B.6 Face width

For shaft angles less than 90°, a face width larger than that given in Figure B.6 may be used. For shaft angles greater than 90°, a face width smaller than that given in Figure B.6 should be used. Generally, the face width is 30 % of the cone distance or 10 m_{et2}, whichever is less. However, design parameters may require values to be larger or smaller. Figure B.6 face widths are based on 30 % of the outer cone distance. For zerol bevel gears, the face width given by Figure B.6 should be multiplied by 0.83 and should not exceed 25 % of the cone distance. For shaft angles substantially less than 90°, care should be exercised to ensure that the ratio of face width to pinion pitch diameter does not become excessive.

In the case of a hypoid, follow the above face width guidelines for the wheel. The hypoid pinion face width is generally greater than the face width of the wheel. Its calculation can be found in 7.5.

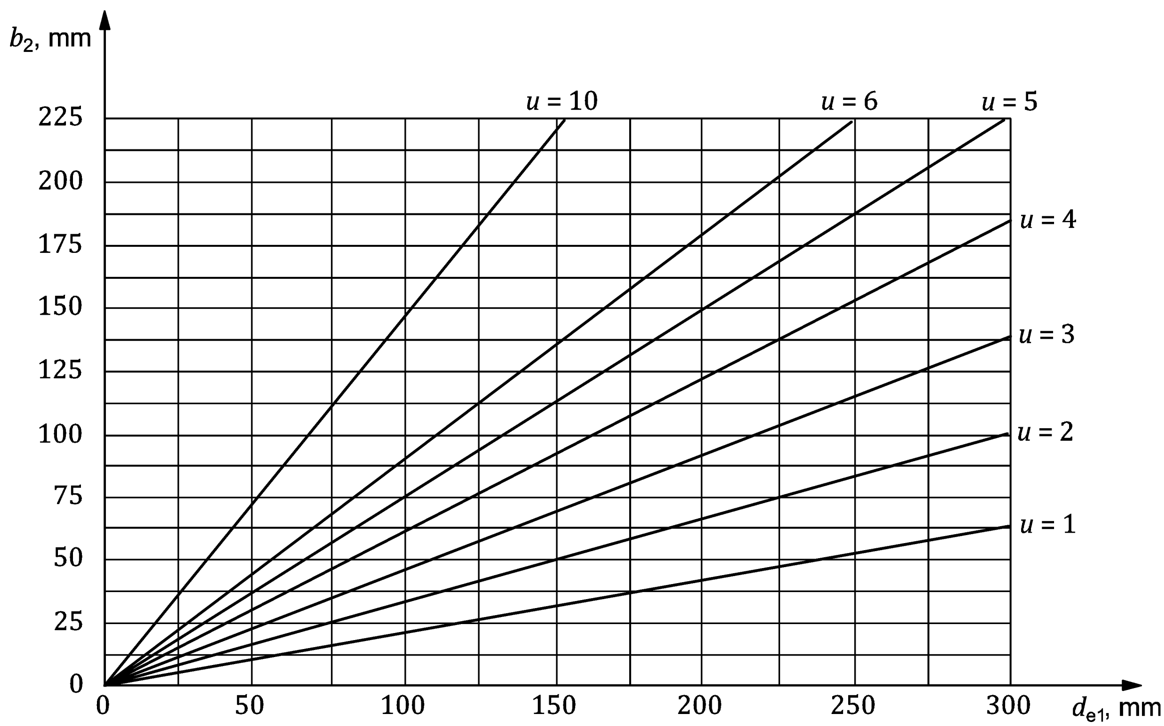


Figure B.6 — Face width of spiral bevel gears operating at 90° shaft angle

B.7 Spiral angle

B.7.1 General

Common design practice suggests that the spiral angle be selected to give a face contact ratio of approximately 2.0. For high-speed applications and maximum smoothness and quietness, face contact ratios greater than 2.0 are suggested, but face contact ratios less than 2.0 are allowed.

