INTERNATIONAL STANDARD

ISO 6336-6

Second edition
2019-11

Calculation of load capacity of spur and helical gears —

Part 6: Calculation of service life under variable load

Calcul de la capacité de charge des engrenages cylindriques à dentures droite et hélicoïdale —

Partie 6: Calcul de la durée de vie en service sous charge variable

Reference number
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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 procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

This second edition cancels and replaces the first edition (ISO 6336-6:2006), which has been technically revised. It also incorporates the Technical Corrigendum ISO 6336-6:2006/Cor.1:2007.

The main changes compared to the previous edition are as follows:

— in Annex A, examples have been revised;

— integration of Annex B "Equivalent cumulative damage".

A list of all parts in the ISO 6336 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

ISO 6336 (all parts) consists of International Standards, Technical Specifications (TS) and Technical Reports (TR) under the general title *Calculation of load capacity of spur and helical gears* (see Table 1).

— International Standards contain calculation methods that are based on widely accepted practices and have been validated.

— Technical Specifications (TS) contain calculation methods that are still subject to further development.

— Technical Reports (TR) contain data that is informative, such as example calculations.

The procedures specified in parts 1 to 19 of the ISO 6336 series cover fatigue analyses for gear rating. The procedures described in parts 20 to 29 of the ISO 6336 series are predominantly related to the tribological behavior of the lubricated flank surface contact. Parts 30 to 39 of the ISO 6336 series include example calculations. The ISO 6336 series allows the addition of new parts under appropriate numbers to reflect knowledge gained in the future.

Requesting standardized calculations according to the ISO 6336 series without referring to specific parts requires the use of only those parts that are currently designated as International Standards (see Table 1 for listing). When requesting further calculations, the relevant part or parts of the ISO 6336 series need to be specified. Use of a Technical Specification as acceptance criteria for a specific design need to be agreed in advance between the manufacturer and the purchaser.

Table 1 — Parts of the ISO 6336 series (STATUS AS OF DATE OF PUBLICATION)

Calculation of load capacity of spur and helical gears	International Standard	Technical Specification	Technical Report
Part 1: Basic principles, introduction and general influence factors	X		
Part 2: Calculation of surface durability (pitting)	X		
Part 3: Calculation of tooth bending strength	X		
Part 4: Calculation of tooth flank fracture load capacity		X	
Part 5: Strength and quality of materials	X		
Part 6: Calculation of service life under variable load	X		
Part 20: Calculation of scuffing load capacity (also applicable to bevel and hypoid gears) — Flash temperature method *(replaces: ISO/TR 13989-1)*		X	
Part 21: Calculation of scuffing load capacity (also applicable to bevel and hypoid gears) — Integral temperature method *(replaces: ISO/TR 13989-2)*		X	
Part 22: Calculation of micropitting load capacity *(replaces: ISO/TR 15144-1)*		X	
Part 30: Calculation examples for the application of ISO 6336-1, 2, 3, 5			X
Part 31: Calculation examples of micropitting load capacity *(replaces: ISO/TR 15144-2)*			X

Calculation of load capacity of spur and helical gears —

Part 6:
Calculation of service life under variable load

1 Scope

This document specifies the information and standardized conditions necessary for the calculation of the service life (or safety factors for a required life) of gears subject to variable loading for only pitting and tooth root bending strength.

If this scope does not apply, refer ISO 6336-1:2019, Clause 4.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 1122-1:1998, *Vocabulary of gear terms — Part 1: Definitions related to geometry*

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

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

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

3 Terms, definitions, symbols and abbreviated terms

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 6336-1 and ISO 1122-1:1998 apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at http://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

3.2 Symbols and abbreviated terms

For the purposes of this document, the symbols and abbreviated terms given in ISO 6336-1, ISO 1122-1:1998 and Table 2 apply.

Table 2 — Symbols and abbreviated terms used in this document

Abbreviated terms	
Term	**Description**
Eh	material designation for case-hardened wrought steel
GG	material designation for grey cast iron
GGG	material designation for nodular cast iron (perlitic, bainitic, ferritic structure)
GTS	material designation for black malleable cast iron (perlitic structure)
IF	material designation for flame or induction hardened wrought special steel
NT	material designation for nitrided wrought steel, nitriding steel
NV	material designation for through-hardened wrought steel, nitrided, nitrocarburized
St	material designation for normalized base steel ($\sigma_B < 800$ N/mm^2)
V	material designation for through-hardened wrought special steel, alloy or carbon ($\sigma_B \geq 800$ N/mm^2)

Symbols		
Symbol	**Description**	**Unit**
a	centre distance[a]	mm
b	face width	mm
d	diameter (without subscript, reference diameter[a])	mm
d_a	tip diameter[a]	mm
F	force or load	N
F_t	(nominal) transverse tangential load at reference cylinder per mesh	N
K	constant, factors concerning tooth load	—
K_A	application factor (Annex A shall apply for pitting and tooth root bending)	—
$K_{F\alpha}$	transverse load factor (bending)	—
$K_{F\beta}$	face load factor (bending)	—
$K_{H\alpha}$	transverse load factor (contact stress)	—
$K_{H\beta}$	face load factor (contact stress)	—
K_γ	mesh load factor	—
K_v	dynamic factor	—
m_n	normal module	mm
N	number of load cycles	—
N_i	number of load cycles to failure for bin i	—
N_L	number of load cycles of S-N curve	—
N_{LF}	number of load cycles for bending damage	—

[a] For external gears a, d, d_a, z_1 and z_2 are positive; for internal gearing, a, d, d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.

Table 2 *(continued)*

Symbols		
Symbol	**Description**	**Unit**
N_{LH}	number of load cycles for pitting damage	—
$N_{L\,ref}$	number of load cycles for endurance limit	—
$n_{D,i}$	number of load cycles for the equivalent damage fatigue curve (Annex B)	—
$n_{D,REF}$	number of load cycles for the equivalent damage fatigue curve (reference) (Annex B)	—
$n_{eq,i}$	equivalent number of load cycles (Annex B)	—
$n_{eq,3REF}$	equivalent number of load cycles (reference) (Annex B)	—
n_i	number of load cycles for bin *i*	—
n_{Hi}	number of load cycles for contact stress for bin *i*	—
n_{Fi}	number of load cycles for tooth root stress for bin *i*	—
$n_{nom,i}$	number of load cycles for nominal stress in bin *i* (Annex B)	—
p	slope of the S-N curve	—
S	safety factor	—
S_F	safety factor for bending	—
S_H	safety factor for pitting	—
T	torque (pinion torque unless specified otherwise)	N·m
T_{eq}	equivalent torque	N·m
T_i	torque for bin *i*	N·m
T_n	nominal torque	N·m
U	sum of individual damage parts	—
U_i	individual damage parts for bin *i*	—
u	gear ratio $(\lvert z_2 / z_1\rvert) \geq 1$[a]	—
x	profile shift coefficient	—
Y	factor related to tooth root bending	—
Y_B	rim thickness factor	—
Y_{DT}	deep tooth factor	—
Y_F	tooth form factor, for the influence on nominal tooth root stress with load applied at the outer point of single pair tooth contact	—
Y_{NT}	life factor for tooth root stress for reference test conditions	—
$Y_{R\,rel\,T}$	relative surface factor, the quotient of the gear tooth root surface factor of interest divided by the tooth root surface factor of the reference test gear, $Y_{R\,rel\,T} = Y_R/Y_{RT}$	—
Y_S	stress correction factor, for the conversion of the nominal tooth root stress, determined for application of load at the outer point of single pair tooth contact, to the local tooth root stress	—
[a] For external gears a, d, d_a, z_1 and z_2 are positive; for internal gearing, a, d, d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		

Table 2 *(continued)*

Symbols		
Symbol	**Description**	**Unit**
Y_{ST}	stress correction factor, relevant to the dimensions of the standard reference test gears	—
Y_β	helix angle factor (tooth root)	—
$Y_{\delta\ rel\ T}$	relative notch sensitivity factor, the quotient of the gear notch sensitivity factor of interest divided by the standard reference test gear factor, $Y_{\delta\ rel\ T} = Y_\delta / Y_{\delta T}$	—
Z	factor related to contact stress	—
Z_B, Z_D	single pair tooth contact factors for the pinion, for the wheel	—
Z_E	elasticity factor	$(N/mm^2)^{0,5}$
Z_H	zone factor	—
Z_L	lubricant factor	—
Z_N	life factor for contact stress	—
Z_{NT}	life factor for contact stress for reference test conditions	—
Z_R	roughness factor affecting surface durability	—
Z_v	velocity factor	—
Z_W	work hardening factor	—
Z_X	size factor (pitting)	—
Z_β	helix angle factor (pitting)	—
Z_ε	contact ratio factor (pitting)	—
z	number of teeth[a]	—
z_n	virtual number of teeth of a helical gear	—
α	pressure angle (without subscript, at reference cylinder)	°
β	helix angle (without subscript, at reference cylinder)	°
σ	normal stress	N/mm^2
σ_D	stress value used to describe the equivalent damage fatigue curve (Annex B)	N/mm^2
σ_F	tooth root stress	N/mm^2
σ_{FG}	tooth root stress limit	N/mm^2
σ_{Fi}	tooth root stress for bin i	N/mm^2
σ_{FP}	permissible bending stress	N/mm^2
$\sigma_{F\ lim}$	nominal stress number (bending)	N/mm^2
σ_G	stress value used to describe the permissible S-N curve	N/mm^2
σ_H	contact stress	N/mm^2
σ_{HG}	pitting stress limit	N/mm^2
σ_{Hi}	contact stress for bin i	N/mm^2
[a] For external gears a, d, d_a, z_1 and z_2 are positive; for internal gearing, a, d, d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.		

Table 2 *(continued)*

Symbols		
Symbol	**Description**	**Unit**
σ_{HP}	permissible contact stress	N/mm^2
σ_P	stress value used to describe the S-N curve (Annex B)	N/mm^2
σ_{REF}	reference permissible stress level	N/mm^2
σ_i	stress for bin i	N/mm^2
$\sigma_{nom,i}$	nominal stress for bin i (Annex B)	N/mm^2

[a] For external gears a, d, d_a, z_1 and z_2 are positive; for internal gearing, a, d, d_a and z_2 have a negative sign, z_1 has a positive sign. All calculated diameters have a negative sign for internal gearing.

4 General

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 dynamic mass elastic properties of the system.

These variable loads (stresses) may be determined by such procedures as

— experimental measurement of the operating loads at the machine in question,

— estimation of the spectrum, if this is known, for a similar machine with a similar operating mode, and

— calculation, using known external excitation and a mass elastic simulation of the drive system, preferably followed by experimental testing to validate the calculation.

