Disclosure to Promote the Right To Information

Whereas the Parliament of India has set out to provide a practical regime of right to information for citizens to secure access to information under the control of public authorities, in order to promote transparency and accountability in the working of every public authority, and whereas the attached publication of the Bureau of Indian Standards is of particular interest to the public, particularly disadvantaged communities and those engaged in the pursuit of education and knowledge, the attached public safety standard is made available to promote the timely dissemination of this information in an accurate manner to the public.

Indian Standard
CALCULATION OF LOAD CAPACITY OF SPUR AND HELICAL GEARs — APPLICATION TO MARINE GEARs
( First Revision )

ICS 21.200; 47.020.05

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BUREAU OF INDIAN STANDARDS
MANAK BHAVAN, 9 BAHADUR SHAH ZAFAR MARG
NEW DELHI 110002

October 2007
NATIONAL FOREWORD

This Indian Standard (First Revision) which is identical with ISO 9083 : 2001 'Calculation of load capacity of spur and helical gears — Application to marine gears' issued by the International Organization for Standardization (ISO) was adopted by the Bureau of Indian Standards on the recommendation of the Transmission Devices Sectional Committee and approval of the Production and General Engineering Division Council.

This standard was first published in 1978 covering the basic requirements of marine gears. In order to harmonize this standard with International Standard, the committee decided to revise this standard to align it with ISO 9083 : 2001 by adoption under dual number.

The text of ISO Standard has been approved as suitable for publication as an Indian Standard without deviations. Certain conventions are, however, not identical to those used in Indian Standards. Attention is particularly drawn to the following:

a) Wherever the words 'International Standard' appear referring to this standard, they should be read as 'Indian Standard'.

b) Comma (,) has been used as a decimal marker, while in Indian Standards the current practice is to use a point (.) as the decimal marker.

In this adopted standard, reference appears to certain International Standards for which Indian Standards also exist. The corresponding Indian Standards, which are to be substituted in their places, are listed below along with their degree of equivalence for the editions indicated:

<table>
<thead>
<tr>
<th>International Standard</th>
<th>Corresponding Indian Standard</th>
<th>Degree of Equivalence</th>
</tr>
</thead>
<tbody>
<tr>
<td>ISO 53 : 1998 Cylindrical gears for general and heavy engineering — Standard basic rack tooth profile</td>
<td>IS 2535 (Part 1) : 2004 Cylindrical gears for general and heavy engineering: Part 1 Standard basic rack tooth profile</td>
<td>Identical</td>
</tr>
<tr>
<td>ISO 54 : 1996 Cylindrical gears for general and heavy engineering — Modules</td>
<td>IS 2535 (Part 2) : 2004 Cylindrical gears for general and heavy engineering: Part 2 Modules</td>
<td>do</td>
</tr>
</tbody>
</table>

(Continued on third cover)
Indian Standard

CALCULATION OF LOAD CAPACITY OF SPUR AND HELICAL GEARS — APPLICATION TO MARINE GEARS

(First Revision)

1 Scope

The formulae specified in this International Standard are intended for the establishment of a uniformly acceptable method for calculating the pitting resistance and bending strength capacity for the endurance of the main-propulsion and auxiliary gears of ships, offshore vessels and drilling rigs, having straight or helical teeth and subject to the rules of classification societies.

The rating formulae in this International Standard are not applicable to other types of gear tooth deterioration, such as plastic yielding, micropitting, scuffing, case crushing, welding and wear, and are not applicable under vibratory conditions where there may be an unpredictable profile breakdown. The bending strength formulae are applicable to fractures at the tooth fillet, but are not applicable to fractures on the tooth working profile surfaces, failure of the gear rim, or failures of the gear blank through web and hub. This International Standard does not apply to teeth finished by forging or sintering. This standard is not applicable to gears having a poor contact pattern.

This International Standard provides a method by which different gear designs can be compared. It is not intended to assure the performance of assembled drive gear systems. It is not intended for use by the general engineering public. Instead, it is intended for use by the experienced gear designer who is capable of selecting reasonable values for the factors in these formulae based on knowledge of similar designs and awareness of the effects of the items discussed.

WARNING — The user is cautioned that the calculated results of this International Standard should be confirmed by experience.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this International Standard. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.


3 Terms, definitions and symbols

For the purposes of this International Standard, the terms and definitions given in ISO 1122-1 apply. For symbols, see Table 1.
Table 1 — Symbols and abbreviations used in this International Standard

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description or term</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$</td>
<td>centre distance</td>
<td>mm</td>
</tr>
<tr>
<td>$b$</td>
<td>facewidth</td>
<td>mm</td>
</tr>
<tr>
<td>$b_B$</td>
<td>facewidth of an individual helix of a double helical gear</td>
<td>mm</td>
</tr>
<tr>
<td>$c_T$</td>
<td>mean value of mesh stiffness per unit facewidth</td>
<td>N/(mm·µm)</td>
</tr>
<tr>
<td>$c'$</td>
<td>maximum tooth stiffness of one pair of teeth per unit facewidth (single stiffness)</td>
<td>N/(mm·µm)</td>
</tr>
<tr>
<td>$d_{1,2}$</td>
<td>reference diameter of pinion, wheel</td>
<td>mm</td>
</tr>
<tr>
<td>$d_{a1,2}$</td>
<td>tip diameter of pinion, wheel</td>
<td>mm</td>
</tr>
<tr>
<td>$d_{b1,2}$</td>
<td>base diameter of pinion, wheel</td>
<td>mm</td>
</tr>
<tr>
<td>$d_{h1,2}$</td>
<td>root diameter of pinion, wheel</td>
<td>mm</td>
</tr>
<tr>
<td>$d_{sh}$</td>
<td>shaft nominal diameter for bending</td>
<td>mm</td>
</tr>
<tr>
<td>$d_{shi}$</td>
<td>internal diameter of hollow shaft</td>
<td>mm</td>
</tr>
<tr>
<td>$d_{w1,2}$</td>
<td>working pitch diameter of pinion, wheel</td>
<td>mm</td>
</tr>
<tr>
<td>$d_{Na1,2}$</td>
<td>diameter of a circle defining the outer extremities of the usable flanks of tip chamfered/rounded gear teeth</td>
<td>mm</td>
</tr>
<tr>
<td>$f_B$</td>
<td>tooth alignment deviation (not including helix form deviation)</td>
<td>µm</td>
</tr>
<tr>
<td>$f_{ma}$</td>
<td>mesh misalignment due to manufacturing deviations</td>
<td>µm</td>
</tr>
<tr>
<td>$f_{PB}$</td>
<td>transverse base pitch deviation (the values of $f_{PB}$ may be used for calculation in accordance with ISO 6336-1, using tolerances complying with ISO 1328-1)</td>
<td>µm</td>
</tr>
<tr>
<td>$f_{sh}$</td>
<td>helix deviation due to elastic deflections</td>
<td>µm</td>
</tr>
<tr>
<td>$h_c$</td>
<td>length of path of contact</td>
<td>mm</td>
</tr>
<tr>
<td>$h$</td>
<td>tooth depth</td>
<td>mm</td>
</tr>
<tr>
<td>$h_{ap}$</td>
<td>addendum of basic rack of cylindrical gears</td>
<td>mm</td>
</tr>
<tr>
<td>$h_{ip}$</td>
<td>dedendum of basic rack of cylindrical gears</td>
<td>mm</td>
</tr>
<tr>
<td>$h_{fo}$</td>
<td>bending moment arm for load application at the outer point of single pair tooth contact</td>
<td>mm</td>
</tr>
<tr>
<td>$l$</td>
<td>bearing span</td>
<td>mm</td>
</tr>
<tr>
<td>$m^*$</td>
<td>relative individual gear mass per unit facewidth referenced to line of action</td>
<td>kg/mm</td>
</tr>
<tr>
<td>$m_n$</td>
<td>normal module</td>
<td>mm</td>
</tr>
<tr>
<td>$m_{red}$</td>
<td>reduced gear pair mass per unit facewidth referenced to the line of action</td>
<td>kg/mm</td>
</tr>
<tr>
<td>$m_t$</td>
<td>transverse module</td>
<td>mm</td>
</tr>
<tr>
<td>$n_{1,2}$</td>
<td>rotation speed of pinion, of wheel</td>
<td>min⁻¹</td>
</tr>
<tr>
<td>$n_E$</td>
<td>resonance speed</td>
<td>min⁻¹</td>
</tr>
<tr>
<td>$\rho_{bn}$</td>
<td>normal base pitch</td>
<td>mm</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description or term</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>---------------------</td>
<td>------</td>
</tr>
<tr>
<td>$P_{bt}$</td>
<td>transverse base pitch</td>
<td>mm</td>
</tr>
<tr>
<td>$p_r$</td>
<td>protuberance of the tool</td>
<td>mm</td>
</tr>
<tr>
<td>$q$</td>
<td>finishing stock allowance of tooth flank</td>
<td>mm</td>
</tr>
<tr>
<td>$q_s$</td>
<td>notch parameter $s_{Fn}/2p_F$</td>
<td>—</td>
</tr>
<tr>
<td>$s$</td>
<td>tooth thickness</td>
<td>mm</td>
</tr>
<tr>
<td>$s_{Fn}$</td>
<td>tooth-root chord at the critical section</td>
<td>mm</td>
</tr>
<tr>
<td>$r_R$</td>
<td>rim thickness</td>
<td>mm</td>
</tr>
<tr>
<td>$u$</td>
<td>gear ratio $u =</td>
<td>z_2/z_1</td>
</tr>
<tr>
<td>$v$</td>
<td>tangential speed (without subscript: at reference circle = tangential speed at pitch circle)</td>
<td>m/s</td>
</tr>
<tr>
<td>$v_p$</td>
<td>velocity parameter</td>
<td>—</td>
</tr>
<tr>
<td>$x_{1,2}$</td>
<td>profile shift coefficient of pinion, wheel</td>
<td>—</td>
</tr>
<tr>
<td>$y_n$</td>
<td>running-in allowance for a gear pair</td>
<td>μm</td>
</tr>
<tr>
<td>$y_0$</td>
<td>running-in allowance (equivalent misalignment)</td>
<td>μm</td>
</tr>
<tr>
<td>$z_n$</td>
<td>virtual number of teeth of a helical gear</td>
<td>—</td>
</tr>
<tr>
<td>$z_{1,2}$</td>
<td>number of teeth of pinion, of wheel $^a$</td>
<td>—</td>
</tr>
<tr>
<td>$A$</td>
<td>auxiliary value for the determination of $f_{sh}$</td>
<td>mm·μm/N</td>
</tr>
<tr>
<td>$B$</td>
<td>total facewidth of a double helical gear including the gap</td>
<td>mm</td>
</tr>
<tr>
<td>$C_a$</td>
<td>tip relief</td>
<td>μm</td>
</tr>
<tr>
<td>$C_B$</td>
<td>basic rack factor (same rack for pinion and wheel)</td>
<td>—</td>
</tr>
<tr>
<td>$C_R$</td>
<td>gear blank factor</td>
<td>—</td>
</tr>
<tr>
<td>$E$</td>
<td>modulus of elasticity, Young's modulus</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$F_m$</td>
<td>the mean transverse load at the reference cylinder ($= F_t K_A K_v$)</td>
<td>N</td>
</tr>
<tr>
<td>$F_t$</td>
<td>(nominal) transverse tangential load at reference cylinder</td>
<td>N</td>
</tr>
<tr>
<td>$F_{th}$</td>
<td>the determinant transverse load at the reference cylinder ($= F_t K_A K_v K_{th}$)</td>
<td>N</td>
</tr>
<tr>
<td>$F_{\beta}$</td>
<td>total helix deviation</td>
<td>μm</td>
</tr>
<tr>
<td>$F_{\beta x}$</td>
<td>initial equivalent misalignment (before running-in)</td>
<td>μm</td>
</tr>
<tr>
<td>$F_{\beta y}$</td>
<td>initial equivalent misalignment (after running-in)</td>
<td>μm</td>
</tr>
<tr>
<td>$K_v$</td>
<td>dynamic factor</td>
<td>—</td>
</tr>
<tr>
<td>$K_A$</td>
<td>application factor</td>
<td>—</td>
</tr>
<tr>
<td>$K_{Fa}$</td>
<td>transverse load factor (root stress)</td>
<td>—</td>
</tr>
</tbody>
</table>
Table 1 — (Continued)

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description or term</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_{F_{B}}$</td>
<td>face load factor (root stress)</td>
<td>—</td>
</tr>
<tr>
<td>$K_{H_{a}}$</td>
<td>transverse load factor (contact stress)</td>
<td>—</td>
</tr>
<tr>
<td>$K_{H_{b}}$</td>
<td>face load factor (contact stress)</td>
<td>—</td>
</tr>
<tr>
<td>$K_{r}$</td>
<td>mesh load factor (takes into account the uneven distribution of the load between meshes for multiple transmission paths)</td>
<td>—</td>
</tr>
<tr>
<td>$M_{1,2}$</td>
<td>auxiliary values for the determination of $Z_{B,D}$</td>
<td>—</td>
</tr>
<tr>
<td>$N_{L}$</td>
<td>number of cycles</td>
<td>—</td>
</tr>
<tr>
<td>$N_{S}$</td>
<td>resonance ratio in the main resonance range</td>
<td>—</td>
</tr>
<tr>
<td>$P$</td>
<td>transmitted power</td>
<td>kW</td>
</tr>
<tr>
<td>$R_{a}$</td>
<td>arithmetic mean roughness value (as specified in ISO 4287)</td>
<td>µm</td>
</tr>
<tr>
<td>$R_{z}$</td>
<td>mean peak-to-valley roughness (as specified in ISO 4287)</td>
<td>µm</td>
</tr>
<tr>
<td>$S_{F}$</td>
<td>factor of safety from tooth breakage</td>
<td>—</td>
</tr>
<tr>
<td>$S_{F_{\text{min}}}$</td>
<td>minimum safety factor (tooth breakage)</td>
<td>—</td>
</tr>
<tr>
<td>$S_{H}$</td>
<td>factor of safety from pitting</td>
<td>—</td>
</tr>
<tr>
<td>$S_{H_{\text{min}}}$</td>
<td>minimum safety factor (pitting)</td>
<td>—</td>
</tr>
<tr>
<td>$T_{1,2}$</td>
<td>pinion torque, wheel torque; (nominal)</td>
<td>Nm</td>
</tr>
<tr>
<td>$Y_{F}$</td>
<td>tooth form factor</td>
<td>—</td>
</tr>
<tr>
<td>$Y_{R_{\text{rel T}}}$</td>
<td>relative surface factor</td>
<td>—</td>
</tr>
<tr>
<td>$Y_{S}$</td>
<td>stress correction factor</td>
<td>—</td>
</tr>
<tr>
<td>$Y_{X}$</td>
<td>size factor (tooth-root)</td>
<td>—</td>
</tr>
<tr>
<td>$Y_{\beta}$</td>
<td>helix angle factor (tooth-root)</td>
<td>—</td>
</tr>
<tr>
<td>$Y_{R_{\text{rel T}}}$</td>
<td>relative notch sensitivity factor</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{v}$</td>
<td>speed factor</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{B,D}$</td>
<td>single pair tooth contact factors for the pinion, for the wheel</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{E}$</td>
<td>elasticity factor</td>
<td>$\sqrt{\text{N/mm}^2}$</td>
</tr>
<tr>
<td>$Z_{H}$</td>
<td>zone factor</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{L}$</td>
<td>lubricant factor</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{R}$</td>
<td>roughness factor affecting surface durability</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{W}$</td>
<td>work-hardening factor</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{X}$</td>
<td>size factor (pitting)</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{\beta}$</td>
<td>helix angle factor (pitting)</td>
<td>—</td>
</tr>
<tr>
<td>$Z_{e}$</td>
<td>contact ratio factor (pitting)</td>
<td>—</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description or term</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>---------------------</td>
<td>------</td>
</tr>
<tr>
<td>$\alpha_n$</td>
<td>normal pressure angle</td>
<td>°</td>
</tr>
<tr>
<td>$\alpha_t$</td>
<td>transverse pressure angle</td>
<td>°</td>
</tr>
<tr>
<td>$\alpha_{wt}$</td>
<td>transverse pressure angle at the working pitch cylinder</td>
<td>°</td>
</tr>
<tr>
<td>$\alpha_{p0}$</td>
<td>normal pressure angle of the basic rack for cylindrical gears</td>
<td>°</td>
</tr>
<tr>
<td>$\beta$</td>
<td>helix angle (without subscript — at the reference cylinder)</td>
<td></td>
</tr>
<tr>
<td>$\beta_b$</td>
<td>base helix angle</td>
<td>°</td>
</tr>
<tr>
<td>$e_{ct}$</td>
<td>transverse contact ratio</td>
<td></td>
</tr>
<tr>
<td>$e_{ctv}$</td>
<td>virtual contact ratio, transverse contact ratio of a virtual gear</td>
<td></td>
</tr>
<tr>
<td>$e_{\alpha}$</td>
<td>axial overlap ratio</td>
<td></td>
</tr>
<tr>
<td>$e_{\gamma}$</td>
<td>total contact ratio ($e_{\gamma} = e_{\alpha} + e_{\beta}$)</td>
<td></td>
</tr>
<tr>
<td>$f_{\beta}$</td>
<td>factor characterizing the equivalent misalignment after running-in</td>
<td></td>
</tr>
<tr>
<td>$\nu_{40,50}$</td>
<td>kinematic viscosity at 40 °C, 50 °C</td>
<td></td>
</tr>
<tr>
<td>$\eta$</td>
<td>viscosity parameter</td>
<td></td>
</tr>
<tr>
<td>$\rho_p$</td>
<td>root fillet radius of the basic rack for cylindrical gears</td>
<td>mm</td>
</tr>
<tr>
<td>$\rho_{rel}$</td>
<td>radius of relative curvature</td>
<td>mm</td>
</tr>
<tr>
<td>$\rho_C$</td>
<td>radius of relative curvature at the pitch surface</td>
<td>mm</td>
</tr>
<tr>
<td>$\rho_T$</td>
<td>tooth-root fillet radius at the critical section</td>
<td>mm</td>
</tr>
<tr>
<td>$\sigma_B$</td>
<td>tensile strength</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_T$</td>
<td>tooth-root stress</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{lim}$</td>
<td>nominal stress number (bending)</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{FE}$</td>
<td>allowable stress number (bending) = $\sigma_{lim} \times_{ST}$</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{FG}$</td>
<td>tooth-root stress limit</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{FP}$</td>
<td>permissible tooth-root stress</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{FO}$</td>
<td>nominal tooth-root stress</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_H$</td>
<td>calculated contact stress</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{lim}$</td>
<td>allowable stress number (contact)</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{HG}$</td>
<td>modified allowable stress number = $\sigma_{FP} \times_{H_{min}}$</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{HP}$</td>
<td>permissible contact stress</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_H0$</td>
<td>nominal contact stress</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\sigma_{2}$</td>
<td>angular velocity of pinion, or wheel</td>
<td>rad/s</td>
</tr>
</tbody>
</table>

