IS 7906-1 (1997): Helical Compression Springs, Part 1: Design and Calculations for Springs Made from Circular Section Wire and Bar [TED 21: Spring]

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(Reaffirmed 2003)

भारतीय मानक

कुंडलाकार संपीडन कमानियां

भाग 1 वृताकार सेक्शन तार तथा छड़ों से बनी कमानियों मे डिजाइन तथा परिकलन

(पहला पुनरीक्षण)

Indian Standard

HELICAL COMPRESSION SPRINGS

PART 1 DESIGN AND CALCULATIONS FOR SPRINGS MADE FROM CIRCULAR SECTION WIRE AND BAR

(First Revision)

ICS 29.160

BUREAU OF INDIAN STANDARDS
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FOREWORD

This Indian Standard (First Revision) was adopted by the Bureau of Indian Standards after the draft finalized by the Springs Sectional Committee had been approved by the Light Mechanical Engineering Division Council.

This standard was originally published in 1976. This first revision incorporates a number of changes which were felt necessary as a result of further experience gained in the manufacture and use of compression spring and due to other development in this field.

The main modifications are:

a) The symbols have been harmonized with the internationally used symbols as per latest standards relating to springs.

b) The contents have been re-arranged in order to distinguish the relationship between individual spring parameters more clearly. The design principles, the types of stresses and permissible stresses have now been dealt within separate clauses.

c) Parameters E, G and ξ have been included or amended.

d) The values of minimum space, S_a, have been altered.

e) The formulae relating to the increase in coil diameter, ΔD_e, and to natural frequency, f_e have been altered.

f) The data relating to the transverse stability have been complemented by being expressed in formulae.

g) The values relating to permissible shear stresses at solid length condition have been altered.

h) Values for the relaxation of cold coiled and hot coiled springs have been included.

j) The creep and fatigue strength diagrams for cold coiled springs have been amended.

This standard is one of a series of standards on helical coiled compression springs. Other standards are as follows:

IS 7906 (Part 2) : 1975	Helical compression springs: Part 2 Specifications for cold coiled springs made from circular section wire and bar
IS 7906 (Part 3) : 1975	Helical compression springs: Part 3 Data sheet for specifications for springs made from circular section wire and bar
IS 7906 (Part 4) : 1987	Helical compression springs: Part 4 Selection of standard cold coiled springs made from circular section wire and bar
IS 7906 (Part 5) : 1989	Helical compression springs: Part 5 Hot coiled springs made from circular section bars — Specification (*first revision*)
IS 7906 (Part 6) : 1976	Helical compression springs: Part 6 Design and calculation for springs made from flat bar steel
IS 7906 (Part 7) : 1989	Helical compression springs: Part 7 Quality requirements for cylindrical coiled compression springs used mainly as vehicle suspension springs
IS 7906 (Part 8) : 1989	Helical compression springs: Part 8 Method of Inspection of hot coiled compression springs made from circular section bars

The object of the present standard is to provide an accurate and rapid method of determining the dimensions of helical springs made from circular section wire and bar. It can be used both for calculating the specification from available data and also for checking purposes. Worked examples have been included to promote understanding of the calculation methods.

(*Continued on third cover*)

Indian Standard

HELICAL COMPRESSION SPRINGS

PART 1 DESIGN AND CALCULATIONS FOR SPRINGS MADE FROM CIRCULAR SECTION WIRE AND BAR

(First Revision)

1 SCOPE

1.1 This standard (Part 1) lays down calculations for design of helical compression springs made from circular section wire and bar.

1.2 It applies to cold and hot coiled compression springs which have working coils with uniform pitch over the working length, are loaded in the direction of the spring axis and work at ordinary room temperature. For operations at considerably higher or lower temperature, reference should be made to the spring manufacturer.

2 REFERENCES

The following standards contain provisions which through reference in this text, constitute provision of this standard. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this standard are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below:

IS No.	*Title*
3195 : 1992	Steel for manufacture of volute and helical springs (for railway rolling stock) (*third revision*)
3431 : 1982	Steel for the manufacture of volute, helical and laminated springs for automotive suspension (*second revision*)
4454	Steel wire cold formed springs
(Part 1) : 1981	Patented and cold drawn steel wire — Unalloyed (*second revision*)
(Part 2) : 1975	Oil hardened and tempered spring steel wire and valve spring wire — Unalloyed (*first revision*)
(Part 3) : 1975	Oil hardened and tempered steel wires — Alloyed (*first revision*)
(Part 4) : 1975	Stainless steel wire for normal corrosion resistance (*first revision*)
7906	Helical compression springs
(Part 2) : 1975	Specification for cold coiled springs made from circular section wire and bar
(Part 3) : 1975	Data sheet for specification for springs made from circular section wire and bar
(Part 5) : 1989	Hot coiled springs made from circular section bars — Specification (*first revision*)

3 SYMBOLS

Following symbols and units shall apply (*see* Fig. 1):

Symbols		Terms	Unit
a_o	=	Space between the active coils of the unloaded spring	mm
D	=	$\frac{D_e + D_i}{2}$ = Mean coil diameter	mm
D_e	=	Outside coil diameter	mm
ΔD_e	=	Increase in coil diameter	mm
D_i	=	Inside coil diameter	mm
d	=	Wire or rod diameter	mm
d_{Max}	=	Maximum diameter of wire	mm
E	=	Modulus of elasticity	N/mm^2
F	=	Spring force (resilience)	N
F_1, F_2	=	Spring forces, correlated to spring lengths L_1, L_2	N

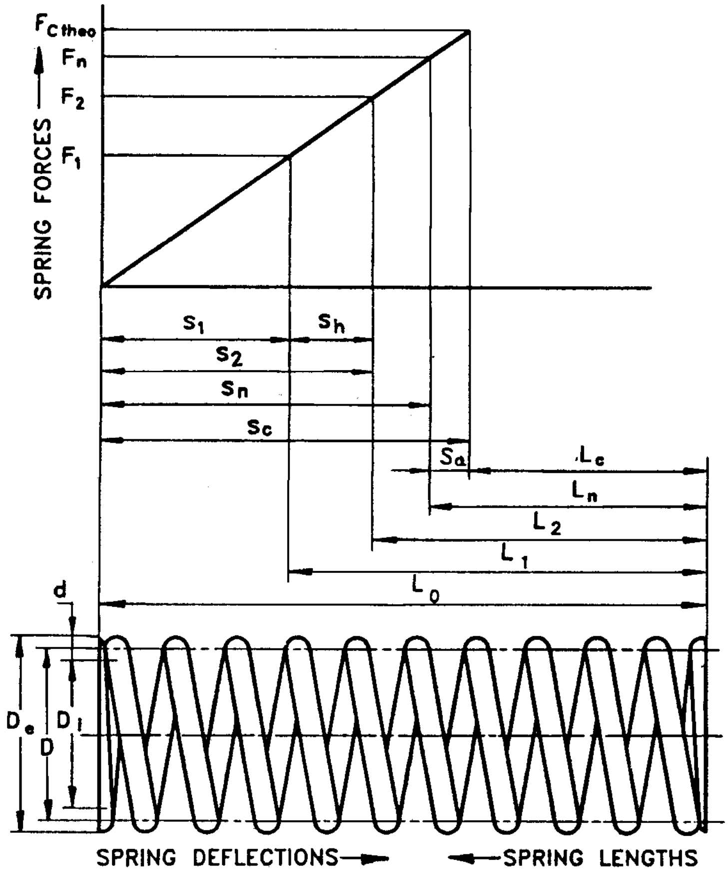

FIG. 1 THEORETICAL COMPRESSION SPRING DIAGRAM FOR CALCULATION AND DESIGN COMPRESSION SPRINGS

Symbols		Terms	Unit
F_n	=	Spring force, correlated to minimum permitted spring length L_n (taking S_a into consideration)	N
$F_{c\ theo}$	=	Theoretical spring force, correlated to the solid length L_c theo NOTE — The actual springs force in the solid length condition is as a general rate greater than theoretical force.	N
F_K	=	Buckling force	N
F_Q	=	Spring force perpendicular to the spring axis	N
f_e	=	Natural frequency of the spring	S^{-1}
G	=	Shear modulus	N/mm^2
k	=	Stress correction factor, taking into account the increase in shear stress as a result of wire curvature as a function of the coiling ratio	

Symbols		Terms	Unit
L	=	Spring length	mm
L_o	=	Length of unloaded spring	mm
L_1, L_2	=	Length of spring under load correlated to spring forces F_1, F_2	mm
L_n	=	Minimum permitted test length of the spring taking S_a into consideration	mm
L_c	=	Solid length, shortest possible spring length (all the coils are in contact with one another)	mm
L_k	=	Bukling length	mm
m	=	Mean distance between centres of adjoining coils (coil spacing; pitch)	mm
N	=	Number of load cycles	
n	=	Number of active coils	
n_t	=	Total number of coils	

Symbols		Terms	Unit
R	=	Spring rate	N/mm
R_m	=	Minimum value of tensile strength	N/mm^2
R_Q	=	Transverse spring rate	N/mm
S_a	=	Sum of minimum gaps between adjoining active coils at spring length L_n	mm
s	=	Spring deflection	mm
$s_1, s_2 .... s_n$	=	Spring deflection correlated to spring forces F_1, $F_2 .... F_n$	mm
s_c	=	Spring deflection, correlated to solid length L_c	mm
s_h	=	Working deflection (stroke) of the spring	mm
s_K	=	Spring deflection correlated to the buckling force F_K	mm
s_Q	=	Spring deflection correlated to the spring force F_Q	mm
v_{st}	=	Impact rate	m/s
W	=	Springing work	N.mm
ω	=	$\frac{D}{d}$ = Spring index	
η	=	$\frac{R_Q}{R}$ = Spring rate ratio	
λ	=	$\frac{L_0}{D}$ = Slenderness ratio	
ξ	=	$\frac{s}{L_0}$ = Relative spring deflection	
ν	=	seating coefficient	
ρ	=	Density	kg/dm^3
g	=	Acceleration due to gravity	m/s^2
τ	=	Shear stress without the influence of the wire curvature being taken into account	N/mm^2
$\tau_1, \tau_2 ... ... \tau_n$	=	Shear stress to the spring forces $F_1, F_2 ..... F_n$	N/mm^2
τ_c	=	Shear stress correlated to solid length L_c	N/mm^2
τ_k	=	Corrected shear stress, taking the influence of the wire curvature by correction factor k into account	N/mm^2
$\tau_{k1}, \tau_{k2} ... ... \tau_{kn}$	=	Corrected shear stress correlated to the spring forces $F_1, F_2, ... F_n$	N/mm^2
τ_{kh}	=	Corrected stress amplitude correlated to working travel S_h	N/mm^2

Symbols		Terms	Unit
τ_kH (. . .)	=	Corrected stress amplitude creep or fatigue strength value with the index specifying the number of load cycles to fracture or the number of ultimate load cycles	N/mm^2
τ_kU (. . .)	=	Corrected minimum stress, creep or fatigue strength value with the index specifying the number of load cycles to fracture or the number of ultimate load cycles	N/mm^2
τ_kO (. . .)	=	Corrected maximum stress, creep or fatigue strength value with the index specifying the number of load cycles to fracture or the number of ultimate load cycles	N/mm^2
τ_{zul}	=	Permissible shear stress	N/mm^2
τ_{st}	=	Increase in shear stress in the event of impact loading	N/mm^2
(. . .)	=	Indicates the stresses relating to number of cycles	

4 DESIGN PRINCIPLES

Before carrying out design calculations, the requirements to be met by the spring being designed should be clearly specified. Specific attention should be paid to the following considerations:

- whether the loading, as a function of time, is static or dynamic;
- minimum number of load cycles to fracture in the case of dynamic loading;
- operating temperature and permissible relaxation;
- effect of transverse loads, impact loads and buckling;
- Whether importance to be given to only one force associated deflection/loaded height or two spring forces and associated deflections/loaded heights or one spring force, the associated deflection/loaded height and spring rate; and
- other influences, such as effect of corrosion space considerations and resonance, etc.

NOTE — Adequate space should be provided for the spring to operate satisfactorily in its intended application.

5 MODE OF LOADING

Before any calculations are made, the type of load and stresses the spring will be subjected to, shall be established.

5.1 Static or Infrequently Varying Load

Spring under static load or infrequently varying load are:

a) all springs which carry only static load;
b) loads variable with time, with negligibly small stresses amplitude (guide line: stress amplitude not exceeding 0.1 times the fatigue strength amplitude); and
c) loads variable with time, with higher stresses amplitude than above, but with number of load cycles not exceeding 10 000.

5.2 Dynamic Loads

5.2.1 Loads variable with time, with number of cycles exceeding 10^4 and with the stress amplitude exceeding 0.1 times the fatigue strength of stroke at:

a) constant stress amplitude, and
b) variable stress amplitude.

shall be deemed to be dynamic stresses.

5.2.2 Compressive springs subjected to dynamic loads and made of spring steel may be divided into two groups.

a) *Group* 1 — Springs with virtually infinite life without failure. These springs shall withstand N not less than 10^7 in the case of cold coiled springs, and N not less than 2×10^6 in the case of hot coiled springs.
   In this case, the stress amplitude is less than the fatigue stroke strength.
b) *Group* 2 — Springs with finite life. Such springs may settle due to fatigue or break with N less than 10^7 in the case of cold coiled springs, and N less than 2×10^6 in the case of hot coiled springs.

In this case, the stress amplitude is more than the fatigue stroke strength.

In such cases, life can be estimated only by conducting endurance test/life cycle test on a number of springs of same design, dimensions and material.

5.3 Transverse Loads

If an axially loaded helical compression spring is subjected to transverse loads (in a direction perpendicular to its axis) then localized stresses will occur in the spring. The effect of the combined stresses (axial and transverse) shall be taken into account during calculation.

NOTE — It is not advisable to load a helical compressions spring transverse to its axis.

5.4 Thermal Loads

The data relating to permissible stresses given in this standard is valid only, if the spring operates between −30°C and 120°C

For springs operating below −30°C the reduction in notch impact strength shall be taken into account. Similarly for springs operating at above 120°C reduction in tensile strength shall be accounted.

5.5 Buckling

Sometimes axially loaded helical compression springs have a tendency to buckle at certain slenderness ratio. Therefore it is necessary to check the buckling behaviour of such springs.

An adequate safety against buckling shall be allowed for in the design of these springs, as it is practically found that buckling starts sooner than calculated theoretically. Compression springs which cannot be designed with an adequate safety against buckling shall be guided inside a tube or over a spindle. 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 total deflection of a spring into regions of deflection in which it is safe in buckling and beyond which certain guides may be necessary.

5.6 Impact Loads

Increased shear stresses will be generated in a helical compression spring if one end of the spring is suddenly accelerated to a high velocity, due to shock or impact. The 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 increased stress depends on the velocity with which the impact is delivered, but not on the dimensions of the spring dimensions.

5.7 Other Influences on Spring Design

5.7.1 *Resonance*

A helical compression spring is prone to natural oscillations by virtue of the inert mass of its active coils and of the elasticity of the material. A distinction is to be made between an oscillation of the first order (fundamental oscillation) and oscillations of higher orders, so-called harmonic oscillations. The frequency of the fundamental oscillation is known as the fundamental frequency, and the frequencies of the harmonic oscillations are integral multiples thereof. While designing springs, care shall be taken to ensure that the frequency of the imposed oscillation (excitation frequency) does not come into resonance with one of the natural frequencies of the spring. In the case of mechanical oscillation the nature of which is known, for examples through the cams resonance may also occur if a harmonic component of the excitation frequencies coincides with one of the natural frequencies of the spring. In cases of resonance, an appreciable increase in stress will arise at certain individual points of the spring, called oscillation nodes. The following measures are recommended to avoid such increases in stress due to resonance phenomena:

a) avoid integral ratios between excitation frequencies and natural frequencies;
b) select the natural frequency of the first order of the spring as high as possible; exclude reasonance with low harmonics of the excitation;
c) use springs with a variable spring rate ;
d) design the cam with a favourable profile (low peak values of the excitation harmonics); and
e) provide for damping by means of shims.

