BS EN 13906-1:2013

BSI Standards Publication

Cylindrical helical springs made from round wire and bar — Calculation and design

Part 1 : Compression springs

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National foreword

This British Standard is the UK implementation of EN 13906-1:2013. It supersedes BS EN 13906-1:2002 which is withdrawn.

The UK participation in its preparation was entrusted to Technical Committee FME/9/3, Springs.

A list of organizations represented on this committee can be obtained on request to its secretary.

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Ressorts hélicoïdaux cylindriques fabriqués à partir de fils ronds et de barres - Calcul et conception - Partie 1: Ressorts de compression

Zylindrische Schraubenfedern aus runden Drähten und Stäben - Berechnung und Konstruktion - Teil 1: Druckfedern

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EUROPEAN COMMITTEE FOR STANDARDIZATION
COMITÉ EUROPÉEN DE NORMALISATION
EUROPÄISCHES KOMITEE FÜR NORMUNG

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Contents

Foreword

This document (EN 13906-1:2013) has been prepared by Technical Committee CEN/TC 407 “Project Committee - Cylindrical helical springs made from round wire and bar - Calculation and design”, the secretariat of which is held by AFNOR.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by January 2014, and conflicting national standards shall be withdrawn at the latest by January 2014.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. CEN [and/or CENELEC] shall not be held responsible for identifying any or all such patent rights.

This document supersedes EN 13906-1:2002.

This European Standard has been prepared by the initiative of the Association of the European Spring Federation ESF.

This European Standard constitutes a revision of EN 13906-1:2002 for which it has been technically revised. The main modifications are listed below:

— updating of the normative references,

— technical corrections.

EN 13906 consists of the following parts, under the general title *Cylindrical helical springs made from round wire and bar — Calculation and design*:

— *Part 1: Compression springs;*

— *Part 2: Extension springs;*

— *Part 3: Torsion springs.*

According to the CEN-CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Bulgaria, Croatia, Cyprus, Czech Republic, Denmark, Estonia, Finland, Former Yugoslav Republic of Macedonia, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, Switzerland, Turkey and the United Kingdom.

Introduction

The revision of EN 13906 series have been initiated by the Association of the European Spring Federation – ESF – in order to correct the technical errors which are in the published standards and to improve them according to the state of the art. However, the revision of the figures is not take part of this work due to the lack of shared (mutual) data to update them. Nevertheless, the customers can have updated data from the manufacturers.

1 Scope

This European Standard specifies the calculation and design of cold and hot coiled cylindrical helical compression springs with a linear characteristic, made from round wire and bar of constant diameter with values according to Table 1, and in respect of which the principal loading is applied in the direction of the spring axis.

Table 1

Characteristic	Cold coiled compression spring	Hot coiled compression spring
Wire or bar diameter	$d \leq 20$ mm	8 mm $\leq d \leq$ 100 mm
Number of active coils	$n \geq 2$	$n \geq 3$
Spring index	$4 \leq w \leq 20$	$3 \leq w \leq 12$

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

EN 10270-1, *Steel wire for mechanical springs — Part 1: Patented cold drawn unalloyed spring steel wire*

EN 10270-2, *Steel wire for mechanical springs — Part 2: Oil hardened and tempered spring steel wire*

EN 10270-3, *Steel wire for mechanical springs — Part 3: Stainless spring steel wire*

EN 10089, *Hot-rolled steels for quenched and tempered springs — Technical delivery conditions*

EN 12166, *Copper and copper alloys — Wire for general purposes*

EN ISO 2162-1:1996, *Technical product documentation — Springs — Part 1: Simplified representation (ISO 2162-1:1993)*

EN ISO 26909:2010, *Springs — Vocabulary (ISO 26909:2009)*

ISO 26910-1, *Springs — Shot peening — Part 1: General procedures*

3 Terms, definitions, symbols, units and abbreviated terms

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in EN ISO 26909:2010 and the following apply.

3.1.1
spring
mechanical device designed to store energy when deflected and to return the equivalent amount of energy when released

[SOURCE: EN ISO 26909:2010, 1.1]

3.1.2
compression spring
spring (1.1) that offers resistance to a compressive force applied axially

[SOURCE: EN ISO 26909:2010, 1.2]

3.1.3
helical compression spring
compression spring (1.2) made of wire of circular, non-circular, square or rectangular cross-section, or strip of rectangular cross-section, wound around an axis with spaces between its coils

[SOURCE: EN ISO 26909:2010, 3.12]

3.2 Symbols, units and abbreviated terms

Table 2 contains the symbols, units and abbreviated terms used in this European Standard.

Table 2 *(1 of 3)*

Symbols	Units	Terms
a_0	mm	gap between active coils of the unloaded spring
$D = \frac{D_e + D_i}{2}$	mm	mean diameter of coil
D_e	mm	outside diameter of spring
ΔD_e	mm	increase of outside diameter of the spring, when loaded
D_i	mm	inside diameter of spring
d	mm	nominal diameter of wire (or bar)
d_{max}	mm	upper deviation of d
E	N/mm² (MPa)	modulus of elasticity (or Young's modulus)
F	N	spring force
F_1, F_2 ...	N	spring forces, for the spring lengths L_1, L_2... (at ambient temperature of 20°C)
$F_{c\ th}$	N	theoretical spring force at solid length L_c NOTE The actual spring force at the solid length is as a rule greater than the theoretical force
F_K	N	buckling force
F_n	N	spring force for the minimum permissible spring length L_n
F_Q	N	spring force perpendicular to the spring axis (transverse spring force)
f_e	s^{-1} (Hz)	natural frequency of the first order of the spring (fundamental frequency)
G	N/mm² (MPa)	modulus of rigidity
k	-	stress correction factor (depending on D/d)
L	mm	spring length
L_0	mm	nominal free length of spring
L_1, L_2...	mm	spring lengths for the spring forces F_1, F_2...

Table 2 *(2 of 3)*

Symbols	Units	Terms
L_n	mm	minimum permissible spring length (depending upon S_a)
L_c	mm	solid length
L_K	mm	buckling length
m	mm	mean distance between centres of adjacent coils in the unloaded condition (pitch)
N	-	number of cycles up to rupture
n	-	number of active coils
n_t	-	total number of coils
R	N/mm	spring rate
R_m	N/mm² (MPa)	minimum value of tensile strength
R_Q	N/mm	transverse spring rate
S_a	mm	sum of minimum gaps between adjacent active coils at spring length L_n
s	mm	spring deflection
s_1, s_2 ...	mm	spring deflections, for the spring forces F_1, F_2 ...
s_c	mm	spring deflection, for the solid length, L_c
s_h	mm	deflection of spring (stroke) between two positions
s_K	mm	spring deflection, for the buckling force F_K (buckling spring deflection)
s_n	mm	spring deflection, for the spring force F_n
s_Q	mm	transverse spring deflection, for the transverse force F_Q
v_{St}	m/s	impact speed
W	Nmm	spring work,
$w = \frac{D}{d}$	-	spring index
η	-	spring rate ratio
λ	-	slenderness ratio
υ	-	seating coefficient
ξ	-	relative spring deflection
ρ	kg/dm³	density
τ	N/mm² (MPa)	uncorrected torsional stress (without the influence of the wire curvature being taken into account)
τ_1, τ_2 ...	N/mm² (MPa)	uncorrected torsional stress, for the spring forces F_1, F_2 ...
τ_c	N/mm² (MPa)	uncorrected torsional stress, for the solid length L_c

Table 2 *(3 of 3)*

Symbols	Units	Terms
τ_{kh}	N/mm² (MPa)	corrected torsional stress range, for the stroke s_h
τ_k	N/mm² (MPa)	corrected torsional stress (according to the stress correction factor k)
τ_{k1}, τ_{k2} ...	N/mm² (MPa)	corrected torsional stress, for the spring forces F_1, F_2 ...
$\tau_{kH\,(...)}$	N/mm² (MPa)	corrected torsional stress range in fatigue, with the subscript specifying the number of cycles to rupture or the number of ultimate cycles
τ_{kn}	N/mm² (MPa)	corrected torsional stress, for the spring force F_n
$\tau_{kO\,(...)}$	N/mm² (MPa)	corrected maximum torsional stress in fatigue, with the subscript specifying the number of cycles to rupture or the number of ultimate cycles
$\tau_{kU\,(...)}$	N/mm² (MPa)	corrected minimum torsional stress in fatigue, with the subscript specifying the number of cycles to rupture or the number of ultimate cycles
τ_n	N/mm² (MPa)	uncorrected torsional stress, for the spring force F_n
τ_{St}	N/mm² (MPa)	impact stress
τ_{zul}	N/mm² (MPa)	permissible static torsional stress

4 Theoretical compression spring diagram

The illustration of the compression spring corresponds to Figure 4.1 from EN ISO 2162-1:1996.

The theoretical compression spring diagram is given in Figure 1.