a For external gear pairs $\alpha, \nu, z_1$ and $z_2$ are positive; for internal gear pairs $\alpha, \nu$ and $z_2$ are negative with $z_1$ positive.
4 Application

4.1 Design, specific applications

4.1.1 General

Gear designers shall recognize that requirements for different applications vary considerably. Use of the procedures of this International Standard for specific applications demands a careful appraisal of all applicable considerations, in particular:

— the allowable stress of the material and the number of load repetitions;
— the consequences of any percentage of failure (failure rate);
— the appropriate factor of safety.

Design considerations to prevent fractures emanating from stress raisers in the tooth flank, tip chipping and failures of the gear blank through the web or hub should be analysed by general machine design methods.

Any variances according to the following shall be reported in the calculation statement.

a) If a more refined method of calculation is desired or if compliance with the restrictions in clause 4.1 is for any reason impractical, relevant factors may be evaluated according to the basic standard or another application standard.

b) Factors derived from reliable experience or test data may be used instead of individual factors according to this International Standard. Concerning this, the criteria for Method A in ISO 6336-1:1996, 4.1.8, are applicable.

In other respects, rating calculations shall be strictly in accordance with this International Standard if stresses, safety factors, etc. are to be classified as being in accordance with this International Standard.

This International Standard is applicable when the wheel blank, shaft/hub connections, shafts, bearings, housings, threaded connections, foundations and couplings conform to the requirements regarding accuracy, load capacity and stiffness forming the basis for the calculation of the load capacity of gears.

Although the method described in this International Standard is mainly intended for recalculation purposes, by means of iteration it can also be used to determine the load capacities of gears. The iteration is accomplished by selecting a load and calculating the corresponding safety factor against pitting, $S_{H1}$, for the pinion. If $S_{H1}$ is greater than $S_{H \text{min}}$, the load is increased, if it is smaller than $S_{H \text{min}}$, the load is reduced. This is done until the load chosen corresponds to $S_{H1} = S_{H \text{min}}$. The same method is used for the wheel ($S_{H2} = S_{H \text{min}}$) and also for the safety factors against tooth breakage, $S_{F1} = S_{F2} = S_{F \text{min}}$.

4.1.2 Gear data

This International Standard is applicable within the following constraints.

a) Types of gear:

— external and internal, involute spur, helical and double helical gears;

— for double helical gears, it is assumed that the total tangential load is evenly distributed between the two helices; if this is not the case (e.g. due to externally applied axial forces), this shall be taken into account; the two helices are treated as two single helical gears in parallel;

— planetary and other gear trains with multiple transmission paths.
b) Range of the transverse contact ratios of actual spur and helical gear pairs:

- $1.2 < e_a < 2.5$ (affects $c', c_r, K_v, K_{Hj}, K_{Fh}, K_{Ha}$ and $K_{Fb}$).

c) Range of helix angles:

- $\beta$ less than or equal to 30° (affects $c', c_r, K_v$ and $K_{Hj}$).

d) Basic racks:

- no restriction\(^1\).

4.1.3 Wheel blank, wheel rim

This International Standard is applicable when $s_R$, the thickness of the wheel rim under the tooth roots of internal and external gears, is $>3.5 \, \text{mm}$.

4.1.4 Materials

These include steels (affects $Z_E$, $\sigma_{H, \text{lim}}$, $\sigma_{FE}$, $K_v$, $K_{Hj}$ and $K_{Fh}$). For materials and their abbreviations used in this International Standard, see Table 2. For other materials, see ISO 6336-1, ISO 6336-2, ISO 6336-3 and ISO 6336-5.

<table>
<thead>
<tr>
<th>Material</th>
<th>Abbreviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Through-hardening steel, alloy or carbon, through-hardened ($\sigma_b \geq 800 , \text{N/mm}^2$)</td>
<td>$V$</td>
</tr>
<tr>
<td>Case-hardened steel, case hardened</td>
<td>$Eh$</td>
</tr>
<tr>
<td>Steel, flame or induction hardened</td>
<td>$IF$</td>
</tr>
<tr>
<td>Nitriding steel, nitrided</td>
<td>$NT$ (nitr.)</td>
</tr>
<tr>
<td>Through-hardening and case-hardening steel, nitrided</td>
<td>$NV$ (nitr.)</td>
</tr>
<tr>
<td>Through-hardening and case-hardening steel, nitrocarburized</td>
<td>$NV$ (nitrocarb.)</td>
</tr>
</tbody>
</table>

4.1.5 Lubrication

The calculation procedures are valid for gears that are spray or oil-bath lubricated using a lubricant approved by the manufacturer/designer of the gears. This validity is further subject to the condition that, at all times of operation, an adequate quantity of approved lubricant is available to the gear mesh. Provision for cooling shall ensure that temperatures assumed for purposes of calculations are not exceeded (affects lubricant film formation i.e. factors $Z_L$, $Z_V$ and $Z_{FH}$).

Provided that sufficient lubricant is available to the mesh, grease lubrication of slow speed auxiliaries is not excluded.

\(^1\) For all practical purposes, it may be assumed that the proportions of the basic rack of the tools are equal to those of the basic rack of the gear.
4.2 Safety factors

It is necessary to distinguish between the safety factor relative to pitting, \( S_H \), and the safety factor relative to tooth breakage, \( S_F \).

For a given application, adequate gear load capacity is demonstrated by the computed values of \( S_H \) and \( S_F \) being equal to or greater than the values \( S_{H\text{min}} \) and \( S_{F\text{min}} \), respectively.

Choice of the value of a safety factor should be based on the degree of confidence in the reliability of the available data and the consequences of possible failures.

Important factors to be considered are:

a) the allowable stress numbers used in the calculation are valid for a given probability of failure (the material values in ISO 6336-5:1996 are valid for 1% probability of damage);

b) the specified quality and the effectiveness of quality control at all stages of manufacture;

c) the accuracy of specification of the service duty and external conditions;

d) tooth breakage is often considered to be a greater hazard than pitting.

Therefore, the chosen value for \( S_{F\text{min}} \) should be greater than the value chosen for \( S_{H\text{min}} \). For calculation of actual safety factor see 6.1.5 (\( S_{H} \), pitting) and 7.1.4 (\( S_{F} \), tooth breakage).

It is recommended that the minimum values of the safety factors should be agreed upon between the purchaser, the manufacturer and the classification authority.

4.3 Input data

The following data shall be available for the calculations:

a) gear data:
   \[ a, z_1, z_2, m_n, d_1, d_2, d_{a1}, d_{a2}, b, x_1, x_2, \alpha_n, \beta, \varepsilon_a, \varepsilon_b \] (see ISO 53:1998, ISO 54:1996);

b) cutter basic rack tooth profile:
   \[ h_a 0, \rho_a 0 \] (see ISO 53:1998);

c) design and manufacturing data:
   \[ C_{a1}, C_{a2}, f_{pb}, S_{H\text{min}}, S_{F\text{min}}, R_{a1}, R_{a2}, R_{z1}, R_{z2} \];

   materials, material hardness and heat treatment details, gear accuracy grades, bearing span \( l \), positions of gears relative to bearings, dimensions of pinion shaft \( d_{sh} \) and, when applicable, helix modification (crowning, end relief);

d) power data:
   \[ P \text{ or } T \text{ or } F, n_1, v_1, \text{ details of driving and driven machines.} \]

Requisite geometrical data can be calculated according to national standards.
Information to be exchanged between manufacturer and purchaser should include data specifying material preferences, lubrication, safety factor and externally applied forces due to vibrations and overloads (application factor).

4.4 Numerical equations

The units listed in clause 3 shall be used in all calculations. Information that will facilitate the use of this International Standard is provided in annex C of ISO 6336-1:1996.

5 Influence factors

5.1 General

The influence factors, $K_v$, $K_{Hv}$, $K_{Hb}$, $K_{Fv}$ and $K_{FB}$, are all dependent on the tooth load. Initially, this is the applied load (nominal tangential load multiplied by the application factor).

The factors are also interdependent and shall therefore be calculated successively as follows:

a) $K_v$ with the applied tangential load $F_t K_A$;

b) $K_{Hb}$ or $K_{FB}$ with the recalculated load $F_t K_A K_v$;

c) $K_{Hv}$ or $K_{Fv}$ (Method B) with the applied tangential load $F_t K_A K_v K_{Hb}$.

When a gear drives two or more mating gears or is double helical, it is necessary to substitute $K_A$ by $K_A K_T$. If possible, the mesh load factor, $K_T$, should preferably be determined by measurement; alternatively its value may be estimated from the available literature.

The simplification of all influence factors in this clause involves the following assumptions (also see clause 4):

a) that the pinion tooth number $z_1 < 50$;

b) that gears are of solid disc type or with heavy rims.

When details are substantially different from any of the above, refer to ISO 6336-1.

5.2 Nominal tangential load, $F_t$, nominal torque, $T$, nominal power, $P$

The nominal tangential load, $F_t$, is determined in the transverse plane at the reference cylinder. It is based on the input torque to the driven machine. This is the torque corresponding to the heaviest regular working condition. Alternatively, the nominal torque of the prime mover can be used as a basis if it corresponds to the torque requirement of the driven machine, or some other suitable basis can be chosen.

\[
F_t = \frac{2000}{d_{12}} T_{12} = \frac{19.098 \times 1000 P}{d_{12} n_{12}} = \frac{1000 P}{v} \quad (1)
\]

\[
T_{12} = \frac{F_t d_{12}}{2000} = \frac{1000 P}{\omega_{12}} = \frac{9549 P}{n_{12}} \quad (2)
\]

\[
P = \frac{F_t}{1000} = \frac{T_{12} \omega_{12}}{1000} = \frac{T_{12} n_{12}}{9549} \quad (3)
\]
When the transmitted load is not uniform, consideration should be given not only to the peak load and its anticipated number of cycles, but also to intermediate loads and their numbers of cycles. This type of load is classed as a duty cycle and may be represented by a load spectrum. In such cases, the cumulative fatigue effect of the duty cycle shall be considered in rating the gearset. A method of calculating the effect of the loads under this condition is given in ISO TR 10495.

This is the maximum tangential load, $F_{\text{tmax}}$, maximum torque, $T_{\text{max}}$, maximum power, $P_{\text{max}}$ in the variable duty range. Its magnitude can be limited by a suitably responsive safety clutch. $F_{\text{tmax}}$, $T_{\text{max}}$, and $P_{\text{max}}$ shall be known when safety from pitting damage and from sudden tooth breakage due to loading corresponding to the static stress limit is to be determined (see 5.5).

It is recommended that the purchaser and manufacturer/designer agree on the value of the application factor with the accord of the classification authority.

If no reliable data, obtained as described in 5.5.2, are available, or even as early as the first design phase, it is possible to use the guideline values for $K_A$ as described in annex C.

The dynamic factor relates the total tooth load, including internal dynamic effects of a "multi-resonance" system, to the transmitted tangential tooth load. Method B of ISO 6336-1:1996 with modifications is used in this International Standard.

In this procedure it is assumed that the gear pair consists of an elementary single mass and spring system comprising the combined masses of pinion and wheel, and the mesh stiffness of the contacting teeth. It is also assumed that each gear pair functions as a single stage pair, i.e. the influence of other stages in a multiple-stage gear system is ignored. This assumption is only tenable when the torsional stiffness (measured at the base radius
of the gears), of the shaft common to a wheel and a pinion is less than the mesh stiffness. See 5.6.3 and annex B.1 for the procedure dealing with very stiff shafts.

Forces caused by torsional vibrations of the shafts and coupled masses are not covered by $K_v$. These forces should be included with other externally applied forces (e.g. with the application factor).

In multiple mesh gear trains there are several natural frequencies. These can be higher or lower than the natural frequency of a single gear pair with only one mesh. When such gears run in the supercritical range, analysis by Method A is recommended. See ISO 6336-1:1996, 6.3.1.

The specific load for the calculation of $K_A$ is $(F_tK_A)/b$.

If $(F_tK_A)/b > 100 \text{ N/mm}$, then $F_m/b = (F_tK_A)/b$.

If $(F_tK_A)/b \leq 100 \text{ N/mm}$, then $F_m/b = 100 \text{ N/mm}$.

When the specific loading $F_tK_A/b$ is $< 50 \text{ N/mm}$, a particular risk of vibration exists (under some circumstances, with separation of working tooth flanks), above all for spur or helical gears of coarse quality grade running at high speed.

