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Cylindrical gears — Calculation of service life under variable load — Conditions for cylindrical gears in accordance with ISO 6336

Engrenages cylindriques — Calcul de la durée de vie en service sous charge variable — Conditions pour les engrenages cylindriques conformément à l'ISO 6336

ISO

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Contents

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The main task of technical committees is to prepare International Standards, but in exceptional circumstances a technical committee may propose the publication of a Technical Report of one of the following types:

- type 1, when the required support cannot be obtained for the publication of an International Standard, despite repeated efforts;
- type 2, when the subject is still under technical development or where for any other reason there is the future but not immediate possibility of an agreement on an International Standard;
- type 3, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example).

Technical Reports of types 1 and 2 are subject to review within three years of publication, to decide whether they can be transformed into International Standards. Technical Reports of type 3 do not necessarily have to be reviewed until data they provide are considered to be no longer valid or useful.

ISO/TR 10495, which is a Technical Report of type 2, was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

Annexes A and B of this Technical Report are for information only.

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Cylindrical gears — Calculation of service life under variable load — Conditions for cylindrical gears in accordance with ISO 6336

1 Scope

This Technical Report is concerned with the calculation of service life (or safety factors for a required life) of gears subject to variable loading. Clauses 4 and 5 give a general discussion of the subject; clauses 6 to 8 present a method which may be conveniently applied at the design stage. Whilst the method is presented in terms of ISO 6336, it is equally applicable to other gear stress calculations (e.g BS 436, DIN 3990, NF E23-015).

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this Technical Report. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this Technical Report are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 701:1976, *International gear rotation - Symbols for geometrical data.*

ISO 1122-1: 1983, *Glossary of gear terms - Part 1: Geometrical definitions.*

ISO 6336-1: 1996, *Calculation of load capacity of spur and helical gears - Part 1: Basic principles, introduction and general influence factors.*

ISO 6336-2: 1996, *Calculation of load capacity of spur and helical gears - Part 2: Calculation of surface durability (pitting).*

ISO 6336-3: 1996, *Calculation of load capacity of spur and helical gears - Part 3: Calculation of tooth bending strength.*

ISO 6336-5: 1996, *Calculation of load capacity of spur and helical gears - Part 5: Strength and quality of material.*

3 Definitions, symbols, quantities and units

For the purposes of ISO TR 10495, the definitions given in ISO 1122-1 apply. Symbols are based on those given in ISO 701. Only symbols for quantities used in ISO TR 10495 are given in table 1.

Table 1 - Symbols used within ISO TR 10495

Symbol	Quantity	Unit
b	Facewidth	mm
d_1	Reference diameter of pinion	mm
e	Inclination of S-N curve	- -
i	Class	- -
I	Class interval	- -
K_A	Application factor	- -
$K_{F\alpha}$	Transverse load distribution factor (bending stress)	- -
$K_{F\beta}$	Face load distribution factor (bending stress)	- -
$K_{H\alpha}$	Transverse load distribution factor (contact stress)	- -
$K_{H\beta}$	Face load distribution factor (contact stress)	- -
K_v	Dynamic factor	- -
m_n	Normal module	mm
n_i	Number of cycles at *i*th stress level (number of counts in class *i*)	- -
n_I	Number of cycles at class interval level *I*	- -
N_I	Number of cycles to failure at class interval level *I*	- -
N_L	Number of cycles to failure	- -
S	Safety factor for stress	- -
$S_{F\ lim}$	Safety factor for bending stress (min.)	- -
$S_{H\ lim}$	Safety factor for contact stress (min.)	- -
T_i	Torque class	Nm
T_I	Pinion torque at top of class interval	Nm
u	Gear ratio	- -
U	Miner sum	- -
U_I	Individual damage part of class interval	- -
Y_F	Tooth form factor	- -
Y_{NT}	Tooth root stress life factor for standard test conditions	- -
$Y_{R\ rel\ T}$	Relative surface condition factor (root)	- -
Y_S	Stress correction factor	- -
Y_{ST}	Stress correction factor for the dimension of the standard test gears	- -
Y_X	Size factor (bending stress)	- -
Y_β	Helix angle factor (bending stress)	- -
$Y_{\delta\ rel\ T}$	Relative notch sensitivity factor	- -
$Z_{B,D}$	Single pair tooth contact factor for pinion or gear	- -
Z_E	Elasticity factor	$(N/mm^2)^{1/2}$
Z_H	Zone factor	- -
Z_L	Lubricant influence factor	- -
Z_{NT}	Contact stress life factor for standard test conditions	- -
Z_R	Roughness factor	- -
Z_v	Speed factor	- -
Z_W	Hardness ratio factor	- -
Z_β	Helix angle factor (contact stress)	- -
Z_ε	Contact ratio factor (contact stress)	- -
$\sigma_{F\ lim}$	Nominal stress number (bending)	N/mm^2
σ_{FI}	Tooth root stress at class interval *I*	N/mm^2
σ_{FP}	Permissible tooth root stress	N/mm^2
$\sigma_{H\ lim}$	Allowable stress number (contact)	N/mm^2
σ_{HI}	Contact stress at class interval *I*	N/mm^2
σ_{HO}	Nominal contact stress	N/mm^2
σ_{HP}	Permissible contact stress	N/mm^2
σ_I	Stress at class interval *I*	N/mm^2
σ_{lim}	Allowable stress	N/mm^2
σ_P	Permissible stress	N/mm^2

