Autodesk Inventor Engineer s Handbook

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Gears Calculation

Bevel Gears Generator

Basic geometric calculation

Input Parameters

Gear type - according to the position of root and head cone

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

Pressure angle (the angle of tool profile) α_t

Helix angle β_m

Axis angle Σ

Tangential module on outer cone met (for metric calculation)

Tangential Diametral Pitch on outer cone Pet (for English units)

NoteModule and Diametral Pitch are reciprocal values.

Unit addendum height ha^*

Unit clearance c^*

Unit dedendum fillet r_f^*

Facewidths b_1, b_2

Unit correction $x = x_1 = - x_2$

Unit change of tooth thickness $x_t = x_{t1} = - x_{t2}$

Auxiliary Geometric Calculations

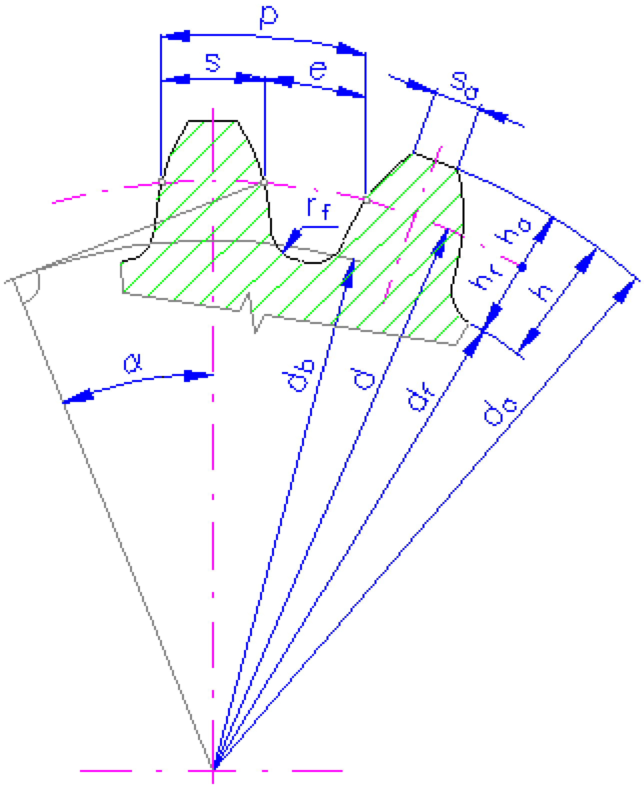
Distribution of Unit Corrections for Single Gears

Design According to the Strength Calculation

Design of Face Width

Calculation of Maximum Dedendum Filleting

Calculated parameters



Normal Pressure Angle in Middle Plane

$$\text{tg}\ \alpha_{nm} = \text{tg}\ \alpha_t \cos \beta_m$$

Pitch angle

$$\delta_1 = \text{arctg}\left(\frac{\sin\Sigma}{u + \cos\Sigma}\right),\ \delta_2 = \Sigma - \delta_1$$

