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INTRODUCTION

In February 1943 the Spring Committee of the SAE War Engineering Board published at the behest of the Ordnance Department the **MANUAL ON DESIGN AND APPLICATION OF HELICAL AND SPIRAL SPRINGS FOR ORDNANCE**. These springs are characterized by lowest weight and smallest size for a limited life. Peacetime production brought back the problems of maximum endurance associated with greater available space and still at minimum cost. The SAE Spring Committee published a new manual in 1958 along the lines of the Ordnance manual to meet these needs.

One of the purposes of the present revision of the manual is to update the information on stress levels, seeking a better balance between cost and service life. This objective has been partially accomplished by including modified Goodman charts to take account of stress range and by relating stress levels to tensile strength, so that data may be applied to many materials. However, a statement of optimum stress levels remains elusive. On the one hand, suggested stress levels are always based on data which contain variables such as the properties of a particular lot of raw materials, design of a test machine, proportions and processing of the specimen, and statistical analysis of test data. On the other hand, service conditions may include such factors as variable amplitude of loading, unfavorable environment, contact with adjacent parts, high speed, or impact loading.

The best information on stress levels is the service records of similar parts which have been changed and improved and stand up well in actual use. This manual can only help to "size" a new part so that suitable space will be allowed. The optimum design must be developed through the usual stages, concluding with field testing.

For a comprehensive study of fatigue and an extensive bibliography, the SAE publication **FATIGUE DESIGN HANDBOOK** is recommended.

Joseph W. Wells

LETTER SYMBOLS

Symbol	Definition	English Units	SI Units
A	angular deflection	turns	turns
b	width (long side)	in	mm
C	spring index (D/d)	—	—
D	mean coil diameter	in	mm
D _a	diameter of arbor	in	mm
D _i	inside diameter	in	mm
D _o	outside diameter	in	mm
d	wire diameter	in	mm
E	modulus of elasticity in tension	psi	kPa
F	deflection	in	mm
f	frequency	Hz	Hz
		(cycles/s)	
G	modulus of elasticity in shear	psi	kPa
g	standard acceleration of gravity	386 in/s ²	9.81 m/s ²
γ	weight per unit volume	lb/in ³	kg/m ³
H	spring height or length	in	mm
K	correction factor	—	—
L	developed length of active material	in	mm
M	moment	lb·in	N·mm
N	number of active coils	—	—
N _t	total number of coils	—	—
n	number of cycles	—	—
P	load or force	lb	N
p	pitch (axial distance center to center of coils)	in	mm
R	spring rate	lb/in; lb·in/deg	N/mm; N·mm/deg
r	radius	in	mm
S	Stress	psi	kPa
T	time	s	s
t	thickness (short side)	in	mm
θ	angular deflection	deg	deg
V	volume of active material	in ³	mm ³
W	weight	lb	kg

CHAPTER I

FUNDAMENTAL CONSIDERATIONS

1. ENERGY

1.1 DEFINITIONS - A mechanical spring is defined as an elastic body which has the primary function to deflect or to distort under load, and to return to its original shape when the load is removed.

The external work imparted to the spring by the product of the two factors "load" and "deflection" is stored by the spring as potential energy and is subsequently recovered (except for frictional losses) as kinetic energy. The complete cycle may require only a fraction of a second or up to several years; two typical examples are whether the spring actuates an engine valve or a circuit breaker.

For almost all springs, the deflection (F) is proportional to the load (P), provided the elastic limit of the material is not exceeded. The spring then is considered to have a constant "load-deflection rate" ($R = P/F$); the energy, stored by the spring at deflection F or load P, is

$$\text{Energy} = \frac{PF}{2} = \frac{P^2}{2R} = \frac{RF^2}{2}$$

The deflection may be measured as translation (as in helical compression or extension springs) or as rotation (as in torsion and flat spiral springs). Whatever the motion type, the energy is invariably the basic quantity which governs all spring designs.

The above formulas are based on the fact that the deflection starts from free height or free position, depending upon the type of spring. Generally, the spring is given some initial deflection (F_1) upon assembly into its application which develops the load (P_1). From this point, the increase of the load to its maximum value (P_2) will cause the spring to be deflected through a distance or "stroke" ($F_2 - F_1$) to the maximum deflection (F_2). The subsequent decrease in load to P_1 will return the spring to its assembled height. In such instances, the energy to be absorbed by the spring during the cycle is

$$\begin{aligned}\text{Energy} &= \frac{P_1 + P_2}{2} (F_2 - F_1) = \frac{(P_2^2 - P_1^2)}{2R} \\ &= (F_2^2 - F_1^2) \frac{R}{2}\end{aligned}$$

1.2 ENERGY CAPACITY OF DIFFERENT SPRING

TYPES - The general problem for the designer is to know the overall energy capacity required of the spring and to bring this in relation to the maximum stress in the spring. It has been

established in fundamental studies that the energy capacity of a spring for a given maximum stress increases in direct proportion to the volume of active spring material. For each type of spring, the energy capacity per unit volume of active material represents the basic index for the efficient utilization of the spring material.

This manual deals with helical compression and extension springs, in which the spring wire is principally subject to torsional stress, and with torsion and flat spiral springs, in which the spring wire is principally subject to bending stress. The two formulas expressing the specific energy capacity for these spring types are as follows:

For the springs subject to torsional stress:

$$\frac{\text{Energy}}{\text{Volume}} = U \frac{S^2}{G}$$

For the springs subject to bending stress:

$$\frac{\text{Energy}}{\text{Volume}} = U \frac{S^2}{E}$$

where:

U = utilization factor, that is, a constant depending upon the cross section of the material and the type of spring

In particular:

U = 1/2 for material under uniform stress, therefore most efficient utilization of material possible; approached by a thin-walled tube used as a torsion bar spring (under torsional loading) and by a ring spring (under compression)

U = 1/4 for round wire torsion bar springs and coil springs

U = 1/6.5 for square wire coil springs

U = 1/6 for rectangular cross sections in uniform circular bending

U = 1/8 for round cross section in uniform circular bending

The formulas show clearly that the specific energy capacity increases with the square of the maximum stress which the material can withstand without permanent set. This is the reason that spring engineers are forever searching for materials and fabricating processes which will permit the springs to be highly stressed.

As for the types of springs to incorporate the highest possible energy capacity, the formulas are not conclusive in them-

selves because the maximum design stresses vary with the type of spring as the diagrams in the later chapters of the manual show. A comparison of different configurations on the basis of volume of metal can be made by selecting suitable stress values for each configuration and solving the appropriate energy/volume equations.

On the basis of these figures, the following sequences of the main spring types treated in the present manual can be set up in the descending order of their specific energy capacity:

1. Round wire compression springs.
2. Flattened round wire or rectangular wire compression springs.
3. Round wire extension springs.
4. Square wire compression springs.
5. Torsion springs and flat spiral springs with rectangular wire section.
6. Square wire extension springs.
7. Torsion springs and flat spiral springs with round wire section.

It will be found that the capacity of the last three spring types is less than one-half the capacity of the round wire compression springs.

1.3 SPACE LIMITATIONS - When the designer is committed to a particular spring type and when he is confronted by a demand for greater energy capacity after he has exhausted all his resources to provide the best values in stress and in utilization factor, then he must turn to an increase in the volume of active spring material. In this endeavor, he is frequently limited by the space in which the spring must work. Therefore, it is valuable to determine the maximum volume of spring material which can be used in a given space.