B.7.2 Spiral bevels

Formulae (B.7) and (B.8) for face contact ratio, ε_β, may be used to select the spiral angle:

$$K_z = \frac{b}{R_e}\left[\frac{\left(2 - \dfrac{b}{R_e}\right)}{2\left(1 - \dfrac{b}{R_e}\right)}\right] \tag{B.7}$$

$$\varepsilon_\beta = \frac{1}{\pi\, m_{et}}\left(K_z \tan\beta_m - \frac{K_z^3}{3}\tan^3\beta_m\right)R_e \tag{B.8}$$

where

R_e is the outer cone distance, in millimetres (mm);

m_{et} is the outer transverse module, in millimetres (mm);

b is the net face width, in millimetres (mm);

β_m is the mean spiral angle at pitch surface.

Figure B.7 may be used to assist in the selection of spiral angle when the face width is 30 % of the outer cone distance.

B.7.3 Hypoids

For hypoid sets, the pinion spiral angle may be calculated using Formula (B.9):

$$\beta_{m1} = 25 + 5\sqrt{\frac{z_2}{z_1}} + 90\frac{a}{d_{e2}} \tag{B.9}$$

where

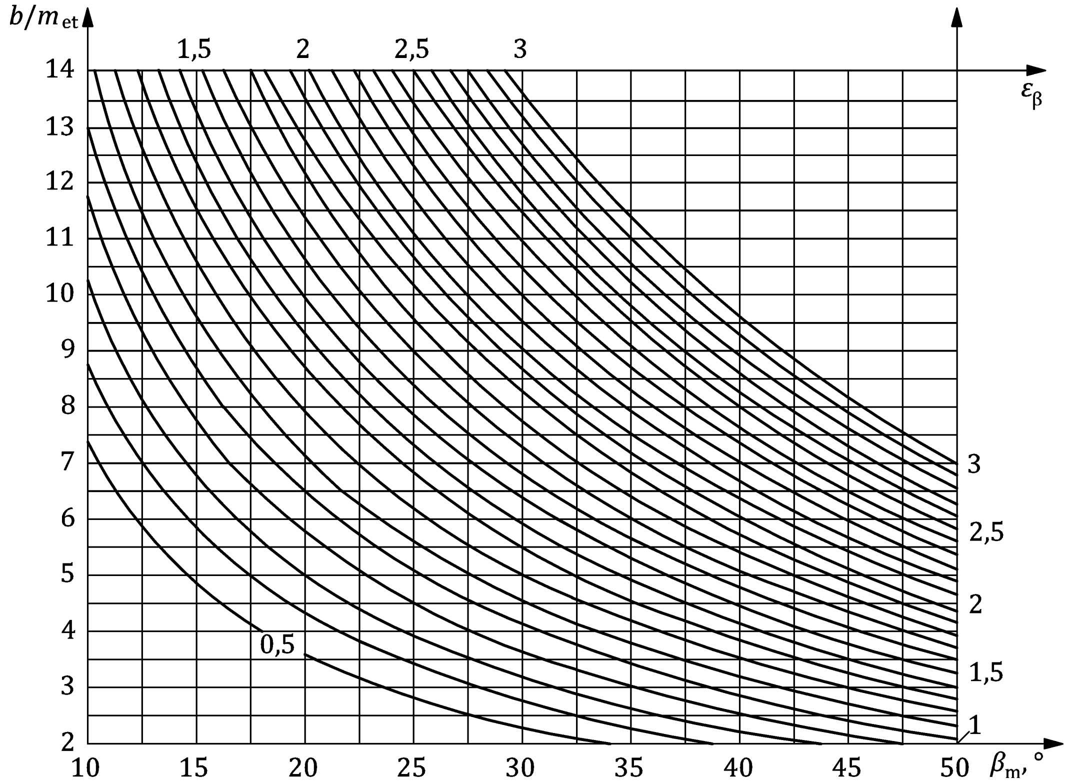
β_{m1} is the pinion mean spiral angle;

z_2 is the number of wheel teeth;

z_1 is the number of pinion teeth;

d_{e2} is the wheel outer pitch diameter, in millimetres (mm).