To obtain the load spectra for fatigue damage calculation, the range of the measured (or calculated) loads is divided into bins or classes. Each bin contains the number of load occurrences recorded in its load range. A widely-used number of bins is 64. These bins can be of an equal size, but it is usually better to use larger bin sizes at the lower loads and smaller bin sizes at the upper loads in the range. In this way, the most damaging loads may be limited to fewer calculated stress cycles and the resulting design is more accurate regarding the effective load. It is recommended that a zero-load bin be included so that the total time used to rate the gears matches the design operating life. For consistency, the usual presentation method is to have the highest torque associated with the lowest numbered bins, such that the most damaging conditions appear towards the top of any table.

The cycle count for the load class corresponding to the load value for the highest loaded tooth is incremented at every load repetition. Table 3 shows as an example of how the torque classes defined in Table 4 can be applied to specific torque levels and correlated numbers of cycles.

Table 3 — Torque classes/numbers of cycles — Example: classes 38 and 39 (see Table 4)

Torque class, T_i N·m	**Number of cycles, n_i**
11 620 ≤ T_{38} ≤ 12 619	n_{38} = 237
10 565 ≤ T_{39} ≤ 11 619	n_{39} = 252

The torques used to evaluate the 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 lifetime, the possibility that there might be torque peaks not frequent enough to be evaluated in that measured spectrum shall be considered. These transient peaks can have an effect on the gear life. Therefore, the evaluated time period could have to be extended to capture extreme load peaks.

Stress spectra concerning bending and pitting can be obtained from the load (torque).

The tooth root stress may also be measured by means of strain gauges in the fillet. The relevant contact stress may be calculated from the measurements.

Table 4 — Example of torque spectrum (with unequal bin sizes for a reducing number of bins) (see Annex C)

Data	Pinion			%	Time	
	Torque N·m		Load cycles			
Bin no.	minimum	maximum			s	h
1	25 502	25 578	0	0,00	0	0
2	25 424	25 501	0	0,00	0	0
3	25 347	25 423	14	0,37	24	0,006 7
4	25 269	25 346	8	0,21	14	0,003 9
5	25 192	25 268	5	0,13	9	0,002 5
6	25 114	25 191	8	0,21	14	0,003 9
7	25 029	25 113	16	0,42	28	0,007 8
8	24 936	25 028	8	0,21	14	0,003 9
9	24 835	24 935	5	0,13	9	0,002 5
10	24 727	24 834	11	0,29	19	0,005 3
11	24 610	24 726	16	0,42	28	0,007 8
12	24 479	24 609	19	0,50	33	0,009 2
13	24 331	24 478	14	0,37	24	0,006 7
14	24 168	24 330	14	0,37	24	0,006 7
15	23 990	24 168	11	0,29	19	0,005 3
16	23 796	23 989	15	0,39	26	0,007 2
17	23 579	23 796	31	0,81	52	0,014 4
18	23 339	23 579	28	0,73	47	0,013 1
19	23 076	23 338	36	0,94	62	0,017 2
20	22 789	23 075	52	1,36	88	0,024 4
21	22 479	22 788	39	1,02	66	0,018 3
22	22 138	22 478	96	2,51	163	0,045 3
23	21 766	22 137	106	2,77	180	0,050 0
24	21 363	21 765	49	1,28	83	0,023 1
25	20 929	21 362	117	3,05	200	0,055 6
26	20 463	20 928	124	3,24	212	0,058 9
27	19 960	20 463	61	1,59	104	0,028 9
28	19 417	19 959	140	3,65	238	0,066 1
29	18 836	19 416	148	3,86	253	0,070 3
30	18 216	18 835	117	3,05	200	0,055 6
31	17 557	18 215	121	3,16	206	0,057 2
32	16 851	17 556	174	4,46	297	0,082 5
33	16 100	16 851	185	4,83	316	0,087 8
34	15 301	16 099	196	5,11	334	0,092 8

Table 4 *(continued)*

Data	Pinion			%	Time	
	Torque N·m		Load cycles			
Bin no.	minimum	maximum			s	h
35	14 456	15 301	207	5,40	352	0,097 8
36	13 565	14 456	161	4,20	274	0,076 1
37	12 620	13 564	168	4,38	286	0,079 4
38	11 620	12 619	237	6,18	404	0,112 2
39	10 565	11 619	252	6,58	429	0,119 2
40	9 457	10 565	263	6,86	449	0,124 7
41	8 294	9 456	275	7,18	468	0,130 0
42	7 070	8 294	178	4,65	303	0,084 2
43	5 783	7 069	103	2,69	176	0,048 9
44	4 434	5 782	7	0,18	12	0,003 3
45	3 024	4 434	0	0,00	0	0
46	1 551	3 023	0	0,00	0	0
47	1	1 550	0	0,00	0	0
48	0	0	0	0,00	6 041 469	1 678,2
		Total ≥	3 832	100,0	6 048 000	1 680

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 fatigue life of a gear is a measure of its ability to accumulate discrete damage until failure occurs.

The fatigue life calculation requires

a) the stress spectrum,

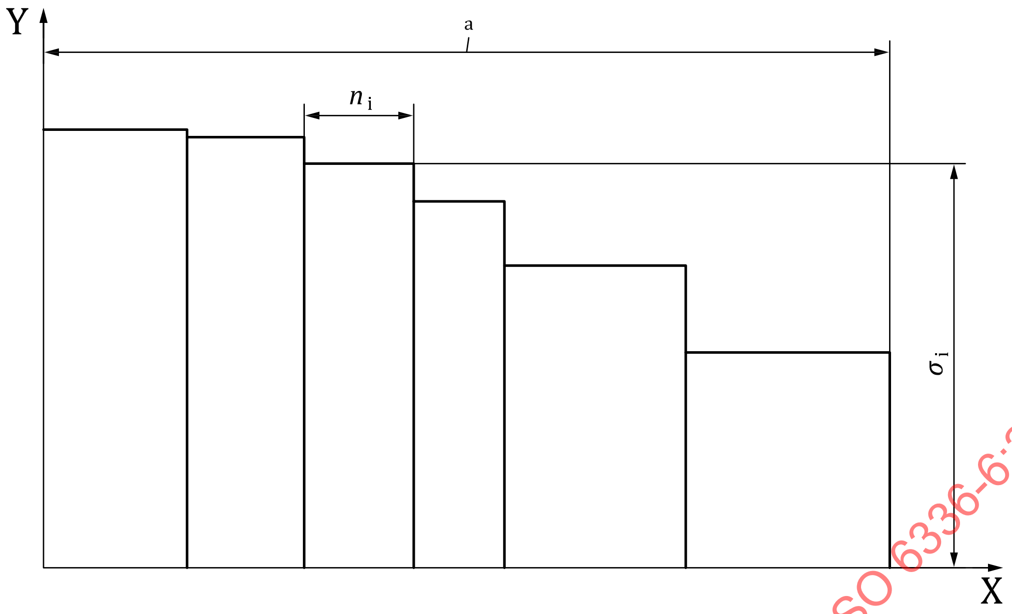
b) material fatigue properties, and

c) a damage accumulation method.

The stress spectrum is discussed in 5.1.

Strength values based on material fatigue properties are chosen from applicable S-N curves. Many specimens shall 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 1. Figure 2 shows a cumulative contact stress spectrum with an S-N curve for specific material fatigue properties.



Key

X cumulative number of applied cycles, N (log)

Y stress, σ (log)

a Load spectrum, $\Sigma\, n_i$, total cycles.

Figure 1 — Example for a cumulative stress spectrum

Linear, non-linear and relative methods are used. Further information can be found in the literature (References [4], [9], [10] and [17]).

4.3 Palmgren-Miner rule

The Palmgren Miner rule — in addition to 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 3 000 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.

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

$$\sum_i \frac{n_i}{N_i} = 1{,}0 \qquad (1)$$

where

n_i is the number of load cycles for bin i;

N_i is the number of load cycles to failure for bin i (taken from the appropriate S-N curve).

If there is an endurance limit (upper, horizontal line beyond the knee in Figure 2), the calculation is only done for stresses above this endurance limit.

If the appropriate S-N curve shows no endurance limit (decreasing line beyond the knee (kink point) in Figure 3), the calculation shall be done for all stress levels. For each stress level, i, the number of cycles to failure, N_i, shall be taken from the corresponding stress level of the S-N curve. Other damage accumulation (including non-linear) hypotheses in addition to the herein described method and permissible damage sums other than one may be used upon agreement of the purchaser and the gear box manufacturer.

5 Calculation of service strength on the basis of single-stage strength according to ISO 6336 series

5.1 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) and has been chosen because it is widely known and easy to apply; the choice does not imply that the method is superior to the others described in the literature (References [4], [9], [10] and [17]).

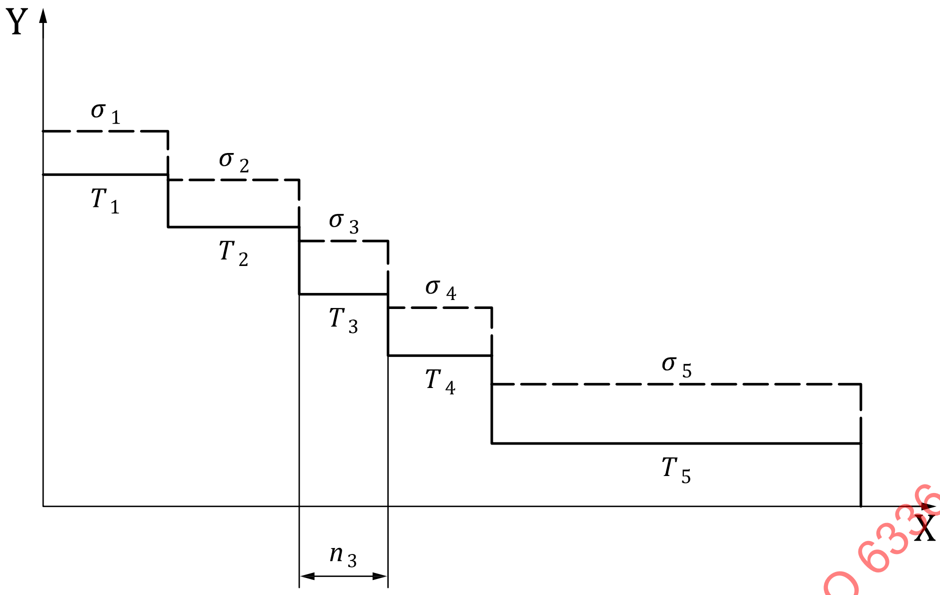
From the individual torque classes, the torques at the upper limit of each torque class and the associated numbers of cycles shall be listed (see Table 5 for an example).

Table 5 — Torque classes/numbers of cycles — Example: classes 38 and 39

Upper limit of torque class[a], T_i N·m	Number of cycles, n_i
$T_{38} < 12\ 620$	$N_{38} = 237$
$T_{39} < 11\ 620$	$N_{39} = 252$

[a] For conservative calculation, sufficiently accurate for a high number of torque classes.

