Autodesk Inventor Engineer s Handbook

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Gears Calculation

Spur Gear Generator

Basic geometric calculation

Input Parameters

Gear type - internal or external gear

Gear ratio and tooth numbers $i = \frac{z_2}{z_1}$

Pressure angle (the angle of tool profile) α

Helix angle β

Module m (for metric calculation)

Diametral Pitch P (for English units)

NoteModule and Diametral Pitch are reciprocal values.

Unit addendum a^*

Unit clearance c^*

Unit dedendum fillet r_f^*

Gear widths b_1, b_2

Unit corrections x_1, x_2

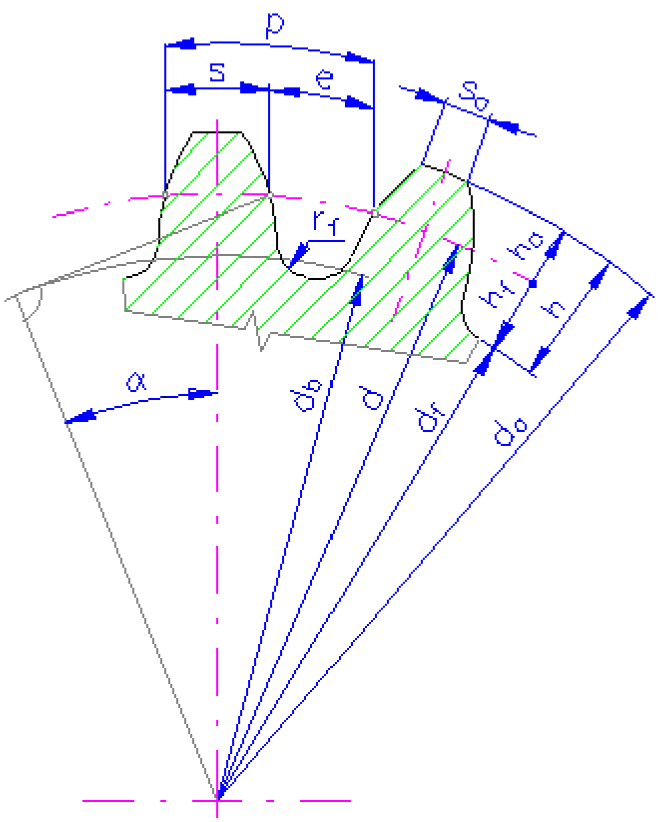
NoteFor internal gears the opposite sign for unit correction is used in following formulas.

Summary of unit correction $\Sigma_x = x_1 + x_2$

Auxiliary Geometric Calculations
Distribution of Unit Corrections for Single Gears
Design of the Module and Tooth Number
Design of Tooth Number
Design According to the Strength Calculation
Calculation of Helix Angle
Calculation of Summary Correction
Design of Gearing Width

Calculation of Maximum Dedendum Filleting

Calculated parameters



Gearing number

for i > 1

u = i

for i < 1

u = 1 / i

Tangential module

$$m_t = \frac{m}{\cos\beta}$$

Normal pitch

p = π m

Axial pitch

$p_t = \pi m_t$

Base pitch

$p_{tB} = p_t \cos \alpha_t$

Helix angle on the basic cylinder

$\sin \beta_b = \sin \beta \cos \alpha$

Axial pressure angle

$$tg\alpha_t = \frac{tg\alpha}{\cos\beta}$$

Rolling/working pressure angle

$$inv\alpha_w = inv\alpha + 2\frac{\Sigma x tg\alpha}{z_1 + z_2}$$

Axial rolling/working pressure angle

$$inv\alpha_{tw} = inv\alpha_t + \frac{2\Sigma x}{z_1 + z_2} tg\alpha$$

Pitch diameter

$d_{1,2} = z_{1,2}\, m_t$

Base circle diameter

$d_{b1,2} = d_{1,2} \cos \alpha_t$

Theoretical center distance

$$a = \frac{d_1 + d_2}{2}$$

Real center distance

$$a_w = a \frac{\cos \alpha_t}{\cos \alpha_{tw}}$$

Feed factor/addendum lowering

$$\Delta y = \Sigma x - \frac{a_w - a}{m}$$

Outside diameter

$d_{a1,2} = d_{1,2} + 2m\,(a^* + x_{1,2} - \Delta_y)$

- for internal gearing the interference check is done as well

$$km = 0.5\left(\sqrt{d_2^2 + d_1^2 \sin^2 \alpha - 2d_1 d_2 \sin^2 \alpha} - d_{a2}\right)$$

for $km > 0$ is accomplished by addendum lowering $d_{a2} = d_{a2} - 2km$

Root diameter

$d_{f1,2} = d_{1,2} - 2m\,(a^* + c^* - x_{1,2})$

Work pitch diameter

$$d_{w1} = \frac{2a_w}{i+1}, \quad d_{w2} = 2a_w - d_{w1}$$

Virtual Number of Teeth

$$z_{v1,2} = \frac{z_{1,2}}{\cos\beta \cos^2 \beta_b}$$

Virtual Pitch Diameter

$d_{n1,2} = z_{v1,2}\, m$

Virtual Base Circle Diameter

$d_{bn1,2} = d_{n1,2} \cos(\alpha)$

Virtual Outside Diameter

$d_{an1,2} = d_{n1,2} + d_{a1,2} – d1,2$

Tooth thickness (measured normally on the pitch diameter)

