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Enclosed gear drives for industrial applications

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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 a 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 an International Standard. Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.

ISO/TR 13593, which is a Technical Report of type 2, was prepared by Technical Committee ISO/TC 60, *Gears*.

This document is being issued in the Technical Report (type 2) series of publications (according to subclause G.3.2.2 of Part 1 of the ISO/IEC Directives, 1995) as a “prospective standard for provisional application” in the field of gearing because there is an urgent need for guidance on how standards in this field should be used to meet an identified need.

This Technical Report is not to be regarded as an “International Standard”. It is proposed for provisional application so that information and experience of its use in practice may be gathered. Comments on the content of this document should be sent to the ISO Central Secretariat.

A review of this Technical Report (type 2) will be carried out not later than three years after its publication with the options of: extension for another three years; conversion into an International Standard; or withdrawal.

Annexes A to F of this Technical Report are for information only.

Enclosed gear drives for industrial applications

1 Scope

This Technical Report is applicable to enclosed speed reducers and increasers for industrial applications, where the designs include spur, helical, herringbone or double helical gears and their combination in single or multistage drives.

This Technical Report provides a method by which gear drive designs can be compared and selected. It is not intended to assure performance of assembled gear drive systems. It is intended for use by experienced gear designers capable of selecting reasonable values for the factors, based on performance knowledge of similar designs and the effects of such items as lubrication, deflection, manufacturing tolerances, metallurgy, residual stress and system dynamics. It is not intended for use by the engineering public at large.

Maintaining an acceptable temperature in the oil sump of an enclosed gear drive is critical to the life of the gear drive. Therefore, this Technical Report for enclosed gear drives considers not only the mechanical rating but also the thermal rating.

The rating methods and influences identified in this Technical Report are limited to enclosed drives of single and multiple stage designs where the pitch line velocities do not exceed 35 m/s and pinion speeds do not exceed 4 500 r/min. In this Technical Report, gear teeth rating is covered only as limited by tooth root bending and contact pressure.

This Technical Report does not cover the design and application of epicyclic drives. It is beyond the scope of this Technical Report to present a detailed analysis of efficiency.

Annexes A to F can be used to make a more detailed analysis of certain rating factors.

2 Normative references

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

ISO 76:1987, *Rolling bearings — Static load ratings*.

ISO 281:1990, *Rolling bearings — Dynamic load ratings and rating life*.

ISO 701, *International gear notation — Symbols for geometrical data*.

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

ISO 3448:1992, *Industrial liquid lubricants — ISO viscosity classification*.

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 materials*.

ISO 6743-6:1990, *Lubricants, industrial oils and related products (class L) — Classification — Part 6: Family C (Gears).*

ISO 8579-1, *Acceptance code for gears — Part 1: Determination of airborne sound power levels emitted by gear units*.

ISO 8579-2, *Acceptance code for gears — Part 2: Determination of mechanical vibrations of gear units during acceptance testing*.

ISO 8821:1989, *Mechanical vibration — Balancing — Shaft and fitment key convention*.

ISO 9085:—[1], *Calculation of load capacity of spur and helical gears — Application for industrial gears*.

ISO 10825, *Gears — Wear and damage to gear teeth — Terminology*.

ISO 12925-1:1996, *Lubricants, industrial oils and related products (class L) — Family C (gears) — Part 1: Specifications for lubricants for enclosed gear systems*.

3 Symbols, terms and definitions

NOTE The symbols, terms, and definitions contained in this document may vary from those used in other ISO standards. Users of this Technical Report should verify that they are using these symbols and terms in the manner indicated herein.

3.1 Symbols

For the purposes of this Technical Report, the symbols given in Table 1 apply.

[1] To be published.