Based on the load spectrum (T_i, n_i), the effective stress levels σ_i are determined by help of the methods described in ISO 6336-2 and ISO 6336-3 for the pinion or the wheel to obtain a stress spectrum (σ_i, n_i) as shown in Figure 2.



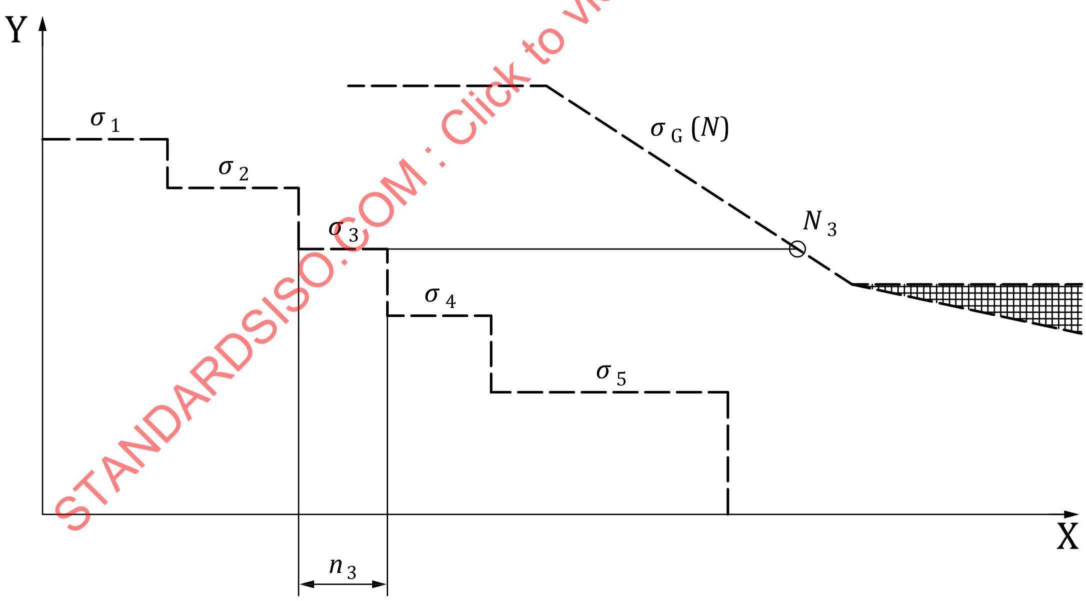
Key

X number of load cycles, *N* (log)

Y stress, *σ* (log) or torque, *T* (log)

Figure 2 — Load and stress spectrum

The stress spectrum (σ_i, n_i) combined with the S-N curve $\sigma_G(N)$, allows the determination of the allowable number of cycles N_i for each stress level σ_i (see Figure 3).



Key

X number of load cycles, *N* (log)

Y stress, *σ* (log)

$\sigma_G(N)$ stress value used to describe the permissible S-N curve

NOTE For each level of stress σ_i with a number of cycles n_i the allowable number of cycles N_i can be determined by help of the methods described in ISO 6336-2 and ISO 6336-3 for the pinion or the wheel.

Figure 3 — Stress spectrum and S-N curve

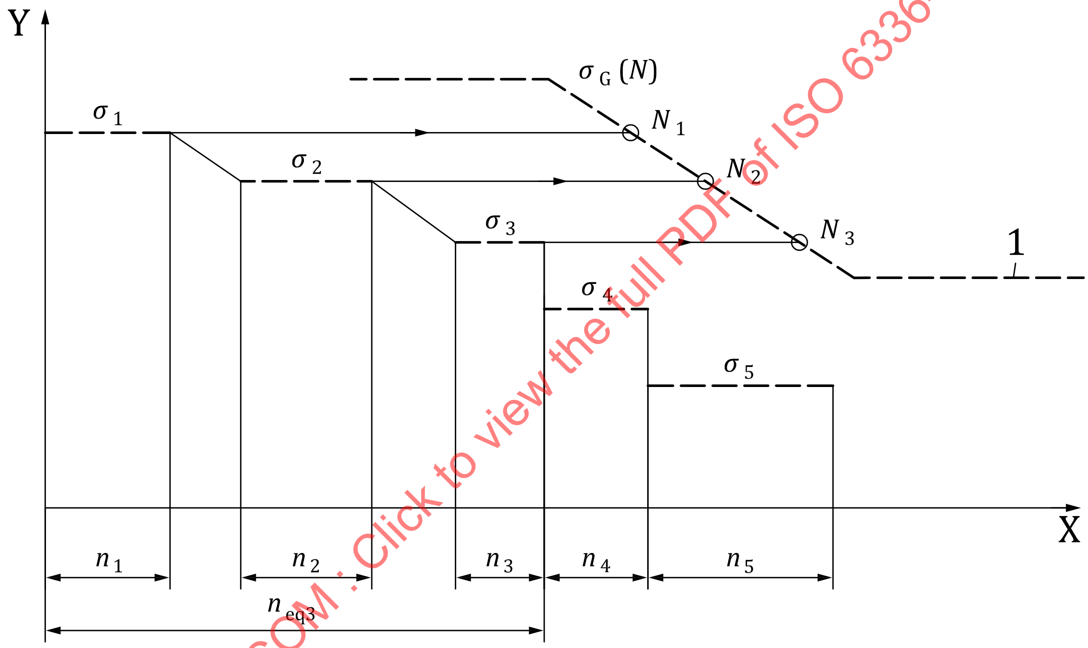
NOTE 1 The representation of the cumulative stress spectrum entirely below the S-N curve does not imply that the part will survive the total accumulative number of stress cycles. This information can be gained from a presentation as shown in Figure 7.

NOTE 2 The value σ_G is either σ_{HG} or σ_{FG}.

NOTE 3 The log scale on the vertical axis are different for the stress and the torque in Figure 2.

To evaluate the cumulative damage graphically, it is necessary to shift the damaging load cycles of each stress bin from stress level σ_i to stress level σ_{i+1} in order to keep the cumulated damage constant. Graphically this is equivalent with drawing a line of the same slope than the S-N curve from the extremity of the stress bin σ_i (see respectively Figures 4 and 5, for the respective cases with and without an endurance limit) to the stress level σ_{i+1}.

The equivalent cumulative damage for the given load spectrum is in Figure 4 the ratio n_{eq3}/N_3 for the stress level σ_3 (respectively n_{eq4}/N_4 for the stress level σ_4 in Figure 5).



Key

X number of cycles, N (log)

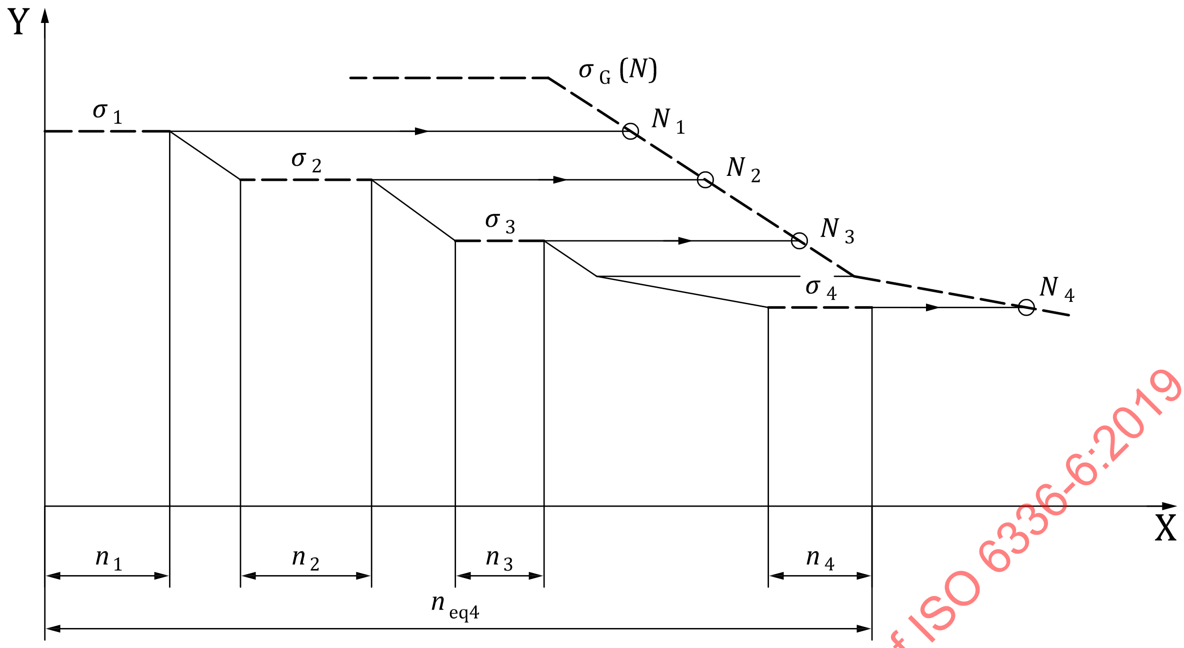
Y stress, σ (log)

1 endurance limit

$\sigma_G(N)$ stress value used to describe the permissible S-N curve

NOTE As level σ_4 and σ_5 are below the endurance limit, these bins don't need to be shifted.

Figure 4 — Cumulated stress spectrum and fatigue curve limit when there is an endurance limit



Key

X number of cycles, N (log)

Y stress, σ (log)

$\sigma_G(N)$ stress value used to describe the permissible S-N curve

Figure 5 — Cumulated stress spectrum and S-N curve for life factor < 1,0 in the range of long life

The stress spectra for the tooth root (σ_{Fi}, n_{Fi}) and the tooth flank (σ_{Hi}, n_{Hi}) 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 the procedure, see 5.2).

With the stress spectra obtained in this way, the calculated values are compared with the strength values (S-N curves, damage lines) determined according to 5.3 using the Palmgren Miner rule, see 4.3. For a graphical representation, see Figure 3.

