

Physical Properties

Young's Modulus

Vulcanized rubber is a solid, three-dimensional network. The more crosslinks there are in the network the greater is the resistance to deformation when a force is applied. Certain fillers, notably the reinforcing blacks, create a structure within the rubber which further resists deformation.

The load-deflexion curves for rubber in tension and compression are approximately linear for strains of the order of a few per cent, and values of Young's modulus E_0 can be obtained from these linear regions. (Shear modulus G can be obtained in a similar manner.) As the curves are continuous through the origin the values of Young's modulus in tension and compression are approximately equal. The shear modulus, G , is about one-third to one-quarter of Young's modulus (see Table 3).

Table 3 HARDNESS AND ELASTIC MODULI

Based on experiments on natural rubber spring vulcanizates similar to A in Table I and containing (above 48 IRHD) SRF black as filler. Note that hardness is subject to an uncertainty of about + 2 degrees.

Hard- ness IRHD+2	Young's modulus E_0 MN/m ²	Shear modulus G MN/m ²	k	Bulk modulus* E MN/m ²
30	0.92	0.30	0.93	1000
35	1.18	0.37	0.89	1000
40	1.50	0.45	0.85	1000
45	1.80	0.54	0.80	1000
50	2.20	0.64	0.73 -	1030
55	3.25	0.81	0.64	1090
60	4.45	1.06	0.57	1150
65	5.85	1.37	0.54	1210
70	7.35	1.73	0.53	1270
75	9.40	2.22	0.52	1330

Shore A (approx.)	lbf/in ²	lbf/in ²		lbf/in ²
35	168	53	0.89	142 000
45	256	76	0.80	142 000
55	460	115	0.64	154000
65	830	195	0.54	171000
75	1340	317	0.52	189 000

k is used in the calculation of compression characteristics (pages 33 and 36). The majority of springs are in the hardness range 40-60 IRHD.

Average design limits: 15% compression, 50% shear.

* See footnote ~+ to Table 2 on previous page.

Note.-Theoretically, with a Poisson's ratio of +, E_0 should equal 30. This is so for soft gum rubbers, but for harder rubbers containing a fair proportion of non-rubber constituents, thixotropic and other effects increase E_0 to about 40.

Incompressibility

The bulk modulus of rubber E (1000-2000 MN/m²) is many times larger than its Young's modulus E_0 , and Poisson's ratio can be taken as 1/2. The very high bulk modulus means that rubber hardly changes in volume even under high loads, so that for most types of deformation there must be space into which the rubber can deform. The more restriction that is made on its freedom to deform the stiffer it will become, a feature used in the design of compression springs.

Strength

The breaking stress of rubber in tension is about 14-28 MN/m² when calculated on the original cross-sectional area. When calculated on the cross-section at break, the breaking stress may be as high as 200 MN/m² which is only a factor of five or so below the corresponding value for steel. A stress well in excess of 160 MN/m² is required to cause the failure of rubber in compression.

Hardness

Hardness measurements are generally used to characterize vulcanized rubbers (see Table 3). For rubber, hardness is essentially a measurement of the reversible, elastic deformation produced by a specially-shaped indenter under a specified load and is therefore related to the Young's modulus of the rubber, unlike metal hardness which is a measure of an irreversible, plastic indentation.

Readings in International Rubber Hardness degrees (LRHD), British Standard Hardness degrees (°BS) and on the Shore Durometer A Scale are approximately the same. This is in contrast to the measurements of metal indentation hardness, where the scales of Brinell and Rockwell B (steel ball), Rockwell A and C (diamond cone) and Vickers (diamond pyramid) are widely different. Hardness is relatively simple and easy to obtain but is subject to some uncertainty, hence the +/- 2 degrees tolerance given in Table 3. Shear modulus values are considerably more accurate, but are less easily obtained.

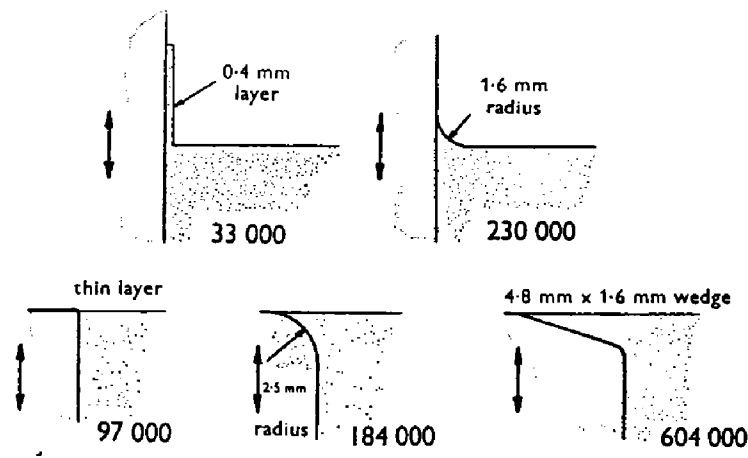
Tensile Stress-Strain Characteristics

Some typical stress-strain curves for natural rubber are shown in Fig. 3. Two features lead to the increased stiffness of high strains; one is due to the sections of molecules between adjacent crosslinks approaching their limiting extension; the other is strain-induced crystallization which occurs in only a few rubbers, notably natural rubber.

Rubber is frequently specified, and its quality controlled during manufacture, by its hardness and its tensile stress-strain properties, i.e. tensile strength, elongation at break and stress at 100%, 300% (etc) strain.

Bonding Rubber to Metal

In a large number of rubber springs the rubber, which is the spring medium, is bonded to metal. Metal parts are generally required for fixing purposes, but may also be a



necessary part of the design—for example the horizontal metal plates which increase the vertical stiffness of bridge bearings. Bonding prevents the rubber from slipping at the load carrying surfaces thus ensuring reliable load-deflection characteristics.

Most modern bonding is carried out by the brass plating method or by the use of proprietary bonding cements. The former, in which the metal is first coated with a layer of brass, is gradually being superseded by the much simpler method of painting cement onto sand-blasted metal surfaces. The coated metal parts when placed in the mould are bonded to the rubber during the vulcanization process. Excellent bond strengths can be obtained by both methods; more often than not the bond is stronger than the rubber. As well as steel many non-ferrous metals and non-metallic materials can be satisfactorily bonded to rubber by the use of suitable cements.

In the design of bonded rubber components the avoidance of sharp corners and stress concentrations is just as important as in the design of metal components (Fig. 15). Generous fillets, having radii at least 10-20% of the smallest overall dimension of the component, should prove beneficial.

Fig. 15.-Effect of reducing stress concentrations on the fatigue life of 25.4 mm square x 6.3 mm thick test pieces or natural rubber subjected to cyclic shear deformations of $\pm 100\%$ strain at 2 Hz. Large flaws visible at the number of cycles shown.

Friction

Bonding, or one of the various forms of mechanical keying, is not always essential as the friction of rubber against a mating surface may be adequate to prevent slip. Plain, unbonded blocks can slip (and thus become softer) under a direct compressive loading. This slippage may not necessarily occur at short times. To avoid the possibility of slip the shape factor (see page 33) of unbonded blocks should be less than half the coefficient of friction.

The coefficient of friction against most dry surfaces is generally about unity, but for design purposes it is usually assumed that slip due to a shear force will not occur if the ratio of maximum shear force to minimum compressive force is less than rubber-steel 0.2, rubber-concrete 0.33.

If water, which is a lubricant for rubber, is present it will normally be squeezed out under load, but the presence of other lubricants and greases should be avoided.

Natural Rubber as a Spring Material

Natural rubber occupies a similar position with regard to rubber springs as spring steel does with metal springs. These two materials are not the sole choices in their respective groups, but they are the obvious and most -widely used ones, particularly where long life and arduous duty are involved. The principle reasons for natural rubber's outstanding success as a spring rubber can be summarized as follows:

It has excellent resistance to fatigue, cut growth and tearing.

It is highly resilient.

It has low heat build-up.

It bonds very efficiently to metals.

It is inexpensive and easy to manufacture.

It has a wider range of operating temperatures than most other rubbers.

Compared with metal springs:

Natural rubber springs require no maintenance.

They have a high energy storage capacity.

They can easily be designed to give different stiffnesses in different directions, or non-linear load-deflection characteristics.

They can accommodate a certain amount of misalignment and are easier to instal.

Although natural rubber is highly resilient, the small amount of inherent hysteresis serves to dampen dangerous resonant vibrations (see Fig. 17).

The various conditions mentioned earlier, which appear unfavourable to natural rubber, need not unduly limit its applications. For example, in many old cars natural rubber mountings are still giving satisfactory service after many years exposure to heat, draught, grease and petrol fumes.

Natural Frequency

The natural frequency (nf) in Hz

of a mounted body on a spring is

$$n_f = \frac{5}{3} / (x)^0$$

where x is the effective deflexion of the spring in centimetres. The effective deflexion is affected (and usually decreases) by the amplitude effect and by non-linear springs (Fig. 16).

