INDIAN REGISTER OF SHIPPING

CLASSIFICATION NOTES

Marine Gears – Calculation of Load Capacity of Involute Parallel Axis Spur and Helical Gears

January 2017

Changes – in JANUARY 2017 Version

Classification Notes “Marine Gears – Calculation of Load Capacity of Involute Parallel Axis Spur and Helical Gears - January 2017”

Section / Clause	Subject/Amendments
1.2.1	Clause amended to provide clarification on the application of the classification note. It is clarified that the requirements of the classification note are applicable to, enclosed gears whose gear set is intended to transmit a maximum continuous power equal to, or greater than: • 220 kW for gears intended for main propulsion • 110 kW for gears intended for essential auxilary services

CLASSIFICATION NOTES

Marine Gears – Calculation of Load Capacity of Involute Parallel Axis Spur and Helical Gears

(This version of the classification note (Jan 2017) is applicable to any marine gear subject to approval and to any type approved marine gear from the date of the first renewal after 1 January 2017. For a Marine gear approved prior to 1 January 2017 where no failure has occurred and no changes in design/scantlings of the gear meshes or materials or declared load capacity data has taken place the requirements of this classification note (version Jan 2017) may be waived.)

January 2017

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Marine Gears – Calculation of Load Capacity of Involute Parallel Axis Spur and Helical Gears

Section 1

Basic Principles – Introduction and General Influence Factors

1.1 Introduction

1.1.1 The following definitions are mainly based on the ISO 6336 standard (hereinafter called “reference standard”) for the calculation of load capacity of spur and helical gears.

1.2 Scope and field of application

1.2.1 These requirements apply to enclosed gears, both intended for main propulsion and for essential auxiliary services, which accumulate a large number of load cycles (several millions), whose gear set is intended to transmit a maximum continuous power equal to, or greater than:

- 220 kW for gears intended for main propulsion
- 110 kW for gears intended for essential auxiliary services

Application of these requirements to the enclosed gears, whose gear set is intended to transmit a maximum continuous power less than those specified above will be specially considered.

1.2.2 The following requirements deal with the determination of load capacity of external and internal involute spur and helical gears, having parallel axis, with regard to surface durability (pitting) and tooth root bending strength and to this purpose the relevant basic equations are provided in Sec 2 and Sec 3.

1.2.3 The influence factors common to said equations in Sec 2 and Sec 3 are described in this section.

1.2.4 The others, introduced in connection with each basic equation, are described in the following Parts 2 and 3.

1.2.5 All influence factors are defined regarding their physical interpretation. Some of the influence factors are determined by the gear geometry or have been established by conventions. These factors are to be calculated in accordance with the equations provided. Other factors, which are approximations, may be calculated according to methods acceptable to IRS.

1.3 Symbols and units

1.3.1 The main symbols used are listed below.

1.3.2 Other symbols introduced in connection with the definition of influence factors are described in the appropriate sections.

SI units have been adopted.

a	centre distance	mm
b	common face width	mm
$b_{1,2}$	face width 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
F_{bt}	nominal tangential load on base cylinder in the transverse section	N
h	tooth depth	mm
m_n	normal module	mm
m_t	transverse module	mm
$n_{1,2}$	rotational speed of pinion, wheel	revs/min (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 velocity 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 ref. 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 ratio	
ε_γ	total contact ratio	

1.4 Geometrical definitions

1.4.1 For 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:

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

1.4.2 For external gears u is positive, for internal gears u is negative.

1.4.3 In the equation of surface durability b is the common face width on the pitch diameter.

1.4.4 In the equation of tooth root bending stress b_1 or b_2 are the face widths at the 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.

1.4.5 The common face width b may be used also in the equation of teeth root bending stress if significant crowning or end relief have been adopted.