Outside pitch diameter

$$d_{e1,2} = m_{et}\, z_{1,2}$$

Outside length of surface line on pitch cone

$$R_e = \frac{d_{e1,2}}{2 \sin \delta_{1,2}}$$

Length of surface line on the mean cone

$$R_m = R_e - 0.5\, b$$

Relative face width

$$\psi_R = \frac{b}{R_e}$$

Tangential module on the mean cone

$$m_{mt} = m_{et} \frac{R_m}{R_e}$$

Normal module on the mean cone

$$m_{mn} = m_{mt} \cos \beta_m$$

Mean pitch diameter

$$d_{m1,2} = m_{mt}\, z_{1,2}$$

Equivalent number of teeth

$$z_{v1,2} = \frac{z_{1,2}}{\cos\delta_{1,2}}$$

Ekvivalent pitch diameter

$$d_{v1,2} = \frac{d_{m1,2}}{\cos \delta_{1,2}}$$

Ekvivalent base diameter

$d_{vb1,2} = d_{v1,2} \cos \alpha_t$

Ekvivalent outside diameter

$d_{va1,2} = d_{v1,2} + 2\, h_{ae1,2}\, m_{mt} / m_{et}$

Ekvivalent center distance

$a_v = 0.5\, (d_{v1} + d_{v2})$

Virtual gear ratio

$$u_v = \frac{z_{v2}}{z_{v1}}$$

Virtual number of teeth

$$d_{n1,2} = \frac{z_{n1,2}}{\cos^3 \beta_m}$$

Virtual pitch diameter

$$d_{n1,2} = \frac{z_{n1,2}}{\cos^3 \beta_m}$$

Virtual base diameter

$d_{bn1,2} = d_{n1,2} \cos \alpha_t$

Virtual outside diameter

$d_{an1,2} = d_{n1,2} + 2\, h_{ae1,2}\, m_{mt} / m_{et}$

Virtual helix angle at the base cylinder

$\sin \beta_b = \sin \beta_m \cos \alpha_{nm}$

Virtual center distance

$a_n = 0.5\, (d_{n1} + d_{n2})$

Dedendum reduction

$k_{1,2} = 0.02\, (17 - z_{n1,2})$

for $k > 0$ the dedendum shortening is done

Addendum

$h_{ae1,2} = m_{et}\, (h_a^* + x_{1,2} - k_{1,2})$

Dedendum

$hf_{e1,2} = m_{et}\, (h_a^* + c^* - x_{1,2})$

Outside diameter

$d_{ae1,2} = d_{e1,2} + 2ha_{e1,2} \cos \delta_{1,2}$

Root diameter

$d_{fe1,2} = d_{e1,2} - 2\, _{hfe1,2} \cos \delta_{1,2})$

Outside diameter at small end

$d_{ai1,2} = d_{ae1,2}\, (1 - \psi_R)$

Vertex distance

$A_{1,2} = R_e \cos \delta_{1,2} - h_{ae1,2} \sin \delta_{1,2}$

Outside bevel angle

$$\delta_{a1,2} = \delta_{1,2} + \operatorname{arctg}\left(\frac{h_{ae1,2}}{R_e}\right)$$

Cutting angle

$$\delta_{f1,2} = \delta_{1,2} - \operatorname{arctg}\left(\frac{h_{fe1,2}}{R_e}\right)$$

Tooth thickness (measured normally on the pitch diameter)

$$s_{e1,2} = m_{et}\left(\frac{\pi}{2} + 2x_{1,2}\operatorname{tg}\alpha_t + x_{t1,2}\right)$$

Chordal facewidth (normal)

$$s_{ke1,2} = s_{e1,2}\cos^2\alpha_t$$

Addendum height above the chord

$$h_{ke1,2} = h_{ae1,2} - \frac{s_{ke1,2}\operatorname{tg}\alpha_t}{2}$$

Unit addendum width (measured normally)

$$s_{a1,2} = \frac{d_{ae1,2}}{m_{en}}\left(\frac{s_{e1,2}}{d_{e1,2}} + \frac{\operatorname{inv}\alpha_t - \operatorname{inv}\alpha_a}{\cos\delta_{1,2}}\right)$$

where:

$$\cos\alpha_a = \frac{d_{e1,2}}{d_{ae1,2}}\cos\alpha_t$$

Operating width of gears

$$b_w = b$$

Factor of mesh duration

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

$$\varepsilon_\alpha = \frac{\sqrt{d_{va1}^2 - d_{vb1}^2} + \sqrt{d_{va2}^2 - d_{vb2}^2} - 2a_v \sin\alpha_t}{2\pi m_{mt} \cos\alpha_t}$$

$$\varepsilon_{\alpha n} = \frac{\varepsilon_\alpha}{\cos^2\beta_b}$$

$$\varepsilon_\beta = \frac{0.85\, b \sin\beta_m}{\pi m_{mn}}$$

Minimum correction without tapering

$$x_{z1,2} = h_{a0}^* - \frac{1 - \dfrac{\cos\alpha_{nm}}{\cos\left(\operatorname{inv}\alpha_{nm} + \dfrac{\pi}{2z_{v1,2}}\right)}}{2\cos\beta_m} z_{v1,2}$$