Table 1 — Symbols used in equations

Symbol	Meaning	Units	Where first used	Subclause
A_C	surface area of gear drive	m^2	Eq 40	7.4.3
A_R	fit holding capacity	N	Eq 21	5.6.3
A_s	stress cross section of fastener	mm^2	Eq 27	5.7.2
a_1	life adjustment factor for reliability	—	Eq 3	5.4.3.3
B_A	altitude factor	—	Eq 41	7.5
B_D	operation time factor	—	Eq 41	7.5
B_{ref}	ambient temperature factor	—	Eq 41	7.5
B_T	non-standard oil sump temperature factor	—	Eq 41	7.5
B_V	ambient air velocity factor	—	Eq 41	7.5
b_k	width of key	mm	Eq 17	5.6.2
D_f	nominal diameter of threaded fastener	mm	Eq 28	5.7.2
d_{he}	outside diameter of hub	mm	Eq 24	5.6.3
d_{hi}	inside diameter of hub	mm	Eq 24	5.6.3
d_{max}	maximum nominal fastener diameter	mm	Table 3	5.7.2
d_{sh}	shaft diameter	mm	Eq 16	5.6.2
d_{she}	shaft outside diameter	mm	Eq 6	5.5.2
d_{shi}	shaft inside diameter	mm	Eq 6	5.5.2
E_H	modulus of elasticity for hub material	N/mm^2	Eq 23	5.6.3
E_S	modulus of elasticity for shaft material	N/mm^2	Eq 23	5.6.3
F_A	applied tensile load	N	Eq 31	5.7.4
F_M	fastener tensile preload	N	Eq 27	5.7.2
f_L	load peak frequency factor		Eq 20	5.6.3
h_k	height of key	mm	Eq 16	5.6.2
I	actual or minimum possible interference fit	mm	Eq 23	5.6.3
i	number of keys	—	Eq 16	5.6.2
K_A	application factor	—	9.5.1	9.5.1
K_J	joint stiffness factor	—	Eq 30	5.7.3
K_{sf}	selection factor	—	Eq 1	4.5.3
K_{tc}	torque coefficient	—	Eq 29	5.7.2
k	heat transfer coefficient	$kW/(m^2 \cdot K)$	Eq 40	7.4.3
L	length of hub	mm	Eq 22	5.6.3
L_{na}	adjusted rating life at $100 - n = R$ % reliability	h	Eq 3	5.4.3.3
L_{10a}	rating life at basic 90 % reliability	h	Eq 3	5.4.3.3
l_g	length of fastener grip	mm	5.7.3	5.7.3
l_{tr}	bearing length of the key	mm	Eq 16	5.6.2
M	bending moment	Nm	Eq 7	5.5.2
M_A	fastener tightening torque	Nm	Eq 29	5.7.2
P_A	input power to gear drive	kW	Eq 34	7.4.1
P_B	bearing power loss	kW	Eq 38	7.4.2
P_H	pressure at common shaft/hub interface	N/mm^2	Eq 22	5.6.3
P_L	load-dependent power losses	kW	Eq 33	7.4.1
P_M	gear mesh power loss	kW	Eq 38	7.4.2
P_{mc}	minimum rated component power	kW	Eq 1	4.5.1
P_N	non-load-dependent power losses	kW	Eq 33	7.4.1
P_n	nominal power of the driven machine or the driving machine	kW	Eq 1	4.5.3