$s_{1,2} = p/2 + 2m_{x1,2}\, tg\, \alpha$

Tooth width on the chord (measured normally)

$s_{c1,2} = s_{1,2} \cos^2 \alpha$

Addendum height above the chord

$$h_{c1,2} = \frac{d_{a1,2} - d_{1,2} - s_{c1,2} tg\alpha}{2}$$

Unit addendum width (measured normally)

$$s_{a1,2} = \frac{d_{a1,2}}{m}\left(\frac{s_{1,2}}{d_{1,2}} + inv\alpha - inv\alpha_a\right)$$

where:

$$\cos\alpha_a = \frac{d_{1,2}}{d_{a1,2}}\cos\alpha$$

Operating width of gearings

$b_w = \min(b_1, b_2)$

Relative width

$$\frac{b_w}{d_{1,2}}$$

Factor of mesh duration

$\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$

Factor of profile mesh duration

$$\varepsilon_\alpha = \frac{\sqrt{d_{a1}^2 - d_{b1}^2} \pm \sqrt{d_{a2}^2 - d_{b2}^2} - 2a_w \sin\alpha_{tw}}{2p_{tb}}$$

(the minus sign applies to internal gearing)

Factor of step mesh duration

$$\varepsilon_\beta = \frac{b_w \sin\beta}{p}$$

Minimum correction without tapering

$$x_{z1,2} = a_0^* - \frac{1 - \dfrac{\cos\alpha}{\cos\left(\operatorname{inv}\alpha + \dfrac{\pi}{2z_{1,2}}\right)}}{2\cos\beta} z_{1,2}$$

where:

$a_0^* = a^* + c^* - r_f^* (1 - \sin\alpha)$

Minimum correction without undercut

$$x_{p1,2} = a_0^* - \frac{z_{v1,2}}{2} \sin^2\alpha$$

Minimum correction with the allowable undercut

$$x_{d1,2} = \frac{5}{6} a_0^* - \frac{z_{v1,2}}{2} \sin^2 \alpha$$

Checking chordal dimension

$$W_{1,2} = ((z_w - 0.5)\,\pi + z_{1,2} \,\mathrm{inv}\, \alpha_t)\, m \cos \alpha) + 2\, x_{1,2}\, m \sin \alpha$$

where:

z_w is the tooth number across which the measure is performed

Checking size across rollers/balls

- for even tooth number

$$M_{1,2} = D_{s1,2} + d_M$$

- for odd tooth number

$$M_{1,2} = D_{s1,2} \cos(90 / z_{1,2}) + d_M$$

where:

d_M	wire (ball) diameter
$D_{s1,2} = \dfrac{d_{b1,2}}{\cos \alpha_{D1,2}}$	diameter of the wire center circle
$\mathrm{inv}\, \alpha_{D1,2} = \mathrm{inv}\, \alpha_t + \dfrac{d_M}{m\, z_{1,2} \cos \alpha} + \dfrac{s_{1,2}}{m\, z_{1,2}} - \dfrac{\pi}{z_{1,2}}$	wire (ball) contact angle

Design of the module and tooth number

Module for external gear

$$m = \frac{2a_w \cos\beta}{43 + 14i}$$

Module for internal gear

$$m = \frac{2a_w \cos\beta}{27(i - 1)}$$

Tooth numbers

$$z_1 = \frac{2a_w \cos\beta}{m(i \pm 1)}, \quad z_2 = iz_1$$

The minus sign (-) applies to the internal gear.

Design of Tooth number:

$$z_1 = \frac{2a_w \cos\beta}{m(i \pm 1)}, \quad z_2 = iz_1$$

The minus sign (-) applies to internal gearing.

.

