INTERNATIONAL STANDARD

ISO 10300-3

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Calculation of load capacity of bevel gears —

Part 3: Calculation of tooth root strength

Calcul de la capacité de charge des engrenages coniques —

Partie 3: Calcul de la résistance du pied de dent

Reference number
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Contents

Page

Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this part of ISO 10300 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 10300-3 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation.*

ISO 10300 consists of the following parts, under the general title *Calculation of load capacity of bevel gears*:

— *Part 1: Introduction and general influence factors*

— *Part 2: Calculation of surface durability (pitting)*

— *Part 3: Calculation of tooth root strength*

Annex A forms an integral part of this part of ISO 10300. Annex B is for information only.

Introduction

Parts 1, 2 and 3 of ISO 10300, taken together with ISO 6336-5, are intended to establish general principles and procedures for the calculation of the load capacity of bevel gears. Moreover, ISO 10300 has been designed to facilitate the application of future knowledge and developments, as well as the exchange of information gained from experience. This part of ISO 10300 gives formulae for bending-strength rating in calculations for the avoidance of tooth breakage.

Failure of gear teeth by breakage can be brought about in many ways — severe instantaneous overloads, excessive pitting, case crushing and bending fatigue are some. The strength ratings determined by the formulae in this part of ISO 10300 are based on cantilever-projection theory modified to consider the following:

— compressive stress at the tooth roots caused by the radial component of the tooth load;

— non-uniform moment distribution of the load, resulting from the inclined contact lines on the teeth of spiral bevel gears;

— stress concentration at the tooth root fillet;

— load-sharing between adjacent contacting teeth;

— lack of smoothness due to a low contact ratio.

The formulae can be used for determining a load rating that will prevent tooth root fillet fracture during the design life of the gear teeth. Nevertheless, if there is insufficient material under the teeth (in the rim), a fracture can occur from the root through the rim of the gear blank or to the bore — a type of failure not covered by this part of ISO 10300. Moreover, special applications could require additional blank material to support the load.

Occasionally, surface distress (pitting or wear) may limit the strength rating, due either to stress concentration around large sharp-cornered pits, or to wear steps on the tooth surface. Neither of these effects are considered in this part of ISO 10300.

In most cases, the maximum tensile stress at the tooth root (arising from bending at the root when the load is applied to the tooth flank) can be used as the criterion for the assessment of the bending tooth root strength, as when the allowable stress number is exceeded the teeth can experience breakage. When calculating the tooth root stresses of straight bevel gears, this part of ISO 10300 starts from the assumption that the load is applied at the tooth tip of the virtual cylindrical gear. The load is subsequently converted to the outer point of single-tooth contact with the aid of the contact-ratio factor Y_ε (see clause 8). The procedure thus corresponds to method C for the tooth root stress of cylindrical gears (see ISO 6336-3).

For spiral bevel gears with a high overlap ratio ($\varepsilon_{v\beta} > 1$), the mid point in the contact zone is regarded as the critical point of load application. There is an interpolation for medium overlap ratio ($0 < \varepsilon_{v\beta} < 1$).

The breakage of a tooth generally means the end of a gear's life. It is often the case that all gear teeth are destroyed as a consequence of the breakage of a single tooth. An S_F, the safety factor against tooth breakage, higher than the safety factor against damage due to pitting, is therefore generally to be preferred (see ISO 10300-1).

Calculation of load capacity of bevel gears —

Part 3: Calculation of tooth root strength

1 Scope

This part of ISO 10300 specifies the fundamental formulae for use in the tooth-bending stress calculation of straight and helical (skew), zerol- and spiral-bevel gears with a minimum rim thickness under the root $\geqslant 3{,}5\ m_{mn}$. All load influences on tooth stress are included, insofar as they are the result of load transmitted by the gearing and able to be evaluated quantitatively. (Stresses such as those caused by the shrink-fitting of gear rims, which are superposed on stresses due to tooth loading, are to be taken into consideration in the calculation of the tooth root stress σ_F or the permissible tooth root stress σ_{FP}.)

The formulae in this part of ISO 10300 are valid for bevel gears with teeth with a transverse contact ratio of $\varepsilon_{v\alpha} < 2$, while the results are valid within the range of the applied factors given in ISO 10300-1 and ISO 6336-3.

This part of ISO 10300 does not apply to stress levels above those permitted for 10^3 cycles, as stresses in that range could exceed the elastic limit of the gear tooth.

CAUTION — The user is cautioned that when the methods are used for large spiral and pressure angles, and for large face width $b > 10\ m_{mn}$, the calculated results of ISO 10300 should be confirmed by experience.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO 10300. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this part of ISO 10300 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 53:1998, *Cylindrical gears for general and heavy engineering — Standard basic rack tooth profile.*

ISO 1122-1:1998, *Vocabulary of gear terms — Part 1: Definitions related to geometry.*

ISO 6336-3, *Calculation of load capacity of spur and helical gears — Part 3: Calculation of tooth bending strength.*

ISO 6336-5, *Calculation of load capacity of spur and helical gears — Part 5: Strength and quality of materials.*

ISO 10300-1:2001, *Calculation of load capacity of bevel gears — Part 1: Introduction and general influence factors.*

ISO 10300-2:2001, *Calculation of load capacity of bevel gears — Part 2: Calculation of surface durability (pitting).*

3 Terms and definitions

For the purposes of this part of ISO 10300, the geometrical gear terms given in ISO 53 and ISO 1122-1, and the following term and definition, apply.

3.1
tooth bending strength
load capacity determined on the basis of the permissible bending stress

4 Symbols and abbreviated terms

For the purposes of this part of ISO 10300, the symbols and abbreviated terms given in Table 1 of ISO 10300-1:2001, and the following abbreviated terms, apply.

Table 1 — Abbreviated terms

Abbreviation	Description
St	steel (σ_B<800 N/mm^2)
V	through-hardened steel ($\sigma_B \geqslant$ 800 N/mm^2)
GG	grey cast iron
GGG (perl., bai., ferr.)	spheroidal cast iron (perlitic, bainitic, ferritic structure)
GTS (perl.)	black malleable cast iron (perlitic structure)
Eh	case-hardening steel, case-hardened
IF (root)	steel and GGG, flame or induction-hardened (including root fillet)
NT (nitr.)	nitriding steels, nitrided
NV (nitr.)	through-hardened and case-hardening steel, nitrided
NV (nitrocar.)	through-hardened and case-hardening steels, nitro-carburized

5 Tooth breakage and safety factors

Tooth breakage usually ends transmission service life. Sometimes the destruction of all gears in a transmission is a consequence of the breakage of one tooth, while in certain instances the transmission path between input and output shafts is broken.

Because of this, the chosen value of the safety factor, S_F, against tooth breakage should be larger than the square of the safety factor, S_H, against pitting (see ISO 10300-1 for general comments on the choice of safety factor).

The value of the minimum safety factor for bending stress, S_{Fmin}, should be $\geqslant$ 1,3 for spiral bevel gears. For straight bevel gears, or where $\beta_m \leqslant 5°$, S_{Fmin} should be $\geqslant$ 1,5. It is recommended that the manufacturer and customer agree on the value of the minimum safety factor.

6 Gear-tooth rating formulae

6.1 General

The capacity of a gear tooth to resist bending shall be determined by the comparison of the following stress values:

— **bending stress**, based on the geometry of the tooth, the accuracy of its manufacture, the rigidity of the gear blanks, bearings and housing, and the operating torque, expressed by the bending stress formula (see 6.2);

— **allowable stress number**, and the effect of the working conditions under which the gears operate, expressed by the permissible bending stress formula (see 6.3).

The determined tooth root stress, σ_F, shall be $\leqslant \sigma_{FP}$, which is the permissible tooth root stress.

NOTE In respect of the allowable stress, reference is made to a stress "number", a designation adopted because pure stress, as determined by laboratory testing, is not calculated by the formulae in this part of ISO 10300. Instead, an arbitrary value is calculated and used throughout, with accompanying changes to the allowable stress number in order to maintain consistency for design comparison.

6.2 Tooth root stress

6.2.1 General

The tooth root stress is determined separately for pinion and wheel:

$$\sigma_F = \sigma_{FO} K_A K_v K_{F\beta} K_{F\alpha} \leqslant \sigma_{FP} \quad (1)$$

where

σ_{F0} is the local tooth root stress defined as the maximum tensile stress arising at the tooth root due to the nominal torque when a perfect gear is loaded.

See ISO 10300-1 for K_A, K_v, $K_{F\beta}$, $K_{F\alpha}$

6.2.2 Local tooth root stress, $\sigma_{F0\text{-}B1}$ — Method B1

The calculation of the local tooth root stress is based on the maximum tensile stress at the tooth root (30° tangent to the tooth root fillet). The determinant position of load application is:

a) the outer limit of single tooth contact ($\varepsilon_{v\beta} = 0$);

b) the mid-point of the zone of contact ($\varepsilon_{v\beta} \geqslant 1$);

c) interpolation between a) and b) ($0 < \varepsilon_{v\beta} < 1$).

The transformation from tip to this determinant position is done by Y_ε:

$$\sigma_{F0\text{-}B1} = \frac{F_{mt}}{b m_{mn}} Y_{Fa} Y_{Sa} Y_\varepsilon Y_K Y_{LS} \quad (2)$$

where

F_{mt} is the nominal tangential force at the reference cone at mid-face width (see ISO 10300-1);

b is the face width;

Y_{Fa} is the tooth form factor (see clause 7), which accounts for the influence of the tooth form on the nominal bending stress for load application at the tooth tip;

Y_{Sa} is the stress correction factor (see clause 7), which accounts for the conversion of the nominal bending stress for load application at tooth tip to the corresponding local tooth root stress. Thus Y_{Sa} accounts for the stress-increasing effect of the notch (in the root fillet), as well as for the fact that the stress condition in the critical root section is complex, but not for the influence of the bending moment arm;

Y_ε is the contact-ratio factor (see clause 8), which accounts for the conversion of the local stress determined for the load application at the tooth tip to the determinant position;

Y_K is the bevel-gear factor, which accounts for smaller values for l_b' compared to total face width b and the inclined lines of contact;

Y_{LS} is the load sharing factor, which accounts for load distribution between two or more pairs of teeth.

6.2.3 Local tooth root stress, $\sigma_{F0\text{-}B2}$ — Method B2

When applying method B2, the combined geometry factor Y_P replaces the factors Y_{Fa}, Y_{Sa}, Y_ε, Y_{LS} and Y_K in the local tooth root stress Equation such that Equation (2) becomes:

$$\sigma_{F0\text{-}B2} = \frac{F_{mt}}{b m_{mn}} Y_P \tag{3}$$

The value of Y_P is determined by:

$$Y_P = \frac{Y_A}{Y_J} \frac{m_{mt} m_{mn}}{m_{et}^2} \tag{4}$$

Substitution in Equation (3):

$$\sigma_{F0\text{-}B2} = \frac{F_{mt}}{b} \frac{m_{mt}}{m_{et}^2} \frac{Y_A}{Y_J} \tag{5}$$

where

Y_A is the bevel-gear adjustment factor for method B2, for standard carburized and case-hardened bevel gears (see annex A);

Y_J is the bending strength geometry factor for method B2 (see 9.2).

The bending-strength geometry factor, Y_J, evaluates the shape of the tooth, the position at which the most damaging load is applied, the stress concentration due to the geometric shape of the root fillet, the sharing of load between adjacent pairs of teeth, the tooth-thickness balance between the wheel and mating pinion, the effective face width due to lengthwise crowning of the teeth, and the buttressing effect of an extended face width on one member of the pair. Both the tangential (bending) and radial (compressive) components of the tooth load are included.

6.3 Permissible tooth root stress

6.3.1 General

The permissible tooth root stress, σ_{FP}, is determined separately for pinion and wheel. It should be calculated on the basis of the strength determined at an actual gear, as this way the reference value for geometrical similarity, course of movement and manufacture will lie within the field of application.

$$\sigma_{FP} = \frac{\sigma_{FE}\, Y_{NT}}{S_{F\min}}\, Y_{\delta\,\mathrm{rel\,T}}\, Y_{R\,\mathrm{rel\,T}}\, Y_X \tag{6}$$

$$\sigma_{FP} = \frac{\sigma_{F\lim}\, Y_{ST}\, Y_{NT}}{S_{F\min}}\, Y_{\delta\,\mathrm{rel\,T}}\, Y_{R\,\mathrm{rel\,T}}\, Y_X \tag{7}$$

where

σ_{FE} is the allowable stress number (bending),

σ_{FE} $= \sigma_{F\lim}\, Y_{ST}$, the basic bending strength of the un-notched specimen under the assumption that the material (including heat treatment) is fully elastic;

$\sigma_{F\lim}$ is the bending stress number for the nominal stress in bending of the test gear, which accounts for material, heat treatment, and surface influence at test gear dimensions (see ISO 6336-5);

Y_{ST} is the stress-correction factor for the dimensions of the standard test gear $Y_{ST} = 2{,}0$;

$S_{F\min}$ is the minimum safety factor (see ISO 10300-1);

$Y_{\delta\,\mathrm{rel\,T}}$ is the relative sensitivity factor (see clause 10) for the allowable stress number, related to the conditions at the standard test gear ($Y_{\delta\,\mathrm{rel\,T}} = Y_\delta / Y_{\delta T}$ accounts for the notch sensitivity of the material);

$Y_{R\,\mathrm{rel\,T}}$ is the relative surface condition factor (see clause 11) ($Y_{R\,\mathrm{rel\,T}} = Y_R / Y_{RT}$ accounts for the surface condition at the root fillet, related to the conditions at the test gear);

Y_X is the size factor for tooth root strength (see clause 12), which accounts for the influence of the module on the tooth root strength;

Y_{NT} is the life factor, which accounts for the influence of required numbers of cycles of operation (see clause 13).

6.3.2 Calculated safety factor

The calculated safety factor against tooth breakage shall be determined separately for pinion and wheel. On the basis of the allowable stress number (bending), determined for the standard test gear:

$$S_F = \frac{\sigma_{FE}\, Y_{NT}}{\sigma_{F0}}\, \frac{Y_{\delta\,\mathrm{rel\,T}}\, Y_{R\,\mathrm{rel\,T}}\, Y_X}{K_A\, K_v\, K_{F\beta}\, K_{F\alpha}} \tag{8}$$

NOTE This is the calculated safety factor with respect to the transmitted torque.

See ISO 10300-1 in reference to the safety factor and the risk (probability) of failure.

7 Tooth form, Y_{Fa}, and correction, Y_{Sa}, factors — Method B1

7.1 General

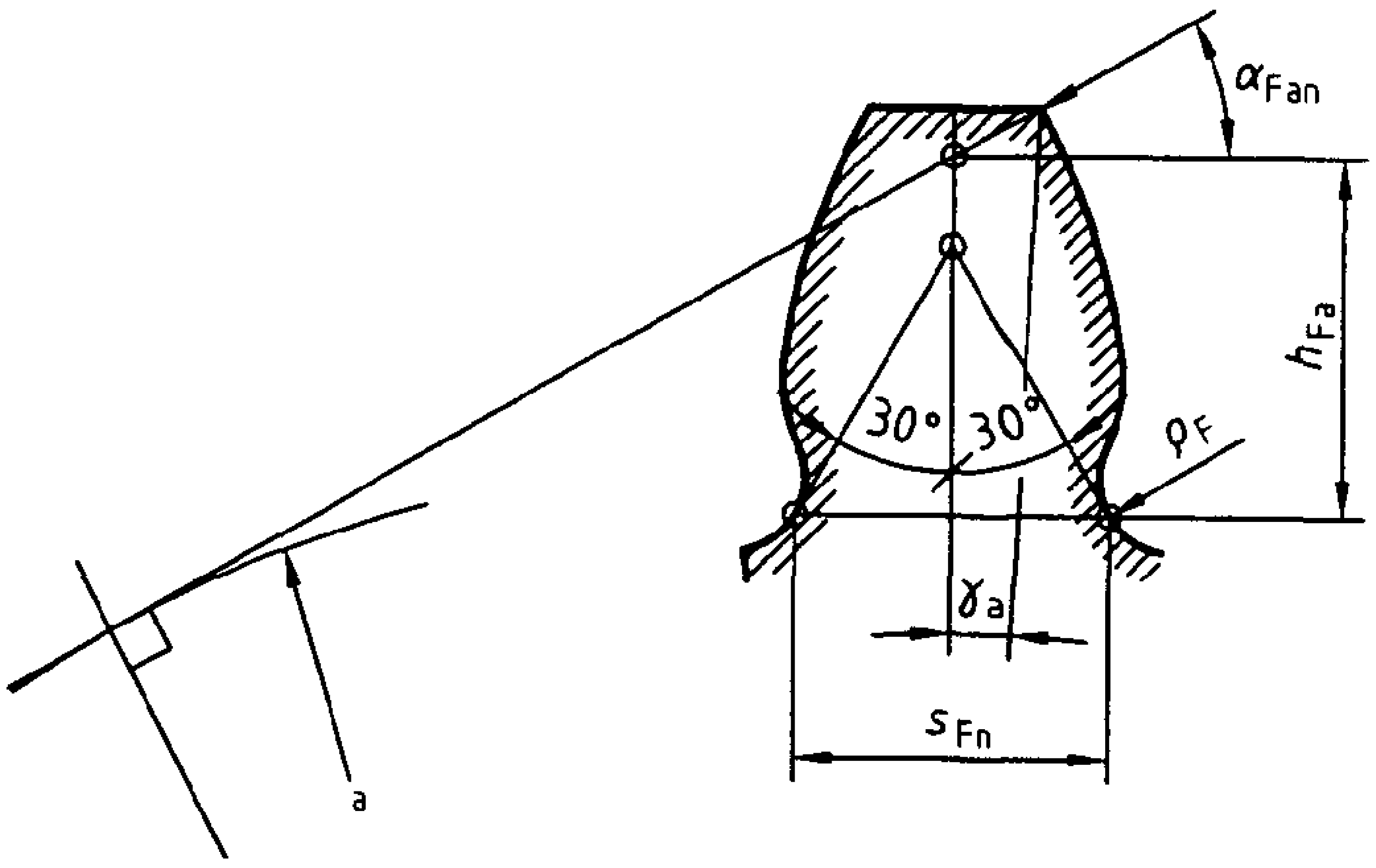
The tooth form factor, Y_{Fa}, accounts for the influence of the tooth form on the nominal bending stress in the case of load application at the tooth tip. It is determined separately for pinion and wheel.