5.7.2 *Influences of Corrosion Friction and Chafing Marks*

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

The service life of dynamically stressed springs in particular is reduced considerably by the influence of corrosion. Suitable coating 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.

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

6 STRESS CORRECTION FACTOR, k

6.1 The distribution of shear stresses in the cross section of the wire/bar of a helical compression spring is non-uniform. The highest stress occurs at the inner edge of the spring wire/bar cross section, due to the curvature of the wire bar (*see* Fig. 2).

6.2 The maximum stress can be approximately calculated with the aid of a stress correction factor k, which is a function of the spring index. This factor shall be taken into account in the design of the maximum stress, minimum stress and stress amplitude of dynamically stressed springs. This factor being a function of spring index ω can be calculated with the aid of an empirical formula (equation 1) or obtained from Fig. 3.

6.3 The stress correction factor can be calculated by Wahl's empirical formula:

$$k \doteq \frac{4\omega - 1}{4\omega - 4} + \frac{0.615}{\omega} \quad ...(1)$$

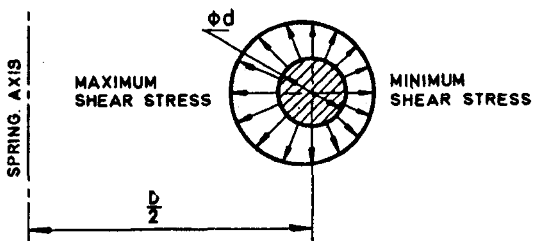

FIG. 2 DISTRIBUTION OF SHEAR STRESSES IN THE CROSS SECTION OF THE WIRE OR ROD

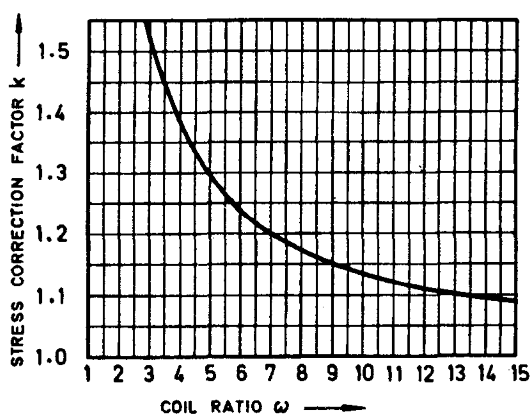

FIG. 3 STRESS CORRECTION FACTOR, K, AS A FUNCTION OF SPRING INDEX, ω

Approximate equation for the relationship between stress correction factor k and spring index ω according to Wahl's:

$$k = \frac{4\omega - 1}{4\omega - 4} + \frac{0.615}{\omega}$$

7 VALUES OF MODULUS OF ELASTICITY, SHEAR MODULUS AND DENSITY USED FOR THE DESIGN OF STEEL SPRINGS

7.1 The values given in the Table 1 are applicable for ambient temperature conditions.

7.2 Effect of operating temperature on modulus of elasticity and shear modulus is shown in Fig. 4 below. This may be considered as a guide.

Table 1 Values of Modulus of Elasticity, Shear Modulus and Density

Material	E N/mm^2	G N/mm^2	ρ kg/dm^3
Patended and cold drawn spring steel wire — Unalloyed, Grades 1, 2, 3 and 4 to IS 4454 (Part 1) : 1981	206 000	81 500	7.85
Oil hardened and tempered spring steel wire and valve spring wire — unalloyed Grades SW and VW to IS 4454 (Part 2) : 1975	200 000	79 500	7.85
Alloyed, oil hardened and tempered valve spring wire and spring steel wire for use under moderately elevated temperatures Grade 1S, 1D, 2S and 2D to IS 4454 (Part 3) : 1975	200 000	79 500	7.85
Hot rolled steel for hot coiled springs (all grades) to IS 3431 : 1982	206 000	78 500	7.85
Hot rolled steel for hot coiled springs (all grades) to IS 3195 : 1992	206 000	78 500	7.85

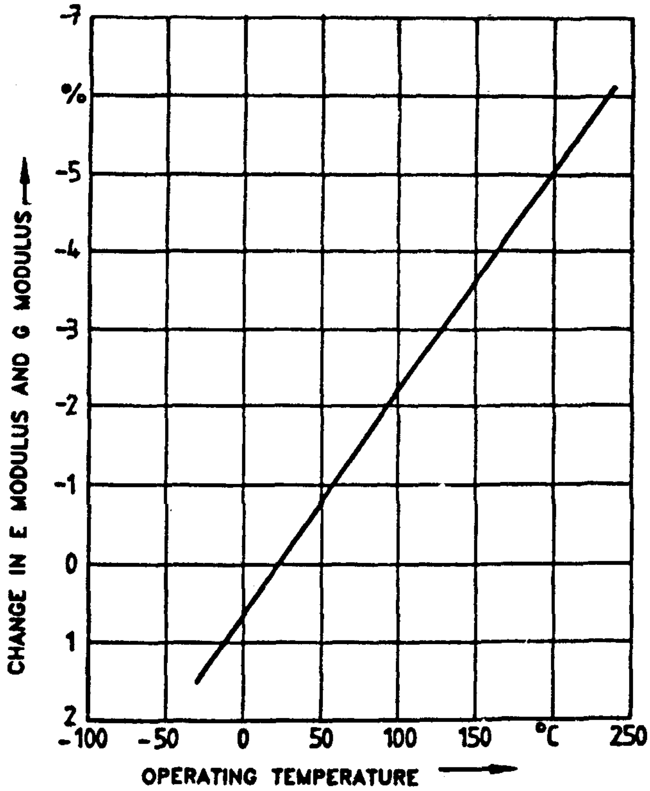

FIG. 4 MODULUS OF ELASTICITY AND SHEAR MODULUS AS A FUNCTION OF THE OPERATING TEMPERATURE : GUIDELINES VALUES AVERAGED FOR THE MATERIALS LISTED IN TABLE 1

8 COMPUTATIONAL FORMULAE

8.1 Work done in deflecting a spring under force '*F*' and associated deflection

$$W = \frac{1}{2} F . s \qquad \text{... (2)}$$

8.2 Spring Force (Resilience)

$$F = \frac{G}{8} \cdot \frac{d^4 . s}{D^3 . n} \qquad \text{... (3)}$$

8.3 Spring Deflection

$$s = \frac{8}{G} \cdot \frac{D^3 . n}{d^4} \cdot F \qquad ... (4)$$

8.4 Spring Rate

$$R = \frac{G}{8} \cdot \frac{d^4}{D^3 . n} \qquad ... (5)$$

8.5 Shear Stresses

$$\tau = \frac{8}{\pi} \cdot \frac{D}{d^3} \cdot F \qquad ... (6)$$

$$\tau = \frac{G}{\pi} \cdot \frac{d}{n . D^2} \cdot s \qquad ... (7)$$

$$\tau_k = k . T \qquad ... (8)$$

NOTE — For static or infrequently loaded spring 'τ' shall be used for evaluting the design of the spring.

For dynamically loaded springs 'τ_k' the corrected shear stress shall be calculated.

8.6 Diameter of Wire or Rod

$$d = \sqrt[3]{\frac{8}{\pi} \cdot \frac{F . D}{\tau_{zul}}} \qquad ... (9)$$

The permissible shear stress τ_{zul} shall be selected according to the design case applicable (*see* **9** for further details).

8.7 Number of Active Coils (n_t)

$$n = \frac{G}{8} \cdot \frac{d^4 . s}{D^3 . F} \qquad ... (10)$$

8.8 Total Number of Coils (n_t)

8.8.1 Total number of coils is the sum of active coils and inactive coils. The number of inactive coils depends upon the type of spring ends and manufacturing process (that is hot or cold coiled).

8.8.2 For cold coiled springs, **3.3** of IS 7906 (Part 2) : 1975 gives the type of spring ends. Generally two inactive colds are required for cold coiled springs with ends closed.

Thus for cold coiled springs

$$n_t = n + 2 \qquad ... (11)$$

8.8.3 For hot coiled springs **4.3** of IS 7906 (Part 5) : 1989 gives various types of spring ends, and **4.4** gives the relation between total turns and active turns. Generally for hot coiled springs having closed ends.

$$n_t = n + 1.5 \qquad ... (12)$$

For other types of ends, **4.4** of IS 7906 (Part 5) : 1989 shall be referred.

8.9 Minimum Space Between Active Coils

The minimum permitted test length (L_n) of a spring is given by $L_n = L_c + S_a$.

where S_a is the sum of the minimum gaps between adjoining active coils and is calculated as follows.

8.9.1 For cold coiled springs as per IS 7906 (Part 2) : 1975:

$$S_a = (0.015 \frac{D_2}{d} + 0.1\, d)\, n \qquad ... 13 (a)$$

In case the spring is subjected to dynamic loading then S_a as calculated above will be increased by factor of 1.5

8.9.2 For hot coiled springs as per IS 7906 (Part 5) : 1989:

$$S_a = 0.02\, (D + d)\, n \qquad ... 13 (b)$$

In case the spring is subjected to dynamic loading than S_a shall be doubled.

8.10 Solid Length (L_c)

8.10.1 For cold coiled springs, refer **3.3** of IS 7906 (Part 2) : 1975.

8.10.2 For hot coiled springs, refer **4.4** of IS 7906 (Part 5) : 1989.

8.11 Increase in Coil Diameter

When a helical compression spring is compressed, the coil diameter increases very slightly. This increase in coil diameter, ΔD_e, can be determined with the aid of the formula given below. This formula is based on the consideration that the spring is compressed to solid length L_c and is freely seated on the spring ends.

$$\Delta D_e = 0.1 \left[\frac{m^2 - 0.8\, m . d - 0.2\, d^2}{D} \right] \qquad ... (14)$$

where

$m = \frac{L_0 - d}{n}$ for springs with ends closed machined flat, and ground.

$m = \frac{L_0 - 2.5\, d}{n}$ for springs with unground ends

8.12 Natural Frequency

The "Natural frequency f_e" of a compression spring, guided at both ends and periodically compressed, can be calculated as follows:

$$f_e = \frac{3\,560\, d}{n\, D^2} \sqrt{\frac{G}{\rho}} \qquad ... (15)$$

8.13 Transverse Stability

8.13.1 Under the combined effect of coaxial and transverse load, a spring becomes deformed as illustrated in Fig. 5 below.

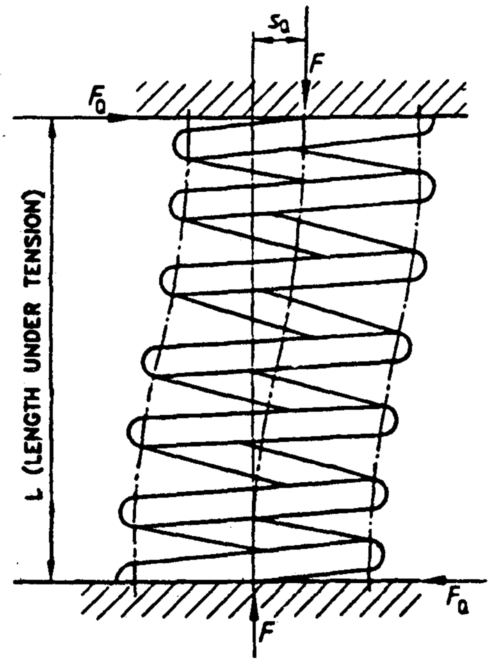

FIG. 5 HELICAL COMPRESSION SPRING UNDER SIMULTANEOUS AXIAL AND TRANSVERSE LOADING

8.13.2 Assuming that the spring ends do not lift off their seatings [*see* formula (23) for this condition], the following formulae shall be used for calculations.

Transverse spring rate:

$$R_Q = \frac{F_Q}{S_Q} \qquad \text{... (16)}$$

Transverse spring deflection :

$$S_q = \frac{F_Q}{R_Q} = \frac{F_Q}{\eta . R} \qquad \text{... (17)}$$

Spring rate ratio η is given by equation

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

where ... (18)

slenderness ratio, $\lambda = \frac{L_0}{D}$ ... (19)

and the relative spring deflection $\xi = \frac{S}{L_0}$ ... (20)

8.13.3 The transverse spring rate, R_Q, is only constant for short transverse spring deflections, for a given loaded height L. It varies with the seating conditions of the spring ends. In applications where the transverse stability of the spring is an important operational factor, the calculated values should be verified in actual practice.

8.13.4 Maximum shear stress considering the combined effect of axial and transverse load is :

$$\tau_{Max} = \frac{8}{\pi d^3} [F . (D + S_Q) + F_Q . (L - d)] \text{... (21)}$$

Maximum corrected shear stress:

$$\tau k_{Max} = k . \tau_{Max} \qquad \text{... (22)}$$

condition for the spring ends resting on their supports:

$$F_Q \frac{L}{2} \leq F . \frac{D - S_Q}{2} \qquad \text{... (23)}$$

8.14 Buckling

8.14.1 Helical compression springs have a tendency to buckle; the associated spring length is designated as buckling length L_k, and the spring deflection up to the point of buckling as buckling spring deflection S_k.

The influence of the seating of the spring ends is allowed for by means of seating coefficient ν, which is specified in Fig. 6 for the main types of seating which occur.

8.14.2 The following formula is applicable for calculating the buckling spring deflection S_K

$$S_K = L_0 \frac{0.5}{1 - \frac{G}{E}} \left[1 - \sqrt{1 - \frac{\frac{G}{E}}{0.5 + \frac{G}{E}} \cdot \left(\frac{\Pi . D}{\nu L_0} \right)^2} \right] \quad \text{... (24)}$$

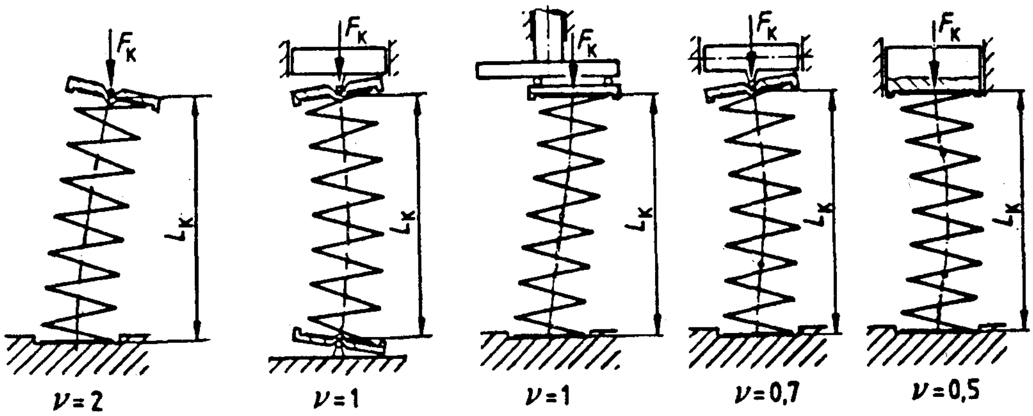

FIG. 6 TYPES OF SEATING AND ASSOCIATED SEATING COEFFICIENTS OF AXIALLY STRESSED HELICAL COMPRESSION SPRINGS

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

8.14.4 Safety against buckling can also be evaluated with the aid of the graph shown in Fig. 7.

8.15 Impact Stress

The increase in shear stress, τ_{st}, resulting from an impact can be calculated with the aid of formula (25) :

$$\tau_{st} = V_{st} . \sqrt{2 \times 10^{-3} . \rho . G} \quad \text{... (25)}$$

This formula does not take into account the reflection of the impact waves at the spring ends, and the effect of a possible striking of the coils against one another.

