SPRINGS, THEIR DESIGN AND CALCULATIONS

12. SPRINGS, THEIR DESIGN AND CALCULATIONS

12.1. *SPRINGS AND THEIR PROPERTIES*

Well-functioning springs are one of the most important prerequisites of a good die function. After all, what good is the drawing operation if the part cannot be stripped off the punch because there is not enough spring power behind the pressure pad? Or—what kind of parts will emerge from a die where the spring stripper is not spring-loaded adequately?

If ample pressure is the absolute basic of a good die operation, then springs are the most vital parts of every die.

12.1.1. Spring Materials

Springs are elements designed to withstand great amounts of deflection and return to their original shape and size on its release. To be capable of such cyclical loading, spring materials must possess very high elastic limits.

Often materials not specifically made for the spring application are utilized for that purpose because their elastic limits are within the above requirements. Steels of medium-carbon and high-carbon content are considered good spring materials. Where a copperbase alloy is required, beryllium copper and phosphor bronze are utilized.

The surface quality of the spring material has a considerable influence on the function of a spring, namely, on its strength and fatigue. Where possible, the surface finish has to be of the highest grade, preferably polished. This is especially important with closely wound springs, where friction between single coils may create minute defects in their surface, which subsequently will cause the spring to crack. Music wire, the highest-quality spring material, is polished, and its surface is almost defect-free.

Of course, the higher quality the material, the more expensive it is. The designer should strive to find the best combination of price versus quality for each particular job.

A brief description of basic spring materials is included in Table 12-1, which provides a rough comparison of properties, usefulness, and some specific aspects. (Additional properties of spring temper alloy steel are presented later in Table 12-8.)

12.1.1.1. *High-Carbon Spring-Steel Wire.*

This group of spring materials is lowest in cost, which may account for its widespread use. It does not take impact loading or shock treatment well. Also it should not be used in extreme temperatures, high or low. Main representatives of this group are listed, with the percent of carbon (C) given.

Music wire, ASTM A228 (0.80 to 0.95 percent C). Good for high stresses caused by cyclic repeated loading. A high-tensile-strength material, available as (cadmium or tin) preplated.

	Young's modulus E[a]		Modulus of rigidity G[a]		Density[a]		Electrical conductivity,[e]	Sizes normally available[b]				Typical surface	Maximum service temperature[d]	
								Min.		Max.				
Common name	MPa 10^3	lb/in^2 10^6	MPa 10^3	lb/in^2 10^6	g/cm^3	(lb/in^3)	% IACS	mm	(in)	mm	(in)	quality[c]	°C	°F
Carbon-steel wires:														
Music[e]	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.10	(0.004)	6.35	(0.250)	1	120	250
Hard drawn[e]	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.13	(0.005)	16	(0.625)	3	150	250
Oil tempered	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.50	(0.020)	16	(0.625)	3	150	300
Valve spring	207	(30)	79.3	(11.5)	7.86	(0.284)	7	1.3	(0.050)	6.35	(0.250)	1	150	300
Alloy-steel wires:														
Chrome vanadium	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.50	(0.020)	11	(0.435)	1,2	220	425
Chrome silicon	207	(30)	79.3	(11.5)	7.86	(0.284)	5	0.50	(0.020)	9.5	(0.375)	1,2	245	475
Stainless-steel wires:														
Austenitic type 302	193	(28)	69.0	(10)	7.92	(0.286)	2	0.13	(0.005)	9.5	(0.375)	2	260	500
Precipitation hardening 17-7 PH	203	(29.5)	75.8	(11)	7.81	(0.282)	2	0.08	(0.002)	12.5	(0.500)	2	315	600
NiCr A286	200	(29)	71.7	(10.4)	8.03	(0.290)	2	0.40	(0.016)	5	(0.200)	2	510	950
Copper-base alloy wires:														
Phosphor bronze (A)	103	(15)	43.4	(6.3)	8.86	(0.320)	15	0.10	(0.004)	12.5	(0.500)	2	95	200
Silicon bronze (A)	103	(15)	38.6	(5.6)	8.53	(0.308)	7	0.10	(0.004)	12.5	(0.500)	2	95	200
Silicon bronze (B)	117	(17)	44.1	(6.4)	8.75	(0.316)	12	0.10	(0.004)	12.5	(0.500)	2	95	200
Beryllium copper	128	(18.5)	48.3	(7.0)	8.26	(0.298)	21	0.08	(0.003)	12.5	(0.500)	2	205	400
Spring brass, CA260	110	(16)	42.0	(6.0)	8.53	(0.308)	17	0.10	(0.004)	12.5	(0.500)	2	95	200
Nickel-base alloys:														
Inconel alloy 600	214	(31)	75.8	(11)	8.43	(0.304)	1.5	0.10	(0.004)	12.5	(0.500)	2	320	700
Inconel alloy X750	214	(31)	79.3	(11.5)	8.25	(0.298)	1	0.10	(0.004)	12.5	(0.500)	2	595	1100
Ni-Span-C	186	(27)	62.9	(9.7)	8.14	(0.294)	1.6	0.10	(0.004)	12.5	(0.500)	2	95	200
Monel alloy 400	179	(26)	66.2	(9.6)	8.83	(0.319)	3.5	0.05	(0.002)	9.5	(0.375)	2	230	450
Monel alloy K500	179	(26)	66.2	(9.6)	8.46	(0.306)	3	0.05	(0.002)	9.5	(0.375)	2	260	500

Figure . TABLE 12-1 Typical Properties of Common Spring Materials

Carbon-steel strip:														
AISI 1050	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.25	(0.010)	3	(0.125)	2	95	200
AISI 1065	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.08	(0.003)	3	(0.125)	2	95	200
AISI 1074, 1075	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.08	(0.003)	3	(0.125)	2	120	250
AISI 1095	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.08	(0.003)	3	(0.125)	2	120	250
Bartex	207	(30)	79.3	(11.5)	7.86	(0.284)	7	0.10	(0.004)	1	(0.040)	1	95	200
Stainless-steel strip:														
Austenitic types 301, 302	193	(28)	69.0	(10)	7.92	(0.286)	2	0.08	(0.003)	1.5	(0.063)	2	315	600
Precipitation hardening 17-7 PH	203	(29.5)	75.8	(11)	7.81	(0.282)	2	0.08	(0.003)	3	(0.125)	2	370	700
Copper-base alloy strip:														
Phosphor bronze (A)	103	(15)	43	(6.3)	8.86	(0.320)	15	0.08	(0.003)	5	(0.188)	2	95	200
Beryllium copper	128	(18.5)	48	(7.0)	8.26	(0.298)	21	0.08	(0.003)	9.5	(0.375)	2	205	400

[a] Elastic moduli, density, and electrical conductivity can vary with cold work, heat treatment, and operating stress. These varia tions are usually minor but should be considered if one or more of these properties is critical.