NOTE In the case of gears with tip and root relief, the actual bending moment arm is slightly smaller, and the calculation is on the safe side.

Bevel gears generally have octoid teeth and a tip and root relief. However, deviations from an involute profile are small, especially in view of the tooth root chord and bending moment arm, and thus they may be neglected when calculating the tooth-form factor.

The distance between the contact points of the 30° tangents at the root fillets of the tooth profile of the virtual cylindrical gear is taken as a cross-section for calculation (see Figure 1).

In this part of ISO 10300, Y_{Fa} and Y_{Sa} are determined for the nominal gear without tolerances. The slight reduction in tooth thickness for backlash between teeth may be neglected for the load capacity calculation. The size reduction shall be taken into account when the outer tooth thickness allowance $A_{sne} > 0{,}05\ m_{mn}$.



a Base circle of virtual cylindrical gear

Figure 1 — Tooth root chord, s_{Fn}, and bending moment arm for load application at the tooth tip, h_{Fa}, of the virtual cylindrical gear tooth profile

7.2 Y_{Fa} for generated gears

7.2.1 General

Equation (9) applies to virtual cylindrical gears in normal section with and without profile shift. The calculation is valid under the following premises:

a) The contact point of the 30° tangent lies on the fillet curve generated by the tool tip radius.

b) The tool is manufactured with a finite tip radius ($\rho_{a0} \neq 0$).

$$Y_{Fa} = \frac{6 \frac{h_{Fa}}{m_{mn}} \cos\alpha_{Fan}}{\left(\frac{s_{Fn}}{m_{mn}}\right)^2 \cos\alpha_n} \qquad (9)$$

See Figure 1 for the respective definitions; see ISO 6336-3 for an evaluation of the decisive normal tooth load and tooth form factor.

7.2.2 Auxiliary quantities

For the calculation of the tooth root chord, s_{Fn}, and bending moment arm, h_{Fa}, first the auxiliary quantities E, G, H and ϑ need to be determined:

$$E = \left(\frac{\pi}{4} - x_{sm}\right) m_{mn} - h_{a0} \tan\alpha_n - \frac{\rho_{a0}(1 - \sin\alpha_n) - s_{pr}}{\cos\alpha_n} \qquad (10)$$

$$G = \frac{\rho_{a0}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm} \tag{11}$$

$$H = \frac{2}{z_{vn}} \left(\frac{\pi}{2} - \frac{E}{m_{mn}} \right) - \frac{\pi}{3} \tag{12}$$

$$\vartheta = \frac{2G}{z_{vn}} \tan\vartheta - H \tag{13}$$

For the solution of the transcendent Equation (13), $\vartheta = \pi/6$ may be inserted as the initial value. In most cases, the Equation already converges after a few iteration steps.

7.2.3 Tooth root chord, s_{Fn}

$$\frac{s_{Fn}}{m_{mn}} = z_{vn} \sin\left(\frac{\pi}{3} - \vartheta \right) + \sqrt{3} \left(\frac{G}{\cos\vartheta} - \frac{\rho_{a0}}{m_{mn}} \right) \tag{14}$$

7.2.4 Fillet radius, ρ_F, at contact point of 30° tangent

$$\frac{\rho_F}{m_{mn}} = \frac{\rho_{a0}}{m_{mn}} + \frac{2G^2}{\cos\vartheta \left(z_{vn} \cos^2\vartheta - 2G \right)} \tag{15}$$

7.2.5 Bending moment arm, h_{Fa}

$$\alpha_{an} = \arccos\left(\frac{d_{vbn}}{d_{van}} \right) \tag{16}$$

$$\gamma_a = \frac{1}{z_{vn}} \left[\frac{\pi}{2} + 2\left(x_{hm} \tan\alpha_n + x_{sm} \right) \right] + \operatorname{inv}\alpha_n - \operatorname{inv}\alpha_{an} \tag{17}$$

$$\alpha_{Fan} = \alpha_{an} - \gamma_a \tag{18}$$

$$\frac{h_{Fa}}{m_{mn}} = \frac{1}{2} \left[\left(\cos\gamma_a - \sin\gamma_a \tan\alpha_{Fan} \right) \frac{d_{van}}{m_{mn}} - z_{vn} \cos\left(\frac{\pi}{3} - \vartheta \right) - \frac{G}{\cos\vartheta} + \frac{\rho_{a0}}{m_{mn}} \right] \tag{19}$$

See annex A of ISO 10300-1:2001 for data of the virtual cylindrical gear in normal section. Dimensions at the basic rack profile of the tooth are shown in Figure 2 of this part of ISO 10300, while Y_{Fa} may be taken from Figure 3 for a basic rack profile of the tool with data $\alpha_n = 20°$, $h_{a0}/m_{mn} = 1{,}25$, and $\rho_{a0}/m_{mn} = 0{,}25$ for $x_{sm} = 0$. Diagrams for other basic rack profiles are given in ISO 6336-3.

See Figures 4 to 6 for the combined tooth form factor $Y_{FS} = Y_{Fa}\, Y_{Sa}$ for generated bevel gears.

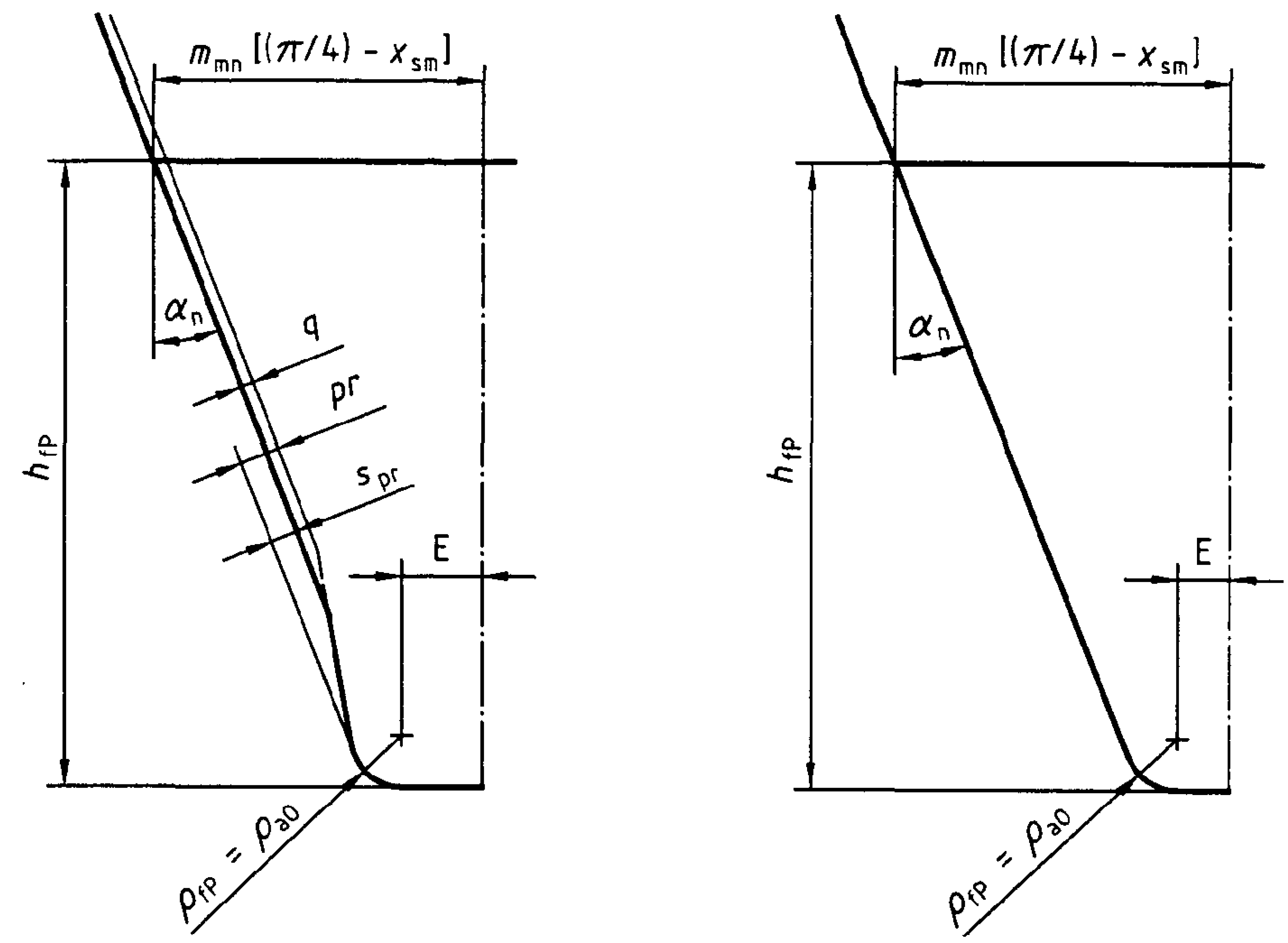


Figure 2 — Dimensions at the basic rack profile of the tooth

7.3 Y_{Fa} for gears made by form cutting

Crown gears can be partly manufactured by the form cutting method (especially for larger gear ratios), by which the profile of the space width of the measured gear is identical to the tool profile (rack profile). Here, the tooth form factor for the crown gear can be directly determined at the tool profile.

Tooth root thickness:

$$s_{Fn2} = \pi\, m_{mn} - 2E - 2\,\rho_{a02} \cos 30° \tag{20}$$

with E according to Equation (10)

fillet radius at contact point of 30° tangent

$$\rho_{F2} = \rho_{a02} \tag{21}$$

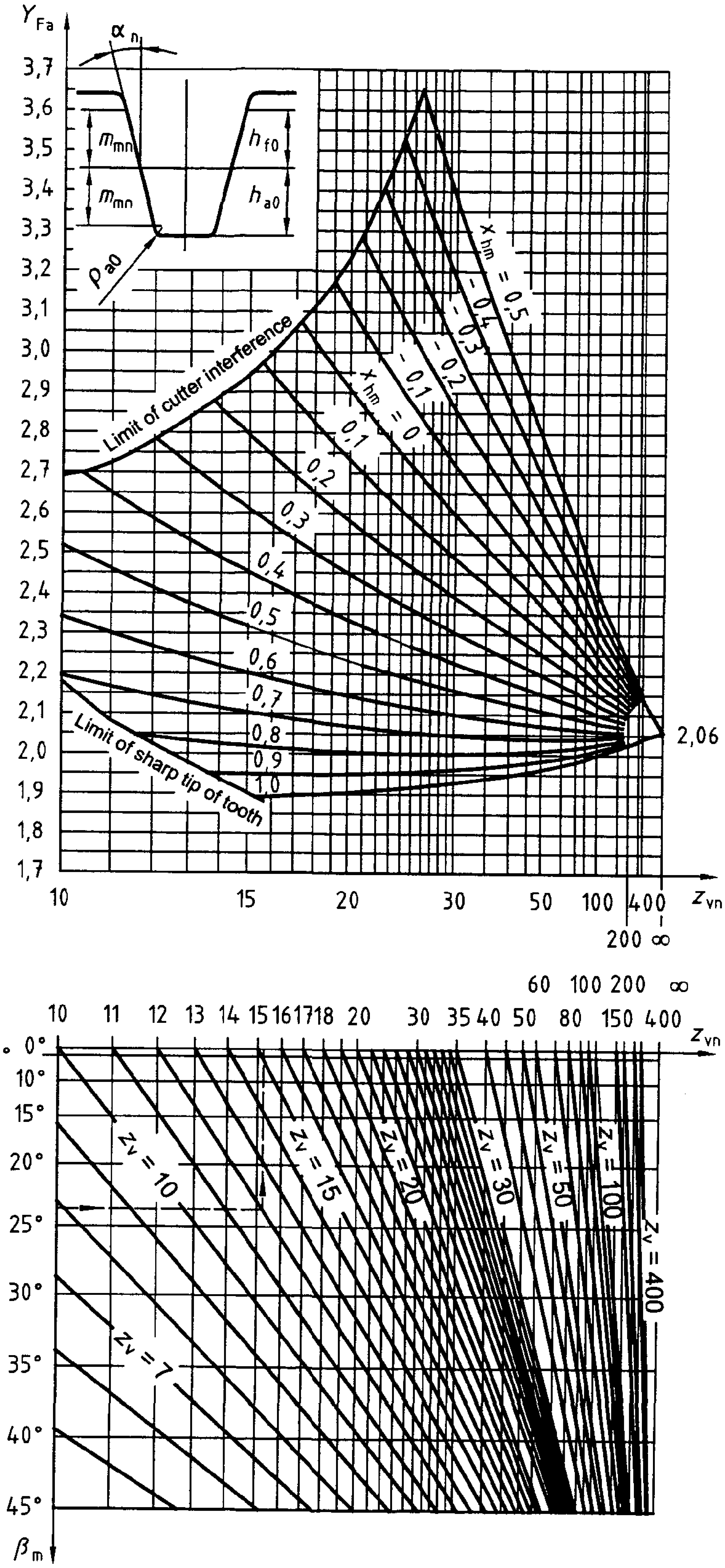
bending moment arm

$$h_{Fa2} = h_{a02} - \frac{\rho_{ao2}}{2} + m_{mn} - \left(\frac{\pi}{4} + x_{sm2} - \tan\alpha_n\right) m_{mn} \tan\alpha_n \tag{22}$$

tooth form factor according to Equation (9), with $\alpha_{Fan} = \alpha_n$

$$Y_{Fa2} = \frac{\dfrac{6\,h_{Fa2}}{m_{mn}}}{\left(\dfrac{s_{Fn2}}{m_{mn}}\right)^2} \tag{23}$$

The tooth form factor of the pertaining bevel pinion manufactured by the generating method may be approximated according to 7.2 in the case of gear ratios $u > 3$.



(Example: β_m = 23,5, z_v = 12 leads to z_{vn} = 15,2)

Figure 3 — Tooth form factor, Y_{Fa}, for generated gears

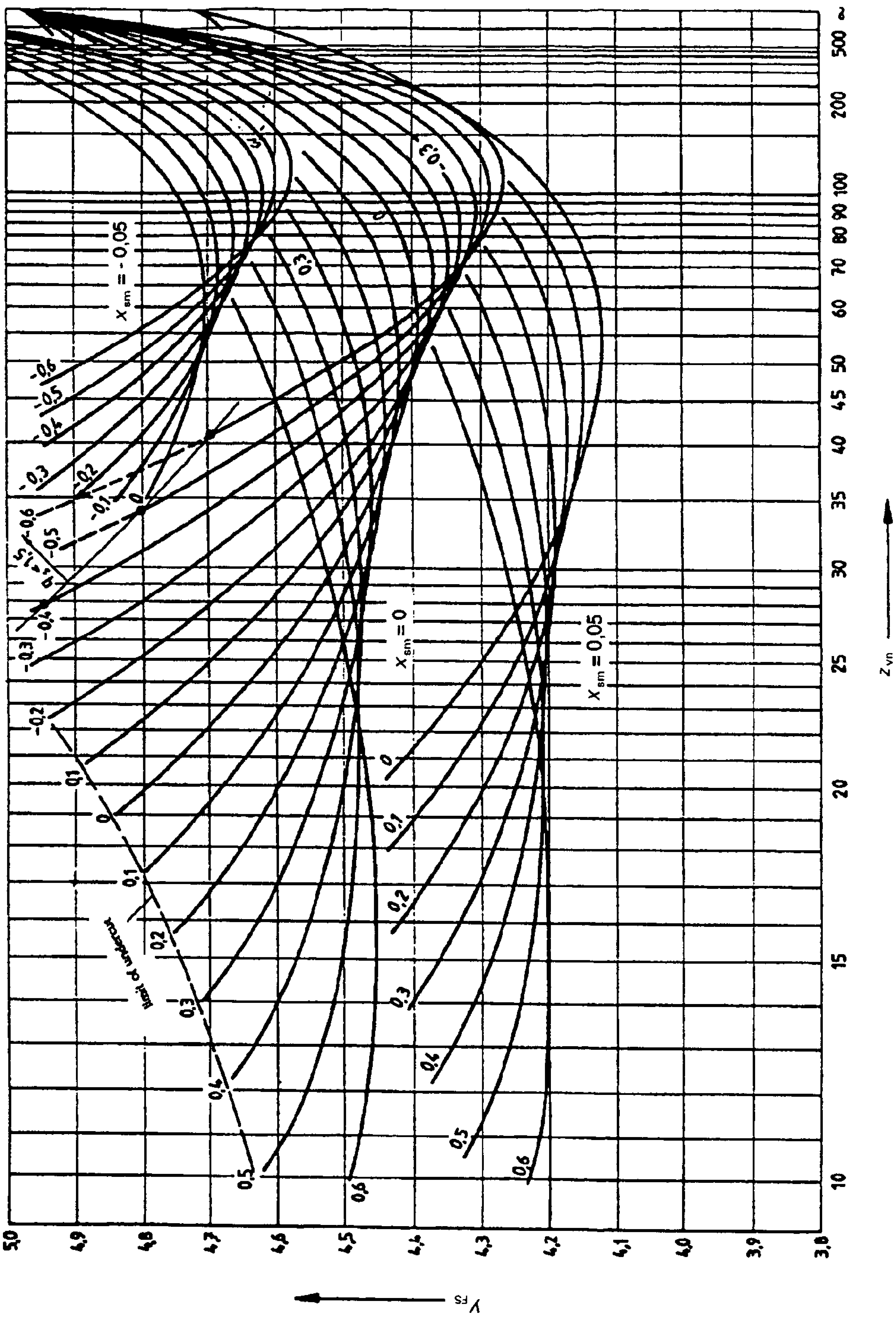


Figure 4 — Combined tooth form factor $Y_{FS} = Y_{Fa} Y_{Sa}$ for generated gears ($\rho_{a0} = 0{,}2\ m_{mn}$)