9 PERMISSIBLE STRESSES

9.1 Permissible Shear Stress at Solid Length

9.1.1 *Cold Coiled Compression Springs*

For manufacturing reasons, it shall be possible to compress all springs down to their solid length. The permissible shear stress at solid length τ_{czul}, shall be $0.56 R_m$. The value of R_m (minimum value of tensile strength) can be obtained from the relevant standards, for the relevant condition, that is hardened and tempered or patented condition.

9.1.2 *Hot Coiled Compression Springs*

Figure 8 shows the values of the permissible shear stress at solid length τ_{czul} for hot coiled springs. The actual stress shall not exceed the permissible shear stress at solid length, which is a function of the strength of the material, and of the wire/bar diameter. This shear stress is without the stress correction factor k.

9.2 Permissible Shear Stress Under Static or Infrequently Varying Load

9.2.1 In such cases, the permissible operating stress is limited by the relaxation which can be tolerated, depending on the specific application concerned.

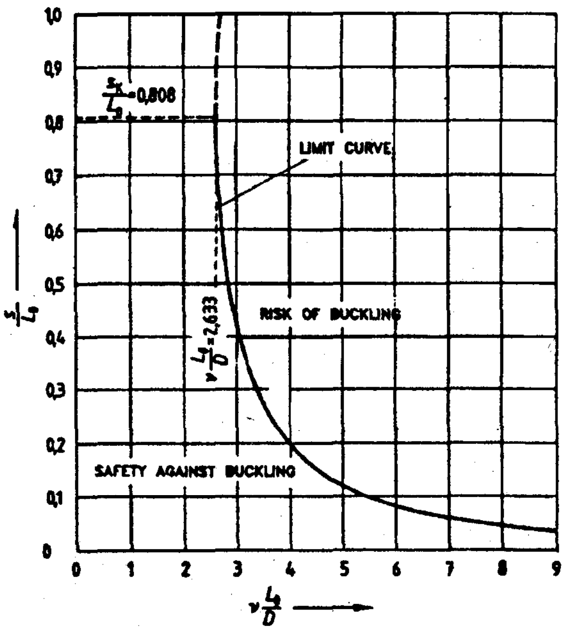

FIG. 7 THEORITICAL BUCKLING LIMIT OF HELICAL COMPRESSION SPRINGS

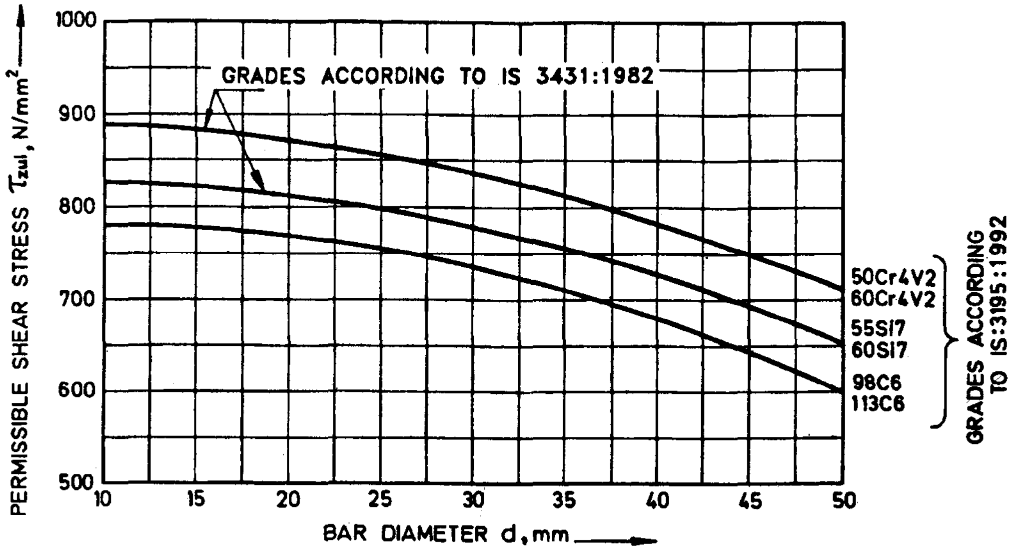

FIG. 8 PERMISSIBLE SHEAR STRESS τ_{czul} FOR HOT COILED HELICAL COMPRESSION SPRINGS MADE FROM STEEL CONFORMING TO IS 3431 : 1982 AND IS 3195 : 1992

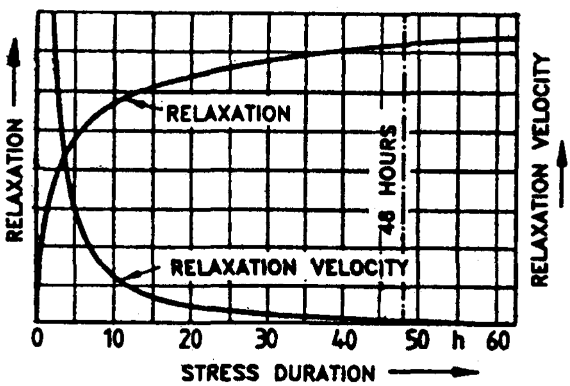

FIG. 9 COURSE OF RELAXATION AND OF RELAXATION VELOCITY OF HELICAL COMPRESSION SPRINGS AS A FUNCTION OF TIME

9.2.1.1 Relaxation is a loss of force at constant clamping length, which is dependent on stress, temperature and time, and in this standard it is represented in the form of a percentage loss related to the initial values. This shall only be verified in cases where requirements have been specified with regard to the constancy of the spring force.

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

9.2.3 Figure 9 illustrates the pattern of the relaxation and of the relaxation velocity in principle. The relaxation values after 48 hours should be regarded as characteristic values, despite the fact that the relaxation is not fully completed at this point in time.

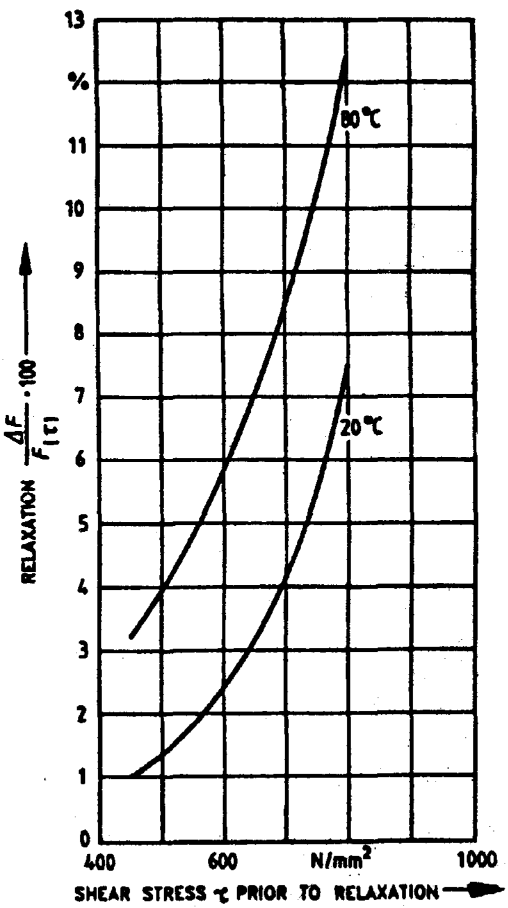

FIG. 10 RELAXATION AFTER 48 HOURS OF HOT COILED HELICAL COMPRESSION SPRINGS MADE FROM STEEL SPECIFIED IN IS 3195 : 1992 AND IS 3431 : 1982 HAVING A STRENGTH DUE TO HEAT TREATMENT OF 1500 N/mm², PRESET AT AMBIENT TEMPERATURE, AS A FUNCTION OF THE OPERATING STRESS AT VARIOUS TEMPERATURES

9.2.4 Data on relaxation is shown in Fig. 10 for hot coiled helical compression springs and in Fig. 11 (a) 11 (b), 11 (c), 11 (d), 11 (e) and 11 (f) for cold coiled helical compression springs as a function of the stress prior to relaxation, and of the operating temperature.

These graphs represent guiding values, and are based on conventional production methods with seragging at ambient temperature. These values can be influenced favourably by using appropriate materials and by pre-setting at higher temperatures and improving the heat treatment for strength.

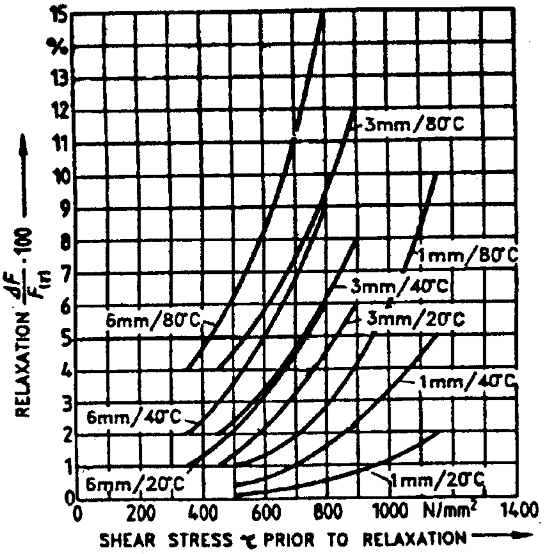

FIG. 11 (a) RELAXATION AFTER 48 HOURS OF COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADES 3 AND 4 WIRE SPECIFIED IN IS 4454 (Part 1) : 1981 PRESET AT AMBIENT TEMPERATURE, AS A FUNCTION OF SHEAR STRESS τ PRIOR TO RELAXATION AT VARIOUS TEMPERATURES, IN °C, AND FOR THE FOLLOWING DIAMETERS OF NON-SHOT- PEENED WIRE :

WIRE DIAMETER : 1 mm, TENSILE STRENGTH : 2 540 N/mm^2

WIRE DIAMETER : 3 mm, TENSILE STRENGTH : 1 970 N/mm^2

WIRE DIAMETER: 6 mm, TENSILE STRENGTH : 1 640 N/mm^2

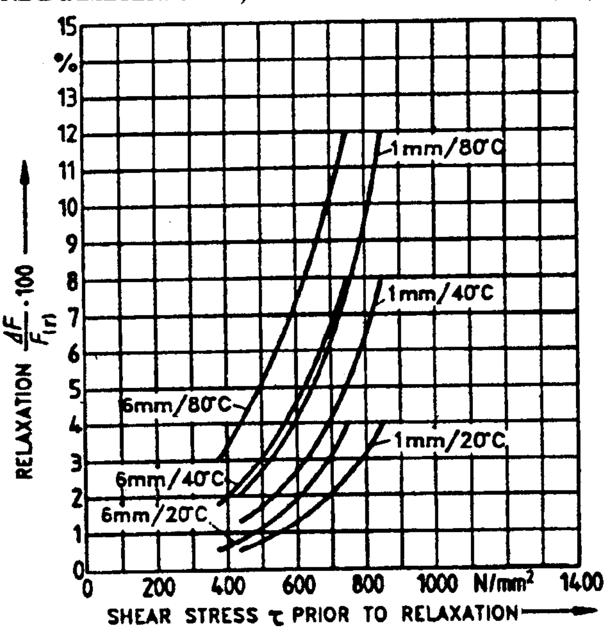

FIG. 11 (b) RELAXATION AFTER 48 HOURS OF COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADE VW WIRE (VALVE SPRING WIRE) AND FROM SW GRADE WIRE (SPRING WIRE) SPECIFIED IN IS 4454 (Part 2) : 1975 PRESET AT AMBIENT TEMPERATURE, AS A FUNCTION OF SHEAR STRESS τ PRIOR TO RELAXATION AT VARIOUS TEMPERATURES, IN °C, AND FOR THE FOLLOWING DIAMETERS OF NON-SHOT-PEENED WIRE:

WIRE DIAMETER : 1 mm, TENSILE STRENGTH : 1 725 N/mm^2

WIRE DIAMETER : 6 mm, TENSILE STRENGTH : 1 560 N/mm^2

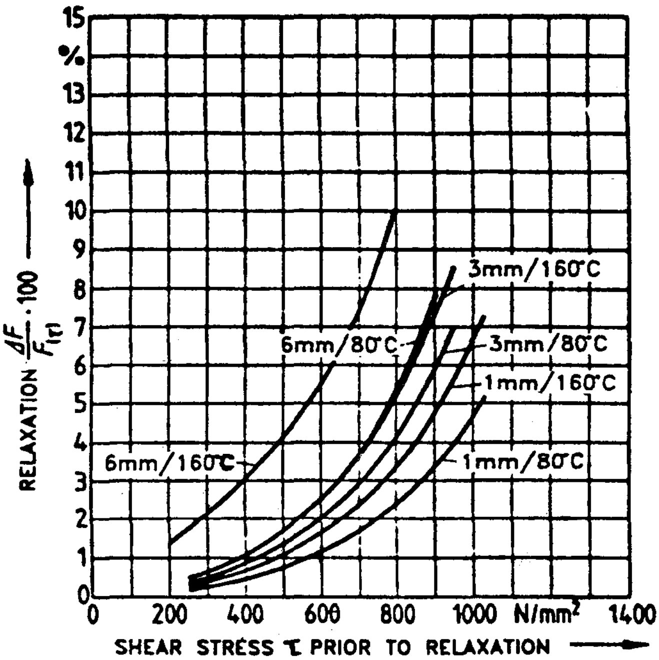

FIG. 11 (c) RELAXATION AFTER 48 HOURS OF COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADE 2S AND 2D OF IS 4454 (PART 3) : 1975, PRESET AT AMBIENT TEMPERATURE, AS A FUNCTION OF SHEAR STRESS τ, PRIOR TO RELAXATION AT VARIOUS TEMPERATURES, IN °C, AND FOR THE FOLLOWING DIAMETERS OF NON-SHOT-PEENED WIRE:
WIRE DIAMETER : 1 mm TENSILE STRENGTH : 2 100 N/mm^2
WIRE DIAMETER : 3 mm TENSILE STRENGTH : 1 950 N/mm^2
WIRE DIAMETER : 6 mm TENSILE STRENGTH : 1 800 N/mm^2

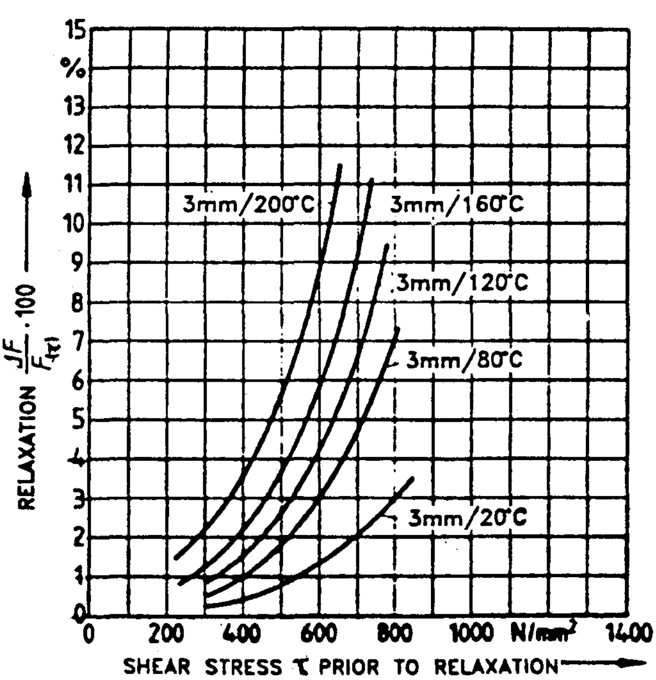

FIG. 11 (d) RELAXATION AFTER 48 HOURS OF COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADE 1S, AND 1D TO IS 4454 (Part 3) : 1975, PRESET AT AMBIENT TEMPERATURE, AS A FUNCTION OF SHEAR STRESS τ, PRIOR TO RELAXATION AT VARIOUS TEMPERATURES, IN °C, AND FOR NON-SHOT-PEENED WIRE OF 3 mm IN DIAMETER AND A TENSILE STRENGTH OF 1 700 N/mm^2