Helix Angle Calculation:

$$\cos\beta = \frac{m(z_1 + z_2)}{2a_w}$$

Other parameters are calculated as in the Basic geometric calculation

Axial pressure angle

$$tg\alpha_t = \frac{tg\alpha}{\cos\beta}$$

Axial rolling/working pressure angle

$$\cos\alpha_{tw} = \frac{m(z_1 + z_2)}{2a_w \cos\beta}\cos\alpha_t$$

Unit summary correction

$$\Sigma x = \frac{inv\alpha_{tw} - inv\alpha_t}{2\,tg\alpha}(z_1 + z_2)$$

$$x_1 = 0.02(30 - z_1) + \frac{\Sigma x}{2},\ x_2 = \Sigma x - x_1$$

Unit dedendum filleting

$$r_f^* = \frac{c^*}{1 - \sin\alpha}$$

Backlash of gears

Backlash types (defined for each gear in a gear set)

Real gears must be manufactured with a specific backlash allowance. Designate the allowance by considering the machining and working conditions.

In spur and helical gearing, there are two alternate ways to obtain the appropriate amount of backlash. First, decrease tooth thickness by sinking the hob deeper into the blank than the theoretical standard depth is. Second, increase the center distance than theoretically computed.

When designing a backlash, consider following factors:

- Lubricant space required.
- Differential expansion between the gear components and the housing.
- Errors in machining. Run-out of both gears, errors in profile, pitch, tooth thickness, helix angle and center distance. The smaller the amount of backlash, the more accurate the machining of the gears is.
- Working conditions such as frequent reversing or overrunning load.

The amount of backlash must not be excessive for the requirements of the job. Ensure it is sufficient so that machining costs are not higher than necessary.

It is customary to make half of the allowance for backlash on the tooth thickness of each gear of a pair, although there are exceptions. For example, on pinions having low numbers of teeth, it provides all the allowance on the driven gear. It avoids weakening the pinion teeth.

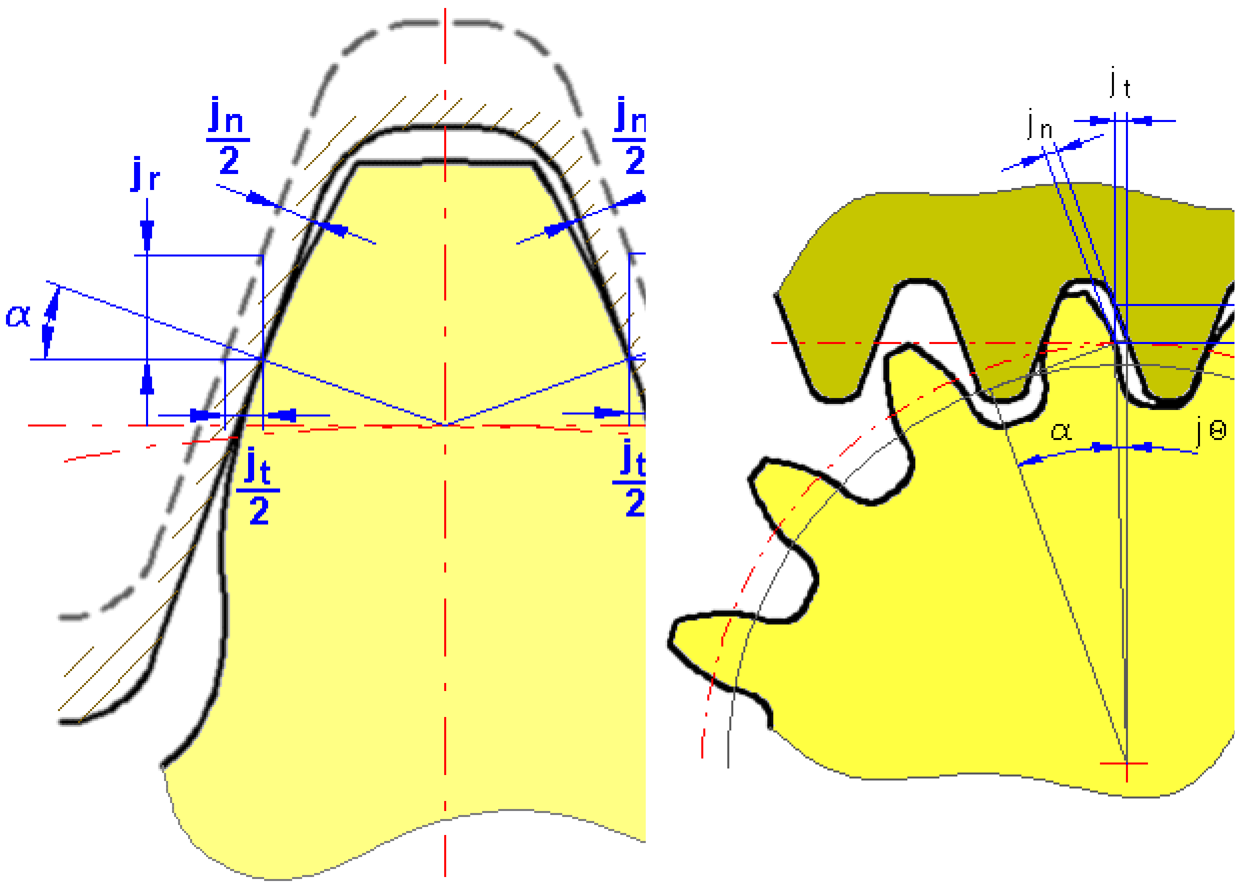
- Circular backlash j_t [mm/in]
- Normal backlash j_n [mm/in]
- Center backlash j_r [mm/in]
- Angular backlash j_Θ [deg]