where:

$$h_{a0}^* = h_a^* + c^* - r_f^* (1 - \sin\alpha_t)$$

Minimum correction without undercut

$$x_{p1,2} = h_{a0}^* - \frac{z_{n1,2}}{2}\sin^2\alpha_{nm}$$

Minimum correction without undercut

$$x_{d1,2} = \frac{5}{6}h^*_{a0} - \frac{z_{n1,2}}{2}\sin^2 \alpha_{nm}$$

Helix angle at end

$\sin\beta_e = \sin\beta_m R_m / R_e$

Normal pressure angle at end

$\tan\alpha_{ne} = \tan\alpha_t \cos\beta e$

Distribution of unit corrections for single gears

By user

User inputs x, x_t

With DIN

$$x = \frac{14 - z_{vn1}}{17}$$

With Merrit

$x = 0.02 \cdot (30 - z_{vn1})$

Complex

$$x = 2 \cdot \left(1 - \frac{1}{u^2}\right) \cdot \sqrt{\frac{\cos^3\beta_m}{z_1}},\ x_t = a + b \cdot (u - 2.5)$$

where:

a, b are constants dependent on helix angle

With compensation of relative slips, such as iteration of nonlinear equation

$$\frac{1}{\sqrt{\left(\frac{d_{va1}}{d_{vb1}}\right)^2 - 1}} - \frac{u_v}{\sqrt{\left(\frac{d_{va2}}{d_{vb2}}\right)^2 - 1}} + \frac{u_v - 1}{tg\alpha_t} = 0$$

Calculation of maximum root filleting

Used calculation
Unit root filleting

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

Calculation of strength proportions

Input values:

Input power P_1

Input speed n_1

Gearing ratio i

Gearing efficiency η

Calculated values

Output: $P_2 = P_1 \eta$ $n_2 = \frac{n_1}{i}$

Metric units

Input moment

$$T_1 = \frac{60000\, P_1}{2 \pi n_1} [Nm]$$

Tangential/circumferential force

$$F_t = \frac{2000\ T_1}{d_{m1}} [N]$$

Speed

$$n_1 = \frac{60000\, P_1}{2 \pi\, T_1} [rpm]$$

Circumferential speed

$$v = \frac{\pi\, d_{m1} n_1}{60000} [m/s]$$

Resonance speed

$$n_{E1} = \frac{1.9110^7}{z_1 d_{m2}} \sqrt{(1+u^2)(0.75\varepsilon_\alpha + 0.25)} [rpm]$$

ANSI (English) units

Input moment

$$T_1 = 550 \frac{60 P_1}{2\pi n_1} [lbft]$$

Tangential/circumferential force

$$F_t = \frac{24\, T_1}{d_{m1}} [lb]$$

Speed

$$n_1 = 550 \frac{60 P_1}{2\pi\, T_1} [rpm]$$

Circumferential speed

$$v = \frac{\pi d_{m1} n_1}{720} [ft/s]$$

Resonance speed

$$n_{E1} = \frac{7.5210^5}{z_1 d_{m2}} \sqrt{(1+u^2)(0.75\varepsilon_\alpha + 0.25)} [rpm]$$

Output moment

$$T_2 = T_1\, i\, \eta$$

Radial force

$$F_{r1a} = F_t * (tg\alpha * \cos\delta_1 - \sin\beta_m * \sin\delta_1) / \cos\beta_m$$

$$F_{r1b} = F_t * (tg\alpha * \cos\delta_2 + \sin\beta_m * \sin\delta_2) / \cos\beta_m$$

$$F_{r2a} = F_t * (tg\alpha * \cos\delta_1 + \sin\beta_m * \sin\delta_1) / \cos\beta_m$$

$$F_{r2b} = F_t * (tg\alpha * \cos\delta_2 - \sin\beta_m * \sin\delta_2) / \cos\beta_m$$