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

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

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

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

$$\left.\begin{aligned} d_{w1} &= \frac{2a}{u+1} \\ d_{w2} &= \frac{2au}{u+1} \end{aligned}\right\} \quad \text{where} \qquad a = 0.5\left(d_{w1} + d_{w2}\right)$$

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

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

$$\text{inv}\, \alpha = \tan \alpha - \frac{\pi \alpha}{180}; \quad \alpha\ [°]$$

$$\operatorname{inv} \alpha_{tw} = \operatorname{inv} \alpha_t + 2 \tan \alpha_n \frac{x_1 + x_2}{z_1 + z_2} \quad \text{or} \quad \cos \alpha_{tw} = \frac{m_t (z_1 + z_2)}{2a} \cos \alpha_t$$

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

the positive sign is used for external gears, the negative sign for internal gears

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

for double helix, b is to be taken as the width of one helix

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

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

1.5 Nominal tangential load, F_t

1.5.1 The nominal tangential load, F_t, tangential to the reference cylinder and perpendicular to the relevant axial plane, is calculated directly from the maximum continuous power transmitted by the gear set by means of the following equations:

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

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

1.6 General influence factors

1.6.1 Application factor, K_A [1)]

a) The application factor, K_A, accounts for dynamic overloads from sources external to the gearing.
b) Where the vessel, on which the reduction gear is being used, is receiving an Ice Class notation, the Application Factor or the Nominal Tangential Force should be adjusted to reflect the ice load associated with the contemplated ice class notation.
c) K_A, for gears designed for infinite life is defined as the ratio between the maximum repetitive cyclic torque applied to the gear set and the nominal rated torque.
d) The nominal rated torque is defined by the rated power and speed and is the torque used in the rating calculations.
e) The factor mainly depends on:
   - characteristics of driving and driven machines;
   - ratio of masses;
   - type of couplings;

- operating conditions (overspeeds, changes in propeller load conditions, etc.).

When operating near a critical speed of the drive system, a careful analysis of conditions must be made.

f) The application factor, K_A, should be determined by measurements or by system analysis acceptable to the Society. Where a value determined in such a way cannot be supplied, the following values can be considered.

- Main propulsion

diesel engine with hydraulic or electromagnetic slip coupling	1.00
diesel engine with high elasticity coupling	1.30
diesel engine with other couplings	1.50

- Auxiliary gears

electric motor, diesel engine with hydraulic or electromagnetic slip coupling	1.00
diesel engine with high elasticity coupling	1.20
diesel engine with other couplings	1.40

1.6.2 Load sharing factor, K_γ

a) The load sharing factor, $K\gamma$ accounts for the maldistribution of load in multiple path transmissions (dual tandem, epicyclic, double helix, etc.)

b) K_γ is defined as the ratio between the maximum load through an actual path and the evenly shared load. The factor mainly depends on accuracy and flexibility of the branches.

c) The load sharing factor, K_γ, should be determined by measurements or by system analysis. Where a value determined in such a way cannot be supplied, the following values can be considered for epicyclic gears:

up to 3 planetary gears	1.00
4 planetary gears	1.20
5 planetary gears	1.30
6 planetary gears and over	1.40

1.6.3 Internal dynamic factor, K_v

a) The internal dynamic factor, K_v, accounts for internally generated dynamic loads due to vibrations of pinion and wheel against each other.

b) K_v is defined as the ratio between the maximum load which dynamically acts on the tooth flanks and the maximum externally applied load ($F_t K_A K_\gamma$).

c) The factor mainly depends on:

- transmission errors (depending on pitch and profile errors);
- masses of pinion and wheel;
- gear mesh stiffness variation as the gear teeth pass through the meshing cycle;
- transmitted load including application factor;
- pitch line velocity;
- dynamic unbalance of gears and shaft;
- shaft and bearing stiffnesses;
- damping characteristics of the gear system.

d) The method of calculation of internal dynamic factor, Kv, given herein may be applied only to cases where all the following conditions are satisfied:
- running velocity in the subcritical range, i.e.:

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

- spur gears $(\beta = 0°)$ and helical gears with $\beta \leq 30°$
- pinion with relatively low number of teeth, $z_1 < 50$
- solid disc wheels or heavy steel gear rim
- This method may be applied to all types of gears if $\frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 3 \text{ m/s}$, as well as to helical gears where $\beta > 30°$.

e) For gears other than the above, reference is to be made to Method B outlined in the reference standard ISO 6336-1.

f) For spur gears and for helical gears with overlap ratio $\varepsilon_\beta \geq 1$

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

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

Numerical values for the factor K_1 are to be as specified in the Table 1.6.3

Table 1.6.3 : Values of the factor K_1 for the calculation of K_v						
	K_1 ISO accuracy grades (See note)					
	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
Note: ISO accuracy grades according to ISO 1328. In case of mating gears with different accuracy grades, the grade corresponding to the lower accuracy should be used.						