Type of Gear Meshes	The Relation between Circular Direction j_t and Normal Direction j_n	The Relation between Circular Direction j_t and Center Direction j_r	The Relation between Circular Direction j_t and Angular Backlash j_Θ
Spur Gear	$j_n = j_t \cos \alpha$	$j_r = \dfrac{j_t}{2 \tan \alpha}$	$j_\Theta = j_t \dfrac{360}{\pi d}$

Helical Gear $j_{nn} = j_{tt} \cos \alpha_n \cos \beta$

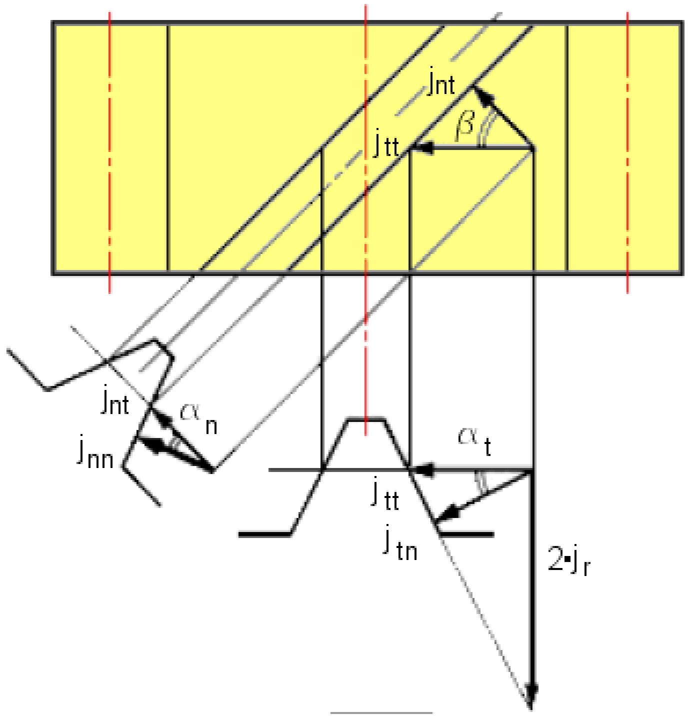
$$j_r = \frac{j_{tt}}{2 \tan \alpha_t}$$

$$j_\Theta = j_{tt} \frac{360}{\pi d}$$



Backlash of Helical Gear Mesh

The helical gear has two kinds of backlash when referring to the tooth space. There is a cross section in the normal direction of the tooth surface “n”, and a cross section in the transverse direction perpendicular to the axis “t”.



j_{nn}	Backlash in the direction normal to the tooth surface
j_{nt}	Backlash in the circular direction in the cross section normal to the tooth
j_{tn}	Backlash in the direction normal to the tooth surface in the cross section perpendicular to the axis
j_{tt}	Backlash in the circular direction perpendicular to the axis
In the plane normal to the tooth:	$j_{nn} = j_{nt} \cos \alpha_n$
On the pitch surface:	$j_{nt} = j_{tt} \cos \beta$

In the plane perpendicular to the axis

$j_{tn} = j_{tt} \cos \alpha_t$

$$j_r = \frac{j_{tt}}{2 \tan \alpha_t}$$

where:

$$\tan \alpha_t = \frac{\tan \alpha_n}{\cos \beta}$$

α Pressure angle

α_n Pressure angle in normal direction, $\alpha = \alpha_n$

α_t Pressure angle in transverse direction

β Helix angle

d Pitch diameter

Calculation of strength proportions

Input values:

Input power P_1

Input speed n_1

Gearing ratio i

Gearing efficiency η

Calculated values

Output: $P_2 = P_1 \eta$

Output speed: $n_2 = \frac{n_1}{i}$

Metric units

Input moment: $M_{k1} = \frac{30000 P_1}{\pi n_1} [Nm]$

Tangential/circumferential force: $F_t = \frac{2000 M_{k1}}{d_{w1}} [N]$

Circumferential speed: $v = \frac{\pi d_1 n_1}{60000} [m/s]$

Resonance speed: $n_{E1} = \frac{1.91 \cdot 10^7}{z_1 \cdot d_2} \cdot \sqrt{\left(1 + u^2\right) \cdot \left(0.75\, \varepsilon_\alpha + 0.25\right)}\ [1/min]$