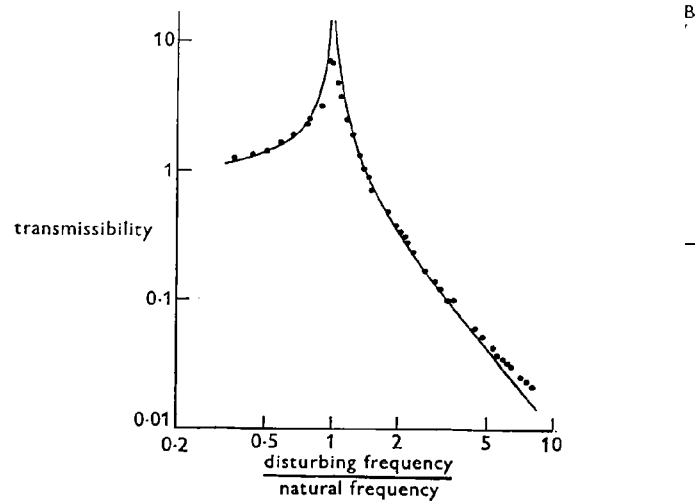


Fig. 16.-OAB is a 'static' load deformation curve. Its exact position and shape will depend on the amount of structure breakdown. The static stiffness at A is the slope of the tangent AC. When a dynamic, small amplitude deformation ED is superimposed on the static deformation at A its stiffness is the slope of AF. Thus the effective dynamic deformation FO is smaller than CC or OG, and the natural frequency will be higher than that predicted from static behaviour. C and Fare generally closer to 0 than illustrated, especially with gum and lightly-filled natural rubber vulcanizates.

Fig. 17.-The transmissibility of a natural rubber containing 40 parts by weight of carbon black. Temperature 19degC

The full line is the theoretical relationship for no damping.

Transmissibility

Transmissibility (T) is the ratio of the amplitude of the vibration on the 'protected' side of the spring to that of the disturbing vibration or, when the spring is on a rigid base, the ratio of the corresponding forces. T depends on the ratio of the disturbing frequency n to the natural frequency n_f of the mounted system (see above).

Under normal operating conditions when $n > 3 n_f$ natural rubber gives attenuation comparable with materials having no damping (Fig. 17). At resonance, when $n = n_f$, the small amount of damping in natural rubber prevents the peak value of T becoming excessive.

Stiffness Characteristics

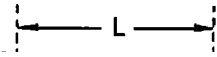
A mathematical theory of rubber-like elasticity has been developed for elastic deformations of up to several hundred per cent. The theory applies strictly only to ideal rubbers, which are completely reversible, incompressible and isotropic, but in practice actual rubbers have been shown to conform quite well. The behaviour of certain kinds of rubber spring may therefore be calculated for deformations of any practicable amount.

However, when the deformation is complex, an exact solution may be unattainable. In some of these cases approximate relationships can be derived from the classical theory of elasticity. This theory, which is the basis of standard engineering practice, only applies to materials in which the strains are very small (a few per cent), but the errors introduced by extrapolation to strains of the order of 10-20 % are not excessive.

In the stiffness equations which follow, terms beyond the first have only been given where their contribution is likely to exceed the variability of the elastic constants. Calculated stiffnesses should be within $\pm 15\%$ of the actual stiffness in most cases; small adjustments can be obtained by slight changes of rubber hardness.

In general, stiffness is particular to a given direction, the stiffnesses in other directions may sometimes be an order of magnitude different as, for example, in bridge bearings. By the suitable selection and location of two or more units the stiffness of a composite spring in three different directions can be varied independently. The design of such springs, of which the inclined shear mounting is a typical example, requires only a knowledge of mechanics and the principal stiffnesses of the units.

SHEAR MOUNTING



$$K = \frac{F}{x} = \frac{GA}{t}$$

$$x_b = \frac{Fl^3}{36AGk_r^2}$$

Shear strain

$$e = \frac{x}{t}$$

Shear stress

$$q = \frac{F}{A}$$

A = cross sectional area

x = shear deflexion

e = shear strain

F = shearing force

G = shear modulus (from Table 3)

K = shear stiffness

q = shear stress

t = thickness of rubber

Fig. 18
Shear mounting

Shear stiffness

Metal plates can be inserted in the rubber without affecting the shear stiffness (see Fig. 31), but their thickness must not be included in 't' the thickness of the rubber.

The maximum working strain depends upon the operating conditions and the type of rubber used. It rarely exceeds 100% and is generally less than 50%.

When the ratio of thickness to length exceeds about 0.25 the deflexion X_b due to bending should be allowed for. It is given by

$$G' = \frac{G}{1 + t^2/36kr^2}$$

(See Reference 22 for more detailed information.)

where
 kr is
the
radius
of
gyration
of the
cross-
section
about
the
neutral
axis.

For a
rectang

ular section $kr^2 = L^2/12$, where L is the dimension in the direction of the force F . For a circular section $kr^2 = D^2/16$ where D is the diameter.

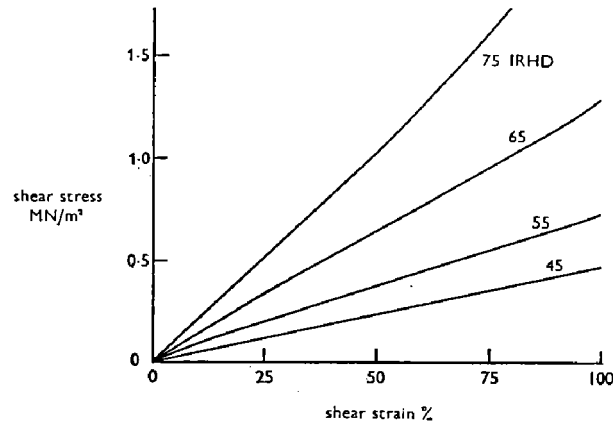
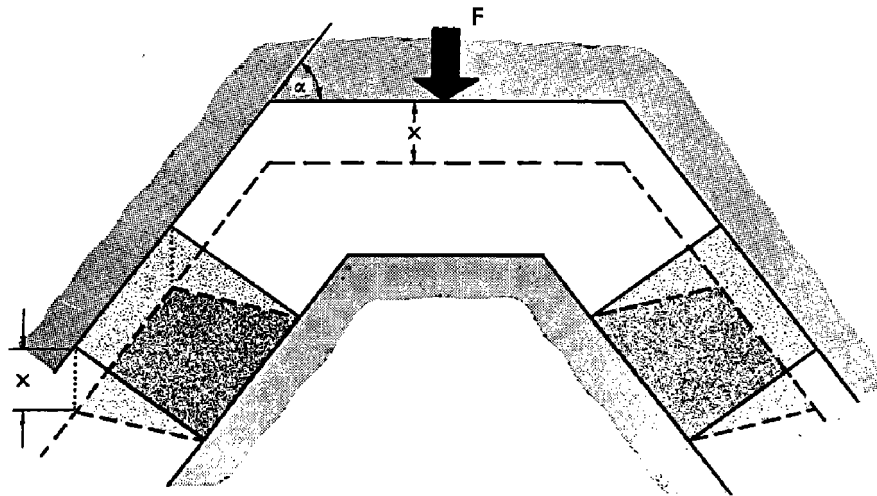


Fig. 20.—The stress-strain curves of rubber in shear are substantially linear over the normal working ranges.

Bending can be allowed for by replacing G by an apparent shear modulus G'' , given by



INCLINED SHEAR MOUNTING

If the left-hand unit alone is considered

a horizontal force H will be necessary to maintain equilibrium under a vertical load F . The system of forces is shown in Fig 22

Fig. 21.-Inclined shear mounting.

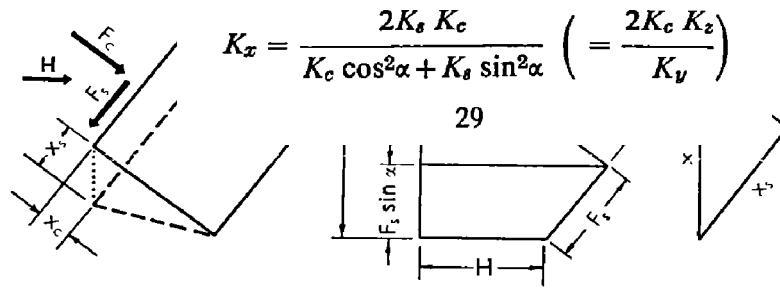


Fig. 22.--Force and deflection diagrams.

$$\frac{1}{2}F = F_c \cos \alpha + F_s \sin \alpha$$

$$x_c = x \cos \alpha \quad x_s = x \sin \alpha$$

$$F_c = K_c x_c \quad F_s = K_s x_s$$

$$\therefore \frac{1}{2}F = K_c x \cos^2 \alpha + K_s x \sin^2 \alpha$$

$$K_y = F/x = 2(K_c \cos^2 \alpha + K_s \sin^2 \alpha)$$

where α = inclination of bearing from horizontal

F = total load

x = deflexion

K_y = overall vertical stiffness

K_c = compression stiffness of each unit (see page 35)

K_s = shear stiffness of each unit (see page 27)

K_x = overall horizontal stiffness (in plane of paper)

K_z = overall horizontal stiffness (perpendicular to plane of paper)

This type of mounting enables different stiffnesses in the vertical and in each of the two horizontal directions to be obtained. In the horizontal direction perpendicular to the plane of the paper the stiffness K_z will be that of the two units in shear, i.e.

$$K_z = 2K_s$$

There are two limiting values for the horizontal stiffness in the plane of the paper K_x . The upper one, when a couple prevents rotation under the action of the horizontal load, is given by a similar expression to K_y , " being replaced by 90° ", i.e.

$$K_x = 2(K_c \sin^2 \alpha + K_s \cos^2 \alpha)$$

The lower value is obtained when there is no restraining couple, i.e.

$$K_s = GA/t = 0.81 \times 0.1 \times 0.1 / 0.025 = 324 \text{ kN/m}$$

the shape factor $S = 10 / (4 \times 2.5) = 1$

$$K_c = E_0 (1 + 2 k S^2) A / t = 3.25 (1 + 2 \times 0.64 \times 1^2) 0.01 / 0.025 = 2960 \text{ kN/m}$$

$$\sin^2 \alpha = 0.93 \quad \cos^2 \alpha = 0.067$$

vertical stiffness

$$K_y = 2(2960 \times 0.067 + 324 \times 0.93) = 1000 \text{ kN/m}$$

transverse stiffness

$$K_z = 2 \times 324 = 648 \text{ kN/m}$$

longitudinal stiffness (two values)

$$K_x = 2(2960 \times 0.93 + 324 \times 0.067) = 5540 \text{ kN/m}$$

and $K_z = 2 \times 2960 \times 648 / 1000 = 3830 \text{ kN/m}$

(1 kgf/cm = 1 kN/m)

75deg to the horizontal. The rubber is 55 IRHD, ie $G=0.81\text{MN/mm}^2$, $E_0=3.25\text{ MN/m}^2$, $k=0.64$.

