Power Transmission Systems

Amended Rules and Guidance

Rules for the Survey and Construction of Steel Ships Part D
Rules for the Survey and Construction of Inland Waterway Ships
Guidance for the Survey and Construction of Steel Ships Part D
Guidance for High Speed Craft
Guidance for the Survey and Construction of Inland Waterway Ships

Reason for Amendment

The requirements for power transmission systems specified in Chapter 5, Part D of the Rules for the Survey and Construction of Steel Ships were originally adopted in 1967. Although these requirements have been amended as necessary over the years, they have never been fundamentally reviewed.

Therefore, as a part of a comprehensive review of the ClassNK Rules, the Society conducted a survey on the needs of power transmission system manufacturers, and the responses received indicated that some clarification was desired regarding these requirements.

In addition, IACS decided to review the way it references standards in its URs and UIs etc. As a result, it adopted UR M56(Rev.4) in February 2021 and UR M56(Corr.1) in October 2021 to amend the way such standards are described in its URs and UIs.

Accordingly, relevant requirements were amended as a part of a comprehensive review of the Rules for the Survey and Construction of Steel Ships, based upon knowledge gained since 1967 and industry requests as well as in accordance with UR M56(Rev.4) and M56(Corr.1).

Outline of Amendment

The main contents of this amendment are as follows:

(1) Moved requirements for gear strength calculations specified in Annex D5.3.5, Part D of Guidance for the Survey and Construction of Steel Ships to Chapter 5, Part D of the Rules for the Survey and Construction of Steel Ships.

(2) Clarified the publication date for standards referenced in some of the requirements specified in the aforementioned Annex D5.3.5.

“Rules for the survey and construction of steel ships” has been partly amended as follows:

Part D MACHINERY INSTALLATIONS

Chapter 5 POWER TRANSMISSION SYSTEMS

5.2 Materials and Construction

5.2.1 Materials

Sub-paragraph -1 has been amended as follows.

1 Materials used for the following components (hereinafter referred to as “the principal components of the power transmission system”) are to comply with the requirements in **Part K**.
(1) Power transmission shafts <u>(including power take-off (PTO) shafts)</u> and gears
(2) Power transmission parts of couplings
(3) Power transmission parts of clutches
(4) Coupling bolts
2 (Omitted)

5.3 Strength of Gears

Paragraph 5.3.1 has been amended as follows.

5.3.1 Application*

The requirements in **5.3** apply to external tooth cylindrical gears having an involute tooth profile. All other gears are to be as deemed appropriate by the Society. In addition, <u>enclosed gear strength calculations are to be in accordance with **Annex 5.3.1 “Calculation of Strength of Enclosed Gears”**.</u>

Paragraph 5.3.5 has been amended as follows.

5.3.5 Detailed Evaluation for Strength*

Special consideration will be given to the gears, notwithstanding the requirements in **5.3.3** and **5.3.4**, provided that detailed data and calculations on their strength are submitted to the Society and considered appropriate. In addition<u>, the wording “detailed data and calculations on their strength” means calculations based on **Annex 5.3.1 “Calculation of Strength of Enclosed Gears”**.</u>

Annex 5.3.1 has been added as follows.

Annex 5.3.1 CALCULATION OF STRENGTH OF ENCLOSED GEARS

1.1 Application and Basic Principles

1.1.1 Application

This annex applies to enclosed gears used for transmission systems which transmit power from main propulsion machinery and prime movers driving generators and essential auxiliaries (excluding auxiliary machinery for specific use, etc., hereinafter the same in this annex).

1.1.2 Basic Principles

The gear strength calculation methods specified in this annex deal with surface durability (pitting) and tooth root bending strength. All influence factors related to strength are defined regarding their physical interpretation. Some of these factors are to be determined either by gear geometry etc,. Other factors are to be approximated according to methods deemed acceptable by the Society.

1.2 Symbols and Units

The main symbols introduced in this annex are listed below.

a : center distance (mm)
b : common facewidth (mm)
$b_{1,2}$: facewidth of pinion , wheel (mm)
d : reference diameter (mm)
$d_{1,2}$: reference diameter of pinion, wheel (mm)
$d_{a1,2}$: tip diameter of pinion, wheel (mm)
$d_{b1,2}$: base diameter of pinion, wheel (mm)
$d_{f1,2}$: root diameter of pinion, wheel (mm)
$d_{w1,2}$: working diameter of pinion, wheel (mm)
F_t : nominal tangential load (N)
h : tooth depth (mm)
m_n : normal module (mm)
m_t : transverse module (mm)
$n_{1,2}$: rotational speed of pinion, wheel (rpm)
P : maximum continuous power transmitted by the gear set (kW)
$T_{1,2}$: torque in way of pinion, wheel (Nm)
u : gear ratio
v : linear speed at pitch diameter (m/s)
$x_{1,2}$: addendum modification coefficient of pinion, wheel
z : number of teeth
$z_{1,2}$: number of teeth of pinion, wheel
z_n : virtual number of teeth
α_n : normal pressure angle at reference cylinder (°)
α_t : transverse pressure angle at reference cylinder (°)
α_{tw} : transverse pressure angle at working pitch cylinder (°)
β : helix angle at reference cylinder (°)

β_b : helix angle at base cylinder (°)
ε_α : transverse contact ratio
ε_β : overlap contact ratio
ε_γ : total contact ratio
σ_H : contact stress at the operating pitch point or at the inner point of single pair contact (N/mm^2)
σ_{HO} : basic value of contact stress (N/mm^2)
K_A : application factor
K_γ : load sharing factor
K_V : dynamic factor
$K_{H\alpha}$: transverse load distribution factor for contact stress
α_{Pn} : normal pressure angle of basic rack for cylindrical gear (°)
h_{fp} : dedendum of basic rack for cylindrical gear (mm)
$K_{H\beta}$: face load distribution factor for contact stress
σ_{HP} : permissible contact stress (N/mm^2)
σ_{Hlim} : endurance limit for contact stress (N/mm^2)
Z_N : life factor for contact stress
Z_L : lubricant factor
Z_V : speed factor
Z_R : roughness factor
Z_W : hardness ratio factor
Z_X : size factor for contact stress
S_H : safety factor for contact stress
Z_B : single pair mesh factor for pinion
Z_D : single pair mesh factor for wheel
Z_H : zone factor
Z_E : elasticity factor ($\sqrt{N/mm^2}$)
Z_ε : contact ratio factor
Z_β : helix angle factor for contact stress
σ_F : tooth root bending stress (N/mm^2)
Y_F : tooth form factor
h_F : bending moment arm for tooth root bending stress for application of load at the outer point of single tooth pair contact (mm)
S_{FN} : tooth root chord in the critical section (mm)
α_{Fen} : pressure angle at the outer point of single tooth pair contact in the normal section (°)
Y_S : stress correction factor
ρ_F : root fillet radius in the critical section (mm)
ρ_{fp} : root fillet radius of the basic rack for cylindrical gears (mm)
S_{pr} : residual fillet undercut (mm)
Y_β : helix angle factor for tooth root bending stress
Y_B : rim thickness factor
s_R : rim thickness of gears (mm)

h : tooth height (mm)
Y_{DT} : deep tooth factor
$K_{F\alpha}$: transverse load distribution factor for tooth root bending stress
$K_{F\beta}$: face load distribution factor for tooth root bending stress
σ_{FP} : permissible tooth root bending stress (N/mm^2)
σ_{FE} : bending endurance limit (N/mm^2)
Y_N : life factor for tooth root bending stress
Y_d : design factor
$Y_{\delta\ relT}$: relative notch sensitivity factor
q_s : notch parameter
ρ' : slip-layer thickness (mm)
Y_{RrelT} : relative surface factor
Y_X : size factor for tooth root bending stress
S_F : safety factor for tooth root bending stress

1.3 Geometrical Definitions

In the case of internal gearing z_2, a, d_2, d_{a2}, d_{b2} and d_{w2} are negative. The pinion is defined as the gear with the smaller number of teeth; therefore, the absolute value of the gear ratio, defined as follows, is always greater or equal to the unity.

$$u = \frac{z_2}{z_1} = \frac{d_{w2}}{d_{w1}} = \frac{d_2}{d_1}$$

In the case of external gears, u is positive. In the case of internal gears, u is negative. In the equation of surface durability, b is the common facewidth on the pitch diameter. In the equation of the tooth root, bending stress b_1 or b_2 are the facewidths at their respective tooth roots. In any case, b_1 and b_2 are not to be taken as greater than b by more than one module (m_n) on either side. The common facewidth b may be used also in the equation of teeth root bending stress if either significant crowning or end relief has been adopted.