5.6.2 Calculation of the parameters required for evaluation of $K_v$

5.6.2.1 Calculation of the reduced mass, $m_{\text{red}}$

a) Calculation of the reduced mass, $m_{\text{red}}$, of a single-stage gear pair

$$m_{\text{red}} = \frac{J_1r_{b1}^2 + J_2r_{b2}^2}{J_1 + J_2}$$

where

$m_{\text{red}}$ is the reduced mass of a gear pair, i.e. of the mass per unit facewidth of each gear, referred to its base radius or to the line of action;

$J_{1,2}$ are the polar moments of inertia per unit facewidth;

$r_{b1,2}$ are the base radii ($= 0.5d_{b1,2}$)

b) Calculation of reduced mass, $m_{\text{red}}$, of a multi-stage gear pair

See clause B.1.

c) Calculations of reduced mass, $m_{\text{red}}$, of gears of less common designs

For information on the following cases, see clause B.1:

- pinion shaft with diameter at mid-tooth depth, $d_{m1}$, about equal to the shaft diameter;
- two rigidly connected, coaxial gears;
- one large wheel driven by two pinions;
- planetary gears;
- idler gears.
5.6.2.2 Determination of the resonance running speed (main resonance) of a gear pair

a) Resonance running speed, $n_{E1}$, of the pinion:

$$n_{E1} = \frac{30 \times 10^{-3}}{\pi z_1} \sqrt{\frac{c_Y}{m_{red}}} \text{ in min}^{-1}$$

with $c_Y$ from annex A

b) Resonance ratio, $N$

The ratio of pinion speed to resonance speed, the resonance ratio, $N$, is determined as follows:

$$N = \frac{n_E}{n_{E1}} = \frac{n_1}{30\times10^3 \sqrt{c_Y}}$$

The resonance running speed may be above or below the running speed calculated from equation (8) because of stiffness that has not been included (e.g., the stiffness of shafts, bearings and housings) and as a result of damping. For reasons of safety, the resonance range is defined by the following:

$$N_S < N \leq 1.15$$

At loads such that $(F_t K_A)/b$ is less than 100 N/mm, the lower limit of resonance ratio $N_S$ is determined:

- if $(F_t K_A)/b < 100$ N/mm, then

$$N_S = 0.5 + 0.35 \frac{F_t K_A}{b \times 100}$$

- if $(F_t K_A)/b \geq 100$ N/mm, then

$$N_S = 0.85$$

5.6.2.3 Gear accuracy and running-in parameters, $B_p$, $B_t$, $B_k$

$B_p$, $B_t$ and $B_k$ are non-dimensional parameters used to take into account the effect of tooth deviations and profile modifications on the dynamic load.

$$B_p = \frac{K_A F_t}{f_{pb \text{ eff}}}$$

$$B_t = \frac{K_A F_t}{f_{t \text{ eff}}}$$

2) The amount $C_a$ of tip relief may only be allowed for gears of accuracy grades in the range 0 to 6 as specified in ISO 1328-1:1995.
$B_k = \left| 1 - \frac{c' C_a}{F_A / b} \right|$ \hspace{1cm} (14)

with

$c'$ as given in annex A;

$C_a$ design amount for profile modification (tip relief at the beginning and end of tooth engagement). A value $C_{ay}$ from running-in shall be substituted for $C_a$ in equation (14) in the case of gears without a specified profile modification. $C_{ay}$ can be obtained from Table 3.

The effective base pitch and profile deviations are those present after running-in. The values of $f'_{pb\text{eff}}$ and $f'_{f\text{eff}}$ are determined by deducting estimated running-in allowances, $\gamma_p$ and $\gamma_f$, as follows:

$f'_{pb\text{eff}} = f_{pb1} - \gamma_p$ or $f'_{pb\text{eff}} = f_{pb2} - \gamma_p$ \hspace{1cm} (15)

whichever is the greater;

$f'_{f\text{eff}} = f_{fa1} - \gamma_f$ or $f'_{f\text{eff}} = f_{fa2} - \gamma_f$ \hspace{1cm} (16)

whichever is the greater.

5.6.2.4 Running-in allowance, $\gamma_{e}$

a) For St, V3: $\gamma_p = \gamma_e = \frac{160}{\sigma_{H11m}} f_{pb}$ \hspace{1cm} (17)

$\gamma_f = \frac{160}{\sigma_{H11m}} f_{fa}$ \hspace{1cm} (18)

b) For Eh, IF, NT (nitr.), NV (nitr.), NV (nitrocarb.)$^3$

$\gamma_p = \gamma_e = 0.075 \ f_{pb}$ \hspace{1cm} (19)

$\gamma_f = 0.075 \ f_{fa}$ \hspace{1cm} (20)

5.6.3 Dynamic factor in the subcritical range ($N \leq N_0$)

In this sector, resonances may exist if the tooth mesh frequency coincides with $N = 1/2$ and $N = 1/3$. The risk of this is slight in the case of precision helical or spur gears, if the latter have suitable profile modification (gears to ISO 1328-1:1995 accuracy grade 6 or better).

When the contact ratio of straight spur gears is small or if the quality is of low grade, $K_e$ can be just as great as in the main resonance-speed range. If this occurs, the design or operating parameters should be altered.

---

3) See Table 2 for an explanation of the abbreviations used.
Resonances at \( N = 1/4, 1/5, \ldots \) are seldom troublesome because the associated vibration amplitudes are usually small.

For gear pairs where the stiffness of the driving and driven shafts is not equal, in the range \( N = 0.2 \ldots 0.5 \), the tooth contact frequency can excite natural frequencies when the torsional stiffness, \( c_r \), of the stiffer shaft, referred to the line of action, is of the same order of magnitude as the tooth stiffness, i.e., if \( c/r_{n}^2 \) is of the order of magnitude of \( c_r \). When this is so, dynamic load increments can exceed values calculated using equation (21).

\[
K_v = (NK) + 1
\quad \text{(21)}
\]
\[
K = (C_{v1}B_p) + (C_{v2}B_t) + (C_{v3}B_k)
\quad \text{(22)}
\]

where

\( C_{v1} \) and \( C_{v2} \) allow for pitch and profile deviations while \( C_{v3} \) allows for the cyclic variation of mesh stiffness.

See Table 3.

A value \( C_{ay} \) resulting from running-in shall be substituted for \( C_a \) in equation (14) in the case of gears without a specified profile modification. The value of \( C_{ay} \) is obtained from Table 3.

See annex A for single tooth stiffness \( c' \).

### 5.6.4 Dynamic factor in the main resonance range \((N_S < N \leq 1.15)\)

High quality helical gears with high total contact ratio can function satisfactorily in this sector. Spur gears of grade 5 or better as specified in ISO 1328-1:1995 shall have suitable profile modification, as specified in ISO 6336-1:1996 clause 6.4.1 item b).

Subject to above, this factor is equal to:

\[
K_v = (C_{v1}B_p) + (C_{v2}B_t) + (C_{v4}B_k) + 1
\quad \text{(23)}
\]

For C parameters refer to Table 3.

### 5.6.5 Dynamic factor in the supercritical range \((N > 1.5)\)

Resonance peaks can occur at \( N = 2, 3 \ldots \) in this range. However, in the majority of cases, vibration amplitudes are small, since excitation forces with frequencies lower than meshing frequency are usually small.

For some gears in this speed range, it is also necessary to consider dynamic loads due to transverse vibration of the gear and shaft assemblies. When the critical frequency is near to the frequency of rotation, and if this condition cannot be avoided, this shall be taken into account in the evaluation of \( K_v \).

\[
K_v = (C_{v5}B_p) + (C_{v6}B_t) + C_{v7}
\quad \text{(24)}
\]

For C parameters, refer to Table 3.
### Table 3 — Equations for the calculation of factors $C_{v1}$ to $C_{v7}$ and $C_{sy}$

<table>
<thead>
<tr>
<th></th>
<th>$1 &lt; \varepsilon_Y \leq 2$</th>
<th>$\varepsilon_Y &gt; 2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{v1}$</td>
<td>0.32</td>
<td>0.32</td>
</tr>
<tr>
<td>$C_{v2}$</td>
<td>0.34</td>
<td>$0.57 - 0.3\varepsilon_Y$</td>
</tr>
<tr>
<td>$C_{v3}$</td>
<td>0.23</td>
<td>$0.096 - 1.56\varepsilon_Y$</td>
</tr>
<tr>
<td>$C_{v4}$</td>
<td>0.90</td>
<td>$0.57 - 0.05\varepsilon_Y$</td>
</tr>
<tr>
<td>$C_{v5}$</td>
<td>0.47</td>
<td>0.47</td>
</tr>
<tr>
<td>$C_{v6}$</td>
<td>0.47</td>
<td>$0.12 - 1.74\varepsilon_Y$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>$1 &lt; \varepsilon_Y \leq 1.5$</th>
<th>$1.5 &lt; \varepsilon_Y \leq 2.5$</th>
<th>$\varepsilon_Y &gt; 2.5$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{v7}$</td>
<td>0.75</td>
<td>$0.125 \sin[\pi (\varepsilon_Y - 2)] + 0.875$</td>
<td>1.0</td>
</tr>
</tbody>
</table>

$$C_{sy} = \frac{1}{18} \left( \frac{\sigma_{H,lim}}{18.45} - 97 \right)^2 + 1.5$$

**NOTE** When the material of the pinion (1) is different from that of the wheel (2), $C_{sy1}$ and $C_{sy2}$ are calculated separately, then $C_{sy} = 0.5 (C_{sy1} + C_{sy2})$.

#### 5.6.6 Dynamic factor in the intermediate range ($1.15 < N < 1.5$)

In this range, the dynamic factor is determined by linear interpolation between $K_v$ at $N = 1.15$, as specified in 5.6.4, and $K_v$ at $N = 1.5$, as specified in 5.6.5.

$$K_v = K_v(N=1.15) + \frac{K_v(N=1.5) - K_v(N=1.15)}{0.35} (1.5 - N)$$

#### 5.7 Face load factor, $K_{Hpb}$

**5.7.1 General**

The face load factor adjusts gear tooth stresses to allow for the effects of uneven load distribution over the facewidth.

Methods C1 and C2 of ISO 6336-1:1996 are used with modifications in this International Standard.

**5.7.2 Face load factor, $K_{Hpb-C1}$**

**5.7.2.1 General**

The use of method C1 is appropriate for gears having the following characteristics:
a) pinion on solid or hollow shaft, \(d_{ph}/d_{sh} < 0.5\), positioned symmetrically between bearings (\(u/l \leq 1\); see Figure 2), (an asymmetrically positioned pinion leads to an additional bending deformation, which shall be evaluated and added to \(f_{ma}\));

b) pinion diameter about equal to shaft diameter;

c) stiff wheel and case, stiff wheel shaft, stiff bearings;

d) a contact pattern which, under load, extends over the entire facewidth;

e) no additional external loads that act on the pinion shaft (e.g. form shaft couplings);

f) running-in allowance \(y_\beta \leq \) maximum \(y_\beta\) as specified in 5.7.2.3. A computed \(F_{px}\) may be verified using the equation:

\[
F_{px} = \frac{K_{H\beta} - 1}{\kappa_{\beta} \left( \frac{c_{y}/2}{F_{m}/b} \right)}
\]  

(26)

It is recommended that the values used for \(f_{ma}\) be verified by inspection checks, such as the tooth contact pattern in the working attitude.

Refer to annex B for application to planetary gears.

5.7.2.2 Mesh misalignment due to manufacturing tolerances, \(f_{ma}\)

a) Assembly of gears without any modification or adjustment:

\[f_{ma} = 1.0 \ f_{H\beta}\]

(27)

b) Gear pairs with provision for adjustment (lapping or running-in under light load, adjustable bearings or appropriate helix angle modification) and gear pairs suitably crowned:

\[f_{ma} = 0.5 \ f_{H\beta}\]

(28)

c) Gear pairs with well designed end relief:

\[f_{ma} = 0.7 \ f_{H\beta}\]

(29)

Of a pair of gears, the larger of the values \(f_{\beta}\) of the pair shall be substituted in equations (27) to (29).

5.7.2.3 Running-in allowance, \(y_\beta\), running-in factor, \(\kappa_{\beta}\)

The amount \(y_\beta\) is that by which the initial equivalent misalignment is reduced by running-in after start of operation. While \(\kappa_{\beta}\) is the factor characterizing the equivalent misalignment after running-in. The use of \(\kappa_{\beta}\) in calculations is valid only as long as \(\kappa_{\beta}\) is proportional to \(F_{px}\).

a) For St, V:

\[
y_\beta = \frac{320}{\sigma_{H\text{lim}}} F_{px}; \quad \kappa_{\beta} = 1 - \frac{320}{\sigma_{H\text{lim}}}
\]

(30)
with $\gamma_b < F_{\beta x}$ and $\kappa_b > 0$

when $v \leq 5 \text{ m/s}$: no restriction;

when $5 \text{ m/s} < v \leq 10 \text{ m/s}$: the upper limit of $\gamma_b$ is $25 \times 600/\sigma_{H\text{lim}}$ corresponding to $F_{\beta x} = 80 \mu$m;

when $v > 10 \text{ m/s}$: the upper limit of $\gamma_b$ is $12 \times 800/\sigma_{H\text{lim}}$ corresponding to $F_{\beta x} = 40 \mu$m;

$\sigma_{H\text{lim}}$ is as specified in ISO 6336-5:1996.

b) For Eh, IF, NT (nitr.), NV (nitr.):

$$\gamma_b = 0.15 F_{\beta x} ; \quad \kappa_b = 0.85$$  \hspace{1cm} (31)

For all speeds, the upper limit is $\gamma_b = 6 \mu$m, corresponding to $F_{\beta x} = 40 \mu$m. When the material of the pinion differs from that of the wheel, $\gamma_{b1}$ and $\kappa_{b1}$ for the pinion, and $\gamma_{b2}$ and $\kappa_{b2}$ for the wheel, shall be determined separately.

The mean of the values:

$$\gamma_b = \frac{\gamma_{b1} + \gamma_{b2}}{2} ; \quad \kappa_b = \frac{\kappa_{b1} + \kappa_{b1}}{2}$$  \hspace{1cm} (32)

is used for the calculation.

5.7.2.4 Determination of the face load factor, $K_{H\beta\cdot C1}$

5.7.2.4.1 Gears with unmodified helices

a) Spur and single helical gears$^4)$:

$$K_{H\beta} = 1 + \frac{4 \times 10^3}{3 \pi} \frac{c_y}{E} \left( \frac{b}{d_1} \right)^2 \left[ 5.12 + \left( \frac{b}{d_1} \right)^2 \left( \frac{l}{b} - \frac{7}{12} \right) \right] + \frac{\kappa_{\beta} c_y f_{ma}}{2 F_m/b}$$  \hspace{1cm} (33)

b) Double helical gears$^4)$ $^5)$:

$$K_{H\beta} = 1 + \frac{4 \times 10^3}{3 \pi} \kappa_{\beta} c_y \left[ 3.2 \left( \frac{2 h_B}{d_1} \right)^2 + \left( \frac{B}{d_1} \right)^4 \left( \frac{l}{B} - \frac{7}{12} \right) \right] + \frac{\kappa_{\beta} c_y f_{ma}}{F_m/b_B}$$  \hspace{1cm} (34)

4) It is assumed that the entire torque is input at one shaft end. If the torque is input at both shaft ends or in between helices of a double helical gear, a more accurate analysis is necessary.

5) The value of $K_{H\beta}$ is for the more severely stressed helix, which is the nearer to the torque end of the pinion; tangential load is divided equally between the two helices; i.e. a small gap width compared to the facewidth $(B - 2 h_B) < 0.5 h_B$. As for the calculation for $K_{H\beta}$, half the tooth width (incorporating half the gap width) is used, and the obtained values are large. Thus, for double helical gears with a large gap width, method C2 of ISO 6336-1 is appropriate in this case, see 5.7.3.
5.7.2.4.2 Gears with modified helices

a) Spur and single helical gears 4)

— with partial helix modification 6) (with compensation for torsional deflection only):

\[ K_{H\beta} = 1 + \frac{4000}{3\pi} \frac{\kappa_{H\beta} \gamma}{E} \left( \frac{b}{d_1} \right)^4 \left( \frac{1}{b} - \frac{7}{12} \right) + \frac{\kappa_{H\beta} \gamma f_{ma}}{2F_m/b} \tag{35} \]

— with full helix modification (with compensation for torsional and bending deflections):

\[ K_{H\beta} = 1 + \frac{\kappa_{H\beta} \gamma f_{ma}}{2F_m/b} \quad \text{and} \quad K_{H\beta} \geq 1.05 \tag{36} \]

b) Double helical gears 4) 5)

— with full helix modification 7) (with compensation for torsional and bending deflections):

\[ K_{H\beta} = 1 + \frac{\kappa_{H\beta} \gamma f_{ma}}{F_m/f_{H\beta}} \quad \text{and} \quad K_{H\beta} \geq 1.05 \tag{37} \]

The validity of equations (33) to (37) depends upon compliance with 5.7.2.1, a) to f).