For each block

Example:
A mine car suspension consists of two bonded rubber blocks 10 cm square by 2.5cm thick each inclined at

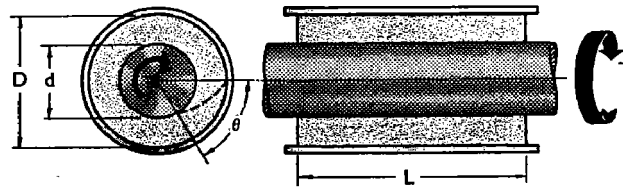


Fig. 24.—Bush mounting in torsion.

TORSI
ON
DISK

Torsional stiffness K_θ

$$K_\theta = \frac{T}{\theta} = \frac{\pi GL}{1/d^2 - 1/D^2}$$

(θ in radians)

or

$$K_\theta = \frac{T}{\theta} = \frac{0.055 GL}{1/d^2 - 1/D^2}$$

(θ in degrees)

where

T = torque

θ = rotation of inner shaft relative to outer

G = shear modulus (from Table 3)

L = length of bush

d = inner diameter of rubber

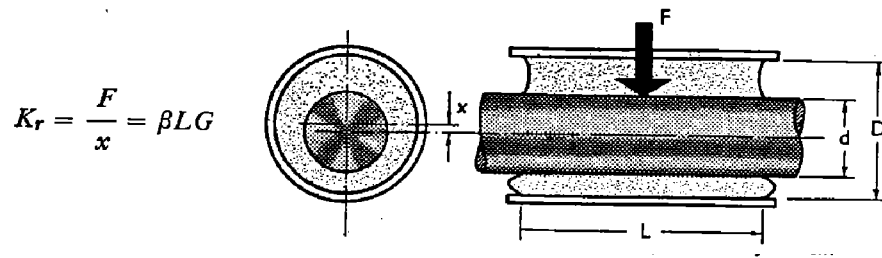
D = outer diameter of rubber

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(For more detailed information see Reference 23)

BUSH MOUNTING



Axial stiffness

K_a

The axial deflexion x of a bush arises primarily from shearing of the rubber, although there may be some contribution from bending.

The axial stiffness, K_a , of a bush which is at least as long as its diameter (ie $L \geq D$), is given by

For shorter bushes (ie $D > L$) the axial stiffness (K_a) given above should be divided by the factor $1 + (D/L)^2$ where " has the following values:

d/D	0.1	0.2	0.4	0.6	0.88
"	0.03	0.03	0.02	0.01	0.0025

Radial stiffness K_r

The radial stiffness K_r has been solved mathematically for only two cases-very long bushes and very short bushes

Fig. 26.-Radial deformation of a bush.

The two values of $\$$ given below correspond to long ($\$L$) and short ($\S) bushes. The stiffness of intermediate length bushes will lie between the extremes given by $\$L$ and $\$S$ for the appropriate ratio of d/D .

d/D	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
$\$L$	9.5	18.3	34	66	135	310	900	3400	32000
$\$S$	5.2	7.9	11.1	15.3	21	30	44	70	150

For more detailed information on bushes see Reference 24, on which the above is based.

COMPRESSION BLOCK

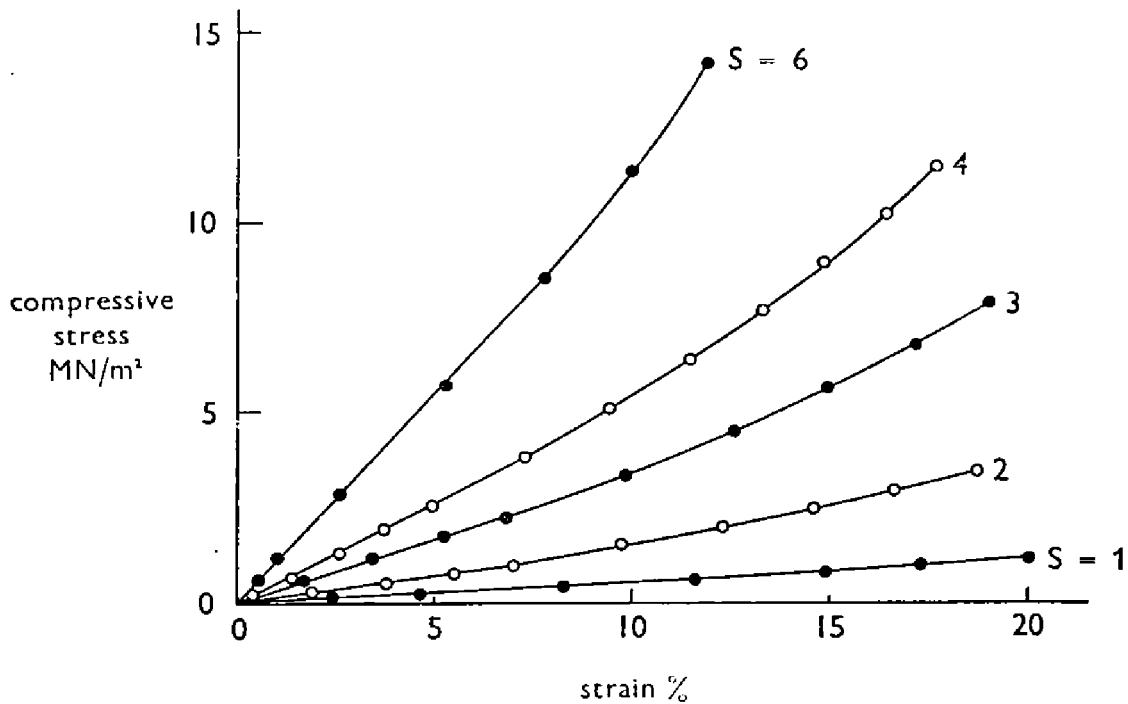


Fig. 27.—Effect of shape factor: Experimental stress-strain curves for 6.3 mm thick disks of rubber (47 IRHD) in compression. The shape factor is shown alongside each curve; the diameter in mm is 25.4 times the shape factor.

The stiffness of rubber in compression, when the loaded surfaces are prevented from slipping (by bonding or by mechanical location), depends upon the shape factor S (see Fig. 27), defined as the ratio of one loaded area to the total force-free area, as shown in Fig. 28.

$$S = \frac{LB}{2t(L+B)}$$

S = shape factor

t = thickness

L = length

B = breadth

For a block of square section (ie $L = B$)
or circular section (diameter = L)

$$\left. \begin{array}{l} \\ \end{array} \right\} S = \frac{L}{4t}$$

The compression modulus E_c depends upon the shape factor S (for derivation see Appendix).

$$E_c = E_0(1 + 2kS^2)^*$$

E_c = compression modulus

S = shape factor

E_0 = Young's modulus (from Table 3)

k = a numerical factor (from Table 3)

*See footnote on following page.