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

$$\tan\beta_b = \tan\beta\cos\alpha_t$$

$$d_{1,2} = \frac{z_{1,2}m_n}{\cos\beta}$$

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

$$\left.\begin{array}{l} d_{w1} = \dfrac{2a}{u+1} \\ d_{w2} = \dfrac{2au}{u+1} \end{array}\right\} \text{where } a = 0.5(d_{w1} + d_{w2})$$

$$z_{n1,2} = \frac{z_{1,2}}{\cos^2\beta_b \cdot \cos\beta}$$

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

$$\text{inv}\alpha = \tan\alpha - \frac{\pi\alpha}{180};\ \alpha(°)$$

$$\mathrm{inv}\alpha_{tw} = \mathrm{inv}\alpha_t + 2\tan\alpha_n \frac{x_1 + x_2}{z_1 + z_2}$$

or

$$\cos\alpha_{tw} = \frac{m_t(z_1 + z_2)}{2a}\cos\alpha_t$$

$$\varepsilon_\alpha = \frac{0.5\sqrt{d_{a1}^2 - d_{b1}^2} \pm 0.5\sqrt{d_{a2}^2 - d_{b2}^2} - a\sin\alpha_{tw}}{\pi m_t \cos\alpha_t}$$

A positive sign is used for external gears, a negative sign for internal gears.

$$\varepsilon_\beta = \frac{b\sin\beta}{\pi m_n}$$

In the case of double helix gears, b is to be taken as the width of one helix.

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

$$v = \frac{\pi d_{1,2} n_{1,2}}{60 \cdot 10^3}$$

1.4 Nominal Tangential Load, F_t

Nominal tangential loads, F_t, which are tangential to cylinders and perpendicular to planes are to be calculated directly from the maximum continuous power transmitted by gear sets using the following equations:

$$T_{1,2} = \frac{30 \cdot 10^3 P}{\pi n_{1,2}}$$

$$F_t = 2000\frac{T_{1,2}}{d_{1,2}}$$

1.5 Loading Factors

1.5.1 Application Factor, K_A

1 The application factor, K_A, accounts for dynamic overloads from source external to the gearing. K_A for gears designed for infinite lifespans is defined as the ratio between maximum repetitive cyclic torques applied to gear sets and nominal rated torques. Nominal rated torque is defined by rated power and speed and is the torque used in rating calculations. This factor mainly depends on the following:

(1) The characteristics of driving and driven machines;

(2) The ratio of masses;

(3) The type of couplings;

(4) Operating conditions (over speed, changes in propeller load conditions, etc.)

2 In cases where drive systems are operating at level near their critical speed, a careful analysis of conditions is to be made. K_A is to be determined either by direct measurements or by a system analysis that is acceptable to the Society. In cases where values determined in such ways cannot be provided, the following values may be used:

(1) Main propulsion

K_A = 1.00 (reciprocating internal combustion engines with hydraulic or electromagnetic slip couplings)

= 1.30 (reciprocating internal combustion engines with high elasticity couplings)

= 1.50 (reciprocating internal combustion engines with other couplings)

However, in cases where vessels using reduction gears are affixed with **Ice Class** Notation, as required in **8.6, Part I of the Rules**.

(2) Auxiliary gears

K_A = 1.00 (electric motors, reciprocating internal combustion engines with hydraulic or electromagnetic slip couplings)

= 1.20 (reciprocating internal combustion engines with high elasticity couplings)

= 1.40 (reciprocating internal combustion engines with other couplings)

1.5.2 Load Sharing Factor, K_γ

The load sharing factor, K_γ, accounts for the maldistribution of loads in multiple path transmissions (dual tandems, epicyclics, double helixes, etc.). K_γ is defined as the ratio between those maximum loads through actual paths and those evenly distributed loads. This factor mainly depends on the accuracy and the flexibility of the branches. K_γ is to be determined by measurements or by system analysis. In cases where values determined in such ways cannot be provided, the following values can be used with respect to epicyclic gears:

K_γ = 1.00 (up to 3 planetary gears)

= 1.20 (4 planetary gears)

= 1.30 (5 planetary gears)

= 1.40 (6 planetary gears and over)

1.5.3 Internal Dynamic Factor, K_V

1 The internal dynamic factor, K_V, accounts for those internally generated dynamic loads due to vibrations of pinions and wheels against each other. K_V is defined as the ratio between those maximum loads which dynamically act on tooth flanks and maximum externally applied loads ($F_t K_A K_V$). This factor mainly depends on the following:

(1) Transmission errors depending on pitch and profile errors;
(2) Masses of pinions and wheels;
(3) Gear mesh stiffness variations as gear teeth pass through meshing cycles;
(4) Transmitted loads including application factors;
(5) Pitch line velocities;
(6) Dynamic unbalance of gears and shafts;
(7) Shaft and bearing stiffness;
(8) Damping characteristics of gear systems.

2 The internal dynamic factor, K_V, is to be calculated as follows; however, this method is to be applied only to cases where all of the following conditions **(1)** to **(4)** are satisfied:

(1) Running speeds in the following subcritical ranges:

$$\frac{v z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 10 \ (m/s)$$

(2) $\beta = 0°$ (In the case of spur gears)

$\beta \leq 30°$ (In the case of helical gears)

(3) pinion with relatively low number of teeth:

$z_1 < 50$

(4) solid disc wheels or heavy steel gear rim

This method may be applied to all types of gears, if $\frac{v z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 3$ (m/s), as well as to helical gears where $\beta > 30°$. For gears other than the above, reference is to be made to Method B outlined in the reference standard *ISO* 6336-1:2019.

(a) For those helical gears with an overlap ratio $\geq$ unity and spur gears, the value of K_V is to be determined as follows:

$$K_V = 1 + \left(\frac{K_1}{K_A \frac{F_t}{b}} + K_2 \right) \cdot \frac{v \cdot z_1}{100} K_3 \sqrt{\frac{u^2}{1+u^2}}$$

K_1 : Factor specified in **Table 5.3-1.**

K_2 : Factors for all *ISO* accuracy grades. Values are as follows:

= 0.0193 (In the case of spur gears)

= 0.0087 (In the case of helical gears)

K_3 : Values are to be calculated as follows:

$$= 2.0 \left(\frac{v \cdot z_1}{100} \sqrt{\frac{u^2}{1+u^2}} \le 0.2 \right)$$

$$= 2.071 - 0.357 \cdot \frac{v \cdot z_1}{100} \sqrt{\frac{u^2}{1+u^2}} \left(\frac{v \cdot z_1}{100} \sqrt{\frac{u^2}{1+u^2}} > 0.2 \right)$$

If $K_A F_t / b$ is less than 100 *N/mm*, this value is assumed to be equal to 100 *N/mm*.

(b) In the case of helical gears with an overlap ratio < unity, the value of K_V is to be obtained by means of linear interpolation as follows:

$$K_V = K_{V2} - \varepsilon_\beta (K_{V2} - K_{V1})$$

K_{V1} : Values for helical gears specified in accordance with **(a)**

K_{V2} : Values for spur gears specified in accordance with **(a)**

In the case of mating gears with different grades of accuracy, the grade corresponding to the lower accuracy is to be used.

Table 5.3-1 Values of K_1

Type of gears	*ISO* grades of accuracy					
	3	4	5	6	7	8
Spur gears	2.1	3.9	7.5	14.9	26.8	39.1
Helical gears	1.9	3.5	6.7	13.3	23.9	34.8

Notes:

ISO accuracy grades according to *ISO* 1328-2:2020.

1.5.4 Face Load Distribution Factors, $K_{H\beta}$ and $K_{F\beta}$

The face load distribution factors, $K_{H\beta}$ for contact stress, and $K_{F\beta}$ for tooth root bending stress account for the effects of the non-uniform distribution of loads across facewidths. $K_{H\beta}$ is defined as the ratio between the maximum load per unit facewidth and the mean load per unit facewidth, and $K_{F\beta}$ is defined as the ratio between the maximum bending stress at tooth root per unit facewidth and the mean bending stress at tooth root per unit facewidth.

