GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D Machinery Installations

Guidance for the Survey and Construction of Steel Ships
Part D 2014 AMENDMENT NO.3

Notice No.68 19th December 2014
Resolved by Technical Committee on 29th July 2014

ClassNK
NIPPON KAIJI KYOKAI

Notice No.68 19th December 2014
AMENDMENT TO THE GUIDANCE 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

Amendment 3-1

Annex D1.1.3-3 GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF AZIMUTH THRUSTERS

1.2 Class Surveys

1.2.2 Periodical Surveys

Sub-paragraph -4 has been amended as follows.

4 Docking Surveys
(1) Visual inspections of steering columns, propeller pods and propellers (including bolt locking and other fastening arrangements) are to be carried out.
(2) Examinations on sealing devices for azimuth steering gears ~~and~~, propeller shafts and propeller blades are to be carried out.
(3) Measurements of the wear down of the bearing are to be carried out. (Except when roller bearings are used as bearings for propeller shafts)

EFFECTIVE DATE AND APPLICATION (Amendment 3-1)

1. The effective date of the amendments is 19 December 2014.
2. Notwithstanding the amendments to the Guidance, the current requirements may apply to the surveys for which the application is submitted to the Society before the effective date.

Amendment 3-2

Annex D5.3.5 GUIDANCE FOR CALCULATION OF STRENGTH OF GEARS

Section 1.2 has been amended as follows.

1.2 Symbols and Units

The main symbols introduced in this Guidance are listed below.

(a to Y_β are omitted)

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

Section 1.3 has been amended as follows.

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.

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

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

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

~~$d_b = d\cos\alpha_t = d_w\cos\alpha_{tw}$~~ $d_{b1,2} = d_{1,2}\cos\alpha_t$

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

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

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

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

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

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.

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

~~$v = \frac{d_1 n_1}{19099}$~~ $v = \frac{\pi\cdot d_{1,2}n_{1,2}}{60\cdot 10^3}$

Section 1.4 has been amended as follows.

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:

~~$T_{1,2} = 9549 \dfrac{P}{n_{1,2}}$~~ $T_{1,2} = \dfrac{30 \cdot 10^3 P}{\pi \cdot n_{1,2}}$

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

1.5 Loading Factors

Paragraph 1.5.1 has been amended as follows.

1.5.1 Application Factor, K_A

1 The application factor K_A accounts for dynamic overloads from source external to the gearing. The value 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:

(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. The application factor 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 (diesel engines with hydraulic or electromagnetic slip couplings)
= 1.30 (diesel engines with high elasticity couplings)
= 1.50 (diesel engines with other couplings)

However, in cases where vessels, on which reduction gear is being used, are receiving Ice Class Notation, <u>as required in **5.6, Part I of the Rules**</u> ~~the following value is to be substituted for the values given above for K_A~~.

~~$K_A = \dfrac{1.10}{1 + J_1 / J_h} \cdot \dfrac{M}{M_0}$~~

~~The values for J_1, J_h, M and M_0 are the same as those specified in **5.4.5, Part I of the Rules**.~~

(2) Auxiliary gears

K_A = 1.00 (electric motors, diesel engines with hydraulic or electromagnetic slip couplings)
= 1.20 (diesel engines with high elasticity couplings)
= 1.40 (diesel engines with other couplings)

Paragraph 1.5.3 has been amended as follows.

1.5.3 <u>Internal</u> ~~D~~<u>d</u>ynamic factor, K_V

1 The <u>internal</u> dynamic factor K_V accounts for those internally generated dynamic loads due to vibrations of pinions and wheels against each other. The value K_V is defined as the ratio between those maximum loads which dynamically act on tooth flanks and maximum externally applied loads

($F_t K_A K_\gamma$). This factor mainly depends on:

(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) Steel gears of heavy rims sections~~

~~(2) $\frac{F_t}{b} > 150 (N/mm)$~~

~~(3) $z_1 < 50$~~

(~~4~~1) Running speeds in the following subcritical ranges:

~~$\frac{vz_1}{100} < 14$ (In the case of helical gears)~~

~~$\frac{vz_1}{100} < 10$ (In the case of spur gears)~~ $\frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 10$ (*m/s*)

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

$\beta \le 30°$ (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 ~~$\frac{vz_1}{100} < 3$~~ $\frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 3$ (*m/s*), as well as to helical gears where $\beta > 30°$. In the case of gears other than those given above, the value of K_V is to be determined, in consideration of results of analyses, by the Society on a case by case basis.