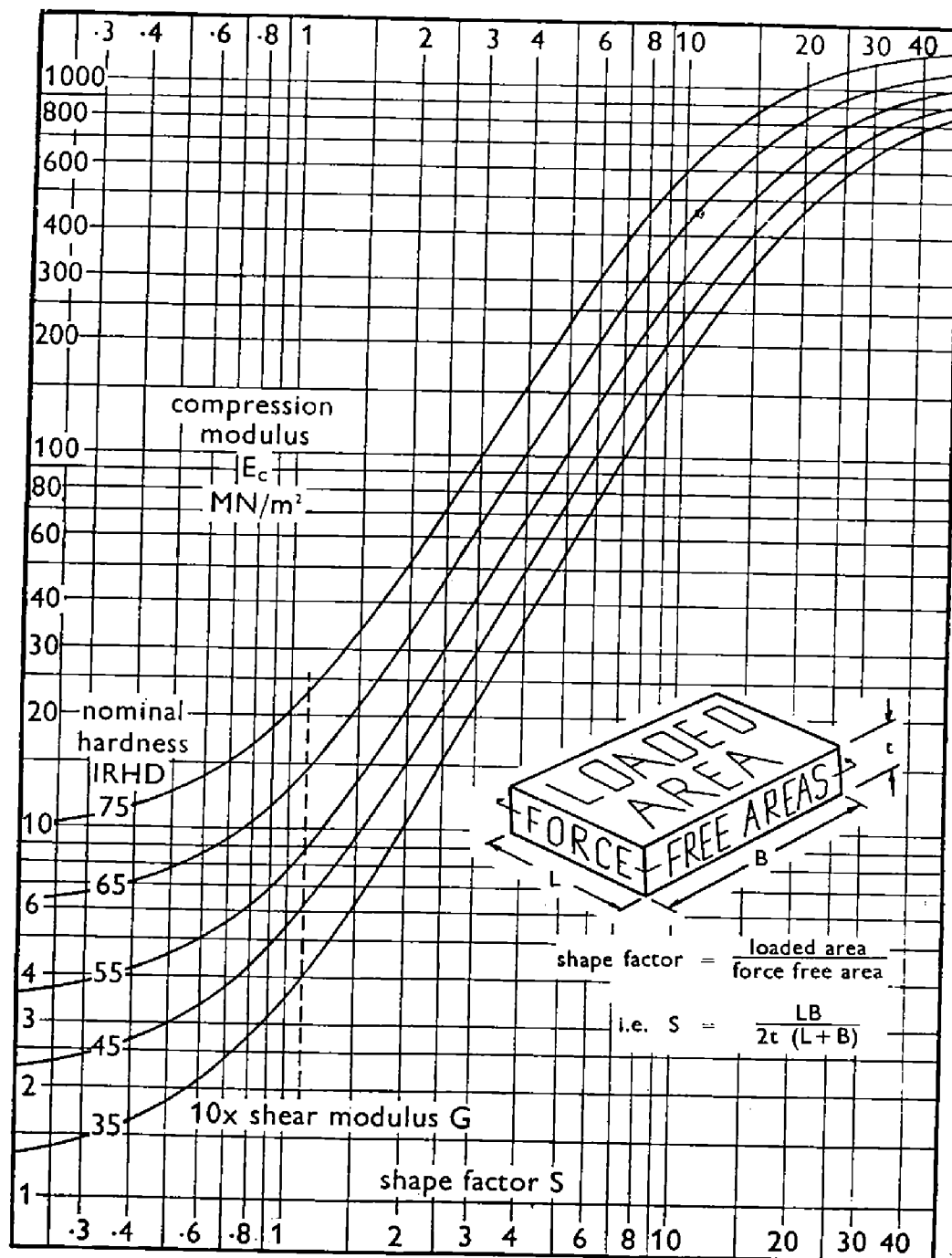


Fig. 28.—Variation of compression modulus E_c with shape factor S for natural rubbers of differing hardnesses (SRF black filler used for 55 IRHD and above).

* Although deformation due to bulk compression can normally be ignored, it can cause a noticeable reduction in E_c when the ratio E_c/E_∞ exceeds about 10%. (E_∞ is the modulus of bulk compression, from Table 3.) To allow for this reduction, use a modified compression modulus obtained by dividing E_c by $1 + (E_c/E_\infty)$. If Wood and Martin's value for bulk modulus is preferred (see footnote to Table 2) then the E_∞ value in Table 3 should be doubled.

When $S > 3$ it may be more convenient to use

$$E_c \simeq 5GS^2$$

where G is the shear modulus (from Table 3).

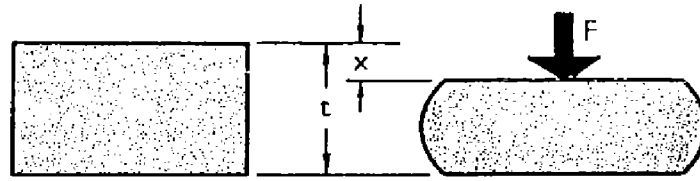


Fig. 29.—Rubber block in compression.

The compression stiffness, K_c , is given by

$$K_c = \frac{F}{x} = \frac{E_c A}{t}$$

where F = compressive load

E_c = compression modulus (corrected, if necessary, for the effect of bulk compression)

A = cross-sectional area

t = thickness

x = deflexion

It is not advisable to rely on friction alone to prevent slip under a compressive load when using unbonded rubber parts, because slip may occur if $S > \mu/2$, where S is the shape factor and μ the coefficient of friction.

The load (F)—deflexion (x) curve of rubber in compression is non-linear. With no slip it has the approximate form

$$F = E_c A e(1 + e)$$

where e , the compressive strain, equals x/t . The non-linearity is usually ignored for strains up to about 10%.

There is as yet no method of calculating that a block will be stable but experience has shown that provided the thickness is less than one-quarter of the least plan dimension there should be no instability.

COMPRESSION STRIP

When a long strip of rubber is compressed the strain in the direction of its length will be negligible.

Shape factor

$$S = \frac{b}{2t}$$

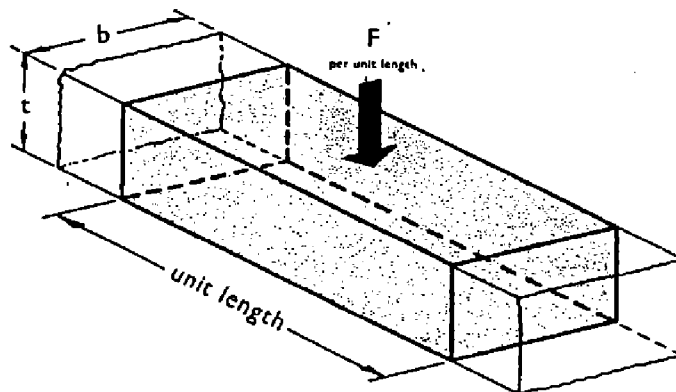


Fig. 30.—Compression strip.

Compression modulus E_c

$$E_c = 4E_0(1 + kS^2)/3$$

The compression stiffness per unit length, K_c , is given by

$$K_c = \frac{F}{x} = \frac{E_cb}{t} = \frac{4bE_0(1 + kS^2)}{3t}$$

where F = load per unit length

E_c = compression modulus (corrected, if necessary, for the effect of bulk compression)

E_0 = Young's modulus (from Table 3)

b = width of strip

t = thickness of strip

x = deflexion

k = a numerical factor (from Table 3)

S = shape factor

The load per unit length (F)—deflexion (x) curve for a compressed strip is non-linear. It has the approximate form

$$F = E_cbe(1 + 3e/2)$$

where e , the compressive strain, equals x/t . As in the case of blocks, the non-linearity is usually ignored for strains up to about 10%.

BRIDGE BEARINGS

The principle requirements of a bridge bearing are:

(i) A high vertical stiffness to prevent appreciable changes in height of the bridge deck under changing load and

(ii) A relatively low horizontal stiffness to prevent excessive loads on the supporting piers due to thermal expansion and contraction of the bridge deck.

The cross-sectional area of a bearing will depend upon the allowable pressure on the support. Knowing this area, the thickness of rubber necessary to limit the horizontal stiffness can be readily determined. The required vertical stiffness is then obtained by insertion of a sufficient number of metal spacer plates. This is the basis of the design of rubber bridge bearings.

The design procedure will depend upon the data provided. As an example suppose a square-sectioned bearing in 60 IRHD rubber has to be designed with the following approximate values:

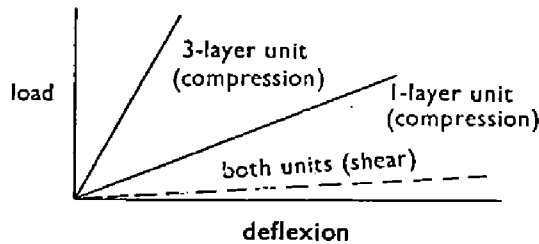
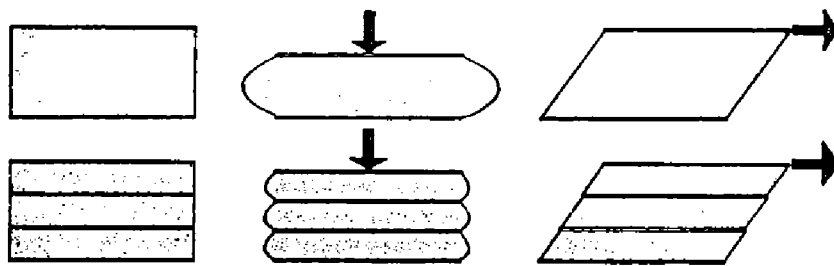


Fig. 31.—The vertical stiffness of a rubber block can be increased by inserting into the block horizontal metal spacer plates which reduce the freedom of the rubber to bulge. The shear stiffness is not altered by the presence of these horizontal plates.

Vertical stiffness	$K_c = 200 \text{ MN/m}$
Horizontal stiffness	$K_s = 3 \text{ MN/m}$
Ratio of side L to total rubber thickness T	$L = 5T$ (for stability)
Shear modulus	$G = 1.06 \text{ MN/m}^2$ (from Table 3)

Thickness of one layer	$t = T/n$
Number of layers	n
Area	$A = L^2$
Shape factor (of one layer)	$S = L/4t$
	$S = L/4t = 5Tn/4T = 5n/4$

$$\text{Shear stiffness} \quad K_s = \frac{GA}{T} = 5GL$$

Approximate compression stiffness

$$K_c = \frac{1}{n} \frac{5GAS^2}{t} = \frac{5GAS^2}{T} = 5K_s S^2 \quad \text{ie} \quad \frac{K_c}{K_s} = \frac{200}{3} = 5S^2 = 5\left(\frac{5n}{4}\right)^2$$

from which n , which must be integral, is 3.

$$\begin{aligned} K_s &= GA/T = 5GL & \therefore L &= K_s/5G = 0.57 \text{ m (say } 0.6 \text{ m)} \\ T &= L/5 = 0.6/5 = 0.12 \text{ m} & t &= T/n = 0.12/3 = 0.04 \text{ m} \end{aligned}$$

The bearing consists of three layers each $60 \text{ cm} \times 60 \text{ cm} \times 4 \text{ cm}$. From these values the calculated stiffnesses are:

$$\begin{aligned} K_s &= 3.18 \text{ MN/m} \\ K_c &= 224 \text{ MN/m (approximate equation for } S > 3) \\ K_c &= 227 \text{ MN/m (from page 35, without bulk modulus effects)} \\ K_c &= 213 \text{ MN/m (from page 35, including bulk modulus effects)} \end{aligned}$$

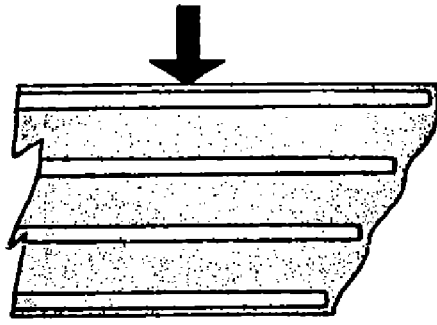


Fig. 32.—Sketch showing part section of a three layer bearing. The thin layer of rubber encasing the 3 rubber layer/4 metal plate unit protects the metal and accommodates irregularities in the contact surfaces. It does not materially affect the stiffness. Note that the edges of the plates have been rounded off.