Table 1 — Symbols used in equations

Symbol	Meaning	Units	Where first used	Subclause
P_P	oil pump power consumption	kW	Eq 39	7.4.2
P_Q	heat dissipation of gear drive	kW	Eq 32	7.4.1
P_S	oil seal power loss	kW	Eq 39	7.4.2
P_T	thermal power rating	kW	Eq 37	7.4.1
P_{Thm}	modified application thermal power rating	kW	Eq 41	7.5
P_V	total power loss	kW	Eq 32	7.4.1
P_{WB}	bearing windage and oil churning power loss	kW	Eq 39	7.4.2
P_{WG}	gear windage and oil churning power loss	kW	Eq 39	7.4.2
p_f	fastener thread pitch	mm	Eq 28	5.7.2
R	reliability level	percent	Eq 4	5.4.3.3
R_e	tensile strength of the key material	N/mm^2	Eq 18	5.6.2
$S_{F\,min}$	minimum safety factor for bending strength	—	9.5.1	9.5.1
$S_{H\,min}$	minimum safety factor for pitting resistance	—	9.5.1	9.5.1
T	shaft torque	Nm	Eq 6	5.5.2
T_a	allowable torque based on the lesser of T_C and T_s	Nm	5.6.2	5.6.2
T_C	allowable torque based on the allowable compressive stress	Nm	Eq 16	5.6.2
T_{max}	maximum torque	Nm	Eq 20	5.6.3
T_{mc}	minimum rated component torque	Nm	Eq 2	4.5.3
T_n	nominal torque of the driven machine or the driving machine	Nm	Eq 2	4.5.3
T_R	torque carried by friction in the interface of shaft and hub	Nm	Eq 21	5.6.3
T_s	allowable torque based on the allowable key shear stress	Nm	Eq 17	5.6.2
t_k	shaft keyway depth	mm	Eq 16	5.6.2
Y_{NT}	life factor for bending strength	—	9.5.1	9.5.1
Z_{NT}	life factor for pitting resistance	—	9.5.1	9.5.1
β_τ	torsional notch factor	—	Eq 10	5.5.3
β_σ	bending notch factor	—	Eq 12	5.5.3
ΔT	temperature differential	K	Eq 40	7.4.3
φ	share of the load	—	Eq 16	5.6.2
η	overall drive efficiency	percent	Eq 36	7.4.1
μ	coefficient of friction	—	Eq 22	5.6.3
ρ_H	Poisson's ratio for hub material	—	Eq 23	5.6.3
ρ_S	Poisson's ratio for shaft material	—	Eq 23	5.6.3
σ_B	material tensile strength	N/mm^2	Eq 10	5.5.3
σ_b	calculated bending shaft stress	N/mm^2	Eq 7	5.5.2
σ_{ba}	allowable bending stress	N/mm^2	Eq 12	5.5.3
σ_f	calculated tensile stress in fastener	N/mm^2	Eq 31	5.7.4
σ_{fa}	allowable tensile stress of fastener	N/mm^2	Eq 30	5.7.3
σ_M	preload tensile stress, recommended	N/mm^2	Eq 26	5.7.2
$\sigma_{p0,2}$	fastener 0,2 % offset yield strength	N/mm^2	Eq 26	5.7.2
σ_s	calculated torsional shaft stress	N/mm^2	Eq 6	5.5.2
σ_{sa}	allowable torsional stress	N/mm^2	Eq 10	5.5.3
σ_{SC}	allowable compressive stress	N/mm^2	Eq 16	5.6.2
τ_{ps}	allowable shear stress	N/mm^2	Eq 17	5.6.2

3.2 Terms and definitions

For the purposes of this Technical Report, the following terms and definitions apply.

3.2.1
gear unit rating
overall mechanical power rating of all static and rotating elements within the enclosed drive, as determined by the minimum rated component power, P_{mc} (weakest part, whether determined by gear teeth, shafts, bolting, housing, etc.)

3.2.2
thermal rating
maximum power that can be continuously transmitted through an enclosed gear drive without exceeding a specified oil sump temperature

NOTE The thermal rating equals or exceeds the actual service transmitted power. Selection factors are not used when determining thermal requirements, see 7.1.

4 Application and design considerations

4.1 Application limitations

In this Technical Report, the gear unit rating, as defined, is the mechanical capacity (selection factor, $K_{sf} = 1{,}0$) of the gear drive components. In some applications it may be necessary to select a gear drive with an increased mechanical rating in order to accommodate adverse effects of environmental conditions, thermal capacity of the drive, external loading or any combination of these factors.

4.2 Rating factors

The allowable stress numbers in this Technical Report are maximum allowed values. Some latitude based upon experience is permissible in the selection of specific factors within this Technical Report. Less conservative values for other rating factors in this Technical Report shall not be used.

