

Specification for

Fine pitch gears —

Part 1: Involute Spur and Helical Gears

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Institution of Engineers and Shipbuilders in Scotland	University of Sheffield
Milling Cutter and Reamer Association	Manufacturers of fine pitch gears
Ministry of Defence, Army Department	
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Foreword

This standard makes reference to the following British Standards:

BS 978, *Fine pitch gears*.

BS 978-2, *Cycloidal type gears*.

BS 978-PD 3376, *Addendum No. 1 to BS 978-2, Double circular arc type gears*.

BS 978-3, *Bevel gears*.

BS 978-5, *Hobs and cutters*.

First published in 1952 and revised in 1962, this British Standard has been further revised under the authority of the Mechanical Engineering Industry Standards Committee.

In the 1962 edition, dual flank composite testing was introduced in the form of an appendix to the specification. As this method of test became more widespread, users of the standard were becoming confused between the application of the composite tolerances and application of the individual elemental tolerances. The prime reason for revision, therefore, has been to give dual flank testing the importance it now deserves by placing it in the mandatory part of the standard as the main accuracy testing procedure. Also care has been taken to ensure that the user is not misled by attempting to reconcile composite errors with elemental errors. It is necessary to retain the elemental tolerances because these will continue to be used in some fields as acceptance criteria and because they are essential as the basis for manufacturing gears and cutting tools¹⁾. Recognizing also that single flank testing is used reference to this is made in an appendix.

Opportunity has been taken to effect improvements in presentation and in the examples of calculations, and the terminology and notation is in accordance with that agreed within the International Organization for Standardization (ISO).

This standard is a companion work to BS 978, “*Fine pitch gears*”, Part 2, “*Cycloidal type gears*” (including addendum PD 3376, “*Double circular arc type gears*”), Part 3, “*Bevel gears*”, and Part 5, “*Hobs and cutters*”.

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Summary of pages

This document comprises a front cover, an inside front cover, pages i to iv, pages 1 to 29 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⁾ BS 978, “*Fine pitch gears*”, Part 5, “*Hobs and cutters*”.

1 General

1.1 Scope

This British Standard relates to involute spur, helical and crossed helical gears, having diametral pitches finer than 20, in four accuracy grades. Examples of application are:

Class A. Astronomical apparatus and scientific instruments; colour printing machinery; control systems (when multiple trains of gears are involved this grade would generally be used for low speed components where the effect of positional accuracy is important).

Class B. Navigational instruments; high speed components of control systems where quietness and smooth running are essential; machine tool speed control; photographic instruments.

Class C. General purpose instruments; accounting and calculating machinery; counters; clockwork mechanisms.

Class D. Textile machinery; switch operating gear; clockwork mechanisms where accuracy is not the primary consideration.

It is recommended that all spur, helical and crossed helical gears for use in horology should be produced to this British Standard with the possible exception of gears for weight or spring-driven mechanisms in which the pinion is the driven member and has 12 or fewer teeth. In such cases it may be preferable to use gears conforming to BS 978-2²⁾.

1.2 Terminology and notation

As the first step towards achieving international understanding of gears terminology, the terms and definitions of Draft ISO Recommendation No. 888³⁾, "International vocabulary of gears", have been applied, together with corresponding notation derived from Draft ISO Recommendation No. 889, "International gear notation, symbols for geometrical data"³⁾.

The notation used is as follows:

a	Centre distance
a_v	Virtual centre distance
d	Reference circle diameter
d_1	Reference circle diameter: pinion
d_2	Reference circle diameter: wheel
d_a	Tip diameter
d_{a1}	Tip diameter: pinion
d_{a2}	Tip diameter: wheel
d_R	Measuring pin diameter
d_b	Base circle diameter
j_n	Normal backlash
j^a	Coefficient for calculation of backlash
l	Length of arc
P_n	Normal diametral pitch
s	Arc tooth thickness
s_1	Arc tooth thickness: pinion
s_2	Arc tooth thickness: wheel
x	Addendum modification coefficient
x_1	Addendum modification coefficient: pinion
x_2	Addendum modification coefficient: wheel
Δx_1	Secondary addendum modification coefficient: pinion
Δx_2	Secondary addendum modification coefficient: wheel
y	Centre distance modification coefficient

²⁾ BS 978, "Fine pitch gears", Part 2, "Cycloidal type spur gears".

³⁾ To be incorporated in BS 2519, "Glossary of terms for toothed gearing".

