INTERNATIONAL STANDARD

ISO 6336-6

First edition
2006-08-15

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
ISO 6336-6:2006(E)

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Published in Switzerland

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

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.

ISO 6336-6 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

ISO 6336 consists of the following parts, under the general title *Calculation of load capacity of spur and helical gears*:

- *Part 1: Basic principles, introduction and general influence factors*
- *Part 2: Calculation of surface durability (pitting)*
- *Part 3: Calculation of tooth bending strength*
- *Part 5: Strength and quality of materials*
- *Part 6: Calculation of service life under variable load*

Calculation of load capacity of spur and helical gears —

Part 6: Calculation of service life under variable load

1 Scope

This part of ISO 6336 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. While the method is presented in the context of ISO 6336 and calculation of the load capacity of spur and helical gears, it is equally applicable to other types of gear stress.

2 Normative references

The following referenced documents are indispensable for the application 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, *Glossary of gear terms — Part 1: Geometrical definitions*

ISO 6336-1:2006, *Calculation of load capacity of spur and helical gears — Part 1: Basic principles, introduction and general influence factors*

ISO 6336-2:2006, *Calculation of load capacity of spur and helical gears — Part 2: Calculation of surface durability (pitting)*

ISO 6336-3:2006, *Calculation of load capacity of spur and helical gears — Part 3: Calculation of tooth bending strength*

3 Terms, definitions, symbols and abbreviated terms

For the purposes of this part of ISO 6336, the terms, definitions, symbols and abbreviated terms given in ISO 6336-1 and ISO 1122-1 apply.

4 General

4.1 Application factors

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 calculation of the service strength.

See Annex B for tables for K_A.

4.2 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 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 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 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 are limited to fewer calculated stress cycles and the resulting gears can be smaller. 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 1 shows as an example of how the torque classes defined in Table 2 can be applied to specific torque levels and correlated numbers of cycles.

Table 1 — Torque classes/numbers of cycles — Example: classes 38 and 39 (see Table 2)

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

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 lifetime, the possibility that there might be torque peaks not frequent enough to be evaluated in that measured spectrum must 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).

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 gauges 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.

Table 2 — Example of torque spectrum (with unequal bin size for reducing number of bins)
(see Annex C)

Data	**Pinion**			**%**	**Time [a]**	
	Torque N·m		**Load cycles [a]**			
Bin no.	min.	max.			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
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

[a] −10 raises and lowers; pinion at 35,2 r/min assumes 1 raise and lower per week.

4.3 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,

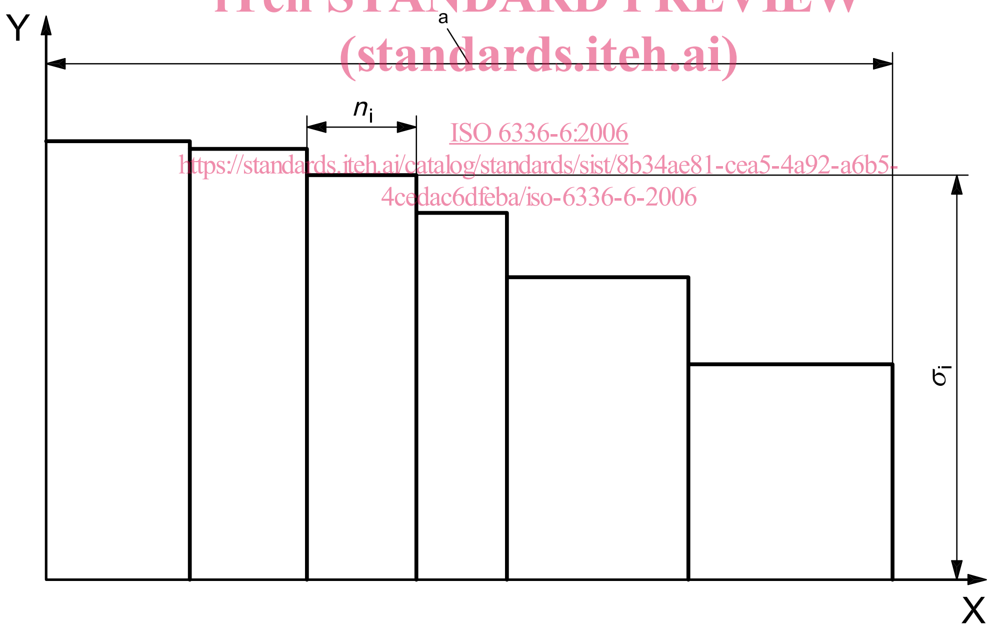
b) material fatigue properties, and

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 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 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

Y stress

a Load spectrum, $\sum n_i$, 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.

4.4 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

$$\sum_i \frac{n_\mathrm{i}}{N_\mathrm{i}} = 1{,}0 \tag{1}$$

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 (lower line beyond the knee in Figure 2), the calculation must be done for all stress levels. For each stress level, i, the number of cycles to failure, N_i, have to be taken from the corresponding part of the S-N curve.