4 Introduction

4.1 Determination of load and stress spectra

Variable loads resulting from a working process, starting process or from operation at or near a critical speed will cause varying stresses at the gear teeth of a drive system. The magnitude and frequency of these loads depend upon the driven machine(s), the driver(s) or motor(s) and the mass elastic properties of the system.

These variable loads (stresses) may be determined by one or more of the procedures listed below:

a) Experimental measurement of the operating loads at the machine in question;
b) Estimation of the spectrum, if this is known, for a similar machine with similar operating mode;
c) Calculation, using known external excitation and a mass elastic simulation of the drive system.

NOTE – Specific data, relevant for the method by which the load or torque measurements are performed, should be marked at the registered results.

To obtain the spectra, the range of the measured (evaluated) loads is divided into classes. A widely used number of classes is 64.

The cycle count for the load class corresponding to the load value for the highest loaded tooth is incremented at every load repetition. Table 2 shows as an example how to apply the torque classes defined in figure 1 to specific torque levels and correlated numbers of cycles.

Table 2 – Example (see figure 1): Classes 111 & 112

Torque class, T_i, Nm	Number of cycles, n_i
$440 \leq T_{111} < 444$	$n_{111} = 2338$
$444 \leq T_{112} < 448$	$n_{112} = 4318$

The torques used to evaluate tooth loading should include the dynamic effects at different rotational speeds.

This spectrum is only valid for the measured or evaluated time period. If the spectrum is extrapolated to represent the required life time, the possibility that there might be torque peaks not frequent enough to be evaluated in that measured spectrum must be considered. These peaks may have an effect on the gear life.

Stress spectra concerning bending and pitting can be obtained from the load (torque) spectrum by using Method II.

Scuffing resistance must be calculated from the worst combination of speed and load.

Wear is a continuous deterioration of the tooth flank and must be considered separately.

Tooth root stress can also be measured by means of strain gages in the fillet. In this case, the derating factors should be taken into account using the results of the measurements. The relevant contact stress can be calculated from the measurements.

4.2 General calculation of service life

The calculated service life is based on the theory that every load cycle (every revolution) is damaging to the gear. The amount of damage depends on the stress level and can be considered as zero for lower stress levels.

The calculated bending or pitting strength fatigue life of a gear is a measure of its ability to accumulate discrete damage until failure occurs.

The fatigue life calculation needs:

a) The stress spectrum;
b) Material fatigue properties;
c) A damage accumulation method.

T_l in Nm	[+0...+4[	[+4...+8[	[+8...+12[	[+12...+16[	[+16...+20[	[+20...+24[
0,00	0	0	0	0	0	0
24,00	0	0	0	0	0	0
48,00	0	0	0	0	0	0
72,00	0	0	0	0	0	0
96,00	0	0	0	0	0	0
120,00	0	706	3469	3081	5109	32
144,00	1	2	438	381	756	903
168,00	2	0	0	0	0	1
192,00	45	350	212	616	16	0
216,00	0	0	0	0	0	0
240,00	0	0	0	0	0	0
264,00	0	0	0	0	19	2108
288,00	2072	3933	4257	6	2	3
312,00	0	0	0	0	0	0
336,00	0	0	0	0	0	0
360,00	0	0	0	0	0	0
384,00	0	0	0	0	0	0
408,00	0	0	0	0	0	0
432,00	26	72	2338*	4318*	3665	1824
456,00	239	477	2553	3216	5576	2109
480,00	932	90	420	1913	2877	2891
504,00	1255	449	67	791	745	2166
528,00	651	518	23	1	0	0
552,00	0	0	8	24	127	520
576,00	751	713	295	42	0	0
600,00	0	0	0	0	0	0
624,00	0	0	0	0	0	3
648,00	218	187	329	469	34	0
672,00	0	0	0	0	0	0
696,00	0	0	0	0	0	0
720,00	0	0	0	0	0	0
744,00	0	0	0	0	0	0
768,00	0	0	0	0	0	0
792,00	0	0	0	0	0	0
816,00	0	0	0	0	0	0
840,00	0	0	0	0	0	0
864,00	0	0	0	0	0	0
888,00	0	0	0	0	0	0
912,00	0	0	0	0	0	0
936,00	0	0	0	0	0	0
960,00	0	0	0	0	0	0
984,00	0	0	0	0	0	0
1008,00	0	0	0	0	0	0

* example presented in table 2

Figure 1 – Torque spectrum (class number = 258)

The stress spectrum is discussed in clause 6.1.