- z Number of teeth
- z_1 Number of teeth: pinion
- z_2 Number of teeth: wheel
- z_v Virtual number of teeth
- α_n Normal pressure angle
- α_{nw} Normal pressure angle, operating
- α_t Transverse pressure angle
- α_{tw} Transverse pressure angle, operating
- β Helix angle at reference cylinder
- β_b Helix angle at base cylinder

^a BS 978, “Fine pitch gears”, Part 5, “Hobs and cutters”.

1.3 Preferred range of normal diametral pitches

The pitch shall be selected from the following preferred range.

Table 1 — Normal diametral pitch P_n
(inch units)

	32	48	100	160
24	36	64	120	180
28	40	80	140	200

1.4 Basic rack tooth profile

The shape and proportions of the basic rack tooth for spur helical and crossed helical gears shall be as shown either in Figure 1 or Figure 2. In the case of helical and crossed helical gears, the shape and proportions of the tooth are on a section at right angles to the pitch cylinder helix.

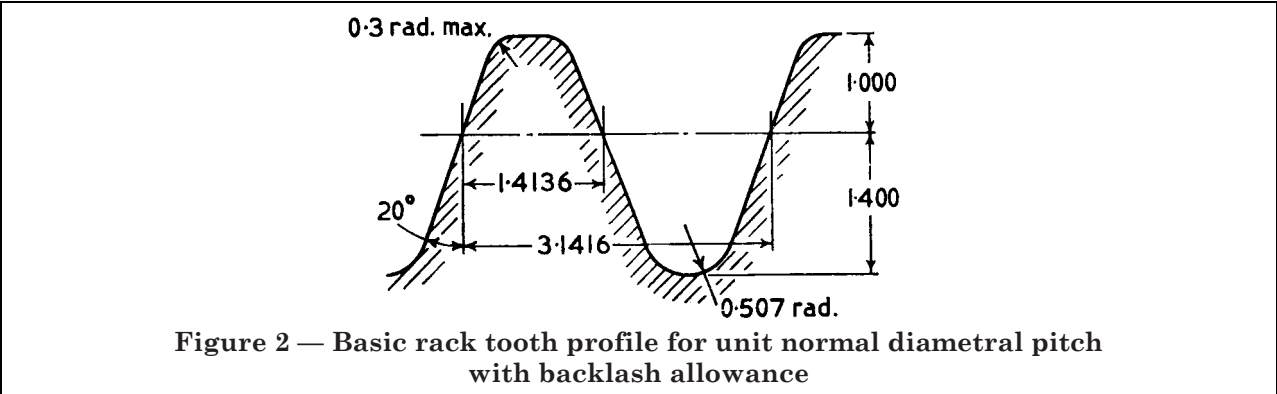
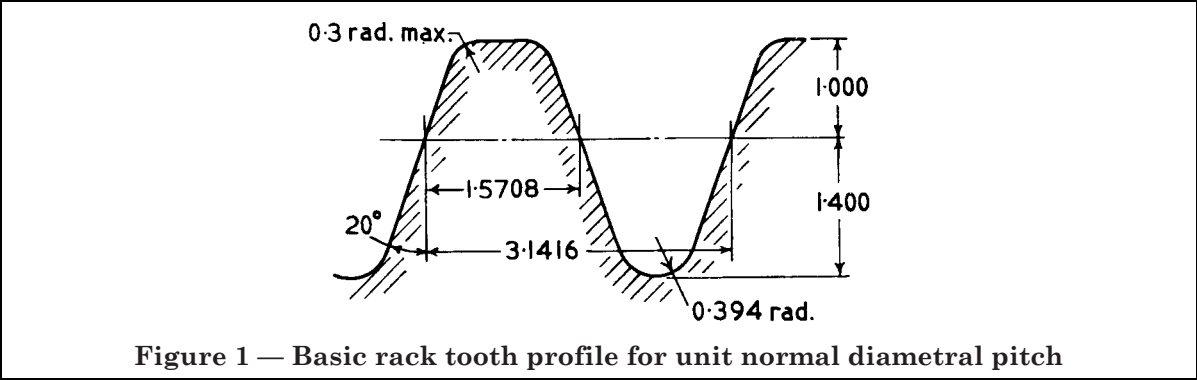


Figure 1 shows the basic rack usually employed where it is desired to have minimum backlash, as in servomechanisms.