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

$$\sout{K_V = 1 + \psi \frac{vz_1}{100}} \quad K_V = 1 + \left(\frac{K_1}{K_A \frac{F_t}{b}} + K_2 \right) \cdot \frac{v \cdot z_1}{100} K_3 \sqrt{\frac{u^2}{1+u^2}}$$

~~The values of ψ are given in the following **Table 5.3-1**~~

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 by following formula:

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

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

If $K_A \frac{F_t}{b}$ 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} : The values for helical gears specified in ~~**Table 5.3-1**~~ accordance with **(a)**

K_{V2} : The values for spur gears specified in ~~**Table 5.3-1**~~ 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 ψ~~

~~Type of gears~~	~~ISO grades of accuracy~~					
	~~3~~	~~4~~	~~5~~	~~6~~	~~7~~	~~8~~
~~Spur gears~~	~~0.022~~	~~0.030~~	~~0.043~~	~~0.062~~	~~0.092~~	~~0.125~~
~~Helical gears~~	~~0.0125~~	~~0.0165~~	~~0.0230~~	~~0.0330~~	~~0.0480~~	~~0.0700~~

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

1.6 Surface Strength

Paragraph 1.6.2 has been amended as follows.

1.6.2 Equations for Basic Contact Stress

(-1 is omitted)

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. The values for 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 given by the following expression, 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 given by the following formula, 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 given by the following expression, 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 given by the following formula, whichever is greater.

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

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

3 Zone Factor, Z_H

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

~~$Z_H = \sqrt{\frac{2\cos\beta_b \cos\alpha_{tw}}{\cos^2\alpha_t \sin\alpha_{tw}}}$~~ <u>$Z_H = \sqrt{\frac{2\cos\beta_b}{\cos^2\alpha_t \tan\alpha_{tw}}}$</u>

(-4 and -5 are omitted)

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:

~~$Z_\beta = \sqrt{\cos\beta}$~~ <u>$Z_\beta = \sqrt{\frac{1}{\cos\beta}}$</u>

where β is the reference helix angle.

Paragraph 1.6.3 has been amended as follows.

1.6.3 Permissible Contact Stress

(-1 is omitted)

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. The value of σ_{Hlim} can be regarded as the level of contact stress which the material will endure without pitting for at least ~~50×10^6~~ 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:

(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.

(-3 to -5 are omitted)

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:

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

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

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

where

R_{z1}, R_{z2} : Respective mean peak to valley roughness for pinions and wheels.

ρ_{red} : Relative radius of curvature. The value for ρ_{red} is to be calculated by following formula:

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

$\rho_{1.2} = 0.5 \cdot d_{b1\cdot2} \cdot \tan\alpha_{tw}$ (In cases where internal gears d_b are negative)

In cases where the roughness stated is ~~a~~ an arithmetic mean roughness, i.e. R_a value, 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: ~~Z_W is to be applied to soft gears only. This factor mainly depends on:~~

~~(1) Hardness of soft gears;~~

~~(2) Alloying elements of soft gears;~~

~~(3) Tooth flank roughness of the harder gears. The hardness ratio factor Z_W is to be calculated as follows:~~

~~$$Z_W = 1.2 - \frac{HB - 130}{1700} \quad (130 \le HB \le 470)$$~~

~~$$= 1.2 \quad (HB < 130)$$~~

~~$$= 1.0 \quad (HB > 470)$$~~

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

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

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

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

where

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

R_{zH} : equivalent roughness (μm)

$$R_{zH} = \frac{R_{z1} \cdot (10/\rho_{red})^{0.33} \cdot (R_{z1}/R_{z2})^{0.66}}{(v \cdot \nu_{40}/1500)^{0.33}}$$

(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.