The mean bending stress at tooth root relates to the considered facewidth b_1 and b_2 respectively. $K_{F\beta}$ can be expressed as a function of the factor, $K_{H\beta}$. $K_{H\beta}$ and $K_{F\beta}$ mainly depend on the following:

(1) Gear tooth manufacturing accuracy;

(2) Errors in mounting due to bore errors;

(3) Bearing clearances;

(4) Wheel and pinion shaft alignment errors;

(5) Elastic deflections of gear elements, shafts, bearings, housing, and foundations which support

the gear elements;

(6) Thermal expansion and distortion due to operating temperature;

(7) Compensating design elements (tooth crowning, end relief, etc.).

The value for $K_{H\beta}$ is to be determined as follows:

if $\frac{F_{\beta y} C_\gamma b}{2F_t K_A K_\gamma K_V} \geq 1$

then $K_{H\beta} = \sqrt{\frac{2F_{\beta y} C_\gamma b}{F_t K_A K_\gamma K_V}}$

if $\frac{F_{\beta y} C_\gamma b}{2F_t K_A K_\gamma K_V} < 1$

then $K_{H\beta} = 1 + \frac{F_{\beta y} C_\gamma b}{2F_t K_A K_\gamma K_V}$

C_γ : tooth stiffness parameter (N/(mm・μm))

The value for C_γ is to be calculated as follows:

$$C_\gamma = 0.8 C_{th} C_R C_B \cos\beta (0.75\varepsilon_\alpha + 0.25)$$

$$C_{th} = \frac{1}{q}$$

$$q = 0.04723 + \frac{0.15551}{z_{n1}} - \frac{0.25791}{z_{n2}} - 0.00635 x_1 - \frac{0.11654}{z_{n1}} x_1$$

$$-0.00193 x_2 - \frac{0.24188}{z_{n2}} x_2 + 0.00529 x_1^2 + 0.00182 x_2^2$$

$C_R = 1$ (In the case of solid disc gears)

$C_R = 1 + \frac{\ln\left(\frac{b_s}{b}\right)}{5e^{\left(\frac{s_R}{5m_n}\right)}}$ (In the case of non-solid disc gears)

b_s : thickness of central web (mm)

s_R : average thickness of rim (mm)

However, in cases where for non-solid disc gears of $\frac{b_s}{b} < 0.2$ or $\frac{s_R}{m_n} < 1.0$, the value for C_R is to be determined by the Society on a case by case basis.

C_B is a basic rack coefficient that accounts for the deviations between the actual basic rack profile of gears and their standard basic rack profile. The value for C_B is to be calculated as follows:

$$C_B = \left[1 + 0.5\left(1.2 - \frac{h_{fp}}{m_n}\right)\right][1\text{-}0.02(20^\circ - \alpha_{Pn})]$$

$F_{\beta y}$ =effective equivalent misalignment (μm). The value for $F_{\beta y}$ is to be calculated as follows:

$$F_{\beta y} = F_{\beta x} - y_\beta$$

In the case of gears that are not surface hardened

$$y_\beta = \frac{320}{\sigma_{Hlim}} F_{\beta x}$$

However, the following conditions are to be satisfied.

$$y_\beta \leq F_{\beta x}$$

$\gamma_\beta \leq 25600/\sigma_{Hlim}$ (5<ν<10 m/sec)

$\gamma_\beta \leq 12800/\sigma_{Hlim}$ (10 m/sec<ν)

In the case of surface hardened gears

$\gamma_\beta = 0.15F_{\beta x}$

However, $\gamma_\beta \leq 6.0$ (μm) is to be satisfied.

$F_{\beta x}$: original effective equivalent misalignment (μm), $F_{\beta x}$ is to be calculated as follows:

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

f_{sh} : takes into account the components of equivalent misalignment resulting from bending and twisting of pinion and pinion shaft, f_{sh} is to be calculated as follows (μm):

In the case of gears without crowning or end relief

$$f_{sh} = 0.023\frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with end relief

$$f_{sh} = 0.016\frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with crowning

$$f_{sh} = 0.012\frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with helix angle modification

$f_{sh} = 0$

However, in all cases f_{sh} is not to be taken as value less than that calculated by the following expressions:

$0.005\frac{F_t K_A K_\gamma K_V}{b}$ (In the case of spur gears)

or

$0.010\frac{F_t K_A K_\gamma K_V}{b}$ (In the case of helical gears)

γ = pinion ratio factor. The value for γ is to be calculated as follows:

$$\gamma = \left[\left|0.7 + K'\frac{\ell S}{d_1^2}\left(\frac{d_1}{d_{sh}}\right)^4\right| + 0.3\right]\left(\frac{b}{d_1}\right)^2 \text{ (In the case of spur and helical gears)}$$

$$\gamma = 2\left[\left|1.2 + K'\frac{\ell S}{d_1^2}\left(\frac{d_1}{d_{sh}}\right)^4\right| + 0.3\right]\left(\frac{b}{d_1}\right)^2 \text{ (In the case of double helical gears)}$$

K', ℓ and S are constant factors used for the calculation of the pinion ratio factor, γ, the bearing span and the distance between mid-plane of pinion and middle of such bearing spans, respectively. Values for K' are given in **Table 5.5-1**.

f_{ma} = the misalignment resulting from manufacturing errors (μm). The value for f_{ma} is to be calculated as follows:

$f_{ma} = 1.0F_\beta$ (In the case of the assembly of gears without any modification or adjustment)

$= 0.7F_\beta$ (In the case of gear pairs with well-designed end relief)

$= 0.5F_\beta$ (In the case of gear pairs with means for adjustment or with helix modifications or suitably crowned)

F_β : tolerance on total helix deviation (μm)

The value for $K_{F\beta}$ is to be determined as follows:

(1) In cases where the hardest contact is at the end of the facewidth, $K_{F\beta}$ is to be calculated as

follows:

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

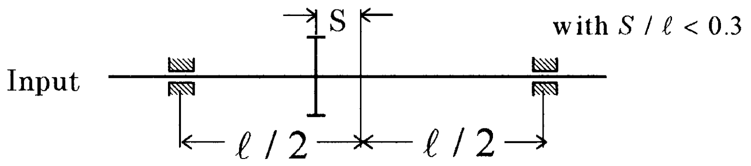
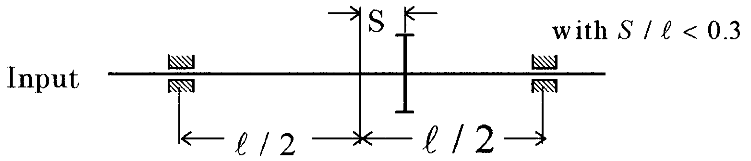
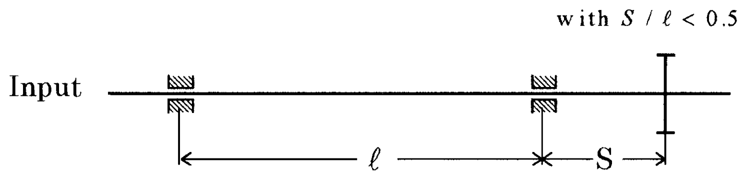
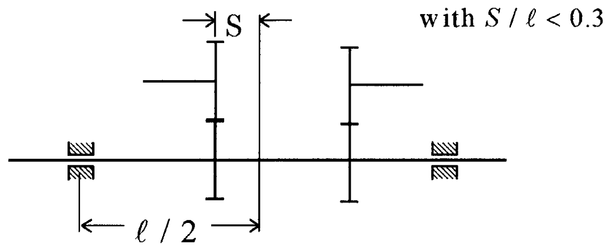
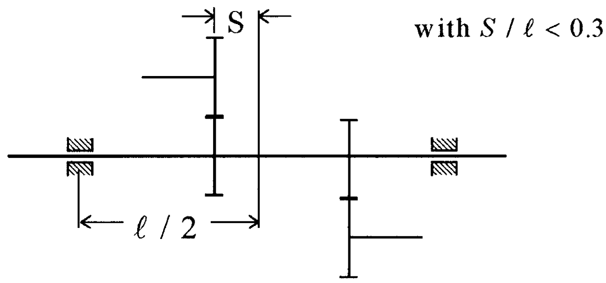
$$N = \frac{\left(\frac{b}{h}\right)^2}{1 + \frac{b}{h} + \left(\frac{b}{h}\right)^2}$$

b/h= facewidth/tooth height ratio, the smaller of b_1/h_1 or b_2/h_2. In the case of double helical gears, the facewidth of only one is to be used. However, in cases where b/h<3.0, b/h is to be taken as 3.0.