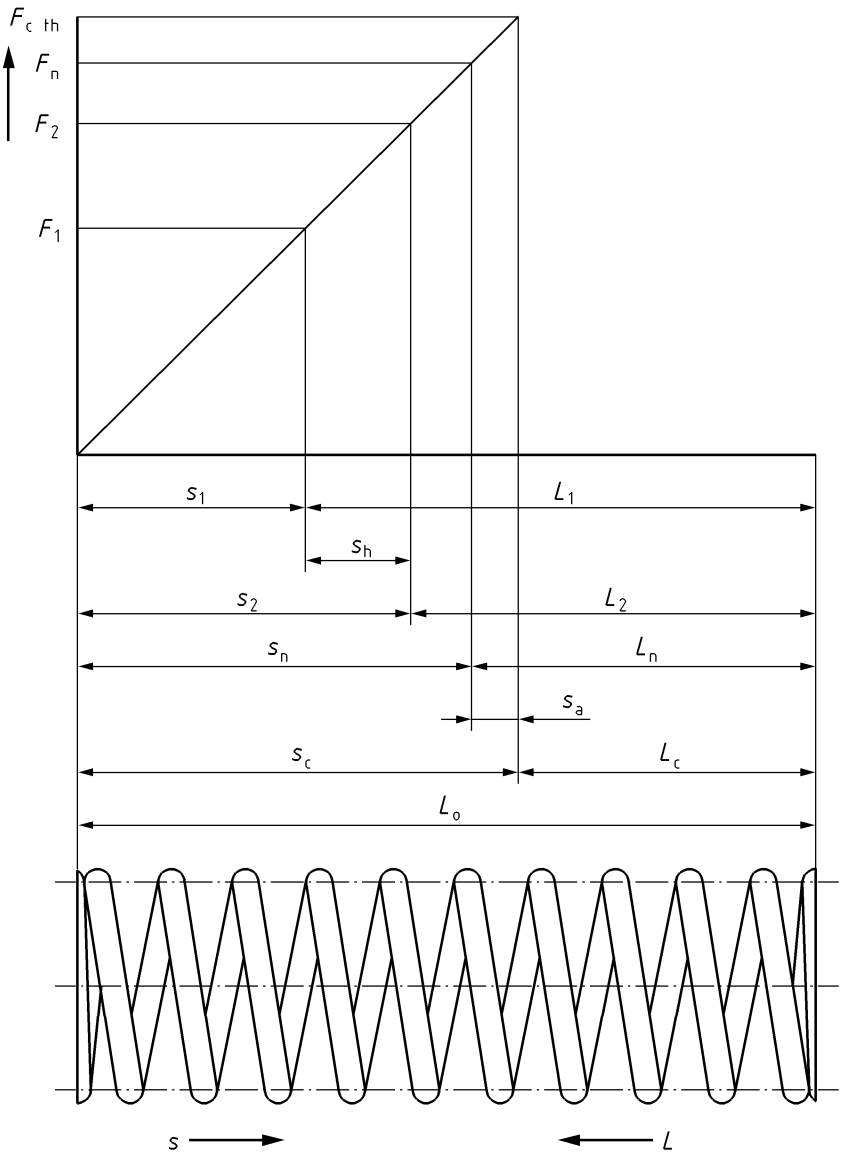


Figure 1 — Theoretical compression spring diagram

5 Design principles

Before carrying out design calculations for a spring, the requirements to be met shall be considered, particularly taking into account and defining:

— a spring force and corresponding spring deflection or two spring forces and corresponding stroke or a spring force, the stroke and the spring rate,

— loading as a function of time: is static or dynamic,

— in the case of dynamic loading the total number of cycles, N, to rupture,

— operating temperature and permissible relaxation,

— transverse loading, buckling, impact loading,

— other factors (e.g. resonance vibration, corrosion).

In order to optimise the dimensions of the spring by taking the requirements into account, sufficient working space should be provided when designing the product in which the spring will work.

6 Types of Loading

6.1 General

Before carrying out design calculations, it should be specified whether they will be subjected to static loading, quasi-static loading, or dynamic loading.

6.2 Static and/or quasi-static loading

A static loading is:

— a loading constant in time.

A quasi-static loading is:

— a loading variable with time with a negligibly small torsional stress range (stroke stress) (e.g. torsional stress range up to 0,1 × fatigue strength);

— a variable loading with greater torsional stress range but only a number of cycles of up to 10^4.

6.3 Dynamic loading

In the case of compression springs dynamic loading is:

Loading variable with time with a number of loading cycles over 10^4 and torsional stress range greater than 0,1 × fatigue strength at:

a) constant torsional stress range;

b) variable torsional stress range.

Depending on the required number of cycles N up to rupture it is necessary to differentiate the two cases as follows:

c) infinite life fatigue in which the number of cycles

  — $N \geq 10^7$ for cold coiled springs;

  — $N \geq 2 \times 10^6$ for hot coiled springs;

  In this case the torsional stress range is lower than the infinite life fatigue limit.

d) limited life fatigue in which

  — $N < 10^7$ for cold coiled springs;

  — $N < 2 \times 10^6$ for hot coiled springs.

  In this case the torsional stress range is greater than the infinite life fatigue limit but smaller than the low cycle fatigue limit.

In the case of springs with a time- variable torsional stress range and mean torsional stress, (set of torsional stress combinations) the maximum values of which are situated above the infinite fatigue life limit, the service life can be calculated as a rough approximation with the aid of cumulative damage hypotheses. In such circumstances, the service life shall be verified by means of a fatigue test.

6.4 Operating temperature

The data relating to the permissible loading of the materials used as given in Clause 10 apply at ambient temperature.

The influence of temperature shall be taken into consideration especially in the case of springs with closely toleranced spring forces. At operating temperatures below -30°C the reduction of the notch impact strength shall also be taken into account.

6.5 Transverse loading

If an axially loaded spring with parallel guided ends is additionally loaded perpendicular to its axis, transverse deflection with localised increase in torsional stress will occur, and this shall be taken into account in the calculation.

6.6 Buckling

Axially loaded springs have a tendency to buckle depending on the slenderness ratio when they are compressed to a certain critical length. Consequently, their buckling behaviour shall be checked. An adequate safety against buckling shall be allowed for in the design of these springs, because the buckling limit is reached in practice sooner than calculated theoretically. Springs which cannot be designed with an adequate safety against buckling shall be guided inside a tube or over a mandrel. Friction will be the inevitable consequence, and damage to the spring will occur in the long run. It is therefore preferable to split up the spring into individual springs, which are safe against buckling, as far as possible, and to guide these springs via intermediate discs over a mandrel or in a tube.

It shall always be borne in mind that the direction of the spring force does not coincide precisely with the geometric axis of the spring. Consequently, the spring will tend to buckle before the theoretical buckling limit has been attained. It is very difficult to allow for this effect by calculation. Buckling occurs in smooth progression.

6.7 Impact loading

Additional torsional stresses will be generated in a spring, if one end of the spring is suddenly accelerated to a high speed, e.g. through shock or impact. This impact wave will travel through the successive coils of the spring and will be reflected at the other end of the spring.

The level of this additional torsional stress depends on the speed with which the impact is delivered, but not on the dimensions of the spring.

6.8 Other factors

6.8.1 Resonance vibrations

A spring is prone to resonance vibrations by virtue of the inert mass of its active coils and of the elasticity of the material. A distinction is made between vibrations of the first order (fundamental vibrations) and vibrations of higher order (harmonic vibrations). The frequency of the fundamental vibration is called the fundamental frequency, and the frequency of the harmonic vibrations are integral multiples thereof.

When calculating springs, subject to high frequency forced vibration, care shall be taken to ensure that the frequency of the forced vibration oscillation (excitation frequency) does not come into resonance with one of the natural frequencies of the spring. In the case of mechanical excitations (e.g. via cams), resonance may also occur if a harmonic component of the excitation frequency coincides with one of the natural frequencies of the spring. In cases of resonance, an appreciable increase in torsional stress will arise at certain individual points of the spring, known as nodes. In order to avoid such increases in torsional stress due to resonance phenomena, the following measurers are advised:

— avoid integral ratios between excitation frequencies and natural frequencies;

— select the natural frequency of the first order of the spring as high as possible; avoid resonance with the low harmonics of the excitation;

— use springs with a progressive characteristic (variable pitch);

— design the cam with a favourable profile (low peak value of the excitation harmonics);

— provide for damping by means of spacers.

6.8.2 Corrosion influences, friction marks

The service life of springs is adversely affected by corrosion influences, by friction and chafing marks.

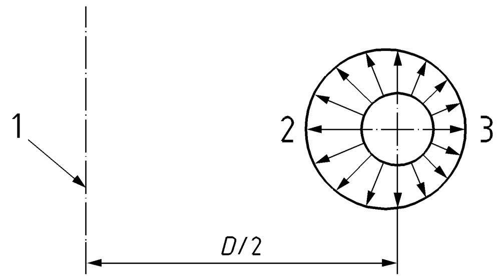
The service life of dynamically loaded springs in particular is reduced considerably by the influence of corrosion. Organic or inorganic coatings can be applied as a protection against corrosion. In the case of electroplated protective coatings, the risk of hydrogen embrittlement shall be borne in mind. Furthermore, various chrome-nickel steels or non-ferrous metals can be used depending on the risk of corrosion involved.

Damage to the surface of the spring, in the form of fretting corrosion, will occur as a result of the friction against surrounding components e.g. when the spring expands, and this will also lead to a considerable reduction in the service life of dynamically loaded springs.

7 Stress correction factor k

The distribution of torsional stresses over the cross section of the wire or bar of a spring is not uniform. The highest torsional stress occurs at the inside coil surface of the spring, due to the curvature of the wire or bar (see Figure 2).

The maximum torsional stress can be determined by approximation with the aid of the stress correction factor "k", which is dependent on the spring index. The factor shall be taken into account in the calculation on the maximum torsional stress, the minimum torsional stress and the torsional stress range of dynamically loaded springs. Its dependency on the spring index can be calculated with the aid of the approximate Formula (1), or obtained from Figure 3.