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

$$U_i = \frac{n_i}{N_\mathrm{i}} \tag{2}$$

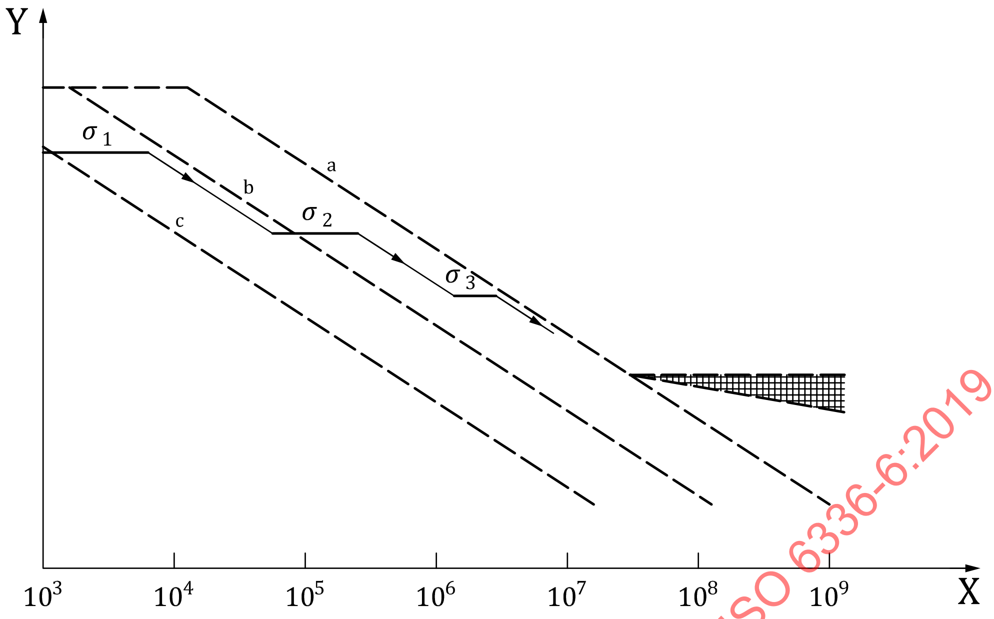
The sum of the individual damage parts, U_i, results in the cumulated damage condition U, which shall be less than or equal to unity.

$$U = \sum_\mathrm{i} U_\mathrm{i} = \sum_i \frac{n_\mathrm{i}}{N_\mathrm{i}} \le 1{,}0 \tag{3}$$

NOTE 4 As noted in 4.3, permissible damage sums other than 1,0 can be used.

NOTE 5 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. Figure 6 shows a presentation from which it can be concluded whether the part will survive the total number of stress cycles.



Key

X number of load cycles, N (log)

Y stress, σ (log)

a Damage sum 100 %.

b Damage sum 10 %.

c Damage sum 1 %.

Figure 6 — Accumulation of damage

In addition, safety factors applied to static load strength should be calculated for the highest stress of the design life. This document is not applicable to stress levels greater than the static stress limit, since stresses in this range can exceed the elastic limit of the gear tooth in bending or in surface contact pressure. In addition, safety factors applied to the static load strength should be calculated for the highest stress of the design life. The highest stress could be either the maximum stress in the load spectrum or an extreme transient load that is not considered in the fatigue analysis. Depending on the material and the load imposed, a single stress cycle above the limited-life range could result in plastic yielding of the gear tooth. The static load strength can be determined according to ISO 6336-2 for pitting and ISO 6336-3 for bending.

5.2 Calculation of stress spectra

For each level i of the torque spectrum, the actual stress, σ_i, shall be determined separately for contact and bending stress in accordance with the following formulae.

— For contact stress (ISO 6336-2:, Method B):

$$\sigma_{\mathrm{H}i} = Z_{\mathrm{H}} \cdot Z_{\mathrm{E}} \cdot Z_{\varepsilon} \cdot Z_{\beta} \cdot Z_{\mathrm{BD}} \cdot \sqrt{\frac{2\,000 \cdot T_i}{d_1^2 \cdot b} \cdot \frac{u+1}{u} \cdot K_{\gamma i} \cdot K_{\mathrm{v}i} \cdot K_{\mathrm{H}\beta i} \cdot K_{\mathrm{H}\alpha i}} \tag{4}$$

— For bending stress (ISO 6336-3:, Method B):

$$\sigma_{\mathrm{F}i} = \frac{2\,000 \cdot T_i}{d_1 \cdot b \cdot m_{\mathrm{n}}} \cdot Y_{\mathrm{F}} \cdot Y_{\mathrm{S}} \cdot Y_{\beta} \cdot Y_{\mathrm{B}} \cdot Y_{\mathrm{DT}} \cdot K_{\gamma i} \cdot K_{\mathrm{v}i} \cdot K_{\mathrm{F}\beta i} \cdot K_{\mathrm{F}\alpha i} \tag{5}$$

The value K_{A}, defined as the 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.

5.3 Determination of pitting and bending strength values

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

Where teeth are loaded in both directions (e.g. idler gear), the values determined for the tooth root strength shall be reduced according to ISO 6336-3.

For contact stress, damage accumulation shall be calculated individually for each flank side if both flanks are loaded differently.

5.4 Determination of safety factors

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

The safety factor, *S*, shall be calculated separately for the pinion and the wheel, each for both bending and pitting. The safety factor is only valid for the required life used for each calculation. Annex C shows an example for calculating *S*.

NOTE Stresses calculated during the iteration process by multiplying the contact stress according to ISO 6336-2 or bending stress according to ISO 6336-3 by the safety factor *S* can be above the static load strength (see for example Figure C.1). This does not mean that these stresses will occur in real operation.

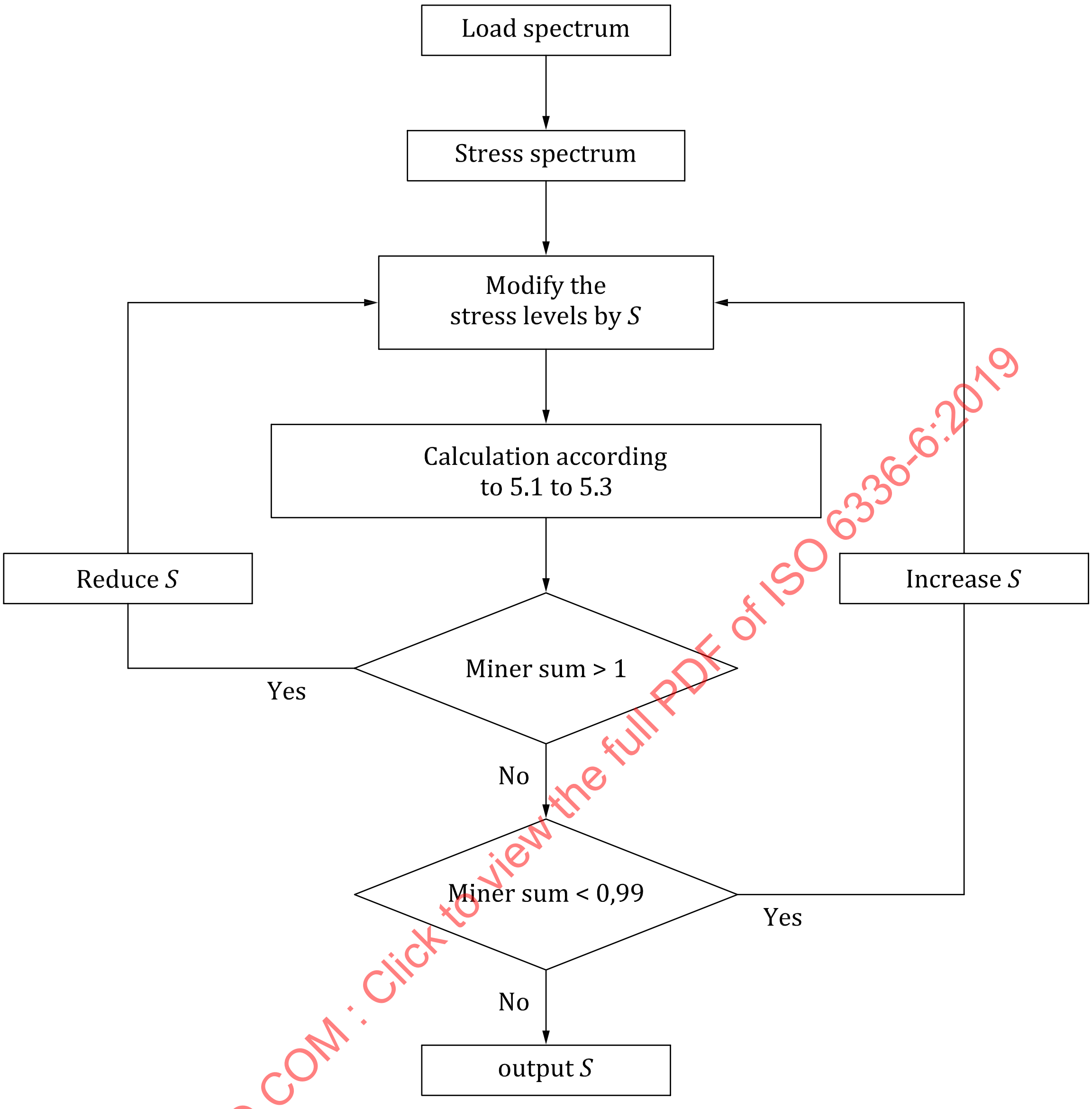


Figure 7 — Flow chart for the determination of the calculated safety factor for a given load spectrum

Annex A
(normative)

Determination of application factor, K_A, from given load spectrum using equivalent torque, T_{eq}

A.1 Purpose

The following calculation method is useful for a first estimation during the gear design stage, where the geometry data of a gear drive is not fixed. This calculation is only valid for the tooth root breakage and pitting damages. If this method is used for determining application factors for other failure modes, this shall be agreed between the purchaser and the gear box manufacturer.

Application factor K_A shall be determined separately for the tooth root breakage and the pitting resistance, both for the pinion and the wheel. The highest of these four values shall be used for a gear rating in conformance with ISO 6336 series.

A.2 Application factor, K_A

The application factor K_A is defined as the ratio between the equivalent torque and the nominal torque:

$$K_A = \frac{T_{eq}}{T_n} \tag{A.1}$$

where

T_n is the nominal torque;

T_{eq} is the equivalent torque.

The equivalent torque can be calculated as a simplification according to Formula (A.2). In general, it is recommended to calculate T_{eq} according to the method described in A.3. As a simplification, Formula (A.2) can be used, but it may be on the unsafe side, especially if high number of load cycles occur. Therefore in all cases, it is recommended to use the method described in A.3.

$$T_{eq} = \left(\frac{n_1 T_1^p + n_2 T_2^p + \ldots}{n_1 + n_2 + \ldots} \right)^{\frac{1}{p}} \tag{A.2}$$

where

n_i is the number of cycles for bin i;

T_i is the torque for bin i;

p is the slope of the S-N curve, see Table A.1.

The slope of the S-N curves used by ISO 6336 series determines that the number of bins to be used in Formula (A.2) may be limited to the bin at which the number of load cycles for endurance limit, $N_{L\ ref}$, is reached, see A.3.3.

A.3 Determination of the equivalent torque, T_{eq}

A.3.1 General

For this procedure, the load spectrum, the slopes of the S-N curves, p, and the number of load cycles for endurance limit, $N_{L\,ref}$, at the reference point shall be known.