(2) In cases of gears where the ends of the facewidth are lightly loaded or unloaded (end relief or crowning), the value for $K_{F\beta}$ is to be calculated as follows:

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

Table 5.5-1 Values of K'

	K'	
	with stiffening[1)]	without stiffening[2)]
	0.48	0.80
	-0.48	-0.80
	1.33	1.33
	-0.36	-0.60
	-0.6	-1.0

Notes:

1) In cases where $d_1/d_{sh} \geq 1.15$, stiffening is assumed.

2) In cases where $d_1/d_{sh} < 1.15$, or where the pinion slides on a shaft or is shrink fitted, no stiffening is assumed.

However, d_{sh} is the external diameter of a solid shaft equivalent to the actual one in bending deflection (mm)

1.5.5 Transverse Load Distribution Factors, $K_{H\alpha}$ and $K_{F\alpha}$

The transverse load distribution factors, $K_{H\alpha}$ for contact stress and $K_{F\alpha}$ for tooth root bending stress, account for the effects of pitch and profile errors on the transversal load distribution between two or more pairs of teeth in mesh. $K_{H\alpha}$ and $K_{F\alpha}$ mainly depend on the following:

(1) Total mesh stiffness;
(2) Total tangential loads, $F_t\ K_A\ K_\gamma\ K_V\ K_{H\beta}$
(3) Base pitch errors
(4) Tip relief
(5) Running-in allowances

$K_{H\alpha}$ and $K_{F\alpha}$ are to be determined as follows:

$$K_{H\alpha} = K_{F\alpha} = \frac{\varepsilon_\gamma}{2}\left(0.9 + 0.4\frac{C_\gamma(f_{pb}-y_\alpha)^b}{F_t K_A K_\gamma K_V K_{H\beta}}\right)\ (\varepsilon_\gamma \le 2)$$

$$K_{H\alpha} = K_{F\alpha} = 0.9 + 0.4\sqrt{\frac{2(\varepsilon_\gamma-1)}{\varepsilon_\gamma}}\frac{C_\gamma(f_{pb}-y_\alpha)^b}{F_t K_A K_\gamma K_V K_{H\beta}}\ (\varepsilon_\gamma > 2)$$

however

$$1.0 \le K_{H\alpha} \le \frac{\varepsilon_\gamma}{\varepsilon_\alpha Z_\varepsilon^2}$$

$$1.0 \le K_{F\alpha} \le \frac{\varepsilon_\gamma}{\varepsilon_\alpha Y_\varepsilon}$$

where

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

The value for $\varepsilon_{\alpha n}$ is the same as that specified in **1.7.2**.

All symbols used in the equation to determine $K_{H\alpha}$ and $K_{F\alpha}$, except for y_α and f_{pb}, are the same as those used in the equation for determining $K_{H\beta}$.

y_α and f_{pb} are to be calculated as follows:

$$y_\alpha = \frac{160}{\sigma_{Hlim}} f_{pb} \text{ (In the case of through hardened gears)}$$

$$= 0.075 f_{pb} \quad \text{(In the case of surface hardened gears)}$$

f_{pb} is to be taken as the larger value of base pitch deviation of pinions or wheels (μm)

1.6 Surface Strength

1.6.1 Equation

The criterion for surface strength is based on the Hertz pressure on operating pitch points or at inner points of single pair contacts. This criterion, as given by the following equation, is that contact stress σ_H is to be equal to or less than permissible contact stress σ_{HP}.

$$\sigma_H = \sigma_{HO}\sqrt{K_A K_\gamma K_V K_{H\alpha} K_{H\beta}} \le \sigma_{HP}$$

where

σ_{HO} is basic value of contact stress for pinions and wheels (N/mm^2).

1.6.2 Equations for Basic Contact Stress

1 Basic contact stresses for pinions and wheels are to be calculated as follows:

$\sigma_{HO} = Z_B Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}}$ (In the case of pinions)

$= Z_D Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}}$ (In the case of wheels)

2 Single Pair Mesh Factors Z_B and Z_D

The single-pair mesh factors, Z_B for pinions and Z_D for wheels, account for the influence on contact stress of tooth flank curvatures at inner points of single pair contacts. These factors transform those contact stresses determined at pitch points to contact stresses considering flank curvatures at inner points of single pair contacts. Z_B and Z_D are to be determined as follows:

(1) In the case of spur gears, the value for Z_B is to be taken as 1.0 or as follows, whichever is greater.

$$M_1 = \frac{\tan\alpha_{tw}}{\left\{\left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - \frac{2\pi}{z_1}\right]\left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_2}\right]\right\}^{\frac{1}{2}}}$$

(2) In the case of helical gears with $\varepsilon_\beta \geq 1$, the value for Z_B is to be taken as 1.0;

(3) In the case of helical gears with $\varepsilon_\beta > 1$, the value for Z_B is to be taken as 1.0 or as follows, whichever is greater:

$Z_B = M_1 - \varepsilon_\beta(M_1 - 1)$

(4) In the case of spur gears, the value for Z_D is to be taken as 1.0 or as follows, whichever is greater.

$$M_2 = \frac{\tan\alpha_{tw}}{\left\{\left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - \frac{2\pi}{z_2}\right]\left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_1}\right]\right\}^{\frac{1}{2}}}$$

(5) In the case of helical gears with $\varepsilon_\beta \geq 1$, Z_D is to be taken as 1.0.

(6) In the case of helical gears with $\varepsilon_\beta < 1$, Z_D is to be taken as 1.0 or as follows, whichever is greater.

$Z_D = M_2 - \varepsilon_\beta(M_2 - 1)$

(7) In the case of internal gears, Z_D is to be taken as 1.0.

3 Zone Factor, Z_H

The zone factor, Z_H, accounts for the influence on Hertzian pressure of tooth flank curvatures at pitch points and relates those tangential forces at reference cylinders to those normal forces at pitch cylinders. Z_H is to be calculated as follows:

$$Z_H = \sqrt{\frac{2\cos\beta_b}{\cos^2\alpha_t \tan\alpha_{tw}}}$$

4 Elasticity Factor, Z_E

The elasticity factor, Z_E, accounts for the influence of the material properties *E* (modulus of elasticity) and υ (Poisson's ratio) on the Hertzian pressure. In the case of steel gears, Z_E is to be calculated as follows:

$Z_E = 189.8(\sqrt{N/mm^2})$

In other cases, reference is to be made to the reference standard *ISO* 6336-2:2019.

5 Contact Ratio Factor, Z_ε

The contact ratio factor, Z_ε, accounts for the influence of transverse contact ratios and overlap ratios on the specific surface loads of gears.

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}} \quad \text{(In the case of spur gears)}$$

$$= \sqrt{\frac{4-\varepsilon_\alpha}{3}(1-\varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}} \quad \text{(In the case of helical gears with } \varepsilon_\beta < 1\text{)}$$

$$= \sqrt{\frac{1}{\varepsilon_\alpha}} \quad \text{(In the case of helical gears with } \varepsilon_\beta \geq 1\text{)}$$

6 Helix Angle Factor, Z_β

The helix angle factor, Z_β, accounts for the influence of helix angles on surface durability, allowing for such variables as distribution of loads along lines of contact. Z_β is dependent only on helix angles and its value can be obtained by the following formula:

$$Z_\beta = \sqrt{\frac{1}{\cos\beta}}$$

where β is the reference helix angle.

1.6.3 Permissible Contact Stress

1 Permissible contact stress, σ_{HP} is to be calculated as follows:

$$\sigma_{HP} = \sigma_{Hlim}\frac{Z_N Z_L Z_V Z_R Z_W Z_X}{S_H}$$

2 Endurance Limit for Contact Stress, σ_{Hlim}

For a given material, σ_{Hlim} is the limit of repeated contact stress which can be permanently endured. σ_{Hlim} can be regarded as the level of contact stress which the material will endure without pitting for at least 5×10^7 load cycles. “Pitting” is defined in the case of non-surface hardened gears, the pitted area > 2 % of total active flank area; in the case of surface hardened gears, the pitted area > 0.5 % of total active flank area, or > 4 % of one particular tooth flank area. The σ_{Hlim} values are to correspond to a failure probability of 1 % or less.