Strength values based on material fatigue properties are chosen from applicable S-N curves. Many specimens must be tested by stressing them repeatedly at one stress level until failure occurs. This gives, after a statistical interpretation for a specific probability, a failure cycle number characteristic of this stress level. Repeating the procedure at different stress levels leads to an S-N curve.

An example of a cumulative stress spectrum is given in figure 2. Figure 3 shows measured cumulative stress spectra for tooth root stress. Figure 4 shows a cumulative contact stress spectrum with an S-N curve for specific material fatigue properties.

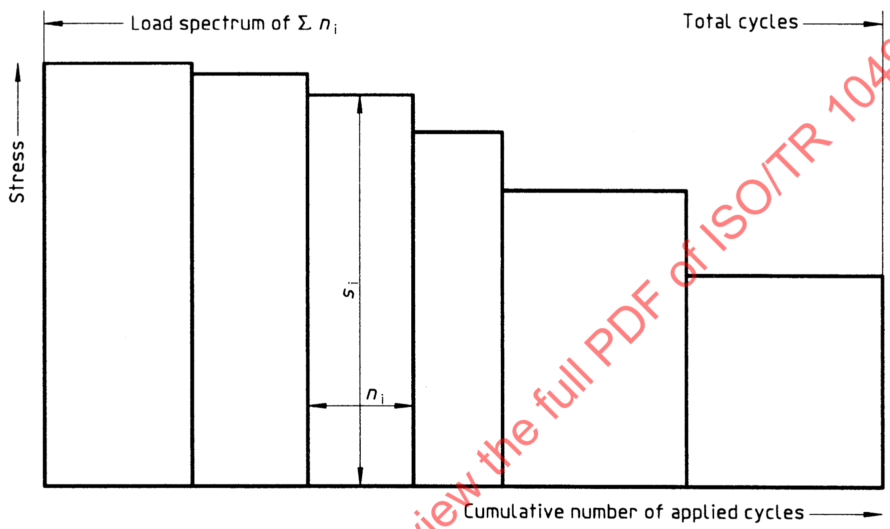


Figure 2 – Example for a cumulative stress spectrum

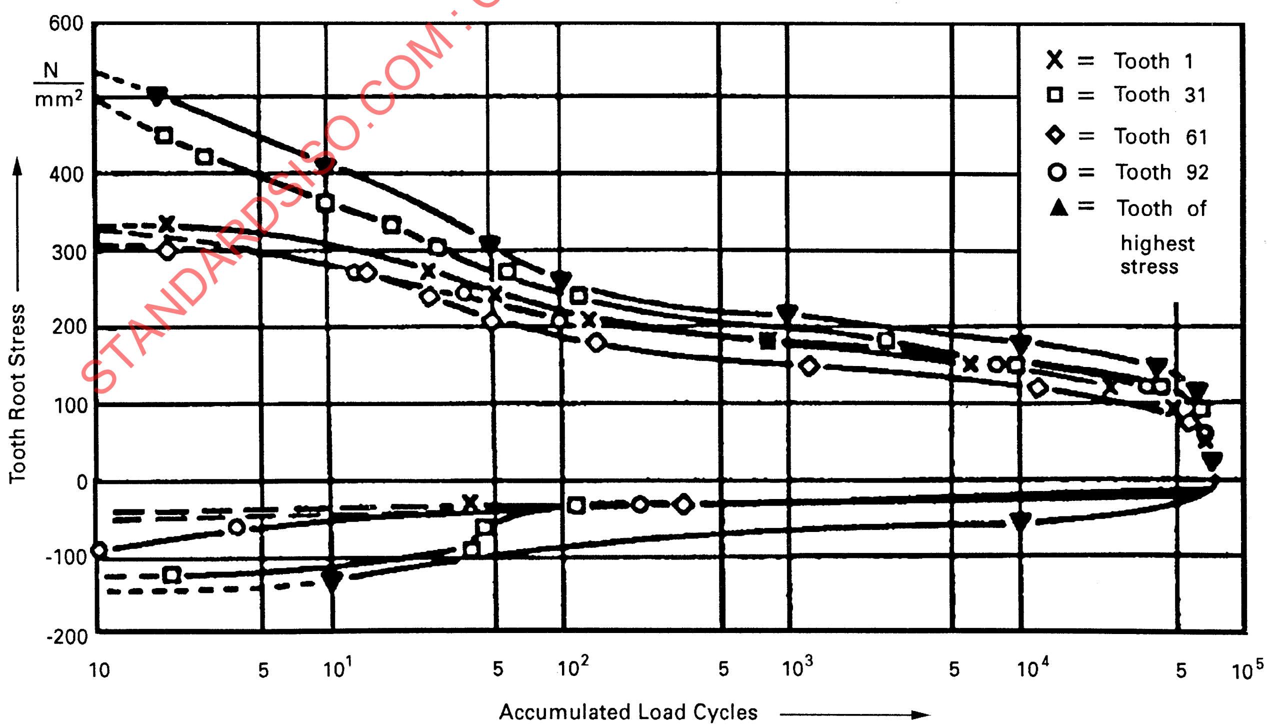
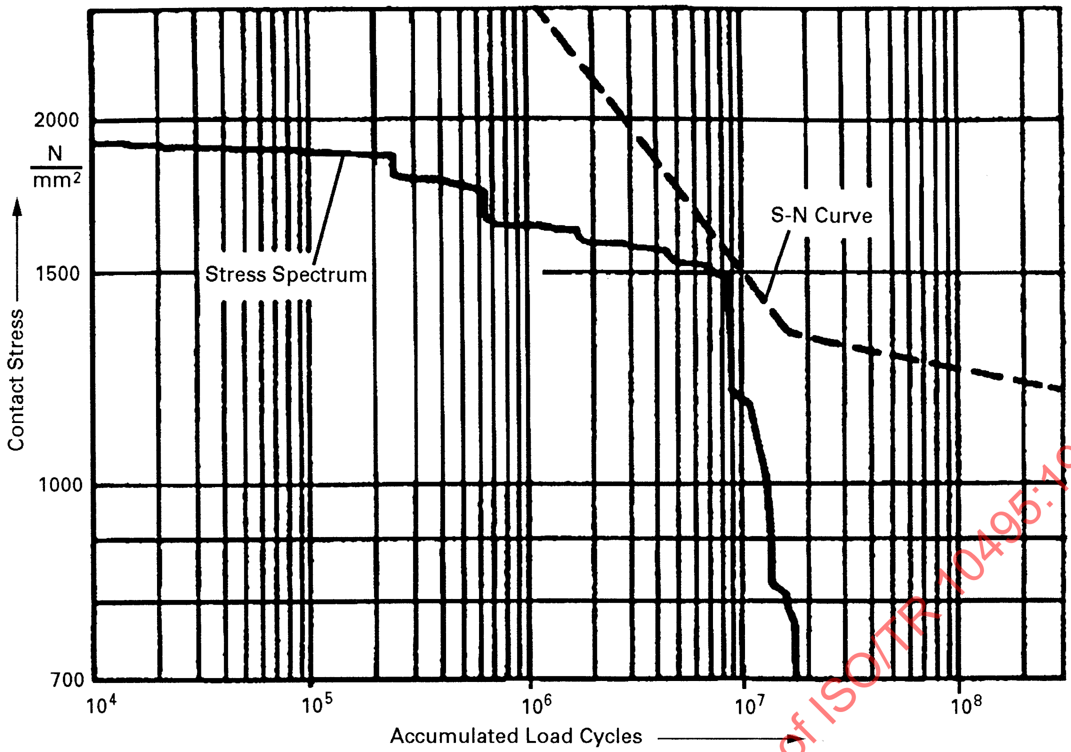


Figure 3 - Measured cumulative tooth root stress spectra for different teeth of one wheel



NOTE – The representation of the cumulative stress spectrum entirely below the S-N curve does not imply that the part will survive the total accumulative number of stress cycles. This information can be gained from a presentation as shown in figure 6.

Figure 4 – Cumulative contact stress spectrum with S-N curve

Linear, non-linear and relative methods are used.

The literature presented in annex B gives a general account of the present state and application of damage accumulation.

4.3 Palmgren-Miner rule

The Palmgren-Miner rule – besides other rules or modifications – is a widely used linear damage accumulation method. It is assumed that the damaging effect of each stress repetition at a given stress level is equal, which means the first stress cycle at a given stress level is as damaging as the last.

The Palmgren-Miner rule operates on the hypothesis that the portion of useful fatigue life used by a number of repeated stress cycles at a particular stress is equal to the ratio of the total number of cycles during the fatigue life at a particular stress level according to the S-N curve established for the material. For example, if a part is stressed for 3000 cycles at a stress level which would cause failure in 100 000 cycles, 3% of the fatigue life would be expended. Repeated stress at another stress level would consume another similarly calculated portion of the total fatigue life.