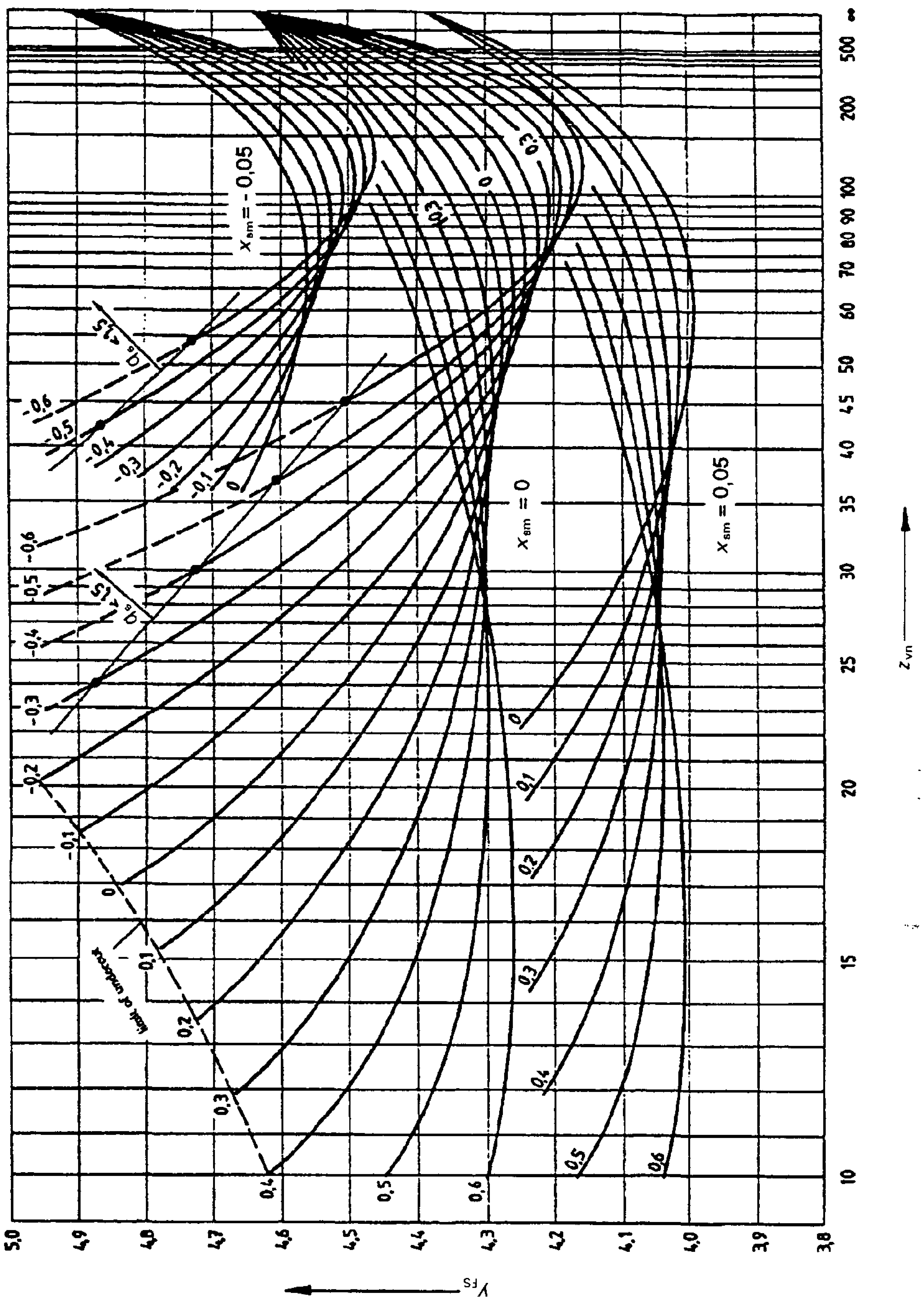


Figure 5 — Combined tooth form factor $Y_{FS} = Y_{Fa} Y_{Sa}$ **for gears generated by basic rack** ($\rho_{a0} = 0,25\ m_{mn}$)

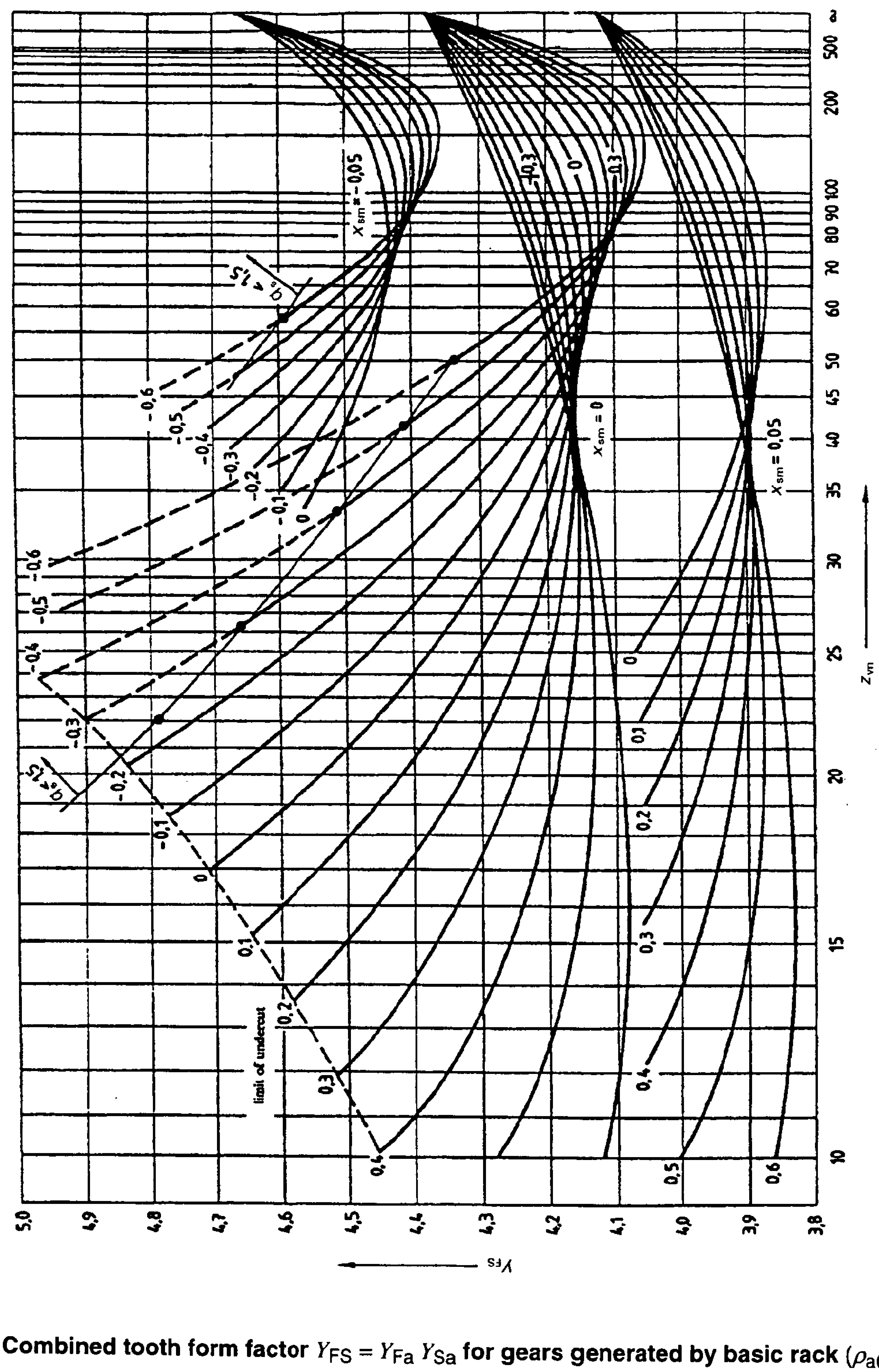


Figure 6 — Combined tooth form factor $Y_{FS} = Y_{Fa} Y_{Sa}$ for gears generated by basic rack ($\rho_{a0} = 0,3\ m_{mn}$)

7.4 Correction factor, Y_{Sa}

By means of the stress correction factor, Y_{Sa}, the nominal bending stress is converted to the local tooth root stress. Y_{Sa} accounts for the stress increasing effect of the notch (= root fillet) as well as for other stress components that arise beside the bending stress. (See ISO 6336-3 for further remarks.)

$$Y_{Sa} = \left(1{,}2 + 0{,}13 L_a\right) q_s^{\left(\frac{1}{1{,}21 + 2{,}3/L_a}\right)} \qquad (24)$$

$$L_a = \frac{s_{Fn}}{h_{Fa}} \qquad (25)$$

$$q_s = \frac{s_{Fn}}{2 \rho_F} \qquad (26)$$

where

s_{Fn} is according to Equation (14) and Equation (20) respectively;

h_{Fa} is according to Equation (19) and Equation (22) respectively;

ρ_F is according to Equation (15) and Equation (21) respectively.

The range of validity of Equation (24) is $1 \leqslant q_s < 8$.

The stress correction factor, Y_{Sa}, may be read from Figure 7 for the basic rack profile of the tool with $\alpha_n = 20°$, $h_{a0}/m_{mn} = 1{,}25$, $\rho_{a0}/m_{mn} = 0{,}25$ for $x_{sm} = 0$. See ISO 6336-3 for the influence of grinding notches.

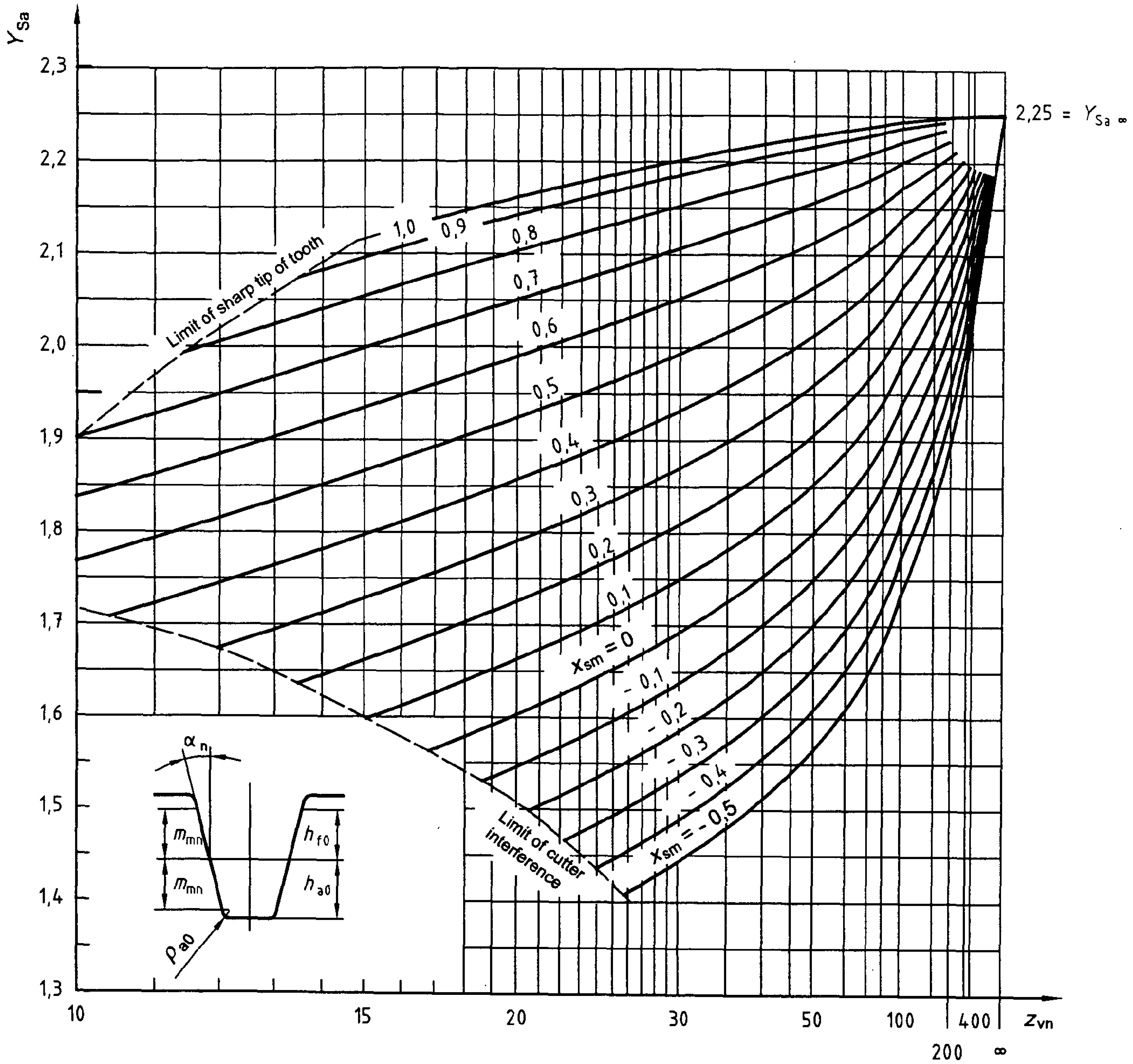


Figure 7 — Stress correction factor for load application at tooth tip, Y_{Sa}

8 Contact-ratio, Y_ε, bevel-gear, Y_K, and load-sharing, Y_{LS}, factors — Method B1

8.1 Contact-ratio factor, Y_ε

The contact-ratio factor, Y_ε, converts the load application at the tooth tip (here the tooth form factor, Y_{Fa}, and stress-correction factor, Y_{Sa}, apply) to the decisive point of load application.

Y_ε is also used for the determination of the transverse load distribution factor $K_{F\alpha}$ (see ISO 10300-1):

$$Y_\varepsilon = 0{,}25 + \frac{0{,}75}{\varepsilon_{v\alpha}} \geqslant 0{,}625 \qquad (\varepsilon_{v\beta} = 0) \qquad (27)$$

$$Y_\varepsilon = 0{,}25 + \frac{0{,}75}{\varepsilon_{v\alpha}} - \varepsilon_{v\beta}\left(\frac{0{,}75}{\varepsilon_{v\alpha}} - 0{,}375\right) \geqslant 0{,}625 \qquad (0 < \varepsilon_{v\beta} \leqslant 1) \tag{28}$$

$$Y_\varepsilon = 0{,}625 \qquad (\varepsilon_{v\beta} > 1) \tag{29}$$

8.2 Bevel-gear factor, Y_K

The bevel gear factor, Y_K, accounts for differences between bevel and cylindrical gears (smaller values of l'_{bm} because of inclined lines of contact):

$$Y_K = \left(\frac{1}{2} + \frac{1}{2}\cdot\frac{l'_{bm}}{b}\right)^2 \frac{b}{l'_{bm}} \tag{30}$$

where

l'_{bm} is the projected length of the middle line of contact [see Equation (A.44) in ISO 10300-1:2001].

8.3 Load sharing factor, Y_{LS}

The load sharing factor, Y_{LS}, accounts for load sharing between two or more pair of teeth.

$$Y_{LS} = Z_{LS}^2 \tag{31}$$

(See ISO 10300-2 for Z_{LS}.)

9 Bending-strength combined geometry factor, Y_P — Method B2

9.1 Graphs and general

Figures in annex B contain graphs for the bevel geometry factor, Y_J, for straight-, zerol- and spiral-bevel gears for a series of gear designs, based on the smaller of the face widths b = 0,3 R_e and 10 m_{et}. These may be used whenever the tooth proportions and thickness, face widths, tool edge radii, pressure and spiral angles of the design, and driving with the concave side, correspond to those in the graphs.

Where the graphs are not applicable, the formulae in 9.2 should be used. Because of the complexity of the calculation, computerization is recommended.

9.2 Formula for the bevel geometry factor, Y_J

The bevel geometry factor, Y_J, is calculated using the following formulae:

$$Y_{J1,2} = \frac{Y_{K1,2}}{\varepsilon_N Y_i} \frac{2 r_{my01,2}}{d_{v1,2}} \frac{b_{ce1,2}}{b_{1,2}} \frac{m_{mt}}{m_{et}} \tag{32}$$

where

$Y_{K1,2}$ is the tooth form factor, including the stress concentration factor of pinion or wheel (see 9.8);

ε_N is the load sharing ratio (see 9.9);

Y_i is the inertia factor for gears with a low contact ratio (see 9.10);

$r_{my01,2}$ is the mean transverse radius to point of load application for pinion or wheel, in millimetres (see 9.4);

$b_{ce1,2}$ is the calculated effective face width of pinion or wheel, in millimetres (see 9.11).

9.3 Point of load application for maximum bending stress, y_3

For most straight-, zerol- and spiral-bevel gears, the maximum bending stress occurs at the equivalent of the highest point of single tooth contact when the modified contact ratio is $\leqslant 2$. When the modified contact ratio is > 2, it is assumed that the line of contact passes through the centre of the path of action. For statically loaded straight-bevel and zerol-bevel gears, such as those used in automotive differentials, the load is applied at the tip of the tooth. In any case, the position is measured along the path of action from its centre, and is designated by y_J. Its distance from the beginning of the path of action is designated by y_3.