The endurance limit mainly depends on the following:

(1) Material composition, cleanliness and defects;
(2) Mechanical properties;
(3) Residual stresses;
(4) Hardening process, depth of hardened zone, hardness gradient;
(5) Material structure (forged, rolled bar, cast).

Endurance limit for contact stress σ_{Hlim} is as given in **Table 6.3-1**. However, for materials having enough data showing their higher endurance limit, values larger than those given in the table may be allowed by the Society in consideration of factors **(1)** through **(5)** mentioned above.

Table 6.3-1 Value of σ_{Hlim} (N/mm^2)

Steel type	σ_{Hlim}
Normalized structural steels	HB+190
Through hardening carbon steels	HB+350
Through hardening alloy steels	1.33HB+367
Induction hardened alloys	0.6HV+850
Nitrided alloys	1000
Soft nitrided alloys	1.14HV+437; however, 950 for HV>450

Nitrided steels	1250
Carburized hardened alloys	1500

Note:
HB: Brinell Hardness; *HV*: Vickers Hardness

3 Life Factor for Contact Stress, Z_N

The life factor for contact stress, Z_N, accounts for the higher permissible contact stress in cases where a limited life (number of cycles) is required. Values larger than 1.0 are to be considered by the Society on a case-by-case basis.

This factor mainly depends on the following:

(1) Material and heat treatment;
(2) Number of cycles;
(3) Influence factors (Z_R, Z_V, Z_L, Z_W, Z_X).

The life factor, Z_N, is to be determined according to Method B outlined in the reference standard *ISO* 6336-2:2019.

4 Lubricant Factor, Z_L

The lubricant factor, Z_L, like the speed factor, Z_V, and roughness factor, Z_R accounts for the influence of the type of lubricant and its viscosity on surface endurance. These factors are to be determined for softer materials in cases where gear pairs are of different hardness. These factors mainly depend on the following:

(1) Viscosity of lubricant in contact zones;
(2) The sum of the instantaneous velocity of tooth surfaces;
(3) Loads;
(4) Relative radius of curvature at pitch points;
(5) Surface roughness of teeth flanks;
(6) Hardness of pinions and wheels.

The value for Z_L is to be calculated as follows:

$$Z_L = C_{ZL} + \frac{4(1.0 - C_{ZL})}{(1.2 + 134/v_{40})^2}$$

where

$$C_{ZL} = \frac{\sigma_{Hlim} - 850}{350} 0.08 + 0.83$$

(In cases where 850 $N/mm^2 \leq \sigma_{Hlim} \leq 1200$ N/mm^2)
= 0.83 (In cases where $\sigma_{Hlim} < 850$ N/mm^2)
= 0.91 (In cases where $\sigma_{Hlim} > 1200$ N/mm^2)

v_{40} : Nominal kinematic viscosity of the oil at 40℃ (mm^2/s)

5 Speed Factor, Z_V

The speed factor, Z_V, accounts for the influence of pitch line velocities on surface endurance. The value for Z_V is to be calculated as follows:

$$Z_V = C_{Z_V} + \frac{2(1.0 - C_{Z_V})}{\sqrt{0.8 + 32/v}}$$

where

$$C_{Z_V} = \frac{\sigma_{Hlim} - 850}{350} 0.08 + 0.85$$

(In cases where 850 $N/mm^2 \leq \sigma_{Hlim} \leq 1200$ N/mm^2)
= 0.85 (In cases where $\sigma_{Hlim} < 850$ N/mm^2)
= 0.93 (In cases where $\sigma_{Hlim} > 1200$ N/mm^2)

6 Roughness Factor, Z_R

The roughness factor, Z_R, accounts for the influence of surface roughness on surface endurance. The value for Z_R is to be calculated as follows:

$$Z_R = \left(\frac{3}{R_{Z10}}\right)^{C_{ZR}}$$

$$R_{Z10} = R_Z \cdot \sqrt[3]{\frac{10}{\rho_{red}}}$$

$$R_Z = \frac{R_{Z1} + R_{Z2}}{2}$$

where

R_{Z1}, R_{Z2} : Respective mean peak to valley roughness for pinions and wheels.

R_Z : Refer to the reference standard *ISO* 6336-2:2019.

ρ_{red} : Relative radius of curvature. The value for ρ_{red} is to be calculated as follows:

$$\rho_{red} = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}$$

$\rho_{1,2} = 0.5 \cdot d_{b1,2} \cdot \tan\alpha_{tw}$ (In the case of internal gears the value d_b is negative)

In cases where the roughness stated is an arithmetic mean roughness, i.e. R_a value, and the conversion $R_Z = 6R_a$ can be applied.

$C_{ZR} = 0.32 - 0.0002\sigma_{Hlim}$ (In cases where 850 $N/mm^2 \leq \sigma_{Hlim} \leq 1200$ N/mm^2)

$= 0.15$ (In cases where $\sigma_{Hlim} < 850$ N/mm^2)

$= 0.08$ (In cases where $\sigma_{Hlim} > 1200$ N/mm^2)

7 Hardness Ratio Factor, Z_W

The hardness ratio factor, Z_W, accounts for the increase in surface durability of soft steel gears meshing with significantly harder gears with smooth surfaces in the following cases:

(1) Surface-hardened pinion with through-hardened wheel

$$Z_W = 1.2\left(\frac{3}{R_{zH}}\right)^{0.15} \quad (HB < 130)$$

$$= \left(1.2 - \frac{HB-130}{1700}\right) \cdot \left(\frac{3}{R_{zH}}\right)^{0.15} \quad (130 \leq HB \leq 470)$$

$$= \left(\frac{3}{R_{zH}}\right)^{0.15} \quad (HB > 470)$$

where

HB : Brinell hardness of the tooth flanks of the softer gear of the pair

R_{ZH} : equivalent roughness (μm)

$$R_{zH} = \frac{R_{Z1}(10/\rho_{red})^{0.33}(R_{Z1}/R_{Z2})^{0.66}}{(v \cdot \nu_{40}/1500)^{0.33}}$$

(2) Through-hardened pinion and wheel

When the pinion is substantially harder than the wheel, the work hardening effect increases the load capacity of the wheel flanks. Z_W applies to the wheel only, not to the pinion.

$Z_W = 1$ ($HB_1/HB_2 < 1.2$)

$= 1 + \left(0.00898\frac{HB_1}{HB_2} - 0.00829\right) \cdot (u-1)$ ($1.2 \leq HB_1/HB_2 \leq 1.7$)

$= 1 + 0.00698 \cdot (u-1)$ ($HB_1/HB_2 > 1.7$)

$HB_{1,2}$: Brinell hardness of the pinion and the wheel respectively.

If gear ratio u>20, then the value u=20 is to be used. In any case, if the calculated Z_W <1, then the value Z_W = 1 is to be used.

(3) In cases other than **(1)** and **(2)** above;

Z_W = 1

8 Size Factor for Contact Stress, Z_X

The size factor for contact stress, Z_X, accounts for the influence of tooth dimensions on permissible contact stress and reflects the inhomogeneity of material properties. This factor mainly depends on the following:

(1) Materials and heat treatments;

(2) Tooth and gear dimensions;

(3) Ratio of case depth to tooth size;

(4) Ratio of case depth to equivalent radius of curvature.

For through hardened gears and for surface hardened gears with adequate case depth relative to tooth size and radius of relative curvature Z_X = 1.0, in cases where the case depth is relatively shallow then a smaller value of Z_X is to be taken.

9 Safety Factor for Contact Stress, S_H

The safety factor for contact stress, S_H, is to be taken as follows:

(1) In the case of main propulsion gears: 1.20

(2) In the case of auxiliary gears: 1.15

In cases where the gears of duplicated independent propulsion or auxiliary machinery, has been duplicated beyond that what is required for its respective class, a reduced value may be taken at the discretion of the Society.