[b] Diameters for wire; thicknesses for strip.

[c] Typical surface quality ratings. (For most materials, special processes can be specified to upgrade typical values.)

1. Maximum defect depth: 0 to 0.5% of d or t .

2. Maximum defect depth: 1.0% of d or t .

3. Defect depth: less than 3.5% of d or t .

[d] Maximum service temperatures are guidelines and may vary owing to operating stress and allowable relaxation.

[e] Music and hard drawn are commercial terms for patented and cold- drawn carbon- steel spring wire.

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Oil-tempered MB grade, ASTM A229 (0.60 to 0.70 percent C). A general-purpose spring steel, frequently used in coiled form. It is not good with shock or impact loading. Can be formed in annealed condition and hardened by heat treatment. Forms a scale, which must be removed if the material is plated.

Hard-drawn MB grade, ASTM A227 (0.60 to 0.70 percent C). Used where cost is essential. Not to be used where long life and accuracy of loads and deflections are important. Can be readily plated.

Oil-tempered HB grade, SAE 1080 (0.75 to 0.85 percent C). With the exception of a higher carbon content and higher tensile strength, this spring steel is almost the same as the previously described MB grade. It is used for more precise work, where a long life, high fatigue, and high endurance properties are needed. If such aspects are not required, an alloy spring steel should be used in replacement.

12.1.1.2. *High-Carbon Spring-Steel Strip.*

The two main types of springs steel in this group are used with an absolute majority of all flat spring. However, both are susceptible to hydrogen embrittlement even when plated and baked afterward.

Cold-rolled blue-tempered spring steel, SAE 1074, plus 1064 and 1070 (0.60 to 0.80 percent C). This steel can be obtained in its annealed or tempered condition. Its hardness should be within 42 to 46 Rockwell hardness Scale C.

Cold-rolled, blue-tempered spring steel, SAE 1095 (0.90 to 1.05 percent C). It is not advisable to purchase annealed, as this type of steel does not always harden properly and spring properties obtained after forming may be marginal. Its hardness range is 47 to 51 Rockwell hardness Scale C.

12.1.1.3. *Alloy Spring Steel.*

A good spring steel for a high-stress application, with impact loading and shock application involved.

Chromium vanadium steel, ASTM A231 takes higher stresses than high-carbon steel. It also has a good fatigue strength and endurance.

Chromium silicon steel, ASTM A401. This material can be groomed to high tensile stress through heat treatment. Applicable where long life is required in combination with shock loading.

12.1.1.4. *Stainless Spring Steel.*

A corrosion-resistant material. With the exception of the 18-8 type, none of these steels should be used for lower-than-zero temperature applications. High-temperature tolerance is up to 550°F.

Stainless spring steel 302, ASTM A313 (18 percent Cr, 8 percent Ni). This material has quite uniform properties and the highest tensile strength of the group. It can be obtained as cold drawn, since it cannot be hardened by heat treatment. The slight magnetic properties are due to cold working, as in annealed form it is nonmagnetic.

Stainless spring steel 304, ASTM A313 (18 percent Cr, 8 percent Ni). Because of its slightly lower carbon content, this material is easier to draw. Its tensile strength is somewhat lower than that of type 302, even though their other properties coincide.

Stainless spring steel 316, ASTM A313 (18 percent Cr, 12 percent Ni, 2 percent Mo). Less corrosion-prone than the 302 type stainless, with its tensile strength about 12 percent lower. Otherwise it is quite similar to the 302 type.

Stainless spring steel 17-7 PH, ASTM A313 (17 percent Cr, 7 percent Ni, with trace amounts of aluminum and titanium). The tensile strength of this material is almost as high as that of music wire. This is achieved through forming in a medium hard condition and precipitation hardening at low temperatures.

Stainless spring steel 414, SAE 51414 (12 percent Cr, 2 percent Ni). Its tensile strength is approximately the same as that of type 316 (above), and it may be hardened through heat treatment. In a high-polished condition this material resists corrosion quite well.

Stainless spring steel 420, SAE 51420 (13 percent Cr). May be obtained in the annealed state, hardened and tempered. Scales in heat treatment. Its corrosion-resistant properties emerge only after hardening. Clear bright surface finish enhances its corrosion resistance.

Stainless spring steel 431, SAE 51431 (16 percent Cr, 2 percent Ni). This material has very high tensile properties, almost on a par with music wire. Such a characteristic is achieved through a combination of heat treatment, followed by cold working.

12.1.1.5. *Copper-Base Spring Alloys.*

This group of spring materials is more expensive than alloy steels or high-carbon materials. They are, however, very useful for their good corrosion resistance and superb electrical properties. An additional advantage is their usefulness in lower-than-zero temperatures.

Spring brass, ASTM B134 (70 percent Cu, 30 percent Zn) cannot be hardened by heat treatment and has generally quite poor spring qualities. Even though it does not tolerate temperatures higher than 150°F, it performs well at subzero. It is the least expensive copperbase spring material, with the highest electrical conductivity, out-weighed by its low tensile strength.

Phosphor bronze (a tin bronze), ASTM B159 (95 percent Cu, 5 percent Sn). This is the most popular copper-based spring material. Its popularity is due to its favorable combination of electrical conductivity, corrosion resistance, good tensile strength, hardness, and low cost.

Beryllium copper, ASTM B197 (98 percent Cu, 2 percent Be) is the most expensive material of this group. It is better formed in its annealed condition and precipitation hardened afterward. The hardened material turns brittle and does not take additional forming. The material has a high hardness and tensile strength. It is used where electrical conductivity is of importance.

12.1.1.6. *Nickel-Base Spring Alloys.*

These alloys take both extremes in temperature, extremely hot and extremely cold, while being corrosion-resistant. For their high resistance to electricity the materials should not be used with electric current. Their field of application lies with precise measuring instruments such as gyroscopes.

Monel (67 percent Ni, 30 percent Cu) cannot be hardened by heat treatment. Its high tensile strength and hardness is obtained through cold drawing and cold rolling. It is almost nonmagnetic and withstands stresses comparable to those beryllium copper can handle. It is the least expensive material of this group.