When $\varepsilon_{v\gamma} \leqslant 2{,}0$

$$y_J = p_{et} \cos \beta_{vb} - \frac{g_\eta}{2} \tag{33}$$

with

$$g_\eta^2 = g_{v\alpha n}^2 \cos^4 \beta_{vb} + b^2 \sin^2 \beta_{vb} \tag{34}$$

when $\varepsilon_{v\gamma} > 2{,}0$

$$Y_J = 0 \tag{35}$$

For statically loaded straight-bevel and zerol-bevel gears (tip loading):

$$y_J = \frac{g_\eta}{2} \tag{36}$$

$$g_J^2 = g_\eta^2 - 4 y_J^2 \tag{37}$$

For straight-bevel and zerol-bevel gears:

$$y_3 = \frac{g_{v\alpha n}}{2} + \frac{g_{v\alpha n}^2 \, y_J}{g_\eta^2} \tag{38}$$

For spiral-bevel gears:

$$y_{31} = \frac{g_{v\alpha n}}{2} + \frac{g_{v\alpha n}^2 \, y_J \cos^2 \beta_{vb} + b \, g_{v\alpha n} \, g_J \, k' \sin \beta_{vb}}{g_\eta^2} \tag{39}$$

$$y_{32} = \frac{g_{v\alpha n}}{2} + \frac{g_{v\alpha n}^2 \, y_J \cos^2 \beta_{vb} - b \, g_{v\alpha n} \, g_J \, k' \sin \beta_{vb}}{g_\eta^2} \tag{40}$$

where constant of location

$$k' = \frac{z_2 - z_1}{3{,}2 z_2 + 4{,}0 z_1} \tag{41}$$

9.4 Radius to point of load application, $r_{my\,01,2}$

Since the point of load application does not usually lie in the mean section of the tooth, the actual radius is determined using the following formulae.

For straight- and zerol-bevel gears:

$$g''_0 = \frac{b g_{\mathrm{v\alpha n}} g_{\mathrm{J}} k'}{g_\eta^2} \tag{42}$$

For spiral-bevel gears:

$$g''_{01} = \frac{b g_{\mathrm{v\alpha n}} g_{\mathrm{J}} k' \cos^2\beta_{\mathrm{vb}} - b^2 y_{\mathrm{J}} \sin\beta_{\mathrm{vb}}}{g_\eta^2} \tag{43}$$

$$g''_{02} = \frac{b g_{\mathrm{v\alpha n}} g_{\mathrm{J}} k' \cos^2\beta_{\mathrm{vb}} + b^2 y_{\mathrm{J}} \sin\beta_{\mathrm{vb}}}{g_\eta^2} \tag{44}$$

$$\tan\alpha_{\mathrm{L}1,2} = \frac{y_{31,2} + a_{\mathrm{vn}} \sin\alpha_{\mathrm{n}} - 0{,}5\sqrt{d^2_{\mathrm{van}2,1} - d^2_{\mathrm{vbn}2,1}}}{0{,}5 d_{\mathrm{vbn}1,2}} \tag{45}$$

where

$\alpha_{\mathrm{L}1,2}$ is normal pressure angle at point of load application for pinion and wheel

$$\xi_{\mathrm{h}1,2} = \frac{180°}{\pi}\left(\frac{s_{\mathrm{mn}1,2}}{d_{\mathrm{vn}1,2}} - \mathrm{inv}\,\alpha_{\mathrm{L}1,2} + \mathrm{inv}\,\alpha_{\mathrm{n}}\right) \tag{46}$$

$$\alpha_{\mathrm{h}1,2} = \alpha_{\mathrm{L}1,2} - \xi_{\mathrm{h}1,2} \tag{47}$$

where $\xi_{\mathrm{h}1,2}$ is the rotation angle used in bending strength calculations for pinion and wheel.

The distance from pitch circle to point-of-load application on the pinion and the wheel-tooth centre line, in millimetres, is:

$$\Delta r_{\mathrm{y}01,2} = 0{,}5\left(\frac{d_{\mathrm{vbn}1,2}}{\cos\alpha_{\mathrm{h}1,2}} - d_{\mathrm{vn}1,2}\right) \tag{48}$$

Mean transverse radius to point of load application:

$$r_{\mathrm{my}01,2} = \frac{d_{\mathrm{v}1,2}}{2}\left(\frac{R_{\mathrm{m}} + g''_{01,2}}{R_{\mathrm{m}}}\right) + \Delta r_{\mathrm{y}01,2} \tag{49}$$

9.5 Tooth fillet radius at root circle, r_{mf}

The minimum tooth fillet radius occurs at the point where the fillet becomes tangent to the root circles, and is given by the following formula.

Fillet radius at root of tooth:

$$r_{\mathrm{mf}1,2} = \frac{\left(h_{\mathrm{fm}1,2} - \rho_{\mathrm{a}01,2}\right)^2}{0{,}5 d_{\mathrm{vn}1,2} + h_{\mathrm{fm}1,2} - \rho_{\mathrm{a}01,2}} + \rho_{\mathrm{a}01,2} \tag{50}$$

9.6 Tooth form factors Y_1 and Y_2

The tooth form factor incorporates both the radial and tangential components of the normal load. Since this factor must be determined for the weakest section, its value must be determined by iteration for pinion and wheel separately.

$$g_{yb1,2} = h_{fm1,2} - \rho_{a01,2} \tag{51}$$

$$g_{01,2} = 0{,}5\, s_{mn1,2} + h_{fm1,2} \tan\alpha_n + \rho_{a01,2} \left(\frac{1 - \sin\alpha_n}{\cos\alpha_n} \right) \tag{52}$$

where $g_{f01,2}$ = assumed value.

For an initial value, make $g_{f01,2\,(1)} = g_{01,2} + g_{yb1,2}$

$$\xi_{1,2} = \frac{360^\circ\, g_{f01,2}}{\pi\, d_{vn1,2}} \tag{53}$$

$$g_{xb1,2} = g_{f01,2} - g_{01,2} \tag{54}$$

$$g_{za1,2} = g_{yb1,2} \cos\xi_{1,2} - g_{xb1,2} \sin\xi_{1,2} \tag{55}$$

$$g_{zb1,2} = g_{yb1,2} \sin\xi_{1,2} + g_{xb1,2} \cos\xi_{1,2} \tag{56}$$

$$\tan = \tau_{1,2} \frac{g_{za1,2}}{g_{zb1,2}} \tag{57}$$

$$s_{N1,2} = 0{,}5\, d_{vn1,2} \sin\xi_{1,2} - \rho_{a01,2} \cos\tau_{1,2} - g_{zb1,2} \tag{58}$$

$$h_{N1,2} = \Delta r_{y01,2} + 0{,}5\, d_{vn1,2} \left(1 - \cos\xi_{1,2}\right) + \rho_{a01,2} \sin\tau_{1,2} + g_{za1,2} \tag{59}$$

Change $g_{f01,2}$ until

$$\frac{s_{N1,2} \cot\tau_{1,2}}{h_{N1,2}} = 2{,}0 \pm 0{,}001 \tag{60}$$

For the second trial, make $g_{f01,2(2)} = g_{f01,2(1)} + 0{,}005\, m_{et}$

For the third and subsequent trials, interpolate.

Tooth-strength factor:

$$x_{N1,2} = \frac{s_{N1,2}^2}{h_{N1}} \tag{61}$$

Tooth-form factor:

$$Y_{1,2} = \frac{2}{3} \left[\frac{1}{m_{et} \left(\dfrac{1}{x_{N1,2}} - \dfrac{\tan\alpha_{h1,2}}{3 s_{N1,2}} \right)} \right] \tag{62}$$

9.7 Stress-concentration and -correction factor, Y_f

The stress-concentration and stress-correction factor, Y_f, depends on:

a) effective-stress concentration;

b) location of the load;

c) plasticity effects;

d) residual-stress effects;

e) material-composition effects;

f) surface finish resulting from gear production and subsequent service;

g) hertz-stress effects;

h) size effects;

i) end-of-tooth effects.

The following stress-concentration and stress-correction factors, derived by Dolan and Broghamer, consider a) and b) only.

$$Y_{f1,2} = L + \left(\frac{2 s_{N1,2}}{r_{mf1,2}} \right)^{M} \left(\frac{2 s_{N1,2}}{h_{N1,2}} \right)^{O} \qquad (63)$$

where

$L = 0{,}325\,454\,5 - 0{,}007\,272\,7\ \alpha_n$

$M = 0{,}331\,818\,2 - 0{,}009\,090\,9\ \alpha_n$

$O = 0{,}268\,181\,8 + 0{,}009\,090\,9\ \alpha_n$

α_n is the pressure angle, in degrees.

Other factors from a) to i) may be included under those covered by Equation (2). Usually, d) and e) are included in allowable bending-stress number σ_{FP}, while h) is included in size factor Y_x, and i) for bevel gears in calculated effective face width, b_{ce} (see 9.11).

9.8 Adjusted tooth-form factor for stress concentration, Y_K

This is simply the combination of the tooth-form factor, $Y_{1,2}$, and the stress-concentration and -correction factor, $Y_{f1,2}$:

$$Y_{K1,2} = \frac{Y_{1,2}}{Y_{f1,2}} \tag{64}$$

9.9 Load-sharing ratio, ε_N

The load-sharing ratio, ε_N, is used to calculate the proportion of the total load carried on the tooth being analysed. It is given by the following formulae:

$$g_J'^3 = g_J^3 + \sum_{k=1}^{k=x} \sqrt{\left[g_J^2 - 4kp_{et}\cos\beta_{vb}\left(kp_{et}\cos\beta_{vb} + 2y_J\right)\right]^3} + \sum_{k=1}^{k=y} \sqrt{\left[g_J^2 - 4kp_{et}\cos\beta_{vb}\left(kp_{et}\cos\beta_{vb} - 2y_J\right)\right]^3} \tag{65}$$

In Equation (65), k is a positive integer which takes on successive values from 1 to x or y, generating all real terms (positive values under the radical) in each series. Imaginary terms (negative values under the radical) should be ignored. For most designs, x and y will not be greater than 2.

$$\varepsilon_N = \frac{g_J^3}{g_J'^3} = \text{load sharing ratio} \tag{66}$$

For statically loaded straight- and zerol-bevel gears:

$$\varepsilon_N = 1{,}0. \tag{67}$$

9.10 Inertia factor, Y_i

The inertia factor, Y_i, allows for the lack of smoothness of the tooth action in dynamically loaded gears with a relatively low contact ratio, and is given as follows:

$$Y_i = \frac{2{,}0}{\varepsilon_{v\gamma}} \quad \text{when } \varepsilon_{v\gamma} < 2{,}0\text{, otherwise, } Y_i = 1{,}0 \tag{68}$$

For statically loaded gears, such as those in vehicle drive-axle differential gears, Y_i equals 1,0 even when $\varepsilon_{v\gamma}$ is less than 2,0.

9.11 Calculated effective face width, b_{ce}

This quantity evaluates the effectiveness of the tooth in distributing the load over the root cross-section, as the instantaneous line of contact frequently does not extend over the entire face width. The following formulae are used to determine the value of the effective face width:

$$g_K = \frac{b\, g_{v\alpha n}\, g_J \cos^2\beta_{vb}}{g_\eta^2} \tag{69}$$

where

g_K is the projected length of the instantaneous line of contact in the lengthwise direction of the tooth, in millimetres.

For the toe increment:

$$\Delta b'_{i1,2} = \frac{b - g_K}{2\cos\beta_m} + \frac{g''_{01,2}}{\cos\beta_m} \tag{70}$$

For the heel increment:

$$\Delta b'_{e1,2} = \frac{b - g_K}{2\cos\beta_m} - \frac{g''_{01,2}}{\cos\beta_m} \tag{71}$$

$\Delta b_{i1,2} = \Delta b'_{i1,2}$ when $\Delta b'_{i1,2}$ and $\Delta b'_{e1,2}$ are both positive

$\Delta b_{i1,2} = \dfrac{b - g_K}{\cos\beta_m}$ when $\Delta b'_{i1,2}$ is positive and $\Delta b'_{e1,2}$ is negative

$\Delta b_{i1,2} = 0$ when $\Delta b'_{i1,2}$ is negative and $\Delta b'_{e1,2}$ is positive

$\Delta b_{e1,2} = \Delta b'_{e1,2}$ when $\Delta b'_{e1,2}$ and $\Delta b'_{i1,2}$ are both positive

$\Delta b_{e1,2} = \dfrac{b - g_K}{\cos\beta_m}$ when $\Delta b'_{e1,2}$ is positive and $\Delta b'_{i1,2}$ is negative

$\Delta b_{e1,2} = 0$ when $\Delta b'_{e1,2}$ is negative and $\Delta b'_{i1,2}$ is positive

$$b_{ce1,2} = h_{N1,2} \cos\beta_m \frac{\pi}{180°}\left[\arctan\left(\frac{\Delta b_{i1,2}}{h_{N1,2}}\right) + \arctan\left(\frac{\Delta b_{e1,2}}{h_{N1,2}}\right)\right] + g_K \tag{72}$$

where $b_{ce1,2}$ is the calculated effective face width, in millimetres.

10 Relative sensitivity factor for allowable stress number, $Y_{\delta\,\mathrm{rel\,T}}$

10.1 General

The dynamic sensitivity factor, Y_δ, indicates the amount by which the theoretical stress peak exceeds the allowable stress number in the case of fatigue breakage. It is a function of the material and relative stress drop. The sensitivity factor can be calculated on the basis of strength values determined at unnotched or notched specimens, or at test gears. If more exact test results (method A) are unavailable, the methods described in this clause shall be used.

10.2 Method B1

The determination of allowable tooth root stresses of bevel gears (and their virtual cylindrical gears) is based on the strength values determined for both bevel and cylindrical test gears. Therefore, the relative sensitivity factor, $Y_{\delta\,\mathrm{rel\,T}}$, is the ratio between the sensitivity factor of the gear to be calculated and the sensitivity factors of the standard test gear. $Y_{\delta\,\mathrm{rel\,T}} = Y_\delta / Y_{\delta T}$ may be taken directly from Figure 8 as a function of q_s (see clause 7) of the gear to be calculated and of the material.

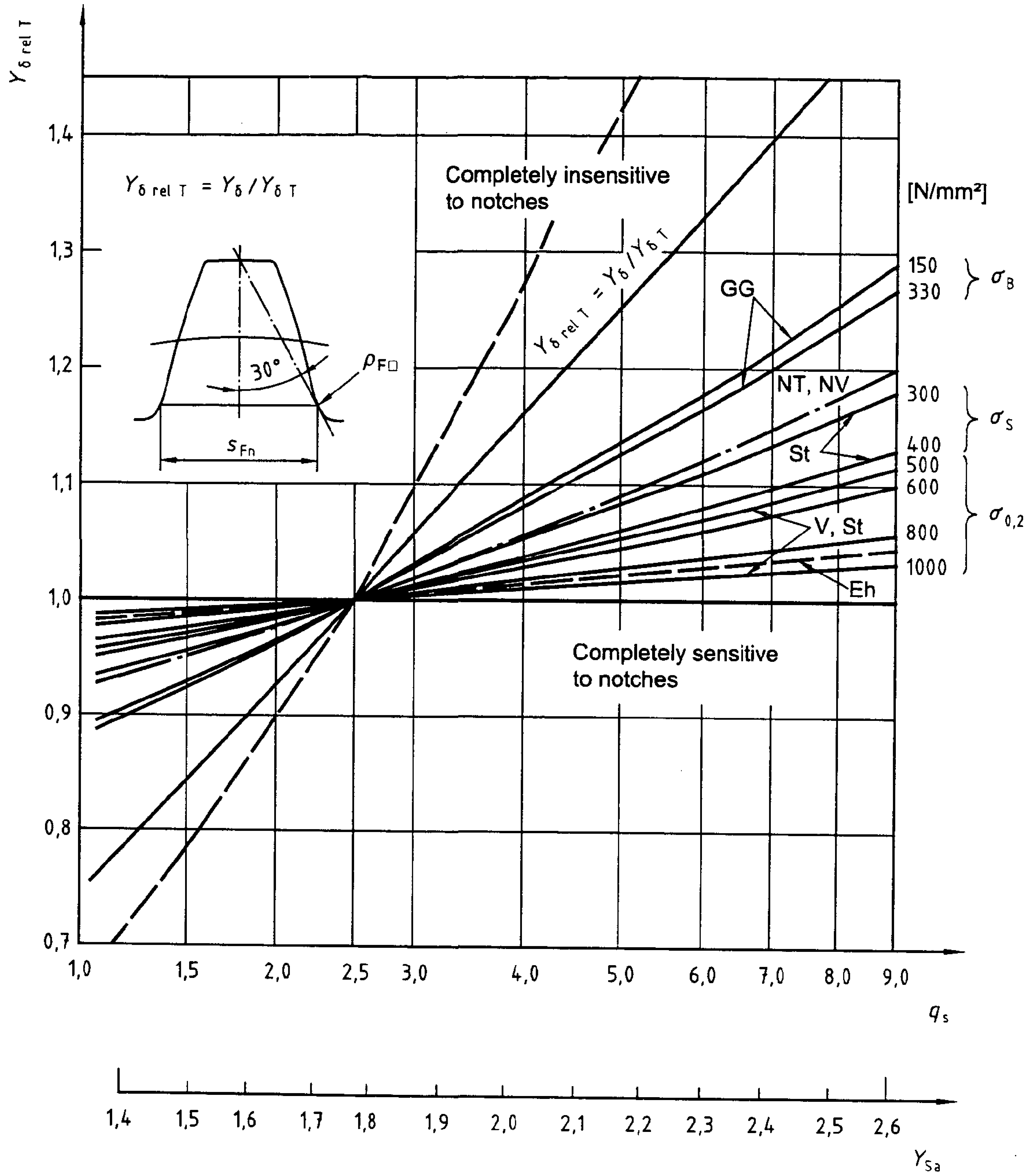


Figure 8 — Relative sensitivity factor in respect of standard-test-gear dimensions for permissible stress number in respect of nominal bending stress

In calculating the relative sensitivity factor, $Y_{\delta\,\mathrm{relT}}$, the following equations, representing the course of the curves in Figure 8, may be used.

$$Y_{\delta\mathrm{relT}} = \frac{1+\sqrt{\rho' \chi^{X}}}{1+\sqrt{\rho' \chi_{T}^{X}}} \qquad (73)$$

$$\chi^{X} = \frac{1}{5}(1+2q_s)$$

$$\chi_{T}^{X} = 1{,}2 \text{ with } q_{sT} = 2{,}5$$

where

ρ' is to be taken from Table 2 as a function of the material.