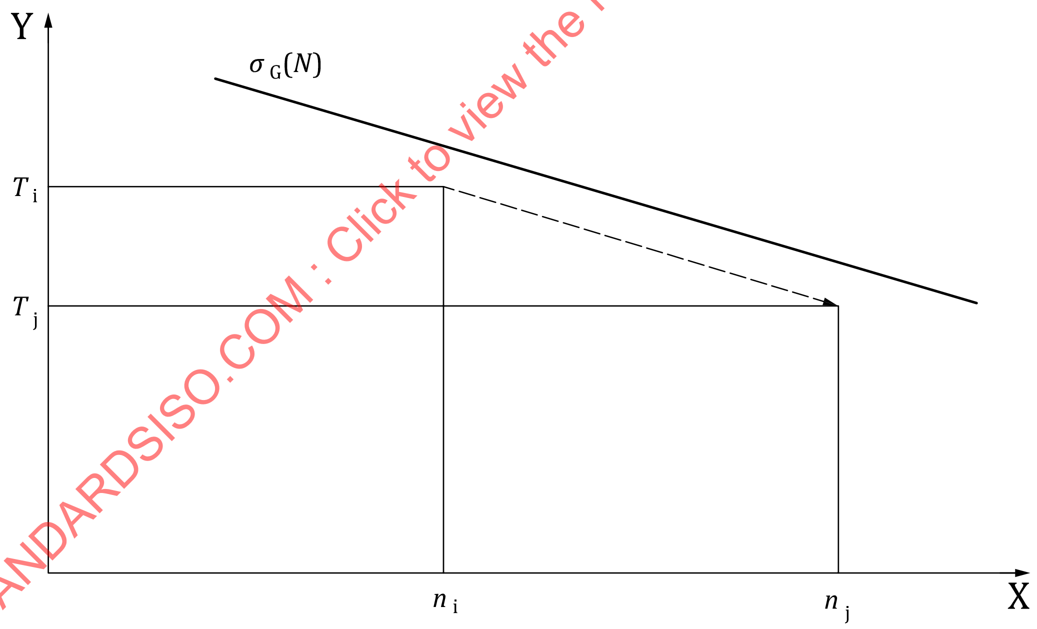
A.3.2 Basis

The following method applies for a design case where the S-N curve is simplified by ignoring all damage which occurs at stresses below some stress limit. It is based upon the fact that while the position of the endurance limit in terms of stress is not known in relation to the gear until the design is available, the position of that endurance limit in terms of cycles does not change as the gear design changes.

It is possible that the reference permissible stress level, σ_{REF}, is below the considered bins, when calculating K_A according to this annex. If, later in the gear design process, it shows, that there is a significant number of unconsidered load cycles above the reference permissible stress level, σ_{REF}, a further detailed calculation according to the main part of this standard is required.

Furthermore, a torque T_i in the bin i can be replaced by a torque T_j in a new bin, j, so that the damage caused by the torque T_i is the same as that caused by the torque T_j. This is shown in Figure A.1 and can be expressed by Formula (A.3).

$$T_i^p n_i = T_j^p n_j \tag{A.3}$$



Key

X number of load cycles, N (log)

Y torque, T (log)

Figure A.1 — Load bins with equal damage behaviour according to Formula (A.3)

A.3.3 Calculation procedure

The load bins shall be denoted as (T_i, n_i) and numbered in descending order of torque, where T_1 is the highest torque. Then the number of cycles n_1 at torque T_1 is equivalent in terms of damage to a larger number of cycles $n_{eq,1}$, at a lower torque T_2, where, according to Formula (A.3):

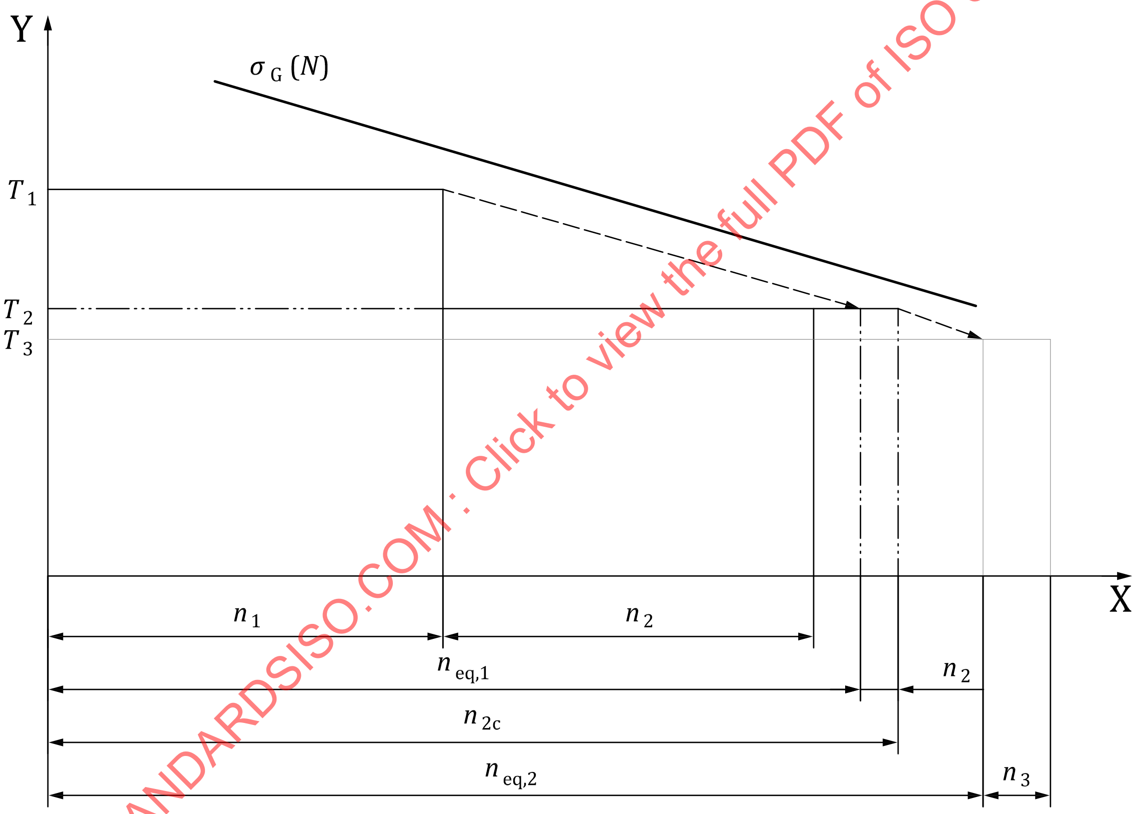
$$n_{eq,1} = n_1 \left(\frac{T_1}{T_2} \right)^p \quad \text{(A.4)}$$

As $n_{2c} = n_2 + n_{eq,1}$, then bins 1 and 2 can be replaced by a single bin (T_2, n_{2c}), see Figure A.2.

Similarly, the cycles n_{2c} at torque T_2 are equivalent to $n_{eq,2}$ at T_3, where

$$n_{eq,2} = n_{2c} \left(\frac{T_2}{T_3} \right)^p \quad \text{(A.5)}$$

As $n_{3c} = n_3 + n_{eq,2}$, then bins 1, 2 and 3 can be replaced by a single bin (T_3, n_{3c}).



Key

X number of load cycles, N (log)

Y torque, T (log)

NOTE The 2 bins of n_2 cycles have different length as the scale of the X axis is logarithmic.

Figure A.2 — Bins (T_1, n_1) and (T_2, n_2) replaced by (T_2, n_{2c})

This procedure shall be stopped when $n_{eq,i}$ reaches the number of load cycles for endurance limit, $N_{L\,ref}$. Once the endurance limit is reached, ensure that no significant numbers of load cycles above the endurance limit have been ignored. If so, further detailed calculation according to the main part of this standard is required.

The required equivalent torque T_{eq} is now between 2 limits:

$$T_i < T_{eq} < T_{i-1} \tag{A.6}$$

or

$$\frac{T_i}{T_n} < K_A < \frac{T_{i-1}}{T_n} \tag{A.7}$$

and can be found by linear interpolation on a log-log basis.

The slope exponent, p, and the number of load cycles for endurance limit, $N_{L\,ref}$, are a function of the material, heat treatment and damage machanism. Values to be used in Formulae (A.2), (A.4) and (A.5) are shown in Table A.1.

Table A.1 — Exponent p of S-N curve and number of load cycles for endurance limit, $N_{L\,ref}$

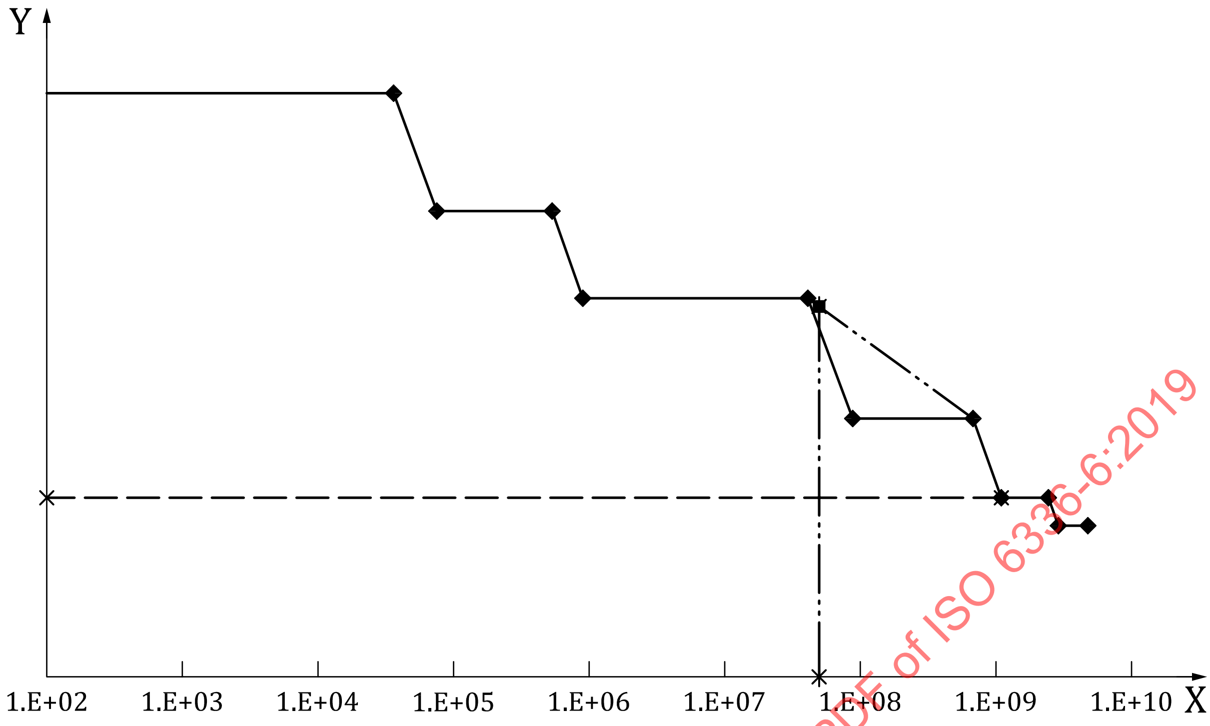
Material (acc. ISO 6336-5)	Pitting		Tooth root bending	
	p[a]	$N_{L\,ref}$	p	$N_{L\,ref}$
St, V, GGG (perl., bai.), GTS (perl.) (limited pitting according to ISO 6336-2)	6,774 8	10×10^6	6,224 9	3×10^6
St, V, GGG (perl., bai.), GTS (perl.) (no pitting according to ISO 6336-2)	6,611 2	50×10^6		
EH, IF (limited pitting according to ISO 6336-2)	6,774 8	10×10^6	8,737 8	3×10^6
EH, IF (no pitting according to ISO 6336-2)	6,611 2	50×10^6		
GG, GGG (ferr.), NT (nitr.), NV (nitr.)	5,709 1	2×10^6	17,035	3×10^6
NV (nitrocar.)	15,716	2×10^6	84,003	3×10^6

[a] Values p for pitting are given for the torque; to convert for the stress, these values shall be doubled.