5.7.3 Face load factor, \( K_{H\beta-C2} \)

5.7.3.1 General

Taken from the basic standard, this method is arranged so that account is taken of the influences on mesh alignment, of elastic deformations of the pinion and of manufacturing inaccuracies.

\( K_{H\beta} \) shall be calculated from the total mesh misalignment after running in; \( F_{by} \), which comprises the following two components.

— **Systematic error** is taken into account by \( f_{sh} \) (mesh misalignment due to shaft deflection). It is primarily caused by pinion shaft deflection, but in principle may include all mechanical deflections able to be evaluated accurately enough in both amount and direction.

— **Random error** is represented by \( f_{ma} \) (mesh misalignment due to manufacturing tolerance). The actual direction and amount of misalignment due to manufacturing cannot be evaluated; only the range is limited by manufacturing tolerance (in reference to gear accuracy grade).

---

6) Torsional deflection can be almost completely compensated for by means of a linear tooth trace or helix angle modification. In addition, crowning is necessary when compensation of bending deflection is required.

7) Full modification of both helices is necessary. Partial modification of the helix angle merely to compensate for torsional deflection is not appropriate for double helical gears which are symmetrically positioned between bearings. Torsional and bending deflections can be almost completely compensated for by means of helix angle modification. However, it is often sufficient if only the helix nearest the torque input end is modified; torsional and bending deflections of the other helix tend to compensate each other. This should be verified.
Application of helix correction and crowning:

- **helix correction** is a lead modification applied to compensate for the systematic error, and while theoretically it is possible to apply a helix correction that exactly matches the calculated deflection for a specific load and so eliminates the \( f_{sh} \) contribution to \( K_{H,\beta} \) for that particular load, in practice, varying loads and errors in the evaluation of \( f_{sh} \) leave a lasting influence on \( K_{H,\beta} \) that has to be taken into account;

- **crowning** is a lead modification comprising the best defensive strategy against the random component of misalignment. Since \( b_{m,\alpha} \) can be in either direction, crowning should be symmetric to the middle of face width.

A more exact and comprehensive analysis in accordance with ISO 6336-1 is recommended if the design does not match the requirements listed in clause 4 or if any of the following items have significant influence on mesh alignment:

- elastic deformations not caused by gear mesh forces but by external loads (e.g. belts, chains, couplings);
- elastic deformations of wheel and wheel shaft;
- elastic deformations and manufacturing inaccuracies of the gear case;
- bearing clearances and deflections;
- arrangements different from those shown in Figure 2;
- any manufacturing or other deformations that indicate the need for a more detailed analysis.

When, by this method, a value of \( K_{H,\beta} \) greater than 2.0 is calculated, the true value will usually be less. However if the calculated value of \( K_{H,\beta} \) is greater than 1.5, the design should be reconsidered (e.g. shaft stiffness increased, bearing positions changed, helix accuracy improved).

### 5.7.3.2 Calculation of \( K_{H,\beta} \)

The specific loading for the calculation of \( K_{H,\beta} \) is \( (F_t K_A K_y)/b \).

If \( (F_t K_A K_y)/b > 100 \) N/mm, then \( F_m/b = (F_t K_A K_y)/b \).

If \( (F_t K_A K_y)/b < 100 \) N/mm, then \( F_m/b = 100 \) N/mm.

\[
K_{H,\beta} = 1 + \frac{F_{py} c_y}{2 F_m/b} \quad \text{applies when } K_{H,\beta} \leq 2 \tag{38}
\]

with the value of \( c_y \) taken from annex A.

Note that \( b \) is the smaller of the facewidths of pinion and wheel measured at the pitch circles. Chamfers or rounding of tooth ends shall be ignored. (For double helical gears, \( b = 2b_B \).)

### 5.7.3.3 Mesh misalignment after running-in, \( F_{py} \)

\[
F_{py} = F_{px} - \beta \tag{39}
\]

where

- \( F_{px} \) is the mesh misalignment before running-in (see 5.7.4);
- \( \gamma_B \) is the running-in allowance (see 5.7.2.3).
5.7.4 Mesh misalignment before running-in, $F_{db}$

The mesh alignment before running-in, $F_{db}$, is the absolute value of the sum of manufacturing deviations, and pinion and shaft deflections, measured in the plane of action.

For gear pairs without verification of the favourable position of the contact pattern:\(^8\):

$$F_{db} = 1,33 B_1 f_{sh} + B_2 f_{ma}$$ \hspace{1cm} (40)

with $B_1$ and $B_2$ taken from Table 4.

For gear pairs with verification of the favourable position of the contact pattern (e.g. by adjustment of bearings):

$$F_{db} = 11,33 B_1 f_{sh} - f_{H5}$$ \hspace{1cm} (41)

where

$f_{H5}$ is the maximum helix slope deviation for ISO accuracy grade 5 (see ISO 1328-1:1995).

By subtracting $f_{H5}$, allowance is made for the compensatory roles of elastic deformation and manufacturing deviations.

### Table 4 — Constants for use in equation (40)

<table>
<thead>
<tr>
<th>No.</th>
<th>Helix modification</th>
<th>Equation constant</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Type</td>
<td>Amount</td>
</tr>
<tr>
<td>1</td>
<td>None</td>
<td>$C_p = 0,5 f_{ma}$</td>
</tr>
<tr>
<td>2</td>
<td>Ventral crowning only</td>
<td>$C_p = 0,5 f_{ma}$</td>
</tr>
<tr>
<td>3</td>
<td>Central crowning only</td>
<td>$C_p = 0,5 (f_{ma} + f_{sh})$</td>
</tr>
<tr>
<td>4(^b)</td>
<td>Helix correction only</td>
<td>Corrected shape calculated to match torque being analysed</td>
</tr>
<tr>
<td>5</td>
<td>Helix correction plus central crowning</td>
<td>Case 2 plus case 4</td>
</tr>
<tr>
<td>6</td>
<td>End relief</td>
<td>Appropriate amount $C_{H5}$</td>
</tr>
</tbody>
</table>

\(^a\) For appropriate crowning, $C_p$, see annex D.

\(^b\) Predominantly applied for applications with constant load conditions.

\(^c\) Valid for very best practice of manufacturing, otherwise higher values appropriate.

5.7.4.1 Minimum values for $K_{H5}$

For gear pairs without helix correction or crowning, the minimum value for $K_{H5}$ is 1,25; for gear pairs with appropriate helix correction and crowning, the minimum value for $K_{H5}$ is 1,10. A favourable contact pattern shall be verified.

\(^8\) With a favourable position of the contact pattern, the elastic deformations and the manufacturing deviations compensate each other (see Figure 1, compensative roles).
IS 8830:2007
ISO 9083:2001

<table>
<thead>
<tr>
<th>Figure</th>
<th>Position of contact pattern</th>
<th>Determination of $F_{px}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>a)</td>
<td>Contact pattern lies towards mid bearing span</td>
<td>$F_{px}$ in accordance with equation (41) (compensative)</td>
</tr>
<tr>
<td>b)</td>
<td>Contact pattern lies away from mid bearing span</td>
<td>$F_{px}$ in accordance with equation (40) (augmentative)</td>
</tr>
<tr>
<td>c)</td>
<td>Contact pattern lies towards mid bearing span</td>
<td>$F_{px}$ in accordance with equation (40) (</td>
</tr>
<tr>
<td>d)</td>
<td>Contact pattern lies away from mid bearing span</td>
<td>$F_{px}$ in accordance with equation (41) (</td>
</tr>
<tr>
<td>e)</td>
<td>Contact pattern lies towards the bearing span</td>
<td>$F_{px}$ in accordance with equation (40) (augmentative)</td>
</tr>
<tr>
<td>f)</td>
<td>Contact pattern lies away from the bearing span</td>
<td>$F_{px}$ in accordance with equation (41) (compensative)</td>
</tr>
</tbody>
</table>

NOTE: Figures a) to d) show the most common mounting arrangement with pinion between bearings. Figures e) to f) show the overhung pinion.

$T^*$ Input or output torqued end, not dependent on direction of rotation.

$B^*$ $B^* = 1$ for spur and single helical gears; $B^* = 1.5$ for double helical gears, the peak load intensity occurs on the helix near to the torqued end.

Figure 1 — Rules for determination of $F_{px}$ with regard to contact pattern position
<table>
<thead>
<tr>
<th>Factor $K'$</th>
<th>with stiffening$^a$</th>
<th>without stiffening$^a$</th>
<th>Figure</th>
<th>Arrangement</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.48</td>
<td>0.8</td>
<td>a)</td>
<td></td>
<td>with $s/l &lt; 0.3$</td>
</tr>
<tr>
<td>$\frac{l}{2}$</td>
<td>$\frac{l}{2}$</td>
<td>T$^*$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

|        | -0.48           | -0.8            | b)     | with $s/l < 0.3$ |
|        | $\frac{l}{2}$ | T$^*$               |        |             |

|        | 1.33            | 1.33             | c)     | with $s/l < 0.3$ |
|        | $\frac{l}{2}$ | T$^*$               |        |             |

|        | -0.36           | -0.6             | d)     | with $s/l < 0.3$ |
|        | $\frac{l}{2}$ | T$^*$               |        |             |

|        | -0.6            | -1.0             | e)     | with $s/l < 0.3$ |
|        | $\frac{l}{2}$ | T$^*$               |        |             |

$^a$ When $d_i/d_{sh} > 1.15$, stiffening is assumed; when $d_i/d_{sh} < 1.15$, there is no stiffening; furthermore, scarcely any or no stiffening at all is to be expected when a pinion slides on a shaft and feather key or similar fitting, nor when normally shrink fitted.

$T^*$ Input or output torqued end, not dependent on direction of rotation.

A dashed line indicates the less deformed helix of a double helical gear. Determine $f_{sh}$ from the diameter in the gaps of double helical gearing mounted centrally between bearings.

Figure 2 — Constant $K'$ to substitute in equations (42) and (43) for calculation of $f_{sh}$
5.7.4.2 Equivalent misalignment, $f_{sh}$

For spur and single helical gears:

$$f_{sh} = \frac{F_m}{b} \, 0.023 \left[ 1 + K \left( \frac{l_s}{d_1^2} \left( \frac{d_1}{d_{sh}} \right)^4 - 0.3 \right) + 0.3 \left( \frac{b}{d_1} \right)^2 \right]$$  \hspace{1cm} (42)

The calculation of $f_{sh}$ for double helical gears is relative to the helix nearest to the shaft end which is driven or which drives the load.

$$f_{sh} = \frac{F_m}{b} \, 0.046 \left[ 1 + K \left( \frac{l_s}{d_1^2} \left( \frac{d_1}{d_{sh}} \right)^4 - 0.3 \right) + 0.3 \left( \frac{b_B}{d_1} \right)^2 \right]$$  \hspace{1cm} (43)

where

- $b = 2b_B$;
- $b_B$ is the width of one helix.

In equations (42) and (43), $K$, $l_s$ and $l$ are according to Figure 2.

In Figure 2, the pinions shown in dashed lines indicate those helices of double helical gears, which have the lower value of $f_{sh}$ and normal shrink fit (for a normal shrink fit, the supporting effect is negligible). The root diameter shall be somewhat greater than the shaft diameter.

5.7.4.3 Misalignment due to manufacturing inaccuracies, $f_{ma}$

The misalignment due to manufacturing inaccuracies $f_{ma}$ equals the helix tolerance $f_{hp}$:

$$f_{ma} = f_{hp}$$  \hspace{1cm} (44)

The greater of the wheel and pinion value should be used.

NOTE As it is theoretically possible that manufacturing tolerances of pinion, wheel and shaft alignment may sum up to the worst case, satisfying load distribution should be verified by e.g. contact pattern control.

5.8 Face load factor, $K_{Fp}$

$$K_{Fp} = k_{H_{Fp}}$$  \hspace{1cm} (45)

a) if $b/h \geq 3$, then

$$N_F = \frac{(b/h)^2}{1 + b/h + (b/h)^2} = \frac{1}{1 + h/b + (h/b)^2}$$  \hspace{1cm} (46)

b) if $b/h < 3$, then

$$N_F = 0.6923$$  \hspace{1cm} (47)
where

\[ b \] is the smaller of the facewidths of pinion and wheel measured at the pitch circles. Chamfers or rounding of tooth ends shall be ignored. For double helical gears the width of one helix, \( \delta_p \), shall be substituted.

\[ h \] is the tooth height from tip to root: \( h = \frac{(d_a - d_i)}{2} \).

### 5.9 Transverse load factors, \( K_{Ha}, K_{Fa} \)

#### 5.9.1 General

The transverse load factors account for the effect of the non-uniform distribution of transverse load between several pairs of simultaneously contacting gear teeth, as follows: \( K_{Ha} \) for surface stress, and \( K_{Fa} \) for tooth-root stress. Method B of ISO 6336-1:1996 is applied.

#### 5.9.2 Determination of the transverse load factors

Equations (48) and (49) are based on the assumption that the base pitch deviations appropriate to the gear accuracy specified are distributed around the circumference of the pinion and wheel, as is consistent with normal manufacturing practice. They do not apply when the gear teeth are intentionally modified.

In the following equations use \( c_\gamma \) from annex A and \( y_\alpha \) from 5.9.4.

- For gears with total contact ratio \( \varepsilon_\gamma \leq 2 \):

\[
K_{Ha} = K_{Fa} = \frac{\varepsilon_\gamma}{2} \left( 0.9 + 0.4 \frac{c_\gamma (f_{pb} - y_\alpha)}{F_{HB}/b} \right)
\]

(48)

- For gears with total contact ratio \( \varepsilon_\gamma > 2 \):

\[
K_{Ha} = K_{Fa} = 0.9 + 0.4 \sqrt{\frac{2(\varepsilon_\gamma - 1)}{\varepsilon_\gamma} \frac{c_\gamma (f_{pb} - y_\alpha)}{F_{HB}/b}}
\]

(49)

In equations (48) and (49), the larger of \( (f_{pb1} - y_{\alpha1}) \) and \( (f_{pb2} - y_{\alpha2}) \) is used.

#### 5.9.3 Limiting conditions for \( K_{Ha} \) and \( K_{Fa} \)

When, in accordance with equations (48) and (49), and

\[
\text{when } K_{Ha} = K_{Fa} > \frac{\varepsilon_\gamma}{\varepsilon_\alpha Z_e^2}, \text{ then for } K_{Ha} \text{ and } K_{Fa}, \text{ substitute } \frac{\varepsilon_\gamma}{\varepsilon_\alpha Z_e^2}
\]

(50)

and when \( K_{Ha} < 1.0, \) and respectively \( K_{Fa} < 1.0, \) then substitute for \( K_{Ha} \) and respectively for \( K_{Fa} \), the limit value 1.0.

It is recommended that the accuracy of helical gears be chosen so that \( K_{Ha} \) and \( K_{Fa} \) are no greater than \( \varepsilon_\alpha \). As a consequence, it may be necessary to limit the base pitch deviation tolerances of gears of coarse quality grade.
5.9.4 Running-in allowance, \( \gamma_a \)

a) For St, V:

\[
\gamma_a = \frac{160}{\sigma_{H\text{lim}}} f_{pb}
\]

- if \( v \leq 5 \text{ m/s} \), no restriction;
- if \( 5 \text{ m/s} < v \leq 10 \text{ m/s} \), the upper limit of \( \gamma_a = 12800/\sigma_{H\text{lim}} \), corresponding to \( f_{pb} < 80 \mu m \);
- if \( v > 10 \text{ m/s} \), the upper limit of \( \gamma_a = 6400/\sigma_{H\text{lim}} \), corresponding to \( f_{pb} < 40 \mu m \).

b) For Eh, IF, NT (nitr.) et NV (nitr.):

\[
\gamma_a = 0.075 f_{pb}
\]

for all speeds with the restriction: the upper limit of \( \gamma_p = 3 \mu m \), corresponding to \( f_{pb} = 40 \mu m \).