For all accuracy grades the factor K_2 is to be in accordance with the following:

- for spur gears, K_2=0.0193
- for helical gears, K_2=0.0087

Factor K_3 is to be in accordance with the following:

If $\frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}} \le 0.2$ then $K_3 = 2.0$

If $\frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}} > 0.2$ then $K_3 = 2.071 - 0.357 \cdot \frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}}$

g) For helical gears with overlap ratio $\varepsilon\beta$<1 the value Kv is determined by linear interpolation between values determined for spur gears ($K_{v\alpha}$) and helical gears ($K_{v\beta}$) in accordance with:

$$K_v = K_{v\alpha} - \varepsilon_\beta \left(K_{v\alpha} - K_{v\beta}\right)$$

Where:

$K_{v\alpha}$ is the K_v value for spur gears, in accordance with f);
$K_{v\beta}$ is the K_v value for helical gears, in accordance with f).

1.6.4 Face load distribution factors, $K_{H\beta}$ and $K_{F\beta}$

a) The face load distribution factors, $K_{H\beta}$ for contact stress, $K_{F\beta}$ for tooth root bending stress, account for the effects of non-uniform distribution of load across the face width.

b) $K_{H\beta}$ is defined as follows:

$$K_{H\beta} = \frac{\text{maximum load per unit face width}}{\text{mean load per unit face width}}$$

c) $K_{F\beta}$ is defined as follows:

$$K_{F\beta} = \frac{\text{maximum bending stress at tooth root per unit face width}}{\text{mean bending stress at tooth roo per unit face width}}$$

d) The mean bending stress at tooth root relates to the considered face width b_1 resp. b_2. $K_{F\beta}$ can be expressed as a function of the factor $K_{H\beta}$.

e) The factors $K_{H\beta}$ *and* $K_{F\beta}$ mainly depend on:

- gear tooth manufacturing accuracy;
- errors in mounting due to bore errors;
- bearing clearances;
- wheel and pinion shaft alignment errors;
- elastic deflections of gear elements, shafts, bearings, housing and foundations which support the gear elements;
- thermal expansion and distortion due to operating temperature;
- compensating design elements (tooth crowning, end relief, etc.).

f) The face load distribution factors, $K_{H\beta}$ for contact stress, and $K_{F\beta}$ for tooth root bending stress, are to be determined according to the Method C outlined in the reference standard ISO 6336-1. Alternative methods acceptable to IRS may be applied.

- In case the hardest contact is at the end of the face width $K_{F\beta}$ is given by the following equations:

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

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

(b/h) = face width/tooth height ratio, the minimum of b_1/h_1 or b_2/h_2. For double helical gears, the face width of only one helix is to be used. When $b/h<3$ the value $b/h=3$ is to be used.

- In case of gears where the ends of the face width are lightly loaded or unloaded (end relief or crowning):

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

1.6.5 Transverse load distribution factors, $K_{H\alpha}$ and $K_{F\alpha}$

a) 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.

b) The factors $K_{H\alpha}$ *and* $K_{F\alpha}$ mainly depend on:
- total mesh stiffness;
- total tangential load F_t, K_A, K_γ, K_v, $K_{H\beta}$;
- base pitch error;
- tip relief;
- running-in allowances.

c) The transverse load distribution factors, $K_{H\alpha}$ for contact stress and $K_{F\alpha}$ for tooth root bending stress, are to be determined according to Method B outlined in the reference standard ISO 6336-1.

Section 2

Surface Durability (Pitting)

2.1 Scope and general remarks

2.1.1 The criterion for surface durability is based on the Hertz pressure on the operating pitch point or at the inner point of single pair contact. The contact stress σ_H must be equal to or less than the permissible contact stress σ_{HP}.