ANSI (English) units

Input moment: $M_{k1} = 30 \frac{550 P_1}{\pi n_1} [lbft]$

Tangential/circumferential force: $F_t = \frac{24 M_{k1}}{d_1} [lb]$

Circumferential speed: $v = \frac{\pi d_1 n_1}{720} [ft/s]$

Resonance speed: $n_{E1} = \frac{7.52 \cdot 10^5}{z_1 \cdot d_2} \cdot \sqrt{\left(1 + u^2\right) \cdot \left(0.75\, \varepsilon_\alpha + 0.25\right)}\ [1/min]$

Output moment

$M_{k2} = M_{k1}\, i\, \eta$

Radial force

$F_r = F_t \operatorname{tg} \alpha_{tw}$

Axial force

$F_a = F_t \operatorname{tg} \beta$

Normal force

$$F_n = \frac{F_t}{\cos\alpha_w \cos\beta}$$

Strength calculation according to Bach

Based on the fixed-end beam calculation, and anticipates that the total circumferential force can only be carried by one tooth.

Allowable load

$F_{all} = \pi\, c\, b\, m \geq F_t$

where:

$c = 0.065\ \sigma_{Ab}$	tooth allowable stress in bending [MPa, psi]
σ_{Ab}	allowable stress in bending (material property)
b	tooth gearing width
m	module
F_t	circumferential force acting on the gearing

Safety factor

$S = F_{all} / F_t$

Strength calculation with Merrit method

Based on the fixed-end beam calculation. Anticipates that the total circumferential force can only be carried by one tooth.

Allowable load

$F_{all} = \pi\, c_{min}\, b_w\, m\, \mu \geq F_t$

where:

$c_{min} = \min(c_b, c_c)$	minimum tooth allowable stress
b_w	operating tooth bearing width
m	module
μ	factor of dependence on the precision degree (table value)
F_t	circumferential force acting on the teeth

Safety factor

$S = F_{all} / F_t$

Bending factor

$$c_b = \frac{\sigma_{Ab} \cdot r_b}{y_b}$$

where:

σ_{Ab} allowable stress in bending (material property)

r_b speed bending factor (table value)

y_b shape bending factor (table value)

Contact factor

$$c_c = \frac{\sigma_{Ac} \cdot r_c}{U \cdot y_c}$$

where:

σ_{Ac}	allowable stress in contact (material property)
r_c	speed pressure factor (table value)
y_c	shape pressure factor (table value)

$U = \left(\frac{m}{10}\right)^{0.2}$ size factor

Strength calculation with CSN 01 4686, ISO 6336 and DIN 3990

Based on the fixed-end beam calculation. Contains the majority of effects.

Safety factors

Contact fatigue

$$S_{H1,2} = \frac{\sigma_{Hlim\,1,2} \cdot Z_{N1,2} \cdot Z_L \cdot Z_R \cdot Z_v \cdot Z_{X1,2} \cdot Z_W}{Z_E \cdot Z_H \cdot Z_{B1,2} \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t \cdot K_H}{b_w \cdot d_1} \cdot \frac{u+1}{u}}}$$

where:

σ_{Hlim} contact fatigue limit (material property)

F_t tangential force acting at teeth

b_w operating face width

d_1 pitch diameter of pinion

Contact during one-time loading

$$S_{HSt1,2} = \frac{\sigma_{HPmax\,1,2}}{Z_E \cdot Z_H \cdot Z_{B1,2} \cdot Z_\varepsilon \cdot \sqrt{\frac{F_t \cdot K_H \cdot K_{AS}}{b_w \cdot d_1} \cdot \frac{u+1}{u}}}$$

where:

σ_{HPmax} permissible contact stress

K_{AS} one-time overloading factor

Bending fatigue

$$S_{F1,2} = \frac{\sigma_{Flim1,2} \cdot Y_{A1,2} \cdot Y_{T1,2} \cdot Y_{N1,2} \cdot Y_{\delta 1,2} \cdot Y_{X1,2} \cdot Y_R}{Y_{Fa1,2} \cdot Y_{Sa1,2} \cdot Y_{Sag1,2} \cdot Y_\beta \cdot Y_\varepsilon \cdot \frac{F_t \cdot K_F}{b_{wF1,2} \cdot m}}$$

where:

σ_{Flim} bending fatigue limit (material property)

$b_{wF1,2} = \min(b_{1,2}, b_w + 2m)$ tooth width for bending

Bending during one-time loading

$$S_{FSt1,2} = \frac{\sigma_{FPmax1,2} \cdot Y_{N1,2} \cdot Y_{X1,2}}{Y_{Fa1,2} \cdot Y_{Sa1,2} \cdot Y_{Sarel1,2} \cdot Y_\beta \cdot Y_\varepsilon \cdot \frac{F_t \cdot K_F \cdot K_{AS}}{b_{wF1,2} \cdot m}}$$

where:

σ_{FPmax} permissible bending stress

Factor calculations

Z_N ... life factor (for contact)

$$Z_{N1,2} = \sqrt[q_H]{\frac{N_{Hlim1,2}}{N_{K1,2}}}$$

$1 \leq Z_N \leq 1.3$ nitridated steels

$1 \leq Z_N \leq 1.6$ other steels

N_{Hlim} base number of load cycles for contact (material property)

$N_{K1,2} = 60\, L_h\, n_{1,2}$ required number of load cycles (speed)

Y_N ... life factor (for bending)

$$Y_{N1,2} = \sqrt[q_F]{\frac{N_{Flim1,2}}{N_{K1,2}}}$$

$1 \leq Y_N \leq 1.6$ nitridated steels

$1 \leq Y_N \leq 2.5$ other steels

N_{Flim} base number of load cycles for bending (material property)

$N_{K1,2} = 60\ L_h\ n_{1,2}$ required number of load cycles (speed)

Z_L ... lubricant factor

DIN and ISO:

$Z_L = C_{ZL} + 4\ (1 - C_{ZL})\ 0.158$

$C_{ZL} = \sigma_{Hlim} / 4375 + 0.6357$

for $\sigma_{Hlim} < 850$ Mpa $C_{ZL} = 0.83$

for $\sigma_{Hlim} > 1200$ Mpa $C_{ZL} = 0.91$

Z_R ... roughness factor

Z_V ... speed factor

CSN:

$Z_v = 0.95 + 0.08 \log v$

ISO and DIN:

$C_{ZV} = C_{ZL} + 0.02$

Z_E ... elasticity factor

$$Z_E = \sqrt{\frac{1}{\pi \cdot \left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2} \right)}}$$

where:

μ Poisson's ratio (material property)

E modulus of elasticity (material property)

Z_H ... zone factor

$$Z_H = \frac{1}{\cos \alpha_t} \sqrt{\frac{2 \cdot \cos \beta_b}{tg \alpha_{tw}}}$$

Z_B ... single pair tooth contact factor

for $\varepsilon_\beta \geq 1$ or internal gearing:

$Z_{B1,2} = 1$

for $\varepsilon_\beta = 0$:

$$Z_{B1,2} = \frac{tg\alpha_{tw}}{\sqrt{\left[\sqrt{\left(\frac{d_{a1,2}}{d_{b1,2}}\right)^2 - 1} - \frac{2\pi}{z_{1,2}}\right] \cdot \left[\sqrt{\left(\frac{d_{a2,1}}{d_{b2,1}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_{2,1}}\right]}}$$

for $\varepsilon_\beta < 1$:

$Z_{B1,2} = Z_{B0} - \varepsilon_b (Z_{B0} - 1)$

where:

$Z_{B0} = Z_{B1,2}$ calculated for $\varepsilon_\beta = 0$

Z_ε ... contact ratio factor

for $\varepsilon_\beta = 0$:

$$Z_\varepsilon = \sqrt{\frac{4 - \varepsilon_\alpha}{3}}$$

for $\varepsilon_\beta < 1$:

$$Z_\varepsilon = \sqrt{\frac{(4 - \varepsilon_\alpha) \cdot (1 - \varepsilon_\beta)}{3} + \frac{\varepsilon_\beta}{\varepsilon_\alpha}}$$

for $\varepsilon_\beta \geq 1$:

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}}$$

Y_ε ... contact ration factor (for bending)

CSN:

for $\varepsilon_\beta < 1$:

$$Y_\varepsilon = 0.2 + \frac{0.8}{\varepsilon_\alpha}$$

for $\varepsilon_\beta \geq 1$:

$$Y_\varepsilon = \frac{1}{\varepsilon_\alpha}$$

ISO and DIN:

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha}$$

Z_β ... helix angle factor (for contact)

CSN:

$Z_\beta = 1$

ISO and DIN:

$$Z_\beta = \sqrt{\cos\beta}$$

Y_β ... helix angle factor (for bending)

$$Y_\beta = 1 - \varepsilon_\beta \frac{\beta}{120°}$$

CSN:

$Y_{\beta min} = 1 - 0.25\,\varepsilon_\beta \geq 0.75$

ISO and DIN:

for $\varepsilon_\beta > 1$ the $\varepsilon_\beta = 1$ is used

for $\beta > 30°$ the $\beta = 30°$ is used

Z_x ... size factor (for contact)

Y_x ... size factor (for bending)