6 Calculation of surface durability (pitting)

6.1 Basic formulae

6.1.1 General

The calculation of surface durability is based on the contact stress, \( \sigma_H \), at the pitch point or at the inner (lowest) point of single pair tooth contact. The higher of the two values obtained shall be used to determine capacity. Contact stress, \( \sigma_H \), and the permissible contact stress, \( \sigma_{HP} \), shall be calculated separately for wheel and pinion; \( \sigma_H \) shall be \( \leq \sigma_{HP} \).

6.1.2 Determination of contact stress, \( \sigma_{H0} \), for the pinion

Contact stress, \( \sigma_{H0} \), for the pinion is calculated as follows:

\[
\sigma_{H0} = Z_B Z_{\text{HP}} Z_k Z_{\text{HV}} Z_{\text{Hv}} Z_{\text{H0}} \leq \sigma_{HP}
\]

with

\[
\sigma_{H0} = Z_H Z_{E} Z_{e} Z_{\beta} \frac{F_{t}}{d_{y} u + 1} \sqrt{u} \quad \text{(use the negative sign for internal gears)}
\]

where

- \( \sigma_{H0} \) is the nominal contact stress at the pitch point; this is the stress induced in flawless (error free) gearing by application of static nominal torque;
- \( b \) is the facewidth (for a double helical gear \( b = 2b_0 \)), and the value \( b \) of mating gears is the smaller of the facewidths at the pitch circles of pinion and wheel, ignoring any intentional transverse chamfers or tooth-end rounding; neither unhardened portions of surface-hardened gear tooth flanks nor the transition zones, shall be included;
- \( Z_B \) is the single pair tooth contact factor for the pinion (see 6.2).
6.1.3 Determination of contact stress, \( \sigma_H \), for the wheel

Contact stress, \( \sigma_H \), for the pinion is calculated as

\[
\sigma_H = Z_D \sigma_{H0} \sqrt{K_A K_Y K_{HA}} \leq \sigma_{HP}
\]

(55)

where

\( Z_D \) is the single pair tooth contact factor for the wheel (see 6.2).

The total tangential load in the case of gear trains with multiple transmission paths, planetary gear systems, or split-path gear trains is not quite evenly distributed over the individual meshes (depending on design, tangential speed and manufacturing accuracy). This shall be taken into consideration by substituting \( K_Y K_A \) for \( K_A \) in equation (53) and equation (55) to adjust the average tangential load per mesh as necessary; see clause 5.

6.1.4 Determination of permissible contact stress, \( \sigma_{HP} \), for long life

In this International Standard, Method B of ISO 6336-2:1996 is used.

\[
\sigma_{HP,ref} = \frac{\sigma_{H,lim}}{S_{H,ref}} Z_L Z_V Z_R Z_W Z_X = \frac{\sigma_{HG}}{S_{H,min}}
\]

(56)

The permissible contact stress (long life) shall be derived from equation (56), with the influence factors \( \alpha_{H,lim} \), \( S_{H,ref} \), \( Z_L \), \( Z_V \), \( Z_R \), \( Z_W \) and \( Z_X \) calculated according to this International Standard. However, according to ISO 6336-2, the values of \( \alpha_{H,lim} \) are validated for \( N_L = 5 \times 10^7 \) load cycles (for St, V, Eh) or \( 2 \times 10^6 \) load cycles [for If, NT (nitr.), NV (nitr.), NV (nitrocar.)]. This number is likely to be exceeded in the life of a marine gear. If this is not the case, refer to ISO 6336-2 for the limited life range. Nevertheless, values of \( \sigma_{HP,ref} \) derived from equation (56) may be substituted for \( \sigma_{HP} \), given optimum conditions, material, lubrication, manufacturing and experience; otherwise the values for \( \sigma_{HP} \) are obtained for material quality MQ according to ISO 6336-5:1996 using equation (57):

For St, V, Eh:

\[
\sigma_{HP} = 0.92 \sigma_{HP,ref} \left( \frac{10^{10}}{N_L} \right)^{0.0157} \frac{\sigma_{HG}}{S_{H,ref}}
\]

(57)

For If, NT (nitr.), NV (nitr.), NV (nitrocar.):

\[
\sigma_{HP} = 0.92 \sigma_{HP,ref} \left( \frac{10^{10}}{N_L} \right)^{0.0098} \frac{\sigma_{HG}}{S_{H,ref}}
\]

6.1.5 Safety factor for surface durability, \( S_H \)

\( S_H \) shall be calculated separately for the pinion and wheel:

\[
S_H = \frac{\sigma_{HG}}{\sigma_H} > S_{H,ref}
\]

(58)

with \( \sigma_{HG} \) for endurance according to equation (57); \( \sigma_H \) shall be in accordance with equation (53) for the pinion, and with equation (55) for the wheel (see 6.1.1).
NOTE This is the calculated safety factor with regard to contact stress (Hertzian pressure). The corresponding factor relative to torque capacity is equal to the square of \( S_{w_i} \).

For the minimum safety factor for surface durability, \( S_{w_{min}} \), and probability of failure, see 4.1.3 of ISO 6336-1:1996.

### 6.2 Single pair tooth contact factors, \( Z_B, Z_D \)

When \( Z_B > 1 \) or \( Z_D > 1 \), the factors \( Z_B \) and \( Z_D \) are used to transform the contact stress at the pitch point of spur gears to the contact stress at the inner (lowest) limit of the single pair tooth contact of the pinion or the wheel. See 6.1.1.

a) Internal gears

\( Z_D \) is always to be taken as unity.

b) Spur gears

Determine \( M_1 \) (quotient of \( \rho_{relC} \) at the pitch point \( \rho_{relB} \) at the inner limit (lowest point) of single tooth pair contact of the pinion) and \( M_2 \) (quotient of \( \rho_{relC} \) by \( \rho_{relD} \) of the wheel) from:

\[
M_1 = \frac{\tan \alpha_w}{\sqrt{\frac{d_{a1}^2}{d_{b1}^2} - 1} - \frac{2\pi}{\zeta_1}} \frac{\frac{d_{a2}^2}{d_{b2}^2} - 1 - (\epsilon_{\alpha} - 1) \frac{2\pi}{\zeta_2}}{\frac{d_{a1}^2}{d_{b1}^2} - 1 - (\epsilon_{\alpha} - 1) \frac{2\pi}{\zeta_1}}
\]

(59)

\[
M_2 = \frac{\tan \alpha_w}{\sqrt{\frac{d_{a2}^2}{d_{b2}^2} - 1} - \frac{2\pi}{\zeta_2}} \frac{\frac{d_{a1}^2}{d_{b1}^2} - 1 - (\epsilon_{\alpha} - 1) \frac{2\pi}{\zeta_2}}{\frac{d_{a2}^2}{d_{b2}^2} - 1 - (\epsilon_{\alpha} - 1) \frac{2\pi}{\zeta_2}}
\]

(60)

(See 6.5.2 for the calculation of the profile contact ratio \( \epsilon_{\alpha} \).)

If \( M_1 > 1 \), then \( Z_B = M_1 \); if \( M_1 < 1 \), then \( Z_B = 1.0 \).

If \( M_2 > 1 \), then \( Z_D = M_2 \); if \( M_2 < 1 \), then \( Z_D = 1.0 \).

c) Helical gears with \( \epsilon_{\beta} \geq 1 \)

\( Z_B = Z_D = 1 \)

d) Helical gears with \( \epsilon_{\beta} < 1 \)

\( Z_B \) and \( Z_D \) are determined by linear interpolation between the values for spur and helical gearing with \( \epsilon_{\beta} \geq 1 \):

\[
Z_B = M_1 - \epsilon_{\beta} (M_1 - 1) ; Z_B \geq 1
\]
\[
Z_D = M_2 - \epsilon_{\beta} (M_2 - 1) ; Z_D \geq 1
\]

(61)

If \( Z_B \) or \( Z_D \) are set to unity, the contact stresses calculated using equations (53) or (55) are the values for the contact stress at the pitch cylinder.

The methods in 6.2 apply to the calculation of contact stress when the pitch point lies in the path of contact. If the pitch point is determinant and lies outside the path of contact, then \( Z_B \) and / or \( Z_D \) both shall be determined for
contact at the adjacent tip circle. For helical gears when \( \varepsilon_\beta \) is less than 1.0, \( Z_B \) and \( Z_D \) shall be determined by linear interpolation between the values (determined at the pitch point or at the adjacent tip circle as appropriate) for spur gears and those helical gears with \( \varepsilon_\beta \geq 1 \).

### 6.3 Zone factor, \( Z_H \)

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

\[
Z_H = \frac{2 \cos \beta \cos \alpha \cos \omega t}{\cos^2 \alpha \sin \omega t}
\]  

(62)

### 6.4 Elasticity factor, \( Z_F \)

The elasticity factor, \( Z_E \), takes into account the influences of the material properties \( E \) (modulus of elasticity) and \( \nu \) (Poisson's ratio) on the contact stress. As this International Standard is only applicable to steel gears, \( Z_E \) is fixed to

\[ Z_H = 189.8 \]  

(63)

### 6.5 Contact ratio factor, \( Z_c \)

#### 6.5.1 General

The contact ratio factor, \( Z_c \), accounts for the influence of the transverse contact and overlap ratios on the surface load capacity of cylindrical gears.

a) Spur gears:

\[
Z_c = \sqrt{\frac{4 - \varepsilon_\alpha}{3}}
\]  

(64)

The conservative value of \( Z_c = 1.0 \) may be chosen for spur gears having a contact ratio less than 2.0.

b) Helical gears:

If \( \varepsilon_\beta < 1 \), then

\[
Z_c = \sqrt{\frac{4 - \varepsilon_\alpha}{3} (1 - \varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}}
\]  

(65)

If \( \varepsilon_\beta \geq 1 \), then

\[
Z_c = \sqrt{\frac{1}{\varepsilon_\alpha}}
\]  

(66)

#### 6.5.2 Transverse contact ratio, \( \varepsilon_\alpha \)

\[
\varepsilon_\alpha = k_d p_{bt}
\]  

(67)

with length of path of contact:
\[ r_a = \frac{1}{2} \left[ \sqrt{d_{a1}^2 - d_{b1}^2} \pm \sqrt{d_{a2}^2 - d_{b2}^2} \right] - a \sin \alpha_{wt} \]  

(68)

and transverse base pitch:

\[ p_{bt} = m_t \pi \cos \alpha_t \]  

(69)

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

Equation (69) is only valid if the path of contact is effectively limited by the tip circle of the pinion and the wheel and not, for example, by undercut tooth profiles.

6.5.3 Overlap ratio, \( \varepsilon_{\beta} \)

\[ \varepsilon_{\beta} = \frac{b \sin \beta}{\pi m_n} \]  

(70)

See 6.1.2 for the definition of facewidth.

6.6 Helix angle factor, \( Z_{\beta} \)

The helix angle factor, \( Z_{\beta} \), takes account of the influence on surface stress of the helix angle.

\[ Z_{\beta} = \sqrt{\cos \beta} \]  

(71)

6.7 Allowable stress numbers (contact), \( \sigma_{H \text{lim}} \)

ISO 6336-5 provides information on commonly used gear materials, methods of heat treatment, and the influence of gear quality on values for allowable stress numbers, \( \sigma_{H \text{lim}} \), derived from test results of standard reference test gears.

See, too, ISO 6336-5 for requirements concerning material and heat treatment for qualities ML, MQ, ME and MX. Material quality MQ shall be chosen for marine gears, unless otherwise agreed.

6.8 Influences on lubrication film formation, \( Z_L, Z_{\nu} \) and \( Z_R \)

6.8.1 General

As described in ISO 6336-2, \( Z_L \) accounts for the influence of nominal viscosity of the lubricant, \( Z_{\nu} \), for the influence of tooth-flank velocities, and \( Z_R \) for the influence of surface roughness on the formation of the lubricant film in the contact zone. Method B of ISO 6336-2:1996 is used in this International Standard.

Factors shall be determined for the softer material when the hardness of meshing gears is different.

6.8.2 Lubricant factor, \( Z_L \)

\( Z_L \) can be calculated using equations (72) to (75):

\[ Z_L = C_{ZL} + \frac{4}{2} \left( \frac{1.0 - C_{ZL}}{2} \right) \]  

(72)

a) If \( \sigma_{H \text{lim}} < 850 \, \text{MPa} \), then
$C_{ZL} = 0.83$  

b) If $850 \text{ N/mm}^2 \leq \sigma_{H\text{lim}} \leq 1200 \text{ N/mm}^2$, then

$$C_{ZL} = \frac{\sigma_{H\text{lim}}}{4.375} + 0.6357 \quad (74)$$

c) If $\sigma_{H\text{lim}} > 1200 \text{ N/mm}^2$, then

$$C_{ZL} = 0.91 \quad (75)$$

Alternatively, $Z_L$ can be calculated from equation (76):

$$Z_L = C_{ZL} + 4(1.0 - C_{ZL}) \nu_1 \quad (76)$$

where

$$\nu_1 = \frac{1}{(1.2 + 80 \nu_50)^2}$$

using the viscosity parameters from Table 5.

### Table 5 — Viscosity parameters

<table>
<thead>
<tr>
<th>ISO viscosity class</th>
<th>VG 32&lt;sup&gt;a&lt;/sup&gt;</th>
<th>VG 46&lt;sup&gt;a&lt;/sup&gt;</th>
<th>VG 68&lt;sup&gt;a&lt;/sup&gt;</th>
<th>VG 100</th>
<th>VG 150</th>
<th>VG 220</th>
<th>VG 320</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal viscosity</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\nu_{40}$ mm²/s</td>
<td>32</td>
<td>46</td>
<td>68</td>
<td>100</td>
<td>150</td>
<td>220</td>
<td>320</td>
</tr>
<tr>
<td>$\nu_{50}$ mm²/s</td>
<td>21</td>
<td>30</td>
<td>43</td>
<td>61</td>
<td>89</td>
<td>125</td>
<td>180</td>
</tr>
<tr>
<td>Viscosity parameter</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\nu_5$</td>
<td>0.040</td>
<td>0.067</td>
<td>0.107</td>
<td>0.158</td>
<td>0.227</td>
<td>0.295</td>
<td>0.370</td>
</tr>
</tbody>
</table>

<sup>a</sup> Only for high speed transmission.

6.8.3 Speed factor, $Z_v$

$Z_v$ can be calculated using equations (77) and (78):

$$Z_v = C_{Zv} + \frac{2(1.0 - C_{Zv})}{\sqrt{0.8 + \frac{32}{\nu^2}}} \quad (77)$$

where

$$C_{Zv} = C_{ZL} + 0.02 \quad (78)$$

see equations (73) to (75) for the values of $C_{ZL}$.