2.2 Basic equations

2.2.1 Contact stress

$$\sigma_H = \sigma_{H0}\sqrt{K_A \cdot K_\gamma \cdot K_v \cdot K_{H\alpha} \cdot K_{H\beta}} \le \sigma_{HP}$$

where:

σ_{H0} = basic value of contact stress for pinion and wheel

$$\sigma_{H0} = Z_B \cdot Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \sqrt{\frac{F_t}{d_1 \cdot b}\frac{(u+1)}{u}}$$ for pinion

$$\sigma_{H0} = Z_D \cdot Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \sqrt{\frac{F_t}{d_1 \cdot b}\frac{(u+1)}{u}}$$ for wheel

where:

Z_B = single pair tooth contact factor for pinion (see 2.3)

Z_D = single pair tooth contact factor for wheel (see 2.3)

Z_H = zone factor (see 2.4)

Z_E = elasticity factor (see 2.5)

Z_ε = contact ratio factor (see 2.6)

Z_β = helix angle factor (see 2.7)

F_t = nominal tangential load at reference cylinder in the transverse section (see Section 1)

b = common face width

d_1 = reference diameter of pinion

u = gear ratio
(for external gears u is positive, for internal gears u is negative)

Regarding factors K_A, K_γ, K_v, $K_{H\alpha}$ and $K_{H\beta}$, see Section 1.

2.2.2 Permissible contact stress

2.2.2.1 The permissible contact stress σ_{HP} is to be evaluated separately for pinion and wheel:

$$\sigma_{HP} = \frac{\sigma_{H\lim} \cdot Z_N}{S_H} \cdot Z_L \cdot Z_v \cdot Z_R \cdot Z_W \cdot Z_X$$

where:

σ_{Hlim}	=	endurance limit for contact stress	(see 2.8)
Z_N	=	life factor for contact stress	(see 2.9)
Z_L	=	lubrication factor	(see 2.10)
Z_v	=	velocity factor	(see 2.10)
Z_R	=	roughness factor	(see 2.10)
Z_W	=	hardness ratio factor	(see 2.11)
Z_X	=	size factor for contact stress	(see 2.12)
S_H	=	safety factor for contact stress	(see 2.13)

2.3 Single pair tooth contact factors, Z_B and Z_D

2.3.1 The single pair tooth contact factors, Z_B for pinion and Z_D for wheel, account for the influence of the tooth flank curvature on contact stresses at the inner point of single pair contact in relation to Z_H.

2.3.2 The factors transform the contact stresses determined at the pitch point to contact stresses considering the flank curvature at the inner point of single pair contact.

2.3.3 The single pair tooth contact factors, Z_B for pinions and Z_D for wheels, are to be determined as follows:

For spur gears, ε_β=0

$Z_B = M_1$ or 1 whichever is the larger value

$Z_D = M_2$ or 1 whichever is the larger value

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

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

For helical gears when $\varepsilon_\beta \geq 1$

$Z_B = 1$
$Z_D = 1$

2.3.4 For helical gears when $\varepsilon_\beta < 1$ the values of Z_B and Z_D are determined by linear interpolation between Z_B and Z_D for spur gears and Z_B and Z_D for helical gears having $\varepsilon_\beta \geq 1$.

Thus:

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

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

For internal gears, Z_D shall be taken as equal to 1.

2.4 Zone factor, Z_H

2.4.1 The zone factor, Z_H, accounts for the influence on the Hertzian pressure of tooth flank curvature at pitch point and transforms the tangential load at the reference cylinder to the normal load at the pitch cylinder.

2.4.2 The zone factor, Z_H, is to be calculated as follows:

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

2.5 Elasticity factor, Z_E

2.5.1 The elasticity factor, Z_E, accounts for the influence of the material properties *E* (modulus of elasticity) and *v* (Poisson’s ratio) on the contact stress.

2.5.2 The elasticity factor, Z_E, for steel gears (E= 206 000 N/mm^2, v= 0.3) is equal to:

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

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

2.6 Contact ratio factor, Z_ε

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

2.6.2 The contact ratio factor, Z_ε, is to be calculated as follows:

Spur gears:

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

Helical gears:

- for $\varepsilon_\beta < 1$

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

- for $\varepsilon_\beta \geq 1$

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

2.7 Helix angle factor, Z_β

2.7.1 The helix angle factor, Z_β, accounts for the influence of helix angle on surface durability, allowing for such variables as the distribution of load along the lines of contact. Z_β is dependent only on the helix angle.