For the flat spiral spring, the solution is comparatively simple, depending only upon the required clearance between coils and on the size of the inner fastening device. For the more important round wire compression and extension springs, the curves in Fig. 1 will help the designer to determine quickly if the space at his disposal is approximately adequate. The curves represent "space efficiency factors" V/V_s , where V is the volume of active spring material (the inactive end coils disregarded) and V_s is the volume of the cylindrical space in which the spring is contained when solid. The curves are plotted against the "spring index" $C = D/d$, where D is the mean coil diameter and d is the wire diameter of the spring.

The curves differ for two basic spring applications. Some springs are under a constant static load throughout most of their life, and a further deflection takes place only infrequently, as in a boiler safety valve spring. In such a spring, the entire (round) wire surface is assumed to be under uniform torsional stress. For this spring the space efficiency factor is

$$V/V_s = \frac{\pi C}{(C+1)^2}$$

Other springs are expected to withstand a great number of

load cycles which subject the wire to large stress ranges, as in most engine valve springs. In such a spring, where the maximum stress range is the criterion for failure, the higher stresses near the inside of the coil due to wire curvature and direct shear become more critical. This is usually taken into account by multiplication of the torsional stress with a "stress correction factor" greater than 1 and dependent upon the spring index C . The "Wahl factor" K which is most widely used for this purpose is expressed by the formula

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

For the spring which must not fail under these "fatigue loading" conditions, the space efficiency factor is

$$V/V_s = \frac{1}{K^2} \frac{\pi C}{(C+1)^2}$$

After a suitable spring index has been estimated, the space efficiency factor may be found on the appropriate curve in Fig. 1. When the factor is multiplied with the space which is available for the solid spring, the result will show the maximum volume of active spring material which may be used in the available space.

To choose a suitable index, it is necessary to find a compromise between the following two trends: With decreasing index, the space efficiency factor will usually be increased, but the maximum deflection of the spring will be reduced.

1.4 NESTED SPRINGS - If the volume of active material, which can be built into one spring within the available space, is insufficient to satisfy the energy requirements for the spring, as expressed in the formula for specific energy capacity, then the designer should consider the use of a second (and possibly a third) spring inside the first one and acting in parallel with it. The springs in such a nest should be wound right and left, alternately to avoid jamming.

The most economical design will be approximated if all springs have equal values of solid height (H_s), free height (H_f), and spring index (C), since their maximum stresses will then be equal. This means that the wire diameters of the springs will be in the same proportion as their mean coil diameters, and their energy capacities will have the same ratio as the squares of the wire diameters. Rather than have equal maximum stresses, it may be more efficient to have the smaller wires assume a somewhat higher stress level because of their greater tensile strength.

Assuming no radial clearance either between the nested springs or between the outer spring and the housing (with diameter (D_o) in which it is to operate, the wire diameters and mean coil diameters can be expressed in terms of D_o as follows.

Subscripts 1, 2, and 3 in the following equations denote the

mensions of the outer, center, and inner springs, respectively.

springs of the nest, then the space efficiency factor for static loading is:

$$d = \frac{D_0}{C+1}$$

$$D_1 = \frac{CD_0}{C+1}$$

For two springs:

$$d_2 = \frac{D_0}{C+1} \cdot \frac{C-1}{C+1}$$

$$D_2 = \frac{CD_0}{C+1} \cdot \frac{C-1}{C+1}$$

$$V/V_s = \frac{\pi C}{(C+1)^2} \left[1 + \left[\frac{C-1}{C+1} \right]^2 \right]$$

$$d_3 = \frac{D_0}{C+1} \cdot \left[\frac{C-1}{C+1} \right]^2$$

$$D_3 = \frac{CD_0}{C+1} \cdot \left[\frac{C-1}{C+1} \right]^2$$

For three springs:

$$V/V_s = \frac{\pi C}{(C+1)^2} \left[1 + \left[\frac{C-1}{C+1} \right]^2 + \left[\frac{C-1}{C+1} \right]^4 \right]$$

If V represents the total volume of active material in all

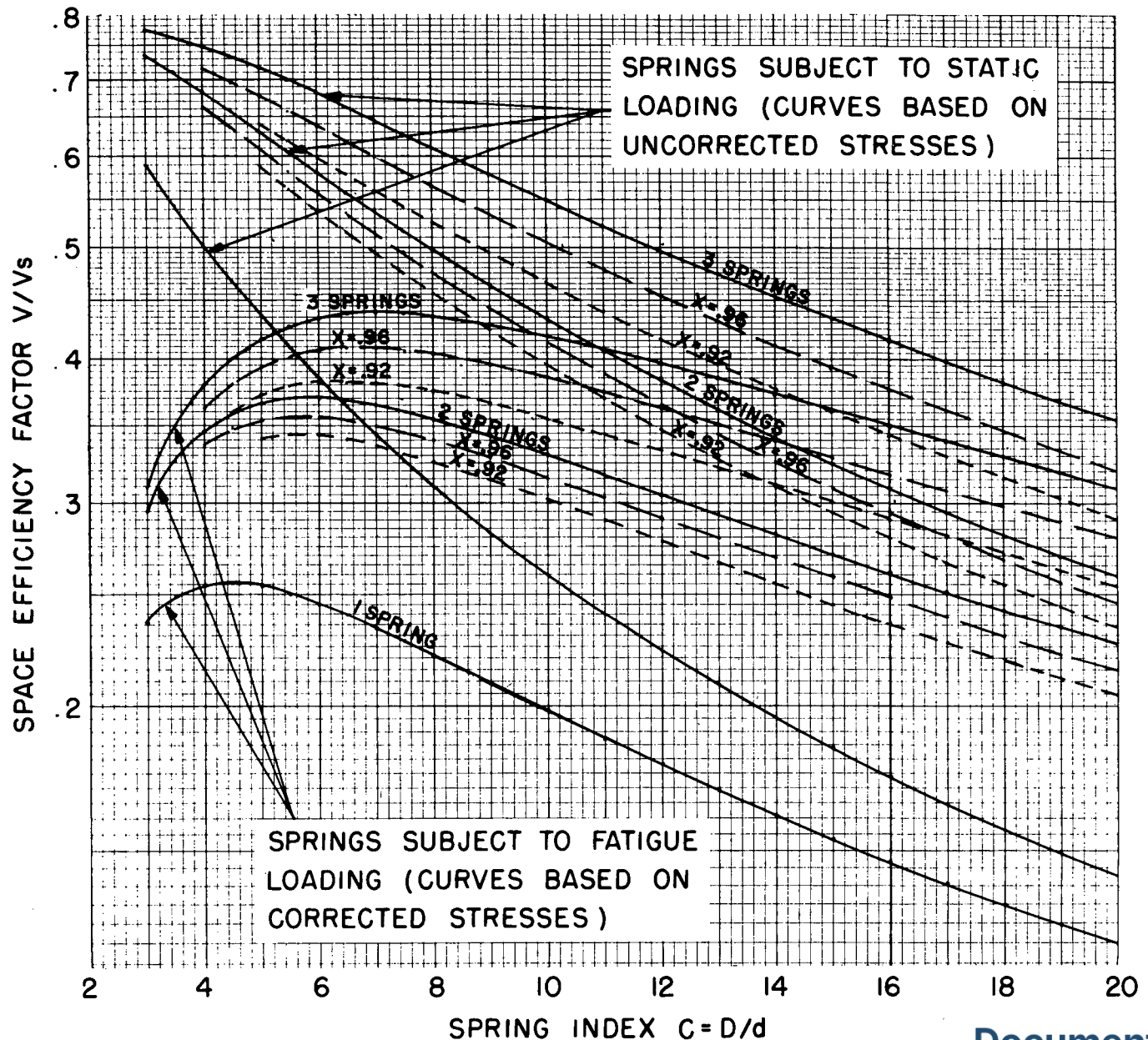


Fig. 1 - Space efficiency of round wire compression and extension springs



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For fatigue loading conditions, these formulas will again have to be multiplied by $1/K^2$. In Fig. 1, the solid lines show the space efficiency factors for spring nests without clearance between the springs.