The wheel spiral angle depends on the hypoid geometry and is calculated using the hypoid formulae in Clause 6.



$$\varepsilon_\beta = \left(0.388\,5 \tan\beta_m - 0.017\,1 \tan^3\beta_m\right) b / m_{et}$$

$$b / R_e = 0.3$$

Figure B.7 — Face contact ratio for spiral bevel gears

B.8 Outer transverse module

The outer transverse module, m_{et2}, is obtained by dividing the outer wheel pitch diameter by the number of teeth in the wheel. Since tooling for bevel gears is not standardized according to module, it is not necessary that the module be an integer.

Annex C
(informative)

Gear dimensions

C.1 Purpose

The purpose of Annex C is to give suggestions for the values of the additional data (see Clause 7) that are necessary to determine the gear dimensions.

C.2 Normal pressure angle

C.2.1 General

There are three normal pressure angles that are to be considered.

— Nominal design pressure angle, α_d, is the start value for the calculation. It may be half of the sum of pressure angles or different on drive and coast side.

— Generated pressure angle, α_n, is the pressure angle of the generating gear; α_n can be found on the tooth flank in the mean normal section.

— Effective pressure angle, α_e, is a calculated value.

The most commonly used design pressure angle for bevel gears is 20°. This pressure angle affects the gear design in a number of ways. Lower generated pressure angles increase the transverse contact ratio, reduce the axial and separating forces and increase the toplands and slot widths. The converse is true for higher pressure angles. Based on the requirements of the application, the engineer may decide to choose higher or lower design pressure angles. Lower effective pressure angles increase the risk of undercut.

For hypoid gears, it could be reasonable to have unequal generated pressure angles on the coast and drive sides, in order to balance the mesh conditions. If full balance of the mesh conditions is recommended, the influence factor of limit pressure angle, $f_{\alpha\mathrm{lim}}$, is set to “1”. Then the limit pressure angle, α_{lim}, is added to the design pressure angle, α_d, on the drive side and subtracted on the coast side in order to obtain the generated normal pressure angle, α_n [see Formula (118) and Formula (119)].

Reducing the generated pressure angles on the drive side may be beneficial for contact ratio, contact stress and axial and radial forces. However, the minimum generated pressure angle should be approximately 9°...10° due to limits of tooling and undercut.

Nevertheless, in all cases, the effective pressure angle, α_e, is calculated according to Formula (120) and Formula (121).

As for bevel (non-hypoid) gears, the limit pressure always is equal to zero, the nominal design pressure angles have the same values as the generated pressure angles. If the effective pressure angles have the same values, the mesh conditions on coast and drive side are equal.

C.2.2 Straight bevels

To avoid undercut, use a nominal design pressure angle of 20° or higher for pinions with 14 teeth to 16 teeth and 25° for pinions with 12 teeth or 13 teeth.

C.2.3 Zerol bevels

On zerol bevels, 22.5° and 25° nominal design pressure angles are used for low tooth numbers, high ratios, or both, to prevent undercut. Use a 22.5° nominal design pressure angle for pinions with 14 teeth to 16 teeth and a 25° nominal design pressure angle for pinions with 13 teeth.

C.2.4 Spiral bevels

To avoid undercut, a 20° design pressure angle or higher for pinions with 12 teeth or fewer teeth may be used.

C.2.5 Hypoids

To balance the mesh conditions on coast and drive side, the influence factor of limit pressure angle should be $f_{\alpha\mathrm{lim}} = 1$. For the use of standard cutting tools, the value of $f_{\alpha\mathrm{lim}}$ may be different from “1”. The nominal design pressure angles 18° or 20° may be used for light-duty drives; higher pressure angles such as 22.5° and 25° for heavy-duty drives.

C.3 Tooth depth components

C.3.1 Data type I

NOTE Data types are described in 7.1.

C.3.1.1 Addendum factor and dedendum factor

In common cases, the addendum factor, k_{hap}, is set to $k_{\mathrm{hap}} = 1$ and the dedendum factor, k_{hfp}, is set to $k_{\mathrm{hfp}} = 1.25$.

C.3.1.2 Profile shift coefficient

To prevent undercut, the profile shift coefficient shall be in the range given in Clause 8.