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

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

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

$HB_{1,2}$: 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 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 and -9 are omitted)

1.7 Bending Strength

Paragraph 1.7.1 has been amended as follows.

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}.

$$\sigma_F = \frac{F_t Y_F Y_s Y_\beta K_A K_\gamma K_V K_{F\alpha} K_{F\beta}}{b m_n} \leq \sigma_{FP}$$

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

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° and reference helix angles up to 30°. In the case of larger pressure angles and larger helix angles, the method of calculation is to be given by the Society on a case by case basis.

1.7.2 Tooth Root Bending Stress for Pinion and Wheel

Sub-paragraph -1 has been amended as follows.

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, the value of S_{pr} is to be taken as zero.

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

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

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

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

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

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

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

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

$$d_n = m_n z_n$$

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

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

$$S_{Fn2} = \frac{m_n}{2}\left[\frac{\pi}{4} + \frac{h_{fP2} - \rho_{fp2}}{m_n}\tan\alpha_n + \frac{\rho_{fp2} - S_{pr}}{m_n\cos\alpha_n} - \frac{\sqrt{3}}{2}\frac{\rho_{fp2}}{m_n}\right]$$

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

$$h_{Fn2} = \frac{d_{en2} - d_{fn2}}{2} - m_n\tan\alpha_n\left[\frac{\pi}{4} + \left(\frac{h_{fp2}}{m_n} - \frac{d_{\theta n2} - d_{fn2}}{2m_n}\right)\tan\alpha_n\right] - \frac{\rho_{fp2}}{2}$$

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

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

~~$h_{fP2} = \frac{d_{n2} - d_{fn2}}{2}$~~

Sub-paragraphs -4 and -5 have been added as follows.

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 should be replaced by a more comprehensive analysis. Factor Y_B is to be determined as follows:

(1) For external gears:

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

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

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:

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

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

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 $\varepsilon_{\alpha n} > 2.5$)

$Y_{DT} = 2.366 - 0.666 \cdot \varepsilon_{\alpha n}$ (*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}

Sub-paragraph -5 has been amended as follows.

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:

~~$Y_{\delta\,relT} = 1.0$ (In cases where $1.5 < q_s < 4$)~~

~~$= \dfrac{1 + 0.45\sqrt{\rho'(1 + 2q_s)}}{1 + 1.1\sqrt{\rho'}}$ (In all other cases)~~

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

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**.

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

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**.

Table 7.3-3 has been amended as follows.

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

Materials	Y_{RreIT}
Case hardened steels, Through hardened steels ($\sigma_B \geq 800$ N/mm^2)	1.120 (for R_Z <1) ~~1.675-0.53~~ 1.674-0.529 (R_Z + 1)$^{0.1}$ (for 1≤ R_Z ≤40)
Normalized steels ($\sigma_B < 800$ N/mm^2)	1.070 (for R_Z <1) ~~5.3-4.2~~ 5.306-4.203 (R_Z + 1)$^{0.01}$ (for 1≤ R_Z ≤40)
Nitriding steels	1.025 (for R_Z <1) ~~4.3-3.26~~ 4.299-3.259 (R_Z + 1)$^{0.0058}$ (for 1≤ R_Z ≤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 ~~a~~ an arithmetic mean roughness, i.e. R_a value, the approximation $R_Z = 6 R_a$ may be applied.

EFFECTIVE DATE AND APPLICATION (Amendment 3-2)

1. The effective date of the amendments is 1 January 2015.

2. Notwithstanding the amendments to the Guidance, the current requirements apply to all gears previously approved by the Society prior to the effective date for which no failure has occurred, and no changes related to strength, such as the scantlings of the gear meshes, materials, etc. have been made.