For practical purposes, it is necessary to provide radial clearances within the nest. This is expressed by a factor X , the ratio of actual diameters to diameters which would allow no clearance. Values of X may be established from consideration of tolerances and diameter changes.

With the assumptions that C , H_s and H_f are equal for all springs, the space efficiency factor for static loading then becomes:

For two springs:

$$V/V_s = \frac{\pi C}{(C+1)^2} \left[1 + X^3 \left[\frac{C-1}{C+1} \right]^2 \right]$$

For three springs:

$$V/V_s = \frac{\pi C}{(C+1)^2} \left[1 + X^3 \left[\frac{C-1}{C+1} \right]^2 + X^6 \left[\frac{C-1}{C+1} \right]^4 \right]$$

Values of X may be as low as 0.92 for springs of less than 1 in (25.4 mm) outside diameter. They may increase by approximately 0.01 with every inch of increase in outside diameter. Thus a second spring with 5 in (127 mm) outside diameter should have ample clearance within the outer spring if it is designed with

$$d_2 = 0.96 \frac{D_0}{C+1} \cdot \frac{C-1}{C+1}$$

$$D_2 = 0.96 \frac{CD_0}{C+1} \cdot \frac{C-1}{C+1}$$

The dashed lines in Fig. 1 indicate the space efficiency values for spring nests with clearance between the springs obtained by using $X = 0.92$ and $X = 0.96$.

If clearance must be provided between the outer spring and the housing, the space efficiency will be further reduced since it will then be necessary to use

$$d_1 = X \frac{D_0}{C+1} \quad D_1 = X \frac{CD_0}{C+1}$$

$$d_2 = X^2 \frac{D_0}{C+1} \cdot \frac{C-1}{C+1} \quad D_2 = X^2 \frac{CD_0}{C+1} \cdot \frac{C-1}{C+1}$$

$$d_3 = X^3 \frac{D_0}{C+1} \left[\frac{C-1}{C+1} \right]^2 \quad D_3 = X^3 \frac{CD_0}{C+1} \left[\frac{C-1}{C+1} \right]^2$$

Fig. 1 confirms the obvious consideration that the use of more than one round wire spring will offer greater reward in a space of large diameter (permitting a high spring index C) than in a space of small diameter. Thus, it appears advisable

to introduce a second spring only for a spring index greater than 4, and a third spring only for a spring index greater than 6.

Changing from a lower to a higher spring index is not efficient for a single spring, but it is rewarding when a second spring is added. In particular, nested springs of high index are necessary if a greater flexibility is required. Then again, in a given space, the higher spring index is equivalent to a smaller wire size, and that generally indicates less manufacturing difficulties and higher permissible stresses.

Helical springs of rectangular wire are used in special applications where a greater volume of material is required in a limited space. Material is not readily available for small lots, and special techniques are required for coiling uniformly so they should not be used randomly in place of round wire springs. However, they may provide a solution to a problem of high loads in small spaces.

Sometimes the designer finds that the available space is inadequate to house a certain spring or a nest of springs which could operate within reasonable stress limits. In that event, he should investigate whether he can satisfy the requirements with a reduced working stroke. This will result in a stress reduction, in that the stress is directly proportional to the working stroke. If this fails, there can be only two solutions to the problem. One is to enlarge the space for the spring, either in diameter or in height. The other is to effect changes in the overall machine design which will reduce the energy requirements.

In a new design, it is essential to compute the required space for each spring at the outset and to make provision for that space in the machine design. This is the only true safeguard against subsequent spring failures. Such failures generally occur when the spring is overstressed because the available space is incompatible with the loading requirements which the spring has to meet.

2. STATIC VERSUS DYNAMIC LOADING

The term "static loading" implies a load application of sufficiently low velocity that the spring is not deflected beyond the equilibrium position and deflections and stresses remain uniformly distributed over the entire spring. For instance, when a mass of weight, W , is gradually lowered on a vertical helical spring of rate, R , a deflection $F = W/R$ will result. The corresponding stress can then be readily found from the spring formulas.

The term "dynamic loading" implies an application in which the velocity of the moving spring end is appreciable. This is true for all impact loadings as well as for sinusoidal loadings under high frequencies. Because of the inertia of the spring coils, the whole spring deflection is not equally distributed among the coils so that some of them must undergo a stress range greater than the corresponding stress range with static loading. For example, let the mass of weight, W , be suddenly released from a point where it just touches the top of the free vertical helical spring of rate, R . In this case, the spring will

undergo a maximum deflection F_1 before it comes to rest at the equilibrium position under deflection F .

When the mass descends a distance F_1 , its potential energy decreases by $W \cdot F_1$; assuming no damping, the strain energy in the spring increases by $R/2 \cdot F_1^2$.

Therefore:

$$W \cdot F_1 = \frac{R}{2} \cdot F_1^2$$

$$F_1 = \frac{2W}{R} = 2F$$

It then follows that, for this instance of dynamic loading, the maximum deflection and consequently the maximum stress is twice as large as would result from the gradual (static) application of the same mass. When the mass is released from some point higher than the top of the free spring, then the maximum stress is more than twice the static stress.

Two basic instances of dynamic loading can be distinguished. In one instance, the stroke is practically constant; this applies to the driving spring of a machine gun since the striking part is finally stopped by a stiff buffer acting in parallel to the driving spring. The other instance is that of a buffer spring, which absorbs almost the full kinetic energy of the striking part and therefore has a variable stroke. In the first instance, the spring itself may take a greater deflection than that prescribed by the striking part owing to the kinetic energy of its coils. In addition, at minimum compressed height, the whole deflection of the spring is not equally distributed among the coils so that some of them must necessarily take a higher than average stress. Therefore, the conventional formulas give too low a maximum stress, and special formulas are required for the maximum dynamic stress.

An example of dynamic loading will serve to illustrate the relationships. A precompressed spring is suddenly compressed by a heavy mass moving with constant velocity, v , over a given stroke. The motion of the front end of the spring causes a surge wave which is propagated along the spring wire with surge velocity c while the coils themselves move axially with velocity v . This wave may be called "single wave," as long as no reflection occurs at the rear end of the spring. The amplitude of this single wave can be expressed by one of the following:

1. Change of load, ΔP , lb (N)
2. Change of stress, ΔS , psi (kPa)
3. Change of deflection per coil, Δf , in (m)

It has been found that all these changes of load, stress, and deflection per coil are proportional to the velocity of loading, v , in/s (m/s). They may be expressed as follows:

$$\frac{\Delta P}{v} = R \cdot T_s = \frac{\pi}{8} \cdot \frac{d^3}{D} \cdot \sqrt{\frac{2\gamma \cdot G}{g}}$$

$$\frac{\Delta S}{v} = \frac{\Delta P}{v} \cdot \frac{\Delta S}{\Delta P} = \sqrt{\frac{2\gamma \cdot G}{g}}$$

$$\frac{\Delta f}{v} = \frac{T_s}{N} = \pi \cdot \frac{D^2}{d} \cdot \sqrt{\frac{2\gamma}{gG}}$$

where:

T_s = travel time of the wave over the whole wire length, or "surge time," s

$g = 386 \text{ in/s}^2 (9.81 \text{ m/s}^2)$

$\gamma = 0.283 \text{ lb/in}^3 (7833 \text{ kg/m}^3)$ for steel

With mean values in pound and inch units

$$\frac{\Delta P}{v} = 51 \frac{d^3}{D} \quad \frac{\Delta S}{v} = 130 \quad \frac{\Delta f}{v} = 35.5 \cdot 10^{-6} \cdot \frac{D^2}{d}$$

or in N, kg, and m units

$$\frac{\Delta P}{v} = 13,800 \frac{d^3}{D} \quad \frac{\Delta S}{v} = 35,300 \quad \frac{\Delta f}{v} = 44.14 \cdot 10^{-3} \cdot \frac{D^2}{d}$$

Here the most interesting result is: The additional stress ΔS is independent of the spring dimensions; for any compression spring made of steel, each inch/second (each m/s) of impact velocity increases the stress by 130 psi (35.3 MPa) or, each ft/s (each m/s) of impact velocity increases the stress by 1560 psi (35.3 MPa).

After reflection of the single wave at the rear end of the spring, the incident and the reflected wave are superimposed by ordinary addition of their stress amplitudes. Therefore, the maximum stress built up in the spring depends on the stress amplitude of the single wave as well as on the number of superimpositions of the single wave over the duration of the compression stroke. While the stress amplitude depends upon the impact velocity only, the number of superimpositions is a function of the surge time T_s , that is, the spring dimensions.

With slow dynamic loading, during compression time, there are many superimpositions of the surge wave, but of small amplitude. Therefore, the dynamic stress will not greatly exceed the static stress.

However, with rapid dynamic loading, the compression time is of the order of the surge time. There are few superimpositions, but the amplitude is high, and, consequently, the dynamic stress can exceed the static stress substantially. In such instances it may be advisable to tune the surge time T_s with regard to the compression time by proper selection of the spring dimensions.

In most instances of dynamic loading, the analysis of stress is not as simple as in the example given above. However, in general, two groups of factors appear to be decisive for the maximum stress built up and the spring life expected:

1. The "loading function" $v(t)$, that is, the time-velocity function of the moving end of the spring, with stroke, compression time, maximum velocity, and maximum acceleration as important parameters.

2. The spring, characterized by its material, maximum static stress, surge time T_s and inherent damping.

The influence of the loading function $v(t)$ on the life of springs has been strikingly proved by fatigue tests: The average life of springs under sinusoidal loading may be reduced as much as 50:1 under impact loading, with the same stroke and frequency. However, the importance of each individual life factor remains to be explored by more fundamental testing.

Gun springs are among the springs which are subjected to severe dynamic loadings. The following methods of design have been successfully tried:

1. Single-wire springs - Proper adjustment of the surge time T_s to the given compression time reduces the maximum dynamic stress.
2. Stranded wire springs, coiled from 3-wire or 7-wire strand - Their greater inherent damping improves the spring life.
3. Two-section springs, that is, a front spring in series with a somewhat stiffer rear spring - In comparison with a single spring, a two-section spring gives more possibility of adjusting the spring assembly to the given loading function since a more equalized distribution of stresses over the wire length can be obtained at the moment of maximum compression. This results in a lower peak stresses and better life than with a single spring.

3. NATURAL FREQUENCY

With dynamic loading of compression springs, the inertia of the spring coils, in addition to their elasticity, becomes effective. When moving one end of a precompressed or free spring in an axial direction, the disturbance of the equilibrium at this end is propagated along the spring wire in a transient wave with a definite surge velocity:

$$c = L \sqrt{\frac{Rg}{W}} = \frac{d}{D} \sqrt{\frac{gG}{2\gamma}} \text{ in/s}$$

With mean values for spring steel, it follows:

$$\begin{aligned} c &= 88,560 \cdot \frac{d}{D} \text{ in/s} \\ &= 7380 \cdot \frac{d}{D} \text{ ft/s} \end{aligned}$$

The travel time of the wave over the whole wire length, or "surge time," is:

$$T_s = \frac{L}{c} = \frac{\pi \cdot N \cdot D}{c} \text{ s}$$

With mean values for spring steel:

$$T_s = 35.5 \cdot 10^{-6} \cdot \frac{D^2}{d} \cdot N$$

or

$$T_s = 35.5 \cdot 10^{-6} \left(\frac{D}{d} \right)^2 H_s$$

where $H_s = N \cdot d$ denotes the active solid height of the spring

In some instances, another formula can be used to good advantage:

$$T_s = \sqrt{\frac{m}{R}} = \sqrt{\frac{W}{g \cdot R}} = 0.051 \sqrt{\frac{W}{R}}$$

where $m = W/g$ is the mass of the spring

T_s increases with the active solid height and with the index of the spring. A spring of index $D/d = 7$ has a surge time of 1.74 ms/in (68.5 ms/m) of active solid height.

When both ends of the spring are supported, as is usual, one cycle of vibration comprises just twice the surge time T_s . Therefore, the natural frequency of the spring, measured in hertz is:

$$f = \frac{1}{2T_s} = 14,100 \cdot \frac{d}{D^2 \cdot N}$$

or

$$f = 14,100 \left(\frac{d}{D} \right)^2 \cdot \frac{1}{H_s}$$

or

$$f = 0.5 \sqrt{\frac{R}{m}} = 9.82 \sqrt{\frac{R}{W}}$$

The natural frequency decreases when the spring index and the solid height are increased. A spring of index $D/d = 7$ and an active solid height of 10 in (0.25 m) has a natural frequency of 29 Hz.

The natural frequency of valve springs as used in automotive engines tends to be somewhat lower than that calculated from the above formulas. Actual measurements reveal that the frequency may be as much as 10% less than that calculated, due evidently to the end effects of introducing more active coils during a spring surge.

NOTE: It is necessary to distinguish clearly between the natural frequency of the spring itself, as described above, and the natural frequency of a mass M lb-s²/in (kg-s²/m) attached to a spring of much smaller mass and of rate R lb/in (kg/m). The natural frequency of such a mass M is

$$f = \frac{1}{2\pi} \sqrt{\frac{R}{M}} = 0.159 \sqrt{\frac{R}{M}} \text{ hertz}$$

The mass M when attached to the spring will cause a static de-

flection

$$F = \frac{Mg}{R} \text{ in(m)}$$

The natural frequency of the mass M relative to the spring deflection F (and measured in cycles/minute) therefore is

$$f = 0.159 \left(60 \sqrt{\frac{g}{F}} \right) = \frac{188}{\sqrt{F}} \text{ cycles/min}$$

4. TEMPERATURE EFFECTS ON SPRINGS

Spring design becomes complicated for extreme temperature applications. Generally, the usual spring materials will operate satisfactorily at very low temperatures under static loading conditions. The combination of extreme cold temperature and impact loading represents a more severe situation; however, it may not be catastrophic because of the resiliency of the spring material.

Under high-temperature conditions, springs undergo load losses which in most instances are due to a decrease in the torsional elastic limit of the material. Springs may be designed to operate satisfactorily at normal temperatures, but under elevated temperatures these same springs will undergo a slow plastic deformation (commonly known as "relaxation" when the spring is at a fixed length, or as "creep" when spring is under a fixed load) and take a permanent set. The amount of set depends upon the working stress level, heat intensity, and the duration at the elevated temperature.