Axial force

$$F_{a1a} = F_t * (tg\alpha * \sin\delta_1 + \sin\beta_m * \cos\delta_1) / \cos\beta_m$$

$$F_{a1b} = F_t * (tg\alpha * \sin\delta_2 - \sin\beta_m * \cos\delta_2) / \cos\beta_m$$

$$F_{a2a} = F_t * (tg\alpha * \sin\delta_1 - \sin\beta_m * \cos\delta_1) / \cos\beta_m$$

$$F_{a2b} = F_t * (tg\alpha * \sin\delta_2 + \sin\beta_m * \cos\delta_2) / \cos\beta_m$$

Normal force

$$F_n = \frac{F_t}{\cos\alpha \cos\beta_m}$$

Strength calculation according to Bach

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

Allowable load

$F_{all} = \pi\ c\ b\ m_{en}\ \xi\ m_{en} \geq F_t$

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_{en}	normal module at end (normal diametral pitch at end inverse value)

$\xi = \frac{(1 - 0{,}5\psi_R)^2}{K}$ width factor

where:

	K = 1.4 for straight teeth, 1.25 for helical teeth
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. Adds an extra contact check.

Allowable load

$F_{all} = \pi\, c_{min}\, b_w\, \xi\, m_{en}\, \mu \geq F_t$

where:

$c_{min} = \min(c_b, c_c)$	minimum tooth allowable stress
b_w	operating facewidth
m_{en}	normal module at end (normal diametral pitch at end inverse value)
μ	factor of dependence on the precision degree (material property)
F_t	circumferential force acting on the teeth

$$\xi = \frac{(1 - 0{,}5\,\psi_R)^2}{K}$$ width factor

where:

K = 1.4 for straight teeth, 1.25 for helical 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 (material property)

y_c shape pressure factor (material property)

$$U = \left(\frac{m}{10}\right)^{0.2}$$ 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. Accessible only for metric units.

Safety factors

Contact fatigue

$$S_{H1,2} = \frac{\sigma_{Hlim1,2} \cdot Z_{N1,2} \cdot Z_L \cdot Z_R \cdot Z_v \cdot Z_{X1,2}}{Z_E \cdot Z_H \cdot Z_{B1,2} \cdot Z_\varepsilon \cdot Z_\beta \cdot Z_K \cdot \sqrt{\frac{F_t \cdot K_H}{b_w \cdot d_{m1}} \cdot \frac{u_v + 1}{u_v}}}$$

where:

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

F_t tangential force acting at teeth

b_w operating face width

Contact during one-time loading

$$S_{HSt1,2} = \frac{\sigma_{HPmax1,2}}{Z_E \cdot Z_H \cdot Z_{B1,2} \cdot Z_\varepsilon \cdot Z_\beta \cdot Z_K \cdot \sqrt{\frac{F_t \cdot K_H \cdot K_{AS}}{b_w \cdot d_{m1}} \cdot \frac{u_v + 1}{u_v}}}$$

where:

σ_{HPmax} contact fatigue limit (material property)

K_{AS} one-time overloading factor

Bending fatigue

$$S_{F1,2} = \frac{\sigma_{Flim1,2} \cdot Y_{A1,2} \cdot Y_{T1,2} \cdot Y_{N1,2} \cdot Y_{\delta 1,2} \cdot Y_{X1,2} \cdot Y_R}{Y_{Fa1,2} \cdot Y_{Sa1,2} \cdot Y_{Sarel1,2} \cdot Y_\beta \cdot Y_\varepsilon \cdot Y_K \cdot \frac{F_t \cdot K_F}{b_{wF1,2} \cdot m_{mn}}}$$

where:

σ_{Flim} bending fatigue limit (material property)

$b_{wF1,2}$ = btooth width for bending

Bending during one-time loading

$$S_{FSt1,2} = \frac{\sigma_{FPmax\,1,2} \cdot Y_{N1,2} \cdot Y_{X1,2}}{Y_{Fa1,2} \cdot Y_{Sa1,2} \cdot Y_{Sarel1,2} \cdot Y_{\beta} \cdot Y_{\varepsilon} \cdot Y_{K} \cdot \frac{F_t \cdot K_F \cdot K_{AS}}{b_{wF1,2} \cdot m_{mn}}}$$

where:

σ_{FPmax}allowable bending stress in dedendum (material property)

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

where:

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)

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

$1 \leq Z_Y \leq 1.6$ nitridated steels

$1 \leq Z_Y \leq 2.5$ other steels

where:

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:

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

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

pro $\sigma_{Hlim} > 1200$ 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: $Z_v = C_{Zv} + 2 \dfrac{1 - C_{Zv}}{\sqrt{0.8 + \dfrac{32}{v}}}$

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

Z_E ... elasticity factor

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

where:

μ Poisson's ratio (material value)

E modulus of elasticity (material value)

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 $\varepsilon_\beta = 0$:

$$Z_{B1,2} = \frac{tg\alpha_t}{\sqrt{\left[\sqrt{\left(\frac{d_{va1,2}}{d_{vb1,2}}\right)^2 - 1} - \frac{2\pi}{z_{v1,2}}\right] \cdot \left[\sqrt{\left(\frac{d_{va2,1}}{d_{vb2,1}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_{v2,1}}\right]}}$$

for $\varepsilon_\beta \geq 1$:

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

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

Z_ε ... contact ratio factor (for contact)

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 ratio factor (for bending)

CSN: for $\varepsilon_\beta < 1$:

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

CSN: for $\varepsilon_{\beta} \geq 1$:

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

DIN and ISO:

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

Z_{β} ... helix angle (for contact)

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

Y_{β} ... helix angle factor (for bending)

CSN:

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

DIN and ISO

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

for $\beta > 30$ deg. the $\beta = 30$ deg.is used

Z_x ... size factor (for contact)

Y_x ... size factor (for bending)

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

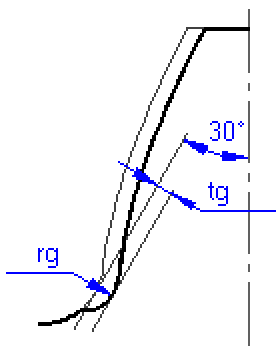
Y_{Sa} ... stress concentration during mesh by tooth end (regression function)

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

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

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 DIN and ISO: 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

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

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

$C_M = 0.8$

$C_R = 1$ for solid gears

$C_B = [1 + 0.5 (1.2 - h_f/m)] [1 - 0.02 (20 \text{ deg.} - \alpha)]$

$E_{steel} = 206\ 000$

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

$$f_{sh1,2} = A_{1,2} \cdot \left(\frac{b_w}{d_{m1,2}}\right)^2 \cdot \left[\left|B_{1,2} + 0.7\right| + 0.3\right] \cdot \frac{F_t}{b_w} \cdot K_A \cdot K_{Hv}$$

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

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

CSN:

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

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

for DIN and ISO:

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

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

for DIN and ISO: $1 \leq K_{F\alpha} \leq \dfrac{\varepsilon_{\gamma}}{\varepsilon_{\alpha} \cdot Y_{\varepsilon}}$

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

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

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

for DIN and ISO: $K_{H\alpha} = K_{F\alpha}$

limit values: $1 \leq K_{H\alpha} \leq \dfrac{\varepsilon_{\gamma}}{\varepsilon_{\alpha} \cdot Z_{\varepsilon}^{2}}$

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

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

where:

s_{ac} allowable contact stress (material property)
F_t tangential force acting at teeth
d_{m1} pitch diameter in middle plane
b_w operating tooth width

Safety factor of bending fatigue

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

where:

s_{at} allowable bending stress
P_{mt} tangential diametral pitch in middle plane
$b_{wF1,2} = \min(b_{1,2}, b_w + 2m)$ 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 size factor
K_s stress cycle factor for pitting resistance
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 alternating factor
C_f surface condition factor
K_R reliability factor
K_T temperature factor
K_B rim thickness factor

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