Key

1 spring axis
2 maximum torsional stress
3 minimum torsional stress

Figure 2 — Distribution of torsional stresses over the cross section of the wire or bar

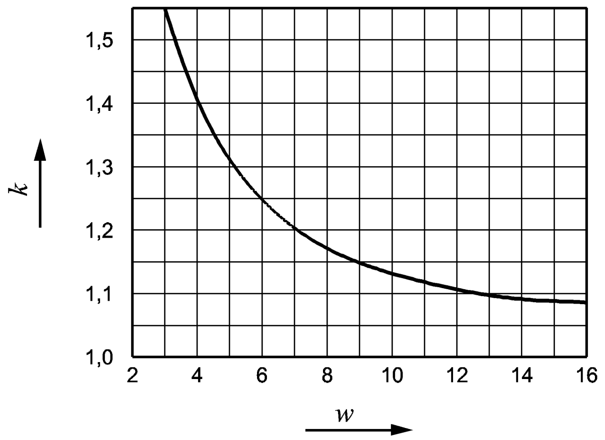


Figure 3 — Stress correction factor k against the spring index w

Approximation formula for the relationship between the stress correction factor k and the spring index w is according to Bergsträsser:

$$k = \frac{w + 0{,}5}{w - 0{,}75} \tag{1}$$

NOTE According to Wahl, an alternative to Formula (1) may also be used, giving approximately the same results:

$$k = \frac{4w - 1}{4w - 4} + \frac{0{,}615}{w}$$

8 Material property values for the calculation of springs

8.1 The material property values are for ambient temperature only and are given in Tables 3 and 4.

Table 3

Material	E N/mm^2 (MPa)	G N/mm^2 (MPa)	ρ kg/dm^3
Spring steel wire according to EN 10270-1	206 000	81 500	7,85
Spring steel wire according to EN 10270-2	206 000	79 500	7,85
Steels according to EN 10089	206 000	78 500	7,85
Copper-tin alloy CuSn6 R950 according to EN 12166 drawn spring hard	115 000	42 000	8,73
Copper-zinc alloy CuZn36 R700 according to EN 12166 drawn spring hard	110 000	39 000	8,40
Copper-beryllium alloy CuBe2 according to EN 12166	120 000	47 000	8,80
Copper-cobalt-beryllium alloy CuCo2Be according to EN 12166	130 000	48 000	8,80

NOTE Table 4 is an extract from EN 10270-3 where the unit has been changed from GPa to MPa.

Table 4 — Reference data for the modulus of elasticity and shear modulus (mean values) [a b c] for stainless steel wire (according to EN 10270-3)

Steel grade		Modulus of elasticity[a]		Shear modulus[b]	
Name	Number	Delivery condition GPa[d]	Condition HT GPa[d]	Delivery condition GPa[d]	Condition HT GPa[d]
X10CrNi18-8	1.4310	180	185	70	73
X5CrNiMo17-12-2	1.4401	175	180	68	71
X7CrNiAl17-7	1.4568	190	200	73	78
X5CrNi18-10	1.4301	185	190	65	68
X2CrNiMoN22-5-3	1.4462	200	205	77	79
X1NiCrMoCu25-20-5	1,4539	180	1 85	69	71

[a] The reference data for the modulus of elasticity (E) are calculated from the shear modulus (G) by means of the formula $G= E/2\ (1+\nu)$ where ν (Poisson's constant) is set to 0,3. The data are applicable for a mean tensile strength of 1 800 MPa. For a mean tensile strength of 1 300 MPa, the values are 6 GPa lower. Intermediate values may be interpolated.

[b] The reference data for the shear modulus (G) are applicable to wires with a diameter ≤ 2,8 mm for measurements by means of a torsion pendulum, for a mean tensile strength of 1 800 MPa. For a mean tensile strength of 1 300 MPa, the values are 2 GPa lower. Intermediate values may be interpolated. Values ascertained by means of an Elastomat are not always comparable with values ascertained by means of a torsion pendulum.

[c] For the finished spring, lower values may be ascertained. Therefore, standards for calculation of springs may specify values different from those given here on the basis of measurement of wire.

[d] 1 MPa = 1 N/mm^2, 1 GPa = 1 kN/mm^2.

8.2 The influence of the operating temperature on the modulus of elasticity and modulus of rigidity is given by the following formula, for averaged values, for the material listed in Tables 3 and 4.

$$G = G_{20} \times \left[1 - r \times (t - 20)\right] \tag{2}$$

with the following r values:

— $0{,}25 \times 10^{-3}$ for springs steel wire according to EN 10270-1, EN 10270-2 and EN 10089;

— $0{,}40 \times 10^{-3}$ for springs steel wire according to EN 10270-3;

— $0{,}40 \times 10^{-3}$ for springs alloy wire according to EN 12166.

9 Calculation formulae

9.1 Spring work

$$W = \frac{F\ s}{2} \tag{3}$$

9.2 Spring force

$$F = \frac{G\,d^4\,s}{8\,D^3\,n} \tag{4}$$

9.3 Spring deflection

$$s = \frac{8\,D^3\;n\,F}{G\,d^4} \tag{5}$$

9.4 Spring rate

$$R = \frac{G\,d^4}{8\,D^3\;n} \tag{6}$$

9.5 Torsional stresses

$$\tau = \frac{8\;D\;F}{\pi\,d^3} \tag{7}$$

$$\tau = \frac{G\;d\;s}{\pi\;\;n\,D^2} \tag{8}$$

$$\tau_k = k\;\tau \tag{9}$$

Whilst the torsional stress τ shall be adopted for the calculation of statically or quasi-statically loaded springs, the corrected torsional stress τ_{k} shall apply for dynamically loaded springs.

9.6 Nominal diameter of wire or bar

To calculate the optimum nominal diameter d of wire or bar then torsional stress τ is replaced with τ_{zul} as shown below.

$$d \geq \sqrt[3]{\frac{8\;F\,D}{\pi\;\tau_{\mathrm{zul}}}} \tag{10}$$

The permissible torsional stress τ_{zul} shall be selected according to the design case concerned (see Clause 10 in this connection).

9.7 Number of active coils

$$n = \frac{G\,d^4\;\;s}{8\,D^3\;\;F} \tag{11}$$

9.8 Total number of coils

The number of closed end coils required depends on the design of the spring ends and on the manufacturing process.

The total number of coils n_{t} should be:

— for cold coiled compression springs $n_{\mathrm{t}} = n + 2$ (12)

— for hot coiled compression springs $n_{\mathrm{t}} = n + 1{,}5$ (13)

A different value for the number of closed coils should be agreed between purchaser and manufacturer.

9.9 Minimum permissible spring length

For the minimum permissible spring length $L_n = L_c + S_a$, the sum of the minimum gaps between adjacent active coils shall be:

— for cold coiled springs: $S_a = n\left(0{,}0015\,\dfrac{D^2}{d} + 0{,}1\,d\right)$ (14a)

— for hot coiled springs: $S_a = 0{,}02\,n\,(D + d)$ (14b)

In the case of dynamically loaded springs, the value S_a in the above formulae shall be doubled for hot coiled springs and increased by a factor 1,5 for cold coiled springs.

9.10 Solid length

The solid length L_c is:

— for cold coiled springs with closed, ground ends

$$L_c \leq n_t\, d_{max} \tag{15}$$

— for cold coiled springs with closed, not ground ends

$$L_c \leq (n_t + 1{,}5)\, d_{max} \tag{16}$$

— for hot coiled springs with closed, ground ends

$$L_c \leq (n_t - 0{,}3)\, d_{max} \tag{17}$$

— for hot coiled springs with open, not ground ends

$$L_c \leq (n_t + 1{,}1)\, d_{max} \tag{18}$$

9.11 Increase of outside diameter of the spring when loaded

When a spring is compressed, the coil diameter increases very slightly. The increase of outside diameter of the spring when loaded, ΔD_e, is determined by the formula below, for solid length L_c and for free seating of the spring.

$$\Delta D_e = 0{,}1\,\frac{m^2 - 0{,}8\,m\,d - 0{,}2\,d^2}{D} \tag{19}$$

where

$m = \dfrac{s_c + n\,d}{n}$ for springs with closed, ground ends

$m = \dfrac{s_c + (n + 1{,}5)\,d}{n}$ for springs with open unground ends

9.12 Fundamental frequency

The natural frequency of the first order of a spring with both ends guided and one end periodically excited in the operating range, is determined by the following formula:

$$f_e = \frac{3560\ d}{n\ D^2}\sqrt{\frac{G}{\rho}} \tag{20}$$

9.13 Transverse loading

Under simultaneous application of forces in the longitudinal direction of the spring axis and perpendicular to it, a spring with parallel and guided ends becomes deformed as illustrated in Figure 4. There is a possibility of movement of base supports.

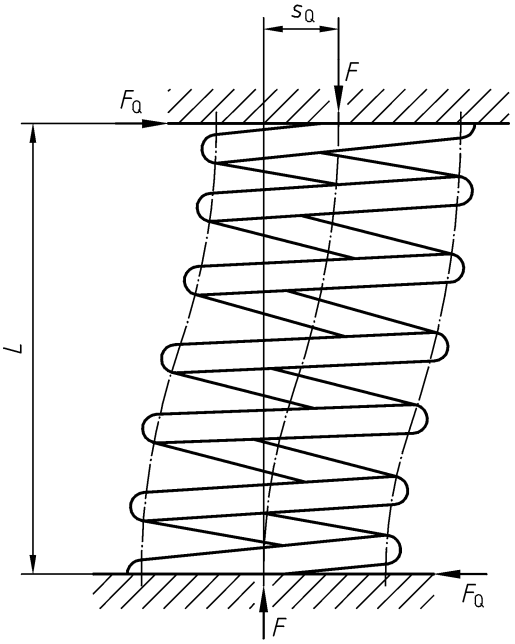


Figure 4 — Spring under simultaneous axial and transverse loading

Assuming that the spring ends do not lift off their seatings (see Formula (28) for the condition), the following formulas apply for the calculation:

Transverse spring rate: $R_Q = \dfrac{F_Q}{s_Q}$ (21)

Transverse spring deflection: $s_Q = \dfrac{F_Q}{R_Q} = \dfrac{F_Q}{\eta\ R}$ (22)

Spring rate ratio:

$$\eta = \frac{R_Q}{R} = \xi\left[\xi - 1 + \frac{\frac{1}{\lambda}}{\frac{1}{2}+\frac{G}{E}}\sqrt{\left(\frac{1}{2}+\frac{G}{E}\right)\left(\frac{G}{E}+\frac{1-\xi}{\xi}\right)}\tan\left\{\lambda\ \xi\sqrt{\left(\frac{1}{2}+\frac{G}{E}\right)\left(\frac{G}{E}+\frac{1-\xi}{\xi}\right)}\right\}\right]^{-1} \tag{23}$$

where:

— slenderness ratio $\lambda = \dfrac{L_0}{D}$ (24)

— relative spring deflection $\xi = \dfrac{s}{L_0}$ (25)

The transverse spring rate, R_Q, is only constant for short transverse spring deflections, for a given length L under compression. It varies with the seating conditions of the spring ends and with their rocking behaviour. In applications where the transverse stability of the spring is an important operational factor, the calculated values should be verified by practical tests.