Thus, in any spring problem involving elevated temperature, the designer should be fully aware of the impending difficulties and make suitable allowances for all the factors involved. In general, the spring stresses should be reduced to the lower practical values. However, there are applications where such modifications are not possible, and in these instances special operations, such as "heat setting" or "hot pressing," may be used.

Heat setting consists of compressing the spring on a fixture, subjecting the compressed spring to a temperature higher than the desired operating temperature for a time suitable to insure complete penetration of the heat, and then cooling to room temperature before releasing.

Hot pressing consists of heating the spring in its free or relaxed position to some temperature in excess of the operating temperature for sufficient time to insure complete penetration; then, while the spring is at the temperature, it is compressed to some height below the installed or operating position and released.

A spring so treated by either method will, of course, take a permanent set. The spring is coiled to a longer free length than normally would be necessary, thus compensating for this set.

The temperature, time, and stress level of either method may be varied to minimize loss of load when subjected to the stresses and elevated temperatures subsequently imposed in operation.

For recommended guidelines on selecting materials to resist the effects of elevated temperatures, see Table 2 in Chapter II.

5. RESIDUAL STRESSES

Residual stresses (sometimes called "trapped stress") may be set up in springs in a variety of ways including heat treatment, shot peening, and presetting (scragging, cold setting). As a rule, the presence of residual stresses of proper sign makes possible the utilization of higher working stresses; in springs, this is of considerable advantage since the energy storage capacity increases as the square of the working stress. For coiled springs, the shot peening and presetting methods of inducing residual stresses are most practical.

5.1 SHOT PEENING - The shot peening process consists essentially of subjecting the spring to a stream of shot moving at high velocity. The peening action of the shot sets up beneficial biaxial compressive stresses in a thin surface layer and also results in cold working this layer. The depth of this cold-worked layer depends, among other things, upon the size and impingement velocity of the shot used, the direction of impingement relative to the surface, and the spring material. Shot peening, when properly conducted, results in relatively large increases of the stress range through which the spring can safely operate under fatigue or repeated loading conditions.

In commercial use, two types of methods are most adaptable. In the earlier method, air is used for the propelling agent as in the common sand blast equipment. The second method is mechanical, the shot being fed on a rapidly rotating wheel having radial blades. Thus, the energy of the wheel is transferred to the shot by the blades and by centrifugal force, the shot in turn impinging on the parts. Tests have shown that either method can produce the same results in increasing fatigue life, but the mechanical method is more economical for high production.

The increase in fatigue life is accomplished in two ways. One consists in stressing the surface layers in compression. Fatigue failures are due to tensile stresses and, if a compressive stress can be set up in advance where subsequently the load application will produce a tensile stress, the part will carry considerable load before it is actually stressed in tension. The second effect is the cold working and strengthening of the surface layers which raises the physical properties where the highest stresses occur. This second effect is of minor importance compared with the first. Because of the twofold beneficial results imparted by shot peening, this surface treatment has done more to increase the life of springs than any alloy steel or other process ever employed.

Shot peening can be successfully carried out on wire sizes in excess of 0.030 in (0.76 mm). But it will be effective only if it is thoroughly and evenly applied over the entire surface, with particular emphasis on those parts of the surface which will undergo the highest load stresses. Thus, on compression springs the inside of the coil is most important, whereas on torsion springs, it is the outside that is important. Time of ex-

posure to the shot is one of the prime factors to obtain effective peening, as indicated by the curve in Fig. 2. For each type of spring, a knowledge of this time factor, with a definite type and size of shot, is needed.

The effects of shot size and peening time can be checked by means of Almen strips and other observations. The height of the arc in the Almen strip is not in itself a measure of fatigue life, but it furnishes a check on uniform operation of the shot peening machine. The SAE Manual on Shot Peening (HS J808a) details the methods for controlling the machine with the help of Almen strips and for measuring coverage. The importance of proper peening coverage cannot be overemphasized.

Since shot peening is a mechanical working process, it can be affected by heat. If springs are heated sufficiently after shot peening (Fig. 3), the benefits will be reduced and the endurance limit will drop, eventually to that of the wire as originally received. The use of shot peened springs in applications involving high temperatures necessitates careful planning. If the spring is heated above 800°F (427°C) during manufacture, the maximum resistance to settling at temperatures usually encountered in automotive practice is obtained. But in this event, the beneficial shot peening stresses are completely removed, and no increase will occur in fatigue life due to the

shot peening operation. Therefore, compromises must be made if an endurance is to be realized which exceeds that of the untreated wire.

5.2 PRESETTING - Residual stresses are most commonly set up in helical compression springs by means of a presetting operation, known also as scragging or cold setting. In this process, the spring is initially coiled to a free length greater than the final desired free length and is then compressed beyond the elastic limit of the material to a point where the spring may be at or near the solid height (Fig. 4, point B). On releasing the load, a permanent set OC is produced while the spring behaves nearly elastically along line BC. In special instances, several additional compressions to point B may be carried out.

Fig. 5 illustrates the beneficial effect of the presetting operation on the stress distribution over the cross section of a round wire spring of large index. Such a spring approximates a straight bar in torsion, and curvature effects may be neglected. For small index springs, a more complicated residual stress pattern may be expected since the residual stresses on the inside of the coil are relatively greater.

Fig. 5A represents the stress distribution over the cross section at a load which gives a stress S somewhat below the yield point.

For purposes of illustration, it is assumed that no strain

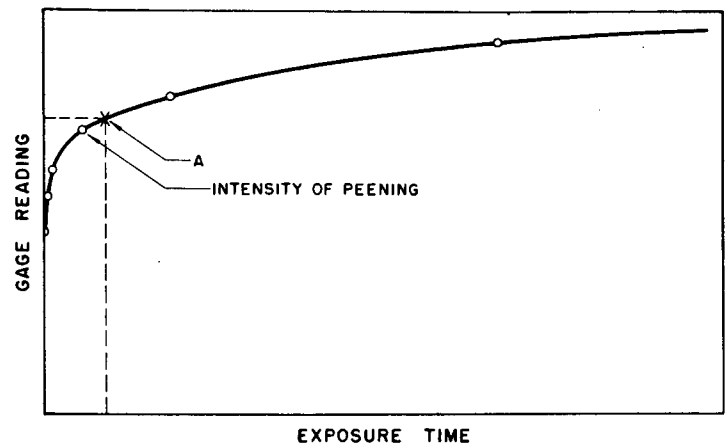


Fig. 2 - Intensity determination curve

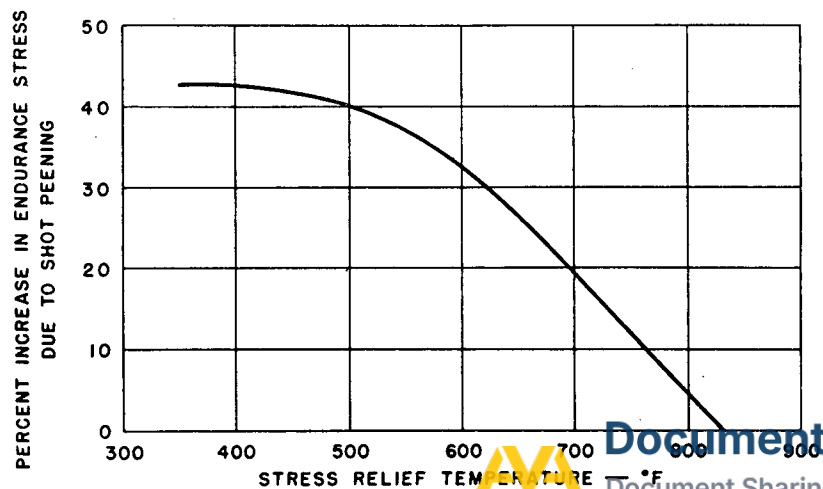


Fig. 3 - Reduction in effectiveness of shot peening by subsequent heating or stress relieving

hardening occurs, even under the higher presetting load (Fig. 4, point B) with which Fig. 5B deals. Here, yielding occurs at stress S , and the distribution of stress will be as indicated by the shaded area f-h-i-k. The nominal stress calculated from the presetting load by the usual formula is equal to S_n , and without yielding the stress distribution resulting from this presetting load would be as indicated by the triangular area f-l-k. Actually, yielding has started at the surface (wire diameter d)