A.4 Example

An example is shown in Figure A.3 and the corresponding Table A.2. In the rightmost column of the table a switch is shown that indicates when the endurance limit has been reached. In this example the application factor K_A is between 1,08 and 1,21. From the fact that on row 3 the value of n_{ie} is very close to the endurance limit, the interpolation will give $K_A = 1{,}201$.

It is important to note that this value of K_A should only be used with the same nominal torque used (950 kN m) and with the life factors which match the number of load cycles for endurance limit used (50×10^6), when doing the gear design.



Key

X number of load cycles, N (log)

Y torque, T (log)

loading spectrum

log-log line for interpolation

$50 \cdot 10^6$ load cycles

nominal torque, T_n

Figure A.3 — Load spectrum with corresponding equivalent torque, T_{eq}

Table A.2 — Example for the calculation of K_A from the load spectrum

Cumulative damage/calculation of K_A							
Flank				Nominal torque T_n in kNm			950
Ratio to this gear u:		75		Number of blade torque			20
Contact per revolution:		1		Slope exponent, p, from Table A.1			6,611 2
Speed in cycles per minute		1 500		Number of load cycles for endurance limit, $N_{L\ ref}$			50,0 × 10^6
Bin i	**Blade torque** T_i	**Torque ratio** T_i/T_n	**Operating hours** h	**Cycles** n_i	**Equivalent from row above** n_{ic}	**Total** $n_{eq,i}$	**Switch**
1	1 400	1,47	0,400	36 000	—	36 000	0
2	1 250	1,32	5,05	454 500	76 154	530 654	0
3	1 150	1,21	450	40 500 000	920 910	41 420 910	0
4	1 025	1,08	6 520	586 800 000	88 635 476	675 435 476	1
5	950	1,00	15 000	1 350 000 000	1 116 233 847	2 466 233 847	1
6	925	0,97	21 500	1 935 000 000	2 941 740 296	4 876 740 296	1
NOTE 1 The calculation is done for pitting (no pitting permissible) which justifies values for the exponent, p.							
NOTE 2 The switch in the rightmost column is equal to 0 if the number of total cycles is less than or equal to 50,0 × 10^6.							
NOTE 3 On each row the equivalent from row above n_{ia} is calculated by Formula A.5.							

The equivalent torque T_{eq} for 50 million load cycles is between the ends of bin 3 and 4 and is obtained by using log-log interpolation as follows:

— The slope exponent for log-log interpolation is given by:

$$p_{interpol} = \frac{\log\left(\frac{T_3}{T_4}\right)}{\log\left(\frac{n_{eq,4}}{n_{eq,3}}\right)} = \frac{\log\left(\frac{1\,150}{1\,025}\right)}{\log\left(\frac{675\,435\,476}{41\,420\,910}\right)} = 0{,}041\,22 \quad \text{(A.8)}$$

— The equivalent torque for log-log interpolation is given by:

$$T_{eq-50\ million} = T_4 \cdot \left(\frac{n_{eq,4}}{50 \cdot 10^6}\right)^{p_{interpol}} = 1\,025 \cdot \left(\frac{675\,435\,476}{50 \cdot 10^6}\right)^{0{,}041\,22} = 1\,141{,}11 \quad \text{(A.9)}$$

Consequently, the application factor is calculated as follows:

$$K_A = \frac{T_{eq-50\ million}}{T_n} = \frac{1\,141{,}11}{950} = 1{,}201 \quad \text{(A.10)}$$

NOTE In this calculation, all load levels between the nominal torque T_n and $T_{eq\text{-}50million}$ are ignored for the calculation of K_A.

Annex B
(informative)

Equivalent cumulative damage

B.1 Purpose

In order to generalize the equivalence in damage on the basis of Figures 2, 3 and 4 it is possible to calculate an equivalent load for the given load spectrum in term of damage under the assumption of linear damage accumulation.

For a given stress level i and corresponding n_i cycles, the individual damage U_i is defined by Formula (2), considering that the fatigue curve for the material allows N_i cycles before damage.

The equivalent cumulative damage of a given load spectrum is given by the sum of individual damages U_i and can be calculated according to Formula (3).

B.2 Determination of the equivalent stress damage curve, σ_{eq}

B.2.1 General

For this procedure, the load spectrum (T_i, n_i), the slopes of the considered S-N curves, p, and the number of load cycles, N_L, at the reference point shall be known (see Table B.1).

Table B.1 — Exponent p and number of load cycles N_L

Material (see ISO 6336-5)	Pitting		Tooth root bending	
	p[a]	N_L	p	N_L
St, V, GGG (perl., bai.), GTS (perl.) (limited pitting according to ISO 6336-2)	6,774 8 8,776 3 7,084 1	$6 \times 10^5 < N_L \le 10^7$ $10^7 < N_L \le 10^9$ $10^9 < N_L \le 10^{10}$ (long life)	6,224 9 49,913	$10^4 < N_L \le 3 \times 10^6$ $3 \times 10^6 < N_L \le 10^{10}$ (long life)
St, V, GGG (perl., bai.), GTS (perl.) (no pitting according to ISO 6336-2)	6,611 2 16,301	$10^5 < N_L \le 50 \times 10^6$ $50 \times 10^6 < N_L \le 10^{10}$ (long life)		
EH, IF (limited pitting according to ISO 6336-2)	6,774 8 8,776 3 7,084 1	$6 \times 10^5 < N_L \le 10^7$ $10^7 < N_L \le 10^9$ $10^9 < N_L \le 10^{10}$ (long life)	8,737 8 49,913	$10^3 < N_L \le 3 \times 10^6$ $3 \times 10^6 < N_L \le 10^{10}$ (long life)
EH, IF (no pitting according to ISO 6336-2)	6,611 2 16,301	$10^5 < N_L \le 50 \times 10^6$ $50 \times 10^6 < N_L \le 10^{10}$ (long life)		
GG, GGG (ferr.), NT (nitr.), NV (nitr.)	5,709 1 26,204	$10^5 < N_L \le 2 \times 10^6$ $2 \times 10^6 < N_L \le 10^{10}$ (long life)	17,035 49,913	$10^3 < N_L \le 3 \times 10^6$ $3 \times 10^6 < N_L \le 10^{10}$ (long life)

[a] Values p for pitting are given for torque; to convert for stress, these values are to be doubled.

Table B.1 *(continued)*

Material (see ISO 6336-5)	Pitting		Tooth root bending	
	p[a]	N_L	p	N_L
NV (nitrocar.)	15,716	$10^5 < N_L \le 2 \times 10^6$	84,003	$10^3 < N_L \le 3 \times 10^6$
	26,204	$2 \times 10^6 < N_L \le 10^{10}$ (long life)	49,913	$3 \times 10^6 < N_L \le 10^{10}$ (long life)

[a] Values p for pitting are given for torque; to convert for stress, these values are to be doubled.

For pitting, the endurance limit is considered:

— beyond 2×10^6 cycles for GG, GGG (ferr.), NT (nitr.), NV (nitr.), NV (nitrocar.);

— beyond 50×10^6 cycles for St, V, GGG (perl., bai.), GTS (perl.), Eh, IF when limited pitting according to ISO 6336-2 is permitted;

— beyond 50×10^6 cycles for St, V, GGG (perl., bai.), GTS (perl.), Eh, IF.

For tooth root bending, the endurance limit is considered beyond 3×10^6 cycles for all materials of Table A.1.

When the long life area is considered, the endurance limit is beyond 10^{10} cycles.

B.2.2 Basis

The first step is to select for which criteria the equivalent stress damage calculation shall be done:

— contact stress at pinion;

— contact stress at wheel;

— bending stress at pinion;

— bending stress at wheel.

The second step is to convert each torque bin (T_i, n_i) in a contact stress bin (σ_{Hi}, n_i) and/or bending stress bin (σ_{Fi}, n_i) on the basis of ISO 6336-2 and ISO 6336-3 (see Formulae (4) and (5) in 5.2).

NOTE The transformation from the torque T_i to the stress σ_i is done by using load distribution factors (transverse and facewidth-direction, $K_{H\alpha i}$ and $K_{H\beta i}$) taking into account the mean rotational speed during the bin number i via the dynamic factor K_{vi};

— For pitting:

  — the calculation of the nominal contact stress $\sigma_{H0,i}$ for the torque T_i and the calculation of the contact stress for the pinion $\sigma_{H1,i}$ and/or the wheel $\sigma_{H2,i}$, according to ISO 6336-2:2019, Formulae (3) to (5);

  — the calculation of the number of cycles for the pinion $n_{H1,i}$ and/or the wheel $n_{H2,i}$ out of the given time and roational speed for which the torque T_i is applied on the concerned tooth flanks;

  — the calculation of the permissible contact stress considering a life factor Z_{NT} equal to 1 for the number of cycles for the pinion σ_{HP1} and/or the wheel σ_{HP2}, according to ISO 6336-2:2019, Formula (6); this calculation gives the reference permissible contact stress level $\sigma_{H\ REF}$ associated to the allowable reference number of cycles $N_{LH\ REF}$ (obtained for the knee of the life factor curve Z_{NT}: beginning of the endurance limit when considered);

  - then, the initial load spectrum (T_i, n_i) is transformed in ($\sigma_{H1,i}$, $n_{H1,i}$) for the pinion or/and ($\sigma_{H2,i}$, $n_{H2,i}$) for the wheel.