χ^X applies to module m_{mn} = 5, with the size influence accounted for by Y_X (see clause 12).

Table 2 — Gliding thickness ρ'

No.	Material		Gliding thickness ρ' mm
1	GG	σ_B = 150 N/mm²	0,312 4
2	GG, GGG (ferr.)	σ_B = 300 N/mm²	0,309 5
3	NT, NV		0,100 5
4	St	σ_S = 300 N/mm²	0,083 3
5	St	σ_S = 400 N/mm²	0,044 5
6	V, GTS, GGG (perl., bain.)	$\sigma_{0,2}$ = 500 N/mm²	0,028 1
7	V, GTS, GGG (perl., bain.)	$\sigma_{0,2}$ = 600 N/mm²	0,019 4
8	V, GTS, GGG (perl., bain.)	$\sigma_{0,2}$ = 800 N/mm²	0,006 4
9	V, GTS, GGG (perl., bain.)	$\sigma_{0,2}$ = 1000 N/mm²	0,001 4
10	Eh		0,003 0

10.3 Method B2

This method is generally sufficiently exact for industrial gears. In the case of gears with $q_s \geqslant 1,5$ it is set as:

$$Y_{\delta\ \mathrm{rel\ T}} = 1,0 \tag{74}$$

For $q_s > 2,5$ the calculation is on the safe side.

The reduction of the allowable tooth root stress expected in case of $q_s < 1,5$ is accounted for by:

$$Y_{\delta\ \mathrm{rel\ T}} = 0,95 \tag{75}$$

The limiting lines for $q_s = 1,5$ are plotted in Figure 4 to Figure 6.

11 Relative surface condition factor, $Y_{R\ \mathrm{rel\ T}}$

11.1 General

The surface condition factor, $Y_{R\ \mathrm{rel\ T}}$, accounts for the dependence of the tooth root strength on the surface condition at the root (predominantly the dependence on the roughness in the root fillet), related to standard-test gear conditions with Rz = 10 µm (see ISO 6336-3 for general remarks).

If no surface-condition factors determined by more exact analysis of all influences are available (method A), the methods described in this clause shall be used.

WARNING — These methods are only valid if there are no scratches or similar defects deeper than 2 Rz.

11.2 Method B1

The relative surface condition factor, $Y_{R\,rel\,T}$, determined by tests with test specimens, may be taken from Figure 9 as a function of roughness height and material. For the calculation, Equations (76) to (81) shall be used.

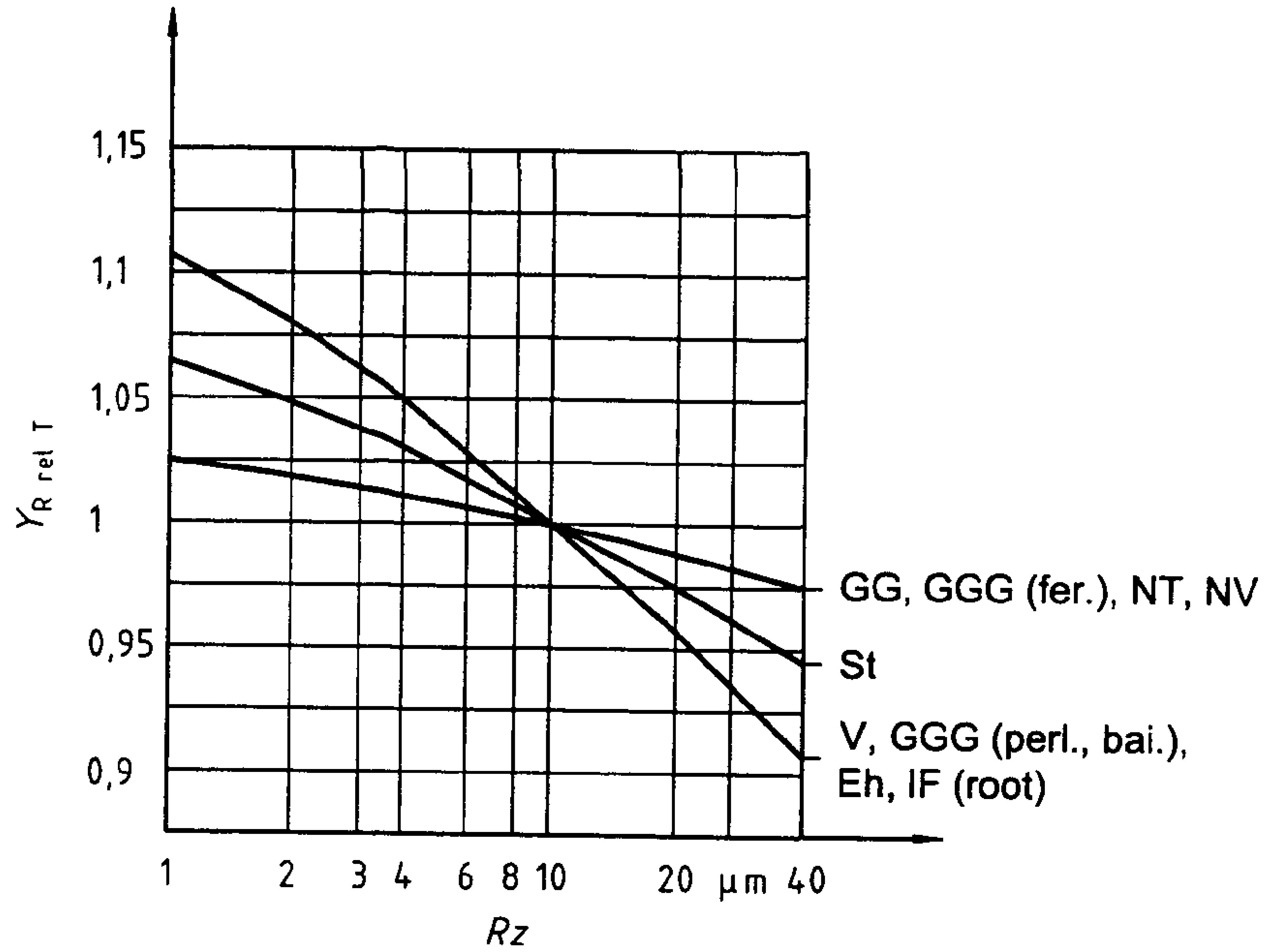


Figure 9 — Surface condition factor, $Y_{R\,rel\,T}$, for permissible stress number relative to test-gear dimensions

Range $Rz < 1$ µm:

For through-hardened and case-hardened steels:

$$Y_{R\,rel\,T} = 1,12 \tag{76}$$

For soft steels:

$$Y_{R\,rel\,T} = 1,07 \tag{77}$$

For grey cast iron, nitrided, and nitro-carburized steels:

$$Y_{R\,rel\,T} = 1,025 \tag{78}$$

Range 1 µm $\leqslant Rz \leqslant$ 40 µm:

For through-hardened and case-hardened steels:

$$Y_{R\,rel\,T} = \frac{Y_R}{Y_{RT}} = 1,674 - 0,529\left(Rz + 1\right)^{1/10} \tag{79}$$

For soft steels:

$$Y_{R\,rel\,T} = \frac{Y_R}{Y_{RT}} = 5,306 - 4,203\left(Rz + 1\right)^{1/100} \tag{80}$$

For grey cast iron, nitrided, and nitro-carburized steels:

$$Y_{\mathrm{R\,rel\,T}} = \frac{Y_\mathrm{R}}{Y_\mathrm{RT}} = 4{,}299 - 3{,}259\left(Rz + 1\right)^{1/200} \qquad (81)$$

11.3 Method B2

For gears with a roughness height of $Rz \leqslant 16$ µm at the root, it may generally be assumed that:

$$Y_{\mathrm{R\,rel\,T}} = 1{,}0 \qquad (82)$$

As shown in Figure 9, the reduction of the allowable stress number is small in the range $10\ \mu m < Rz \leqslant 16\ \mu m$. In the case of $Rz < 10$ µm, the calculation according to Equation (82) is on the safe side.

12 Size factor, Y_X

12.1 General

The size factor, Y_X, accounts for the decrease in strength with increasing size (size effect).

The main factors of influence for Y_X are:

— tooth size;

— diameter of the part;

— ratio of tooth size to diameter;

— area of stress pattern;

— material and heat treatment;

— ratio of case depth to tooth thickness.

If no test values of personal or other proven experience are available, Y_X may be approximated according to Figure 10 as a function of the normal module m_{mn} and the material.

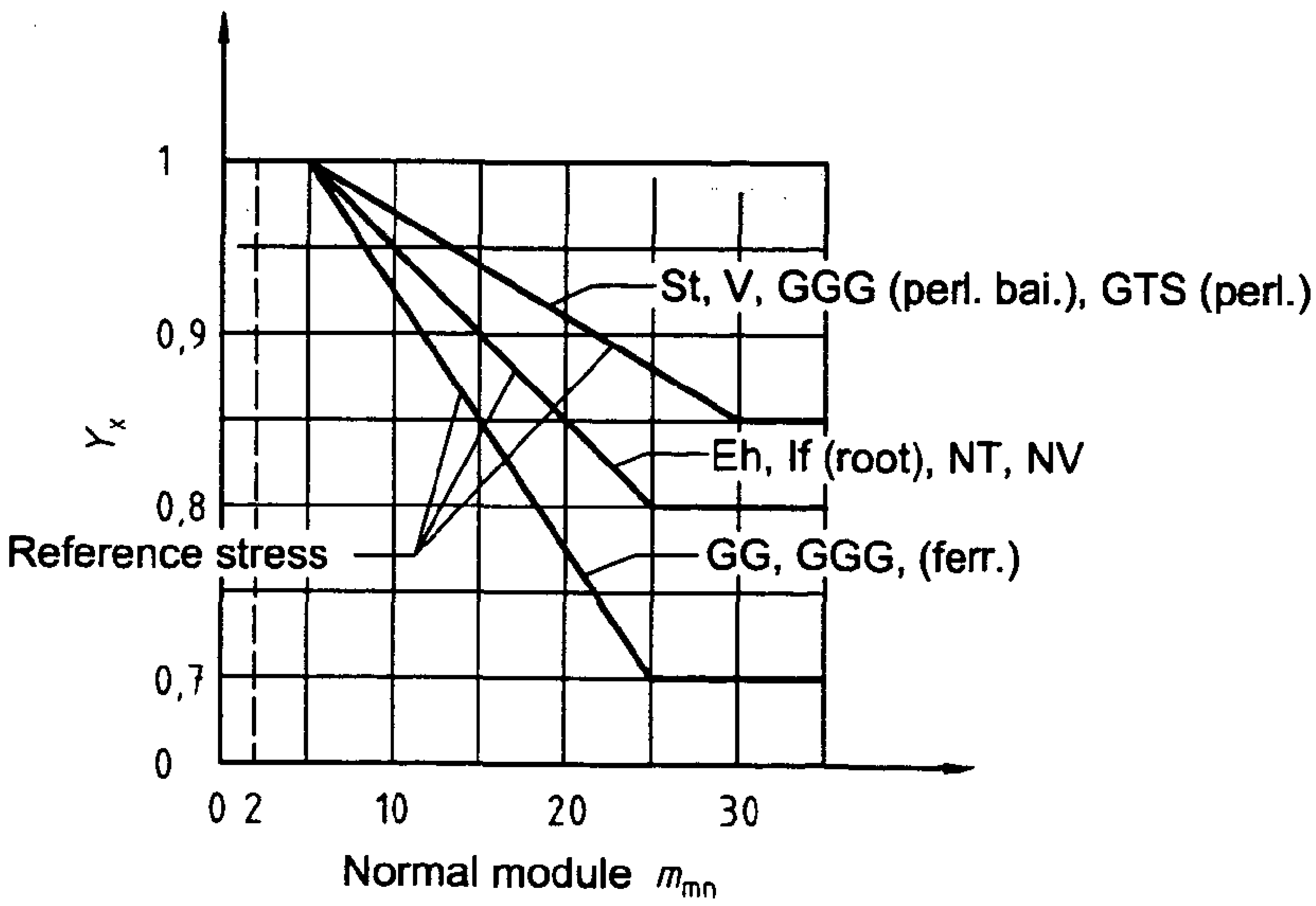


Figure 10 — Size factor, Y_X, for bending stress (allowable stress number)

When Y_X is calculated, Equation (83) to Equation (85), which approximately represent the course of the curves in Figure 10, may be used.

12.2 Formulae

12.2.1 Structural and through-hardened steels, spheroidal cast iron, perlitic malleable cast iron

$$Y_X = 1{,}03 - 0{,}006 m_{mn} \tag{83}$$

with the restriction $0{,}85 \leqslant Y_X \leqslant 1{,}0$

12.2.2 Case, flame, induction-hardened steels, nitrided or nitro-carburized steels

$$Y_X = 1{,}05 - 0{,}01 m_{mn} \tag{84}$$

with the restriction $0{,}80 \leqslant Y_X \leqslant 1{,}0$

12.2.3 Grey cast iron

$$Y_X = 1{,}075 - 0{,}015 m_{mn} \tag{85}$$

with the restriction $0{,}70 \leqslant Y_X \leqslant 1{,}0$.

13 Life factor, Y_{NT}

13.1 General

The life factor, Y_{NT}, accounts for higher bending stress, which may be tolerable for a limited life (number of load cycles), compared with the allowable stress at 3×10^6 cycles.

The principal influence factors for Y_{NT} are:

- material and heat treatment (see ISO 6336-5);
- number of load cycles (service life) N_L;
- failure criteria;
- required smoothness of operation;
- gear-material cleanliness;
- material ductility and fracture toughness;
- residual stress.

For the purposes of ISO 10300, the number of load cycles N_L is defined as the number of mesh contacts, under load, of the gear tooth being analysed. The allowable stress numbers are established for 3×10^6 tooth-load cycles at 99 % reliability.

A Y_{NT} value of unity may be used, where justified by experience, beyond 3×10^6 cycles.

When using unity Y_{NT}, use of the optimum conditions for material quality and manufacturing, together with an appropriate safety factor, should also be considered.

13.2 Method A (Y_{NT-A})

The S-N, or damage, curve derived from facsimiles of the actual gear is determinant for the establishment of limited life. Under such circumstances, the factors $Y_{\delta\ rel\ T}$, $Y_{R\ rel\ T}$ and Y_X are, in effect, already included in the S-N/damage curves, and therefore 1,0 is to be substituted for each in the calculation of permissible stress.

13.3 Method B (Y_{NT-B})

13.3.1 General

For this method, the life factor Y_{NT} of the standard-reference-test gear is used as an aid in the evaluation of permissible stress for limited life or reliability. The factors $Y_{\delta\ rel\ T}$, $Y_{R\ rel\ T}$ and Y_X are not included, and thus the modifying effect of these factors on limited life shall be considered (see ISO 6336).

13.3.2 Graphed values

Graphed values of Y_{NT} may be read from Figure 11 for the static and endurance strengths as a function of material and heat treatment. Values from a large number of tests are presented as typical damage or crack-initiation curves for surface-hardened and nitride-hardened steels, or curves of yield stress for structural and through-hardened steels.

13.3.3 Determination by calculation

The data of life factor, Y_{NT}, for static and endurance strengths shall be taken from Table 3. Y_{NT} for limited-life stress is determined by means of interpolation between the values for endurance and static strength limits. (Evaluation of Y_{NT} is described in ISO 6336.)

WARNING — Stress levels above those permissible for 10^3 cycles should be avoided, since stresses in this range may exceed the elastic limit of the gear tooth.