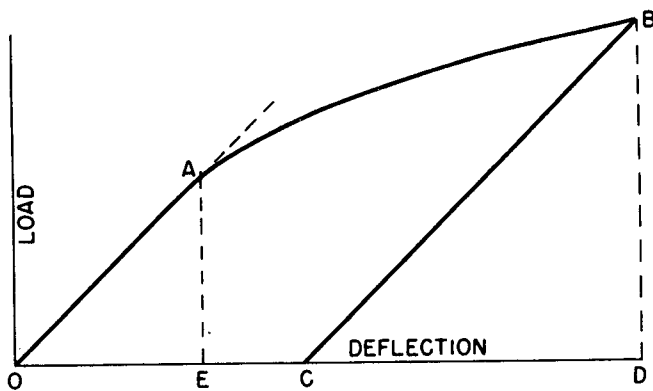


Fig. 4 - Typical load-deflection diagram of helical spring during presetting

and has progressed inward to a diameter

$$d_1 = d \cdot \sqrt[3]{4 - 3S_n/S_y}$$

In Fig. 5B, the ratio S_n/S_y has been chosen as 31/24. In that instance, d_1 becomes $d/2$.

When the presetting load is reduced to zero (Fig. 4, point C), residual stresses are set up as indicated by the shaded area in Fig. 5C. These are obtained by subtracting the ordinates of the dashed line f-l from those of the broken line f-h-i in Fig. 5B. The reason for this is that on removal of the load, the spring will behave approximately elastically. Therefore, removal of the load is equivalent to subtracting the elastic stress distribution f-l from the plastic distribution f-h-i.

If the spring is loaded subsequent to presetting, the resultant stress distribution is found by adding the load stress distribution to that of Fig. 5C. For example, a load may be applied to the spring which gives a stress S , the same as that shown in Fig. 5A. The resultant stress distribution under that load is indicated by the shaded area f-o-p-q in Fig. 5D. It will be seen that, due to the presetting operation in this instance, the peak stress has been reduced from a value S (Fig. 5A) to a lower value S_4 (Fig. 5D).

Still dealing with a round wire helical compression spring of large index and still assuming that no strain hardening occurs,

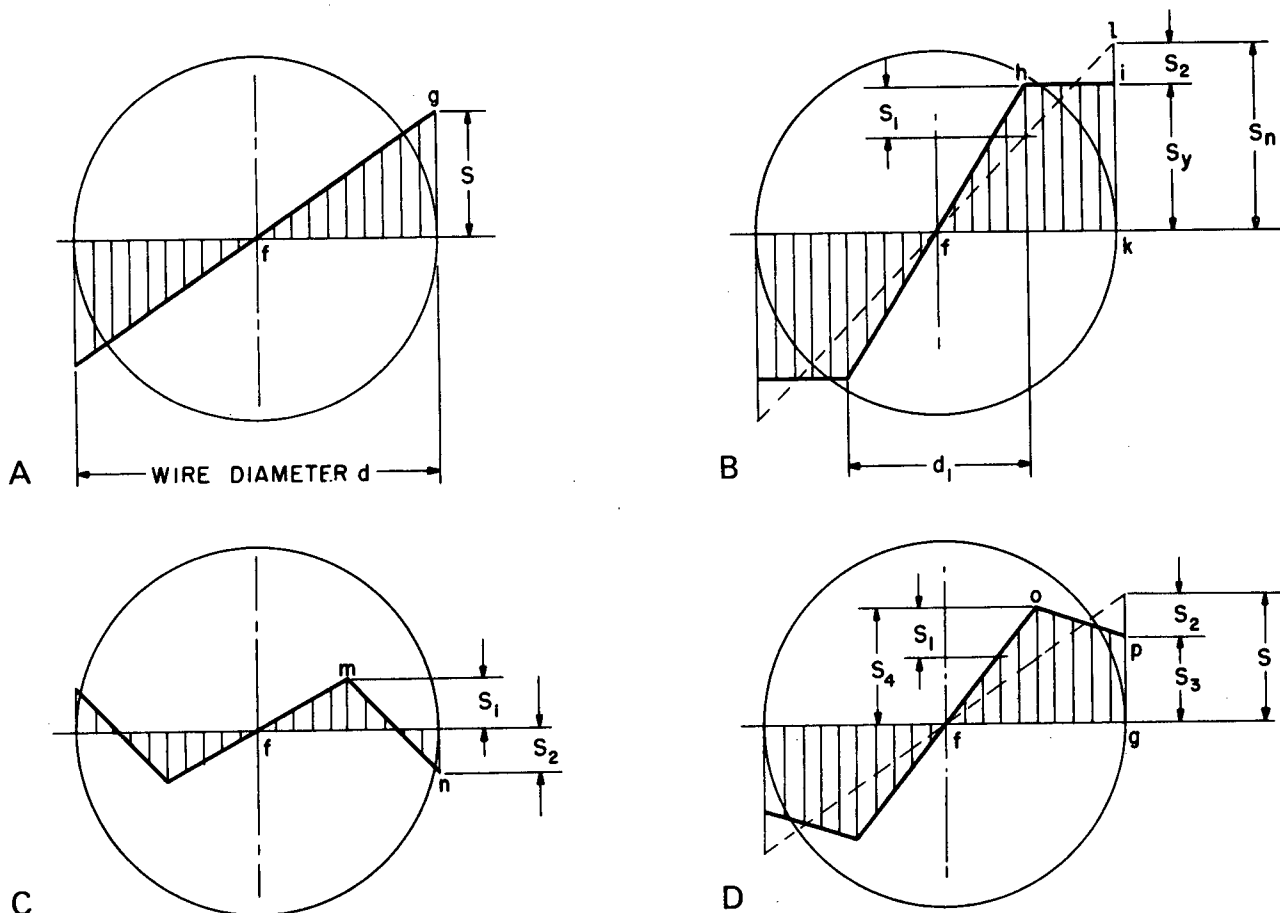


Fig. 5 - Effect of presetting on stress distribution over cross section of helical spring



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it may be considered that the material yields at an essentially constant stress equal to the shearing yield point and that complete yielding, over the entire cross section, will occur theoretically at a load equal to 1.33 times the load at which yielding first starts. This would indicate that, for these theoretical conditions, the maximum presetting load would be about 33% above the load corresponding to the torsional yield stress and that, after presetting, the spring would behave elastically up to a load corresponding to a stress 1.33 times the torsional yield point. Because of curvature and strain hardening effects, the ratio of the maximum presetting load to the load at which yielding first starts tends to be higher than 1.33 in actual springs, and, in some instances, for the smaller spring indexes, it may approach 2. Since the spring will behave elastically nearly up to the presetting load, this means that the elastic limit of a spring may be increased considerably by presetting; much of this increase is attributable to the residual stresses set up in the presetting operation. In actual practice, presetting deflections equal to 2, and in some instances even 3, times the elastic limit deflection may be used (OD = 2 to 3 times OE in Fig. 4). In other instances, however, lower values of presetting deflection may be required to avoid excessive distortion of the spring.