- For tooth root bending:
  - the calculation of the nominal bending stress $\sigma_{F0,i}$ for the torque T_i and then the calculation of the bending stress for the pinion $\sigma_{F1,i}$ and/or the wheel $\sigma_{F2,i}$, according to ISO 6336-3:2019, Formula (3);
  - the calculation of the number of cycles for the pinion $n_{F1,i}$ and/or the wheel $n_{F2,i}$ out of the given time and roational speed for which the torque T_i is applied on the concerned tooth flanks;
  - the calculation of the permissible bending stress considering a life factor Y_{NT} equal to 1 for the number of cycles for the pinion σ_{FP1} and/or the wheel σ_{FP2}, according to ISO 6336-3:2019, Formula (5); this calculation gives the reference permissible bending stress level $\sigma_{F\ REF}$ associated to the allowable reference number of cycles $N_{LF\ REF}$ = 3,0 × 10^6 cycles (same value for all materials) (obtained for the knee of the life factor curve Y_{NT}: beginning of the endurance limit when considered);
  - then, the initial load spectrum (T_i, n_i) is transformed in ($\sigma_{F1,i}$, $n_{F1,i}$) for the pinion or/and ($\sigma_{F2,i}$, $n_{F2,i}$) for the wheel.

To be general, the transformed load spectrum is noted as ($\sigma_{nom,i}$, $n_{nom,i}$) and the reference permissible stress level is noted as σ_{REF} in the rest of the document.

Further on, a stress level $\sigma_{nom,i}$ in the bin i can be replaced by the stress level $\sigma_{nom,j}$ in a new bin j, so that the damage caused by the stress level $\sigma_{nom,j}$ is the same as that caused by the stress level $\sigma_{nom,i}$. This is shown in Figure B.1 and can be expressed by Formula (B.1).

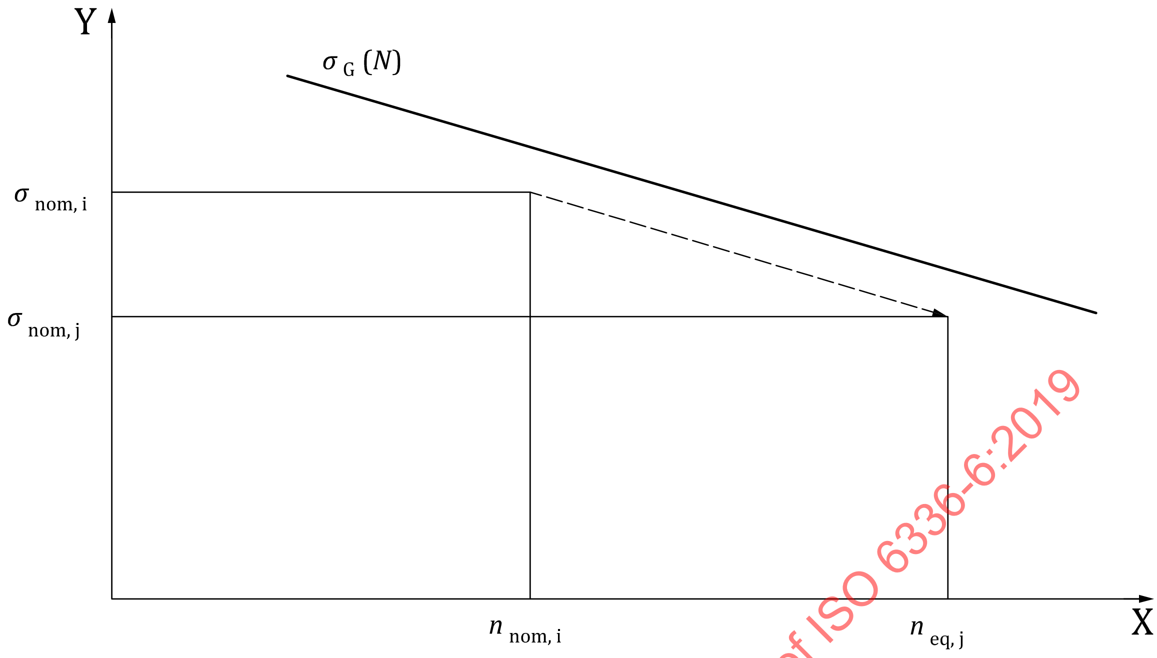
$$\sigma^p_{nom,i} \cdot n_{nom,i} = \sigma^p_{nom,j} \cdot n_{nom,j} \tag{B.1}$$

B.2.3 Calculation procedure

The load bins shall be denoted as ($\sigma_{nom,i}$, $n_{nom,i}$) and numbered in descending order of stress, where $\sigma_{nom,1}$ corresponds to the highest torque. Then the number of cycles $n_{nom,1}$ at the torque T_1 is equivalent in terms of damage to a larger number of cycles $n_{eq,1}$, at the lower stress $\sigma_{nom,2}$, according to Formula (B.2):

$$n_{eq,1} = n_{nom,1} \cdot \left(\frac{\sigma_{nom,1}}{\sigma_{nom,2}} \right)^p \tag{B.2}$$

Then bins 1 and 2 can be replaced by a single bin ($\sigma_{nom,2}$, $n_{eq,1} + n_{nom,2}$), see Figure B.2.

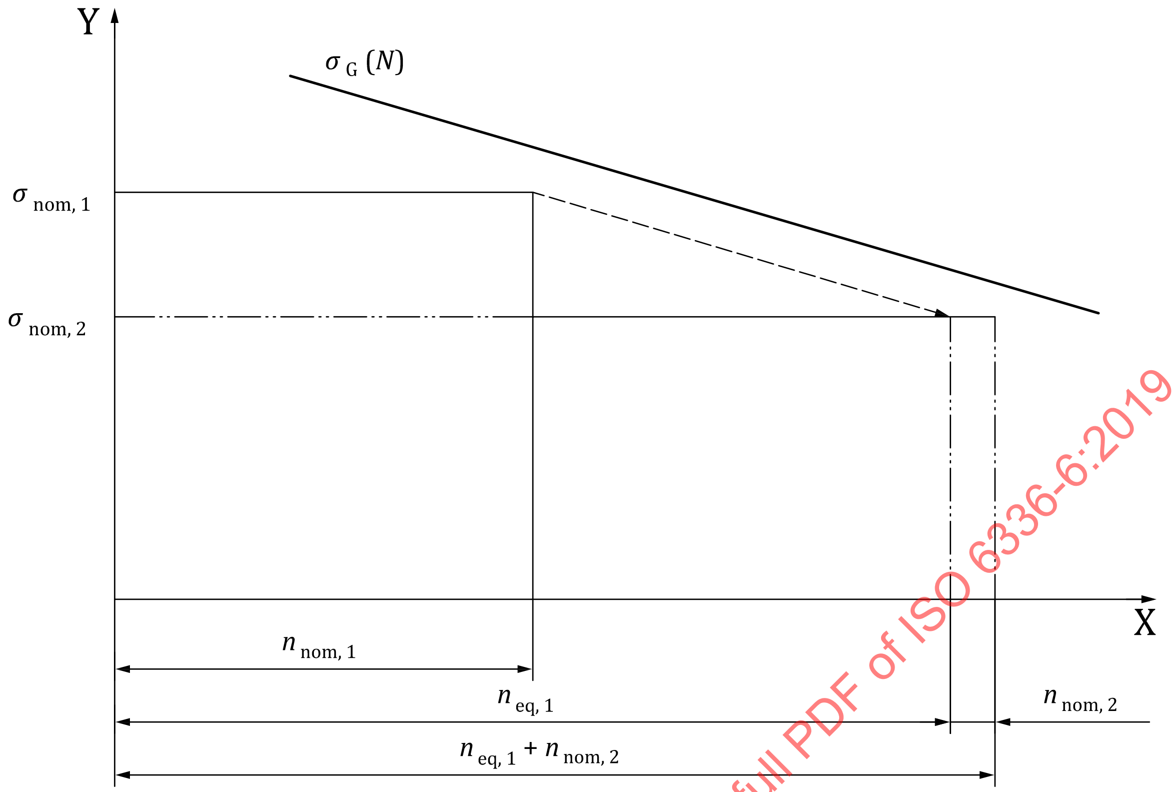


Key

X number of load cycles, N (log)

Y stress, σ (log)

Figure B.1 — Load bins with equal damage behaviour according to Formula (B.1)



Key

X number of load cycles, N (log)

Y stress, σ (log)

Figure B.2 — Bins ($\sigma_{nom,1}$, $n_{nom,1}$) and ($\sigma_{nom,2}$, $n_{nom,2}$) replaced by ($\sigma_{nom,2}$, $n_{eq,1}$ + $n_{nom,2}$)

This procedure can be done for all bins of the spectrum.

B.3 Equivalent damage curve σ_D(N)

No endurance limit is always considered.

The initial complete load spectrum of Figure 2 is transformed in the cumulated stress spectrum shown in Figure B.3. In terms of damage, it is equivalent to the stress level $\sigma_{nom,5}$ with a number of cycles $n_{D,5} = n_{eq,5}$. From this point ($\sigma_{nom,5}$, $n_{D,5}$) it is possible to draw an equivalent damage fatigue curve σ_D(N) parallel to the permissible fatigue curve σ_P(N).

The initial complete spectrum of Figure 2 has also the following equivalent cumulative damage for each level of stresses:

— at stress level $\sigma_{nom,5}$ with a number of cycles $n_{D,5}$;

— at stress level $\sigma_{nom,4}$ with a number of cycles $n_{D,4}$;

— at stress level σ_{REF} with a number of cycles $n_{D,REF}$;

— at stress level $\sigma_{nom,3}$ with a number of cycles $n_{D,3}$;

— at stress level $\sigma_{nom,2}$ with a number of cycles $n_{D,2}$;

— at stress level $\sigma_{nom,1}$ with a number of cycles $n_{D,1}$.