NOTE – The used material fatigue characteristics and endurance data should be related to a specific and required failure probability, e.g. 1%, 5% or 10%.

When 100% of the fatigue life is expended in this manner, the part could be expected to fail. The order in which each of these individual stress cycles is applied is not considered significant in Palmgren-Miner analysis.

Failure could be expected when:

$$\sum_{I} \frac{n_I}{N_I} = 1{,}0 \qquad \text{...(1)}$$

n_I Number of cycles at class interval level I

N_I Number of cycles to failure at interval level of class I (taken from the appropriate S-N curve)

If there is an endurance limit (upper, horizontal line beyond the knee in figure 5), the calculation is only done for stresses above this endurance limit.

If the appropriate S-N curve shows no endurance limit (lower line beyond the knee in figure 5), the calculation must be done for all stress levels. For each stress level, *l*, the number of cycles to failure, N_L, has to be taken from the corresponding part of the S-N-curve.

5 General calculation of service life, Method I

This method serves only for recalculation.

The load or stress spectrum of the gearing shall be determined by measurement, system analysis or experience.

For the calculation of cumulative damage, linear, non-linear, and relative methods may be used, provided that accuracy and reliability is appropriate to the application.

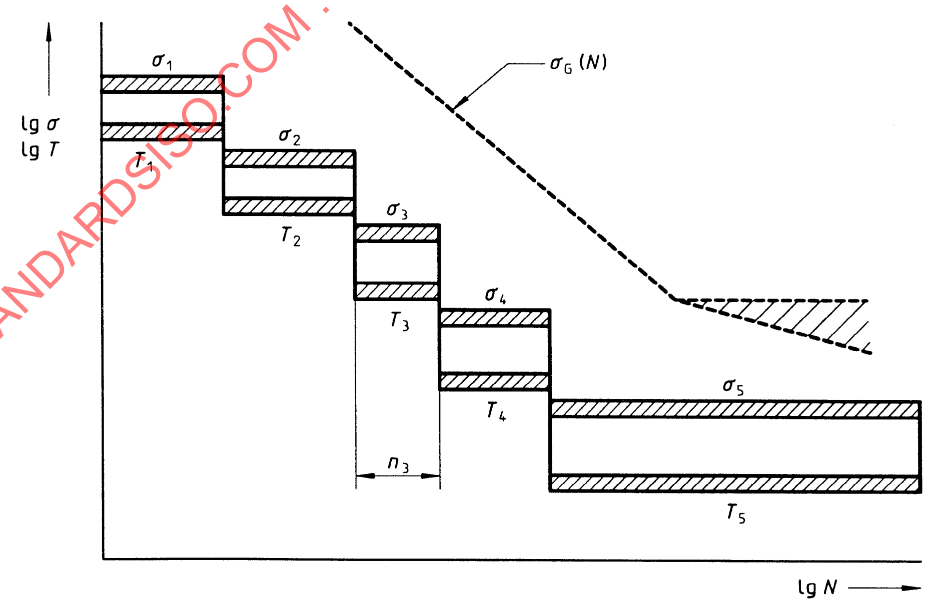
Annex B lists literature which gives data on the present state and application of calculation for service life.

6 Calculation of service strength on the basis of single-stage strength; calculation according to ISO 6336, Method II

6.1 Basic principles

This method is only valid for recalculation. It describes the application of linear cumulative damage calculations according to the Palmgren-Miner rule (see 4.3).

This method has been chosen because it is widely known and easy to apply; the choice does not imply that the method is superior to others described in the literature.



NOTE – The representation of the cumulative stress spectrum entirely below the S-N curve does not imply that the part will survive the total accumulative number of stress cycles. This information can be gained from a presentation as shown in figure 6.