C.3.2 Data type II

NOTE Data types are described in 7.1.

C.3.2.1 Depth factor

Normally, a depth factor, k_{d}, of 2.000 is used to calculate mean working depth, h_{mw}, but it can be varied to suit design and other requirements. Table C.1 gives the suggested depth factors based on pinion tooth numbers.

Table C.1 — Suggested depth factor, k_d

Type of gear	Depth factor	Number of pinion teeth
Straight bevel	2.000	12 or more
Spiral bevel	2.000	12 or more
	1.995	11
	1.975	10
	1.940	9
	1.895	8
	1.835	7
	1.765	6
Zerol bevel	2.000	13 or more
Hypoid	2.000	11 or more
	1.950	10
	1.900	9
	1.850	8
	1.800	7
	1.750	6

C.3.2.2 Clearance factor

While the clearance is constant along the entire length of the tooth, the calculation is made at mean point. Normally, the value of 0.125 is used for the clearance factor, k_c, but it can be varied to suit the design and other requirements.

During the manufacturing of fine pitch gearing, m_{et2} = 1.27 and finer, 0.051 mm should be added to the clearance of the teeth which are to be finished in a secondary machining operation. This 0.051 mm should not be included in the calculations.

C.3.2.3 Mean addendum factor

This factor apportions the working depth between the pinion and wheel addendums. The pinion addendum is usually longer than the wheel addendum, except when the numbers of teeth are equal. Longer addendums are used on the pinion to avoid undercut. Suggested values for shaft angles $\Sigma = 90°$ for c_{ham} are found in Table C.2. Other values based on sliding velocity, topland or point width limits, or matching strength between two members, can be used. Clause 8 gives the limits for the mean addendum factor to prevent undercut on pinion and wheel. For Table C.2, the equivalent ratio u shall be calculated.

Wheel offset angle in axial plane, η

$$\eta = a\sin\left(\sin\zeta_m \cos\delta_2\right) \tag{C.1}$$

Equivalent ratio, u_a

$$u_a = \sqrt{\frac{\cos\delta_1 \ \tan\delta_2 \ \cos\eta}{\cos\delta_2}} \tag{C.2}$$

Table C.2 — Mean addendum factor for shaft angles Σ = 90°, c_{ham}

Type of gear	Mean addendum factor	Number of pinion teeth
Straight bevel	$0.210 + 0.290/u_a^2$	12 or more
Spiral bevel and hypoid	$0.210 + 0.290/u_a^2$	12 or more
	$0.210 + 0.280/u_a^2$	11
	$0.175 + 0.260/u_a^2$	10
	$0.145 + 0.235/u_a^2$	9
	$0.130 + 0.195/u_a^2$	8
	$0.110 + 0.160/u_a^2$	7
	$0.100 + 0.115/u_a^2$	6
Zerol bevel	$0.210 + 0.290/u_a^2$	13 or more

C.4 Tooth thickness components

C.4.1 Data type I

C.4.1.1 Thickness modification coefficient

Values for the thickness modification coefficient, x_{smn}, can be found, regarding the bending strength balance between pinion and wheel. After the thickness modification, a successful cutting process shall be ensured.