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

5.1 Basic principles

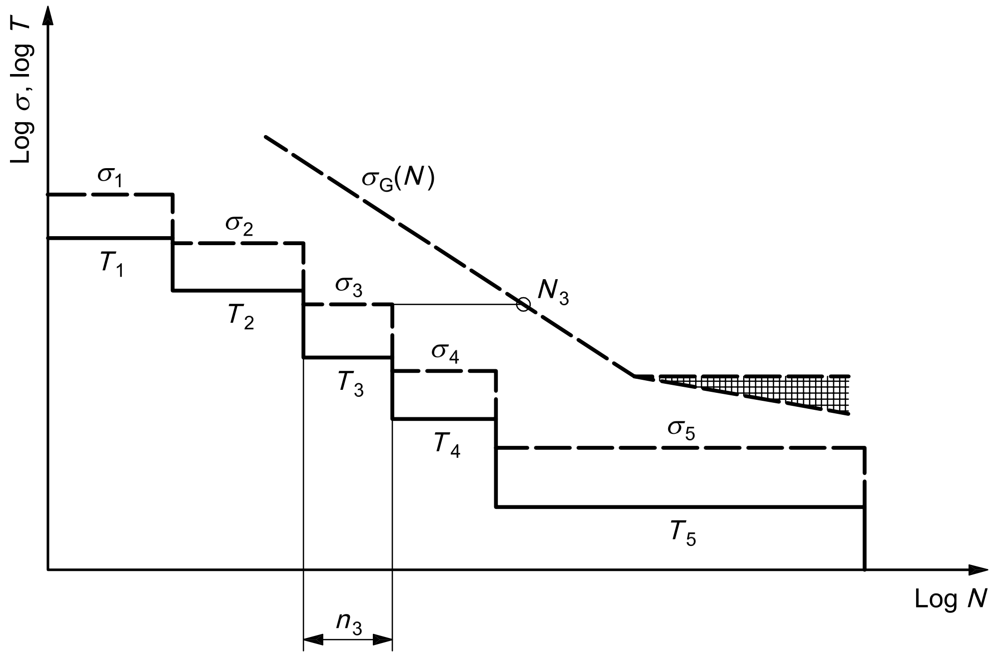
This method is only valid for recalculation. It describes the application of linear cumulative damage calculations according to the Palmgren-Miner rule (see 4.4) and has been chosen because it is widely known and easy to apply; the choice does not imply that the method is superior to others described in the literature.

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 an example).

Table 3 — 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.



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 3.

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

Figure 2 — Torque spectrum and associated stress spectrum with S-N

The stress spectra for tooth root and tooth flank (σ_{Fi}, σ_{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 stress spectra obtained in this way, the calculated values are compared with the strength values (S-N curves, damage line) 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:

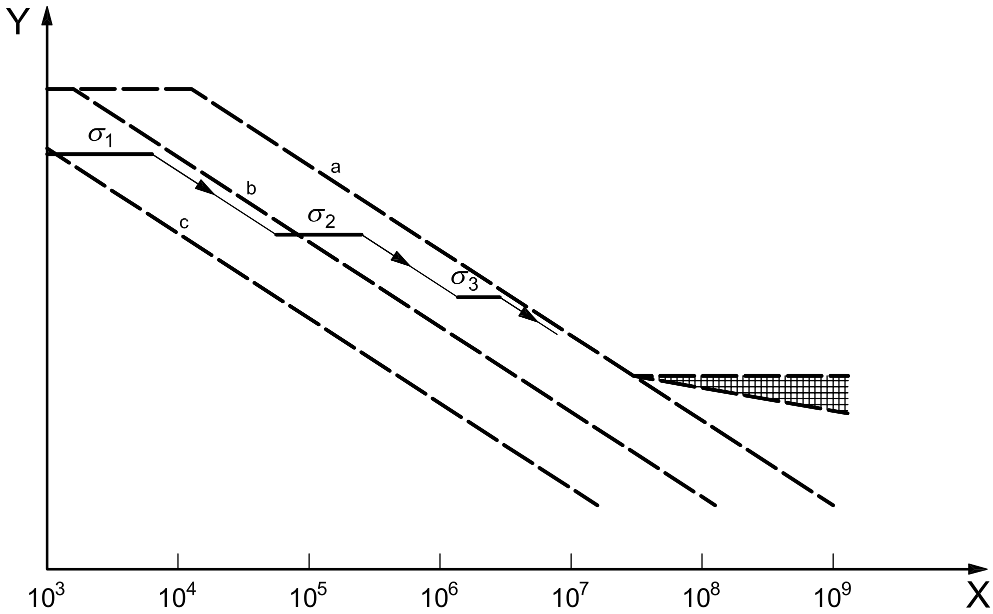
$$U_i = \frac{n_i}{N_i} \tag{2}$$

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

$$U = \sum_i U_i = \sum_i \frac{n_i}{N_i} \leqslant 1{,}0 \tag{3}$$

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.



Key

X number of load cycles, N_L

Y stress

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

a 100 % damage.

b 10 % de damage.

c 1 % de damage.

Figure 3 — Accumulation of damage

In addition, safety factors applied to static load strength should be calculated for the highest stress of the design life. ISO 6336 does not extend to stress levels greater than those permissible at 10^3 cycles or less, since stresses in this range can exceed the elastic limit of the gear tooth in bending or in surface compression. 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 greater than the limit level at $< 10^3$ cycles could result in plastic yielding of the gear tooth.

5.2 Calculation of stress spectra

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

— For contact stress (ISO 6336-2:2006, Method B):

$$\sigma_{Hi} = Z_H\, Z_E\, Z_\varepsilon\, Z_\beta\, Z_{BD} \sqrt{\frac{2\,000\, T_i}{d_1^2\, b}\,\frac{u+1}{u}\, K_{vi}\, K_{H\beta i}\, K_{H\alpha i}} \tag{4}$$

— For bending stress (ISO 6336-3:2006, Method B):

$$\sigma_{Fi} = \frac{2\,000\, T_i}{d_1\, b\, m_n}\, Y_F\, Y_S\, Y_\beta\, K_{vi}\, K_{F\beta i}\, K_{F\alpha i} \tag{5}$$

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.

5.3 Determination of pitting and bending strength values

S-N curves for 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 tooth root strength must be reduced according to ISO 6336-3.

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

5.4 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 4.

The safety factor, S, has to 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.