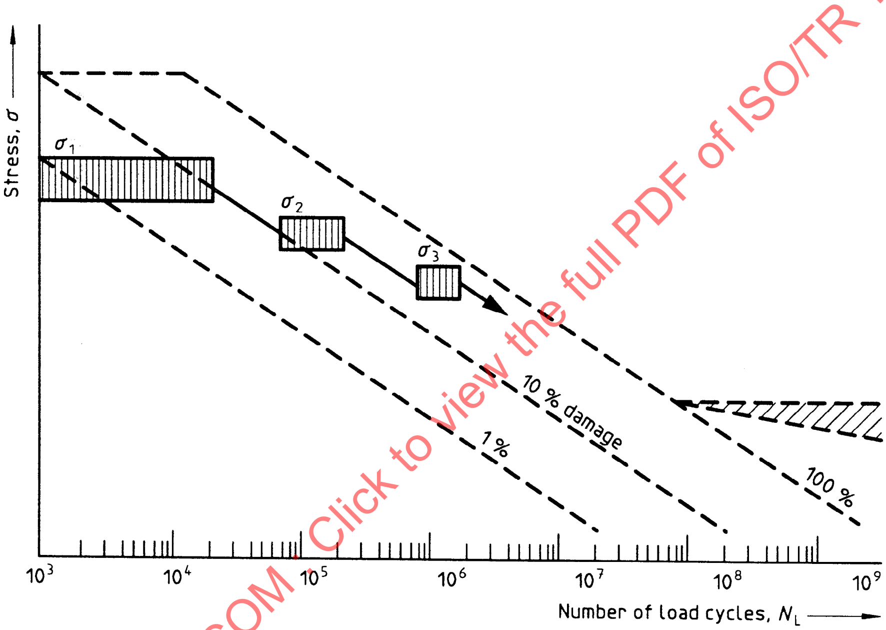
Figure 5 – Torque spectrum and associated stress spectrum with S-N curve

From the individual torque classes, the torques at the upper limit of each torque class and the associated numbers of cycles shall be listed (see table 3 for example).

Table 3 – Example

Upper limit of torque class*, T_i, Nm	Number of cycles, n_i
$T_{111} < 444$ $T_{112} < 448$	$n_{111} = 2338$ $n_{112} = 4318$
* For conservative calculation, sufficiently accurate for a high number of torque classes.	

The stress spectra for tooth root and tooth flank (σ_{Fl}, σ_{Hl}) with all relative factors are formed on the basis of this torque spectrum. The load-dependent K-factors are calculated for each new torque class (for procedure, see 6.2).



NOTE – From this presentation it can be concluded whether the part will survive the total number of stress cycles.

Figure 6 – Accumulation of damage

With stress spectra obtained in this way, the calculated values are compared with the strength values (S-N curves, damage line) determined as described in clause 6.3 using the Palmgren-Miner rule, see 4.3. For a graphical representation, see figure 5.

For all values of σ_l, individual damage parts are defined as follows:

$$U_l = \frac{n_l}{N_l} \qquad ...(2)$$

The sum of the individual damage parts, U_l, results in the damage condition U. U must be less than or equal to unity.

$$U = \sum_l U_l = \sum_l \frac{n_l}{N_l} \leq 1,0 \qquad ...(3)$$

(If the stress spectra includes a safety factor – see 6.4 – ΣU_l should lie between 0,95 and 1,05)

NOTE – The calculation of speed-dependent parameters is based, for each load level, on a mean rotational speed. This also refers to the determination of the S-N curve.

This calculation process shall be applied to each pinion and wheel for both bending and contact stress. In addition, safety factors applied to static load strength should be calculated for the highest stress of the spectrum. At stresses higher than those permitted at the upper limit of Z_{NT}, Y_{NT} curves, short cycle fatigue may prevail, yield may occur, and the cumulative fatigue methods do not apply.

6.2 Calculation of the stress spectra

For each level l of the torque spectrum, the actual stress, σ_l, is to be determined separately for bending and contact stress in accordance with the following equations:

- for bending stress (Method B of ISO 6336-3):

$$\sigma_{Fl} = \frac{2\,000\, T_l}{d_1\, b\, m_n} Y_F\, Y_S\, Y_\beta\, K_{vl}\, K_{F\beta l}\, K_{F\alpha l} \quad \text{...(4)}$$

- for contact stress (Method B of ISO 6336-2):

$$\sigma_{Hl} = Z_H\, Z_E\, Z_\varepsilon\, Z_\beta\, Z_{BD} \sqrt{\frac{2\,000\, T_l}{d_1^2\, b} \frac{u+1}{u} K_{vl}\, K_{H\beta l}\, K_{H\alpha l}} \quad \text{...(5)}$$

The value K_A, defined as application factor, is set equal to unity (1,0) for this calculation, as all the application load influences should be taken into account by stress levels included in the calculation method.

6.3 Determination of bending and pitting strength values

S-N curves for bending and pitting strength can be determined by experiment or by the rules of ISO 6336-2 and ISO 6336-3.

The probability of damage is to be determined by the appropriate application standard.

Where teeth are loaded in both directions (e.g. idler gear), the values determined for tooth root strength must be reduced according to ISO 6336-3. For random reverse torques, see clause 7.

6.4 Determination of safety factors

In the general case, safety factors cannot directly be deduced from the Miner sum, U. They are to be determined by way of iteration. The procedure is shown in figure 7.