4.3 Metallurgy

The factors for gears affected by material conditions and quality are defined in ISO 6336-5.

4.4 System analysis

The system of connected rotating parts shall be compatible, free from critical speeds, torsional or other types of vibration within the specified operating speed range, no matter how induced. The enclosed gear drive designer or manufacturer is not responsible for this analysis, unless agreed to in the purchase contract.

4.5 Gear unit rating

4.5.1 Unit rating application

The gear unit rating is the overall mechanical power rating of all static and rotating elements within the enclosed drive. The minimum rated component power, P_{mc} (weakest part, whether determined by gear teeth, shafts, bolting, housing, etc.), of the enclosed drive determines the gear unit rating. The load histogram for determining the gear unit rating shall consist of 10 000 cycles at 200 % load plus 10 000 h at 100 % load. The gear unit rating shall also include the effects of the allowable overhung load at a specified distance from the end of the gearbox where the overhung load is applied.

NOTE It is the responsibility of the user to specify peak load conditions so that the drive can be selected such that the peak torque does not exceed that specified in 4.6.

Unity selection factor ($K_{sf} = 1{,}0$) is used in determining the gear unit rating. Refer to clause 9 for a discussion of the selection factor, K_{sf}.

4.5.2 Gear unit rating requirements

The gear unit rating implies that all items within the unit have been designed to meet or exceed the unit rating. Gear and pinion ratings shall be in accordance with the bending strength and pitting resistance ratings as specified in 5.2.

4.5.3 Application of gear unit rating

The required gear unit rating of an enclosed drive is a function of the application and assessment of variable factors that affect the overall rating. These factors include environmental conditions, severity of service and life. Refer to clause 9 for further explanation.

The application of the enclosed drive requires that its unit rating meet the requirements of the actual service conditions. This is accomplished by the proper selection of a selection factor, K_{sf}, based on field data or experience.

The values shown in annex A may be used as a guide. The gear unit rating required for the considered application is then obtained by satisfying:

$$P_{\mathrm{mc}} \geqslant P_{\mathrm{n}}\, K_{\mathrm{sf}} \tag{1}$$

where

P_{n} is the nominal power of the driven machine or the driving machine. See clause 9 and annex A.

Similarly, when rating by torques:

$$T_{\mathrm{mc}} \geqslant T_{\mathrm{n}}\, K_{\mathrm{sf}} \tag{2}$$

If the nominal power or the nominal torque of the driven machine is used for the gear unit rating and $P_{\mathrm{n\ driver}}$ is greater than $P_{\mathrm{n\ driven\ machine}}$, the maximum torque appearing in the whole system should be checked. During acceleration (or at other times) the maximum torque should not exceed 200 % of the nominal torque of the driven machine, see 4.6.

4.6 Momentary overloads

When the enclosed drive is subjected to momentary overloads, direct on-line motor starts, braking, stall conditions and low-cycle fatigue, the conditions should be evaluated to assure that the strength limitations of any component are not exceeded.

With respect to the gear bending strength for momentary overloads, the maximum allowable stress is determined by the allowable fatigue limitations of the material. Shaft, bearing and housing deflections have a significant effect on gear mesh alignment during momentary overloads. The enclosed drive shall be evaluated to assure that the reactions to momentary overloads do not result in excessive misalignment causing localized high stress concentrations and/or permanent deformation. In addition, the effects of external loads such as overhung, transverse and thrust loads shall be evaluated.

Gear drives rated to this Technical Report shall be able to accommodate peak loads whose magnitude does not exceed 200 % of P_{mc} applied for a number of stress cycles not exceeding 10 000. The minimum face load factor, $K_{\mathrm{H}\beta}$, determined for 100 % load applies to the analysis at 200 %.

4.7 Efficiency estimate

When an efficiency estimate of the enclosed drive is calculated, it should be determined based on the transmitted power and specified operating conditions. The estimate should include the effects of the components within the enclosed drive and shaft driven accessories agreed to by manufacturer and user. Unless specifically agreed to between the user and manufacturer, the prime mover, couplings, external driven loads, motor driven accessories, etc., are not included in the enclosed drive efficiency estimate. See clause 7 for calculations.