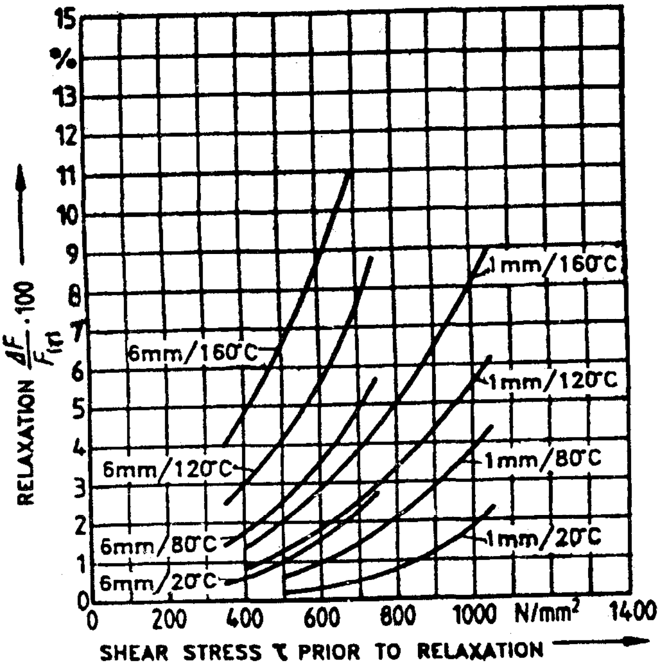

FIG. 11 (e) RELAXATION AFTER 48 HOURS OF COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADE 1 TO IS 4454 (Part 4) : 1975, PRESET AT AMBIENT TEMPERATURE, AS A FUNCTION OF SHEAR STRESS τ PRIOR TO RELAXATION AT VARIOUS TEMPERATURES, IN °C, AND FOR THE FOLLOWING DIAMETERS OF NON-SHOT-PEENED WIRE
WIRE DIAMETER : 1 mm TENSILE STRENGTH : 2 000 N/mm²
WIRE DIAMETER : 6 mm TENSILE STRENGTH : 1 500 N/mm²

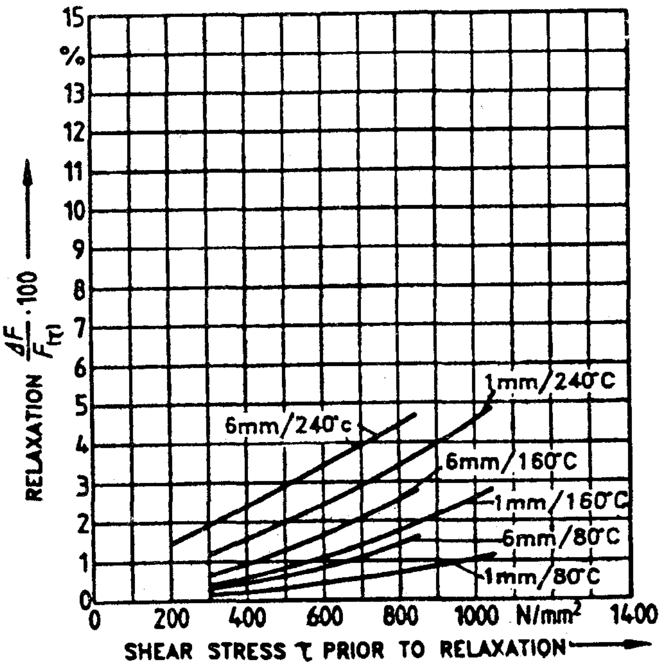

FIG. 11 (f) RELAXATION AFTER 48 HOURS OF COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADE 2 TO IS 4454 (Part 4) : 1975, PRESET AT AMBIENT TEMPERATURE, AS A FUNCTION OF SHEAR STREES τ PRIOR TO RELAXATION AT VARIOUS TEMPERATURES, IN °C, AND FOR THE FOLLOWING DIAMETERS OF NON-SHOT-PEENED WIRE:
WIRE DIAMETER : 1 mm TENSILE STRENGTH : 2 100 N/mm²
WIRE DIAMETER: 6 mm TENSILE STRENGTH : 1 650 N/mm²

9.3 Permissible Stress Under Dynamic Loading

9.3.1 In case of dynamically loaded springs, the permissible amplitude of stress is limited by the required number of load cycles and by the given rod or wire diameter.

9.3.2 Stress correction factor *k* shall be taken into consideration in the calculation. Stress amplitude τ_{kh} as illustrated in Fig. 12 is the difference between τ_{k1} and τ_{k2}.

9.3.3 For a given value of $\tau_{k1} = \tau_{ku}$, τ_{k2} shall not exceed τ_{KO} that is stress amplitude τ_{kh} for the desired working stroke of the compression spring shall not exceed the value of the creep or fatigue stroke strength T_{kH} which can be obtained from Figures 13 to 24. In the case of dynamically stressed helical compression springs, the solid length stress τ_{czul} shall also be verified taking into account the possibility that additional stresses may be superimposed as a result of natural oscillations of the spring body.

All dynamically stressed springs should be shot-peened. Shot-peening is feasible, as a general rule, in respect of helical compression springs with a wire diameter, *d*, exceeding < 1 mm, a spring index, ω, of less than 15 and space between adjoining coils, a_o, greater than *d*.

9.3.4 The values given in the following strength diagrams are not applicable to springs operating under the influence of corrosion or friction.

10 EXAMPLES OF SPRING DESIGN CALCULATIONS

10.1 Example of spring design calculation of compression spring subjected to static or infrequently varying load are given in Annex A.

10.2 Examples of spring design calculation of a compression spring subjected to dynamic load is given in Annex B.

11 DESIGN OF VARIABLE RATE SPRINGS

Variable rate springs often improve the working characteristics of an assembly. There are two methods outlined in 11.1 and 11.2.

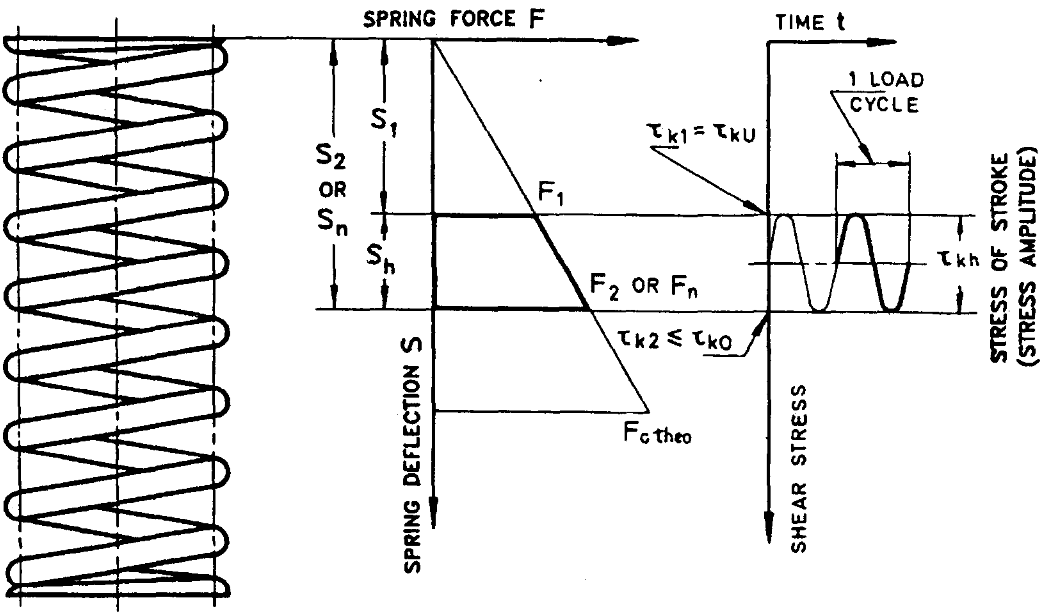

FIG. 12 OSCILLATION DIAGRAM OF A HELICAL COMPRESSION SPRING SUBJECTED TO OSCILLATING STRESS

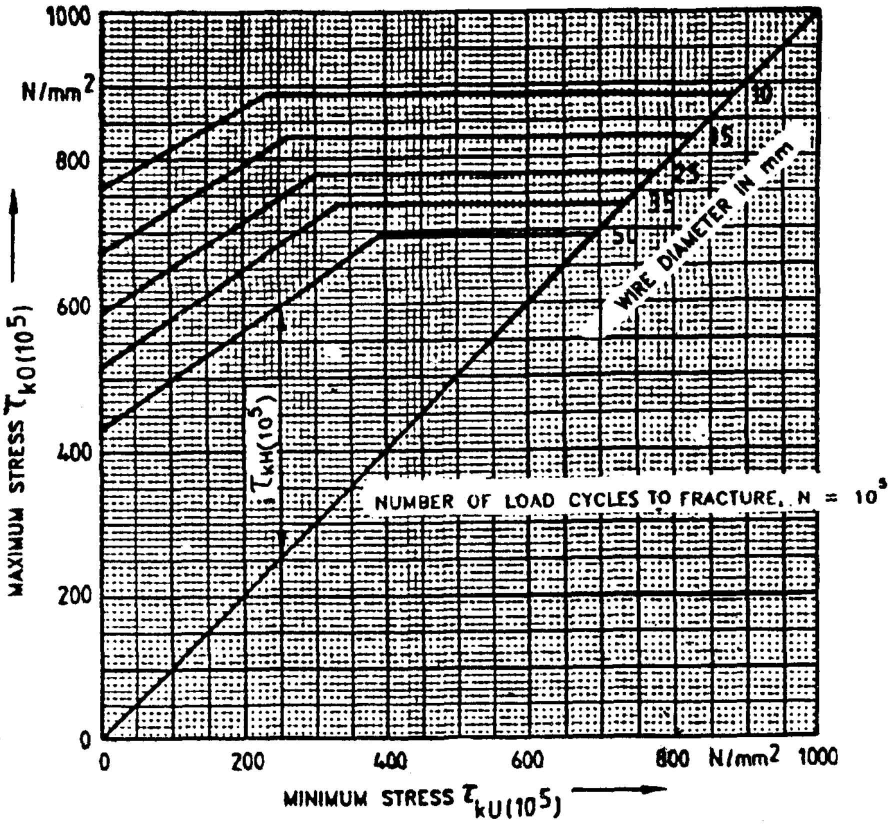

FIG. 13 CREEP STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR HOT COILED HELICAL COMPRESSION SPRINGS MADE FROM HIGH GRADE STEEL SPECIFIED IN IS 3195 : 1992/IS 3431 : 1982 WITH GROUND OR BRIGHT TURNED SURFACE, SHOT-PEENED

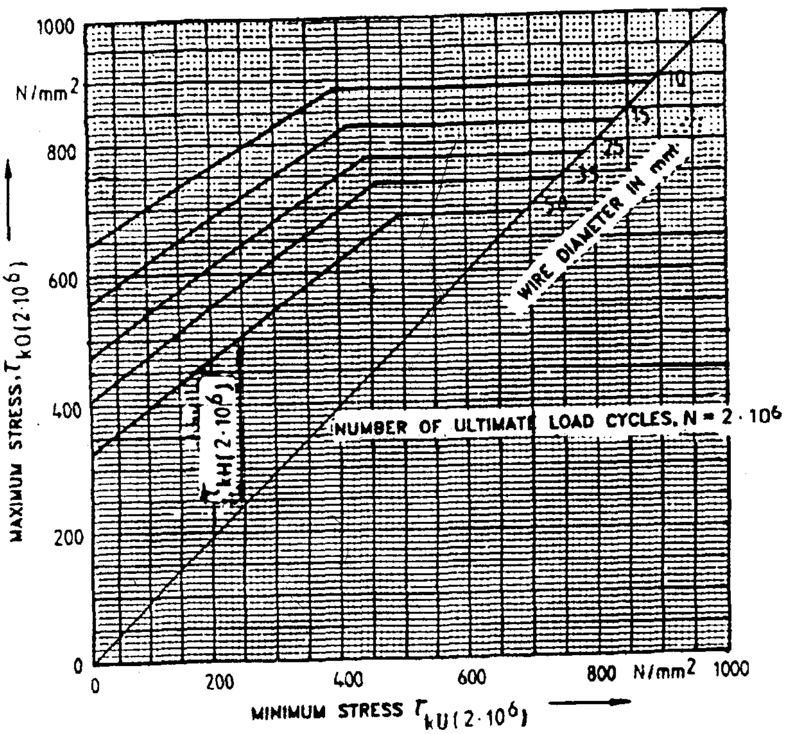

FIG. 14 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR HOT COILED HELICAL COMPRESSION SPRINGS MADE FROM HIGH GRADE STEEL SPECIFIED IN IS 3195 : 1992/IS 3431 : 1982 WITH GROUND OR BRIGHT TURNED SURFACE, SHOT-PEENED

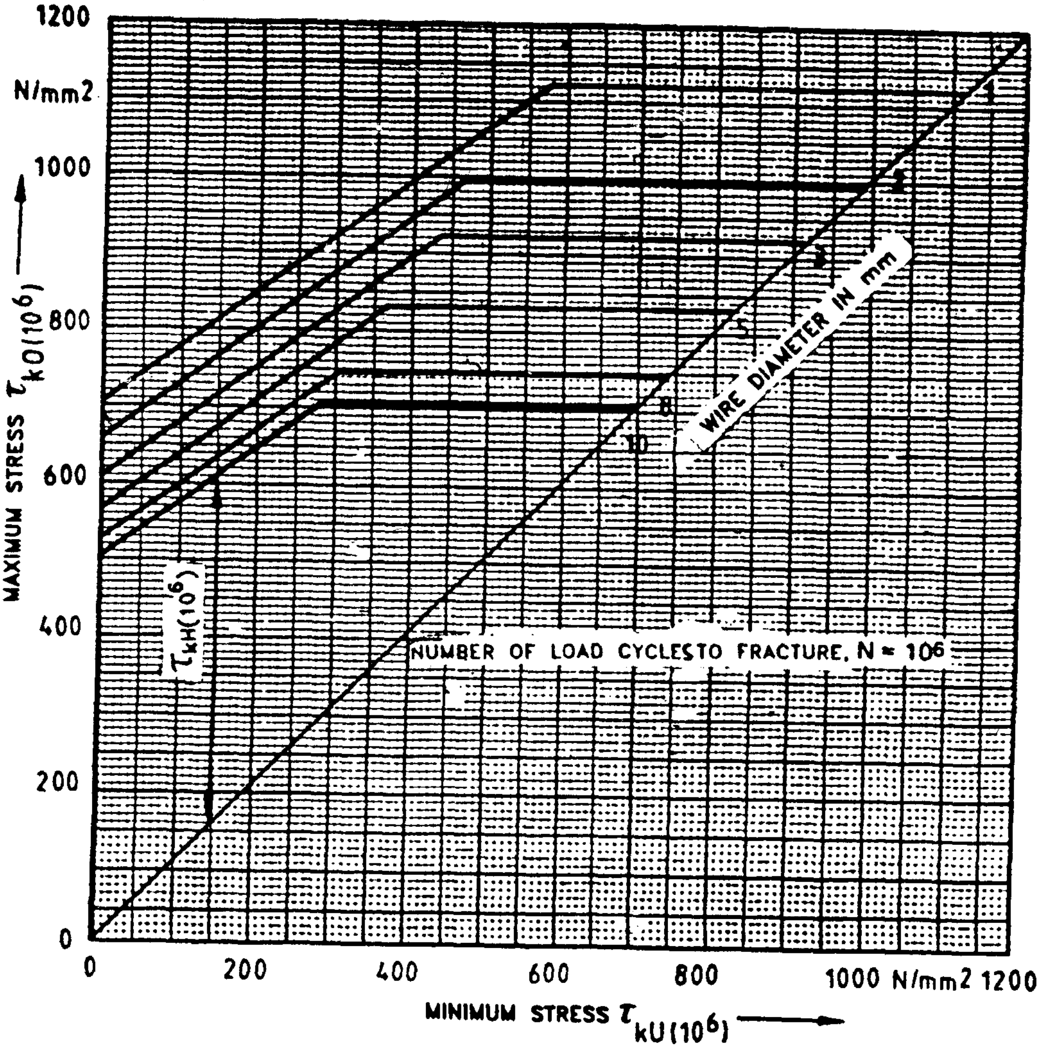

FIG. 15 CREEP STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADE 3 AND 4 PATENTED DRAWN SPRING STEEL WIRE SPECIFIED IN IS 4454 (Part 1) : 1981 SHOT-PEENED