Maximum torsional stress (incorporates torsional stresses arising from axial and transverse loading) is given by the following formula:

$$\tau_{max} = \frac{8}{\pi\ d^3}\ [F\,(D + s_Q) + F_Q\,(L - d)] \tag{26}$$

Maximum corrected torsional stress

$$\tau k_{max} = k\ \tau_{max} \tag{27}$$

Condition necessary for the spring ends resting on their supports

$$F_Q\,\frac{L}{2} \leq F\,\frac{D\text{-}s_Q}{2} \tag{28}$$

9.14 Buckling

Some springs have a tendency to buckle; the critical spring length at which buckling starts is known as the buckling length L_k and the spring deflection up to the point of buckling is known as the buckling spring deflection s_K.

The influence of the seating of the spring ends is taken into account by means of seating coefficient ν, which is specified in Figure 5 for the most common types of seating which occur.

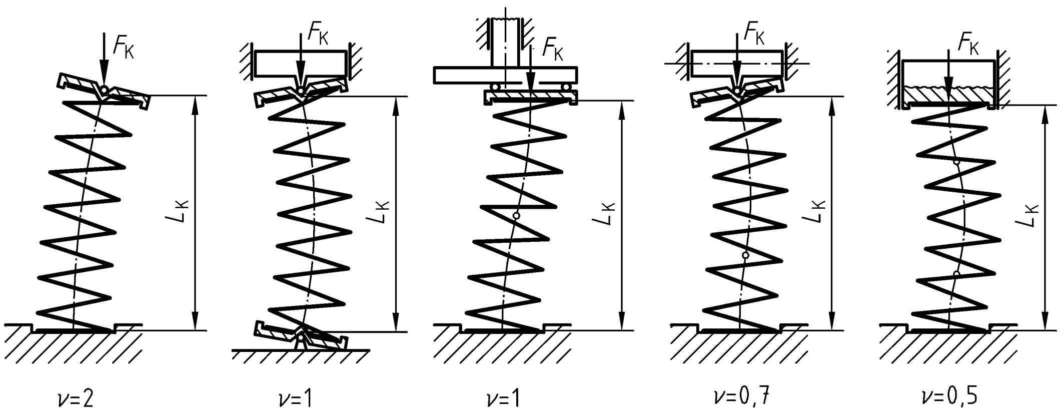


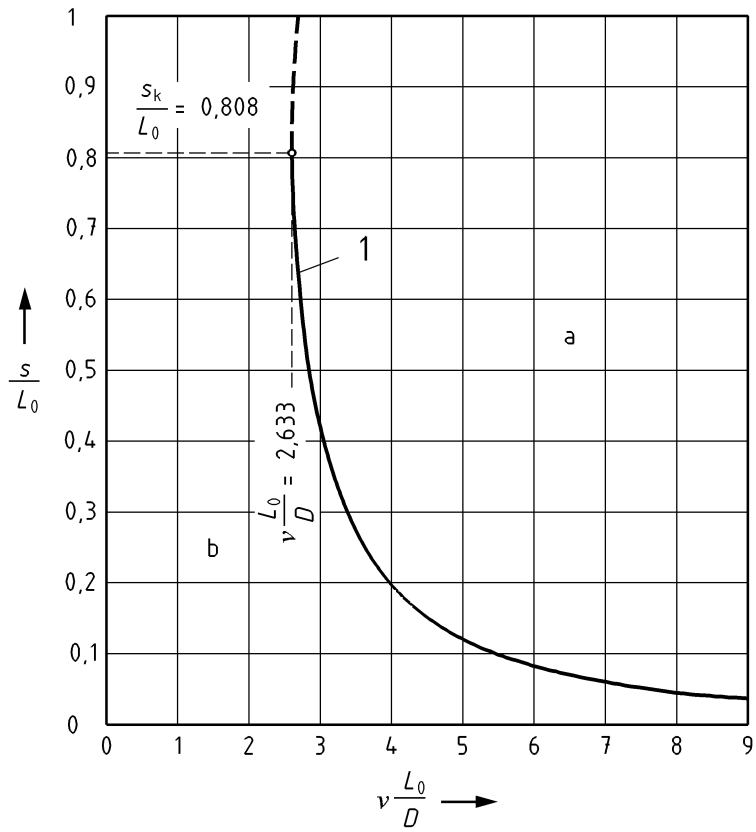
Figure 5 — Types of seating and associated seating coefficients ν of axially loaded springs

The buckling spring deflection is determined by the following formula:

$$s_K = L_0\,\frac{0{,}5}{1-\frac{G}{E}}\left[1-\sqrt{1-\frac{1-\frac{G}{E}}{0{,}5+\frac{G}{E}}\left(\frac{\pi D}{\nu\ L_0}\right)^2}\,\right] \tag{29}$$

Safety against buckling is achieved in theory for an imaginary square root value and for $\frac{s_K}{s} > 1$.

Safety against buckling can also be evaluated with the graph shown in Figure 6. On the right-hand side of the limit curve the spring is unstable, on the left-hand side the spring is stable.



Key

1 limit curve

a risk of buckling

b no buckling

Figure 6 — Theoretical buckling limit of helical compression springs

9.15 Impact stress

The impact torsional stress τ_{St} is determined by the following formula:

$$\tau_{St} = \nu_{St}\sqrt{2 \times 10^{-3}\rho\ G} \tag{30}$$

Formula (30) does not take into account the reflection of the impact waves and the effect of coil clashing.

10 Permissible torsional stresses

10.1 Permissible torsional stress at solid length

10.1.1 Cold coiled springs

For manufacturing reasons, it should be possible to compress all springs down to their solid length. The uncorrected permissible torsional stress at solid length, $\tau_{c\ zul}$ is usually $\tau_{c\ zul} = 0,56\ R_m$. The value of R_m (minimum value of tensile strength) is determined from the relevant standards referred to in Table 3. The strength values used in the calculation shall be the tensile strength values for the tempered condition or for the artificially aged condition.

10.1.2 Hot coiled springs

Figure 7 shows the values of the uncorrected permissible torsional stress at solid length, $\tau_{c\ zul}$ for hot coiled springs. The actual torsional stress which arises shall not exceed the uncorrected permissible torsional stress at solid length, which is a function of the strength of the material used and of the wire or bar diameter.

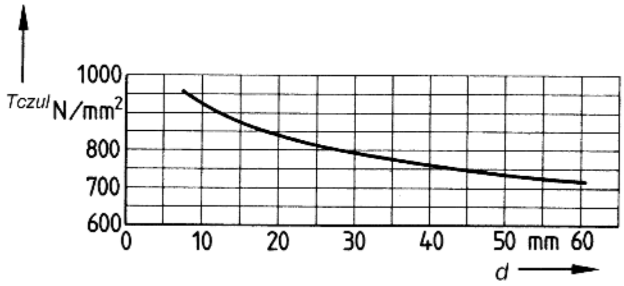


Figure 7 — Permissible torsional stress at solid length for hot coiled springs made from special steel specified in EN 10089 as a function of the wire or bar diameter

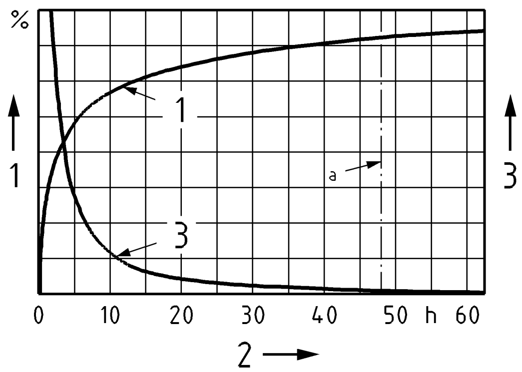
10.2 Permissible torsional stress under static or quasi-static loading

In the case of statically or quasi-statically loaded springs the permissible operating torsional stress is limited by the relaxation which can be tolerated, depending on the particular application concerned.

Relaxation is a loss of force at constant length, which is dependent on stress, temperature and time and in the present European Standard is represented in the form of a percentage loss related to the initial values. It shall only be verified in cases where stringent requirements have been specified with regard to the stability with time of the spring force.

The operating torsional stress shall be calculated without consideration of the stress correction factor k.

Figure 8 illustrates the pattern of the relaxation and of the relaxation velocity in principle. The relaxation values after 48 h are regarded as characteristic values, despite the fact the relaxation is not fully completed at this point in time.

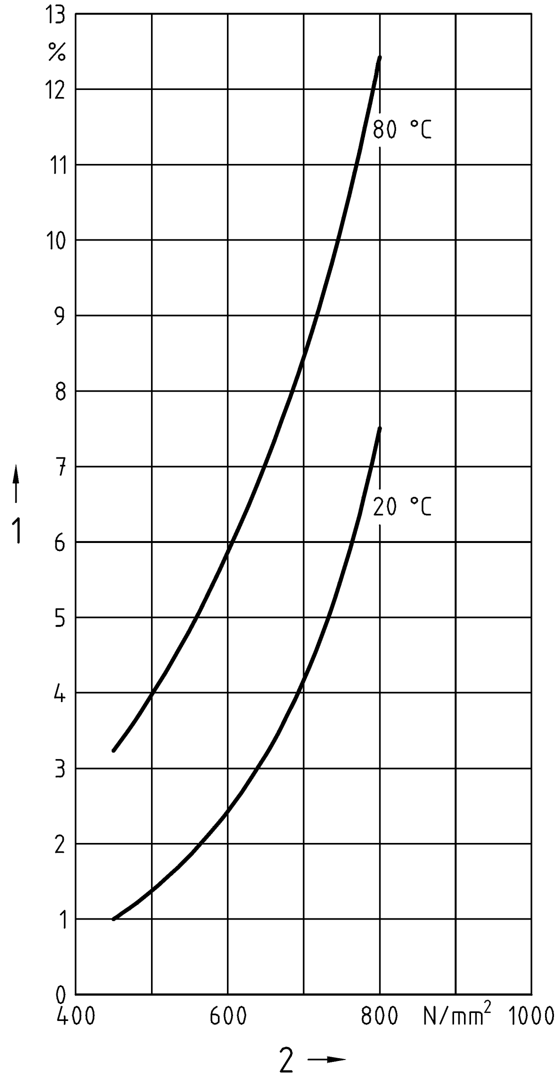


Key

1 relaxation
2 loading time
3 relaxation velocity
a 48 h

Figure 8 — Pattern of relaxation and relaxation velocity as a function of time

Data on the relaxation are plotted in Figure 9 for hot coiled springs. For cold coiled springs examples of relaxation as a function of the torsional stress prior to relaxation and or the operating temperature is included in Annex A.