Alternatively, $Z_v$ can be calculated from equation (79):

$$Z_v = C_{Zv} + 2(1.0 - C_{Zv}) \nu_p \quad (79)$$

where the velocity parameter $\nu_p = 1/(0.8 + 32/\nu)^{0.5}$
6.8.4 Roughness factor, \( Z_R \)

6.8.4.1 Calculation of \( Z_R \)

\( Z_R \) may be calculated using the following equations:

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

where

\[
Z_R = \left[ \frac{1.29 a^{1.3}}{R_{z1} + R_{z2}} \right]^{C_{ZR}}
\]

6.8.4.2 Roughness values

\[
R_z = \frac{R_{z1} + R_{z2}}{2}
\]

\( R_{z1,2} \) is measured on several tooth flanks. The mean roughness \( R_{z1} \) (pinion flank) and the mean roughness \( R_{z2} \) (wheel flank) shall be determined for their surface condition after manufacture, including any running-in treatment planned as a manufacturing, commissioning or in-service process, when safe to assume that this will take place. If the stated roughness is an \( Ra \) value (= CLA value; = AA value), the following approximation may be used for the conversion:

\[
Ra = CLA = AA = \frac{R_z}{6}
\]

\[
R_{z10} = R_z \left( \frac{10}{P_{red}} \right)
\]

\[
P_{red} = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}
\]

where

\[
\rho_{1,2} = 0.5 d_{b1,2} \tan \alpha_1
\]

(also applicable for internal gears, \( d_b \) then being negative sign)

6.8.4.3 Material dependent index, \( C_{ZR} \)

a) If \( \sigma_{H, lim} < 850 \text{ N/mm}^2 \), then

\[
C_{ZR} = 0.15
\]

b) If \( 850 \text{ N/mm}^2 \leq \sigma_{H, lim} \leq 1200 \text{ N/mm}^2 \), then

\[
C_{ZR} = 0.32 - 0.0002 \sigma_{H, lim}
\]
6.9 Work hardening factor, $Z_W$

As described in ISO 6336-2, the work hardening factor, $Z_w$, takes account of the increased surface durability due to the meshing of a steel wheel (structural steel, through-hardened steel) with a pinion significantly (= 200 HV or more) harder than the wheel and having smooth tooth flanks ($R_z \leq 6 \, \mu m$, otherwise effects of wear are not covered by this International Standard). Method B of ISO 6336-2:1996 is applied, as follows.

If $HB < 130$, then

$$Z_W = 1.2$$

(90)

If $130 \leq HB \leq 470$, then

$$Z_W = 1.2 - \frac{HB - 130}{1700}$$

(91)

If $HB > 470$, then

$$Z_W = 1.0$$

(92)

where $HB$ is the Brinell hardness of the tooth flanks of the softer gear of the pair.

6.10 Size factor, $Z_X$

By means of $Z_x$, account is taken of statistical evidence indicating that the stress levels at which fatigue damage occurs decrease with an increase of component size (larger number of weak points in structure), as a consequence of the influence on subsurface defects of the smaller stress gradients that occur (theoretical stress analysis) and the influence of size on material quality (effect on forging process, variations in structure, etc.). Important influence parameters are:

a) material quality (furnace charge, cleanliness, forging);

b) heat treatment, depth of hardening, distribution of hardening;

c) radius of flank curvature;

d) module: in the case of surface hardening; depth of hardened layer relative to the size of teeth (core supporting-effect).

For through-hardened gears and for surface-hardened gears with adequate case depth relative to tooth size and radius of relative curvature, the size factor, $Z_x$, is taken to be 1.0.

7 Calculation of tooth bending strength

7.1 Basic formulae

7.1.1 General

As specified in ISO 6336-3, the maximum tensile stress at the tooth-root may not exceed the permissible bending stress for the material. This is the basis for rating the bending strength of gear teeth.
The actual tooth-root stress, $\sigma_F$, and the permissible bending stress, $\sigma_{FP}$, shall be calculated separately for pinion and wheel; $\sigma_F$ shall be less than $\sigma_{FP}$.

### 7.1.2 Determination of tooth root stress $\sigma_F$

Method B of ISO 6336-3:1996 is used in this International Standard.

Tooth root stress, $\sigma_F$, is calculated as follows:

$$\sigma_F = \sigma_{F0} K_A K_v K_{FB} K_{Fa} \frac{F_t}{b m_n}$$

(93)

with

$$\sigma_{F0} = \frac{F_t}{b m_n} Y_F Y_s Y_p$$

(94)

In the case of gear trains with multiple transmission paths, planetary gear systems or split-path gear trains, the total tangential load is not quite evenly distributed over the individual meshes (depending on design, tangential speed and manufacturing accuracy). This shall be taken into consideration by substituting $K_A K_v$ for $K_A$ in equation (93) to adjust the average tangential load per mesh as necessary (see clause 5).

When the facewidth $b$ (for a double helical gear $b = 2 b_B$) is larger than that of its mating gear, the bending strength of its teeth shall be based on the smaller facewidth plus a length, not exceeding one module of any extension at each end. However, if it is foreseen that, because of crowning or end relief, contact does not extend to the end of face, then the smaller facewidth shall be used for both pinion and wheel. Facewidth $b$ is the facewidth at the root cylinder of the gear.

### 7.1.3 Determination of permissible tooth root stress, $\sigma_{FP}$

$$\sigma_{FP} = \frac{\sigma_{FE} Y_{rel} Y_{rel} Y_{X}}{S_{F min}}$$

(95)

According to ISO 6336-3, the values of $\sigma_{F lim}$ and $\sigma_{FE}$ are validated for $N_L = 3 \times 10^6$ load cycles. This number is likely to be exceeded in the life of a marine gear. If this is not the case, refer to ISO 6336-3 for the limited life range. Nevertheless, values of $\sigma_{FP}$ derived from equation (95) may be substituted for $\sigma_{FP}$, given optimum conditions, material, manufacturing and experience; otherwise the value for $\sigma_{FP}$ is obtained by equation (96).

$$\sigma_{FP} = 0.92 \sigma_{FP} \left( \frac{10^{10}}{N_L} \right)^{0.01} \frac{\sigma_{FG}}{S_{F min}}$$

(96)

### 7.1.4 Safety factor for bending strength, $S_F$

The factor $S_F$ shall be calculated using the following equation:

$$S_F = \frac{\sigma_{FG}}{\sigma_F} \geq S_{F min}$$

(97)

$S_F$ is calculated separately for pinion and wheel, with $\sigma_{FG}$ calculated in accordance with equation (95) or (96) as appropriate, and $\sigma_F$ obtained with equation (93).

More information on the safety factor and probability of failure can be found in ISO 6336-1:1996, 1.3.
7.2 Form factor, $Y_F$

7.2.1 General

$Y_F$ is the form factor by means of which the influence of tooth form on nominal bending stress is taken into account. $Y_F$ is relevant to the application of load at the outer limit of single pair tooth contact. (method B of ISO 6336-3:1996).

Values of $Y_F$ are determined for spur gears and the virtual spur gears of helical gears. Virtual spur gears have the virtual number of teeth $Z_n$. See 7.2.4 for the calculation of $Z_n$ and other virtual gear parameters.

$Y_F$ shall be determined separately for wheel and pinion from the following equation (see Figure 3).

$$Y_F = \frac{6h_{Fe} \cos \alpha_{Fen}}{m_n} \left( \frac{F_{bn}}{b \cos \beta_n} \right)^2 \cos \alpha_n$$

Figure 3 — Determination of dimensions of tooth-root chord at the critical section

The equations given here apply to all basic rack tooth profiles with and without undercut, but with the following restrictions:

a) the contact point of the 30° tangent lies on the tooth-root fillet;

b) the basic rack profile of the gearing has a root fillet;

c) the teeth were generated using tools such as hobs or planer-cutters having rack form teeth.

a  Base circle.
7.2.2 Parameters required for the determination of $Y_F$

First, determine the auxiliary values $E$, $G$ and $H$:

$$E = \frac{\pi}{4} m_n - h_{Ip} \tan \alpha_n + \frac{S_{pr}}{\cos \alpha_n} - \frac{r_{Ip}}{\cos \alpha_n} (1 - \sin \alpha_n)$$ (99)

where

$r_{pr} = pr - q$ (see Figure 4);

$r_{pr} = 0$, when gears are not undercut (see Figure 4)

$$G = \frac{r_{Ip}}{m_n} - \frac{h_{Ip}}{m_n} + \frac{z_n}{m_n}$$ (100)

$$H = \frac{2}{z_n} \left( \frac{\pi - E}{2 m_n} - \frac{\pi}{3} \right)$$ (101)

Next, use $G$ and $H$ together with $\theta = \pi/6$ as a seed value (on the right hand side) in equation (102).

$$\theta = \frac{2G}{z_n} \tan \theta - H$$ (102)

Use the newly calculated $\theta$ and again apply equation (102). Continue using equation (102) until there is no significant change in successive values of $\theta$. Generally, the function converges after two or three iterations of equation (102). Use this final value of $\theta$ in equations (103), (104) and (105).
Tooth-root normal chord, \( s_{Fn} \):

\[
\frac{s_{Fn}}{m_n} = \frac{z_n \sin \left( \frac{\pi}{3} - \theta \right) + \sqrt{3} \left( \frac{G}{\cos \theta} - \frac{\rho_{FP}}{m_n} \right)}{m_n}
\]

Radius of root fillet, \( \rho_F \):

\[
\frac{\rho_F}{m_n} = \frac{\rho_{FP}}{m_n} + \frac{2G^2}{\cos \theta (z_n \cos^2 \theta - 2G)}
\]

Bending moment arm, \( h_{Fe} \):

\[
\alpha_{en} = \arccos \left( \frac{d_{en}}{d_{en}} \right)
\]

\[
\gamma_e = \frac{0.5 \pi + 2 \times x \times \tan \alpha_n - \arccos \alpha_n}{z_n}
\]

\[
\alpha_{Fen} = \alpha_{en} - \gamma_e = \frac{0.5 \pi + 2 \times x \times \tan \alpha_n}{z_n}
\]

\[
\frac{h_{Fe}}{m_n} = 0.5 \left[ \cos \gamma_e \tan \alpha_{Fen} - \sin \gamma_e \tan \alpha_{Fen} \right]
\]

\[
= \frac{d_{en}}{m_n} - \frac{z_n \cos \left( \frac{\pi}{3} - \theta \right) - \frac{G}{\cos \theta} + \frac{\rho_{FP}}{m_n}}{m_n}
\]

7.2.3 Internal gears

It is assumed that the value of the form factor of a special rack can be substituted as an approximate value of the form factor of an internal gear. The profile of such a rack should be a version of the basic rack profile, modified in such a way that it would generate the normal profile, including tip and root circles, of an exact counterpart gear of the internal gear. The load direction angle is \( \alpha_n \) (see Figure 5).

Figure 5 — Parameters for determination of form factor, \( Y_F \), of an internal gear
The values to be used in equation (98) are determined as follows:

Tooth-root normal chord, $s_{Fn2}$:

$$s_{Fn2} = \frac{2}{mn} \left[ \frac{\pi}{4} + \frac{\rho_{IP2}}{m_n} \tan \alpha_n + \frac{\rho_{IP2} - \rho_{IPr}}{m_n \cos \alpha_n} - \frac{\rho_{IP2} \cos \pi}{m_n} \right]$$

(109)

where

- $\rho_{IP2}$ is tool radius (see below).

Bending moment arm, $h_{Fe2}$:

$$\frac{h_{Fe2}}{mn} = \frac{d_{en2}^2 - d_{tn2}^2}{2 m_n} \left[ \frac{\pi}{4} \left( \frac{h_{IP2}}{m_n} - \frac{d_{en2}^2 - d_{tn2}^2}{2 m_n} \right) \tan \alpha_n \right] \tan \alpha_n - \frac{\rho_{IP2}}{m_n} \left( 1 - \sin \frac{\pi}{6} \right)$$

(110)

where

- $\rho_{IP2}$ is tool radius (see below);
- $d_{en2}$ is derived from equation (121) adding the subscript 2;
- $d_{tn2}$ is derived in the same way as $d_{an}$ [equation (121), note that $d_{an2} - d_{2} = d_{n2} - d_{2}$].

Obtain $h_{IP2}$ from equation (111), refer to equation (113) and related information for $\rho_{IP2}$.

$$h_{IP2} = \frac{d_{n2} - d_{tn2}}{2}$$

(111)

Root fillet radius, $\rho_{F2}$, tool radius, $\rho_{IP2}$:

When the root fillet radius, $\rho_{F2}$, is known, it shall be used. Otherwise:

$$\rho_{F2} = \rho_{IP2} = \frac{c_p}{1 - \sin \alpha_n} = \frac{h_{IP2} - h_{N2}}{1 - \sin \alpha_n} = \frac{d_{N2} - d_{12}}{2(1 - \sin \alpha_n)}$$

(112)

($d_{N2}$ represents the diameter of a circle near the tooth-root, containing the limits of the usable flanks of an internal gear).

If sufficient data are not available, the following approximation may be used:

$$\rho_{F2} = \rho_{IP2} = 0,15m_n$$

(113)

Ensure that the correct sign is used; see the footnote in Table 1.

7.2.4 Parameters for the virtual gear

$$\beta_B = \arccos \sqrt{1 - (\sin \beta \cos \alpha_n)^2}$$

(114)

$$z_n = \frac{z}{\cos^2 \beta_B \cos \beta}$$

(115)
The value of $z$ is positive for external gears and negative for internal gears (see clause 3, footnote 2).

### 7.3 Stress correction factor, $Y_S$

The stress correction factor, $Y_S$, is used to convert the nominal bending stress to local tooth root stress. $Y_S$ shall be determined separately for pinion and wheel. $Y_S$ is valid in the range $1 \leq q_s < 8$.

$$Y_S = (1,2 + 0.13 \cdot L) \cdot q_s$$  \hfill (123)

where

$$L = \frac{S_Fn}{h_{Fe}}$$  \hfill (124)

$$q_s = \frac{S_Fn}{2 \cdot \rho_F}$$  \hfill (125)

with

- $S_Fn$ from equation (103) for external gears, equation (109) for internal gears;
- $h_{Fe}$ from equation (108) for external gears, equation (110) for internal gears;
- $\rho_F$ from equation (104) for external gears, equation (113) for internal gears.
7.4 Helix angle factor, $Y_\beta$

The tooth-root stress of a virtual spur gear, calculated as a preliminary value, is converted by means of the helix factor, $Y_\beta$, to that of the corresponding helical gear. By this means, the oblique orientation of the lines of mesh contact is taken into account (lesser tooth-root stress).

If $\varepsilon_\beta > 1$ and $\beta < 30^\circ$, then

$$Y_\beta = 1 - \frac{\beta}{120^\circ}$$

If $\varepsilon_\beta > 1$ and $\beta > 30^\circ$, then

$$Y_\beta = 0.75$$

If $\varepsilon_\beta < 1$ and $\beta < 30^\circ$, then

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

If $\varepsilon_\beta < 1$ and $\beta > 30^\circ$, then

$$Y_\beta = 1 - 0.25 \varepsilon_\beta$$

7.5 Tooth-root reference strength, $\sigma_{FE}$

ISO 6336-5 provides information on values of $\sigma_{fe}$, $\sigma_{im}$ and $\sigma_{FE}$ for the more popular gear materials. The requirements for heat treatment processes and material quality for quality grades ML, MQ and ME are also included.

The quality MQ shall be used for marine gears unless otherwise agreed.

7.6 Relative notch sensitivity factor, $Y_{\delta rel T}$

$Y_{\delta rel T}$ approximately indicates the overstress tolerance of the material in the root fillet region. Method B of ISO 6336-3:1996 is used in this International Standard.

$$Y_{\delta rel T} = \frac{1 + \sqrt{\rho \chi}}{1 + \sqrt{\rho \chi_T}}$$

where

- $\rho$ is the slip-layer thickness taken from Table 6 as a function of the material;
- $\chi_T$ is the value for the standard reference test gear: $\chi_T = 1.2$;
- $\chi'$ is the relative stress gradient calculated using the following equation:

$$\chi' = \frac{\sigma_{FE}}{Y_{\delta rel T}}$$

9) Applies for module $m = 5$ mm. The influence of size is covered by the factor $Y_X$ (see 7.8).
\[ \chi^* = 0.2(1+2q_s) \]  

where

\[ q_s \] is the notch parameter obtained from equation (125).