COMPRESSION OF ROLLERS

When a curved surface of a rubber component is compressed against a rigid plane the stiffness generally increases as the area of contact increases during the deformation. Thus the load deformation characteristics tend to be markedly non-linear. These are described for various shapes. For the rollers (solid, hollow and rubber covered) the relationships apply for plane strain conditions, *ie* for length \gg rubber thickness. For more detailed information see Reference 26.

Solid Rubber Rollers (Fig. 33)

$$\frac{F}{dE_0} = 1.25 \left(\frac{x}{d}\right)^{3/2} + 50 \left(\frac{x}{d}\right)^6$$

F = load per unit length

x = compression

d = cross-sectional diameter

E_0 = Young's modulus

Hollow Rubber Rollers

For hollow rollers with thin rubber walls there is a sharp increase in stiffness at a compression equal to the inner diameter (d_1). The initial stiffness is given by classical elasticity theory; the stiffness after the turn up is approximately that of a compression strip of width $d/2$ and thickness $d-d_1$.

Rubber Covered Rollers (Fig. 34)

The relationships between the compression x of the rubber covering (thickness t) of rubber covered rollers of overall diameter d are shown in Fig. 34 for rollers of differing t/d ratios.

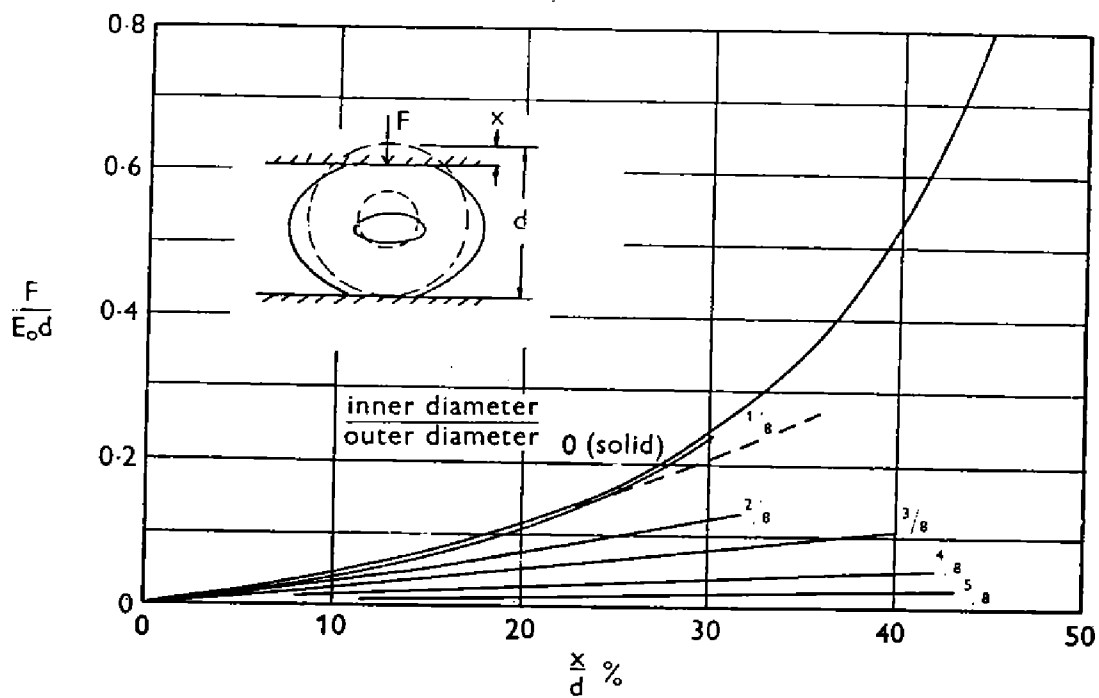


Fig. 33.—Compression characteristics of solid and hollow rubber rollers. That for the solid rubber corresponds to the quoted equation; ignoring the last (sixth power) term, as shown by the broken line, is sufficiently accurate for most purposes. F is the load per unit length.

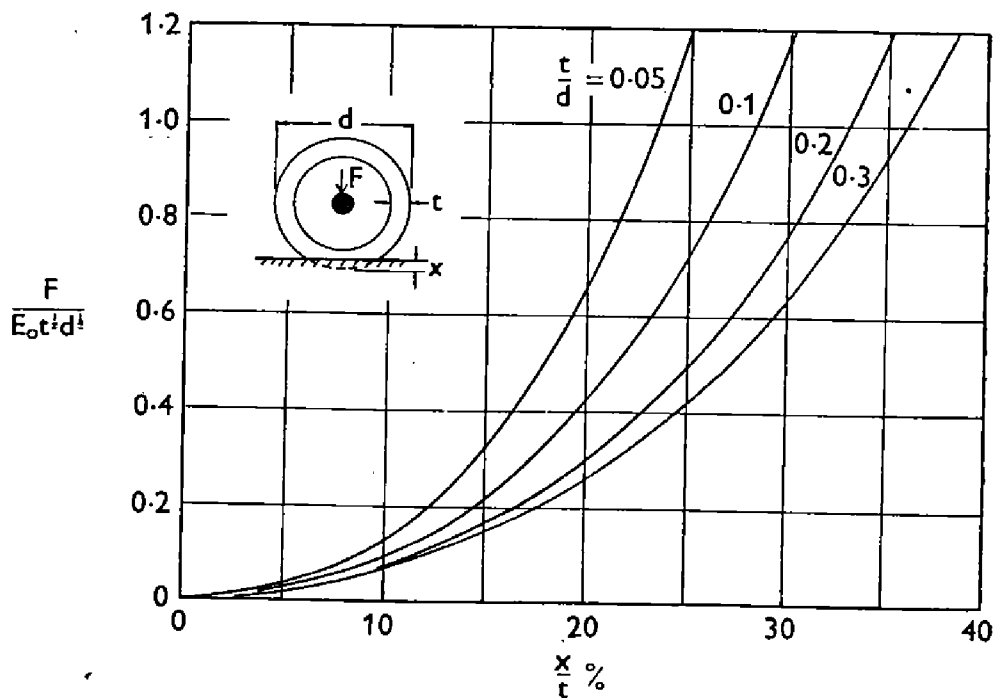


Fig. 34.—Compression of rubber covered rollers of differing cover thickness/overall diameter ratios. F is the load per unit length.

Rubber Sphere (Fig. 35)

Some rubber components (eg bump stops) have hemispherical caps. The compression is approximately half that of the sphere shown in Fig. 35.

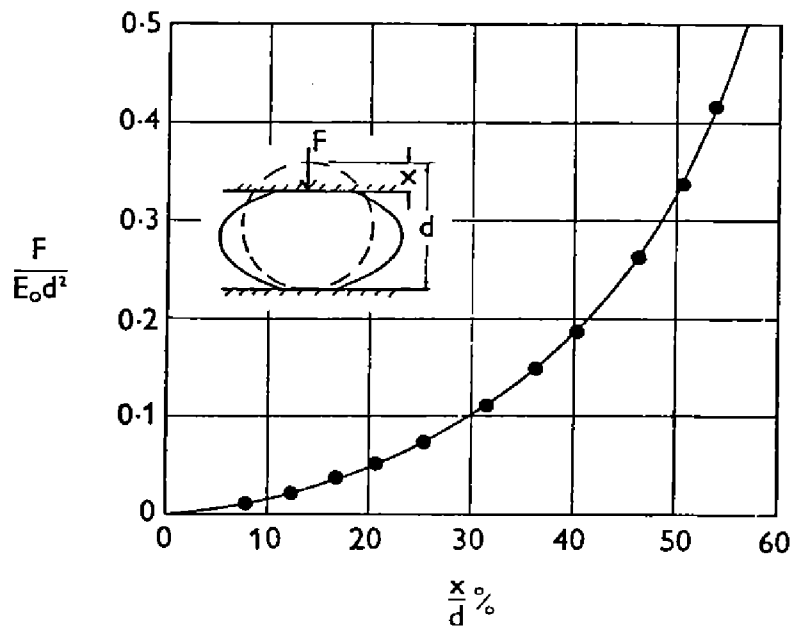


Fig. 35.—Compression of a rubber sphere.