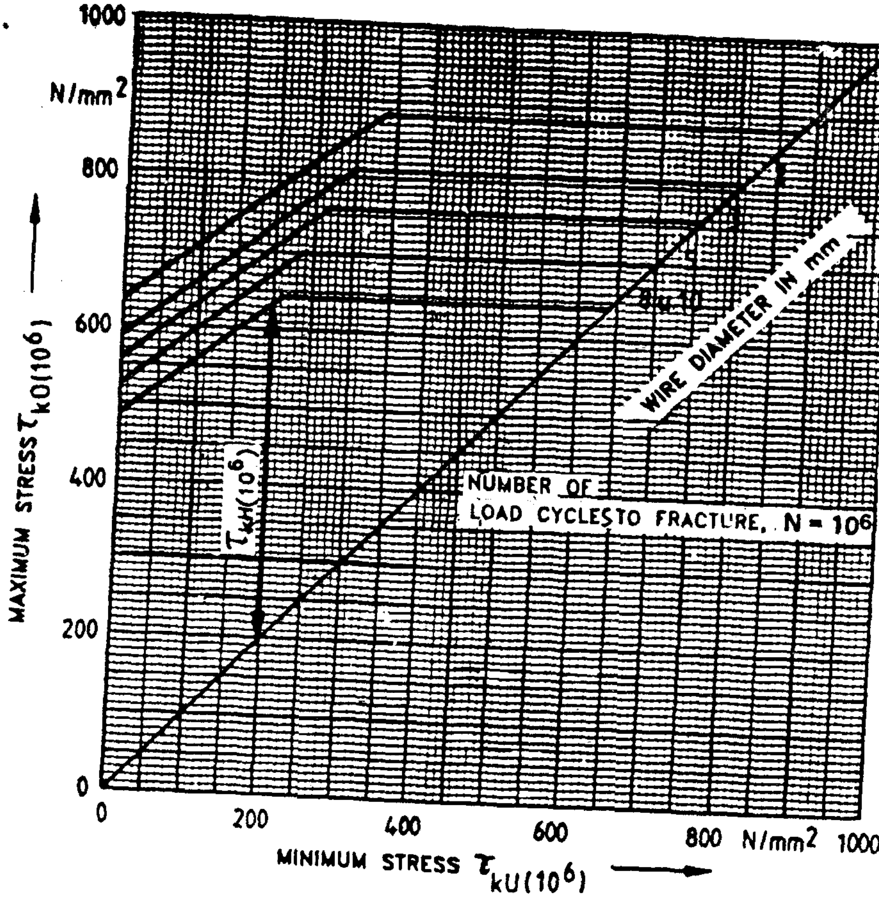

FIG. 16 CREEP STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM QUENCHED AND TEMPERED SPRING WIRE SPECIFIED IN IS 4454 (Part 2) : 1975 SHOT-PEENED

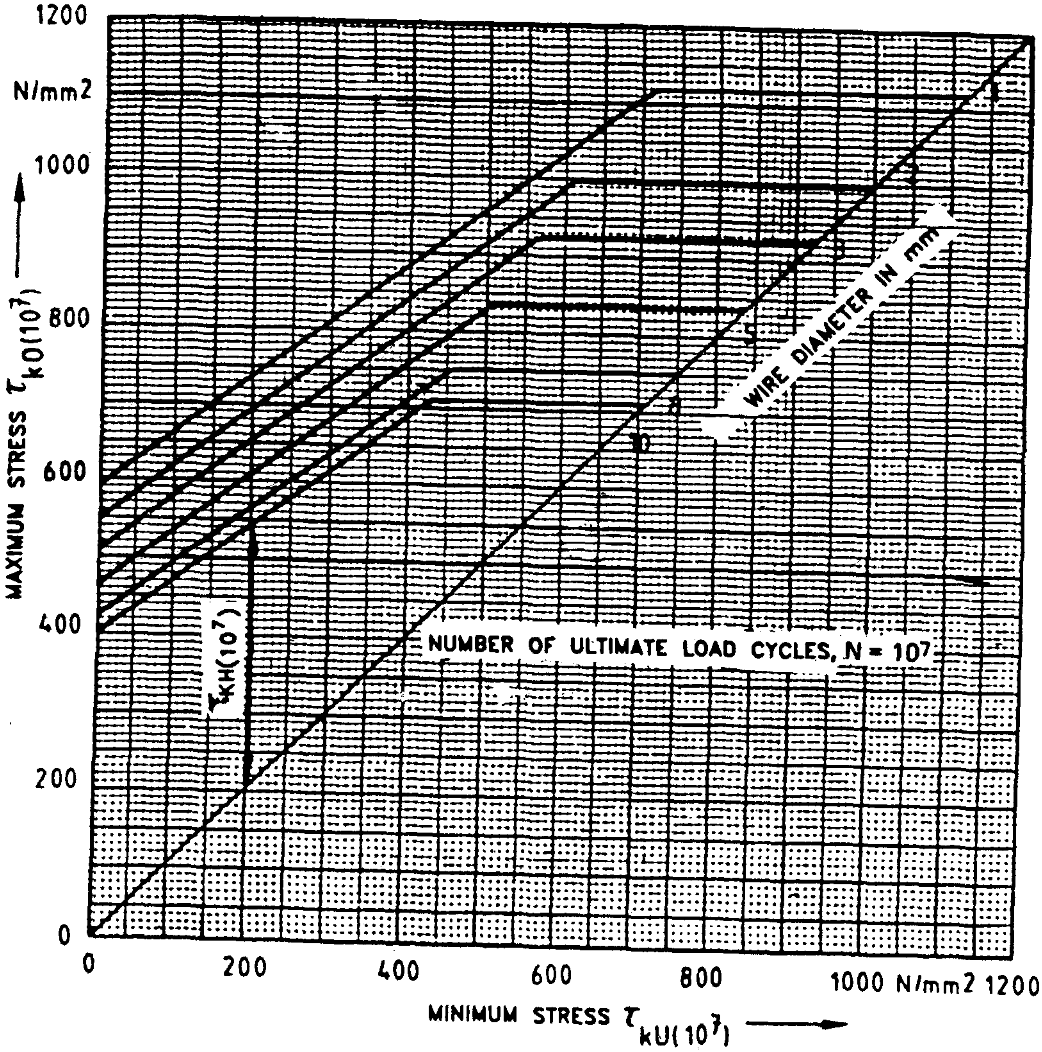

FIG. 17 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADE 3 AND 4 PATENTED DRAWN SPRING STEEL WIRE SPECIFIED IN IS 4454 (Part 1) : 1981 SHOT-PEENED

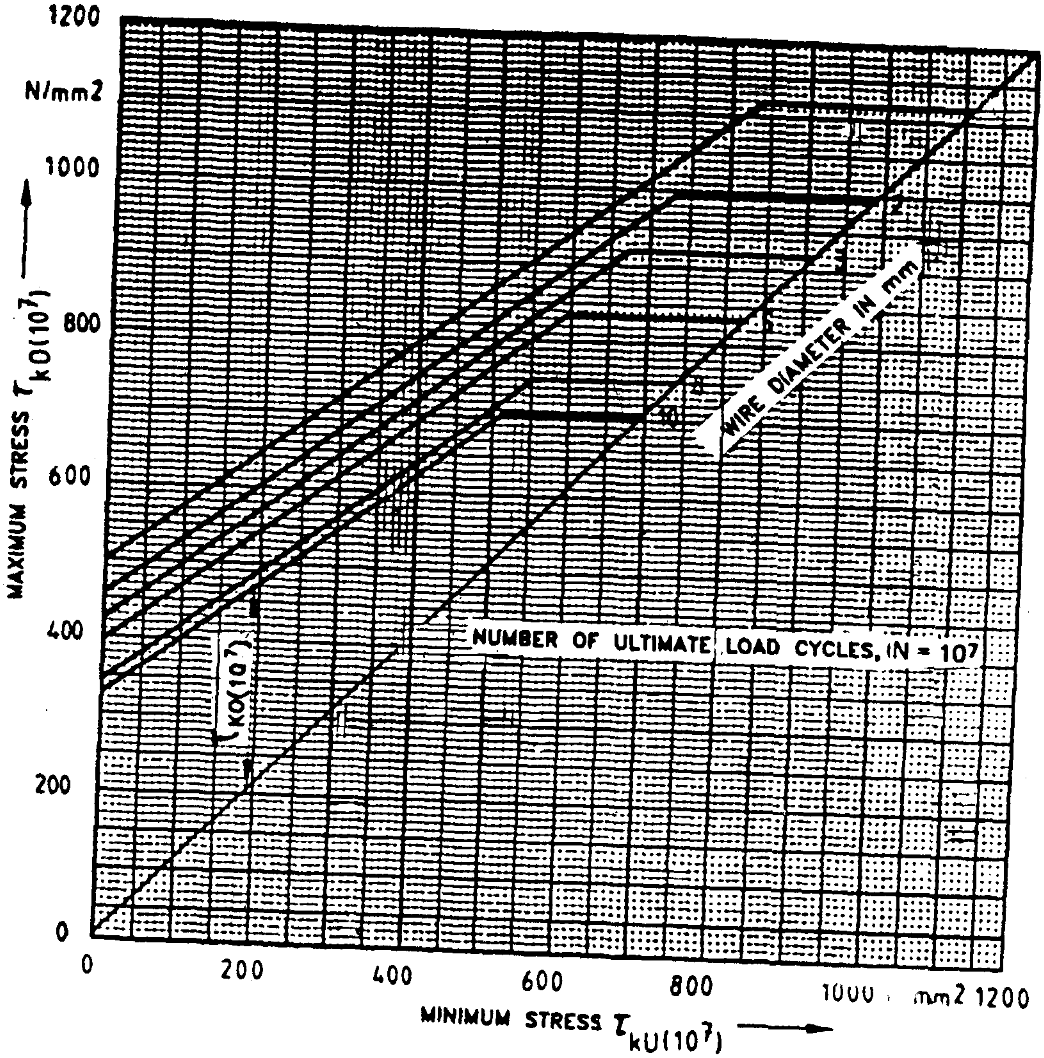

FIG. 18 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM GRADE 3 AND 4 PATENTED DRAWN SPRING STEEL WIRE SPECIFIED IN IS 4454 (Part 1) : 1981 NOT SHOT-PEENED

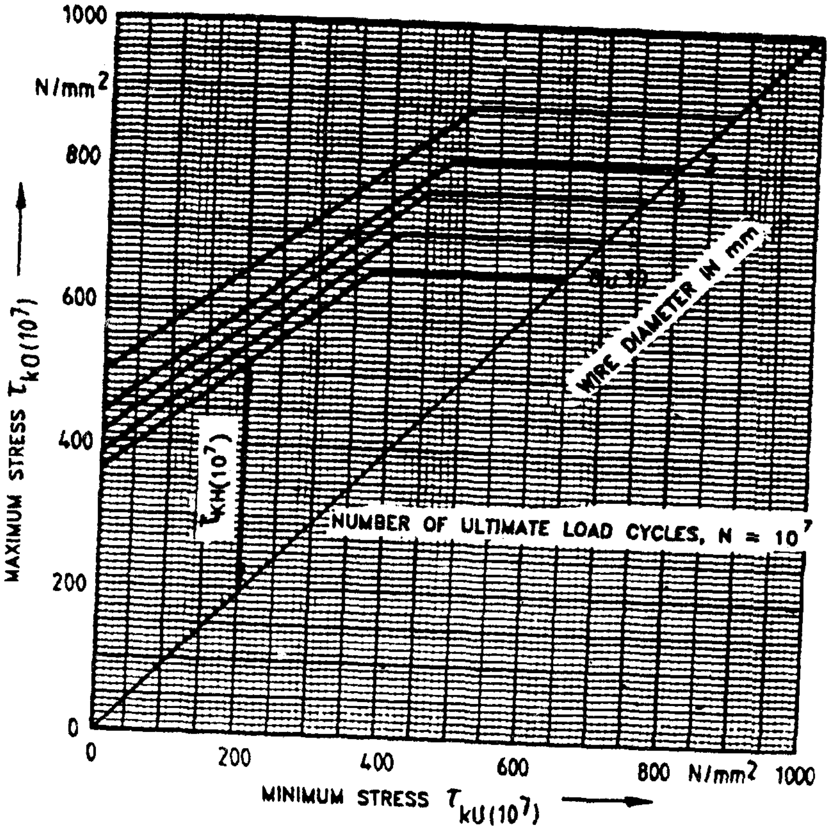

FIG. 19 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM QUENCHED AND TEMPERED SPRING WIRE SPECIFIED IN IS 4454 (Part 2) : 1975 SHOT-PEENED

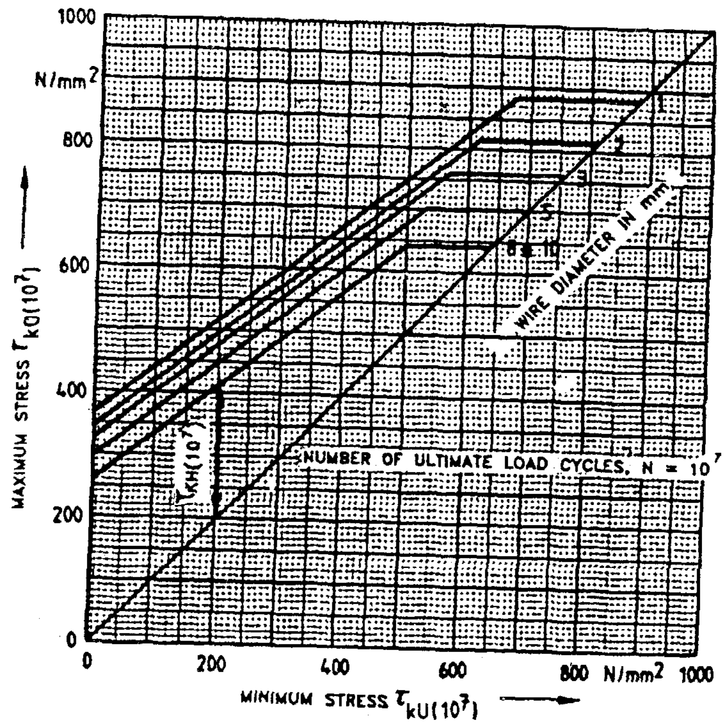

FIG. 20 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM QUENCHED AND TEMPERED SPRING WIRE SPECIFIED IN IS 4454 (Part 2) : 1975 NOT SHOT-PEENED