2.7.2 The helix angle factor, Z_β, is to be calculated as follows:

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

Where β is the reference helix angle.

2.8 Endurance limit for contact stress, σ_{Hlim}

2.8.1 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 5×10^7 load cycles.

2.8.2 For this purpose, pitting is defined by:

- for not surface hardened gears:
  pitted area > 2% of total active flank area
- for surface hardened gears:
  pitted area > 0,5% of total active flank area, or
  > 4% of one particular tooth flank area.

2.8.3 The σ_{Hlim} values are to correspond to a failure probability of 1% or less.

2.8.4 The endurance limit mainly depends on:
- material composition, cleanliness and defects;
- mechanical properties;
- residual stresses;
- hardening process, depth of hardened zone, hardness gradient;
- material structure (forged, rolled bar, cast).

2.8.5 The endurance limit for contact stress σ_{Hlim}, is to be determined, in general, making reference to values indicated in the standard ISO 6336-5, for material quality MQ.

2.9 Life factor, Z_N

2.9.1 The life factor Z_N, accounts for the higher permissible contact stress in case a limited life (number of cycles) is required.

2.9.2 The factor mainly depends on:
- material and heat treatment;
- number of cycles;
- influence factors (Z_R, Z_v, Z_L, Z_W, Z_X).

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

2.10 Influence factors of lubrication film on contact stress, Z_L, Z_v and Z_R

2.10.1 The lubricant factor, Z_L, accounts for the influence of the type of lubricant and its viscosity. The velocity factor, Z_v, accounts for the influence of the pitch line velocity. The roughness factor, Z_R, accounts for the influence of the surface roughness on the surface endurance capacity.

2.10.2 The factors may be determined for the softer material where gear pairs are of different hardness.

2.10.3 The factors mainly depend on:
- viscosity of lubricant in the contact zone;
- the sum of the instantaneous velocities of the tooth surfaces;
- load;
- relative radius of curvature at the pitch point;

- surface roughness of teeth flanks;
- hardness of pinion and gear.

2.10.4 The lubricant factor, Z_L, the velocity factor, Z_V, and the roughness factor Z_R are to be calculated as follows:

a) Lubricant factor, Z_L

The factor, Z_L, is to be calculated from the following equation:

$$Z_L = C_{ZL} + \frac{4(1 - C_{ZL})}{\left(1.2 + \frac{134}{v_{40}}\right)^2}$$

In the range 850 N/mm^2 ≤ $\sigma_{H\lim}$ ≤ 1200 N/mm^2, C_{ZL} is to be calculated as follows:

$$C_{ZL} = \left(0.08 \frac{\sigma_{H\lim} - 850}{350}\right) + 0.83$$

If σ_{Hlim} < 850 N/mm^2, take C_{ZL} = 0.83

If σ_{Hlim} > 1200 N/mm^2, take C_{ZL} = 0.91

Where:

v_{40} = nominal kinematic viscosity of the oil at 40°C, mm^2/s

b) Velocity factor, Z_v

The velocity factor, Z_v, is to be calculated from the following equations:

$$Z_v = C_{ZV} + \frac{2(1 - C_{ZV})}{\sqrt{0.8 + \frac{32}{v}}}$$

In the range 850 N/mm^2 ≤ σ_{Hlim} ≤ 1200 N/mm^2, C_{ZV} is to be calculated as follows:

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

c) Roughness factor, Z_R

The roughness factor, Z_R, is to be calculated from the following equations~~s~~:

$$Z_R = \left(\frac{3}{R_{z10}}\right)^{C_{ZR}}$$

Where:

$$R_z = \frac{R_{z1} + R_{z2}}{2}$$

The peak-to-valley roughness determined for the pinion R_{z1} and for the wheel R_{z2} are mean values for the peak-to-valley roughness R_z measured on several tooth flanks (R_z as defined in the reference standard ISO 6336-2).

$$R_{z10} = R_z \sqrt[3]{\frac{10}{\rho_{red}}}$$

relative radius of curvature:

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

Wherein:

$$\rho_{1,2} = 0.5 \cdot d_{b1,2} \cdot \tan \alpha_{tw}$$

(also for internal gears, d_b negative sign)