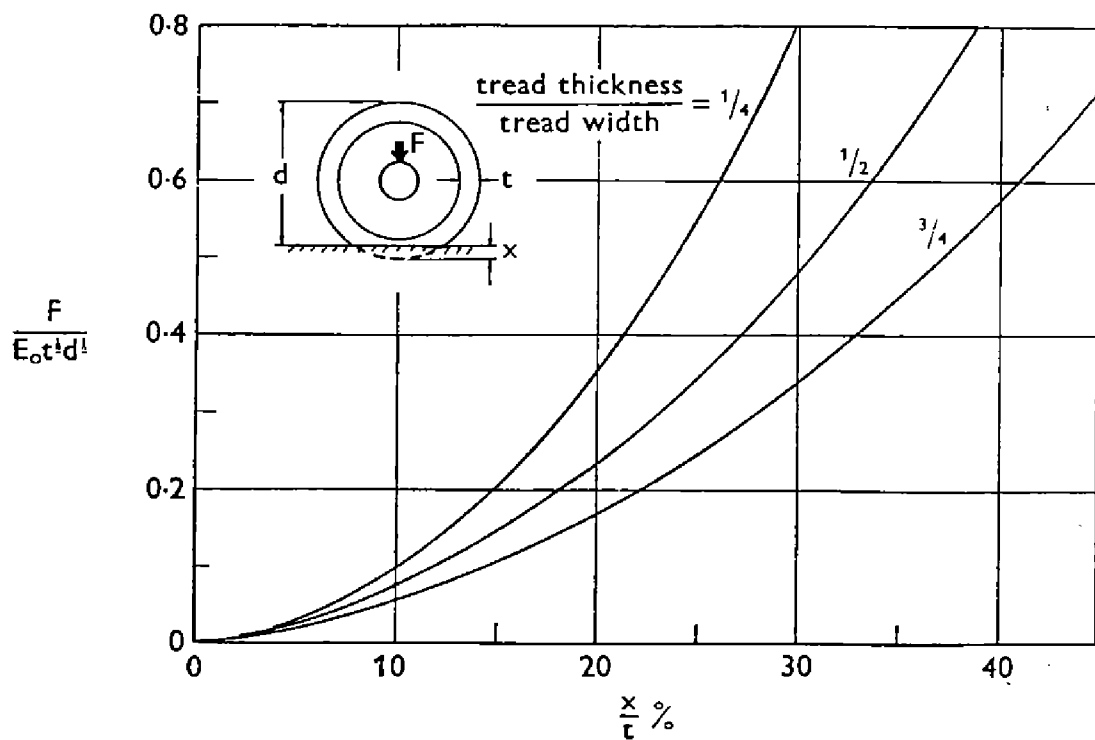


Fig. 36.—Compression of wheels with solid rubber tyres. (Experimental data for wheels with a diameter d five times the tread width.)

Solid Rubber Tyres

The stiffness per unit width of tyre is less than that for a roller of the same cross-section because the rubber can deform sideways (*ie* the effective shape factor is less). Some typical experimental results are shown in Fig. 36.

F = load per unit width of tyre

E_0 = Young's modulus

b = width of tyre

x = compression of tyre

t = thickness of tyre

d = outer diameter of tyre

COMPRESSION OF SOLID RUBBER RINGS

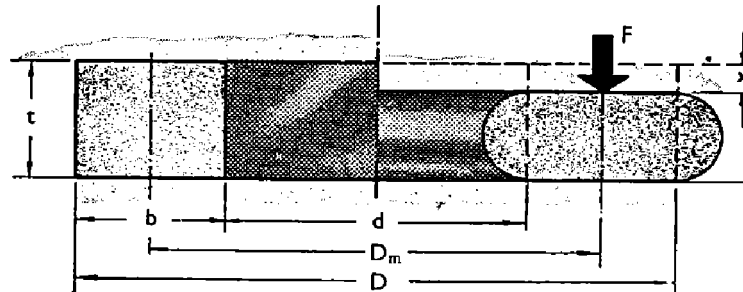
Circular section

The force per unit length F for a solid rubber cylinder should be multiplied by πD where D is the mean diameter of the ring.

The width of the flattened surface of rubber increases during the compression, and is approximately $2.4x$, where x is the diametral compression. From this the average contact stress can be determined.

Rectangular section

Fig. 37.—Rubber annulus of rectangular section.



$$K = \frac{F}{x} = \frac{4}{3} \frac{E_0 \pi (D^2 - d^2)}{4t} \left\{ 1 + k \left(\frac{D-d}{4t} \right)^2 \right\}$$

$$= \frac{4}{3} E_0 \pi D_m \frac{b}{t} \left\{ 1 + \frac{kb^2}{4t^2} \right\}$$

K = stiffness

F = load

x = deflection

E_0 = Young's modulus (Table 3)

k = a numerical factor (Table 3)

D = external diameter

d = internal diameter

D_m = mean diameter = $\frac{1}{2}(D+d)$

b = radial width of section = $\frac{1}{2}(D-d)$

t = thickness

(This expression arises from the compression stiffness of a rubber strip.)