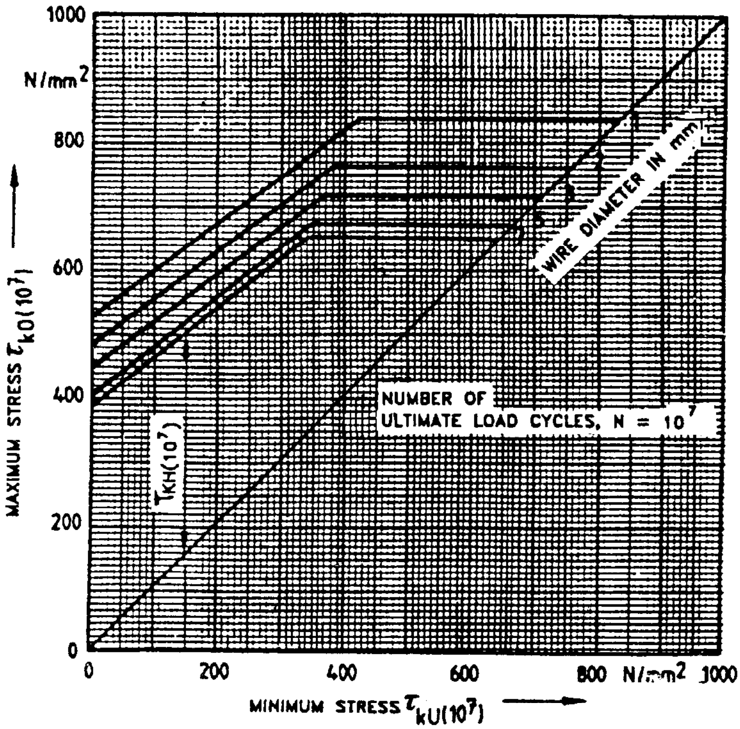

FIG. 21 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM QUENCHED AND TEMPERED VALVE SPRING WIRE SPECIFIED IN IS 4454 (Part 2) : 1975 NOT SHOT-PEENED

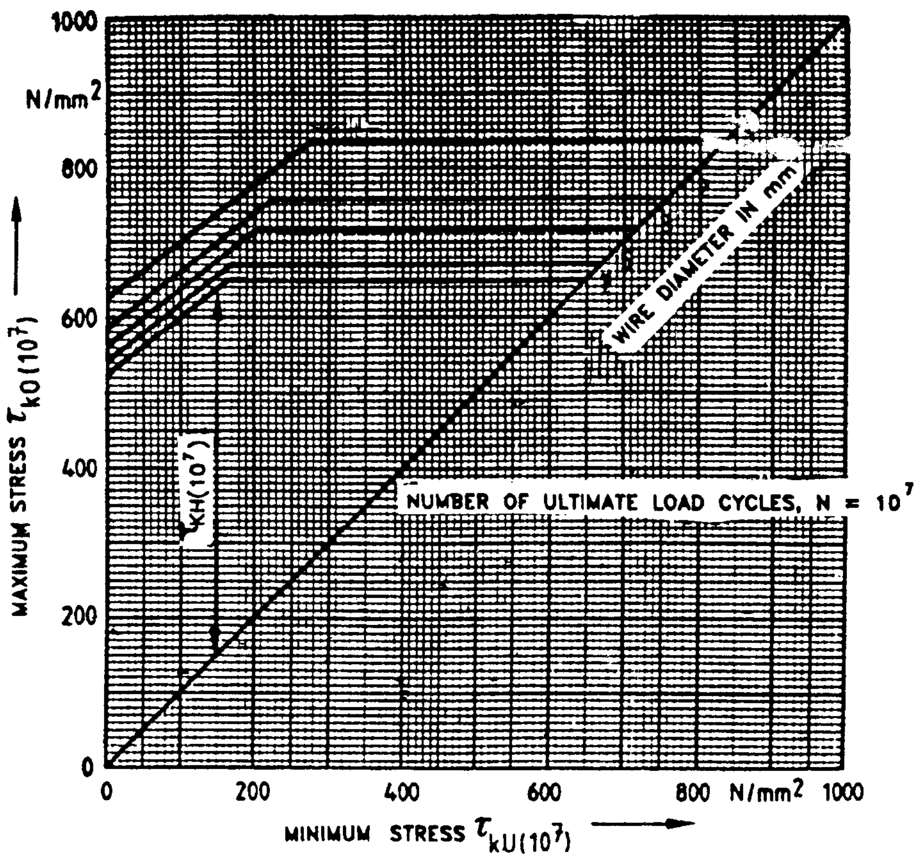

FIG. 22 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM QUENCHED AND TEMPERED VALVE SPRING WIRE SPECIFIED IN IS 4454 (Part 2) : 1975 NOT SHOT-PEENED

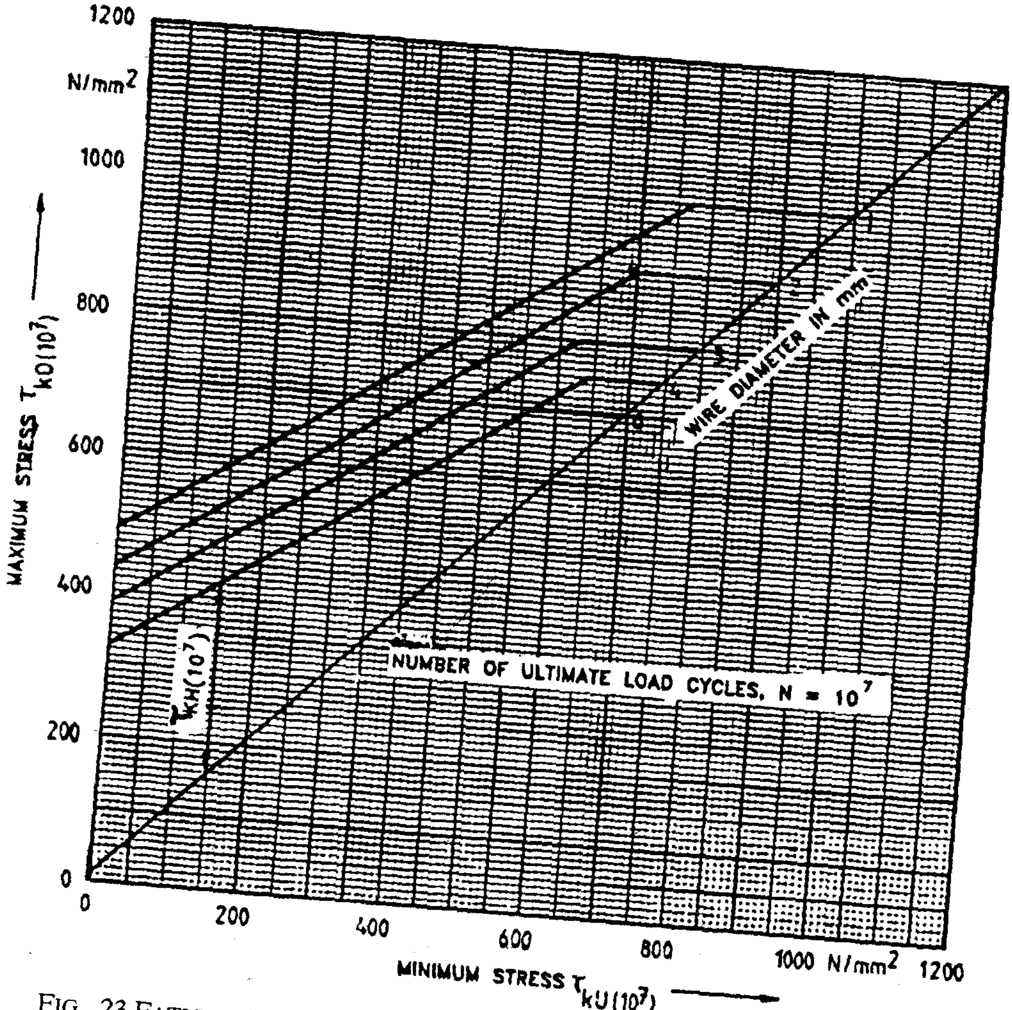

FIG. 23 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM STAINLESS SPRING STEEL WIRE GRADE 1 SPECIFIED IN IS 4454 (Part 4) : 1975 NOT SHOT-PEENED

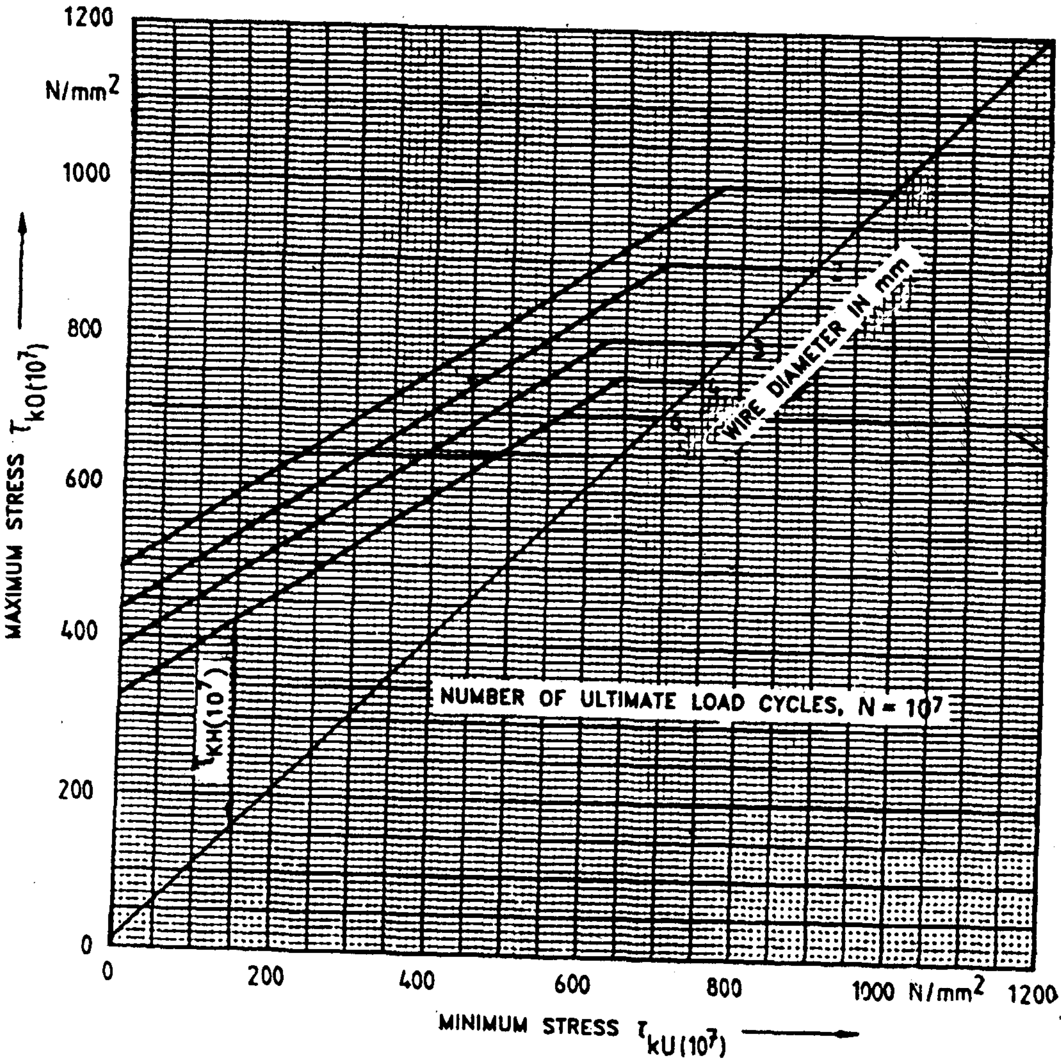

FIG. 24 FATIGUE STRENGTH DIAGRAM (GOODMAN DIAGRAM) FOR COLD COILED HELICAL COMPRESSION SPRINGS MADE FROM STAINLESS SPRING STEEL WIRE GRADE 2 SPECIFIED IN IS 4454 (Part 4) : 1975 NOT SHOT-PEENED

11.1 Preloading of Springs in Series

This types of loading gives decreasing spring rate. This could be obtained by having springs with the same or different spring rates in series. The total spring rate obtained is less than either of the two spring rates alone., If R_1 and R_2 are the spring rates for two springs, the total spring rate R is given by:

$$\frac{1}{R_t} = \frac{1}{R_1} + \frac{1}{R_2}$$

$$\text{or } R_t = \frac{R_1 \times R_2}{R_1 + R_2}$$

11.1.1 In such cases preloads for both the springs should be specified.

11.1.2 Number of Working Coils

$$n_1 = \frac{G \times (d_1)^4}{8 \times (D_1)^3 \times R_1}$$

$$\text{and} \quad n_2 = \frac{G \times (d_2)^4}{8 \times (D_2)^3 \times R_2}$$

11.1.3 All the other formulae given in this standard shall apply.

11.2 Two springs, each with different pitch are combined in such a fashion that the coils of one bottom out and become inactive after undergoing a certain amount of deflection. This method gives increasing spring rate. The springs may or may not be preloaded, but preloading is advisable. In practice only one spring is used with a varying pitch in coils (*see* Fig. 25)

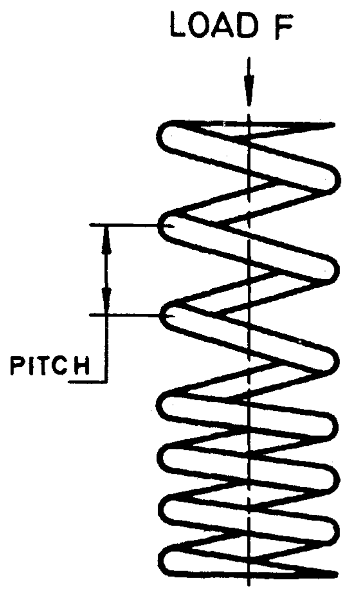

FIG. 25 VARIABLE PITCH SPRING

11.2.1 All the other formulae given in this standard shall apply.

12 NOMOGRAMS FOR PRELIMINARY DESIGN CALCULATION

Nomograms as shown in Fig. 26 and 27 for helical springs embody the wire or bar diameter d, the shear stress τ, the load F and the mean coil diameter D in the relationship given by Equation (6). The two nomograms are an aid to be used in preliminary design calculations *see* examples given in 12.2. It is not possible to make an exact calculation of spring data with their aid.

12.1 Nomogram as shown in Fig. 26 is intended for the preliminary calculation of the smaller sizes of spring and nomogram as shown in Fig. 27 for the preliminary calculation of the larger sizes.

12.2 The two inner scales for F and D and the two outer scales for τ and d are always used together in pairs. Each pair of scales is joined by a straight line. The straight line through F and D and the straight line through τ and d must intersect the pivot line in the same point.