Table 6 — Values for slip-layer thickness \( \rho' \)

<table>
<thead>
<tr>
<th>Material (^a)</th>
<th>( \rho' ) mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>NT (nitr.), NV (nitr.), NV (nitrocar.)</td>
<td>0.1005</td>
</tr>
<tr>
<td>V</td>
<td></td>
</tr>
<tr>
<td>yield point ( \sigma_s = 500 \text{ N/mm}^2 )</td>
<td>0.0281</td>
</tr>
<tr>
<td>yield point ( \sigma_s = 600 \text{ N/mm}^2 )</td>
<td>0.0194</td>
</tr>
<tr>
<td>limit of proportionality ( \sigma_{p,2} = 800 \text{ N/mm}^2 )</td>
<td>0.0064</td>
</tr>
<tr>
<td>limit of proportionality ( \sigma_{p,2} = 1000 \text{ N/mm}^2 )</td>
<td>0.0014</td>
</tr>
<tr>
<td>Eh, If</td>
<td>0.0030</td>
</tr>
</tbody>
</table>

\(^a\) See Table 2 for an explanation of abbreviations used.

7.7 Relative surface factor, \( Y_{\text{R rel T}} \)

The surface factor, \( Y_{\text{R rel T}} \), accounts for the influence on tooth-root stress of the surface condition in the tooth-roots. Primarily, this is dependent on surface roughness in the tooth-root fillets.

The influence of surface condition on tooth-root bending strength does not depend solely on the surface roughness in the tooth-root fillets, but also on the size and shape (the problem of 'notches within a notch'). This subject has not been sufficiently well studied to date for it to be taken into account in this International Standard. The method applied here is only valid when scratches or similar defects deeper than \( 2 \times R_z \) are not present.

NOTE \( 2 \times R_z \) is the preliminary estimated value.

Besides surface texture, other known influences on tooth bending strength include residual compressive stresses (shot peening), grain boundary oxidation and chemical effects. When fillets are shot peened, perfectly shaped, or both, a value slightly greater than that obtained from the graph should be substituted for \( Y_{\text{R rel T}} \). When grain boundary oxidation or chemical effects are present, a smaller value than that indicated by the graph should be substituted for \( Y_{\text{R rel T}} \).

\( Y_{\text{R rel T}} \) when \( R_z < 1 \mu \text{m} \)

a) For V, Eh, IF when \( R_z < 1 \mu \text{m} \)

\[ Y_{\text{R rel T}} = 1.12 \]  

b) for NT (nitr.), NV (nitr.), NV (nitrocar.) when \( R_z < 1 \mu \text{m} \)

\[ Y_{\text{R rel T}} = 1.025 \]

c) for V, Eh, IF if \( R_z \geq 1 \mu \text{m} \)

\[ Y_{\text{R rel T}} = 1.674 - 0.529 (R_z + 1)^{0.1} \]  

\[ (132) \]

\[ (133) \]
7.8 Size factor, $Y_X$

$Y_X$ is used to allow for the influence of size on

- the probable distribution of weak points in the material structure,
- the stress gradients that in materials theory decrease with increasing dimensions,
- material quality, and
- as regards the quality of forging, presence of defects, etc.

$Y_X$ is calculated in accordance with Table 7.

<table>
<thead>
<tr>
<th>Table 7 — Size factor (root), $Y_X$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
</tr>
<tr>
<td>----------</td>
</tr>
<tr>
<td>V</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Eh, IF,</td>
</tr>
<tr>
<td>NT (nitr.)</td>
</tr>
<tr>
<td>NV (nitr.)</td>
</tr>
<tr>
<td>NV (nitrocar.)</td>
</tr>
</tbody>
</table>

*a* See Table 2 for an explanation of the abbreviations used.
Annex A
(normative)

Tooth stiffness parameters $c'$ and $c_Y$

A.1 General

A tooth stiffness parameter represents the requisite load over 1 mm facewidth, directed along the line of action\(^{10}\) to produce in line with the load, the deformation amounting to 1 \(\mu\)m, of one or more pairs of deviation-free teeth in contact.

Single stiffness, $c'$, is the maximum stiffness of a single-tooth pair of a spur gear pair. It is approximately equal to the maximum stiffness of a tooth pair in single pair contact\(^{11}\). Single stiffness $c'$ for helical gears is the maximum stiffness normal to the helix of one tooth pair.

Mesh stiffness, $c_p$, is the mean value of stiffness of all the teeth in a mesh.

Method B from ISO 6336-1:1996, used in this International Standard, is applicable in the range $x_1 \geq x_2 \leq 2$.

A.2 Single stiffness $c'$

A.2.1 Calculation of $c'$

For specific loading, $F_t \frac{K_A}{b} \geq 100 \text{ N/mm}^2$:

$$c' = 0.8 \frac{c'b}{b} C_R C_B \cos \beta$$  \hspace{1cm} (A.1)

A.2.2 Theoretical single stiffness, $c'_{th}$

$$c'_{th} = \frac{1}{q'}$$  \hspace{1cm} (A.2)

where

$$q' = C_1 + C_2 \frac{z_{n1}}{z_{n2}} + C_3 \frac{z_{n1}}{z_{n2}} + (C_4 x_1) + (C_5 x_1)^2 + (C_6 x_2) + (C_7 x_2)^2 + (C_8 x_2^2) + (C_9 x_2^2)$$  \hspace{1cm} (A.3)

10) The tooth deflection can be determined approximately using $F_t (F_m F_{bt} ...)$ instead of $F_{bt}$. Conversion from $F_t$ to $F_{bt}$ (load tangent to the base cylinder) is covered by the relevant factors, or the modifications resulting from this conversion can be ignored when compared with other uncertainties (e.g. tolerances on the measured values).

11) $c'$ at the outer limit of single pair tooth contact, can be assumed to approximate the maximum value of single stiffness when $\varepsilon_0 > 1.2$. 

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A.2.3 Gear blank factor, \( C_R \)

\( C_R = 1 \) for gears made from solid disc blanks. For other gears:

\[
C_R = 1 + \frac{\ln (b_i/b)}{5e^{iR/(b_mn)}}
\]  
(A.4)

Boundary conditions:

- when \( b_i/b < 0.2 \), substitute \( b_i/b = 0.2 \);
- when \( b_i/b > 1.2 \), substitute \( b_i/b = 1.2 \).

See Figure A.1 for symbols.

A.2.4 Basic rack factor, \( C_B \)

\( C_B \) can be obtained from equation (A.5):

\[
C_B = \left[ 1 + 0.5 \left( 1.2 - \frac{h_{pm}}{m_n} \right) \right] \left[ 1 - 0.02 (20° - \alpha_{pn}) \right]
\]  
(A.5)

A.2.5 Additional information

a) Internal gears: approximate values of the theoretical single stiffness of internal gear teeth can be determined from equations (A.2), (A.3), by the substitution of infinity for \( z_n2 \).

b) Specific load \( (F_t K_A/b) < 100 \text{ N/mm} \)

\[
c' = 0.8 c_i C_R C_B \cos \beta \left( \frac{F_t K_A}{100 b} \right)^{0.25}
\]  
(A.6)

c) The above is based on steel gear pairs, for other materials and material combinations, refer to ISO 6336-1:1996, clause 9.

A.2.6 Mesh stiffness, \( c_\gamma \)

For spur gears with \( \epsilon_\alpha \geq 1.2 \) and helical gears with \( \beta \leq 30^\circ \), the mesh stiffness:

\[
c_\gamma = c' \left( 0.75 \epsilon_\alpha + 0.25 \right)
\]  
(A.7)

with \( c' \) according to equation (A.1).
Figure A.1 — Wheel blank factor, $C_R$; mean values for mating gears of similar or stiffer wheel blank design
Annex B
(normative)

Special features of less common gear designs

B.1 Dynamic factor, $K_v$, for planetary gears

B.1.1 General

In gear trains which include multiple mesh gears such as idler gears and in epicyclic gearing, planet and sun gears, there are several natural frequencies. These can be higher or lower than the natural frequency of a single gear pair which has only one mesh.

Although values of $K_v$ determined with the formulae in this International Standard shall be considered as unreliable, they can nevertheless be useful as preliminary assessments. It is recommended that if possible they be re-assessed using a more accurate procedure.

Method A should be preferred for the analysis of less common transmission designs. Refer to 6.1.1 of ISO 6336-1:1996 for further information.

B.1.2 Calculation of the relative mass of a gear with external teeth

Refer to 5.6.2.

B.1.3 Resonance speed determination for less common gear designs

B.1.3.1 General

The resonance speed determination for less common gear designs should be made using Method A. However, other methods may be used to approximate the effects. Some examples are the following:

a) pinion shaft with diameter at mid-tooth depth, $d_{m1}$, about equal to the shaft diameter;

b) two rigidly connected, coaxial gears;

c) one large wheel driven by two pinions;

d) planetary gears;

e) idler gears.

B.1.3.2 Pinion shaft diameter equal to diameter at mid-tooth depth, $d_{m1}$

The high torsional stiffness of the pinion shaft is to a great extent compensated by the shaft mass. Thus the resonance speed can be calculated in the normal way, using the mass of the pinion (toothed portion) and the normal mesh stiffness $C_T$.

B.1.3.3 Two, rigidly connected, coaxial gears

The mass of the larger of the connected gears shall be included.
B.1.3.4 One large wheel driven by two pinions

As the mass of the wheel is normally much greater than the masses of the pinions, each mesh can be considered separately, i.e:

— as a pair comprising the first pinion and the wheel, and
— as a pair comprising the second pinion and the wheel.

B.1.3.5 Planetary gears

Because of the many transmission paths that include varieties of stiffness other than mesh stiffness, the vibratory behaviour of planetary gears is very complex. The calculation of dynamic load factors using simple formulae, such as method B, is generally quite inaccurate. Nevertheless, method B, modified as follows, can be used for a first estimate of $K_V$. This estimate should, if possible, be verified by means of a subsequent detailed theoretical or experimental analysis, or on the basis of operating experience. See also the introductory comments to this annex.

a) Sun gear/planet gear

The reduced mass for the determination of the resonance speed, $n_{E1}$, of the sun gear is given by:

$$m_{red} = \frac{J_{pla}^* J_{sun}^*}{\left( \frac{p J_{pla}^* r_{sun}^2}{J_{sun}^* r_{pla}^2} \right)}$$  \hspace{1cm} (B.1)

where

- $J_{pla}^*$ and $J_{sun}^*$ are the moments of inertia per unit facewidth of one planet gear and the sun gear respectively in kilogram square millimetres per millimetre ($\text{kg} \cdot \text{mm}^2/\text{mm}$).

- $r_{sun} = 0.5 d_{sun}$

- $r_{pla} = 0.5 d_{pla}$

- $p$ is the number of planet gears in the gear stage under consideration.

The value, $m_{red}$, determined from equation (B.1), shall be used in the equation for calculating $N$ (see 5.6.2.2) where a mesh stiffness approximately equal to a single planetary gear shall be used for the mesh stiffness $c_y$ and the number of teeth on the sun gear shall be used for $z_1$.

Concerning planetary gears, it should be noted that $F_t$ in equations (12) to (14) for $B_p$, $B_h$, $B_k$ (see 5.6.2.3) is equal to the total tangential load applied to the sun gear, divided by the number of planet gears.

b) Planet gear/annulus gear rigidly connected to the gear case

In this case, the mass of the annulus gear can be assumed to be infinite. Thus, the relative mass becomes equal to the referred mass of the planet gear. This can be determined as follows:

$$m_{red} = \frac{J_{pla}^*}{r_{pla}^2}$$  \hspace{1cm} (B.2)

with the notation as above.
c) Planet gear/rotating annulus gear

In this case, the referred mass of the annulus gear can be determined as for an external wheel, and the planet gear relative mass calculated in accordance with equation (B.2). The procedure described in B.1.3.4 shall be used when the annulus gear meshes with several planet gears.

B.1.3.6 Idler gears

Approximate values can be obtained from the following when the driving and driven gears are roughly of the same size, the idler gear also about the same size or a little larger:

- reduced mass

\[ m_{\text{red}} = \frac{2}{\left( \frac{n_1^2 + 2n_2^2 + n_3^2}{J_1 + J_2 + J_3} \right)} \]  

\[ (B.3) \]

- mesh stiffness

\[ c_y = 0.5 (c_{1.2} + c_{1.3}) \]  

\[ (B.4) \]

where

\[ J_1, J_2, J_3 \] are the moments of inertia per unit facewidth of the pinion, the idler and the wheel respectively in kilogram square millimetres per millimetre (kg·mm\(^2\)/mm);

\[ c_{1.2} \] is the mesh stiffness of the driver and idler gear pair;

\[ c_{1.3} \] is the mesh stiffness of the driver and idler gear pair (see annex A for the determination of \( c_y \)).

More accurate analysis is recommended if the reference speed is in the range \( 0.6 < c < 1.5 \).

If the idler is substantially larger than the driving and driven gears or, if the driving gear or driven gear is substantially smaller than the two other gears, \( K_v \) can be calculated separately for each meshing pair, i.e.

- for the driver-idler gear combination, and

- for the idler-driven gear combination.

Values of \( m_{\text{red}} \) calculated in accordance with the above may be substituted in equation (7) of 5.6.2.2 to determine the resonance speed.

An accurate analysis is recommended for cases not mentioned here.

B.2 Face load factors, \( K_{\text{H}} \), \( K_{\text{F}} \), for simple planetary gears

The face load factor takes into account the effects of the non-uniform distribution of load over the gear facewidth on the surface stress (\( K_{\text{H}} \)) and tooth-root stress (\( K_{\text{F}} \)).
According to 7.2.3.1 a) and 7.6.1 of ISO 6336-1:1996, Method C1 is suitable for the gears of single planetary gearsets in which the following features are found.12)

Either the sun or the planet carrier and sometimes the annulus gear is free to float; otherwise a comparable division of load between the individual planet gears is achieved by greater accuracy of manufacture, flexibility or both. If necessary, refer to the above mentioned clauses for details.

Determine:
- mesh misalignment due to manufacturing deviation $f_{ma}$ in accordance with 5.7.2.2,
- running-in factor $x_{p}$ in accordance with 5.7.2.3,
- mesh stiffness in accordance with annex A.