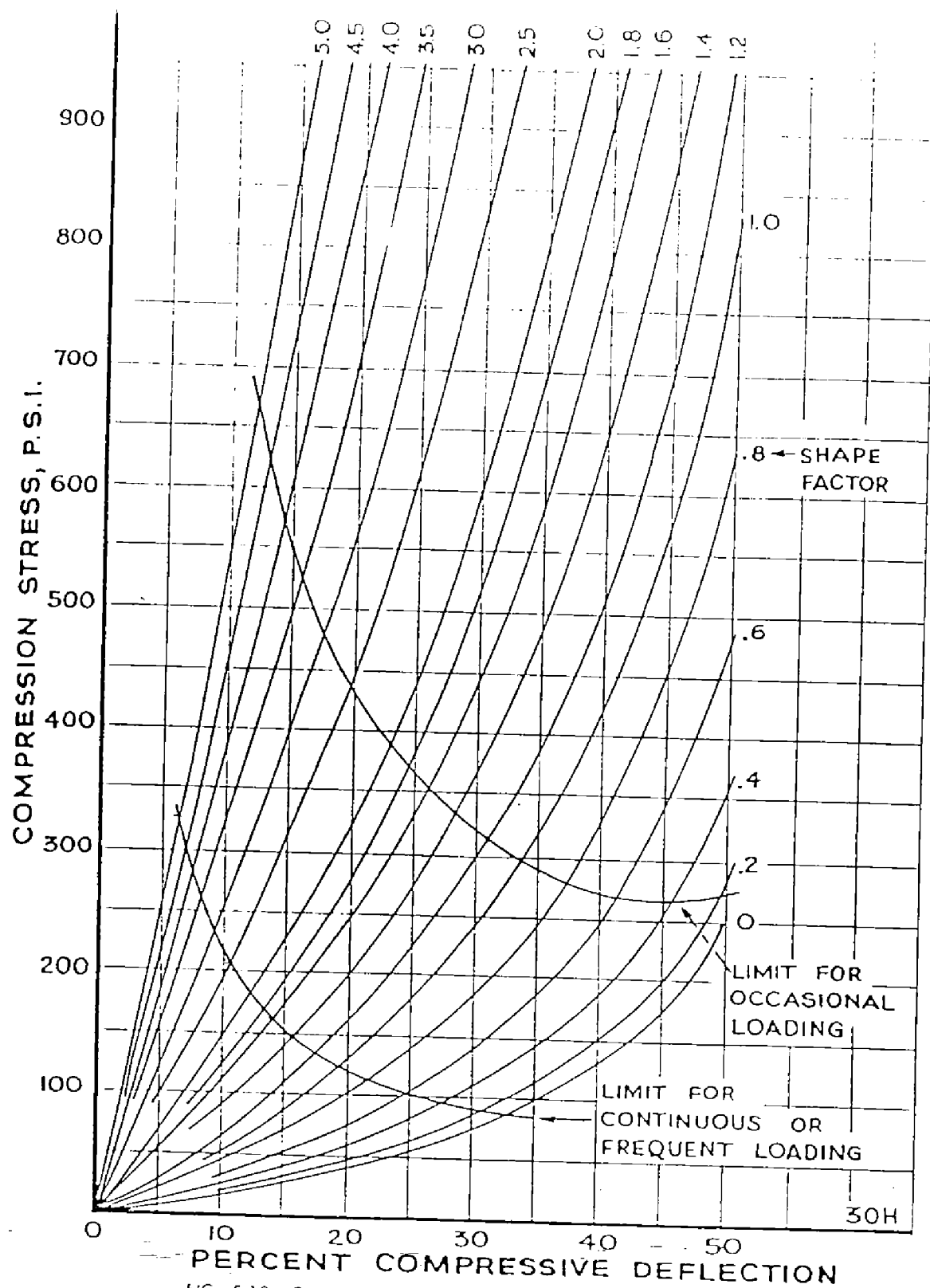


FIG. 5-10 Compression Stress-Strain Curves for 30 Durometer Rubber

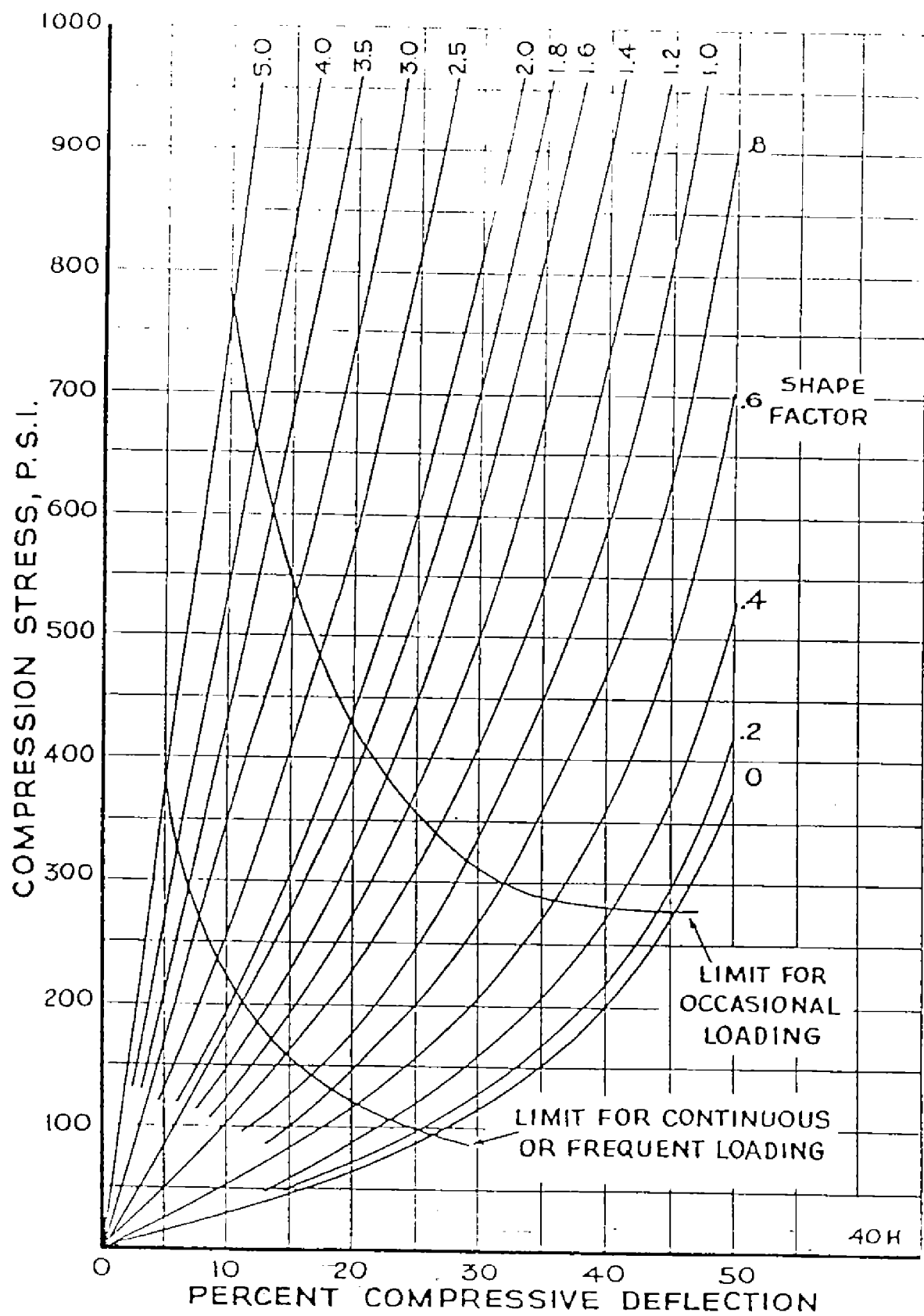


FIG. 5-11 Compression Stress-Strain Curves for 40 Durometer Rubber

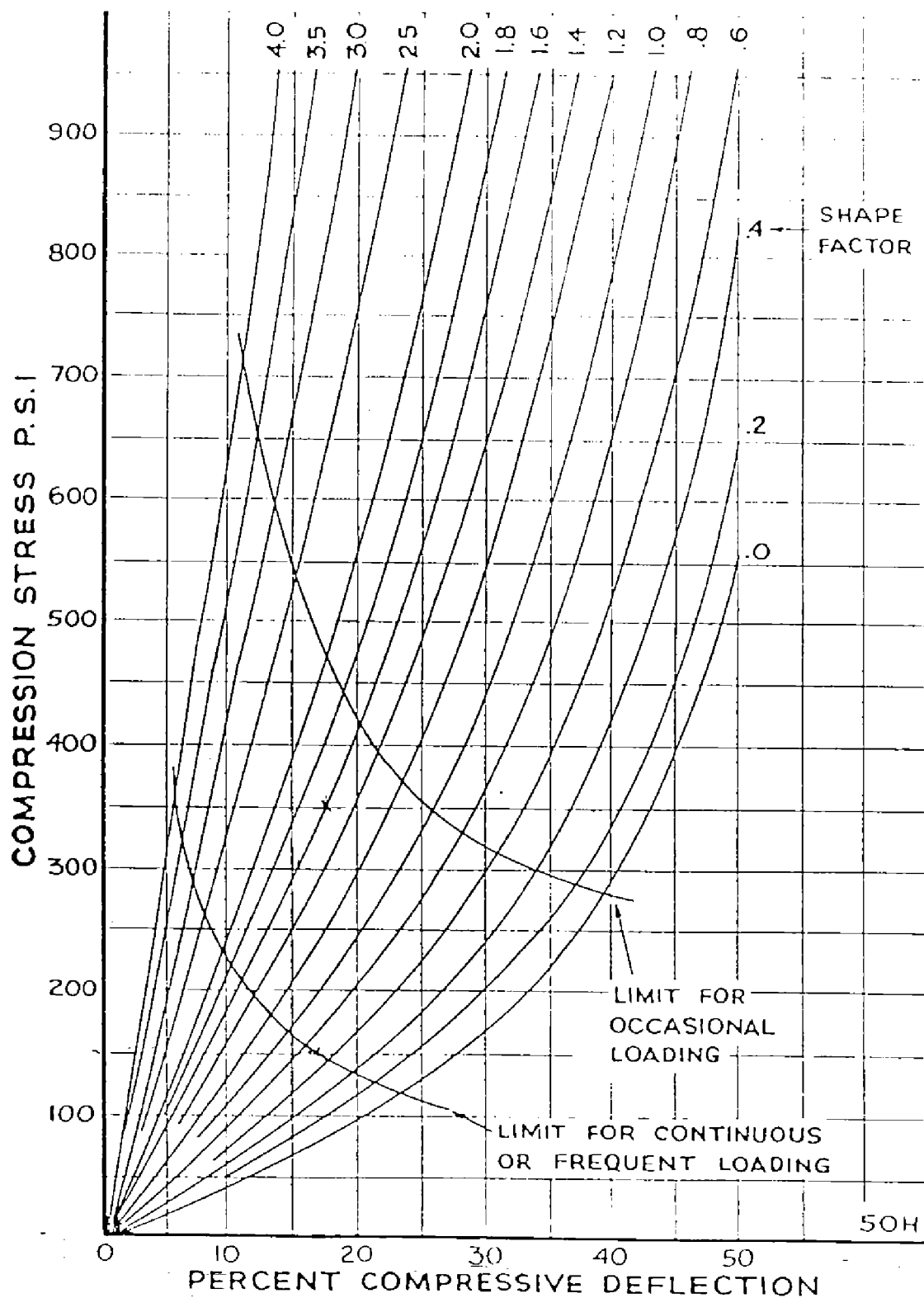


FIG. 5-12 Compression Stress-Strain Curves for 50 Durometer Rubber