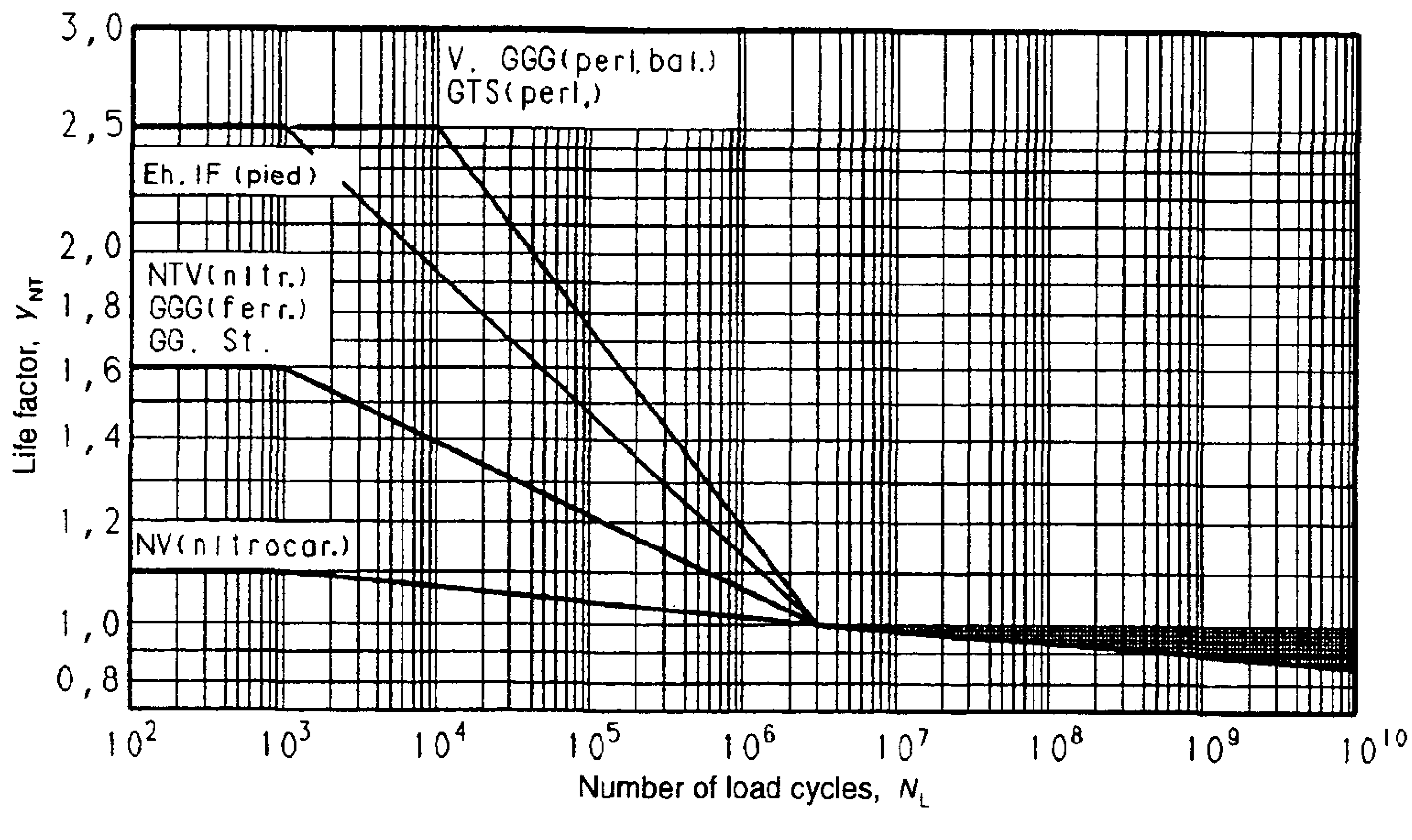


Figure 11 — Life factor, Y_{NT} (standard-reference test gears)

Table 3 — Life factor, Y_{NT}, for static and endurance stress

Material	Number of load cycles, N_L	Life factor, Y_{NT}
V, GGG (perl. bai.), GTS (perl.)	$N_L \leqslant 10^4$, static	2,5
	$N_L = 3 \times 10^6$, endurance	1,0
	$N_L = 10^{10}$, endurance	0,85
	Optimum conditions; material, manufacturing, and experience	1,0
Eh, IF (root)	$N_L \leqslant 10^3$, static	2,5
	$N_L = 3 \times 10^6$, endurance	1,0
	$N_L = 10^{10}$, endurance	0,85
	Optimum conditions; material, manufacturing, and experience	1,0
St, NTV (nitr.), GG, GGG (ferr.)	$N_L \leqslant 10^3$, static	1,6
	$N_L = 3 \times 10^6$, endurance	1,0
	$N_L = 10^{10}$, endurance	0,85
	Optimum conditions; material, manufacturing, and experience	1,0
NV (nitrocar.)	$N_L \leqslant 10^3$, static	1,1
	$N_L = 3 \times 10^6$, endurance	1,0
	$N_L = 10^{10}$, endurance	0,85
	Optimum conditions; material, manufacturing, and experience	1,0

Annex A
(normative)

Bevel gear adjustment factor, Y_A — Method B2

A.1 General

This normative annex specifies two possibilities for the use of the bevel-gear-adjustment factor, Y_A, the factor needed to adjust the results of method B2 to the results of method B1. Without this adjustment the stress numbers for the nominal stress in bending of the test gears according to ISO 6336-5 may not be used. The user of method B2, shall quote the derivation of Y_A.

A.2 Preliminary value for adjustment factor, Y_A

As a preliminary value:

$$Y_{A1,2} = 1{,}2 \quad \text{(A.1)}$$

may be used.

This value of Y_A was determined for gears with $m_{mn} = 5$ mm and with normal proportions, *i.e.* $\alpha_n = 20°$, $\beta_m = 35°$, $z_1 \geqslant 15$ and carburized material.

A.3 Complex Equation for adjustment factor, Y_A

The main difference between method B1 and B2 is that B2 covers not only bending stress but also compression stress [compare Equation (9) and Equation (62)]. The tooth-form factor according to method B2 can be looked upon as the difference between bending-stress factor, Y_B, and compression-stress factor, Y_C:

$$\frac{1}{Y_{1,2}} = \frac{m_{et}}{m_{mn}}\left(Y_{B1,2} - Y_{C1,2}\right) \quad \text{(A.2)}$$

where

$$Y_{B1,2} = m_{mn}\frac{3h_{N1,2}}{2s_{N1,2}^2} \quad \text{(A.3)}$$

$$Y_{C1,2} = m_{mn}\frac{\tan\alpha_{h1,2}}{2s_{N1,2}} \quad \text{(A.4)}$$

with

$Y_{1,2}$ see Equation (62)

$h_{N1,2}$ see Equation (59)

$s_{N1,2}$ see Equation (58)

$\alpha_{h\,1,2}$ see Equation (47)

Considering that $\cos \alpha_{Fan} \approx \cos \alpha_h \approx \cos \alpha_n$, $h_{Fa} \approx h_N$ [1)], and $s_{Fn} \approx 2\, s_N$,

it follows that:

$$Y_B \approx Y_{Fa}\, Y_\varepsilon \tag{A.5}$$

with

Y_{Fa} see Equation (9)

Another difference between methods B1 and B2 is the approach to stress correction [compare Equation (24) with Equation (63)]. A good approximation (including all other differences between these methods) is:

$$Y_{Sa} \approx \frac{1}{2,3} Y_f^2 \tag{A.6}$$

Comparison of B1 and B2 results in:

$$Y_{Fa}\, Y_{Sa}\, Y_\varepsilon \approx Y_A\, Y_f \frac{1}{Y} \frac{m_{mn}}{m_{et}} \tag{A.7}$$

With assumptions according to Equation (A.5) and Equation (A.6) it follows:

$$Y_B \frac{Y_f^2}{2,3} = Y_A\, Y_f \left(Y_B - Y_C\right) \tag{A.8}$$

From here, the adjustment factor Y_A is:

$$Y_{A1,2} = \frac{Y_f}{2,3\left(1 - \dfrac{s_{N1,2}}{3 h_{N1,2}} \tan\alpha_n\right)} \tag{A.9}$$

[1)] The approximation $h_{Fan} \approx h_N$ may be made only if the effect of contact-ratio factor Y_ε (see 8.1) is also considered (see Equation A.5 to Equation A.7).

Annex B
(informative)

Graphs of geometry factor, Y_J — Method B2

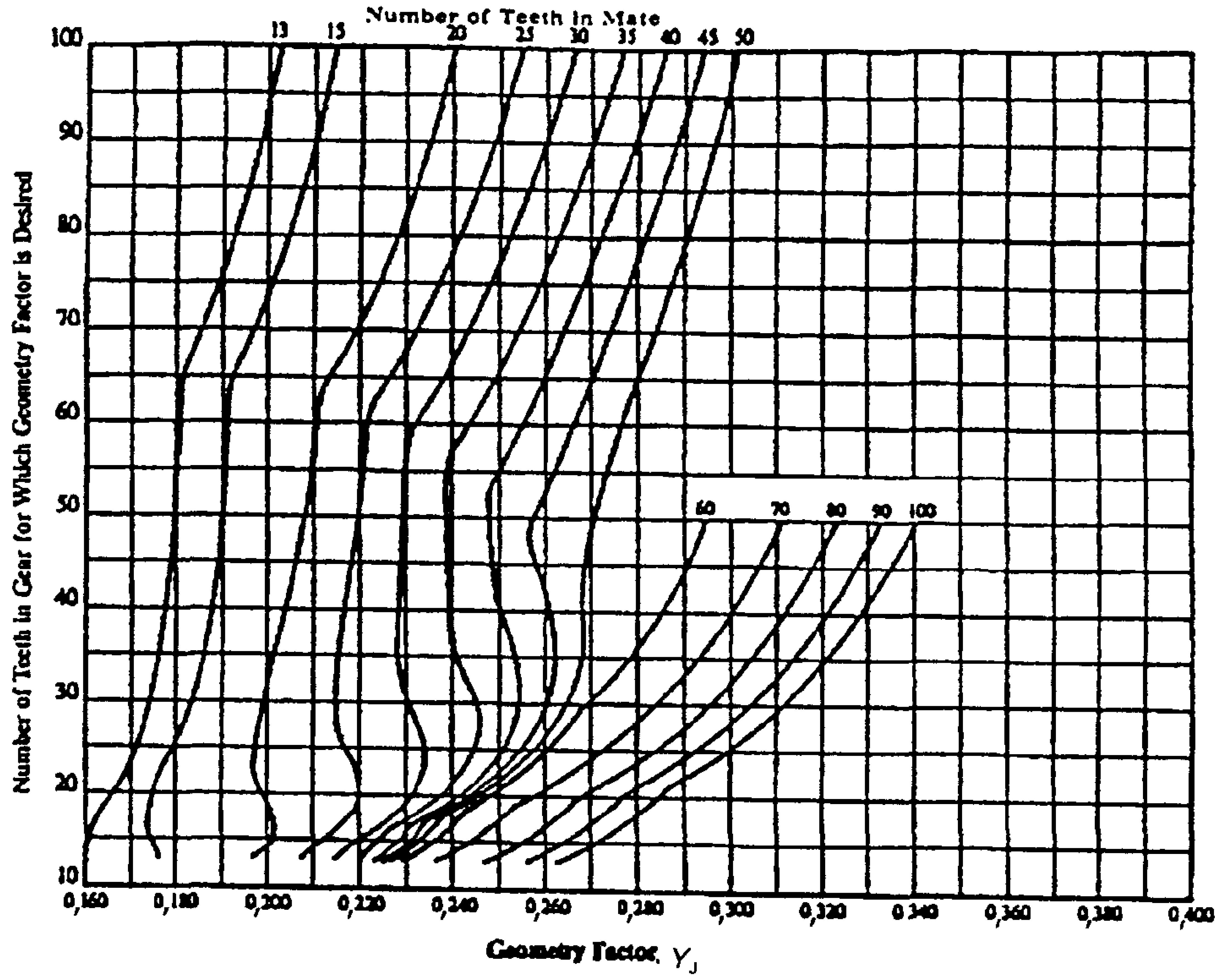


Figure B.1 — Geometry factor, Y_J, for straight-bevel gears with 90° shaft angle, 20° pressure angle, and 0,12 m_{et} tool edge radius

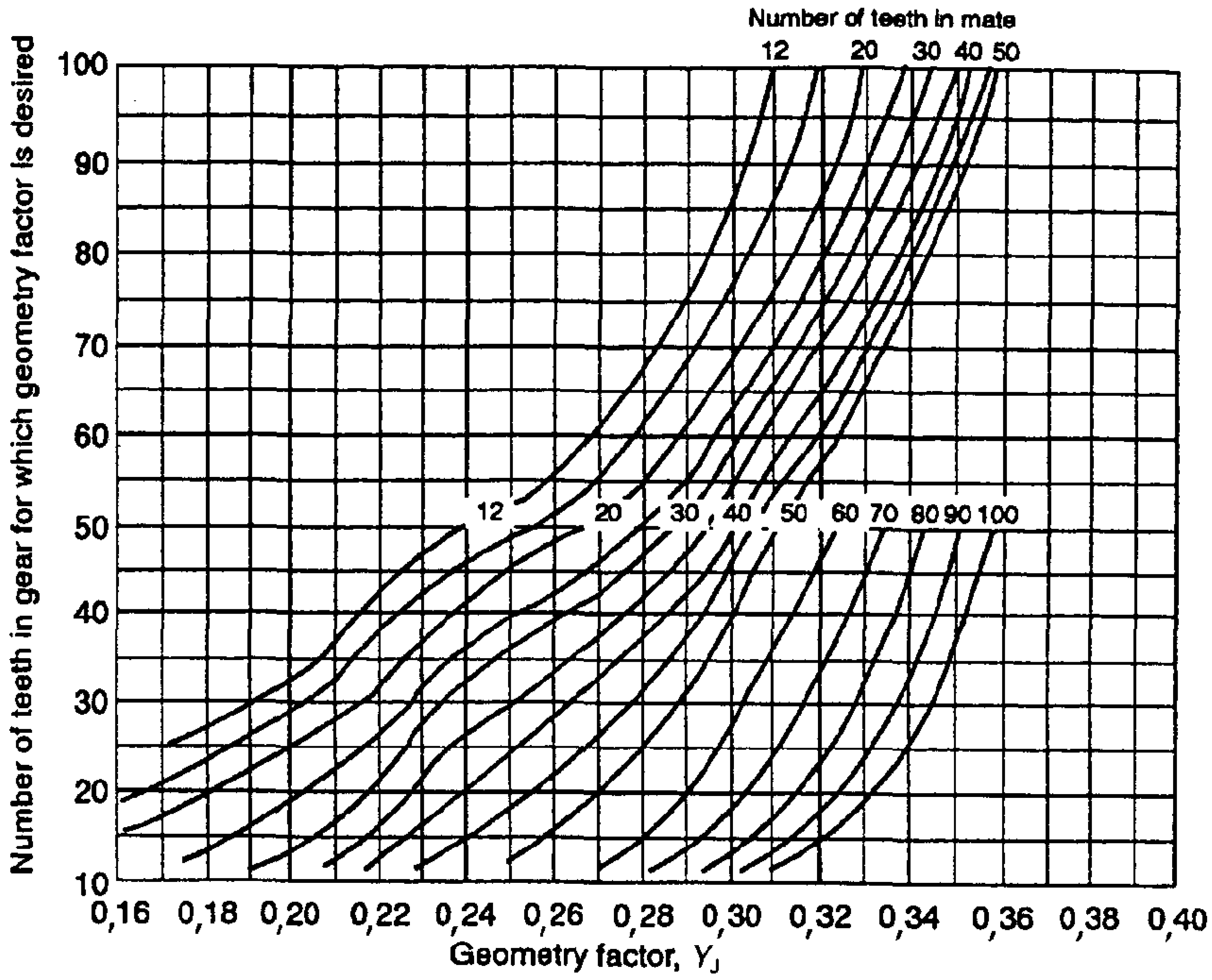


Figure B.2 — Geometry factor, Y_J, for spiral-bevel gears with 90° shaft angle, 20° pressure angle, 35° spiral angle and 0,12 m_{et} tool edge radius

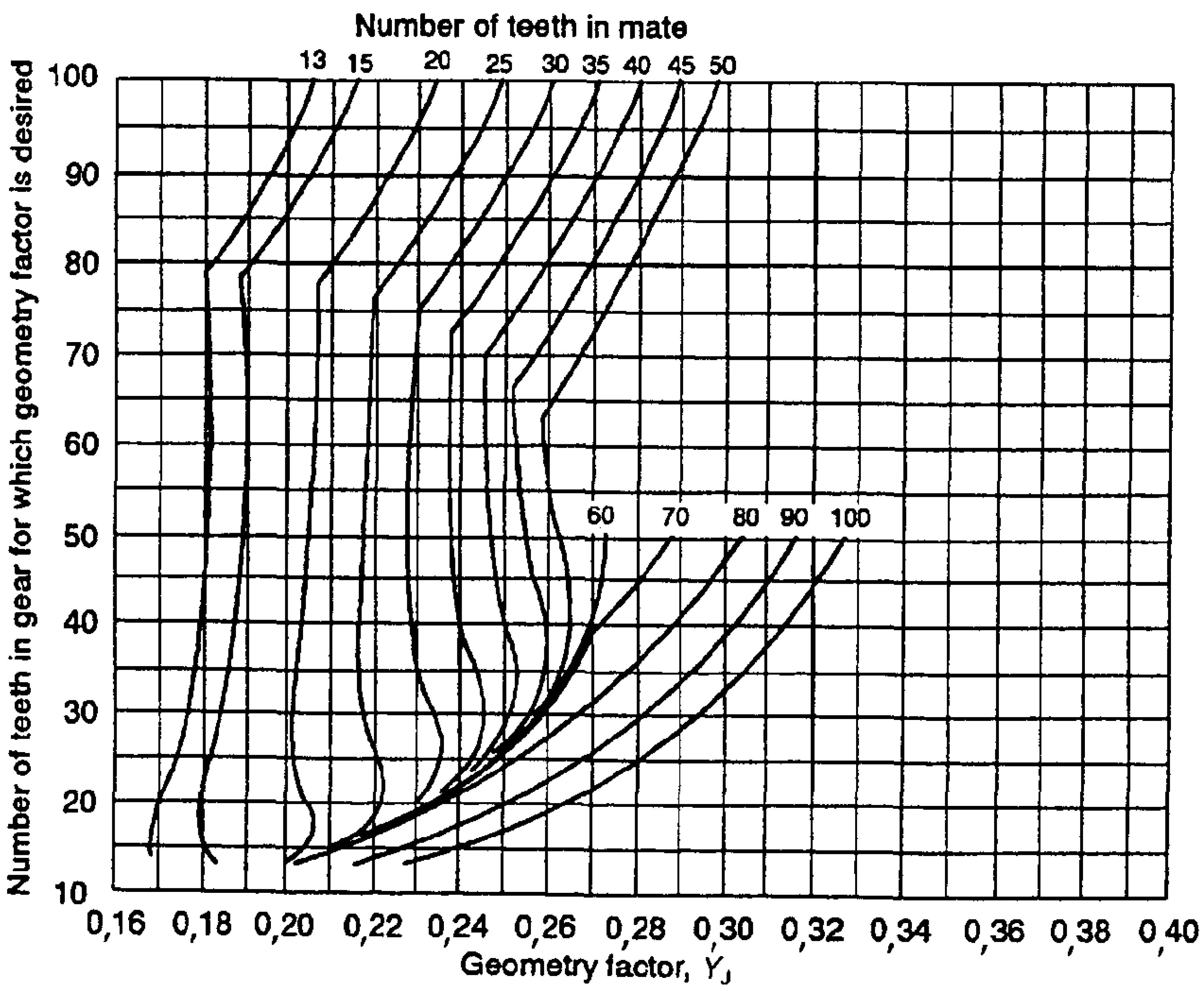


Figure B.3 — Geometry factor, Y_J, for fine-pitch zerol-bevel gears with 90° shaft angle, 20° pressure angle, and 0,12 m_{et} tool edge radius