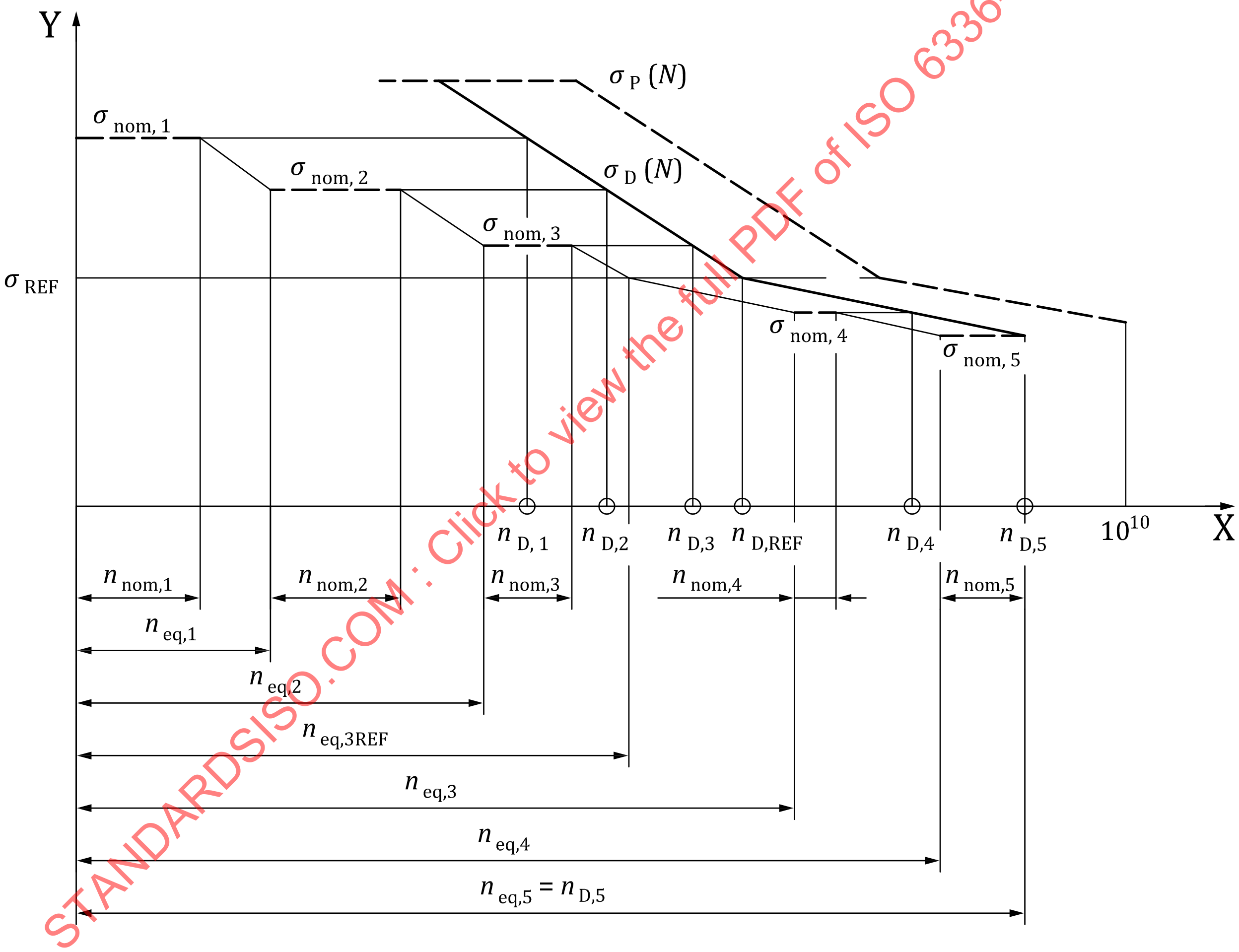
Those different stress levels $\sigma_{nom,i}$ associated with their respective number of cycles $n_{D,i}$ are all equivalents in terms of damage to the damage obtained for the application of the initial complete load spectrum.

Each equivalent number of cycles $n_{eq,i}$ for each level of stress $\sigma_{nom,i}$ is located on the same parallel to the S-N curve of the material represented in ISO 6336 series by the life factors.

At the stress level σ_{REF} the slope of the S-N curve (the life factor curve) might change depending on the material and the chosen parameters. The transformation in equivalent number of cycles ($n_{eq,3REF}$ on the Figure B.3) shall consider this change.

NOTE 1 After $n_{eq,3REF}$ cycles the slope of the curve changes from the p defined for “limited life” to values p_{LF} from the corresponding life factor given for “long life” for the shown example (see Table B.1).

NOTE 2 In order to be conservative, stress levels lower than the minimum level of the S-N curve at 10^{10} cycles, such as $\sigma_{nom,5}$ in Figure B.3, are integrated as it can contribute to the cumulative damage.



Key

X number of cycles, N (log)

Y stress, σ (log)

$\sigma_D(N)$ stress value used to describe the equivalent damange fatigue curve

$\sigma_P(N)$ stress value used to describe the S-N curve dependant from the number of load cycles

Figure B.3 — Generalization of equivalent damages when there is no endurance limit

The following are some remarks on the calculation:

— Determination of $n_{eq,3REF}$ on the Figure B.3:

In this calculation the reference permissible stress level σ_{REF} is already determined taking into account the safety factor $S_{H\,min}$ for pitting, $S_{F\,min}$ for bending ($\sigma_{REF} = \sigma_{HG}/S_{H\,min}$, $\sigma_{REF} = \sigma_{FG}/S_{F\,min}$).

On this basis, the calculation of the equivalent number of cycles $n_{eq,3REF}$ is given by:

$$n_{eq,3REF} = \left(n_{eq,2} + n_{nom,3}\right)\left(\frac{\sigma_{nom,3}}{\sigma_{REF}}\right)^{p} \tag{B.3}$$

— Then the slope of the curve changes from p (defined for "limited life") to p_{LF} from the corresponding life factor given for "long life" and

$$n_{eq,3} = n_{eq,REF}\left(\frac{\sigma_{REF}}{\sigma_{nom,4}}\right)^{p_{LF}} \tag{B.4}$$

— Formula of equivalent damage fatigue curve $\sigma_D(N)$ for « long life area »

$$\sigma_D(N) = \sigma_{REF} \cdot \left(\frac{n_{eq,3REF}}{N}\right)^{1/p_{LF}} \quad \text{for } N < n_{eq,3REF} \tag{B.5}$$

— Formula of equivalent damage fatigue curve $\sigma_D(N)$ for « limited life area »

$$\sigma_D(N) = \sigma_{REF}\left(\frac{n_{eq,3REF}}{N}\right)^{1/p} \quad \text{for } N \geq n_{eq,3REF} \tag{B.6}$$

B.4 Use of cumulative damage concept for accelerated tests

The method previously described in B.2 and B.3 for equivalent damage can be used in order to accelerate the testing duration to qualify products in terms of endurance fatigue.

When accelerating tests for a defined type of failure it is recommended to check that the increase of loadings will not lead to other potential damages on the tested component itself and all others implied in the test.

NOTE 2 cases can be considered:

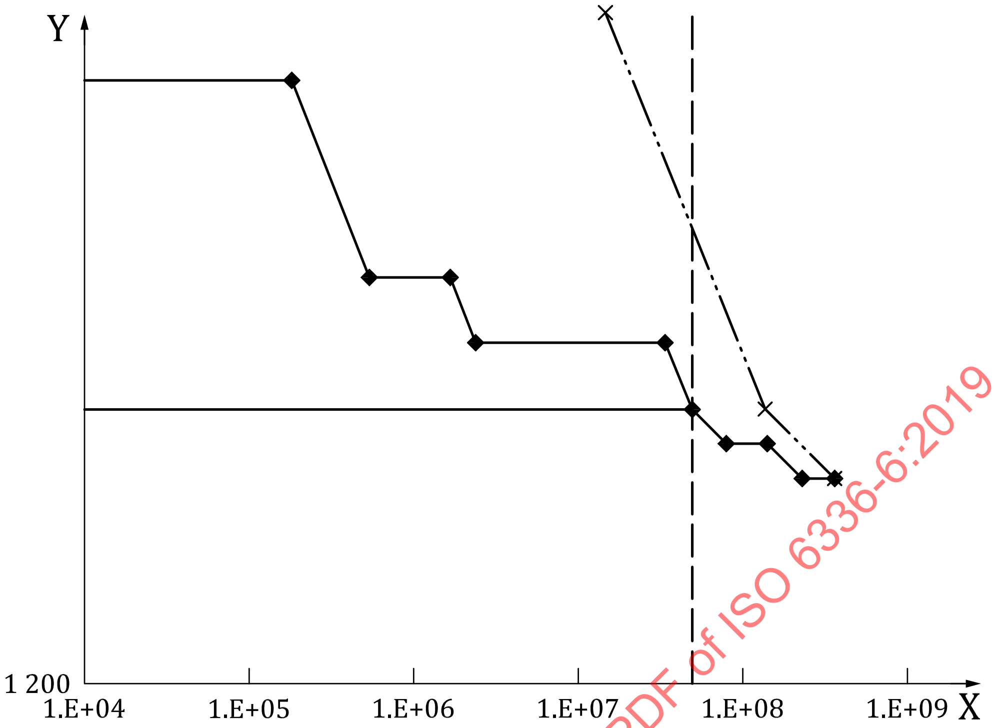
— either it can be proved that there is no damage for the considered loading spectrum and in that case the safety factors considered to calculate the reference permissible stress level are greater or equal to 1;

— or it can be proved that there is damage for the considered loading spectrum and in that case the safety factors considered to calculate the reference permissible stress level are equal to 1.

B.5 Example of equivalent damage fatigue curve $\sigma_D(N)$

An example is shown in Figure B.4 and the corresponding Table B.2. The right hand column of the table indicates the life factor. In this example the number of load cycles for endurance limit are 50×10^6 and therefore between Z_N = 0,933 and Z_N = 1,03.

Here the examined criterion is the contact pressure, and the cumulated loading spectrum is transformed in a contact stress spectrum which is cumulated as indicated in Figure B.4. Then from the point of the cumulated spectrum with the highest number of cumulated cycles the slope of life factor allows to determine the equivalent damage curve: each point of this curve is equivalent in terms of the contact fatigue damage to the full spectrum.



Key

X number of load cycles, *N* (log)

Y torque, *T* (log)

—◆— loading spectrum

——— reference permissible stress level

— — — $50 \cdot 10^6$ limit

—×— equivalent damage curve

Figure B.4 — Load spectrum with a corresponding equivalent damage curve

Table B.2 — Example for the calculation of the equivalent damage curve

Equivalent damage curve							
Flank			Reference permissible contact stress level, $\sigma_{nom,i}$, in MPa:				1 360
			Number of blade torque:				20
Ratio to this gear *u*:		3	Slope exponent, *p*, from Table A.1:				13,222 4
Contacts per revolution:		1	Slope exponent, *p*, from Table A.1: (> 50,0 × 10^6 cycles)				32,602
Speed in cycles per minute		1 500	Number of load cycles for endurance limit, $N_{L\ ref}$:				50,0 × 10^6
Bin *i*	**Blade torque** *T*	**Nominal contact stress** $\sigma_{nom,i}$	**Operating hours** *h*	**Cycles** n_i	**Equivalent from row above** n_{ic}	**Total** $n_{eq,i}$	Z_N **Factor**
1	1 400	1 555	2	180 000	—	180 000	1,530
2	1 275	1 431	12	1 080 000	544 204	1 624 204	1,296
3	1 125	1 390	350	31 500 000	2 375 556	33 875 556	1,030
3a	—	1 350	—	—	50 000 000	50 000 000	1,000
4	1 060	1 330	500	45 000 000	80 690 792	125 690 792	0,933
5	1 040	1 310	800	72 000 000	205 985 135	277 985 135	0,878
NOTE The calculation in Table B.2 is done for pitting (no pitting permissible).							