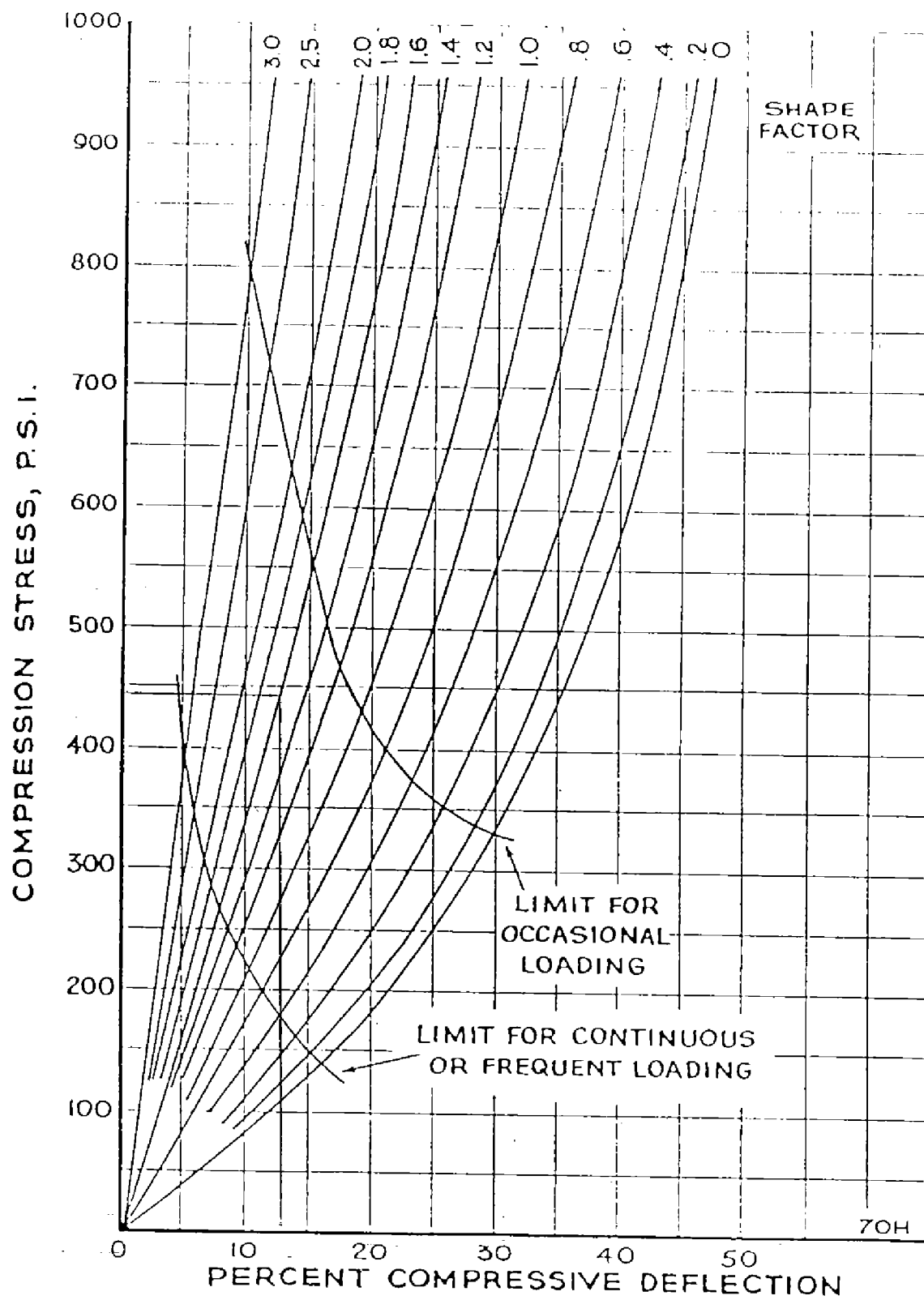


FIG. 5-14 Compression Stress-Strain Curves for 70 Durometer Rubber

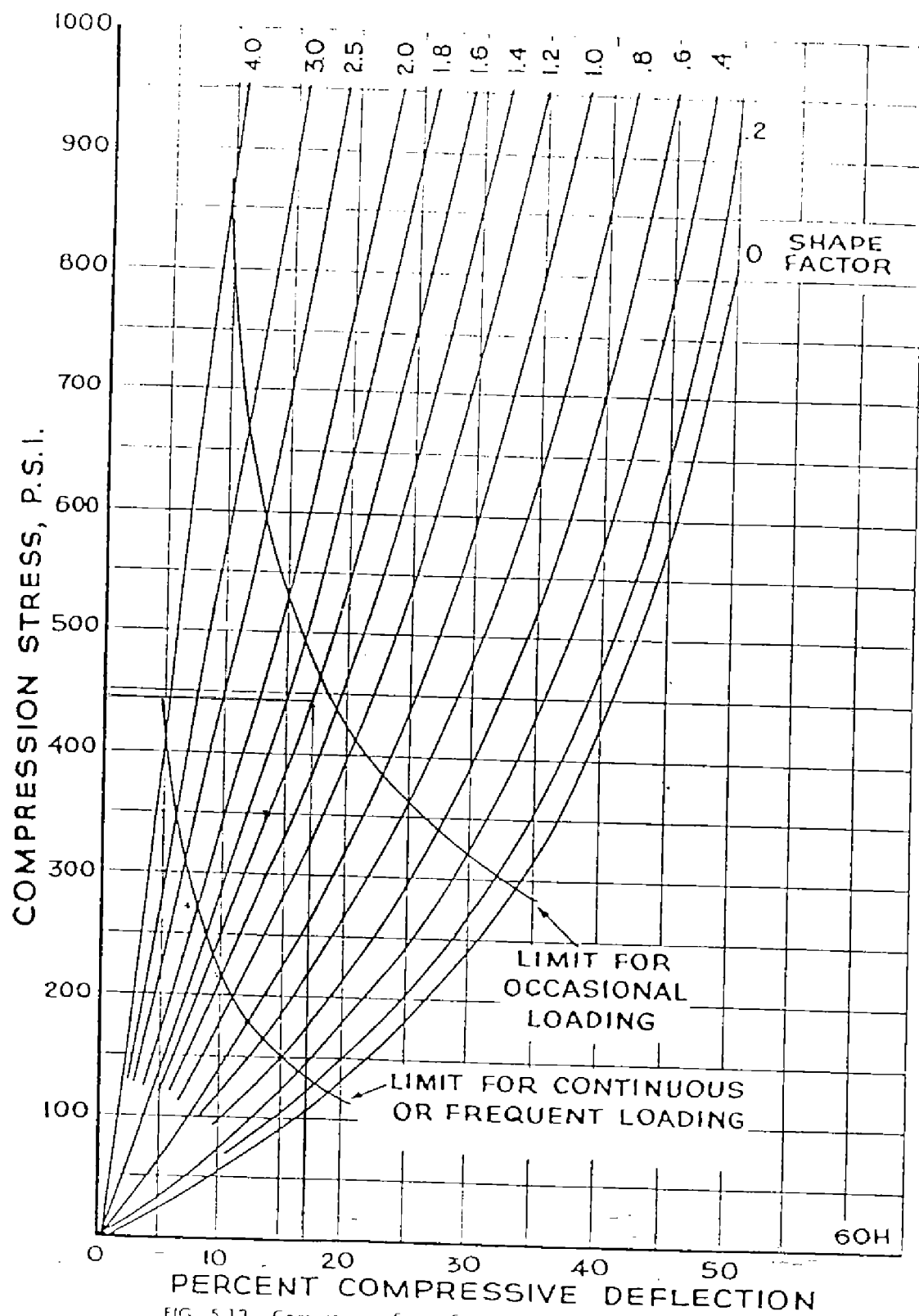


FIG. 5-13 Compression Stress-Strain Curves for 60 Durometer Rubber

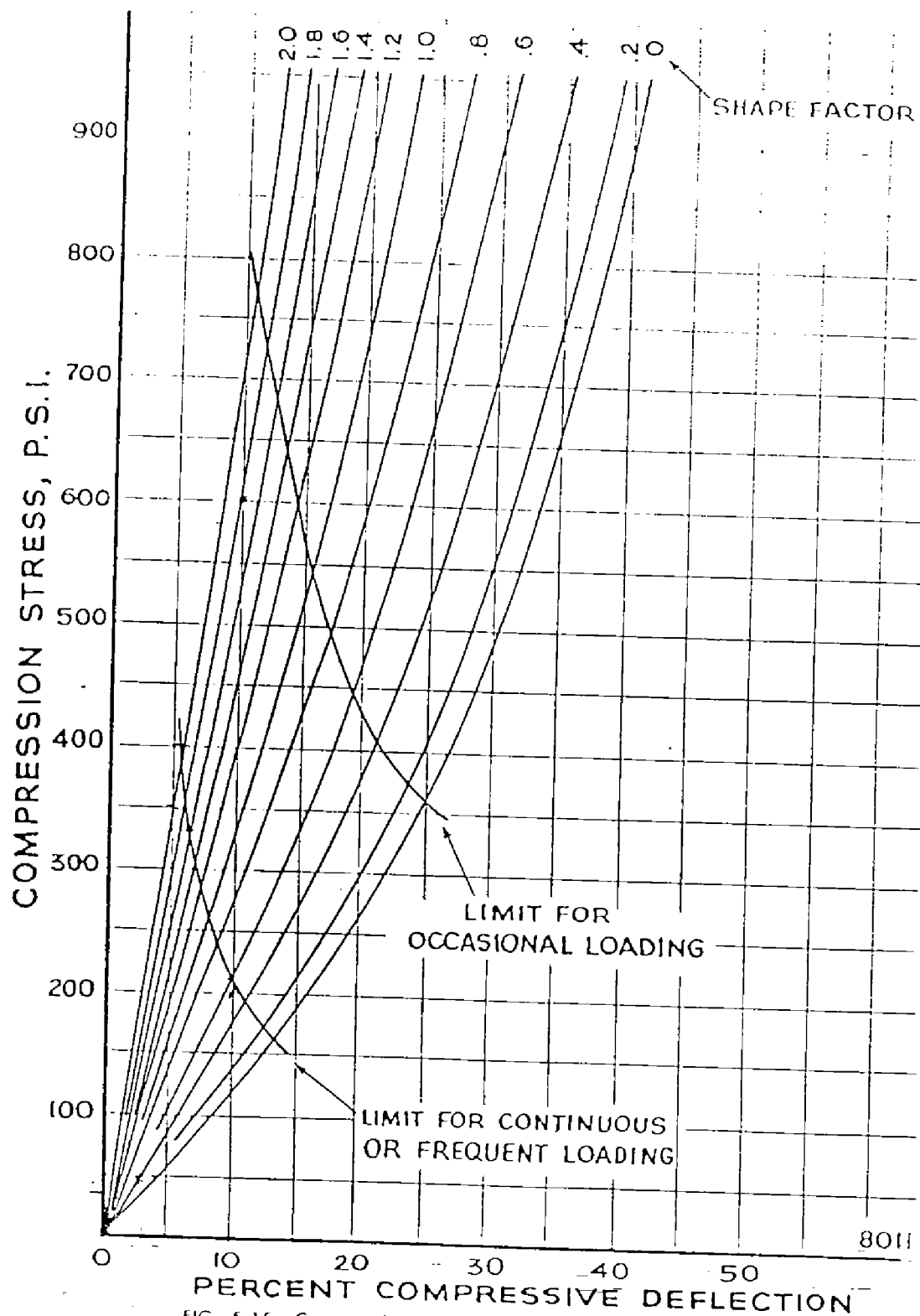


FIG. 5-15 Compression Stress-Strain Curves for 80 Durometer Rubber