K-Monel (66 percent Ni, 29 percent Cu, 6 percent Al). The material is nonmagnetic, and the small amount of aluminum makes it a precipitation-hardening applicant. Otherwise it is very similar to previously described monel. It can be formed soft and hardened afterward by application of an age-hardening heat treatment.

Inconel (78 percent Ni, 14 percent Cr, 7 percent Fe) has higher tensile strength and hardness than K-monel, both of these properties being attributable to cold drawing and cold rolling, as it cannot be hardened by heat treatment. It can be used at temperatures of up to 700°F. It is a very popular alloy because of its corrosion resistance, even though its cost is higher than that of the stainless-steel group, yet not so costly as beryllium copper.

Inconel-X (70 percent Ni, 16 percent Cr, 7 percent Fe, with small amounts of titanium, columbium, and aluminum). This nonmagnetic material should be precipitation hardened at high temperatures. It is operable up to 850°F.

Duranickel (98 percent Ni) takes slightly lower temperatures than inconel. It is non-magnetic, resistant to corrosion, and has a high tensile strength. It can be precipitation hardened.

12.1.2. Heat Treatment of Springs

Heat treatment of finished springs is done in two stages. First, following the forming process, a low-temperature heat treatment of 350 to 950°F (175 to 510°C) is applied. Such a treatment causes the material to stabilize dimensionally, while removing some residual stresses developed during the forming operation. Residual stresses come in two groups: Some of them are beneficial to the part's functionality; others are detrimental to it.

A second heat treatment is done at higher temperatures, ranging between 1480 and 1650°F (760 and 900°C). This heat treatment strengthens the material, which is still annealed after forming. Typical heat-treatment temperatures for specific materials are shown in Table 12-2. Usually, a 20 to 30-min-long exposure to these temperatures is considered adequate.

Table 12-2. Typical Heat Treatment for Springs after Forming

		Heat treatment	
Materials		°C	°F
Patented and cold-drawn steel wire		190–230	375–450
Tempered steel wire:			
	Carbon	260–400	500–750
	Alloy	315–425	600–800

Austenitic stainless-steel wire	230–510	450–950
Precipitation-hardening stainless wire (17-7 PH):		
Condition C	480/1 h	900/1 h
Condition A to TH 1050	760/1 h, cool to 15°C, followed by 565/1 h	1400/1 h, cool to 60°F, followed by 1050/1 h
Monel:		
Alloy 400	300–315	575–600
Alloy K500, spring temper	525/4 h	980/4 h
Inconel:		
Alloy 600	400–510	750–950
Alloy X-750:		
No. 1 temper	730/16 h	1350/16 h
Spring temper	650/4 h	1200/4 h
Copper-base, cold-worked (brass, phosphor bronze, etc.)	175–205	350–400
Beryllium copper:		
Pretempered (mill hardened) solution	205	400
Annealed, temper rolled or drawn	315/2–3 h	600/2–3 h
Annealed steels:		
Carbon (AISI 1050 to 1095)	800–830*	1475–1525*
Alloy (AISI 5160H 6150, 9254)	830–885*	1525–1625*

*Time depends on heating equipment and section size. Parts are austenitized, then quenched and tempered to the desired hardness.

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Hardened high-carbon steel parts, when electroplated, are prone to cracking. This is caused by the action of hydrogen atoms, which intermingle with the material's metallic lattice and affect its structure. Such an occurrence is called hydrogen embrittlement. To prevent hydrogen embrittlement in plated springs, heat treatment at low temperatures is used prior to plating, with a baking operation added after forming.

Beryllium copper is strengthened after forming by the application of an age-hardening process; with other materials, tempering may sometimes be utilized.

12.1.3. Corrosion Resistance

Coatings (zinc, cadmium, and their alloys) are frequently utilized to prevent corrosion damage to springs. These coatings not only act as a blockade between the material and the outside environment. They also protect the part cathodically, often even when scratched or otherwise topically damaged.

Electroplating is another method of protection used with application of metallic coatings. This type of surface finish, however, causes hydrogen embrittlement to appear, and care should be taken to minimize the part's susceptibility. As a means of protection, there should be no stress points in the part, such as sharp corners, sharp bends, or sharp-cornered cuts. Hardness should be at the minimum allowable level, and residual stresses within the material should be relieved by application of the highest possible heat-treating temperatures. After plating, parts should be baked at low temperatures for approximately 2 to 3 h.

Mechanical plating offers an adequate amount of protection against corrosion and hydrogen embrittlement as well. Such surface treatment should be used for parts suffering from high residual stresses after the forming operation. Its drawback lies in the difficulties with plating of tight or inaccessible areas—all part surfaces must be well exposed and clean.

12.1.4. Fatigue and Reliability

Fatigue in springs is a process that develops slowly and insidiously over the span of three stages: (1) crack induction, (2) crack increase, and (3) failure of the material. It is obvious that fatigue is an irreversible process, detrimental to the functionability of the part. Its development is caused by the emergence of cyclic stresses, accompanied by plastic strains, so common in springs. It may also be caused by the quenching process during springs manufacture.

Residual stresses, as found in the spring material after bending, may either increase or diminish its fatigue resistance. This variation in their influence is due to the fact that there actually are two types of residual stresses within the material.

Stresses which counterbalance those accompanying the spring operation are beneficial to the part's longevity. For example, in a compressed coil spring, where a residual tension is encountered at its core, some residual stresses of the compressive type should ideally be near its surface. A condition like this may create an environment within the material of the spring, allowing for increased loads and improving the spring's resistance to fatigue.

However, if the residual stresses are in another (opposite) direction, their contribution to the load-carrying capacity and fatigue resistance of the spring will be negative.

Favorable residual stresses are often introduced to the spring material by the spring manufacturer. After the first stress-relieving heat treatment, a slight plastic deformation is purposely caused to the parts, following the direction of the spring's own elastic deformation later in service. Unfortunately, such prestressing cannot be preformed with all springs, as its subsequent increase in production costs cannot always be justified.

Plated steel springs emerge from the plating operation free from residual stresses, which cannot be reintroduced afterward.

For removal of various residual stresses located near the surface, shot peening is utilized. This procedure, however, decreases the load-carrying capacity of the spring, as it lowers the material's yield strength.

Reliability is a fatigue-dependent value, where the decrease in the spring's reliability is always caused by defects produced by fatigue.

Reliability of springs operating at higher temperatures is negatively influenced by so-called *stress relaxation* . It is the decrease in the load-carrying capacity and deflecting capacity of a spring held or cycled under a load. Higher temperatures also affect the tensile strength, fatigue, and modulus of the material.