If the roughness stated is an arithmetic mean roughness, i.e. R_a value (=CLA value) (=AA value) the following approximate relationship can be applied:

$$R_a = CLA = AA = R_z / 6$$

In the range 850 N/mm^2 ≤ σ_{Hlim} ≤ 1200 N/mm^2, C_{ZR} is to be calculated as follows:

$$C_{ZR} = 0.32 - 0.0002 \cdot \sigma_{H\lim}$$

If σ_{Hlim} < 850 N/mm^2, take C_{ZR} = 0.150

If σ_{Hlim} > 1200 N/mm^2, take C_{ZR} = 0.080

2.11 Hardness ratio factor, Z_W

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

a) Surface-hardened pinion with through-hardened wheel

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

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

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

Where:

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

R_{zH} = equivalent roughness, μm

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

ρ_{red} = relative radius of curvature (see clause 2.10 c)

b) 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.

If $HB_1/HB_2 < 1.2$ $$Z_W = 1$$

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

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

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 = 1.0 is to be used.

2.12 Size factor, Z_X

2.12.1 The size factor, Z_X, accounts for the influence of tooth dimensions on permissible contact stress and reflects the non-uniformity of material properties.

2.12.2 The factor mainly depends on:

- material and heat treatment;
- tooth and gear dimensions;
- ratio of case depth to tooth size;
- ratio of case depth to equivalent radius of curvature.

2.12.3 For through-hardened gears and for surface-hardened gears with adequate casedepth relative to tooth size and radius of relative curvature $Z_X = 1$. When the casedepth is relatively shallow then a smaller value of Z_X should be chosen.

2.13 Safety factor for contact stress, S_H

2.13.1 The safety factor for contact stress, S_H, can be assumed by the Society taking into account the type of application.

The following guidance values can be adopted:

- Main propulsion gears: 1.20 to 1.40
- Auxiliary gears: 1.15 to 1.20

2.13.2 For gearing of duplicated independent propulsion or auxiliary machinery, duplicated beyond that required for class, a reduced value can be assumed at the discretion of the IRS.

Section 3

Tooth Root Bending Strength

3.1 Scope and general remarks

3.1.1 The criterion for tooth root bending strength is the permissible limit of local tensile strength in the root fillet. The root stress σ_F and the permissible root stress σ_{FP} shall be calculated separately for the pinion and the wheel.

3.1.2 σ_F must not exceed σ_{FP}.

3.1.3 The following formulae and definitions apply to gears having rim thickness greater than 3.5m_n.

3.1.4 The result of rating calculations made by following this method are acceptable for normal pressure angles up to 25° and reference helix angles up to 30°.

3.1.5 For larger pressure angles and large helix angles, the calculated results should be confirmed by experience as by Method A of the reference standard ISO 6336-3.

3.2 Basic equations

3.2.1 Tooth root bending stress for pinion and wheel

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

where:

Y_F = tooth form factor (see clause 3.3)
Y_S = stress correction factor (see clause 3.4)
Y_β = helix angle factor (see clause 3.5)
Y_B = rim thickness factor (see clause 3.6)
Y_{DT} = deep tooth factor (see clause 3.7)
$F_t, K_A, K_\gamma, K_v, K_{F\alpha}, K_{F\beta}$ (see Sec 1)
b (see Sec 1, clause 1.4)
m_n (see Sec 1, clause 1.3)

3.2.2 Permissible tooth root bending stress for pinion and wheel

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

where:

σ_{FE} = bending endurance limit
Y_d = design factor
Y_N = life factor
$Y_{\delta relT}$ = relative notch sensitivity factor
Y_{RrelT} = relative surface factor
Y_X = size factor
S_F = safety factor for tooth root bending stress

3.3 Tooth form factor, Y_F

3.3.1 The tooth form factor, Y_F, represents the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact. Y_F shall be determined separately for the pinion and the wheel. In the case of helical gears, the form factors for gearing shall be determined in the normal section, i.e. for the virtual spur gear with virtual number of teeth Z_n.