1.7 Bending Strength

1.7.1 Equation

The tooth root bending stress σ_F and the permissible tooth root bending stress σ_{FP} are to be calculated separately for the pinion and the wheel. The criterion for tooth root bending strength, as given by the following equation, is that the tooth root bending stress in the tooth root fillet σ_F is to be equal to or less than the permissible tooth root bending stress σ_{FP}.

$$\sigma_F = \frac{F_t}{bm_n} Y_F Y_S Y_\beta Y_B Y_{DT} K_A K_\gamma K_V K_{F\alpha} K_{F\beta} \leq \sigma_{FP}$$

However, the following definitions and equations apply only to those gears having a rim thickness greater than 3.5 m_n. The results of calculations using the following method are acceptable for normal pressure angles up to 25 degrees and reference helix angles up to 30 degrees. In the case of larger pressure angles and larger helix angles, the calculated results are to be confirmed by experience as by Method A of the reference standard *ISO* 6336-3:2019.

1.7.2 Tooth Root Bending Stress for Pinion and Wheel

1 Tooth Form Factor, Y_F

The tooth form factor, Y_F, accounts for the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact. Y_F is to be determined separately for the pinion and the wheel. In the case of helical gears, the form factors for gearing are to be determined in normal sections (i.e. for virtual spur gears with virtual numbers of teeth, Z_n).The value for Y_F is to be calculated as follows:

$$Y_F = \frac{6\frac{h_F}{m_n}\cos\alpha_{Fen}}{\left(\frac{S_{Fn}}{m_n}\right)^2\cos\alpha_n}$$

S_{Fn}, h_F and α_{Fen} are to be calculated as follows:

$$S_{Fn} = m_n z_n \sin\left(\frac{\pi}{3} - \theta\right) + \sqrt{3} m_n \left(\frac{G}{\cos\theta} - \frac{\rho_{fp}}{m_n}\right)$$

$$G = \frac{\rho_{fp}}{m_n} - \frac{h_{fp}}{m_n} + x$$

$$\theta = \frac{2G}{z_n}\tan\theta - \frac{2}{z_n}\left(\frac{\pi}{2} - \frac{E}{m_n}\right) + \frac{\pi}{3}$$

$$E = \frac{\pi}{4} m_n - h_{fp}\tan\alpha_n + \frac{S_{Pr}}{\cos\alpha_n} - (1 - \sin\alpha_n)\frac{\rho_{fp}}{\cos\alpha_n}$$

S_{pr} is illustrated in **Fig. 7.2-1.** However, in cases where racks are without undercuts, S_{pr} is to be taken as zero.

$$h_F = \frac{m_n}{2}\left[(\cos\gamma_e - \sin\gamma_e\tan\alpha_{Fen})\frac{d_{en}}{m_n} - z_n\cos\left(\frac{\pi}{3} - \theta\right) - \frac{G}{\cos\theta} + \frac{\rho_{fp}}{m_n}\right]$$

$$\alpha_{Fen} = \alpha_{en} - \gamma_e$$

$$\gamma_e = \frac{0.5\pi + 2x\tan\alpha_n}{z_n} + \mathrm{inv}\alpha_n - \mathrm{inv}\alpha_{en}$$

$$\alpha_{en} = \arccos\left(\frac{d_{bn}}{d_{en}}\right)$$

$$d_{en} = 2\frac{z}{|z|}\left\{\left[\sqrt{\left(\frac{d_{an}}{2}\right)^2 - \left(\frac{d_{bn}}{2}\right)^2} - \frac{\pi d\cos\beta\cos\alpha_n}{|z|}(\varepsilon_{\alpha n} - 1)\right]^2 + \left(\frac{d_{bn}}{2}\right)^2\right\}^{\frac{1}{2}}$$

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

$$\beta_b = \arccos\sqrt{1 - \sin^2\beta\cos^2\alpha_n}$$

$$d_{b_n} = d_n\cos\alpha_n$$

$$d_n = m_n z_n$$

$$d_{a_n} = d_n + d_a - d$$

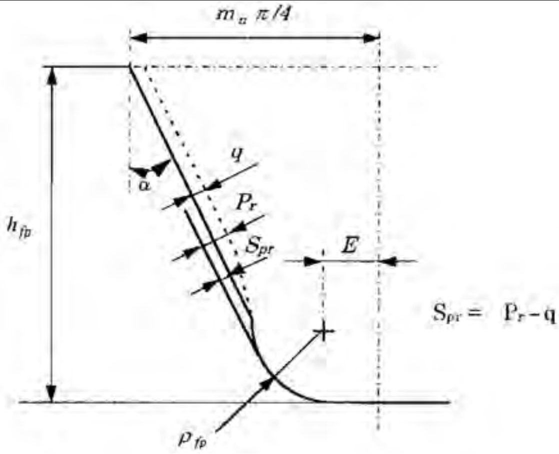
In the case of internal gears, the following coefficients used for determining the form factor are to be calculated as follows:

$$S_{Fn2} = m_n z_n \sin\left(\frac{\pi}{6} - \theta\right) + m_n\left(\frac{G}{\cos\theta} - \frac{\rho_{fp}}{m_n}\right)$$

$$h_{Fn2} = \frac{m_n}{2}\left[(\cos\gamma_e - \sin\gamma_e\tan\alpha_{Fen})\frac{d_{en}}{m_n} - z_n\cos\left(\frac{\pi}{6} - \theta\right) - \sqrt{3}\left(\frac{G}{\cos\theta} - \frac{\rho_{fp2}}{m_n}\right)\right]$$

$$d_{en2} = 2\frac{z}{|z|}\left\{\left[\sqrt{\left(\frac{d_{an}}{2}\right)^2 - \left(\frac{d_{bn}}{2}\right)^2} - \frac{\pi d_2 \cos\beta \cos\alpha_n}{|z|}(\varepsilon_{\alpha n} - 1)\right]^2 + \left(\frac{d_{bn2}}{2}\right)^2\right\}^{\frac{1}{2}}$$

Fig. 7.2-1 Dimensions and Profile of Basic Rack



2 Stress Correction Factor, Y_S

The stress correction factor, Y_S, is used to convert nominal bending stress into local tooth root stress, taking into account that not only bending stresses arise at roots. Y_S applies to those load applications at the outer points of single tooth pair contacts. Y_S is to be determined separately for pinions and for wheels. The value for Y_S is to be determined as follows (in the effective range: $1 \leq q_S < 8$).

$$Y_s = \left(1.2 + 0.13\frac{S_{Fn}}{h_F}\right)q_s^L$$

$$L = \frac{1}{1.21 + 2.3\frac{h_F}{S_{Fn}}}$$

where

However, since q_S is a notch parameter, it is to be calculated as follows:

$$q_S = \frac{S_{Fn}}{2\rho_F}$$

However, since ρ_F is the root fillet radius in the critical section (mm), it is to be calculated as follows:

$$\rho_F = \rho_{fp} + \frac{2m_n G^2}{\cos\theta(z_n\cos^2\theta - 2G)}$$

For the calculation of ρ_F, the procedure outlined in the reference standard *ISO* 6336-3:2019 is to be used.

3 Helix Angle Factor for Bending Stress, Y_β

The helix angle factor for bending stress, Y_β, is used to convert the stress calculated for point loaded cantilever beams representing gear teeth to the stress induced by loads along oblique load lines into cantilever plates which represent helical gear teeth. The value for Y_β can be calculated as follows:

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

where β is the reference helix angle in degree.
However, in cases where $\beta > 30°$ then $\beta = 30°$; and,
Also, in cases where $\varepsilon_\beta > 1$ then ε_β is to be taken as 1.0.

4 Rim thickness factor, Y_B

The rim thickness factor, Y_B, is a simplified factor used to de-rate thin rimmed gears. For critically loaded applications, this method is to be replaced by a more comprehensive analysis. Factor Y_B is to be determined as follows:

(1) For external gears:

$$Y_B = 1.6 \cdot \ln\left(2.242 \frac{h}{s_R}\right) \qquad (0.5 < s_R/h < 1.2)$$

$$= 1 \qquad (s_R/h \geq 1.2)$$

s_R : rim thickness of external gears (*mm*)
h : tooth height (*mm*)
The case $s_R/h \leq 0.5$ is to be avoided.

(2) For internal gears:

$$Y_B = 1.15 \cdot \ln\left(8.324 \frac{m_n}{s_R}\right) \qquad (1.75 < s_R/m_n < 3.5)$$

$$= 1 \qquad (s_R/m_n \geq 3.5)$$

s_R : rim thickness of internal gears (*mm*)
The case $s_R/m_n \leq 1.75$ is to be avoided.