Stresses and high operating temperatures will in time produce stress relaxation in springs. In opposition to such an influence is the type of alloy: More alloyed materials were found less susceptible to the damage caused by temperature increases.

In static applications, the load-carrying ability of a spring may be impaired by its yield strength and resistance to stress relaxation. To increase the static load-carrying capacity, a longer than necessary spring length should be selected and precompressed to solid in assembly. This process is called *set removal* or *presetting* of the spring, and it may increase the load-carrying ability by 45 to 65 percent. By presetting the spring, favorable residual stresses are introduced into the material. Their type and direction correspond with the spring's own natural (elastic) deformation, attributable to its function.

12.2. *SPRINGS IN DIE DESIGN*

Types of springs most often used in die and fixture design are coil springs of the compression type. Marginally, extension coil springs and flat springs are utilized.

Compression springs are wound as an open helix (Fig. 12-1) with an open pitch to resist the compressive force applied against it. Overall shapes of these springs are most often straight and cylindrical. But variations in the outline and winding, such as barrel-shaped, conical, hourglass, and variable-pitch springs can be encountered (Fig. 12-2).

Extension springs form a tight helix, and their pitch is limited to the wire thickness (Fig. 12-3). Flat springs may come in many types and shapes (Fig. 12-4).

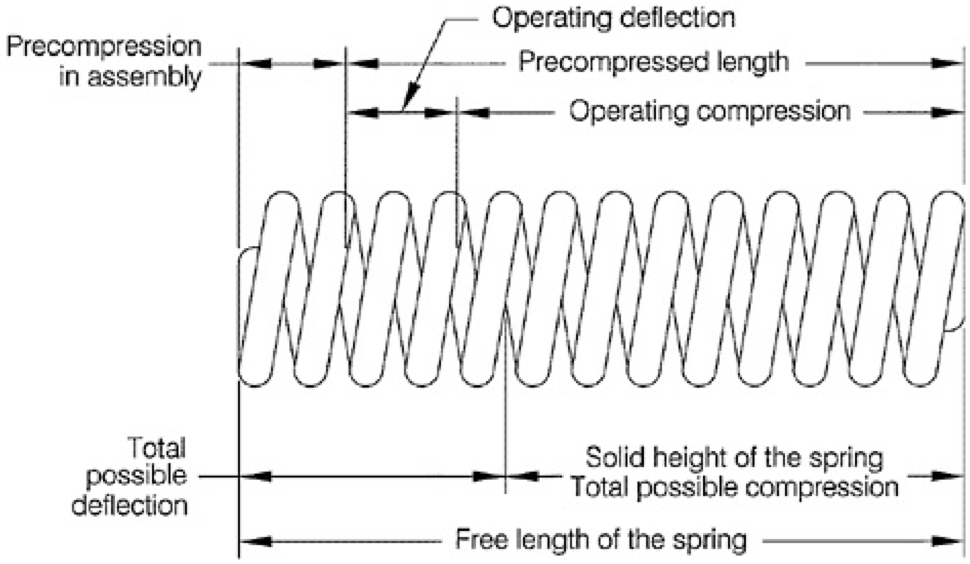

Figure 12-1. Compression spring and its properties.

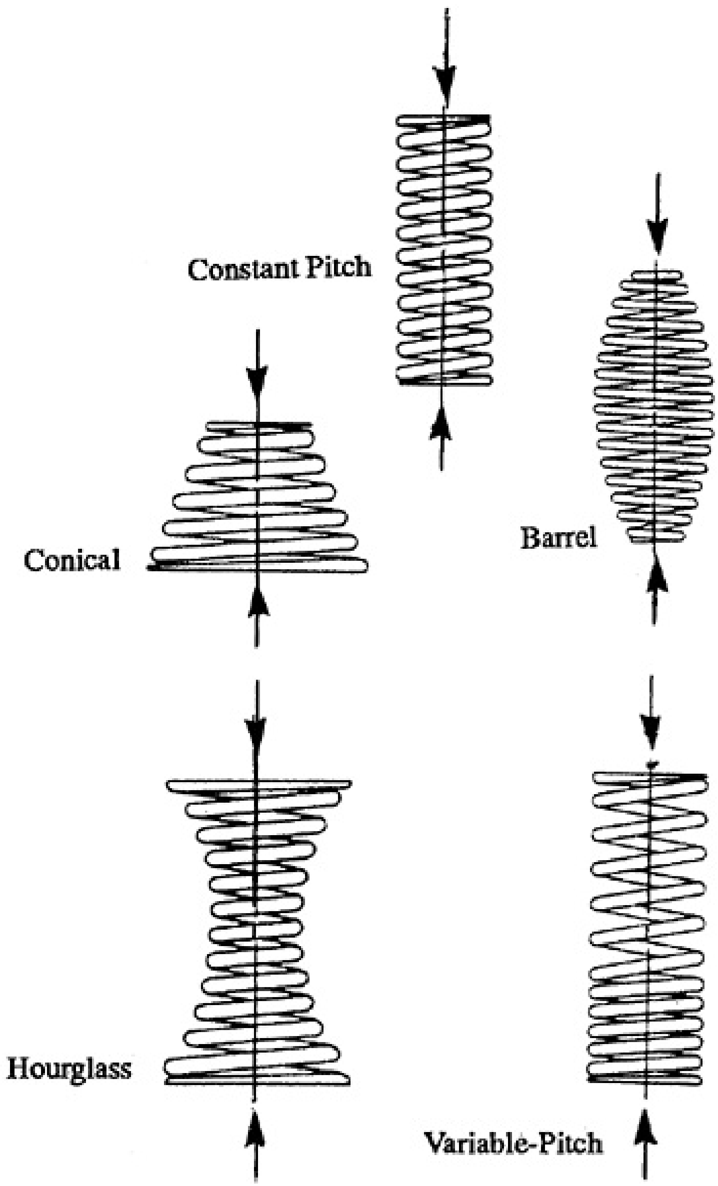

Figure 12-2. Helical compression springs, round and rectangular wire. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

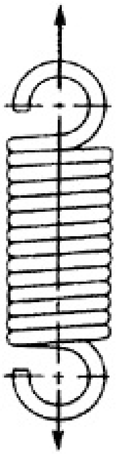

Figure 12-3. Helical extension spring. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