If the relevant portion of the σ/N line is one straight line on a log/log scale, then with

$$\sigma_1 = \sigma_2 \left(\frac{N_2}{N_1} \right)^e \quad \text{...(6)}$$

as the equation of that portion, the safety factor is calculated as:

$$S = \frac{1}{U^e} \quad \text{or} \quad \frac{1}{S^{(1/e)}} = \sum_l \frac{1}{S_l^{(1/e)}} \quad \text{...(7)}$$

where S_l is safety factor calculated for T_l and n_l.

The safety factor has to be calculated separately for pinion and wheel, for both bending and pitting. The safety factor is only valid for the required life used for each calculation.

7 Random reverse torques

Random reverse torques must be included in the stress spectra. They have an effect on the tooth root strength comparable to loading from both sides, as mentioned in 6.3.

Reverse torques affect the contact stress spectrum of the rear flank. Damage accumulation has to be considered separately for each flank side.

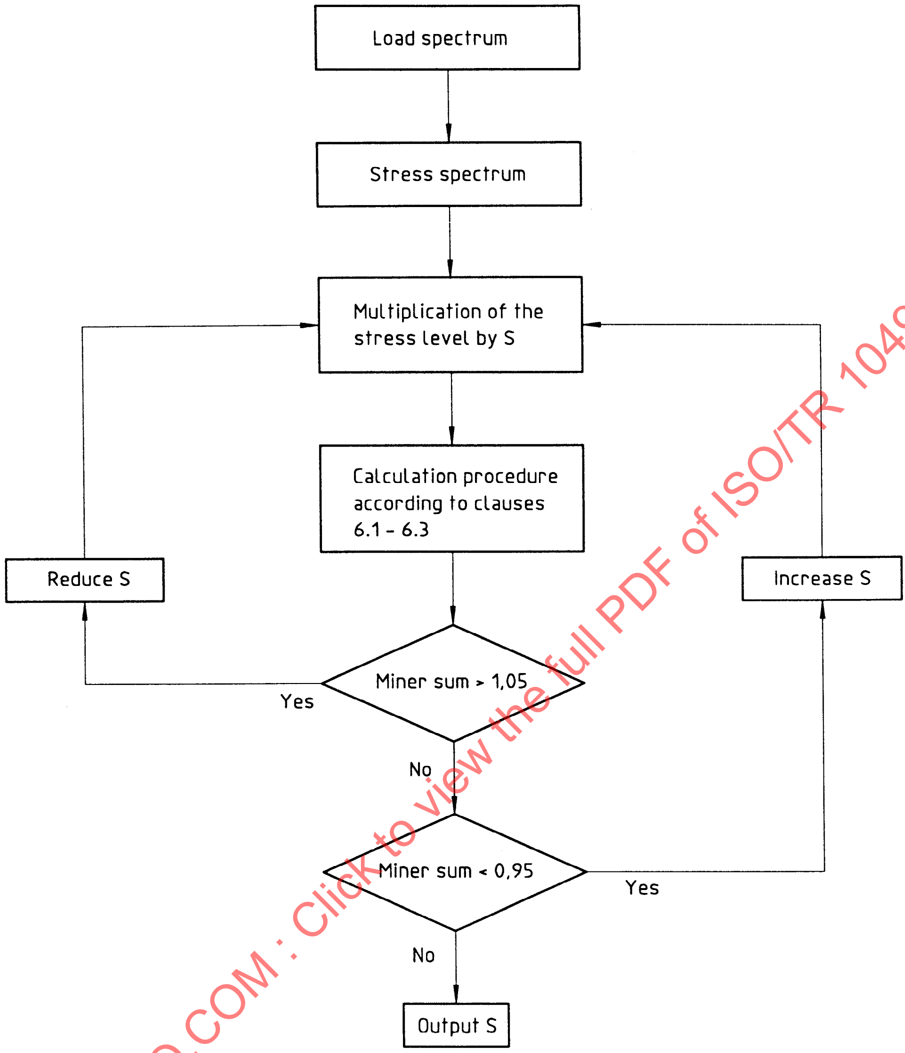


Figure 7 — Flow chart for the determination of the calculated safety factor for a given load spectrum

8 Reference values for application factor, K_A

If no load spectra are available, application factors from experience with similar machines may be used, depending on the operating mode of the driving and driven machine instead of calculating the service strength.

See annex A for tables for K_A.

Annex A
(informative)

Guide values for the application factor, K_A

The application factor, K_A, is used to modify the value of the tangential force F_t to take into account loads additional to nominal loads which are imposed on the gears from external sources. The empirical guidance values in table A.1 can be used (for industry gears and high speed gears).