5 Deep tooth factor, Y_{DT}

The deep tooth factor, Y_{DT}, adjusts the tooth root stress to take into account high precision gears and contact ratios within the range of virtual contact ratio $2.05 \leq \varepsilon_{\alpha n} \leq 2.5$.

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

Factor Y_{DT} is to be determined as follows:

$Y_{DT} = 0.7$(*ISO* accuracy grade ≤ 4 and $\varepsilon_{\alpha n} > 2.5$)

$Y_{DT} = 2.366 - 0.666\,\varepsilon_{\alpha n}$(*ISO* accuracy grade ≤ 4 and $2.05 < \varepsilon_{\alpha n} \leq 2.5$)

$Y_{DT} = 1.0$(In all other cases)

1.7.3 Permissible Tooth Root Bending Stress, σ_{FP}

1 Permissible tooth root bending stress σ_{FP} is to be calculated as follows:

$$\sigma_{FP} = \frac{\sigma_{FE} Y_d Y_N Y_{\delta relT} Y_{RrelT} Y_X}{S_F}$$

2 Bending Endurance Limit, σ_{FE}

For a given material, σ_{FE} is the local tooth root stress which can be permanently endured. According to the reference standard *ISO* 6336-5:2016, the number of 3×10^6 cycles may be regarded as the beginning of the endurance limit. σ_{FE} is defined as unidirectional pulsating stress with a minimum stress zero (neglecting any residual stresses due to heat treatment). Other conditions such as alternating stress or prestressing etc. are covered by the design factor Y_d. σ_{FE} values are to correspond to a failure probability of 1 % or less. These endurance limits mainly depends on the following:

(1) Material composition, cleanliness and defects;

(2) Mechanical properties;
(3) Residual stresses;
(4) Hardening process, depth of hardened zone, hardness gradient
(5) Material structure (forged, rolled bar, cast)

The value for σ_{FE} is to be calculated as follows:

$\sigma_{FE} = 2\sigma_{Flim}$

Values for σ_{Flim} are given in **Table 7.3-1.** However, for materials having enough data showing their higher endurance limit, values larger than those given in the table may be allowed by the Society in consideration of the factors **(1)** through **(5)** mentioned above.

3 Design Factor, Y_d

The design factor, Y_d, takes into account the influence of load reversing and shrink fit prestressing on tooth root strength, relative to tooth root strength with unidirectional loads as defined for σ_{FE}. Y_d for load reversing is to be determined as follows:

Y_d = 1.00 (In general cases)
= 0.90 (In the case of gears with occasional part loads in reversed directions, such as the main gears in reversing gearboxes)
= 0.70 (In the case of idler gears)

4 Life Factor for Bending Stress, Y_N

The life factor for bending stress, Y_N, accounts for the higher tooth root bending stress permissible in cases where limited life is required. Values greater than 1.0 will be considered by the Society on a case-by-case basis.

This factor mainly depends on the following:

(1) Material and heat treatment;
(2) Number of load cycles (service life);
(3) Influence factors($Y_{\delta relT}$, Y_{RrelT}, Y_X).

Y_N is to be determined according to Method B outlined in the reference standard *ISO* 6336-3:2019.

5 Relative Notch Sensitivity Factor, $Y_{\delta relT}$

The relative notch sensitivity factor, $Y_{\delta relT}$, indicates the extent of the influence of concentrated stress on fatigue endurance limits. This factor mainly depends on materials and relative stress gradients. This factor is to be calculated as follows:

$$Y_{\delta relT} = \frac{1 + \sqrt{0.2\rho'(1 + 2q_s)}}{1 + \sqrt{1.2\rho'}}$$

q_s : notch parameter
ρ' : slip-layer thickness (*mm*)

However, the values to be used for ρ' are those given in **Table 7.3-2**.

6 Relative Surface Factor, Y_{RrelT}

The relative surface factor, Y_{RrelT}, takes into account the influence of surface conditions in tooth root fillets on root strength and mainly depends on peak to valley surface roughness. The value for Y_{RrelT} is to be determined as shown in **Table 7.3-3**.

7 Size Factor for Bending Stress, Y_X

The size factor for bending stress, Y_X, takes into account decreases of strength with increasing size. This factor mainly depends on the following:

(1) Material and heat treatment;
(2) Tooth and gear dimensions;
(3) Ratio of case depth to tooth size.

The value for Y_X is to be determined as shown in **Table 7.3-4**.

8 Safety Factor for Tooth Root Bending Stress, S_F

The safety factor for tooth root bending stress, S_F, is to be taken as follows:

(1) 1.55 for main propulsion gears;

(2) 1.40 for auxiliary gears;

In addition, in cases where the gears of duplicated independent propulsion or auxiliary machinery which are duplicated beyond that required for their respective classes, a reduced value may be used at the discretion of the Society.

Table 7.3-1 Values of $\sigma_{F\lim}$ (N/mm^2)

Steel type	σ_{Flim}
Normalized structural steels	0.45*HB*+70
Through hardening carbon steels	0.25*HB*+160
Through hardening alloy steels	0.45*HB*+180
Induction hardened alloys	0.14*HV*+285; however, 365 for *HV*>570
Nitrided alloys	365
Soft nitrided alloys	0.66*HV*+88; however, 385 for *HV*>450
Nitrided steels	420
Carburized hardened alloys	465; however, 500 for *HRC*>30 in core

HB: Brinell Hardness; *HV*: Vickers Hardness; *HRC*: C Scale Rockwell Hardness

Table 7.3-2 Values of ρ' (mm)

Materials	ρ'
Nitriding steels, surface or through hardened	0.1005
Steels having yield strength about 300 N/mm^2	0.0833
Steels having yield strength about 400 N/mm^2	0.0445
Through hardened steels having yield strength about 500 N/mm^2	0.0281
Through hardened steels having yield strength 600 N/mm^2	0.0194
Through hardened steels having $\sigma_{0.2}$ about 800 N/mm^2	0.0064
Through hardened steels having $\sigma_{0.2}$ about 1000 N/mm^2	0.0014
Surface hardening steels, surface hardened	0.0030

Table 7.3-3 Values of Relative Surface Factor, Y_{RrelT}

Materials	Y_{RrelT}
Case hardened steels, Through hardened steels ($\sigma_B \geq 800$ N/mm^2)	1.120 (for R_Z <1) $1.674 - 0.529(R_Z + 1)^{0.1}$ (for 1≤ $R_Z \leq 40$)
Normalized steels ($\sigma_B < 800$ N/mm^2)	1.070 (for R_Z <1) $5.306 - 4.203(R_Z + 1)^{0.01}$ (for 1≤ $R_Z \leq 40$)
Nitriding steels	1.025 (for R_Z <1) $4.299 - 3.259(R_Z + 1)^{0.0058}$(for1≤ $R_Z \leq$40)

Notes

1: R_Z mean peak to valley roughness of tooth root fillets (μm).

2: This method is only valid when scratches or similar defects deeper than 2 R_Z do not exist.

3: If the roughness stated is an arithmetic mean roughness, i.e. R_a value, the approximation R_Z=6R_a may be applied.

Table 7.3-4 Size Factor for Bending Stress, Y_X

m_n	Materials	Y_X
$m_n \leqq 5$	General	1.00
$5 < m_n < 30$	Normalized and through hardened steels	$1.03 - 0.006m_n$
$m_n \geqq 30$		0.85
$5 < m_n < 25$	Surface hardened steels	$1.05 - 0.010m_n$
$m_n \geqq 25$		0.80

"Rules for the survey and construction of inland waterway ships" has been partly amended as follows:

Part 7 MACHINERY INSTALLATIONS

Chapter 3 POWER TRANSMISSION SYSTEMS

3.2 Materials and Construction

3.2.1 Materials*

Sub-paragraph -1 has been amended as follows.

1 Materials used for the following components (hereinafter referred to as "the principal components of the power transmission system") are to comply with the requirements in **Part K of the Rules for the Survey and Construction of Steel Ships**.

(1) Power transmission shafts (including power take-off(PTO) shafts) and gears

(2) Power transmission parts of couplings

(3) Power transmission parts of clutches

(4) Coupling bolts

2 (Omitted)

3.3 Strength of Gears

Paragraph 3.3.1 has been amended as follows.