Z_W ... work hardening factor

Y_{Fa} ... form factor

$$Y_{Fa} = 6 \cdot \frac{\frac{h_{Fa}}{m_n} \cdot \cos\alpha_{Fan}}{\left(\frac{s_{Fn}}{m_n}\right)^2 \cdot \cos\alpha}$$

where:

h_{Fa} bending arm of a force acting on the tooth end

s_{Fn} thickness of dedendum dangerous section of alternate gear

α_{Fan} bending angle at the end of straight tooth of alternate gear

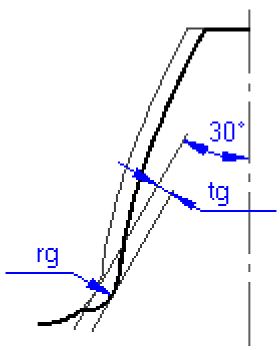
Y_{Sa} ... stress correction factor

$Y_{Sa} = (1.2 + 0.13\ L_a)\ q_s^{\ exp}$

$$exp = \frac{1}{1.21 + \frac{2.3}{L_a}}$$

Y_{Sag} ... teeth with grinding notches factor

$$Y_{Sag} = \frac{1{,}3}{1{,}3 - 0{,}6\sqrt{\frac{t_g}{r_g}}}$$



Y_{δ} ... notch sensitivity factor (depends on the material and curvature radius of dedendum transition)

Y_R ... tooth root surface factor

K_H ... additional loads factor (for contact)

$K_H = K_A K_{Hv} K_{Hb} K_{Ha}$

K_F ... additional loads factor (for bending)

$K_F = K_A K_{Fv} K_{Fb} K_{Fa}$

K_A ... application factor (external dynamic forces)

K_{Hv} ... dynamic factor (internal dynamic forces) for contact

K_{Fv} ... dynamic factor (internal dynamic forces) for bending

$$K_{Fv} = K_{Hv} = 1 + \left(\frac{K_P}{K_A \cdot F_t / b_w} + K_Q \right) \cdot \frac{z_1 \cdot v}{100} \cdot \sqrt{\frac{u^2}{1+u^2}}$$

for CSN:

at $K_A F_t / b_w < 150$ considering $K_A F_t / b_w = 150$

for ISO and DIN:

at $K_A F_t / b_w < 100$ considering $K_A F_t / b_w = 100$

where: K_P , K_Q ... table values

$K_{H\beta}$ ... face load factor (for contact)

for CSN:

$$K_{H\beta} = 1 + \frac{C \cdot f_{ky}}{K_A \cdot K_{Hv} \cdot f_{z0}}$$

where:

c = 0.4	gears with hardened tooth sides
c = 0.3	non-hardened gears

$$f_{z0} = \frac{F_t \cdot Z_\varepsilon^2}{b_w \cdot c' \cdot \cos \alpha_t}$$

$f_{ky} = | f_{sh1} + f_{sh2} | + f_{kZ} - y_\beta$

$$f_{kZ} = \sqrt{0.98 f_\beta^2 + f_y^2 + (f_x \cdot tg\alpha_t)^2} \cdot \cos\alpha_t \cdot \cos\beta_b$$

f_b, f_x, f_y ... teeth tolerance

y_β ... table value

for ISO and DIN:

for $\frac{b_w \cdot F_{\beta y} \cdot c_\gamma}{2 \cdot K_A \cdot K_{Hv} \cdot F_t} \geq 1$:

$$K_{H\beta} = 1 + \frac{b_w \cdot F_{\beta y} \cdot c_\gamma}{2 \cdot K_A \cdot K_{Hv} \cdot F_t}$$

otherwise (< 1):

$$K_{H\beta} = \sqrt{\frac{2 \cdot b_w \cdot F_{\beta y} \cdot c_\gamma}{K_A \cdot K_{Hv} \cdot F_t}}$$

$F_{\beta y} = F_{\beta x} \chi_\beta$

for gears with hardened tooth sides $\chi_\beta = 0.85$

for others

$$\chi_\beta = 1 - \frac{320}{\sigma_{Hlim}} \geq 0$$

$F_{\beta x} = 1.33 f_{sh} + f_{ma}$

$q' = 0.04723 + 0.15551/z_{v1} + 0.25791/z_{v2} - 0.00635\, x_1 - 0.11654\, x_1/z_{v1} - 0.00193\, x_2 - 0.24188\, x_2/z_{v2} + 0.00529\, x_1^2 + 0.00182\, x_2^2$

$$c' = \frac{C_M\, C_R\, C_B \cos\beta}{q'} \frac{\frac{2E_1 E_2}{E_1 + E_2}}{E_{steel}}$$

for $F_t K_A / b_w < 100$ the values are interpolated

for ISO $c' = c' [(F_t K_A / b_w) / 100]^{0.25}$

for DIN $c' = c' (F_t K_A / b_w) / 100$

$C_M = 0.8$

$C_R = 1$ for solid gears

$C_B = [1 + 0.5 (1.2 - h_f/m)] [1 - 0.02 (20° - \alpha)]$

$E_{steel} = 206\ 000$

$c_\gamma = c' (0.75\ \varepsilon_\alpha + 0.25)$

$$f_{sh1,2} = A_{1,2} \cdot \left(\frac{b_w}{d_{1,2}}\right)^2 \cdot \left[\left|B_{1,2} + 0.7\right| + 0.3\right] \cdot \frac{F_t}{b_w} \cdot K_A \cdot K_{Hv}$$