Example 1 (*see* Fig. 26)

Given load F = 1 850 N, mean coil diameter D = 60 mm

Find wire diameter d and shear stress τ

Join the point for F = 1 850 with the point for line on the point of intersection of the first line with the pivot line. The second line gives the shear stress τ for various values of d. The result obtained for $d \approx 8$ mm is $\tau \approx 570$ N/mm^2.

Example 2 (*see* Fig. 27)

Given load F = 31 800 N, mean coil diameter, D = 130 mm and τ = 700 N/mm^2.

Find wire or bar diameter d.

The straight line joining the point for F = 31 800 N with the point for D = 130 mm intersects the pivot line in a point through which a second straight line is drawn with one end passing through the point for τ = 700 N/mm^2 The point of intersection of this straight line with the d scale gives d = 25 mm.

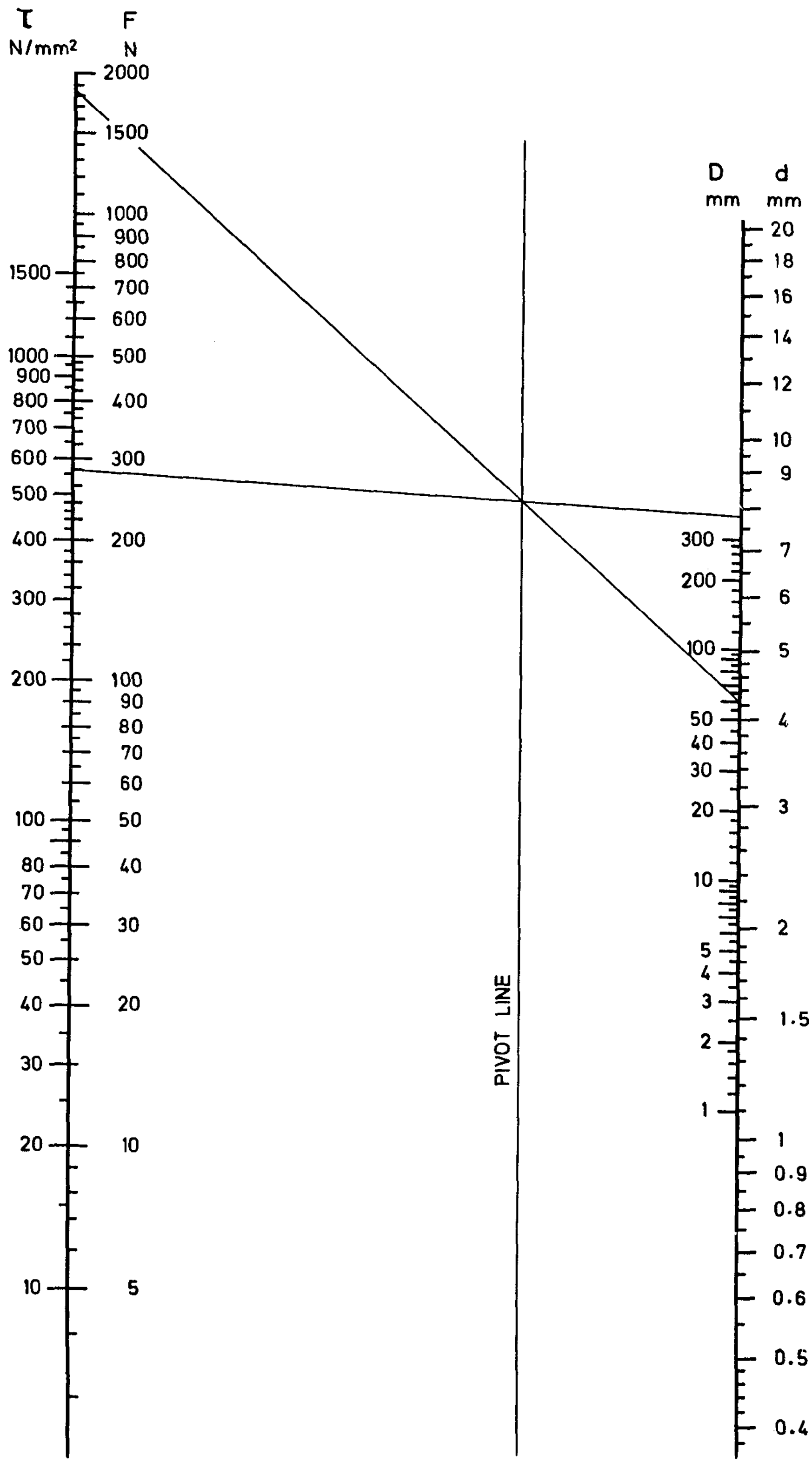

FIG. 26 NOMOGRAM FOR PRELIMINARY DESIGN CALCULATION OF SMALL SIZE SPRINGS

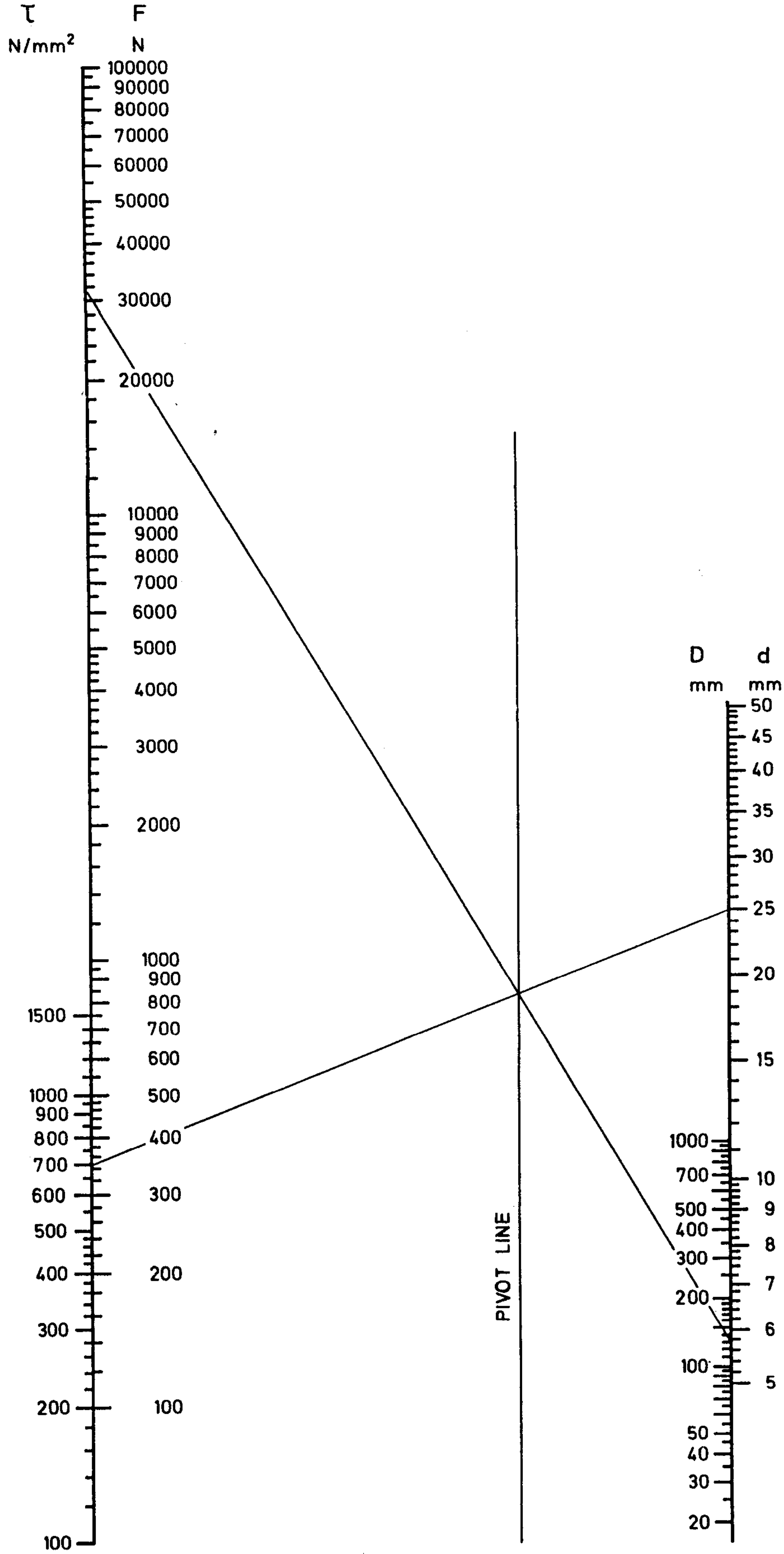

FIG. 27 NOMOGRAM FOR PRELIMINARY DESIGN CALCULATIONS OF LARGE SIZE SPRINGS

ANNEX A

(*Clause* 10.1)

EXAMPLES OF DESIGN OF COMPRESSION SPRINGS SUBJECTED TO A STATIC OR INFREQUENTLY VARYING LOAD

A-1 DESIGN OF A COLD COILED COMPRESSION SPRING SUBJECTED TO A STATIC OR INFREQUENTLY VARYING LOAD

A compression spring is required for a load F_n = 1 850 N and a deflection S_n = 90 mm. The mounting space has an inside diameter of 72 mm. There are no special requirements to be met by the spring.

It is thus necessary to determine the requisite spring data, that is d, n, L_o, τ and the spring material.

A-1.1 Wire Diameter, d

From Equation (9)

$$d = \sqrt[3]{\frac{8F.D}{\pi.T}} = \sqrt[3]{\frac{8F_n.D}{\pi.Tzul}}$$

In Equation (9) τ has to be replaced by the permissible shear stress τ_{zul} the values of which can be taken from **9** or the nomogram (*see* **12**) as follows:

Using the nomogram as shown in Fig. 26 draw a straight line joining the specified load F_n = 1 850 N and the diameter D = 60 mm which is assumed to be in agreement with the space available. This line cuts the pivot line at a point which serves as pivot for a second straight line. By rotating this second line, coordinate values of the shear stress τ and the diameter d can be found. Diameter d = 8 mm, for example, yields τ = 570 N/mm^2.

According to **9**, spring steel wire of Grade 2, IS 4454 (Part 1) : 1981 with τ_{zul} = 620 N/mm^2 is suitable.

$$\text{Hence } d = \sqrt[3]{\frac{8 \times 1\,850 \times 60}{\pi \times 620}} = 7.7 \text{ mm}$$

In fact the next larger wire diameter according to IS 4454 (Part 1) : 1975, namely d = 8 mm is selected.

A-1.2 Number of Active Coils, n

From Equation (10) :

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

Taking G = 81 500 N/mm^2 the following result is obtained:

$$n = \frac{81\,500}{8} \times \frac{(8)^4 \times 90}{(60)^3 \times 1\,850} = 9.4$$

Which is rounded off to 9.5 so that n = 9.5.

Owing to the increase of n = 9.5, F_n is reduced inversely proportional to 1 830 N.

A-1.3 Total Number of Coils

According to IS 7906 (Part 2) : 1975:

$$n_t = n+2 = 9.5 + 2 = 11.5$$
$$n_t = 11.5$$

A-1.4 Sum S_a of Minimum Spaces Between Individual Working Coils

From **8.9**

$$\omega = \frac{D}{d} = \frac{60}{8} = 7.5$$

Putting the values in Equation 13(a),

$$S_a = 13.3 \text{ mm}$$

Which is rounded to 14 so that S_a = 14 mm.

A-1.5 Solid Length, L_c

From IS 7906 (Part 2) : 1975, the solid length of a spring with ground ends is:

$$L_c = n_t \times d_{Max} = 11.5 \times 8.06 = 92.7 \text{ mm}$$

A-1.6 Length L_n Under Load

$$L_n = L_c + S_a = 92.7 + 14 = 106.7 \text{ mm}$$

A-1.7 Unloaded Length L_o

$$L_o = L_n + f_n = 106.7 + 90 = 196.7 \text{ mm}$$

A-1.8 Checking the Shear Stress, τ_{kn}

From Equation (6):

$$\tau_{kn} = \frac{8.D.F_n}{\pi.d^3} = \frac{8 \times (60) \times 1\,830}{\pi \times (8)^3}$$

$$= 547 \text{ N/mm}^2$$

A-1.9 Checking the Shear Stress, T_c

From Equation (6):

$$T_c = \frac{8 \times D.F_c}{\pi.d^3}$$

The theoretical load F_c can be found by means of the ratio:

$$\frac{F_c}{F_n} = \frac{S_c}{f_n}$$

Putting $S_c = f_n + S_a$ = 90 + 14 = 104 mm, gives

$$F_c \text{ theo} = \frac{F_n \times S_c}{f_n} = \frac{1\,830 \times 104}{90} = 2\,115 \text{ N}$$

$$\text{Hence } \tau_c = \frac{8 \times 60 \times 2\,115}{\pi \times (8)^3} = 631 \text{ N/mm}^2$$

A-1.10 Buckling Resistance (Fig. 7)

$$\xi = \text{Relative deflection} = \frac{\text{Deflections } S}{\text{Length } L_0}$$

The value of S to be inserted in this expression is the one corresponding to the maximum allowable load, namely S_n

ξ Relative deflection

$$= \frac{S_n}{L_0} = \frac{90}{196.7} = 0.458$$

$$\text{Degree of slenderness} = \frac{L_0}{D} = \frac{196.7}{60} = 3.28$$

Taking $\nu = 1$

The point having these two coordinates in Fig. 3 lies below curve. The spring is therefore buckle-proof.

A-1.11 Summary of Spring Data

The essential spring data sheet be entered in the data sheet given in IS 7906 (Part 3) : 1975.

A-2 DESIGN OF A HOT COILED COMPRESSION SPRING SUBJECTED TO A STATIC OR INFREQUENTLY VARYING LOAD

A compression spring is required for a load $F_2 = F_n = 26\,500$ N at a deflection $f_2 = f_n = 120$ mm. The mean coil diameter D is required to be 130 mm. The load sustained by the spring alternates between a variable pre-load F_1 and the load F_2. The number of load cycles between F_1 and F_2 in 24 hours does not exceed 50. The loading involved is therefore of the infrequently varying type. It is thus necessary to find the requisite spring data, that is d, n, L_o, τ and the spring material.

A-2.1 Bar Diameter, *d*

From Equation (9):

$$d = \sqrt[3]{\frac{8F.D}{\pi\tau}} = \sqrt[3]{\frac{8F_c \text{theo}.D}{\pi.\tau_{zul}}}$$

In Equation (9) the permissible shear stress τ_{zul} from Fig. 8 shall be substituted for τ. As τ_{zul} in Fig. 8 pertains to the solid length the theoretical load F_c shall be substituted for F in Equation (9). Since F_c is not known yet it is assumed to be 20 percent greater than F_n

$$F_c = 1.2\ F_n$$

For the time being, the quantity S_a = sum of minimum spaces will be left out of account. Not before the diameter d and the number of working coils n are known, the exact value of the theoretical load F_c which depends on S_a can be found and used for calculating the shear stress τ_c.

As a first approximation for the permissible shear stress the value $\tau_{zul} = 750$ N/mm^2 may be inserted in Equation (9). This is an average value for hot coiled compression springs having bar diameters up to 35 mm and made of Grades 2 and 3 steel to IS 3431 : 1982.

$$\text{Hence } d = \sqrt[3]{\frac{8 \times 1.2 F_n \times D}{\pi \times \tau_{zul}}}$$

$$= \sqrt[3]{\frac{8 \times 1.2 \times 26\,500 \times 130}{\pi \times 750}}$$

$$= 24.3$$

In fact, the next larger bar diameter according to IS 3431 : 1982 namely $d = 25$ mm, is selected. With the aid of the nomogram (*see* **12**) the above equation can also be solved without calculation. For example, by taking a point on the F scale representing the load $F_c = 1.2 \times 26\,500 = 31\,800$ N and drawing a straight line to the point representing 130 mm on the D, scale it is found that a second straight line drawn from the point $\tau = 750$ N/mm^2 and the passing through the same point of intersection on the pivot line yields a diameter $d = 25$ mm.