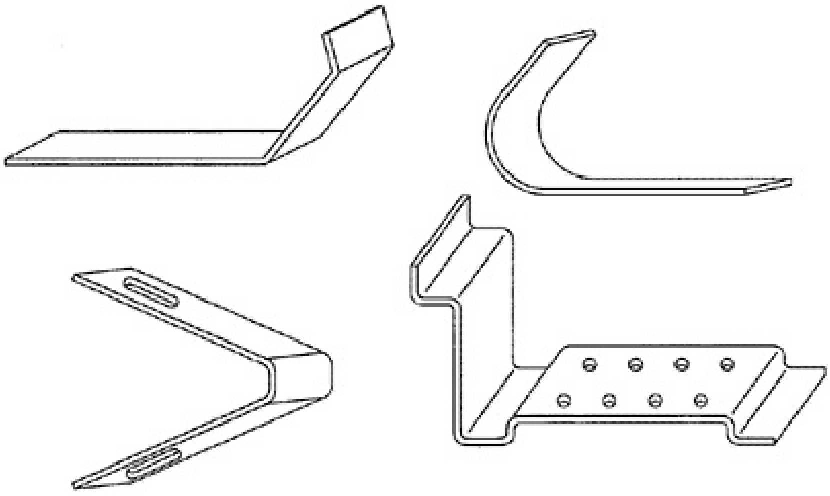

Figure 12-4. Flat spring samples.

12.3. *HELICAL COMPRESSION SPRINGS*

These are abundant in die and fixture design, being used to support spring pads, spring strippers, and other spring-loaded arrangements.

12.3.1. Spring-Related Terminology

A certain terminology has been developed over the years, describing various spring attributes, which is used throughout the industry. Terms like *spring diameter, mean diameter, pitch, squareness*, and *parallelism*, among others, are explained further in the text.

Spring diameter can be either the outside diameter (OD) or inside diameter (ID) or mean diameter (D) of the spring. Mean diameter is equal to the value of OD plus ID divided by 2. It is used for calculations of stress and deflection.

Where the OD is specified, the number is given with regard to the spring’s working environment, in this case the cavity, where the spring would be retained. With specification of ID, the size of the coil-supporting pin, which is to fit inside the coil, is important.

Minimum clearances between the spring and its cavity or between the spring and the supporting pin (per diameter) are

$0.10D$ where D_{cavity} is less than 0.512 in. (13 mm)

$0.05D$ where D_{cavity} is greater than 0.512 in. (13 mm)

This is to allow for the increase in diametral size which occurs with the load application on the spring. This increase, seen as a bulging of the spring, is usually quite small, yet it must be taken into account if the function of the spring is not to be impaired. To calculate the increase in size, the following formula is provided:

$$\mathrm{OD}_{\text{solid}} = \sqrt{D^2 + \frac{p^2 - d^2}{\pi^2}} + d$$

(12-1)

where the values are as shown in Fig. 12-5.

12.3.1.1. *Buckling of Compression Springs.*

Long springs may buckle unless they are supported by a pin coming through their center. Buckling may occur where the length of a spring unsupported by any pin exceeds the value of *four times its diameter* . Critical buckling conditions are given in Fig. 12-6. Critical buckling will occur with values to the right of each line.

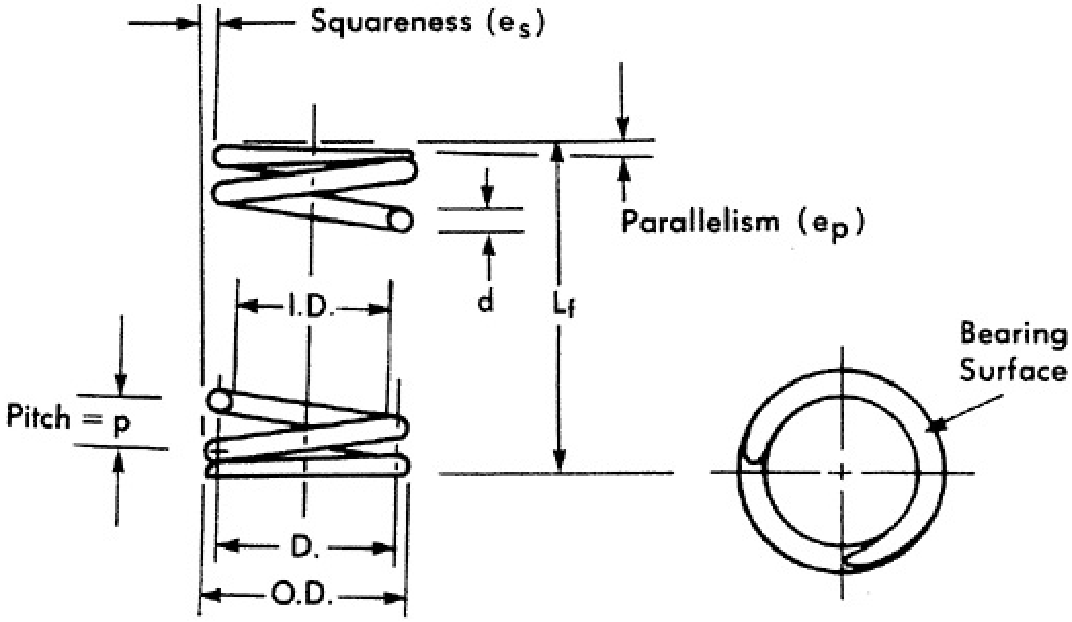

Figure 12-5. Dimensional terminology for helical compression springs. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

Curve *A* depicts those springs whose one end is positioned against a flat plate while its other end is free to tip, as shown in Fig. 12-7. This curve limits the occurrence of buckling conditions to the right and above its location.

Buckling occurrence is lower in springs retained between two parallel plates, as shown in section *B* of Fig. 12-7. *B*-line buckling, as observed in the graph in Fig. 12-6, is lessened accordingly.

In cases where large deflections are required, several springs supported by an inner core consisting of rods or shoulder screws may be utilized.

Spring index C is the ratio of mean diameter to the wire diameter, or

$$C = \frac{D}{d}$$

(12-2)

With spring cross sections other than round, this formula is altered as shown in Fig. 12-8.

The preferable spring index value is 4 to 12. Springs with high indexes may become tangled, requiring individual packaging for shipment, especially where their ends are not squared. Springs with indexes lower than 4 are difficult to form.

12.3.1.2. *Types of Spring Ends.*

A wide variety of spring ends may be selected, such as plain ends, plain ends ground, square ends, and square ends ground (Fig. 12-9).

A bearing surface of at least 270° serves to reduce buckling. Squared and ground spring ends have a bearing surface of 270 to 330°. Additional grinding of these ends is undesirable, as it may result in further thinning of these sections.