A, B ... table values depend on the arrangement of teeth gears, shafts, and bearings

$K_{F\beta}$ ... face load factor (for bending)

$K_{F\beta} = (K_{H\beta})^{NF}$

where:

$$NF = \frac{(b_w/h)^2}{(b_w/h)^2 + (b_w/h) + 1}$$

$h = 2\ m/\varepsilon_\alpha$ spur gears

$h = 2\ m$ helical gears

K_{Fa} ... transverse load factor (for bending)

for $\varepsilon_\gamma < 2$:

$$K_{F\alpha} = \frac{\varepsilon_\gamma}{2} \cdot \left[0.9 + 0.4 \cdot \frac{c_\gamma \cdot b_w \cdot \left(\left|f_{pb}\right| - \left|y_\alpha\right|\right)}{F_t \cdot K_A \cdot K_{Hv} \cdot K_{H\beta}}\right]$$

for $\varepsilon_\gamma > 2$:

at $K_A F_t / b_w < 100$ considering $K_A F_t / b_w = 100$

$$K_{F\alpha} = 0.9 + 0.4 \cdot \sqrt{\frac{2 \cdot (\varepsilon_\gamma - 1)}{\varepsilon_\gamma}} \cdot \frac{c_\gamma \cdot b_w \cdot \left(\left|f_{pb}\right| - \left|y_\alpha\right|\right)}{F_t \cdot K_A \cdot K_{Hv} \cdot K_{H\beta}}$$

limit values:

for CSN: $1 \leq K_{F\alpha} \leq \varepsilon_\gamma$

$$1 \le K_{F\alpha} \le \frac{\varepsilon_\gamma}{\varepsilon_\alpha \cdot Y_\varepsilon}$$

$K_{H\alpha}$ ... transverse load factor (for contact)

for CSN:

$K_{H\alpha} = 1$ for straight teeth

$K_{H\alpha} = K_{F\alpha}$ for helical teeth

DIN and ISO:

$K_{H\alpha} = K_{F\alpha}$

for limit values:

$$1 \le K_{H\alpha} \le \frac{\varepsilon_\gamma}{\varepsilon_\alpha \cdot Z_\varepsilon^2}$$

Strength Calculation according to ANSI/AGMA 2001-D04:2005

Based on the fixed-end beam calculation. Includes the majority of effects.

Safety factor of contact fatigue

$$S_{H1,2} = \frac{s_{ac} \cdot Z_{N1,2} \cdot C_{H1,2}}{C_p \cdot K_T \cdot K_R \cdot \sqrt{F_t \cdot K_o \cdot K_v \cdot K_{s1,2} \cdot \frac{K_{m1,2}}{d_{w1} \cdot b_w} \frac{C_{f1,2}}{I}}}$$

where:

s_{ac} allowable contact stress number (material property)
F_t tangential force acting at teeth
d_{w1}operating pitch diameter of pinion
b_w operating tooth width

Safety factor of bending fatigue

$$S_{F1,2} = \frac{s_{at1,2} \cdot Y_{N1,2} \cdot Y_{A1,2}}{K_T \cdot K_R \cdot F_t \cdot K_o \cdot K_v \cdot K_{s1,2} \cdot \frac{P_t}{b_{wF1,2}} \frac{K_{m1,2} \cdot K_{B1,2}}{J_{1,2}}}$$

where:

s_{at} allowable bending stress
P_t tangential diametral pitch
$b_{wF1,2} = \min(b_{1,2}, b_w + 2m)$operating tooth width

Factor Calculations

$$C_p = \sqrt{\frac{1}{\pi \cdot \left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}\right)}}$$

where:

μ Poisson's ratio (material property)

E modulus of elasticity (material property)

I geometry factor for pitting resistance

Z_N stress cycle factor for pitting resistance
C_H hardness ratio factor
K_o overload factor
K_v dynamic factor
K_s size factor
K_m load distribution factor

$K_m = 1 + C_{mc} (C_{pf} C_{pm} + C_{ma} C_e)$

C_{mc} - Lead Correction Factor

C_{pf} - Pinion proportion factor

C_{ma} - Mesh Alignment Factor

C_e - Mesh Alignment Correction Factor

J geometry factor for bending strength
Y_N stress cycle factor for bending strength
Y_a reverse loadingfactor
C_f surface condition factor
K_R reliability factor
K_T temperature factor
K_B rim thickness factor

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