3.3.2 The tooth form factor, Y_F, is to be calculated as follows:

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

Where:

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 normal chord in the critical section	mm
α_{Fen}	=	pressure angle at the outer point of single tooth pair contact in the normal section	°

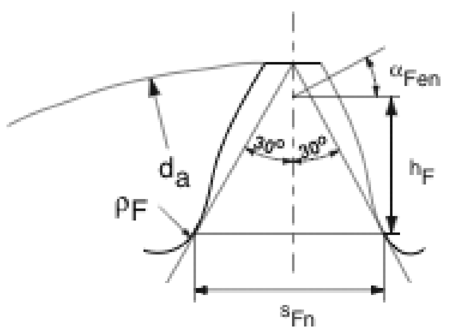

Fig. 3.3 : Dimensions of h_F, s_{Fn} and α_{Fen} for external gear

3.3.3 For the calculation of h_F, s_{Fn} and α_{Fen}, the procedure outlined in the reference standard ISO 6336-3 (Method B) is to be used.

3.4 Stress correction factor, Y_S

3.4.1 The stress correction factor Y_S, is used to convert the nominal bending stress to the local tooth root stress, taking into account that not only bending stresses arise at the root.

3.4.2 Y_S applies to the load application at the outer point of single tooth pair contact. Y_S shall be determined separately for the pinion and for the wheel. The stress correction factor, Y_S, is to_be determined with the following equation (having range of validity: $1 \le q_s \le 8$):

$$Y_S = (1.2 + 0.13L) q_s^{\left(\frac{1}{1.21 + 2.3/L}\right)}$$

Where:

$$q_s = \frac{s_{Fn}}{2\rho_F}$$

q_s = notch parameter,

ρ_F = root fillet radius in the critical section, mm

L = s_{Fn}/h_F

For h_F and s_{Fn} see clause 3.1

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

3.5 Helix angle factor, Y_β

3.5.1 The helix angle factor, Y_β, converts the stress calculated for a point loaded cantilever beam representing the substitute gear tooth to the stress induced by a load along an oblique load line into a cantilever plate which represents a helical gear tooth.

3.5.2 The helix angle factor, Y_β is to be calculated as follows:

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

where:

β = reference helix angle in degrees.

The value 1.0 is substituted for ε_β when ε_β > 1.0, and 30° is substituted for β > 30°.

3.6 Rim thickness factor, Y_B

3.6.1 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:

a) for external gears:

if $s_R / h \geq 1.2$ $\quad Y_B = 1$

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

where:

s_R = rim thickness of external gears, mm

h = tooth height, mm

The case $s_R / h \leq 0.5$ is to be avoided.

b) for internal gears:

if $s_R / m_n \geq 3.5$ $\quad Y_B = 1$

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

where:

s_R = rim thickness of internal gears, mm

The case $s_R / m_n \leq 1.75$ is to be avoided.

3.7 Deep tooth factor, Y_{DT}

3.7.1 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 $2.05 \leq \varepsilon_{\alpha n} \leq 2.5$, where:

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

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

if ISO accuracy grade ≤ 4 and $\varepsilon_{\alpha n} > 2.5$ $\quad Y_{DT} = 0.7$

if ISO accuracy grade ≤4 and $2.05 < \varepsilon_{\alpha n} \leq 2.5$

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

in all other cases $\quad Y_{DT} = 1.0$

3.8 Bending endurance limit, σ_{FE}

3.8.1 For a given material, σ_{FE} is the local tooth root stress which can be permanently endured.

3.8.2 According to the reference standard ISO 6336-5 the number of 3×10^6 cycles is regarded as the beginning of the endurance limit.

3.8.3 σ_{FE} is defined as the unidirectional pulsating stress with a minimum stress of zero (disregarding residual stresses due to heat treatment). Other conditions such as alternating stress or prestressing etc. are covered by the design factor Y_d.

3.8.4 The σ_{FE} values are to correspond to a failure probability 1% or less.

3.8.5 The endurance limit mainly depends on:
- material composition, cleanliness and defects;
- mechanical properties;
- residual stresses;
- hardening process, depth of hardened zone, hardness gradient;
- material structure (forged, rolled bar, cast).

3.8.6 The bending endurance limit, σ_{FE} is to be determined, in general, making reference to values indicated in the reference standard ISO 6336-5, for material quality *MQ*.

3.9 Design factor, Y_d

3.9.1 The design factor, Y_d, takes into account the influence of load reversing and shrinkfit prestressing on the tooth root strength, relative to the tooth root strength with unidirectional load as defined for σ_{FE}.