Springs with squared ends only, where no grinding is involved, are naturally cheaper. This type of end should be reserved to springs with

- Wire diameters less than 0.020 in. (0.5 mm)
- Index numbers greater than 12
- Low spring rates

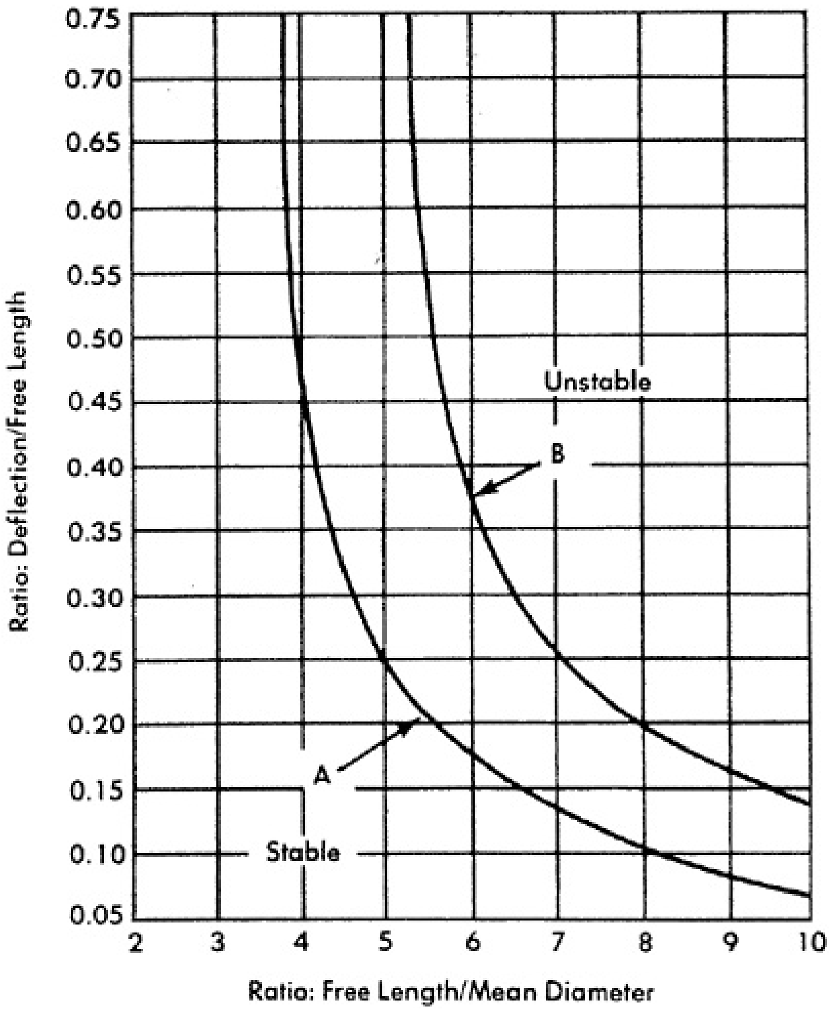

Figure 12-6. Critical buckling condition curves. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

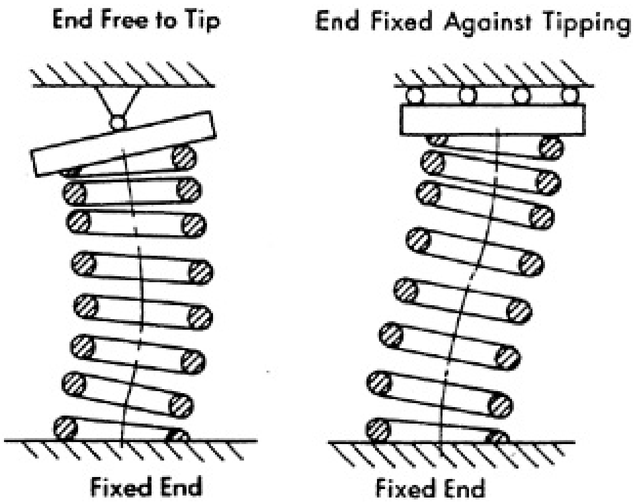

Figure 12-7. End conditions used to determine critical buckling. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

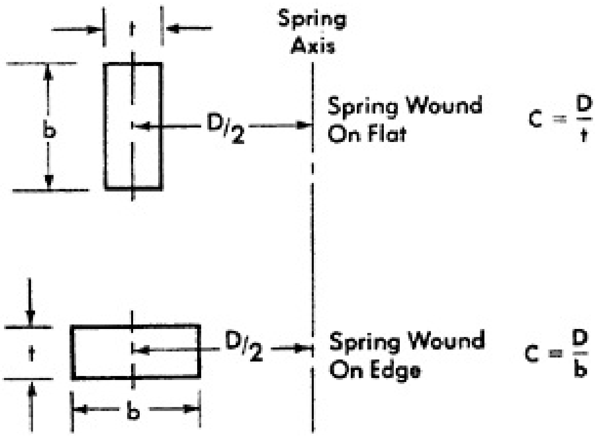

Figure 12-8. Rectangular wire compression spring wound on flat or edge. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

12.3.1.3. *Number of Active Coils N_A.*

Springs with squared ends have the number of active coils approximately equal to the total number of coils minus 2. Springs with plain ends usually have more of their coils active, the exact number being dependent on their seating method. Guidelines for selection of end types for a particular spring application are given in Table 12-3.

12.3.1.4. *Solid Height.*

Solid height of a spring is the length of all coils, pressed together. Solid height of a ground spring can be obtained by multiplying the coil diameter by the number of coils. Nonground springs have solid height equal to the number of coils plus 1, multiplied by the wire diameter.

Plating or other coating will increase the solid height of a spring. The safe amount to add for such an increase is approximately one-half of the wire diameter per spring.

12.3.1.5. *Direction of Coiling.*

Helical compression springs are either right- or left-hand wound (Fig. 12-10).

To assess the direction of coiling, the index finger of the right hand should be bent to resemble the shape of a coiled spring, with its tip ending in approximately the same location as the end of the coil. Such a spring, if matching the finger's arrangement, is right-hand-wound. An opposite-side arrangement is a left-hand-wound spring.