Key

1 relaxation $\frac{\Delta F}{F(\tau)} \times 100$

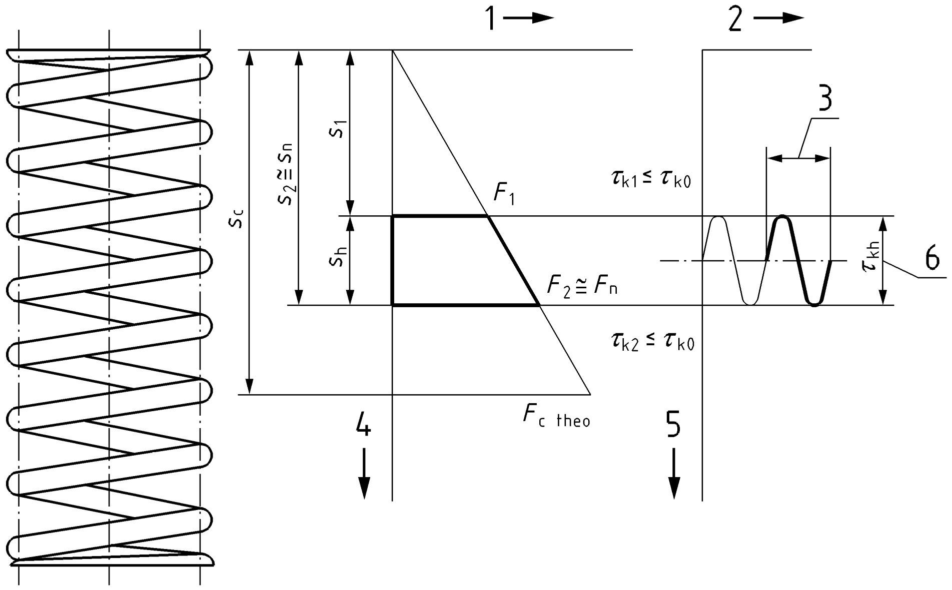
2 shear stress τ prior to relaxation

Figure 9 — Relaxation after 48 h of hot coiled springs made from steel according to EN 10089, having a strength due to the heat treatment of 1 500 N/mm² (MPa), preset at ambient temperature, as a function of the operating stress at various temperatures

10.3 Permissible stress range under dynamic loading

In the case of dynamically loaded springs, the permissible stress range is limited by the required minimum number of cycles and by the given wire or bar diameter.

The stress correction factor k shall be taken into consideration for the calculation. Stress range τ_{kh}, as illustrated in Figure 10 is the difference between τ_{k1} and τ_{k2}.



Key

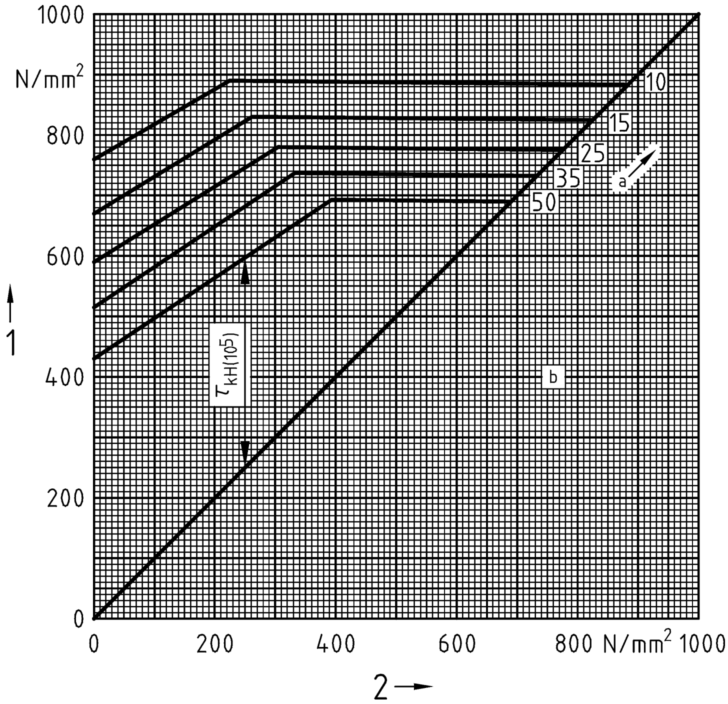
1 F
2 t = time
3 one load cycle
4 s
5 τ
6 amplitude

Figure 10 — Oscillation diagram of a spring subjected to dynamic stresses

For a given value of $\tau_{k1} = \tau_{kU}$, τ_{k2} shall not exceed τ_{kO} i.e. stress range τ_{kH} for the desired deflection of spring (stroke) between two positions s_h shall not exceed the value of the low cycle or infinite life fatigue limit $\tau_{kh\ (....)}$ which is determined from Figures 11 to 22. In the case of dynamically loaded springs, the solid length torsional stress $\tau_{c\ zul}$ shall also be verified taking into account the possibility that additional torsional stresses may be superimposed as a result of resonance in the spring body.

All dynamically loaded springs should be shot peened. Shot-peening is feasible, as a rule, in respect of springs with a nominal diameter of wire, d > 1 mm, a spring index, w < 15 and a gap between active coils, $a_0 > d$. The process of shot peening shall be defined in accordance with ISO 26910-1. The peening intensity and coverage should be as agreed between the purchaser and the supplier.

The values given in Figures 11 to 22 are not applicable to springs operating under the influence of corrosion or friction.



Key

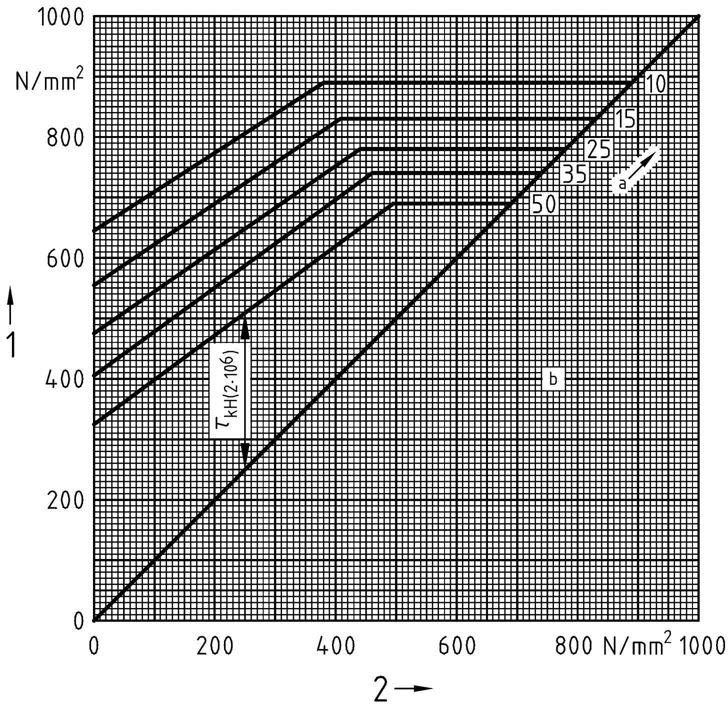
1 $\tau_{kO(10^5)}$

2 $\tau_{kU(10^5)}$

a d

b $N = 10^5$

Figure 11 — Low cycle fatigue strength diagram (Goodman diagram) for hot coiled springs, made from special quality steel specified in EN 10089, with ground or bright turned surface, shot peened



Key

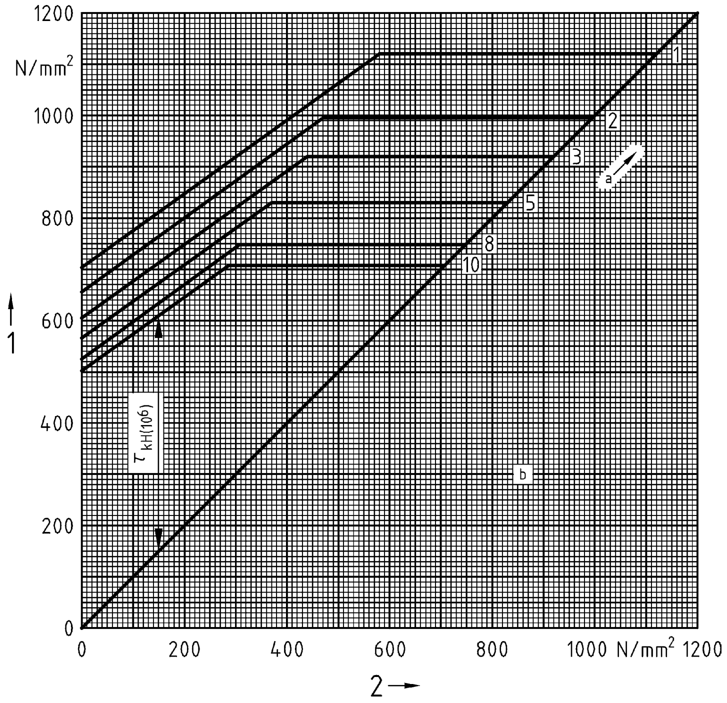
1 $\tau_{kO(2 \cdot 10^6)}$

2 $\tau_{kU(2 \cdot 10^6)}$

a d

b $N = 2 \times 10^6$

Figure 12 — Infinite life fatigue strength diagram (Goodman diagram) for hot coiled springs, made from special quality steel specified in EN 10089, with ground or bright turned surface, shot peened