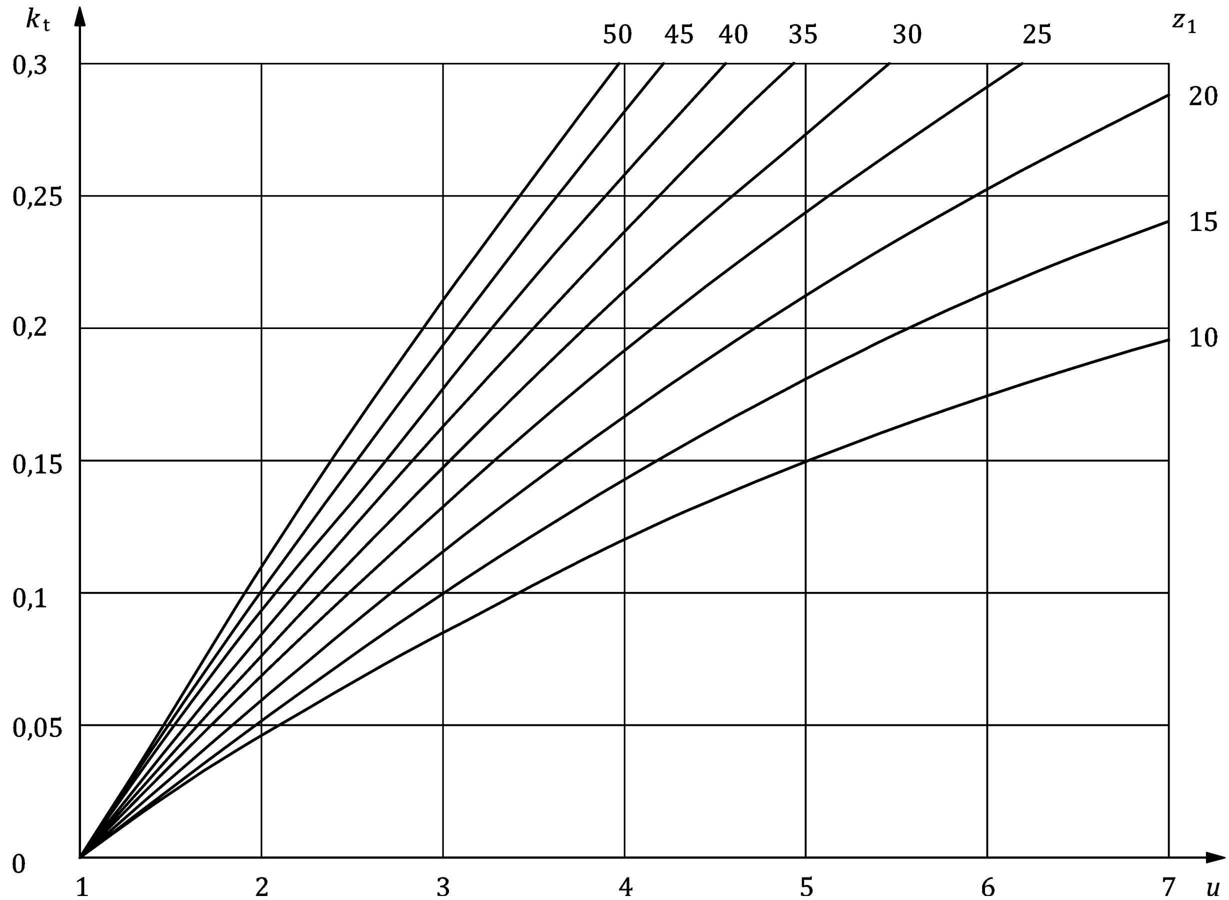
C.4.2 Data type II

C.4.2.1 Thickness factor

The mean normal circular thickness is calculated at the mean point. Values of k_t based on balanced bending stress are found by using the graph in Figure C.1. Other values of k_t may be used if a different strength balance is desired.

C.4.2.2 Outer normal backlash

Suggested minimum values of the outer backlash are given in Table C.3. It will be noted that the backlash allowance is proportional to the module. Two ranges of values are given: one for ISO accuracy grades 4 to 7, the other for ISO accuracy grades 8 to 12, according to ISO 1328-1.



$k_t = -0.088 + 0.092u - 0.004u^2 + 0.001\,6(z_1 - 30)(u - 1)$.

Figure C.1 — Thickness factor, k_t

Table C.3 — Typical minimum normal backlash measured at outer cone

Outer transverse module	Minimum normal backlash mm ISO accuracy grades 4 to 7	Minimum normal backlash mm ISO accuracy grades 8 to 12
25.00 to 20.00	0.61	0.81
20.00 to 16.00	0.51	0.69
16.00 to 12.00	0.38	0.51
12.00 to 10.00	0.30	0.41
10.00 to 8.00	0.25	0.33
8.00 to 6.00	0.20	0.25
6.00 to 5.00	0.15	0.20
5.00 to 4.00	0.13	0.15
4.00 to 3.00	0.10	0.13
3.00 to 2.50	0.08	0.10
2.50 to 2.00	0.05	0.08
2.00 to 1.50	0.05	0.08
1.50 to 1.25	0.03	0.05
1.25 to 1.00	0.03	0.05

C.5 Addendum angle and dedendum angle of wheel

C.5.1 Sum of dedendum angles, $\Sigma\theta_f$

The sum of the dedendum angles of pinion and wheel is a calculated value that is established by the depthwise taper which is chosen in accordance with the cutting method. The formulae for calculating this value are listed in Table C.4.