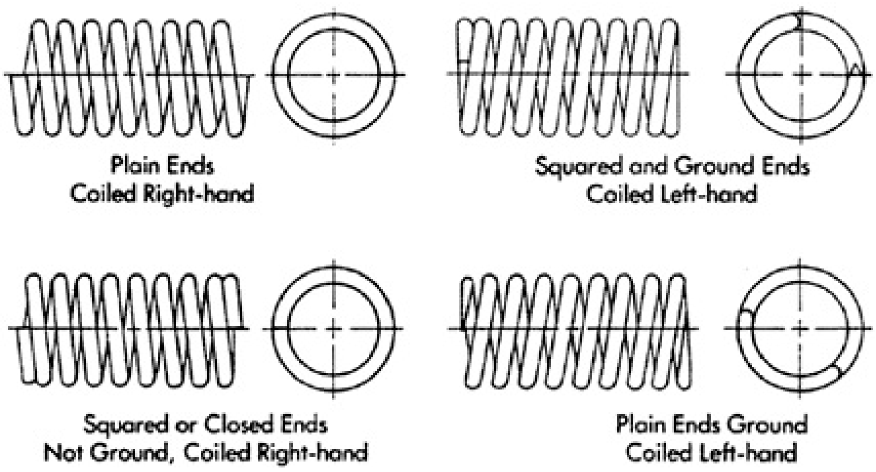

Figure 12-9. Types of ends for helical compression springs. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

Table 12-3. Guidelines for Dimensional Characteristics of Compression Springs

Dimensional characteristics	Type of ends: Open or plain (not ground)	Open or plain (ground)	Squared only	Squared and ground
Solid height L_s	$(N_t + 1)d$	$N_t d$	$(N_t + 1)d$	$N_t d^*$
Pitch p	$\frac{L_f - d}{N_a}$	$\frac{L_f}{N_t}$	$\frac{L_f - 3d}{N_a}$	$\frac{L_f - 2d}{N_a}$
Active coils N_a	$\frac{L_f - d}{p}$	$\frac{L_f}{p} - 1$	$\frac{L_f - 3d}{p}$	$\frac{L_f - 2d}{p}$
Total coils N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length L_f	$p\,N_t + d$	$p\,N_t$	$p\,N_a + 3d$	$p\,N_a + 2d$

*For small index springs lower solid heights are possible.

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12.3.1.6. *Squareness and Parallelism.*

Squared and ground springs are usually square within 3° when measured in their free form. However, squareness of free springs may differ from those under a load.

Parallelism has a considerable effect on the function of an unsupported spring. For illustration of squareness and parallelism, refer to Fig. 12-5.

12.3.1.7. *Hysteresis.*

Hysteresis means the loss of mechanical energy in a spring, which is exposed to cyclic loading and unloading within its elastic range. The known reason for such behavior is the friction between coils, or the friction between the spring and its support during compression.

12.3.2. Variable-Diameter and Variable-Pitch Springs

Variable-diameter springs are as shown in Fig. 12-2, respectively conical, hourglass, and barrel-shaped springs, are utilized where the solid height of the spring must be low or where greater lateral stability and resistance to surging are needed.

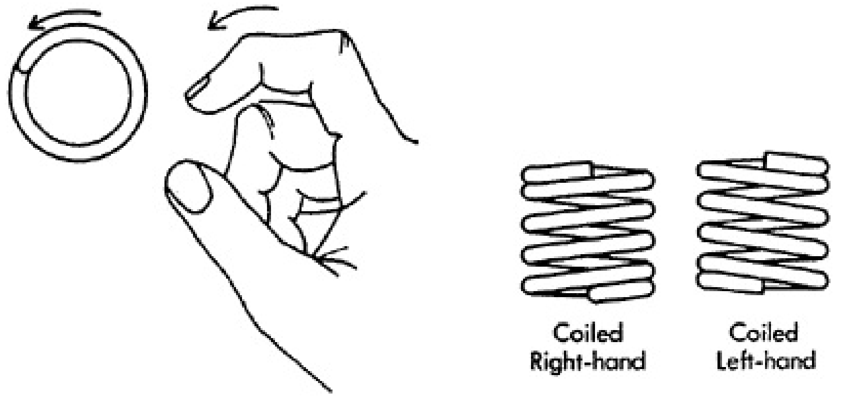

Figure 12-10. Direction of coiling of helical compression springs. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

The conical spring's solid height may be as low as one coil diameter, for these springs can be designed in such a fashion as to allow each coil to nest in the preceding coil. The spring rate can be made uniform by varying the pitch along the spring length.

When calculating the highest amount of stress at a predetermined load, the mean diameter of the largest active coil should be used.

Solid height of a spring L_S, made from a round wire, tapered in shape but not telescoping, with its ends squared and ground, can be estimated by using the formula

$$L_S = N_a\sqrt{d^2 - u^2} + 2d$$

(12-3)

where u is equal to

$$u = \frac{\text{OD large end} - \text{OD small end}}{2N_a}$$

where N_a is the number of active coils. To approach this calculation properly, each spring has to be considered to amount to several springs in series. Equation (12-11) may be used for such a purpose.

Barrel-shaped and hourglass springs can be calculated the same way, considering them to be two conical springs, which they incidentally are.

Variable-pitch springs are utilized where the natural spring frequency is near or corresponds with that of the cyclic rate of the load application. As coils of lesser pitch become inactive during the spring's function, the natural frequency of the spring will change. This will result in minimizing of surging and spring resonance. Spring resonance is addressed in Sec. 12-4-6.

12.3.3. Commercial Tolerances

Standard commercial tolerances for free length of a spring, diameter, and load are presented in Tables 12-4, 12-5, and 12-6. These tolerances are a good combination of the manufacturing costs and the spring's quality and performance.

Table 12-4. Free Length Tolerances of Squared and Ground Helical Compression Springs

	Tolerances: ± in./in. of free length						
	Spring index (D/d)						
Number of active coils per in. (mm)	**4**	**6**	**8**	**10**	**12**	**14**	**16**
0.02 (0.5)	0.010	0.011	0.012	0.013	0.015	0.016	0.016
0.04 (1)	0.011	0.013	0.015	0.016	0.017	0.018	0.019
0.08 (2)	0.013	0.015	0.017	0.019	0.020	0.022	0.023
0.2 (5)	0.016	0.018	0.021	0.023	0.024	0.026	0.027
0.3 (8)	0.019	0.022	0.024	0.026	0.028	0.030	0.032
0.5 (12)	0.021	0.024	0.027	0.030	0.032	0.034	0.036
0.6 (15)	0.022	0.026	0.029	0.032	0.034	0.036	0.038
0.8 (20)	0.023	0.027	0.031	0.034	0.036	0.038	0.040