In cold wound helical torsion springs, residual stresses are also set up by the coiling operation; tension residual stresses are set up on the inside of the coil and compression stresses on the outside. If the spring is loaded by a torque so as to wind up the coil, it is clear that the residual stresses, in this instance, will also subtract from the maximum load stresses near the outer fibers. To take advantage of the residual stress pattern, torsion springs should therefore be loaded in the windup direction.

Residual stresses set up in cold-wound helical compression springs as a consequence of the winding process are usually removed or reduced by a low-temperature, stress-relieving (bluing) treatment. These stresses tend to augment the effects of the torsional working stresses in such springs, and hence their removal is beneficial.

If springs are subject to elevated temperatures during operation, residual stresses may be greatly reduced or even lost. For example, shot peening begins to lose its effectiveness at temperatures around 500°F (260°C), with complete loss at temperatures around 800°F (427°C) on steel springs. This indicates that, at these temperatures, residual stresses are considerably reduced, and the benefits of presetting will be reduced correspondingly.

6. FATIGUE DURABILITY

Some springs are required to function cyclically with accompanying repetitive stresses. Failure in these springs is commonly known as fatigue. Generally, these springs perform an essential function and should be designed to have high fatigue durability. This applies particularly to valve springs for internal combustion engines and to injector springs for diesel engines. Designing a spring of this type requires consideration

of stress ranges, environmental conditions, coordination of the surge time to the compression time, harmonic analysis, and specialized surface treatments such as shot peening.

When determining the wire size, it is not sufficient to check for maximum stress alone. It is essential that the stress range be within a safe limit to assure high endurance properties. (see Figs. 10 and 11.)

In the choice of the material for springs that are to have prolonged fatigue durability, first consideration should be given to carbon steel valve spring quality wire. In the usual sizes, it equals any alloy steel and is more dependable. In most instances, the use of pretempered steel is advisable to avoid hardening cracks that can occur in material hardened and tempered after coiling.

The alloy steels are usually more subject to seams and have greater tendency toward quench cracks than carbon steels. Therefore, it is advisable to subject such springs to magnaflux inspection. While this process is not ordinarily used in springs for the automotive industry, it is seldom omitted on aircraft engine springs.

7. COMMON CAUSES OF SPRING FAILURE

A spring designed and produced with due regard to all the foregoing details should not fail. In spite of this, a rare failure may occur on a properly designed spring. It can usually be traced to the following causes:

1. Surface defects - They occur as hardening cracks, seams, laps, pits, die marks, and tool marks. Of these, magnetic particle inspection will locate only the first three. The other defects must be found by careful inspection of the raw materials, usually by etching both ends of every bundle of wire and by visual inspection of the finished part. The difficult part of this procedure is that the small sharp pits or marks may be overlooked while the broad, round bottom marks, which are harmless, are easily observed. For critical springs, such as automotive valve springs, it is common practice to inspect the wire continuously by an electromagnetic device which rejects springs with defects.

2. Corrosion - This is a more common cause of spring breakage than is usually understood by the spring user. In valve springs, it may be caused by condensation of the products of combustion on the springs. A small pit is formed or hydrogen embrittlement has occurred, from the oxidation of the steel. It is not generally recognized how fast this process can lead to breakage. Springs which were perfect have failed in 3-6 h in an engine which was started and stopped before it could run long enough to scavenge the crankcase. In applications of this type, it will be necessary to provide corrosion protection for the springs. (Refer to section 8 on surface protection.)

3. Improper heat treatment - It is a threat which must not be overlooked, even though it is not frequently encountered. If springs are overheated, the grain structure will be coarse and the fatigue life poor. Failure to heat to a proper temperature, or to heat long enough to get all the carbides into solu-

tion, will result in ferrite patches and low endurance limit. Automatic time-temperature control in modern heat treating departments has almost completely eliminated this once common cause of breakage.

4. Decarburization - This is the least frequent offender in the entire list of defects. Partial decarburization is usually present in spring wire, at least to a slight extent. However, properly shot peened springs have the same life whether partially decarburized or not, provided no large ferrite areas are present. But, when the examination by a competent metallographer reveals a full ferrite ring around the circumference, the wire is subject to rejection because such a surface condition will ultimately lead to premature failure.

The foregoing data cover the general considerations which deal with fatigue durability springs, and some of the special methods which have been developed to meet the conditions where the consequences of breakage may be disproportionately serious.

8. SURFACE PROTECTION

The most widely used metals in the spring industry are the carbon and low-alloy steels. These are also the least corrosion resistant. The wide acceptance of these steels as spring material directs consideration toward choice of the proper type of finish for a particular spring application. The most important factors to be considered are:

1. Corrosive severity of the spring environment.
2. The degree of corrosion protection required to give spring life expectancy.
3. Effect of coating and its method of application on the mechanical properties of the spring. (See guidelines in following paragraphs to minimize hydrogen embrittlement.)
4. Comparative corrosion resistance and the cost of applying the coating.
5. Effect of the protective coating on associated members.
6. Availability of the protective materials and the equipment necessary for the application of this coating.

8.1 PROTECTION FROM MILDLY CORROSIVE ENVIRONMENT - For many applications, springs are coated with one of several slushing oils or rustproofing compounds. A more durable coating, if required, is a phosphate coating with rustproofing compound or organic finish. The organic finish can be compounded with low, medium, or high oil content in order to give the degrees of protection required, though baking may be required to dry the springs coated with compounds containing the higher oil content. An example of an organic coating with a low oil concentration is the asphaltic-based material known as Gilsonite paint or Japan Black, while the medium and high oil concentration paints are alkyd enamels. The addition of zinc chromate pigment to these synthetic enamels may be used to further enhance their corrosive protection.

When springs are protected by applying a zinc or iron phosphate coating, the thickness of the coating depends, among other factors, on the precleaning process. When an emulsion

cleaner is used, a very fine coating is formed; an alkaline cleaner produces a thicker, hard, tight coating; if an acid is used, the coating is coarse and very thick. A uniform coating depends on good cleaning and chemical control. This phosphate coating is a basis for the subsequent coating with a rust preventive oil, wax, grease, or organic finish providing good protection from corrosion. The mechanism of these coatings is the ability of the minute pits, surface irregularities, and the crystalline structure of the zinc and iron phosphate crystals to retain the subsequent coating, thereby providing maximum corrosion protection. The phosphating process must be tailored to protect the spring from acid pickling, both in the precleaning sequence and during the phosphating process by controlling the free acid ratio.

When the end use of the spring is in a mildly corrosive atmosphere, a black oxide finish provides some degree of corrosion protection. A black oxide coating provides some of the same mechanics to hold rust preventive materials as the zinc and iron phosphate coatings, but with an overall lesser degree of protection. When black oxide coatings are applied they have the advantage of causing very slight, if any, dimensional changes and no hydrogen embrittlement.