4.8 Reverse loading

The effect of torque reversals on an enclosed drive is taken into account by choosing an adequate selection factor for the considered application, e.g. travel drive. In a detailed rating analysis, the effect of reverse loading may be considered alternatively at component level.

5 Components

5.1 Rating considerations

The components of a gear drive shall be designed with due consideration for all loads likely to be encountered during operation. These include not only the torque loads imposed on the components through the gearing, but also external loads, i.e. overhung loads, external thrust loads, dynamic loads such as from cast overhung pinions, etc. These components shall also be designed to withstand any assembly forces which might exceed the operating loads. During the design process, the operating loads shall be considered to occur in the worst possible direction and in the worst possible loading combinations, including a 200 % momentary peak starting load.

Component rating shall be within the limits specified in this Technical Report. Where user requirements or specifications dictate different design criteria, such as higher bearing life, this shall be by contractual agreement.

Alternative component rating methods based on test data or field experience are allowed. The gear manufacturer shall indicate and document all modifications which are used.

Gear unit ratings may also include allowable overhung load values which are usually designated to act at a distance of one shaft diameter from the face of the housing or enclosure component. Stresses in related parts resulting from these overhung loads shall also be within limits set by this Technical Report.

For the purposes of this Technical Report, where component capacities are being determined, the calculations are specifically related to the gear unit rating as defined in 4.5.1.

NOTE A separate computation is required to relate the gear unit rating to application conditions.

5.2 Housing

The combined assembly of gears, shafts and bearings shall be enclosed by a housing of such design and construction as to provide the rigidity required for proper gear alignment. The housing shall maintain alignment under rated internal and external loading.

For housings with low speed centre distances greater than 460 mm, at least two reference surfaces should be machined parallel to the mounting surfaces for the purpose of levelling the gear drive.

5.3 Gears

5.3.1 Rating criteria

The fundamental formulas for enclosed gear drives shall be in accordance with ISO 9085.

The calculation method for each gear rating factor has the ability to be modified. The gear designer shall indicate all modifications to ISO 9085 that are used.

Pitting resistance is a function of the Hertzian contact (compressive) stresses between two curved surfaces or tooth surfaces and is proportional to the square root of the applied tooth load. Bending strength is measured in terms of the bending (tensile) stress in a cantilever plate and is directly proportional to this same load. The difference in nature of the stresses induced in the tooth surface areas and at the tooth root is reflected in a corresponding difference in allowable limits of contact and bending stress numbers for identical materials and load intensities.

The term "gear failure" is itself subjective and a source of considerable disagreement. One observer's "failure" may be another observer's "wearing-in". For a more complete discussion see ISO 10825.

5.3.1.1 Reverse loading

For gears which are reverse loaded on every cycle, see ISO 6336-5.

5.3.1.2 Localized yielding

This Technical Report 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 compressive stress. 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.

5.4 Bearings

5.4.1 Bearing selection

Shafts may be mounted in bearings, of any size, type and capacity to properly carry the radial and thrust loads which would be induced under the most severe operating conditions.

5.4.2 Fluid film bearings

Fluid film bearings should be designed for bearing pressures not in excess of 6 N/mm^2 on projected area. Journal velocities should not exceed 8 m/s with lubricant supplied un-pressurized. Higher values may be used when the manufacturer has experience or test data.

5.4.3 Roller and ball bearing selection

5.4.3.1 Selection criteria

Roller and ball bearings shall be selected to have a minimum L_{10a} life of 5 000 h based on gear unit rating and gear drive selection factor equal to unity, according to the bearing manufacturers calculations. The L_{10a} life is the operating time that 90 % of apparently identical bearings will equal or exceed before a subsurface originated fatigue spall reaches a predetermined size.