Figure 2 is used where it is desired to have backlash to avoid any risk of jamming and where the load is usually uni-directional, for example, in clockwork mechanisms. When non-topping hobs are used it is permissible to use a hob conforming to Figure 1 basic rack and to cut at greater depth than normal to obtain the required tooth thickness.

For internal gears (Figure 1 and Figure 2 basic racks) the addendum shall normally be 0.850 instead of 1.000. In cases when the tips of the teeth have a radius of 0.25 or greater (up to 0.3 max. as shown) the 1.000 addendum may be applied.

2 Design details

2.1 Outside diameter and centre distance

For the normal case when both $z_1 \sec^3\beta$ and $z_2 \sec^3\beta$ are greater than 17 the dimensions are as follows:

$$d_1 = \frac{z_1 \sec\beta}{P_n} \quad (1) \quad d_2 = \frac{z_2 \sec\beta}{P_n} \quad (2)$$

$$d_{a1} = \frac{z_1 \sec\beta + 2}{P_n} \quad (3) \quad d_{a2} = \frac{z_2 \sec\beta + 2}{P_n} \quad (4)$$

$$a = \frac{(z_1 + z_2) \sec\beta}{2P_n} \quad (5)$$

(For spur gears, $\sec\beta = 1$)

2.2 Modification of addendum and centre distance

2.2.1 For an explanation of the principles followed in connection with addendum modification see Appendix A.

2.2.2 Where no limitations are imposed, the value of x for any gear whose virtual number of teeth z_v is less than 17 is given by the formulae:

$$x = 1 - \frac{z_v \sin^2 \alpha}{2} \quad (6) \quad \text{(See Table 2 and Figure 3 and Figure 4)}$$

$$\text{where } z_v = z \sec^2\beta_b \approx z \sec^3\beta \quad (7)$$

This gives values in accordance with the lower limiting line XX in Figure 3 and a small and normally negligible amount of undercutting is thereby permitted.

The addendum modification coefficient for helical pinions from 6 to 16 teeth and helix angles up to 45° is shown in Figure 4.

2.2.3 When the pinion is modified as above and there are no other limitations, the wheel is unmodified and its dimensions are obtained from the formulae in **2.1**; but the dimensions of the pinion and centre distance are as follows:

$$d_{a1} = \frac{z_1 \sec\beta + 2(1 + x_1)}{P_n} \quad (8)$$

$$s_1 = \frac{1.5708 + 0.7279x_1}{P_n} \quad \text{(Figure 1 type gears)} \quad (9)$$

$$s_1 = \frac{1.4136 + 0.7279x_1}{P_n} \quad \text{(Figure 2 type gears)} \quad (10)$$

$$a = \frac{(z_1 + z_2) \sec\beta + 2x_1}{2P_n} \quad (11)$$

Parallel axis external gear pairs designed as above have backlash at this centre distance. (See Appendix A.)

Table 2 — Data for spur pinions of unit diametral pitch having fewer than 17 teeth

No. of teeth z_1	Addendum modification coefficient x_1	Tip diameter d_{a1}	Increase in circular tooth thickness at standard pitch dia.	Recommended minimum number of teeth in mating wheel z_2
8 ^a	0.5321	11.0642	0.3874	26
9 ^a	0.4736	11.9472	0.3448	25
10	0.4151	12.8302	0.3022	24
11	0.3566	13.7132	0.2596	23
12	0.2982	14.5963	0.2170	22
13	0.2397	15.4793	0.1744	21
14	0.1812	16.3623	0.1319	20
15	0.1227	17.2453	0.0893	19
16	0.0642	18.1284	0.0467	18

^a Pinions having 8 or 9 teeth generated from basic rack Figure 1 have narrow crest width, as also have pinions of 10 and 11 teeth generated from basic rack Figure 2. 10 is the smallest number of teeth that can be generated at full depth from basic rack Figure 2.

2.2.4 Enlarged spur pinions with minimum backlash. Spur pinions enlarged according to Table 2 may be engaged with wheels in which x_2 is zero, at centre distance derived from Table 3. Gears derived from Figure 1 basic rack will have no backlash at this centre distance. A small and negligible reduction of tip-to-root clearance is obtained at this centre distance.

Table 2 is not applicable to helical gears, for which the formulae in Appendix A are required.