Table C.4 — Sum of dedendum angles, $\Sigma\theta_f$

Depthwise taper	**Sum of dedendum angles** (degrees)	
Standard	$\Sigma\theta_{fs} = \arctan\left(\frac{h_{fm1}}{R_{m2}}\right) + \arctan\left(\frac{h_{fm2}}{R_{m2}}\right)$	(C.3)
Uniform depth	$\Sigma\theta_{fU} = 0$	
Constant slot width	$\Sigma\theta_{fC} = \left(\frac{90 m_{et}}{R_{e2} \tan\alpha_n \cos\beta_m}\right)\left(1 - \frac{R_{m2} \sin\beta_{m2}}{r_{c0}}\right)$	(C.4)
Modified slot width	$\Sigma\theta_{fM} = \Sigma\theta_{fC}$ or $\Sigma\theta_{fM} = 1.3\ \Sigma\theta_{fS}$, whichever is smaller	(C.5)

C.5.2 Angles, θ_{a2} and θ_{f2}

The sum of the dedendum angles is apportioned between the pinion and the wheel using the formulae in Table C.5. The desired depthwise taper dictates which formulae are to be used when determining the dedendum angles of each member.

Table C.5 — Angles, θ_{a2} and θ_{f2}, wheels

Depthwise taper	**Angles** (degrees)
Standard	$\theta_{a2} = \arctan\left(\frac{h_{fm1}}{R_{m2}}\right)$ (C.6) $\theta_{f2} = \Sigma\theta_{fS} - \theta_{a2}$ (C.7)
Uniform depth	$\theta_{a2} = \theta_{f2} = 0$
Constant slot width	$\theta_{a2} = \Sigma\theta_{fC}\frac{h_{am2}}{h_{mw}}$ (C.8) $\theta_{f2} = \Sigma\theta_{fC} - \theta_{a2}$ (C.9)
Modified slot width	$\theta_{a2} = \Sigma\theta_{fM}\frac{h_{am2}}{h_{mw}}$ (C.10) $\theta_{f2} = \Sigma\theta_{fM} - \theta_{a2}$ (C.11)

Annex D
(informative)

Analysis of forces

D.1 Purpose

The purpose of Annex D is to estimate the forces at the mesh that result from the gear geometry and the transformed torque.

D.2 Analysis of forces

The gear tooth forces result in tangential, axial and radial components, for the purpose of determining the forces and moments which act on shafts and bearings. The axial and radial forces are dependent on the curvature of the loaded tooth flank. Use Table D.1 to determine the loaded flank. The formulae to calculate the forces are presented as follows.

Table D.1 — Loaded flank

Driver hand of spiral	Rotation of driver	Loaded flank	
		Driver	Driven
Right	Clockwise	Convex	Concave
	Anticlockwise (counterclockwise)	Concave	Convex
Left	Clockwise	Concave	Convex
	Anticlockwise (counterclockwise)	Convex	Concave

D.3 Tangential force

The tangential force on a wheel is

$$F_{mt2} = \frac{2\,000\,T_2}{d_{m2}} \tag{D.1}$$

where

F_{mt2} is the tangential force at the mean diameter on the wheel, in newtons (N);

T_2 is the torque transmitted by the wheel, in newton metres (N·m).

The tangential force on the mating pinion is given by Formula (D.2):

$$F_{mt1} = \frac{F_{mt2}\cos\beta_{m1}}{\cos\beta_{m2}} = \frac{2\,000\,T_1}{d_{m1}} \tag{D.2}$$

where

F_{mt1} is the tangential force at the mean diameter on the pinion, in newtons (N).