A-2.2 Number of Working Coils, *n*

From Equation (10):

$$n = \frac{G.d^4.S_2}{8.(D)^3 F_2}$$

Taking $G = 78\,500$ N/mm^2 (according to Table 1) the following result is obtained:

$$n = \frac{78\,500 \times (25)^4 \times 120}{8 \times (130)^3 \times 26\,500} = 8.05$$

which is rounded to $n = 8$.

Generally speaking it is not necessary for the number of working coils to be rounded to greater accuracy than 1/4 of a coil. In accordance with IS 7906 (Part 5) : 1989 however, the aim should be to make the total number of coils and in 1/2. The result should therefore be rounded, where necessary, to a full working coil, since according to IS 7906 (Part 5) : 1989 the difference between the total number of coils and the number of working coils should be 1.5. When the number of coils is small, this practice of rounding makes a check calculation necessary.

A-2.3 Total Number Of Coils, n_t

From IS 7906 (Part 5) : 1989

$$n_t = n + 1.5 = 8 + 1.5 = 9.5$$

A-2.4 Sum S_a of Minimum Spaces Between Individual Working Coils

From **8.9**

$$\omega = \frac{D}{d} = \frac{130}{25} = 5.2 \text{ and}$$

$$S_a = 20 \text{ mm}$$

A-2.5 Solid Length, L_c

From IS 7906 (Part 5) : 1989 the solid length of springs made from rolled bars is:

$$L_c \leq (n_t - 0.3) \times d_{Max}$$
$$\leq (9.5 - 0.3) \times 25.35$$
$$\leq 233.2 \text{ mm}$$

A-2.6 Load Length, L_n

$$L_n = L_2 = L_c + S_a = 230 + 20 = 250 \text{ mm}$$

A-2.7 Unloaded Length, L_o

$$L_o = L_2 + S_2 = 250 + 120 = 370 \text{ mm}$$

A-2.8 Checking Shear Stress, τ_c

From Equation (6):

$$= \frac{8 . D . F_c}{\pi . d^3}$$

The theoretical load F_c at solid length L_c can be derived from the ratio:

$$\frac{F_c}{F_2} = \frac{S_c}{f_2}$$

Putting $S_c = S_2 + S_a = 120 + 20 = 140$ mm, gives

$$F_c = \frac{F_2 . S_c}{f_2} = \frac{26\,500 \times 140}{120} = 31\,000 \text{ N}$$

$$\text{Hence } \tau_c = \frac{8 \times 130}{\pi \times (25)^3} \times 31\,000 = 657 \text{ N/mm}^2$$

This value of shear stress is permitted by Fig. 8.

In the event of the shear stress τ_c exceeding the τ_{zul} values permitted by Fig. 8 for Grades 2 and 3 steel, the designer shall decide whether Grades 2 and 3 is necessary for economic reasons or whether Grade 1 steel may be specified. In the first case a recalculation is needed in order to lower the shear stress τ_c in the interest of spring life it is very often advisable to specify Grade 1 steel. For hot coiled springs made of bar having diameters above 40 mm it is recommended that only Grade 1 steel should be specified.

A-2.9 Buckling Resistance (Fig. 7)

$$\xi = \text{Relative deflection} = \frac{\text{Deflections } s}{\text{length } L_0}$$

The value of deflection S to be inserted in this expression is the one corresponding to the maximum allowable load, namely S_n

$$\text{Relative deflection } \xi = \frac{S_n}{L_0} = \frac{120}{384} = 0.312$$

$$\text{Degree of slenderness} = \frac{L_0}{D} = \frac{120}{130} = 2.95$$

Taking $\nu = 1$

The point of intersection of these two values in Fig. 7 lies below-curve. The spring is thus buckle proof.

A-2.10 Summary of Spring Data

The data comprising the spring specification shall be entered in the form according to IS 7906 (Part 3) : 1975.

ANNEX B

(*Clause* 10.2)

EXAMPLE OF DESIGN OF A COLD COILED COMPRESSION SPRING SUBJECTED TO LOADING

B-1 A compression spring is required with virtually infinite life for loads $F_1 = 300$ and $F_2 = F_n = 650$ N at a stoke $S = 14$ mm. The mounting space has an inside diameter of 37 mm. The load cycle frequency $N = G.\ H_z$.

It is thus necessary to determine the requisite spring data, that is d, n, L_o, R_{kh} and the spring material

B-1.1 Approximate Determination of τ_{k1}, τ_{k2}, and τ_{kh}

The calculation shall start with the required stroke S and the corresponding stress range τ_{kh}. The latter shall not be greater than the stress range for fatigue strength τ_{kh} of the chosen material. In addition, the shear stress τ_{k2} should not be greater than the upper limit of fatigue strength τ_{ko}.

B-1.1.1 The spring concerned is required to have virtually infinite life which means that the number of load cycles sustained is not less than 10 million, and therefore oil hardened and tempered valve spring wire Grade VW to IS 4454 (Part 2) : 1975 which has been shot peened is selected (*see* Fig. 19). The following relationship applies (*see* Fig. 12):

$$\frac{\tau_{k1}}{S_1} = \frac{\tau_{k2}}{S_2}$$

The deflection S_1 is obtained from the relationship:

$$\frac{F_1}{S_1} = \frac{F_2 - F_1}{S_h}$$

$$S_1 = \frac{F_1 . S_h}{F_2 . F_1} = \frac{300 \times 14}{350} = 12 \text{ mm}$$

$$S_2 = S_1 + S_h = 12 + 14 = 26 \text{ mm}$$

The value of τ_{k2} needed for the calculation is obtained from the fatigue strength diagram (*see* Fig. 19) at τ_{ko}. Since the wire diameter d is not known at this stage, it is necessary for the time being to insert an estimated figure for τ_{k2} = 650 N/mm².

The relationship between f_1 and f_2 stated above gives

$$\tau_{k1} = \frac{s_1}{s_2} \cdot \tau_{k2} = \frac{12}{26} \times 650 = 300 \text{ N/mm}^2$$

Hence $\tau_{kh} = \tau_{k2} - \tau_{k1} = 650 - 300 = 350$ N/mm²
These stresses range within the limits of Fig. 19.

B-1.1.2 *Wire Diameter, d*

From Equation (9) :

$$d = \sqrt[3]{k \cdot \frac{8 . F . D}{\pi . \tau_k}}$$

$$= \sqrt[3]{k \cdot \frac{8 . (F_2 - F_1) . D}{\pi . \tau_{kh}}}$$

In Equation (9) the stress range τ_{kh} is substituted for τ_k. Similarly, the corresponding load difference ($F_2 - F_1$) shall be substituted for F. The mean coil diameter D is already fixed approximately by the specified mounting space which has a diameter of 37 mm. Taking an estimated figure of 4.5 mm for the wire diameter and allowing for a variation of about 1 mm in this figure, a mean coil diameter of D = 31 mm can be assumed. An approximate value for d can also be found as a function of τ_{k2} with the aid of the nomogram.

For the stress correction factor k the provisional figure obtained from Fig. 3 by using the estimated values:

$$\omega = \frac{D}{d} = \frac{31}{4.5} = 6.9 \text{ is } k = 1.2$$

Equation (9) therefore yields the following result:

$$d = \sqrt[3]{1.2 \times \frac{8 \times 350 \times 31}{\pi \times 350}} = 4.54 \text{ mm}$$

The value actually chosen is d = 4.5 mm.

B-1.1.3 *Number of Working Coils, n*

From Equation (10):

$$n = \frac{G . d^4 . s_2}{8 . (D)^3 . F_2}$$

Taking G = 81 500 N/mm² (according to Table 1) the following result is obtained:

$$n = \frac{81\,500}{8} \times \frac{(4.5)^4 \times 26}{(31)^3 \times 650} = 5.95 = 6.0$$

B-1.1.4 *Total Number of Coils, n_t*

From IS 7906 (Part 2) : 1975 (*see* **8.8**) it is seen that:

$$n_t = n + 2 = 6.0 + 2 = 8.0$$

B-1.1.5 *Sum S_a of Minimum Spaces Between Individual Working Coils*

From **8.9**

$\omega = 6.9$ $\qquad$ S_a = 4.32 mm

The value actually chosen is S_a = 4.5 mm

B-1.1.6 *Block Length, L_c*

From IS 7906 (Part 2) : 1975 (*see* **8.10**) the block length when the spring ends are ground is:

$$L_c \leq n_t \times d_{Max} = 8 \times 4.54 = 36.3 \text{ mm}$$

B-1.1.7 *Lengths L_2, L_1 and L_o*

$$L_2 = L_n = L_c + S_a = 36.3 + 4.5 = 40.8 \text{ mm}$$

$$L_1 = L_2 + S = 40.8 + 14 = 54.8 \text{ mm}$$

$$L_o = L_1 + f_1 = 54.8 + 12 = 66.8 \text{ mm}$$

B-1.1.8 *Checking the Shear Stress, τ_c*

From Equation (6) :

$$\tau_c = \frac{8 . D . F_c}{d^3}$$

The theoretical load F_c at block length s_c can be derived from the relationship:

$$\frac{F_c}{F_n} = \frac{S_c}{S_n}$$

Putting $S_c = S_n + S_a = 26 + 4.5 = 30.5$ mm

$$F_c = \frac{F_n . S_c}{S_h} = \frac{650 \times 30.5}{26} = 763 \text{ N}$$

$$\text{Hence } \tau_c = \frac{8 \times 31 \times 763}{(4.5)^3} = 662 \text{ N/mm}^2$$

Figure gives the permissible shear stress as τ_{zul} = 700 N/mm²

B-1.1.9 *Checking the Shear Stresses τ_{k1}, τ_{k2} and τ_{kh}*

From Equation (8) :

$$\tau_k = k \frac{8 . D . F}{d^3}$$

$$= 1.2 \times \frac{8 \times 31}{(4.5)^3} \times F = 1.04 \times F, \text{ N/mm}^2$$

$$\tau_{k1} = 1.04 \times F_1 = 1.04 \times 300 = 312 \text{ N/mm}^2$$

$$\tau_{k2} = 1.04 \times F_2 = 1.04 \times 630 = 676 \text{ N/mm}^2$$

$$\tau_{kh} = \tau_{k2} - \tau_{k1} = 676 - 312 = 364 \text{ N/mm}^2$$

The above value are permitted by Fig. 19. The first calculation performed does not always yield the desired and permitted results. In such cases the calculation shall be repeated by starting from different assumptions.

B-1.1.10 *Resistance to Buckling* (*see* Fig. 7)

$$\xi \text{ Relative deflection } = \frac{\text{Deflection } s}{\text{Length } L_0}$$

The value of S to be inserted in this expression is that deflection which corresponds to the maximum load occurring namely S_2.

$$\text{Relative deflection } = \frac{S_2}{L_0} = \frac{26}{66.8} = 0.38$$

$$\text{Degree of slenderness } = \frac{L_0}{D} = \frac{66.8}{31} = 2.15$$

Taking $\nu = 1$

The point of intersection of these two values in Fig. 7 lies below curve. The spring is thus buckle-proof.

B-1.1.11 *Summary of Spring Data*

The data comprising the spring specification shall be entered in the form according to IS 7906 (Part 3) : 1975.

B-2 RECALCULATION OF A COLD COILED COMPRESSION SPRING TO ALTERNATING LOADING

B-2.0 Given: Compression spring with d = 4.5 mm, 31.0 mm,

L_o = 74 mm, n = 8.5, nt = 10.5, Ends closed and ground

At F_1 = 100 N, f_1 = 6 mm,

At F_2 = 334 N, f_2 = 20 mm,

Stroke S = 14 mm.

Spring shot — peened.

Material — hardened and tempered valve spring wire, Grade VW to IS 4454 (Part 2) : 1975.

B-2.1 The corresponding shear stresses are τ_{k1} = 104 N/mm^2 and τ_{k2} = 347 N/mm^2. The stress range $\tau_{kh} = \tau_{k2} - \tau_{k1}$ = 374 − 104 = 243 N/mm^2.
Figure 19 represents the fatigue strength diagram for shot peened springs made of hardened and tempered valve spring wire. The calculated stress range τ_{kh} = 243 N/mm^2 is shown in this diagram.

It is seen that maximum possible stress range for fatigue strength is not fully exploited. To determine the maximum possible stress range for fatigue strength τ_{kH} it is first necessary to calculate the ratio of the loads F_1 and F_2. This is found to be:

$$\frac{F_1}{F_2} = \frac{100}{334} = 0.3$$

The corresponding 0.3 line is indicated in Fig. 19. Its point of intersection with the interpolated line representing the stress range for fatigue strength for a wire diameter of 4.5 mm gives τ_{ko} as 710 N/mm^2. Vertically below this line τ_{ku} is found to be 215 N/mm^2.

From these values the maximum possible stress range for fatigue strength is calculated as:

$$\tau_{kH} = \tau_{ko} - \tau_{ku} = 710 - 215 = 495 \text{ N/mm}^2.$$

This check shows that the same conditions (F_1, F_2, S and D) could be met by using a lighter spring and therefore hence one which is an improvement from the economic viewpoint. Calculation on the lines of the example in **B-1** would yield the following new data d = 3.5 mm, L_o = 43.25 mm, n = 3.13 rounded to 3.25, n_t = 5.25.

The smaller number of coils now obtained would mean an inadmissibly high shear stress T_c on loading to the solid length L_c if the free length L_o were to be left unchanged at 74 mm. This is why the shorter length L_o = 43 mm is needed.

(*Continued from second cover*)

The main requirements to be satisfied in the design of helical springs are maximum possible dependability and life combined with lowest possible weight and cost. The expenditure of time and effort involved in the calculation procedure also needs to be reduced as much as possible. Nomograms have been included at the end of the standard, which can be used as a rapid method of arriving at provisional figures.

In spring calculations, a distinction is made between springs subjected to a static or infrequently varying load and springs subjected to dynamic loads (*see* **5**).

The cross section of the wire or bar of which a helical spring is made, is stressed mainly in torsion. In the calculations, therefore, the shear stresses likely to occur under load are compared with the permissible shear stresses. The stressing imposed on the spring, consisting in fact of an overwhelmingly torsional element and a negligibly small bending element, is more severe on the inside of the coil than on the outside. This maximum stress is taken into account in the calculation by introducing stress correction factor k. Tests have shown, however, that stress correction factor k can be omitted in calculations concerned with springs subjected to a static or infrequently varying load.

If the shear stresses occurring in the spring remain below the permissible range of stress or below the fatigue strength values in the case of springs subjected to alternating load, the spring will not fall or become fatigued within the envisaged life.

The permissible shear stresses and fatigue strength for cold and hot coiled springs have been included for spring materials meeting general requirements.

In this standard the unit of force used is newton (N) and that for stress is N/mm^2.

1 kgf = 9.806 65 N (exactly)

or 1 kgf $\approx$ 9.81 N (approximately)

$\approx$ 10 N (within 2 percent error)

1 N/mm^2 = 1 MN/m^2.

= 1 MPa [1 pascal] (Pa) = 1N/m^2

$\approx$ 0.1 kgf/mm^2 (within 2 percent error)

In the preparation of this standard, considerable assistance has been derived from DIN 2089 Part 1 : 1984 'Helical compression spring made from round wire or rod, calculation and design', issued by DIN Deutsches Institute für Normung, Germany.