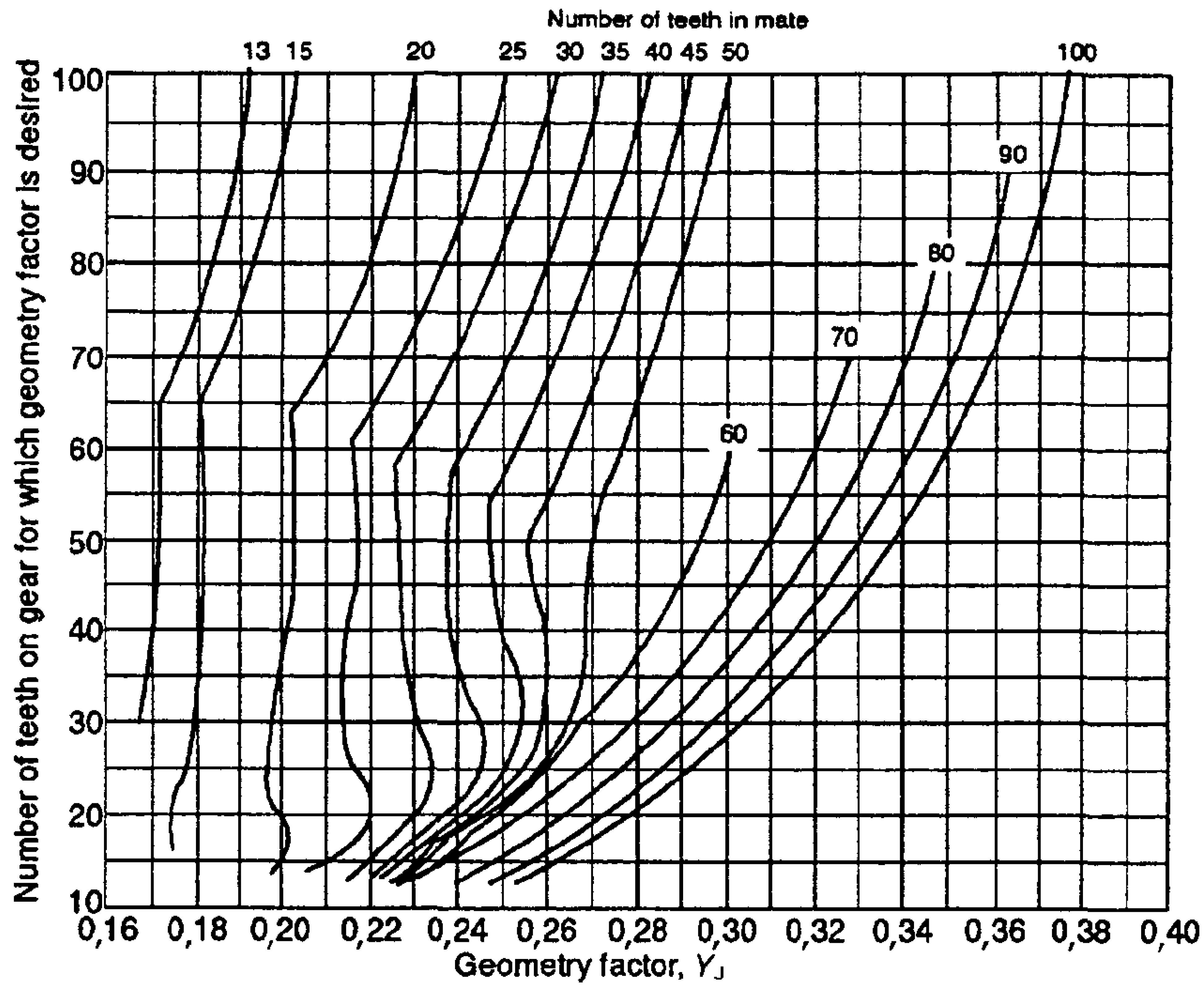


Figure B.4 — Geometry factor, Y_J, for coniflex straight-bevel gears with 90° shaft angle and 20° pressure angle

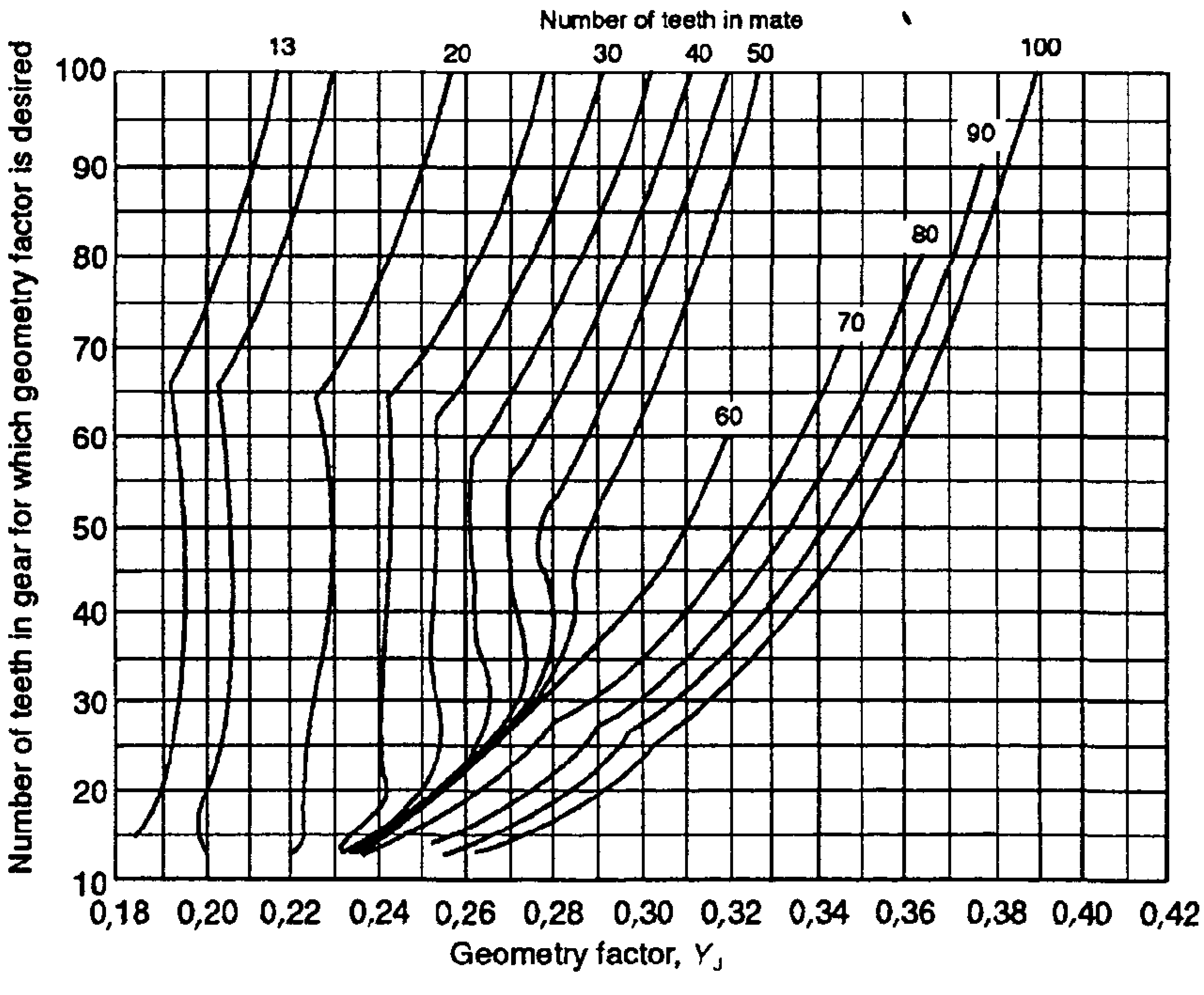


Figure B.5 — Geometry factor, Y_J, for coniflex straight-bevel gears with 90° shaft angle and 25° pressure angle

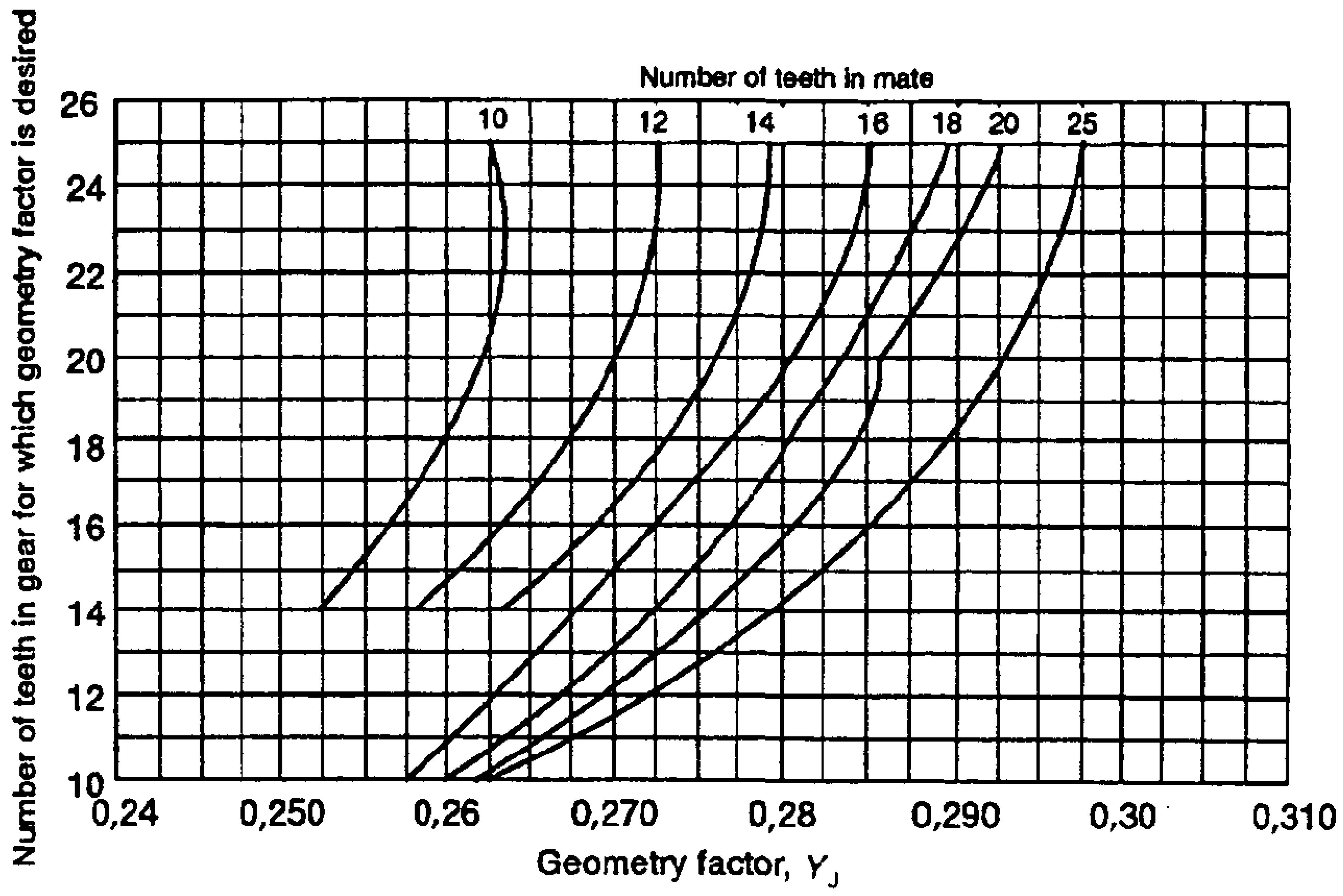


Figure B.6 — Geometry factor, Y_J, for coniflex straight-bevel differential gears with 90° shaft angle and 22,5° pressure angle

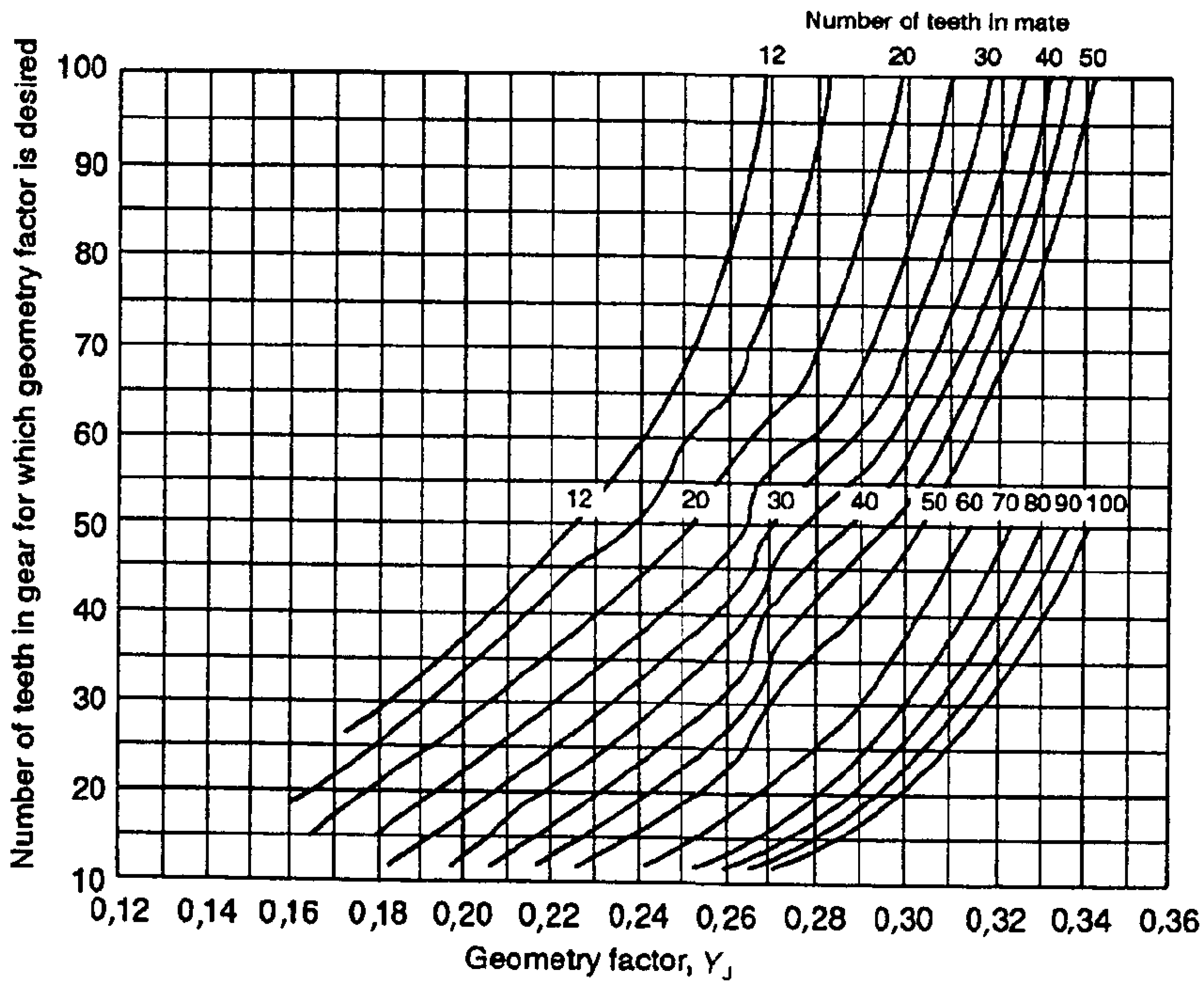


Figure B.7 — Geometry factor, Y_J, for spiral-bevel gears with 90° shaft angle, 20° pressure angle, and 25° spiral angle