Any unequal division of the total tangential load between the planet gears is covered by factor $K_{y}$ (see clause 5). Thus, for these gears, $F_{m} = (F_{t} K_{A} K_{y} K_{H})$, and with $F_{t}$ being the nominal tangential load transmitted per mesh, also the sum of the loads over both helices of double helical gears.

a) Spur and single helical gears (see footnote 5)

- Gear pair without helix modification, sun gear (Z)/planet (P), mounted on a fixed, rigid planet pin:

$$K_{H_{p}} = 1 + \frac{4000}{3\pi} K_{p} K_{y} \left( \frac{b}{d_{z}} \right) ^2 5.12 + \frac{K_{y} c_{y} f_{ma}}{2F_{m}/b}$$  \hspace{1cm} (B.5)

- For the same gear pair with helix modification (torsional deflection only compensated):

$$K_{H_{p}} \text{ in accordance with equation (36) and 5.7.2.4.2, and } K_{H_{p}} \geq 1.05.$$  \hspace{1cm} (B.6)

- Gear pair without helix modification, sun gear (Z)/planet (P) with journals, mounted with bearings in the planet carrier:

$$K_{H_{p}} = 1 + \frac{4000}{3\pi} K_{p} K_{y} \left[ 5.12 \frac{b}{d_{z}} \right] ^2 + \frac{2}{b} \frac{b}{d_{p}} \left( \frac{l_{p}}{b} - \frac{7}{12} \right) + \frac{K_{y} c_{y} f_{ma}}{2F_{m}/b} \hspace{1cm} (B.6)$$

- For the same gear pair with full helix modification (bending and torsional deflection fully compensated):

$$K_{H_{p}} \text{ in accordance with equation (36) of 5.7.2.4.2, and } K_{H_{p}} \geq 1.05.$$  \hspace{1cm} (B.7)

- Gear pair without helix modification, annulus gear (H)/planet (P) with journals, mounted with bearings in the planet carrier:

$$K_{H_{p}} = 1 + \frac{8000}{3\pi} K_{p} K_{y} \left( \frac{b}{d_{p}} \right) ^4 \left( \frac{l_{p}}{b} - \frac{7}{12} \right) + \frac{K_{y} c_{y} f_{ma}}{2F_{m}/b} \hspace{1cm} (B.7)$$

12) Restoring forces in toothed couplings are ignored. Restoring forces which lead to uneven distribution of load over the facewidth can occur when transmission elements are rigid and friction characteristics of flexible couplings are unsatisfactory.
For the same gear pair with helix modification (bending deflection only compensated):

\[ K_{\text{H}\beta} \text{ in accordance with equation (36) of 5.7.2.4.2, and } K_{\text{H}\beta} \geq 1.05. \]

Gear pair with or without helix modification, annulus gear (H)/planet (P) mounted on a fixed, rigid planet pin:

\[ K_{\text{H}\beta} \text{ in accordance with equation (36) of 5.7.2.4.2, and } K_{\text{H}\beta} \geq 1.05. \]

b) Double helical gears (see 5.7.2.4, with footnotes 4 and 5)

Gear pair without helix modification, sun gear (Z)/planet (P) mounted on a fixed, rigid planet pin:

\[ K_{\text{H}\beta} = 1 + \frac{4000}{3\pi} \beta \frac{c_7}{E} \left( \frac{2B_B}{d_z} \right)^2 \exp \left( -\frac{2\beta c_7 f_{\text{ma}}}{F_m/b_B} \right) \tag{B.8} \]

For the same gear pair with helix modification (torsional deflection only compensated, see 5.7.2.4, footnote 4):

\[ K_{\text{H}\beta} \text{ in accordance with equation (37) of 5.7.2.4.2, and } K_{\text{H}\beta} \geq 1.05. \]

Gear pair without helix modification, sun gear (Z)/planet (P) with journals, mounted with bearings in a planet carrier:

\[ K_{\text{H}\beta} = 1 + \frac{4000}{3\pi} \beta \frac{c_7}{E} \left[ 3,2 p \left( \frac{2B_B}{d_z} \right)^2 + 2 \left( \frac{B}{d_p} \right)^4 \left( \frac{I_p}{B} - \frac{7}{12} \right) \right] + \frac{\kappa_B c_7 f_{\text{ma}}}{F_m/b_B} \tag{B.9} \]

For the same gear pair with full helix modification (bending and torsional deflection fully compensated, see footnote 7):

\[ K_{\text{H}\beta} \text{ in accordance with equation (37) of 5.7.2.4.2, and } K_{\text{H}\beta} \geq 1.05. \]

Gear pair without helix modification, annulus gear (H)/planet (P) with journals, mounted with bearings in a planet carrier:

\[ K_{\text{H}\beta} = 1 + \frac{8000}{3\pi} \beta \frac{c_7}{E} \left( \frac{B}{d_p} \right)^4 \left( \frac{I_p}{B} - \frac{7}{12} \right) + \frac{\kappa_B c_7 f_{\text{ma}}}{F_m/b_B} \tag{B.10} \]

For the same gear pair with helix modification (bending deflection only compensated):

\[ K_{\text{H}\beta} \text{ in accordance with equation (37) of 5.7.2.4.2, and } K_{\text{H}\beta} \geq 1.05. \]

Gear pair with or without helix modification, annulus gear (H)/planet (P) mounted on a fixed, rigid planet pin:

\[ K_{\text{H}\beta} \text{ in accordance with equation (37) of 5.7.2.4.2, and } K_{\text{H}\beta} \geq 1.05. \]
Annex C
(informative)

Guide values for application factor, $K_A$

C.1 Establishment of application factors

Application factors can best be established from a thorough analysis of service experience with a particular application (see ISO/TR 10495). For marine gears, the rules of the classification authorities shall be observed, as these are founded on extensive service experience. For the main propulsion gears of sea-going ships, a thorough analytical investigation should be made.

The factor $K_A$ is used to modify the value $F_t$, to take into account loads additional to nominal loads imposed on the gears from external sources. If it is not possible to determine the equivalent tangential load (see clause 5.3) by comprehensive system analysis or from measured values using a suitable cumulative damage criterion; the empirical guidance values in C.2 can be used.

For marine gears, which are subjected to cyclic peak torques (torsional vibrations) and designed for infinite life, the application factor can be defined as the ratio between the peak cyclic torque and the nominal rated torque. The nominal rated torque is defined by the rated power and speed; it is the torque used in the load capacity calculations.

If the gear is subjected to a limited number of known loads in excess of the amount of the peak cyclic torques, this influence may be covered directly by means of a cumulative fatigue criterion, as mentioned above, or by means of an increased application factor representing the influence of the load spectrum.

It is recommended that the purchaser and manufacturer or designer agree on the value of the application factor in agreement with the applicable classification authority.

C.2 Approximate values for the application factors

$K_A$ used in the preparation of preliminary designs can be chosen from the following values:

--- for diesel driven main propulsion gears, $K_A = 1.35$;
--- for turbine driven main propulsion gears, $K_A = 1.1$.

For geared transmissions of auxiliary machinery such as those listed in table C.1, the following values may be used:

--- for diesel driven auxiliaries, $K_A = 1.5$;
--- for turbine and electric motor driven auxiliaries, $K_A = 1.25$;
--- for turbine driven electricity generators, $K_A = 1.1$.

<table>
<thead>
<tr>
<th>Table C.1 — Auxiliary machinery</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electricity generators</td>
</tr>
<tr>
<td>Side thrusters</td>
</tr>
<tr>
<td>Azimuth thrusters</td>
</tr>
<tr>
<td>Any other equipment which is essential to the safety of a ship or other similar marine unit</td>
</tr>
</tbody>
</table>
Annex D
(informative)

Guide values for crowning and end relief of teeth of cylindrical gears

D.1 General

Well designed crowning and end relief have a beneficial influence on the distribution of load over the facewidth of a gear (see 5.7). Design details should be based on a careful estimate of the deformations and manufacturing deviations of the gearing of interest. If deformations are considerable, helix angle modification might be superposed over crowning or end relief, but well designed helix modification is preferable.

D.2 Amount of crowning, \( C_{\beta} \)

The following non-mandatory rule is drawn from experience; the amount of crowning (see Figure D.1) necessary to obtain acceptable distribution of load can be determined as follows.

Subject to the limitations \( 10 \, \mu m \leq C_{\beta} \leq 40 \, \mu m \) plus a manufacturing tolerance of 5 \( \mu m \) to 10 \( \mu m \), and that the value \( b_{ca} / b \) would have been greater than 1 had the gears not been crowned: \( C_{\beta} = 0.5 \, f_{px \, cv} \).

![Figure D.1 — Amount of crowning, \( C_{\beta} \), and width, \( b \)](image)

In order to avoid excessive loading of tooth ends, the crowning amount shall be calculated as:

\[
C_{\beta} = 0.5( f_{sh} + f_{H\beta} ) \tag{D.1}
\]

When the gears are of such stiff construction that \( f_{sh} \) can for all practical purposes be neglected, or when the helices have been modified to compensate for deformation at mid-facewidth, the following value can be substituted:

\[
C_{\beta} = 0.5 f_{H\beta} \tag{D.2}
\]

Subject to the restriction \( 10 \, \mu m \leq C_{\beta} \leq 25 \, \mu m \) plus a manufacturing tolerance of about 5 \( \mu m \), 60 % to 70 % of the above values are adequate for extremely accurate and reliable high speed gears.
D.3 Amount $C_{(ll)}$ and width $b_{(ll)}$ of end relief

D.3.1 Method C1

This method is based on an assumed value for the equivalent misalignment of the gear pair, without end relief, and on the recommendations for the amount of gear crowning.

a) Amount of end relief (see Figure D.2)

For through hardened gears: $C_{(ll)} = F_{p x c v}$ plus a manufacturing tolerance of 5 μm to 10 μm.

Thus, by analogy with $F_{p x c v}$ in clause D.1, $C_{(ll)}$ should be approximately:

$$C_{(ll)} = f_{sh} + 1.5 f_{H b} \quad (D.3)$$

For surface hardened and nitrided gears: $C_{(ll)} = 0.5 F_{p x c v}$ plus a manufacturing tolerance of 5 μm to 10 μm.

Thus, by analogy with $F_{p x c v}$ in clause D.1, $C_{(ll)}$ should be approximately:

$$C_{(ll)} = 0.5 \left( f_{sh} + 1.5 f_{H b} \right) \quad (D.4)$$

![Figure D.2 — Amount $C_{(ll)}$ and width $b_{(ll)}$ of end relief](image)

When the gears are of such stiff construction that $f_{sh}$ can for all practical purposes be neglected, or when the helices have been modified to compensate deformation, proceed in accordance with equation (D.2).

For very accurate and reliable gears with high tangential velocities, 60 % to 70 % of the above values is appropriate.

b) Width of end relief

For approximately constant load and higher tangential velocities: $b_{(ll)}$ is the smaller of the values $(0.1 b)$ or $(1.0 m)$

The following is appropriate for variable loading, low and average speeds:

$$b_{red} = (0.5 \text{ to } 0.7) b \quad (D.5)$$

D.3.2 Method C2

This method is based on the deflection of gear pairs, assuming uniform distribution of load over the facewidth:

$$\delta_{sh} = F_m/(bc_v), \text{ or } F_m = F_1 K_A K_v \quad (D.6)$$
For highly accurate and reliable gears with high tangential velocities, the following are appropriate:

\[ C_{(II)} = (2 \text{ to } 3) \delta_{bh} \]  
\[ b_{red} = (0.8 \text{ to } 0.9)b \]  

For similar gears of lesser accuracy:

\[ C_{(III)} = (3 \text{ to } 4) \delta_{bh} \]  
\[ b_{red} = (0.7 \text{ to } 0.8)b \]
Check and interpretation of tooth contact pattern

E.1 Scope and field of application

This annex describes a procedure for checking the tooth contact of marine gear units (accuracy grade 6 or better) without load or under partial load condition.

E.2 Test methods

E.2.1 General

There are two methods described for determining the tooth contact pattern:

— contact test (check of mesh without load);
— load test (contact pattern at defined load level).

E.2.2 Contact test

The contact test is an economical method for determining the sum of all manufacturing deviations. The contact test is usually performed in the completely mounted condition. If no housing is available, especially in the case of large gears, a test rig may be used. Typical applications are

— large gears for marine transmissions, and
— gears mounted on board.

The main influence factors on the tooth contact without load are given in Table E.1.

<table>
<thead>
<tr>
<th>Tooth deviations</th>
<th>Housing deviations</th>
<th>Shaft deviations</th>
<th>Bearing tolerances</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pitch deviation</td>
<td>Shaft angle deviation</td>
<td>Axial runout</td>
<td>Bearing clearance</td>
</tr>
<tr>
<td>Profile deviation</td>
<td>Shaft slope deviation</td>
<td></td>
<td>Concentricity</td>
</tr>
<tr>
<td>Lead deviation</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

E.2.3 Load test

The load test is applied for highly loaded gears with profile or lead modifications, or both, for comparison of the actual contact pattern with the data obtained by calculation. For the test, the load is increased in reasonable steps in order to be able to predict the load distribution at full load. At the lowest load stage the gears shall already have reached their final position. A typical sequence of load stages is: 5 %, 25 %, 50 %, 75 %, 100 % (maximum value as high as possible).

The load-dependent influence factors with influence on the tooth contact are given in Table E.2.
Table E.2 — Load-dependent influence factors influencing the tooth contact

<table>
<thead>
<tr>
<th>Tooth deviations</th>
<th>Housing deviations</th>
<th>Shaft deviations</th>
<th>Bearing tolerances</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tooth deformation</td>
<td>Housing stiffness</td>
<td>Shaft deflection</td>
<td>Bearing stiffness</td>
</tr>
<tr>
<td>Hertzian deformation</td>
<td>Housing temperature</td>
<td>Shaft distortion</td>
<td></td>
</tr>
<tr>
<td>Gear blank deformation</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tooth wear</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

E.2.4 Procedure

It is usual for both contact and load tests that at least three sets of teeth (for the whole plane of contact) are considered. Either the wheel or the pinion is painted with a suitable contrast colour. After several revolutions without load or at the actual load stage the transmission of colour to the mating gear (without or with moderate load), or the abrasion of the colour (high load), is used to evaluate the contact pattern.

E.2.5 Paints

E.2.5.1 Contact test

See Table E.3.

Table E.3 — Suitable paints (contact test)

<table>
<thead>
<tr>
<th>Suitable paints</th>
<th>Manufacturer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lukas Tuschierfarbe</td>
<td>Dr. Schönfeld &amp; Co.</td>
</tr>
<tr>
<td>Diamant Tuschierfarbe</td>
<td>Schleifmittelwerk Kahl</td>
</tr>
<tr>
<td>Eosol Tuschierpaste</td>
<td>Emil Otto/Fabrik</td>
</tr>
<tr>
<td>Kruel Tuschierfarbe</td>
<td>Fa. C. Kreul</td>
</tr>
<tr>
<td>Norma Olfarbe</td>
<td>H. Schminke &amp; Co.</td>
</tr>
<tr>
<td>Yellow Gear Marking</td>
<td>Prescott &amp; Comp. Ltd.</td>
</tr>
</tbody>
</table>

NOTE The above are examples of products available commercially. This information is given for the convenience of users of this International Standard and does not constitute an endorsement of ISO of these products.

E.2.5.2 Load test

The contrast colour for load tests shall meet the following requirements:

- good contrast on metal surface;
- high temperature resistance;
- oil resistant;
- high tensile strength;
- high adhesion.

See Table E.4.
Table E.4 — Suitable paints (load test)

<table>
<thead>
<tr>
<th>Suitable paints</th>
<th>Manufacturer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dykem Red Layout DX-296</td>
<td>The Dykem Company</td>
</tr>
<tr>
<td>Eosol Anreißfarbe</td>
<td>Emil Otto/Fabrik</td>
</tr>
<tr>
<td>Pelikan Anreißfarbe</td>
<td>Pelikanwerke</td>
</tr>
<tr>
<td>Regensburger Getriebeprüflack</td>
<td>Regensburger Lackfabrik</td>
</tr>
<tr>
<td>Copper sulphate</td>
<td></td>
</tr>
</tbody>
</table>

NOTE The above are examples of products available commercially. This information is given for the convenience of users of this International Standard and does not constitute an endorsement of ISO of these products.

### E.3 Evaluation of expected values for contact test and load test

#### E.3.1 Face contact width

The optimum contact pattern is determined on the basis of the lead modification gained by Method A, B or C of ISO 6336-1:1996. If a linear helix modification is applied the face contact width is calculated as

\[
h_P = \frac{s_c}{f_{korr}} \times 100 \tag{E.1}
\]

where

- \( h_P \) is the face contact width in percent;
- \( s_c \) is the film thickness of the contrast colour, in micrometres (\( \mu m \));
- \( f_{korr} \) is the lead correction value, in micrometres (\( \mu m \)).

#### E.3.2 Profile contact width

The optimum contact width over the profile is determined corresponding to the actual profile and lead corrections, and the referring tolerances. Reasonably, the load value should be advised where full profile contact is expected.

### E.4 Examination of contact pattern

Examination of the contact pattern is subjective and should therefore always be carried out together with the use of complete investigation records. For both the contact test and load test, the optimum contact pattern can be adjusted during the test by means of eccentric bearing rings or by adding shims at the supports to distort the housing.
Bibliography


13) To be published.
The technical committee responsible for the preparation of this standard has reviewed the provisions of following International Standards referred in this adopted standard and has decided that they are acceptable for use in conjunction with this standard:

<table>
<thead>
<tr>
<th>International Standard</th>
<th>Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>ISO 1328-1:1995</td>
<td>Cylindrical gears — ISO system of accuracy — Part 1: Definitions and allowable values of deviations relevant to corresponding flanks of gear teeth</td>
</tr>
<tr>
<td>ISO 6336-5:1996</td>
<td>Calculation of load capacity of spur and helical gears — Part 5: Strength and quality of material</td>
</tr>
</tbody>
</table>

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the result of a test or analysis, shall be rounded off in accordance with IS 2:1960 ‘Rules for rounding off numerical values (revised)’. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.
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Amendments Issued Since Publication

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