Alternatively, spur pinions enlarged according to Table 2 or helical pinions enlarged according to Figure 4 may be engaged with wheels in which $x_2 = -x_1$, in which case the centre distance is as obtained in 2.1, and

$$s_2 = \frac{1.5708 - 0.7279x_1}{P_n} \quad (\text{Figure 1 type gears}) \quad (12)$$

$$s_2 = \frac{1.4136 - 0.7279x_1}{P_n} \quad (\text{Figure 2 type gears}) \quad (13)$$

$$d_{a2} = \frac{z_2 \sec \beta + 2(1 - x_1)}{P_n} \quad (14)$$

Such gears derived from Figure 1 basic rack will have zero backlash at nominal dimensions.

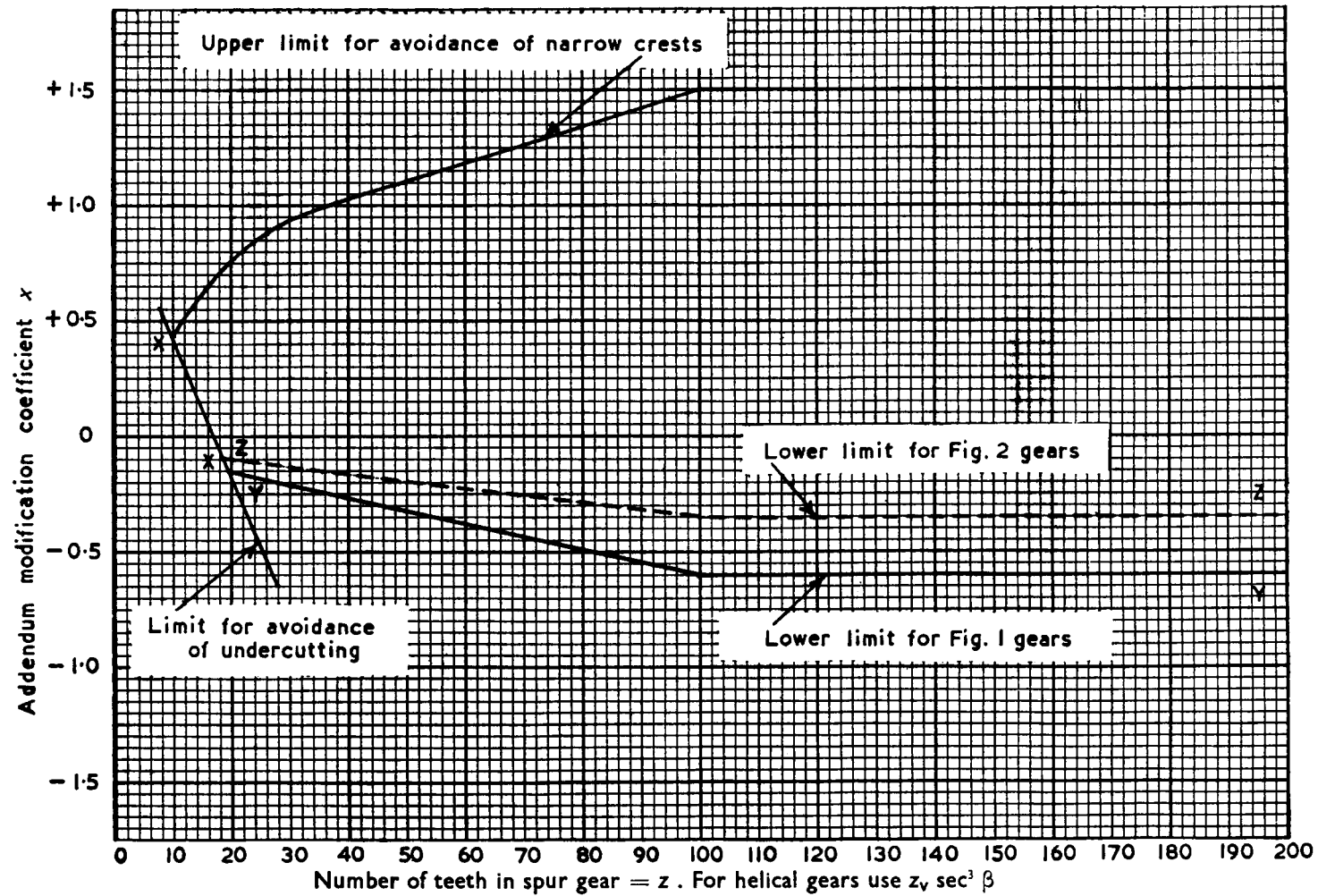


Figure 3 — Limiting values of addendum modification coefficient