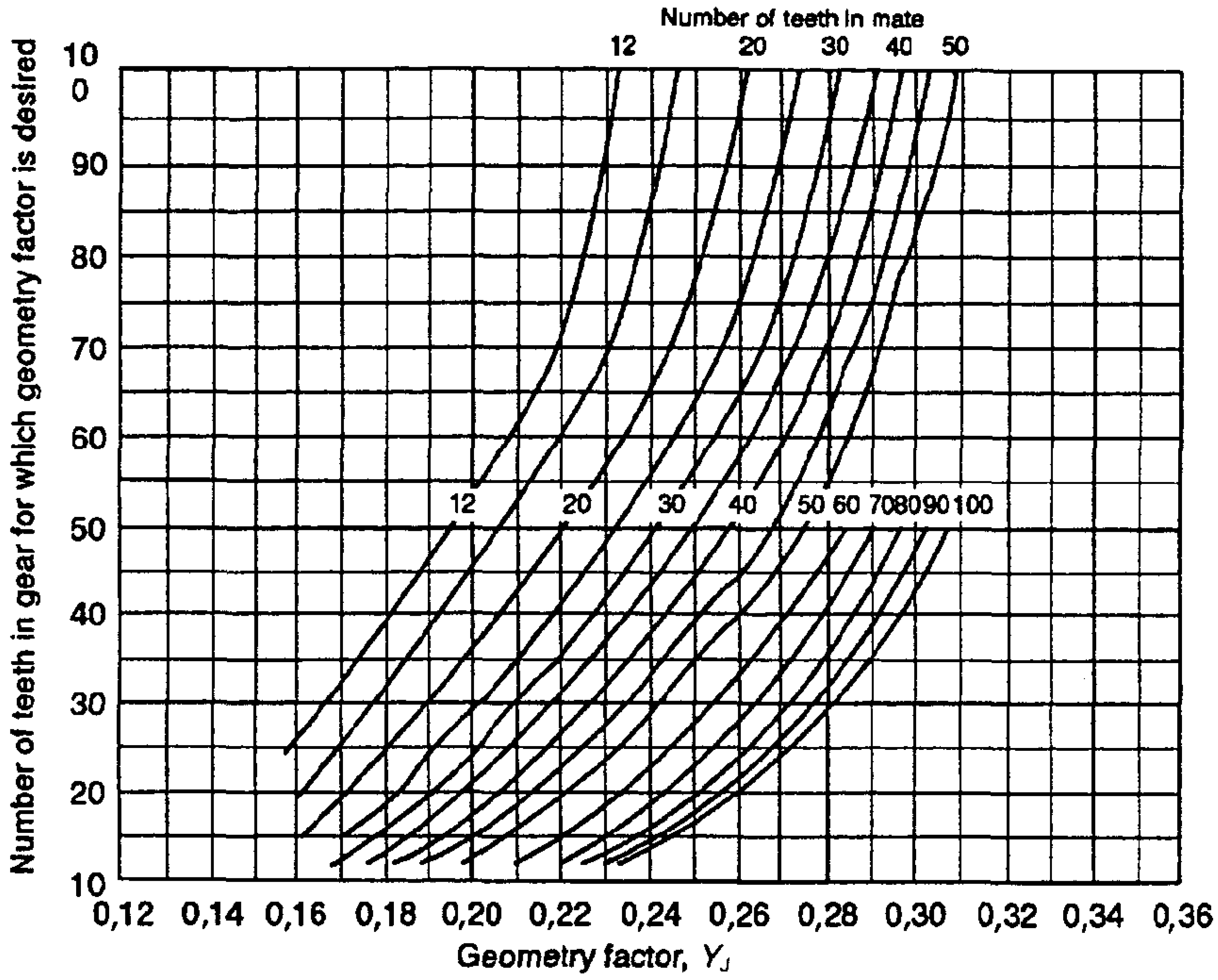


Figure B.8 — Geometry factor, Y_J, for spiral-bevel gears with 90° shaft angle, 20° pressure angle, and 15° spiral angle

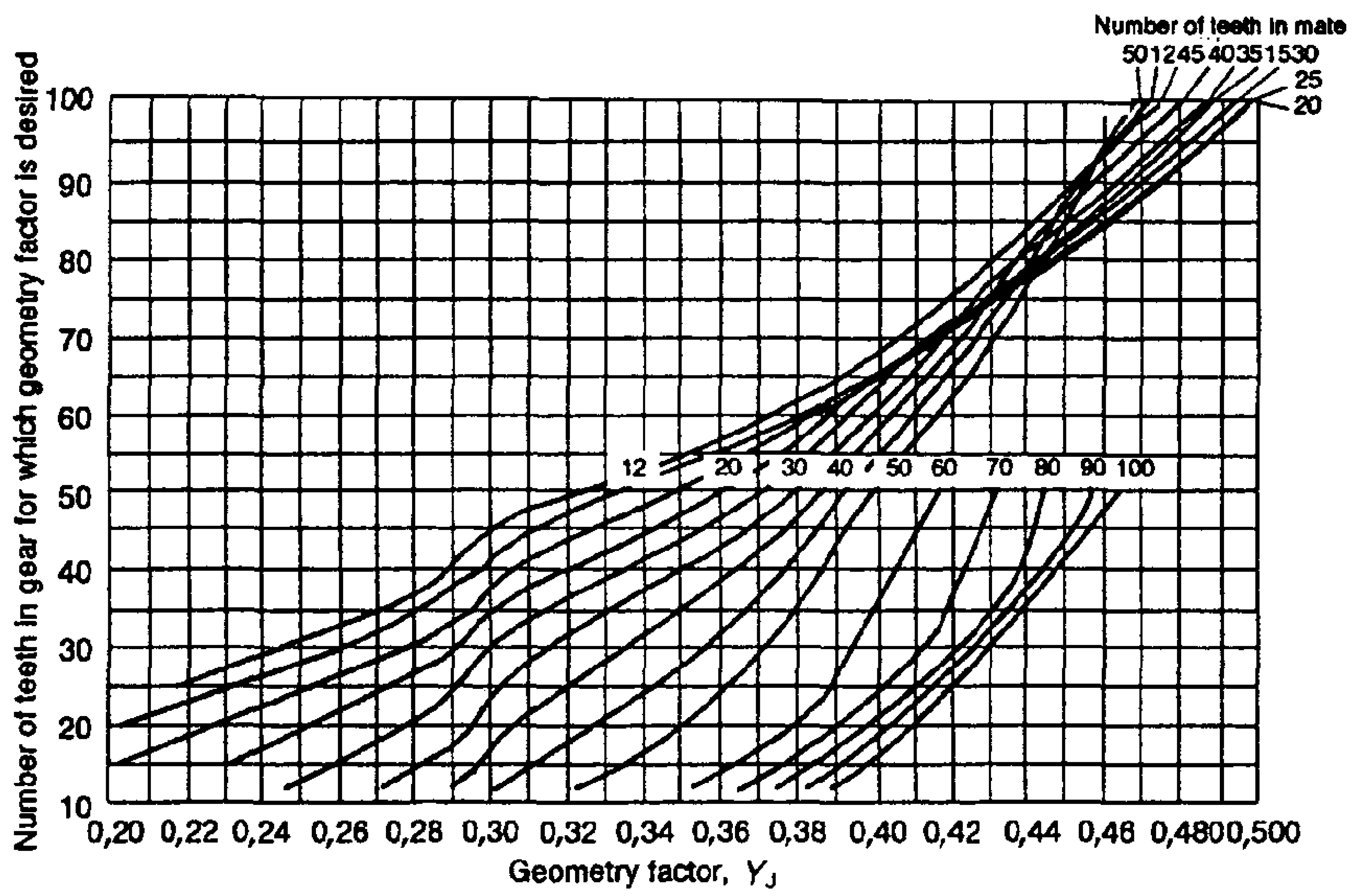


Figure B.9 — Geometry factor, Y_J, for spiral-bevel gears with 90° shaft angle, 25° pressure angle, and 35° spiral angle

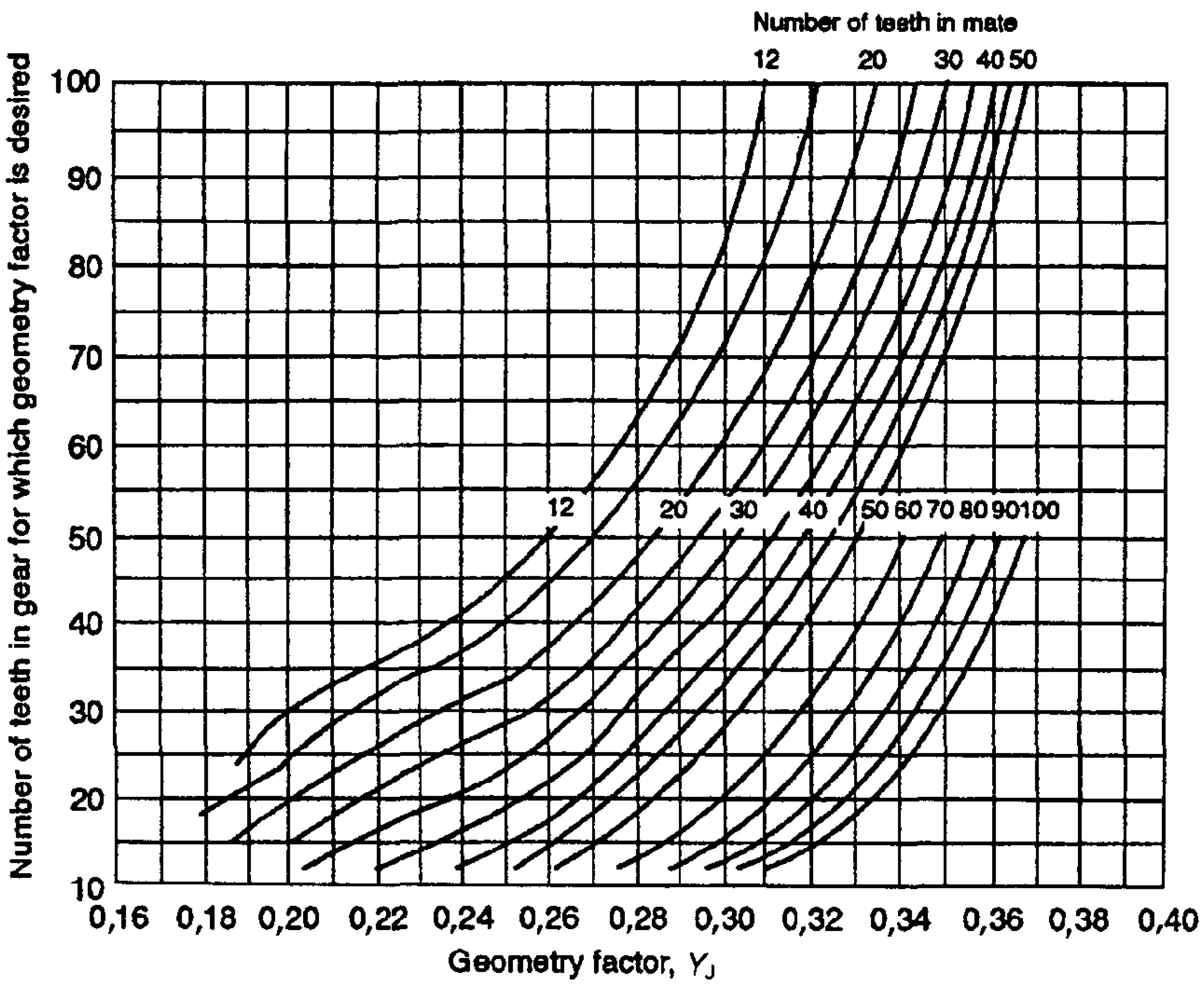


Figure B.10 — Geometry factor, Y_J, for spiral-bevel gears with 60° shaft angle, 20° pressure angle, and 35° spiral angle

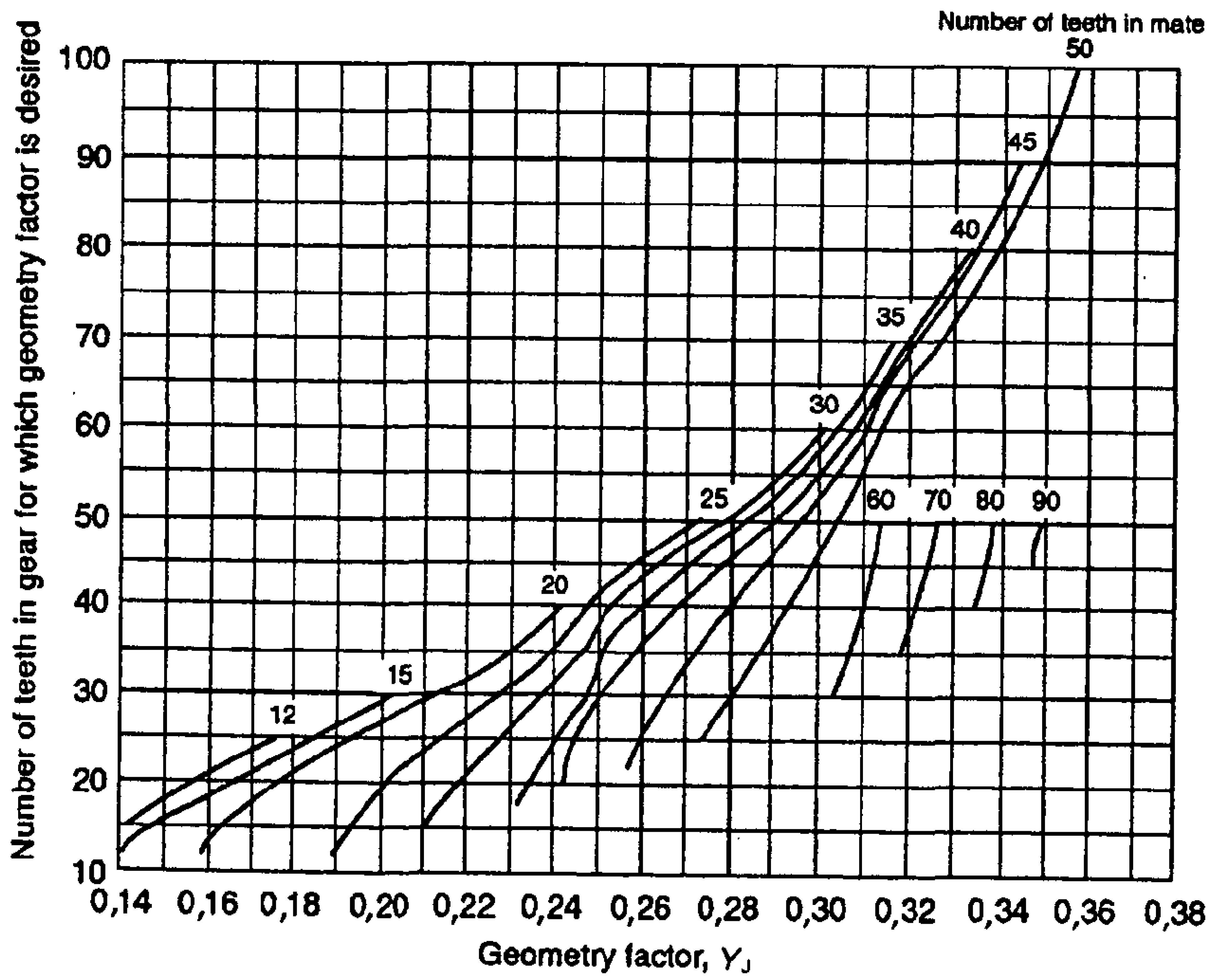


Figure B.11 — Geometry factor, Y_J, for spiral-bevel gears with 120° shaft angle, 20° pressure angle, and 35° spiral angle

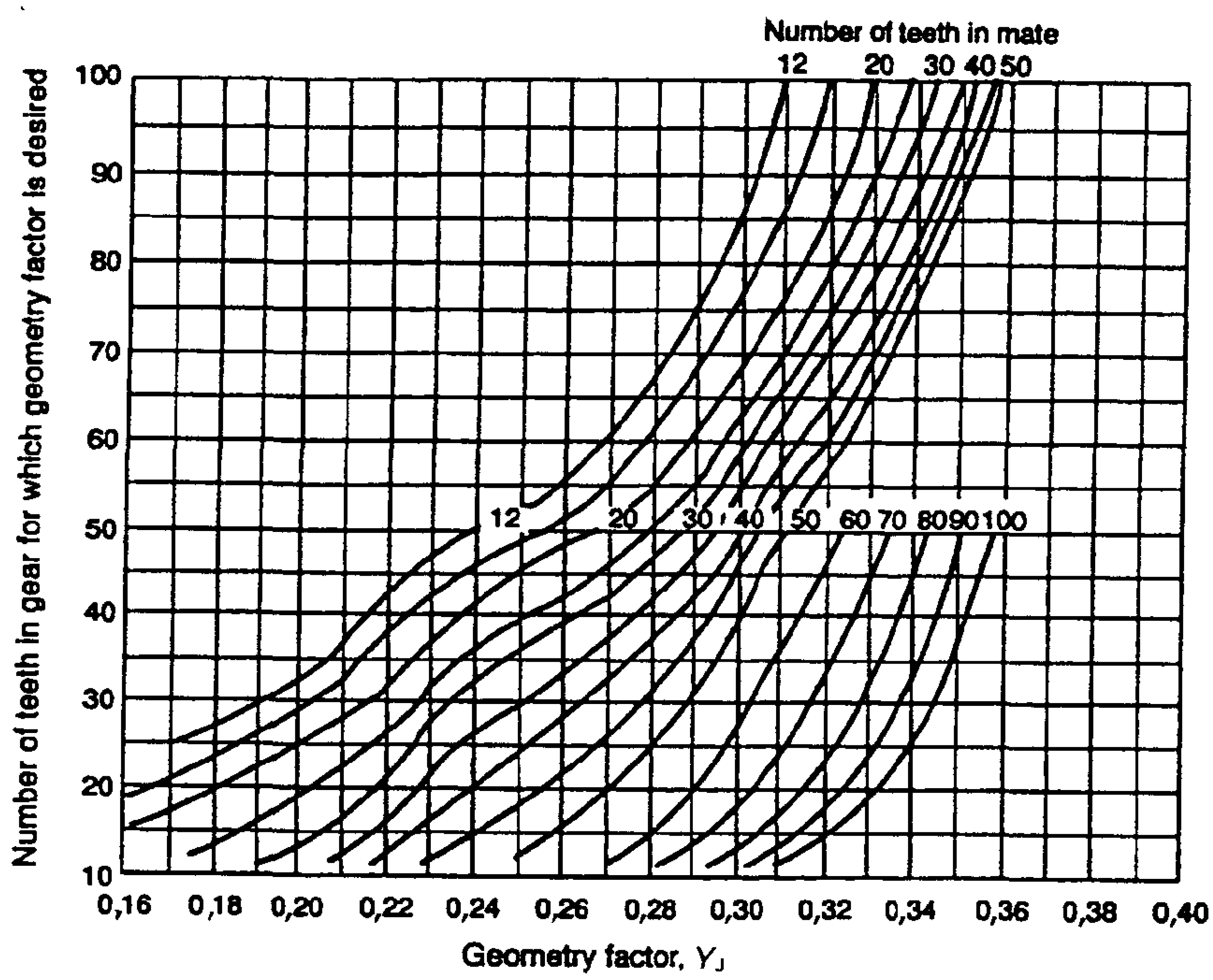


Figure B.12 — Geometry factor, Y_J, for spiral-bevel gears with 90° shaft angle, 20° pressure angle, and 35° spiral angle

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