Table A.1 - Application factor, K_A

Working characteristics of the driving machine	Working characteristics of the driven machine			
	Uniform	Light shocks	Moderate shocks	Heavy shocks
Uniform	1,00	1,25	1,50	1,75
Light shocks	1,10	1,35	1,60	1,85
Moderate shocks	1,25	1,50	1,75	2,00
Heavy shocks	1,50	1,75	2,00	2,25 or higher

The values apply to the nominal torque of the machine under consideration, or alternatively to the nominal torque of the driving motor, as long as this corresponds to the torque demand of the driving machine (see ISO 6336-1, clause 4.1).

The values only apply to transmissions which operate outside the resonance speed range under relatively steady loading. If operating conditions involve unusually heavy loading, motors with high starting torques, intermittent service or heavy repeated shock loading, or service brakes with a torque greater than the driving-motor, the safety of the static and limited life load capacity of the gears shall be verified (see ISO 6336-1, ISO 6336-2 and ISO 6336-3).

Examples:

- Turbine/generator
  In this system short circuit torques of up to 6 times the nominal torque can occur. Such overloads can be shed by means of safety couplings.

- Electric motor/compressor
  If pump frequency and torsional natural frequency coincide, considerable alternating stresses can occur.

- Heavy plate and billet rolling mills
  Initial pass-shock-torques up to 6 times the rolling torque shall be taken into account in these cases.

- Drives with synchronous motors
  Alternating torques up to 5 times the nominal torque can occur briefly (approximately 10 amplitudes) on starting; however, hazardous alternating torques can often be completely avoided by the appropriate detuning measures.

Information and numerical values provided here cannot be generally applied. The magnitude of the peak torque depends on the mass spring system, the forcing term, safety precautions (safety coupling, protection for unsynchronized switching of electrical machines), etc.

Thus, in critical cases, careful analysis should be demanded. It is then recommended that agreement is reached on suitable actions.

If special application factors are required for specific purposes, these shall be applied (e.g. because of a variable duty list specified in the purchase order, for marine gears according to the rules of a classification authority).

Where there are additional inertial masses, torques resulting from the flywheel effect are to be taken into consideration. Occasionally, braking torque provides the maximum loading and thus influences calculation of load capacity.

It is assumed the gear materials used should have adequate overload capacity. When materials used have only marginal overload capacity, designs should be laid out for endurance at peak loading.

The K_A value for moderate, average, and heavy shocks can be reduced by using hydraulic couplings or torque matched elastic couplings, and especially vibration attenuating couplings when the characteristics of the couplings so permit.

Table A.2 - Examples for driving machines with various working characteristics

Working characteristics	Driving machine
Uniform	Electric motor (e.g. DC motor), steam or gas turbine with uniform operation[1)] and small rarely occurring starting torques[2)]
Light shocks	Steam turbine , gas turbine, hydraulic or electric motor (large, frequently occurring starting torques[2)])
Moderate shocks	Multiple cylinder internal combustion engines
Heavy shocks	Single cylinder internal combustion engines

1) Based on vibration tests or on experience gained from similar installations.
2) See service life graphs Z_{NT}, Y_{NT} for the material in ISO 6336-2 and ISO 6336-3. Consideration of momentarily acting overload torques, see examples under table A.1.

Table A.3 - Industrial gears: Examples of working characteristics of driven machine

Working characteristics	Driven machines
Uniform	Steady load current generator; uniformly loaded conveyor belt or platform conveyor; worm conveyor; light lifts; packing machinery; feed drives for machine tools; ventilators; light-weight centrifuges; centrifugal pumps; agitators and mixers for light liquids or uniform density materials; shears; presses, stamping machines[1)]; vertical gear, running gear[2)].
Light shocks	Non-uniformly (i.e. with piece or batched components) loaded conveyor belts or platform conveyors; machine tool main drives; heavy lifts; crane slewing gear; industrial and mine ventilators; heavy centrifuges; centrifugal pumps; agitators and mixers for viscous liquids or substances of non-uniform density; multi-cylinder piston pumps; distribution pumps; extruders (general); calendars; rotating kilns; rolling mill stands[3)], (continuous zinc and aluminum strip mills, wire and bar mills).
Moderate shocks	Rubber extruders; continuously operating mixers for rubber and plastics; ball mills (light); wood-working machines (gang saws, lathes); billet rolling mills[3), 4)]; lifting gear; single cylinder piston pumps.
Heavy shocks	Excavators (bucket wheel drives); bucket chain drives; sieve drives; power shovels; ball mills (heavy); rubber kneaders; crushers (stone, ore); foundry machines; heavy distribution pumps; rotary drills; brick presses; de-barking mills; peeling machines; cold strip[3), 5)]; briquette presses; breaker mills.

1) Nominal torque = maximum cutting, pressing or stamping torque.
2) Nominal torque = maximum starting torque.
3) Nominal torque = maximum rolling torque.
4) Torque from current limitation.
5) K_A up to 2,0 because of frequent strip cracking.