Key

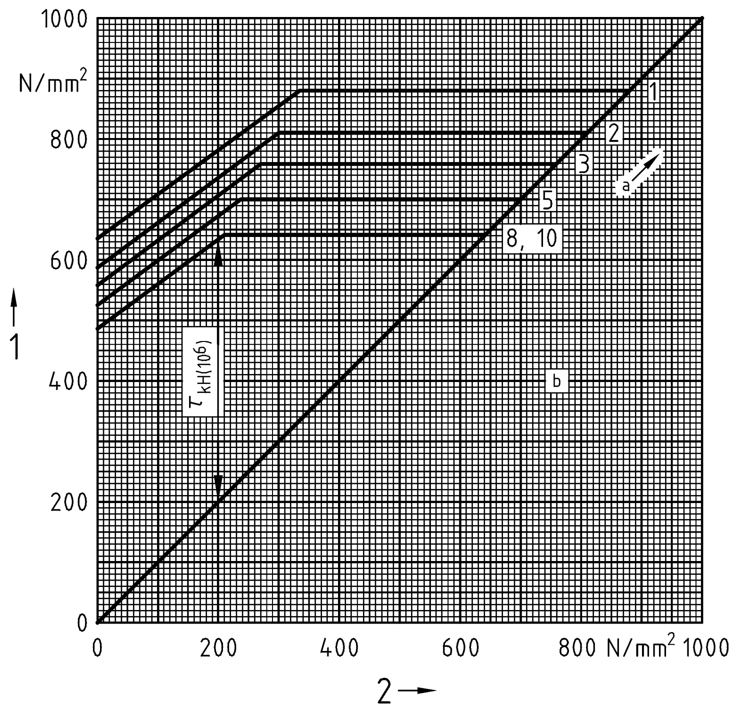
1 $\tau_{kO}\left(10^6\right)$

2 $\tau_{kU}\left(10^6\right)$

a d

b $N = 10^6$

Figure 13 — Low cycle fatigue strength diagram (Goodman diagram) for cold coiled springs, made from class DH or SH patented and drawn spring steel wire specified in EN 10270-1, shot peened



Key

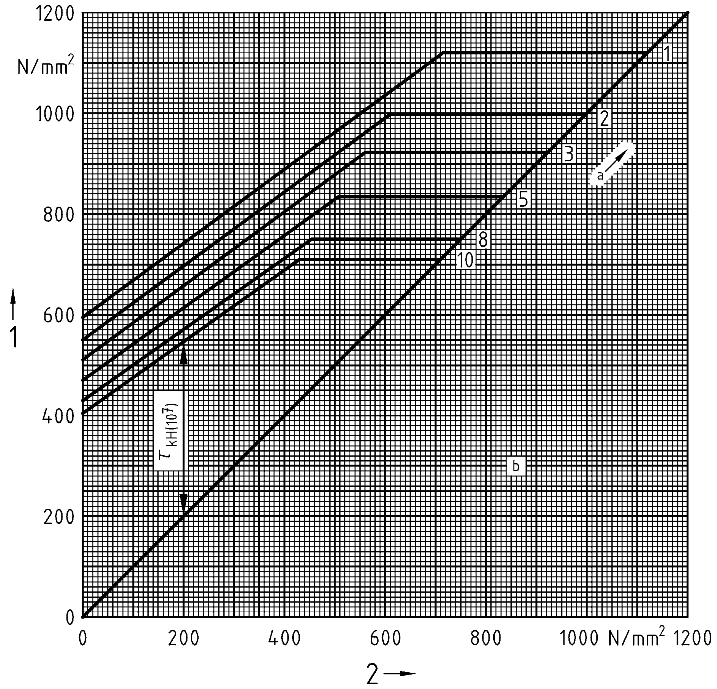
1 $\tau_{kO}\left(10^6\right)$

2 $\tau_{kU}\left(10^6\right)$

a d

b $N = 10^6$

Figure 14 — Low cycle fatigue strength diagram (Goodman diagram) for cold coiled springs, made from class FD or TD oil hardened and tempered wire specified in EN 10270-2, shot peened



Key

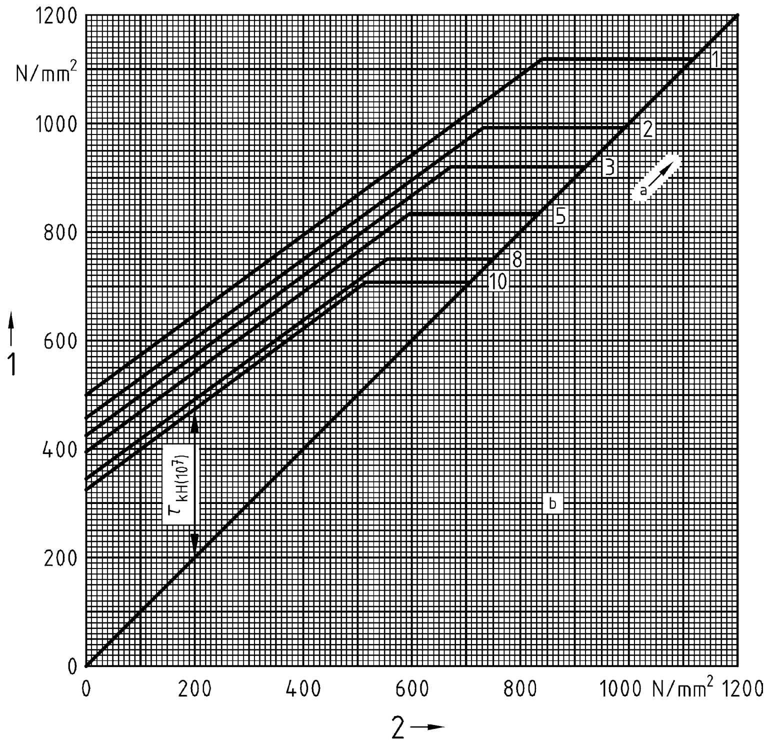
1 $\tau_{kO(10^7)}$

2 $\tau_{kU(10^7)}$

a d

b $N = 10^7$

Figure 15 — Infinite life fatigue strength diagram (Goodman diagram) for cold coiled springs, made from class DH or SH patented and drawn spring steel wire specified in EN 10270-1, shot peened



Key

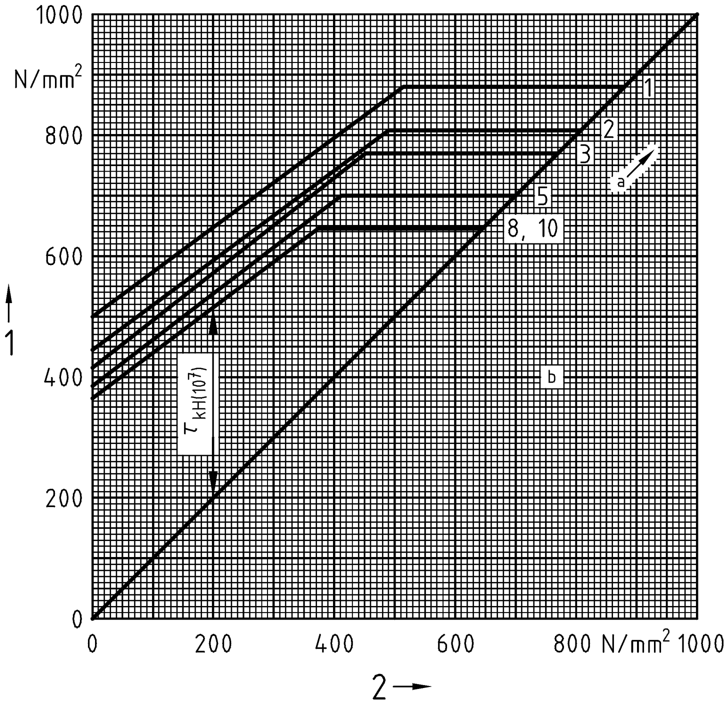
1 $\tau_{kO(10^7)}$

2 $\tau_{kU(10^7)}$

a d

b $N = 10^7$

Figure 16 — Infinite life fatigue strength diagram (Goodman diagram) for cold coiled springs, made from class DH or SH patented and drawn spring steel wire specified in EN 10270-1, not shot peened



Key

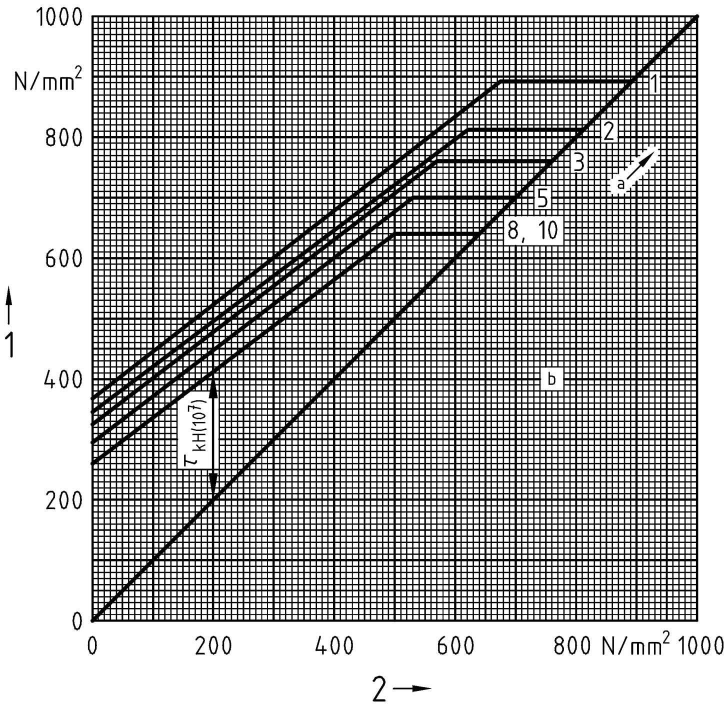
1 $\tau_{kO}\left(10^7\right)$

2 $\tau_{kU}\left(10^7\right)$

a d

b $N = 10^7$

Figure 17 — Infinite life fatigue strength diagram (Goodman diagram) for cold coiled springs, made from class FD or TD oil hardened and tempered wire specified in EN 10270-2, shot peened