When selecting bearings, the following parameters shall be considered:

— lubrication,

— temperature,

— load zone,

— alignment,

— bearing material.

5.4.3.2 Other considerations

The life calculation methods used by bearing manufacturers are based upon subsurface fatigue damage which leads to spalling. Other types of bearing damage which may occur include, but are not limited to, surface originated spalling due to bruises from lubricant contamination, failure of cages, plastic yielding, brinelling due to extreme momentary overload, and scoring or scuffing due to momentary lack of lubricant film.

5.4.3.3 Reliability

Bearing life at reliability levels other than 90 % is calculated by:

$$L_{na} = a_1 L_{10a} \quad (3)$$

where

L_{na} is the adjusted rating life at $100 - n = R$ percent reliability;

L_{10a} is rating life at basic 90 % reliability, factors a_2 and a_3 included;

a_1 is life adjustment factor for reliability, as in ISO 281.

for reliability $R \geqslant 90$ %,

$$a_1 = 4{,}48 \sqrt[1{,}5]{\ln\left(\frac{100}{R}\right)} \tag{4}$$

for reliability $R < 90$ %,

$$a_1 = 6{,}84 \sqrt[1{,}17]{\ln\left(\frac{100}{R}\right)} \tag{5}$$

Equations 4 and 5 for a_1 are based on the Weibull distribution, fitted to the data of leading bearing manufacturers.

5.5 Shafting

5.5.1 Design criteria

Shafts should be designed to adequately withstand the internal loads (generated by the gear meshes) and the external loads. Both the strength and the stiffness of the shafts are important. Adequate shaft strength will prevent fatigue or plastic deformation, while adequate stiffness will maintain gear and bearing alignment.

5.5.2 Shaft stress calculation

Nominal shaft stresses are calculated as follows. The applicability of equations 6 and 7 to the design of thin wall shafts where the ratio $d_{shi}/d_{she} > 0{,}9$ has not been established.

$$\sigma_s = \frac{16\,000\, T\, d_{she}}{\pi\left(d_{she}^4 - d_{shi}^4\right)} \tag{6}$$

$$\sigma_s = \frac{32\,000\, M\, d_{she}}{\pi\left(d_{she}^4 - d_{shi}^4\right)} \tag{7}$$

where

σ_s is calculated torsional shaft stress, in N/mm^2;

T is shaft torque, in Nm;

d_{she} is shaft outside diameter, in mm;

d_{shi} is shaft inside diameter, in mm;

σ_b is calculated bending shaft stress, in N/mm^2;

M is bending moment, in Nm.

For solid shafting, equations 6 and 7 simplify to:

$$\sigma_s = \frac{16\,000\, T}{\pi\, d_{she}^3} \tag{8}$$

$$\sigma_b = \frac{32\,000\, M}{\pi\, d_{she}^3} \tag{9}$$

5.5.3 Allowable stress

The calculated stresses due to bending and torsion shall not exceed the allowable stress values determined by equations 10 through 15. These equations are a simplified version of DIN 743 and are subject to the following limitations.

— Equations 10 through 15 apply for shaft diameters in the following range:

$25 \leqslant d_{she} \leqslant 150$ mm

For shaft diameters outside of this range the following conditions apply:

If $d_{she} \leqslant 25$ let $d_{she} = 25$ mm

If $150 \leqslant d_{she} \leqslant 500$ let $d_{she} = 150$ mm

— Equations 14 and 15 apply only for:

$d_{she}^{0,36} \times \sigma_B > 2\,600$ N/mm^2

The equations for the allowable stress values have been developed based on the following conditions:

— state of the art shaft design is utilized which should result in keeping the effective stress concentration factors below the maxima listed for each equation;

— repeated torsional stress (zero to maximum) and reversed bending stress;

— equations 11 and 13 apply only to shaft sections with little stress concentration effect;

— the effects of a variable load spectrum is considered by the use of an appropriate selection factor, K_{sf};

— momentary overloads shall be limited to 200 % of P_{mc} applied for a number of stress cycles not exceeding 10 000;

— the material requirements are as specified in 5.4.3.