3.9.2 The design factor, Y_d, for load reversing, is to be determined as follows:

$Y_d = 1.0$	in general;
$Y_d = 0.9$	for gears with occasional part load in reversed direction, such as main wheel in reversing gearboxes;
$Y_d = 0.7$	for idler gears

3.10 Life factor, Y_N

3.10.1 The life factor, Y_N, accounts for the higher tooth root bending stress permissible in case a limited life (number of cycles) is required.

3.10.2 The factor mainly depends on:
- material and heat treatment;
- number of load cycles (service life);
- influence factors ($Y_{\delta relT}$,Y_{RrelT} , Y_X).

3.10.3 The life factor, Y_N, is to be determined according to Method B outlined in the reference standard ISO 6336-3.

3.11 Relative notch sensitivity factor, $Y_{\delta relT}$

3.11.1 The relative notch sensitivity factor, $Y_{\delta relT}$, indicates the extent to which the theoretically concentrated stress lies above the fatigue endurance limit. The factor mainly depends on material and relative stress gradient.

3.11.2 The relative notch sensitivity factor, $Y_{\delta relT}$, is to be determined as follows:

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

where:

q_s = notch parameter (see clause 3.4)

ρ' = slip-layer thickness, mm, from the following table

Material		ρ', mm
case hardened steels, flame or induction hardened steels		0.0030
through-hardened steels[1)], yield point R_e=	500 N/mm²	0.0281
	600 N/mm²	0.0194
	800 N/mm²	0.0064
	1000 N/mm²	0.0014
nitrided steels		0.1005
[1)]The given values of ρ' can be interpolated for values of R_e not stated above		

3.12 Relative surface factor, Y_{RrelT}

3.12.1 The relative surface factor, Y_{RrelT}, takes into account the dependence of the root strength on the surface condition in the tooth root fillet, mainly the dependence on the peak to valley surface roughness.

3.12.2 The relative surface factor, Y_{RrelT} is to be determined as follows:

$R_z < 1$	$1 \le R_z \le 40$	Material
1.120	$1.674 - 0.529\,(R_z+1)^{0.1}$	case hardened steels, through - hardened steels ($\sigma_B \ge 800$ N/mm^2)
1.070	$5.306 - 4.203\,(R_z+1)^{0.01}$	normalised steels ($\sigma_B < 800$ N/mm^2)
1.025	$4.299 - 3.259\,(R_z+1)^{0.0058}$	nitrided steels

Where:

R_z = mean peak-to-valley roughness of tooth root fillets, μm

σ_B = tensile strength, N/mm^2

3.12.3 The method applied here is only valid when scratches or similar defects deeper than 2R_z are not present.

3.12.4 If the roughness stated is an arithmetic mean roughness, i.e. R_a value (=*CLA* value) (=*AA* value) the following approximate relationship can be applied:

$$R_a = CLA = AA = R_z / 6$$

3.13 Size factor, Y_X

3.13.1 The size factor, Y_X, takes into account the decrease of the strength with increasing size.

3.13.2 The factor mainly depends on:
- material and heat treatment;
- tooth and gear dimensions;
- *ratio of case depth to tooth size.*

3.13.3 The size factor, Y_X, is to be determined as follows:

$Y_X = 1.00$	for $m_n \leq 5$	generally
$Y_X = 1.03 - 0.06\ m_n$	for $5 < m_n < 30$	normalised and through-hardened steels
$Y_X = 0.85$	for $m_n \geq 30$	
$Y_X = 1.05 - 0.010\ m_n$	for $5 < m_n < 25$	surface hardened steels
$Y_X = 0.80$	for $m_n \geq 25$	

3.14 Safety factor for tooth root bending stress, S_F

3.14.1 The safety factor for tooth root bending stress, S_F, can be assumed by IRS taking into account the type of application.

3.14.2 The following guidance values can be adopted:

- Main propulsion gears: 1.55 to 2.00
- Auxiliary gears: 1.40 to 1.45

3.14.3 For gearing of duplicated independent propulsion or auxiliary machinery, duplicated beyond that required for class, a reduced value can be assumed at the discretion of the IRS.