3.3.1 Application*

The requirements in **3.3** apply to external tooth cylindrical gears having an involute tooth profile. All other gears are to be as deemed appropriate by the Society. In addition enclosed gear strength calculations are to be in accordance with **Annex 5.3.1 "Calculation of Strength of Enclosed Gears", Part D of the Rules for the Survey and Construction of Steel Ships.**

Paragraph 3.3.5 has been amended as follows.

3.3.5 Detailed Evaluation for Strength*

Special consideration will be given to the gears, notwithstanding the requirements in **3.3.3** and **3.3.4**, provided that detailed data and calculations on their strength are submitted to the Society and considered appropriate. In addition, the wording "detailed data and calculations on their strength" means calculations based on **Annex 5.3.1 "Calculation of Strength of Enclosed Gears", Part D of the Rules for the Survey and Construction of Steel Ships.**

"Guidance for the survey and construction of steel ships" has been partly amended as follows:

Part D MACHINERY INSTALLATIONS

D5 POWER TRANSMISSION SYSTEMS

D5.2 Materials and Construction

Paragraph D5.2.3 has been added as follows.

D5.2.3 General Construction of Gearings

The words "enough" and "sufficient" referred to in **5.2.3, Part D of the Rules** mean being designed in accordance with national or international standards such as *JIS*.

D5.3 Strength of Gears

Paragraph D5.3.1 has been amended as follows.

D5.3.1 Application

In the case of bevel gears, the wording "deemed appropriate by the Society" in **5.3.1, Part D of the Rules** means as follows:

(1) (Omitted)

(2) Strength of the interior of gear teeth

The Vickers hardness (*HV*) of the interior of gear teeth is not to be less than the value calculated by the following formula. However, this requirement does not apply to bevel gears for which the tip diameter (outer end) is smaller than 1,100 *mm*:

If $\frac{z}{w} < 0.79$ then $\frac{z}{w}$ is to be taken as 0.79.

$$HV = 1.11 S_H p \left[\frac{z}{w} - \frac{\left(\frac{z}{w}\right)^2}{\sqrt{1 + \left(\frac{z}{w}\right)^2}} \right]$$

HV : Vickers hardness

S_H : Safety factor for contact stress, is to comply with the requirements in **Annex ~~D5.3.5~~5.3.1 "~~GUIDANCE FOR~~ Calculation of Strength of Enclosed Gears" 1.6.3-9, Part D of the Rules.**

p : Real hertzian stress (*MPa*). The upper limit of the value of p used in this calculation is to be 1,500 *MPa*.

$p = AS_c$

S_c: Contact stress (*MPa*), to be calculated according to *ISO* 10300 standards.

A: If S_c is calculated according to *ISO* 10300 standards, then the coefficients are to be determined, in consideration of analysis results, by the Society on a case by case basis. In addition, if S_c is calculated according to *ISO* 10300 standards, A is to be taken as 1.32

w : Half the hertzian contact width (*mm*), to be calculated by the following formula:

$$w = \frac{p\rho_C}{56300}$$

$$\rho_C = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}$$

$\rho_1 = 0.5 d_{vn1} \sin\alpha_n$

$\rho_2 = 0.5 d_{vn2} \sin\alpha_n$

$$d_{vn1} = d_{m1} \frac{\sqrt{1+u^2}}{u} \frac{1}{\cos^2\beta_{vb}}$$

d_{m1} : Mean pitch diameter of pinion（*mm*）

u : Gear ratio

$\beta_{vb} = \arcsin(\sin\beta_m \cos\alpha_n)$

β_m : Mean spiral angle

α_n : Normal pressure angle

$d_{vn2} = u^2 d_{vn1}$

z : Depth from teeth surface to evaluation point (*mm*)

Paragraph D5.3.5 has been deleted.

~~**D5.3.5 Detailed Evaluation for Strength**~~

~~It is acceptable that the bending and surface strength of gears are calculated based on **Annex D5.3.5 "GUIDANCE FOR CALCULATION OF STRENGTH OF GEARS"**.~~

Paragraph D5.4 has been added as follows.

D5.4 Gear Shafts and Flexible Shafts

D5.4.1 Gear Shafts

The word "sufficient" in **5.4.1-1(2), Part D of the Rules** means being designed in accordance with national or international standards such as *JIS*.

Paragraph D5.4.3 has been added as follows.

D5.4.3 Couplings and Coupling Bolts

The word "sufficient" in **5.4.3, Part D of the Rules** means being designed in accordance with national or international standards such as *JIS*.

Annex D5.3.5 has been deleted.

~~Annex D5.3.5 GUIDANCE FOR CALCULATION OF STRENGTH OF GEARS~~

~~(Omitted)~~

"Guidance for high speed craft" has been partly amended as follows:

Part 9 MACHINERY INSTALLATIONS

Chapter 4 has been added as follows.

Chapter 4 POWER TRANSMISSION SYSTEMS

4.1 General

4.1.4 General Construction of Gears

The word "sufficient" in **4.1.4, Part 9 of the Rules** means being designed in accordance with national or international standards such as *JIS*.

"Guidance for the survey and construction of inland waterway ships" has been partly amended as follows:

Part 7 MACHINERY INSTALLATIONS

Chapter 3 POWER TRANSMISSION SYSTEMS

3.2 Materials and Construction

Paragraph 3.2.3 has been added as follows.

3.2.3 General Construction of Gears

The words "enough" and "sufficient" in **3.2.3, Part 7 of the Rules** mean being designed in accordance with national or international standards such as *JIS*.

3.3 Strength of Gears

Paragraph 3.3.1 has been amended as follows.

3.3.1 Application

In the case of bevel gears, the wording "deemed appropriate by the Society" in **3.3.1, Part 7 of the Rules** means as follows:

(1) The bending strength at the root sections of gear teeth and limiting tooth surface strength are to be according to *ISO* standards or as deemed appropriate by the Society.

(2) Evaluation of the strength of the interior of gear teeth may be required where deemed necessary by the Society. In such cases, the Vickers hardness (*HV*) of the interior of gear teeth is not to be less than the value obtained from the following formula. However, this requirement does not apply to bevel gears for which the tip diameter (outer end) is smaller than 1,100 *mm*:

If $\frac{z}{w} < 0.79$ then $\frac{z}{w}$ is to be taken as 0.79.

$$HV = 1.11 S_H p \left[\frac{z}{w} - \frac{\left(\frac{z}{w}\right)^2}{\sqrt{1 + \left(\frac{z}{w}\right)^2}} \right]$$

HV: Vickers hardness

S_H: Safety factor for contact stress is to comply with the requirements in **Annex ~~D5.3.5~~5.3.1 "~~GUIDANCE FOR~~ Calculation of Strength of Enclosed Gears" 1.6.3-9, Part D of the ~~Guidance~~Rules for the Survey and Construction of Steel Ships**.

p: Real hertzian stress (*MPa*). The upper limit of the value of p used in this calculation is to be 1,500 *MPa*.

$p = AS_c$

S_c: Contact stress (*MPa*), to be calculated according to *ISO 10300* standards.

A: If S_c is calculated according to *ISO 10300* standards, then the coefficients are to be determined, in consideration of analysis results, by the Society on a case by case basis. In addition, if S_c is calculated according to *ISO 10300* standards, A is to be taken as 1.32

w: Half the hertzian contact width (*mm*), to be calculated by the following formula:

$$w = \frac{p\rho_C}{56300}$$

$$\rho_C = \frac{\rho_1\rho_2}{\rho_1 + \rho_2}$$

$$\rho_1 = 0.5d_{vn1}\sin\alpha_n$$

$$\rho_2 = 0.5d_{vn2}\sin\alpha_n$$

$$d_{vn1} = d_{m1}\frac{\sqrt{1+u^2}}{u}\frac{1}{\cos^2\beta_{vb}}$$

d_{m1}: Mean pitch diameter of pinion（*mm*）

u: Gear ratio

β_{vb} = arcsin(sin$\beta_m$cosα_n)

β_m: Mean spiral angle

α_n: Normal pressure angle

$$d_{vn2} = u^2 d_{vn1}$$

z: Depth from teeth surface to evaluation point (*mm*)

Paragraph 3.3.5 has been deleted.

~~**3.3.5 Detailed Evaluation for Strength**~~

~~It is acceptable that the bending and surface strength of gears are calculated based on **Annex D5.3.5 "GUIDANCE FOR CALCULATION OF STRENGTH OF GEARS", Part D of the Guidance for the Survey and Construction of Steel Ships**.~~