For through hardened materials:

if $0,09 \times (\sigma_B)^{0,4} < \beta_\tau \leqslant 0,113 \times (\sigma_B)^{0,4}$

$$\sigma_{sa} = [2,22 - 0,35 \times \log(d_{she})] \times \sigma_B^{0,6} \tag{10}$$

if $\beta_\tau \leqslant 0,09 \times (\sigma_B)^{0,4}$

$$\sigma_{sa} = [2,61 - 0,35 \times \log(d_{she})] \times \sigma_B^{0,6} \tag{11}$$

if $0,10 \times (\sigma_B)^{0,4} < \beta_\sigma \leqslant 0,175 \times (\sigma_B)^{0,4}$

$$\sigma_{ba} = [1,88 - 0,30 \times \log(d_{she})] \times \sigma_B^{0,63} \tag{12}$$

if $\beta_\sigma \leqslant 0,10 \times (\sigma_B)^{0,4}$

$$\sigma_{ba} = [2,40 - 0,31 \times \log(d_{she})] \times \sigma_B^{0,66} \tag{13}$$

For carburized and case hardened materials:

If $\beta_\tau \leqslant 0,113 \times {\sigma_B}^{0,4}$

$$\sigma_{sa} = [1,43 - 0,36 \times \log(d_{she})] \times \sigma_B^{0,68} \tag{14}$$

if $\beta_\sigma \leqslant 0,175 \times {\sigma_B}^{0,4}$

$$\sigma_{ba} = [6{,}02 - 1{,}58 \times \log(d_{she})] \times \sigma_B^{0{,}57} \quad (15)$$

where

σ_B is the material tensile strength, in N/mm^2;

σ_{sa} is the allowable torsional stress, in N/mm^2;

σ_{ba} is the allowable bending stress, in N/mm^2;

β_τ is the torsional notch factor;

β_σ is the bending notch factor.

For applications beyond the limits, a more detailed analysis may be required.

5.5.4 Material requirements

For through hardened materials the basis for defining allowable stress is the minimum surface hardness at the critically stressed section. The minimum hardness at a depth from the surface of 1/4 the radius of the critical section shall be 75 % of the minimum hardness at the surface.

For case hardened materials the basis for defining allowable stress is the minimum core hardness at a distance of three times the effective case depth below the surface in the critically stressed section.

For both through hardened and case hardened materials, the hardness will be converted to tensile strength by the conversion table in ISO 6336-5:1996, annex C.

The material for shafts shall meet the requirements of Grade ML of ISO 6336-5:1996. Materials with hardness greater than 241 BHN (255 HV) shall undergo magnetic particle inspection. Indications longer than 1 mm are not permitted in the critically stressed areas.

Ground surfaces shall be free from grinding temper in the critically stressed areas.

The hardness at the specified radius may be determined by measuring the hardness at the same radius on a representative test bar coupon of the same alloy which has been heat treated with the product shaft(s). The coupon shall have the same diameter as the shaft when it is heat treated. See ISO 6336-5:1996, 6.3.

Selection of the appropriate alloy grade shall be based on expected quench rate at the critical section, critical section size, and Jominy hardenability. See ISO 6336-5:1996, annex B for more information.

Statistical or other verifiable process control methods may be substituted for the detailed quality requirements when justified by the manufacturer's experience. See ISO 6336-5:1996, clauses 0, 4, 5.1, and 6.1 for more information.

5.5.5 Deflection

Shaft deflections shall be analyzed regardless of stress levels to ensure satisfactory tooth and bearing contact.

5.6 Keys

5.6.1 Application limits

This calculation method is applicable for keyed connections within the following limits (see Figure 1):

$b_k / d_{sh} \leqslant 0{,}36$

$(h_k - t_k) / t_k \leqslant 0{,}81$

$(h_k - t_k) / b_k \leqslant 0{,}45$

number of keys, $i \leqslant 2$