contact patterns with the possible gearbox axis errors. It is recommended that provision be made to adjust the axial positions of both members by shimming or other means.

Table 4 — Tooth thickness tolerance

Class	Reduction in nominal thickness	
	Maximum	Minimum
	μm	μm
A	$(0.2d + 4 p_t + 50) \cos \beta_v$	$(0.1d + 2 p_t + 25) \cos \beta_v$
B	$(0.3d + 6 p_t + 75) \cos \beta_v$	$(0.1d + 2 p_t + 25) \cos \beta_v$
C	$(0.4d + 8 p_t + 100) \cos \beta_v$	$(0.1d + 2 p_t + 25) \cos \beta_v$
Special cases:		
Engine timing gears	$(0.075d + 1.5 p_t + 19) \cos \beta_v$	$(0.037d + 0.075 p_t + 10) \cos \beta_v$
Commercial cut differential or planetary gears	$(0.25d + 5 p_t + 63) \cos \beta_v$	$(0.15d + 3 p_t + 38) \cos \beta_v$

NOTE 1 Where no reversal of torque occurs under load, the maximum reduction in nominal thickness for classes A, B and C may be doubled.

NOTE 2 The tolerances specified refer to the allowable variation between similar gears and do not represent permissible difference in thickness between any two teeth of one gear, which is controlled indirectly by the pitch tolerances in Table 2.

7 Load capacity

7.1 General. The permissible load capacity of a pair of gears shall be that of the pinion or the wheel, whichever is the lesser. The permissible load capacity of a gear shall be limited either by considerations of surface stress (conveniently referred to as “wear”) or of bending stress in the teeth (described as “strength”).

NOTE 1 Consequently, the determination of the permissible load capacity of a pair of gears involves four calculations concerned with wear and strength of pinion and wheel. These assume satisfactory contact patterns under the critical loaded condition.

NOTE 2 Bevel gears should be set up correctly on assembly but spiral bevel gears will often give tolerance to some mounting errors.

NOTE 3 Examples illustrating the application of the requirements of 7.2 to 7.11 are given in Appendix A.

7.2 Determination of load capacity

7.2.1 Wear. The permissible power, in kW, of a pair of bevel gears connecting perpendicular shafts shall be limited by wear to the lower of the following values:

$$a) \frac{X_{c,1} \sigma_{cm,1} Z_1 b n_1 z_1 m^{1.8}}{21 \times 10^6 \epsilon_\beta^*} \left(1 - \frac{b}{R}\right);$$

$$b) \frac{X_{c,2} \sigma_{cm,2} Z_2 b n_2 z_2 m^{1.8}}{21 \times 10^6 \epsilon_\beta^*} \left(1 - \frac{b}{R}\right);$$

7.2.2 Strength. The permissible power, in kW, of a pair of bevel gears connecting perpendicular shafts shall be limited by strength to the lower of the following values:

$$a) \frac{X_{b,1} \sigma_{bm,1} Y_1 b n_1 z_1 m^2}{21 \times 10^6 \beta^*} \left(1 - \frac{b}{R}\right)$$

$$b) \frac{X_{b,2} \sigma_{bm,2} Y_2 b n_2 z_2 m^2}{21 \times 10^6 \beta^*} \left(1 - \frac{b}{R}\right)$$

7.2.3 Permissible tangential load. The permissible tangential load, in N, for wear or strength shall be equal to the permissible power, in kW, divided by the pitch line speed, in m/s, and multiplying by 10^3 .

7.2.4 The expected life. The permissible power, or tangential load, obtained in accordance with 7.2.1, 7.2.2 and 7.2.3 shall apply to an expected total running life of 26 000 h. If it is specified or desirable that bevel gears shall be designed on the basis of any other life, the permissible power, or tangential load, shall be multiplied by the following expressions, see Table 5:

a) for wear:

$$\left(\frac{27\,000}{1000 + H_{ec}}\right)^{1/3};$$

b) for strength:

$$\left(\frac{26\,200}{200 + H_{eb}}\right)^{1/7}.$$

where H_{ec} and H_{eb} are the equivalent running times, for wear and strength respectively, corresponding to the expected life.

Unless otherwise specified or determinable, the nominal rating (e.g. the continuous load capacity of the motor) shall apply continuously throughout the expected total running time.