Frequently, it is advantageous to use precoated spring wire where mildly corrosive conditions are encountered. These wires are coated by dipping into molten metal or by electroplating. They may be coated and then drawn to size, or they may be coated at the finished size. Cadmium, tin, and zinc are the metals employed for the wire coating process. The commercial importance of the process is considerable because the parts formed from such wires may be used immediately without having to undergo additional operations normally associated with the plating of finished parts. In the case of extension springs, the use of precoated wire is preferred because the tightly wound coils prevent applying a satisfactory coating after coiling.

8.2 ELECTROPLATING - Many springs are being plated with a variety of metals of which cadmium and zinc have generally proved to be the most satisfactory.

Cadmium and zinc plate are bluish-white metals having good ductility and providing a protective coating on springs for most applications. When properly applied, cadmium or zinc plating does not flake or peel. They protect the base metal by sacrificial galvanic action. The protective life of both zinc and cadmium plate may be extended by adding a chromate coating. The degree of protection will still be dependent upon the thickness of the plate which may vary from 0.00015-0.0005 in (0.0038-0.013 mm). Neither of these metals is resistant to attack from strong acids or alkalis. Both zinc and cadmium are poisonous when vaporized. Neither should be used on springs which come in contact with food or drink.

The end use of the spring and the environment determines which coating is to be applied. When either zinc or cadmium is indicated, zinc may be chosen because it is less expensive; the choice may be cadmium because of the longer time before producing white corrosion products. Cadmium can be baked at a higher temperature for relief of hydrogen embrittlement.

without the hazard of blistering. Copper plate on springs should be used only for identification purposes as it provides only slight corrosion protection for steels. The use of a copper, nickel, or chrome plate may be indicated for decorative reasons but is not generally applicable. Springs requiring nickel plate may be plated using the sulfamate, Watts, or electroless process. When the nickel corrodes, it forms green nickel corrosion products, and the protection depends on the thickness and continuity of the plate. Plated nickel is dense and prevents the escape of hydrogen which enters the steel during the precleaning, pickling, and plating processes. Hard chrome plating of springs is generally unsatisfactory because

of the heavy thickness (0.002 in (0.051 mm) or more) required to give corrosion protection. Chrome plate is hard with low ductility; therefore, it is subject to peeling and cracking while causing reduction in spring fatigue life.

When springs are exposed to severe environmental conditions such as salt water, acid, and caustic atmosphere or solutions, the most resistant coating is lead alloy. Lead coatings should be 0.001 in (0.025 mm) in thickness to afford maximum protection. Heavy tin coatings, exceeding 0.001 in (0.025 mm) in thickness, should be used for springs coming in contact with foodstuff because of the resistance of tin to weak organic acids and its low toxicity. Lead and tin coat-

Table 1 - Corrosion Protection for Springs

Criteria for Determining Protection to be Used	Recommended Protection in Order of Preference	Dimensional Buildup, Minimum/Surface, in (mm)	Methods of Applying Protection	Special Advantages, Precautions, and Limitations
Directly exposed to weather, not making close fits	Phosphate coat and paint, or paint alone	0.001-0.002 (0.025-0.051)	Dip, spray, or brush	Close fits will peel paint
Directly exposed to weather, making close fits	Cadmium or zinc plate	0.0005 (0.013)	Electroplate	Corrosion protection good. Relief of hydrogen embrittlement essential after electroplating*
Not directly exposed to weather, making close fits	Cadmium or zinc plate	0.00015 (0.0038)	Electroplate	Same as above
Not directly exposed to weather, making close fits	Phosphate coat plus oil	0.0003-0.0008 (0.008-0.020)	Dip	Corrosion protection poorer than phosphate coat plus paint
Relatively mild corrosive conditions	Precoated spring wire	Thickness is variable and is dependent on size	Hot dip or electroplate	Fair all purpose coating applied to spring wire suitable for mildly corrosive conditions
Where corrosive conditions are very mild	Oxide black	Less than 0.0002 (0.005)	Alkaline oxidizing solution, 30-60 min at 300°F (139°C)	
Mildest corrosive conditions	Slushing oil or rust-proofing compounds	Dependent on method of application and/or viscosity of material	Dip or spray	Good for protection prior to installation and for short period after installation

*See text for alternate methods of plating and recommendation for relief of hydrogen embrittlement.

ings are electropositive to steel (more noble in the electromotive series); therefore, these coatings will only protect the steel spring when the coating is continuous and of a sufficient thickness. Lead and tin plates are soft materials; therefore, abrading of the coating must be avoided in order to provide the necessary continuous thick protective coating.

8.3 TECHNIQUES FOR MINIMIZING HYDROGEN EMBRITTLEMENT - Hydrogen may be introduced either during the cleaning steps in preparation for plating or it may be introduced during the plating process. If springs can be fabricated without producing heavy scale or film, the first hazard will be greatly reduced.

Residual stresses should be reduced as much as possible before plating.

If scale removal is required, a mechanical process is preferred. Unless the parts would distort, they may be blasted with shot, grit, glass, aluminum oxide, etc., or they may be tumbled or vibrated with various abrasive media.

When either a sulfuric or hydrochloric acid solution is used to remove scale from springs, the use of a suitable organic pickle inhibitor will eliminate excessive pickling and tend to reduce hydrogen embrittlement. Springs with a hardness over Rc 40 should not be descaled in acid.

The resistance of the passage of hydrogen from the steel increases as the thickness of the plate increases; therefore, excessive plate thickness should be avoided as well as the conditions producing hydrogen embrittlement. The following procedure should be followed to achieve this. The anodic or cathodic cleaning and pickling cycle should be as short as possible to reduce possible cracking of the spring in highly stressed areas. The zinc or cadmium plate should be initially deposited at a low amperage until the springs are completely coated. The amperage should then be raised, as the plating operation is continued until the specified thickness is achieved.

In spite of these precautions, springs which have been cadmium or zinc plated have been exposed to a severe hydrogen embrittlement environment during the preplate and plating steps. To produce a satisfactory spring, conditions must be

set up to force the hydrogen to pass from the steel spring through the cadmium or zinc plate. The time and temperature required to relieve hydrogen embrittlement may vary from 1 h at 350°F (177°C) to as much as 24 h or longer at 425°F (219°C) depending on the tensile strength, hardness, residual stress, size of the spring, and thickness of the zinc or cadmium plate. This operation should be performed as soon as possible after plating.

The following processes for applying metallic coatings may be used with no exposure to hydrogen or substantially less exposure than occurs with conventional electroplating: mechanical plating, peen plating, vacuum plating, and low-embrittling electroplating.

8.4 EFFECT OF FINISH ON ENVIRONMENT - The choice of finish depends also on the conditions encountered in use. Each individual application presents a different problem since the type of protective finish suitable for one may be wholly inadequate for another. For example, consider the question, "Should zinc- or cadmium-coated springs be used in hydraulic systems using petroleum hydraulic fluids?" While the zinc- or cadmium-coated spring will function perfectly for years in this medium, conditions can develop causing the petroleum hydraulic fluid to break down producing acid, sludge, and varnish by-products. These breakdown products are caused by temperature and time. When a breakdown occurs, it will be accelerated by the presence of zinc or cadmium. Zinc or cadmium metal will act as a catalyst speeding up the formation of acids, sludges, and varnishes. These conditions may cause little or no etching of the steel spring but will prevent normal operation of the hydraulic system due to the sludge and varnish deposits.

It is essential to consider all of the factors involved in order to select the finish best suited for a particular purpose. If possible, the spring maker should be consulted before final decision, so as to coordinate the finish with the manufacturing procedure.

Table 1 can be used as a general guideline for selecting finishes.