Key

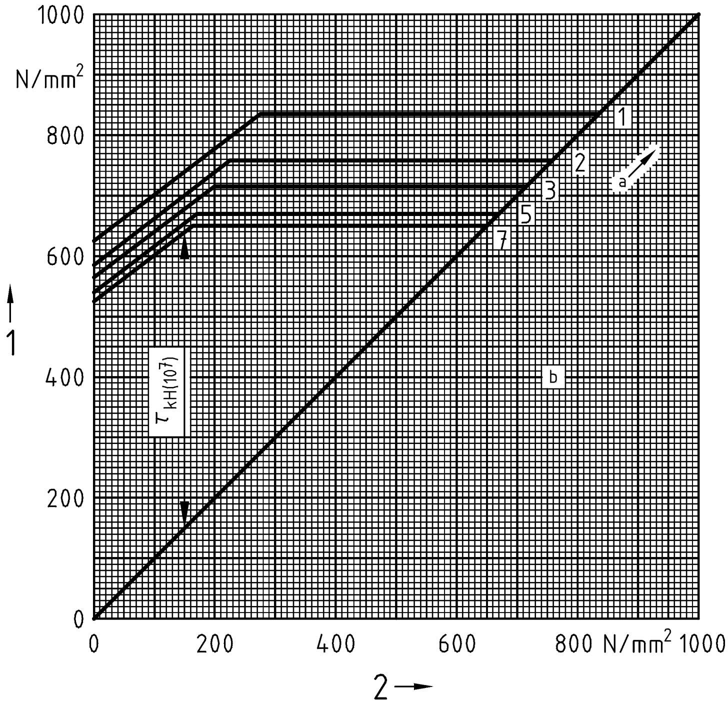
1 $\tau_{kO}\left(10^7\right)$

2 $\tau_{kU}\left(10^7\right)$

a d

b $N = 10^7$

Figure 18 — Infinite life fatigue strength diagram (Goodman diagram) for cold coiled springs, made from class FD or TD oil hardened and tempered wire specified in EN 10270-2, not shot peened



Key

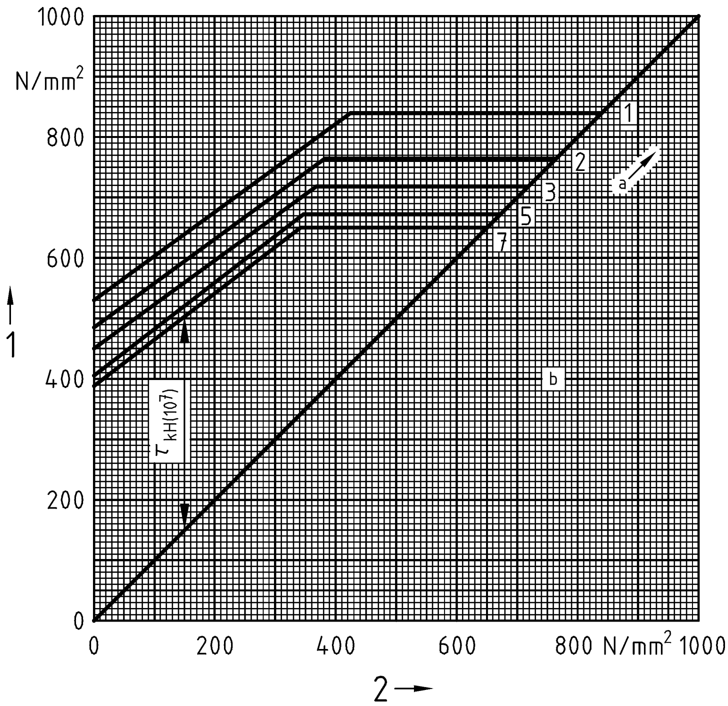
1 $\tau_{kO}(10^7)$

2 $\tau_{kU}(10^7)$

a d

b $N = 10^7$

Figure 19 — Infinite life fatigue strength diagram (Goodman diagram) for cold coiled springs, made from class VD valve spring quality oil hardened and tempered wire specified in EN 10270-2, shot peened



Key

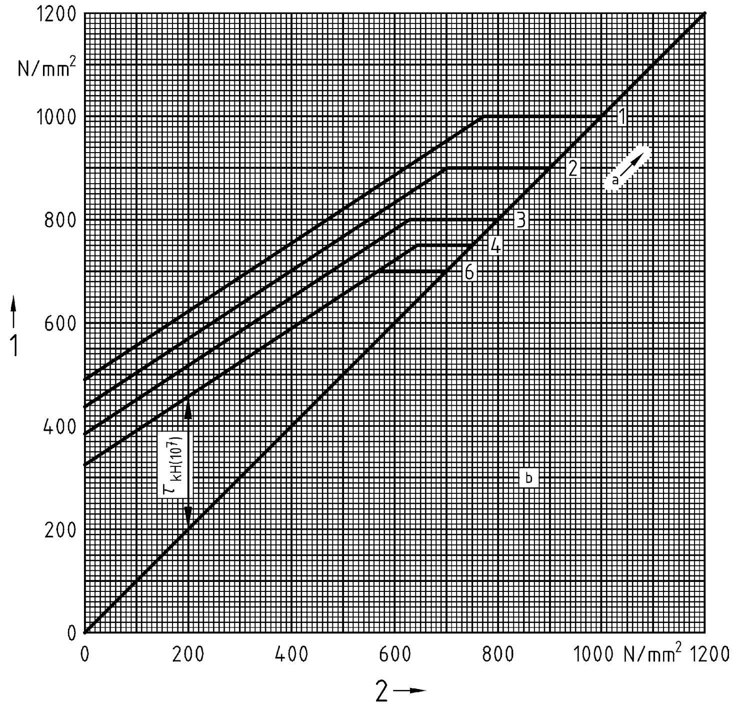
1 $\tau_{kO}(10^7)$

2 $\tau_{kU}(10^7)$

a d

b $N = 10^7$

Figure 20 — Infinite life fatigue strength diagram (Goodman diagram) for cold coiled springs, made from class VD valve spring quality oil hardened and tempered wire specified in EN 10270-2, not shot peened



Key

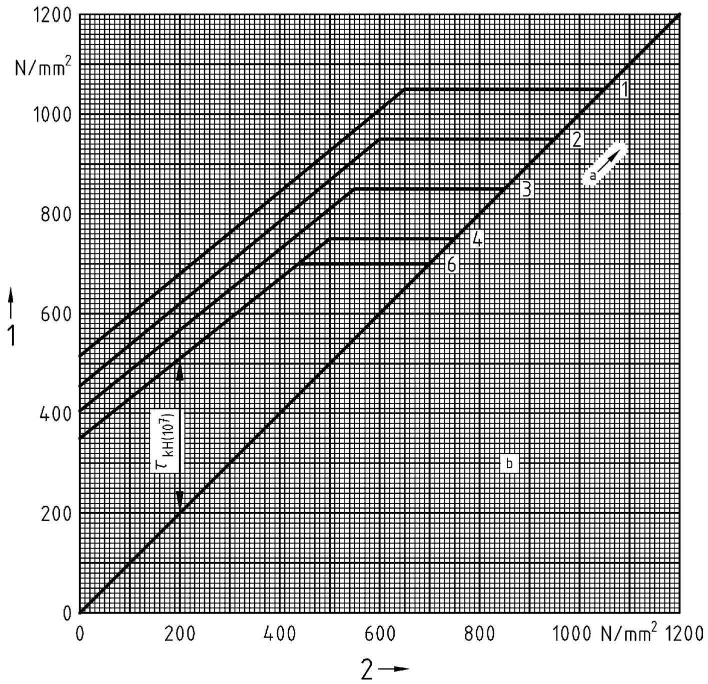
1 $\tau_{kO}\left(10^7\right)$

2 $\tau_{kU}\left(10^7\right)$

a d

b $N = 10^7$

Figure 21 — Infinite life fatigue strength diagram (Goodman diagram) for cold coiled springs, made from X10CrNi18-8, spring steel wire, material number 1.4310, specified in EN 10270-3, not shot peened



Key

1 $\tau_{kO}\left(10^7\right)$

2 $\tau_{kU}\left(10^7\right)$

a d

b $N = 10^7$

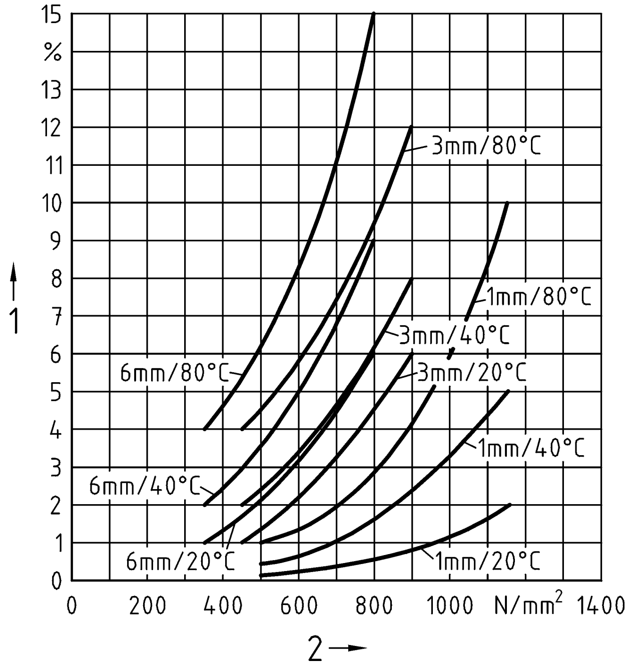
Figure 22 — Infinite life fatigue strength diagram (Goodman diagram) for cold coiled springs, made from X7CrNiAl17-7, spring steel wire, material number 1.4568, specified in EN 10270-3, not shot peened

Annex A
(informative)

Examples of relaxation for cold coiled springs

The following graphs represent guideline values, which presuppose conventional production methods, with setting at ambient temperature. These values can be influenced favourably by using appropriate materials by pre-setting at higher temperature and by increasing the strength resulting from the quenching and tempering heat treatment.

The relaxation after 48 h of cold coiled springs made from grades SH and DH wire specified in EN 10270-1, preset at ambient temperature, as a function of the torsional stress τ prior to relaxation at various temperatures, in degrees Celsius, and for the diameters 1 mm, 3 mm and 6 mm in the unpeened state is given in Figure A.1.



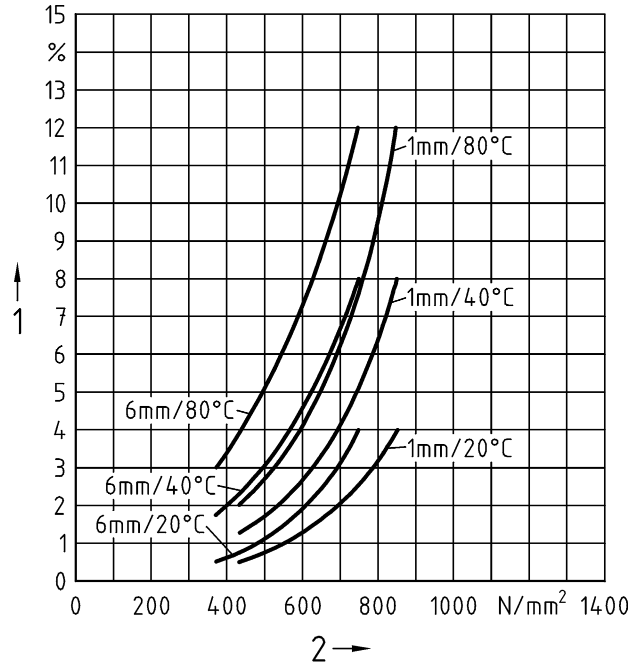
Key

1 relaxation $\frac{\Delta F}{F(\tau)} \times 100$

2 shear stress τ prior to relaxation

Figure A.1

The relaxation after 48 h of cold coiled springs made from grade VDC, wire (valve spring wire) and for FDC, grade wire specified in EN 10270-2, preset at ambient temperature, as a function of the torsional stress τ prior to relaxation at various temperatures, in degrees Celsius, and for the diameters 1 mm and 6 mm in the unpeened state is given in Figure A.2.



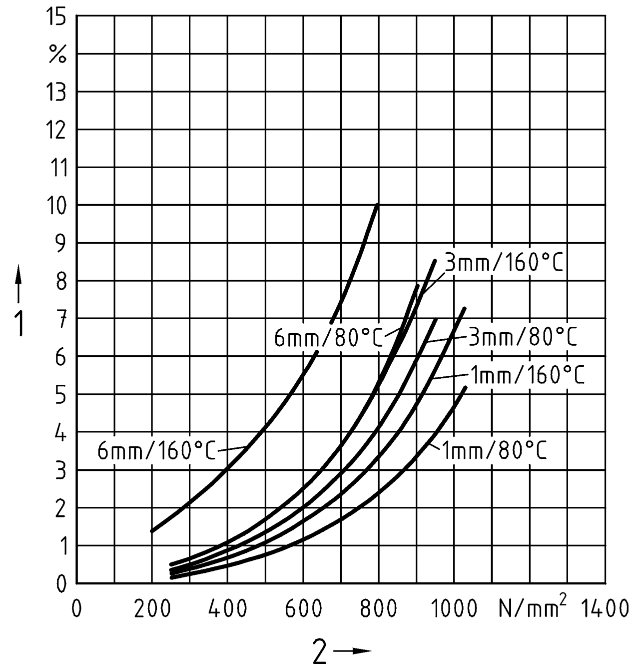
Key

1 relaxation $\frac{\Delta F}{F(\tau)} \times 100$

2 shear stress τ prior to relaxation

Figure A.2

The relaxation after 48 h of cold coiled springs made from SiCr grade (alloyed steel) specified in EN 10270-2, preset at ambient temperature, as a function of the torsional stress τ prior to relaxation at various temperatures, in degrees Celsius, and for the diameters 1 mm, 3 mm and 6 mm in the unpeened state is given in Figure A.3.



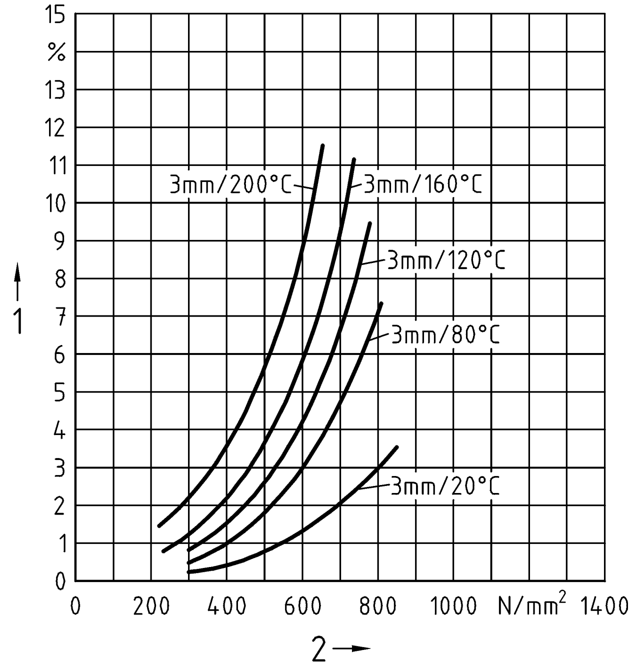
Key

1 relaxation $\frac{\Delta F}{F(\tau)} \times 100$

2 shear stress τ prior to relaxation

Figure A.3

The relaxation after 48 h of cold coiled springs made from CrV grade (Alloyed steel) specified in EN 10270-2, preset at ambient temperature, as a function of the torsional stress τ prior to relaxation at various temperatures, in degrees Celsius, and for the diameter 3 mm in the unpeened state is given in Figure A.4.



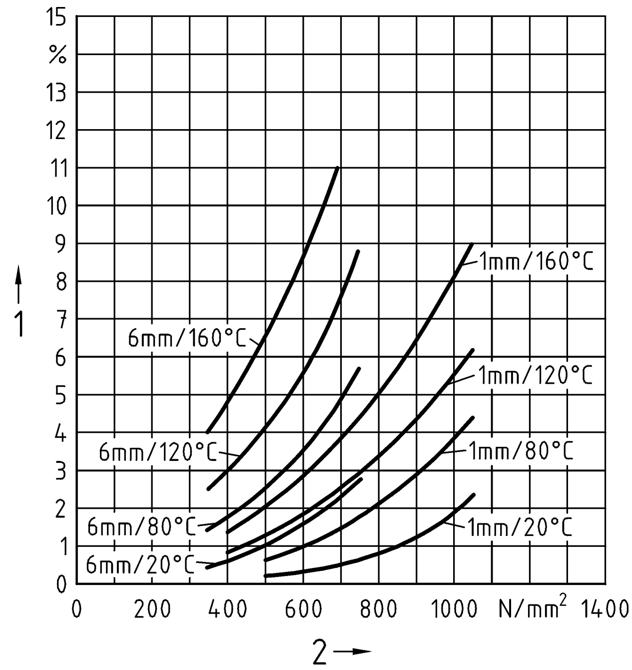
Key

1 relaxation $\frac{\Delta F}{F(\tau)} \times 100$

2 shear stress τ prior to relaxation

Figure A.4

The relaxation after 48 h of cold coiled springs made from grade 1.4310 wire specified in EN 10270-3, preset at ambient temperature, as a function of the torsional stress τ prior to relaxation at various temperatures, in degrees Celsius, and for the diameters 1 mm and 6 mm in the unpeened state is given in Figure A.5.



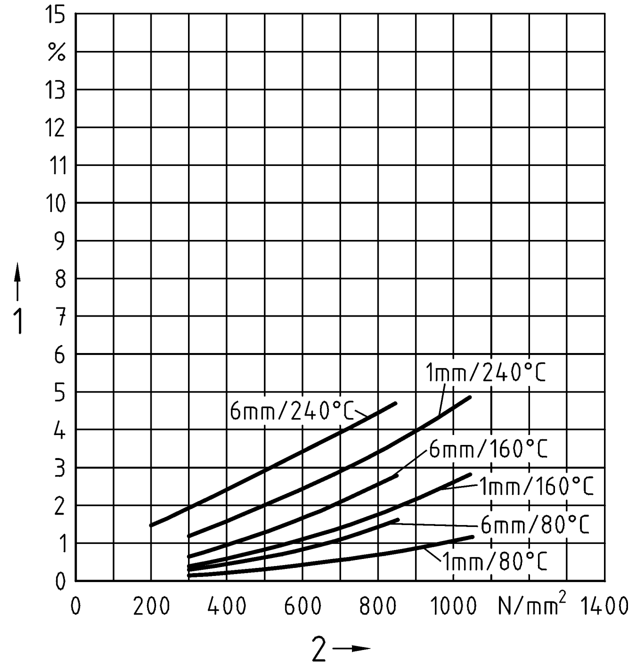
Key

1 relaxation $\frac{\Delta F}{F(\tau)} \times 100$

2 shear stress τ prior to relaxation

Figure A.5

The relaxation after 48 h of cold coiled springs made from grade 1.4568 wire specified in EN 10270-3, preset at ambient temperature, as a function of the torsional stress τ prior to relaxation at various temperatures, in degrees Celsius, and for the diameters 1 mm and 6 mm in the unpeened state is given in Figure A.6.



Key

1 relaxation $\dfrac{\Delta F}{F(\tau)} \times 100$

2 shear stress τ prior to relaxation

Figure A.6

Bibliography

[1] EN 15800, *Cylindrical helical springs made of round wire — Quality specifications for cold coiled compression springs*

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