For springs less than 12.7 mm (0.500 in) long, use the tolerances for 12.7 mm (0.500 in).
For closed ends not ground, multiply above values by 1.7.
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Table 12-5. Coil Diameter Tolerances of Helical Compression and Extension Springs

Wire dia, mm (in.)	Tolerances: ± mm (in.) Spring index (D/d) 4	6	8	10	12	14	16
0.38	0.05	0.05	0.08	0.10	0.13	0.15	0.18
(0.015)	(0.002)	(0.002)	(0.003)	(0.004)	(0.005)	(0.006)	(0.007)
0.58	0.05	0.08	0.10	0.15	0.18	0.20	0.25
(0.023)	(0.002)	(0.003)	(0.004)	(0.006)	(0.007)	(0.008)	(0.010)
0.89	0.05	0.10	0.15	0.18	0.23	0.28	0.33
(0.035)	(0.002)	(0.004)	(0.006)	(0.007)	(0.009)	(0.011)	(0.013)
1.30	0.08	0.13	0.18	0.25	0.30	0.38	0.43
(0.051)	(0.003)	(0.005)	(0.007)	(0.010)	(0.012)	(0.015)	(0.017)
1.93	0.10	0.18	0.25	0.33	0.41	0.48	0.53
(0.076)	(0.004)	(0.007)	(0.010)	(0.013)	(0.016)	(0.019)	(0.021)
2.90	0.15	0.23	0.33	0.46	0.53	0.64	0.74
(0.114)	(0.006)	(0.009)	(0.013)	(0.018)	(0.021)	(0.025)	(0.029)
4.34	0.20	0.30	0.43	0.58	0.71	0.84	0.97
(0.171)	(0.008)	(0.012)	(0.017)	(0.023)	(0.028)	(0.033)	(0.038)
6.35	0.28	0.38	0.53	0.71	0.90	1.07	1.24
(0.250)	(0.011)	(0.015)	(0.021)	(0.028)	(0.035)	(0.042)	(0.049)
9.53	0.41	0.51	0.66	0.94	1.17	1.37	1.63
(0.375)	(0.016)	(0.020)	(0.026)	(0.037)	(0.046)	(0.054)	(0.064)
12.70	0.53	0.76	1.02	1.57	2.03	2.54	3.18
(0.500)	(0.021)	(0.030)	(0.040)	(0.062)	(0.080)	(0.100)	(0.125)

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The squareness tolerances, as noted, is 3°. Spring life is presented in lieu of fatigue values.

12.4. *CALCULATION OF COMPRESSION SPRINGS*

All spring design begins with the application of Hooke's law. This law states that any force acting upon the material is directly proportional to the material's deflection, provided such deflection is within the range of that material's elastic limit.

12.4.1. Stress Calculation

Compression springs made of round wire subject this wire to a stress classified as a torsional stress. The basic formula to calculate such a stress *S* is, according to Bernoulli-Euler,

$$S = \frac{Mc}{J}$$

(12-4)

where c = distance from neutral axis at center of section to outside of material, or one-half

of material thickness for a round wire

J = polar moment of inertia

Table 12-6. Load Tolerances of Helical Compression Springs

	Tolerances: ±% of load. Start with tolerance from Table 12-4, multiplied by L_F														
	Deflection from free length to load, mm (in.)														
Length tolerance, ±mm (in.)	1.27 (0.050)	2.54 (0.100)	3.81 (0.150)	5.08 (0.200)	6.35 (0.250)	7.62 (0.300)	10.2 (0.400)	12.7 (0.500)	19.1 (0.750)	25.4 (1.00)	38.1 (1.50)	50.8 (2.00)	76.2 (3.00)	102 (4.00)	152 (6.00)
0.13 (0.005)	12	7	6	5											
0.25 (0.010)		12	8.5	7	6.5	5.5	5								
0.51 (0.020)		22	15.5	12	10	8.5	7	6	5						
0.76 (0.030)			22	17	14	12	9.5	8	6	5					
1.0 (0.040)				22	18	15.5	12	10	7.5	6	5				
1.3 (0.050)					22	19	14.5	12	9	7	5.5				
1.5 (0.060)					25	22	17	14	10	8	6	5			
1.8 (0.070)						25	19.5	16	11	9	6.5	5.5			
2.0 (0.080)							22	18	12.5	10	7.5	6	5		
2.3 (0.090)							25	20	14	11	8	6	5		
2.5 (0.100)								22	15.5	12	8.5	7	5.5		
5.1 (0.200)										22	15.5	12	8.5	7	5.5
7.6 (0.300)											22	17	12	9.5	7
10.2 (0.400)												21	15	12	8.5
12.7 (0.500)												25	18.5	14.5	10.5

First load test at not less than 15% of available deflection.

Final load test at not more than 85% of available deflection.

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Polar moment of inertia for a round section is

$$J = \frac{\pi d^4}{32}$$

where d = wire diameter

M = torsional moment, calculated as follows:

$$M = PR = \frac{PD}{2}$$

where P = load on spring, lb

D = mean diameter

Adding a stress-correcting factor K_W changes this formula to

$$S = \frac{8PD}{\pi d^3} K_W = \frac{2.546PD}{d^3} K_W$$

(12-5)

The sudden emergence of the stress-correcting factor K_W is due to the nonuniform distribution of torsional stress across the cross section of the wire. This is caused by the curvature of the coil and a direct shear load. Maximum torsional stress can be found at the inner surface of the spring, and its value is assessed with the aid of stress-correcting factor K_{W1} or K_{W2} (see Table 12-7), attributable to Dr. A. M. Wahl of Westinghouse Electric Co. The formula to calculate this correction factor is as follows:

$$K_{W1} = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

(12-6)

In some conditions, where resultant stresses are distributed more uniformly around the cross section, the stress-correcting factor K_{W2} can be used as

$$K_{W2} = 1 + \frac{0.5}{C}$$

(12-7)

Where elevated temperatures are encountered in the spring-operating environment, the stress distribution is more uniform around the cross section and can therefore be estimated, referring to Fig. 12-11.

Table 12-7. Maximum Allowable Torsional Stresses for Helical Compression Springs in Static Applications

Materials	Max % of tensile strength	
	Before set removed (K_{w1})	**After set removed (K_{w2})**
Patented and cold-drawn carbon steel	45	60–70
Hardened and tempered carbon and low-alloy steel	50	65–75
Austenitic stainless steels	35	55–65
Nonferrous alloys	35	55–65

Bending or buckling stresses are not included.
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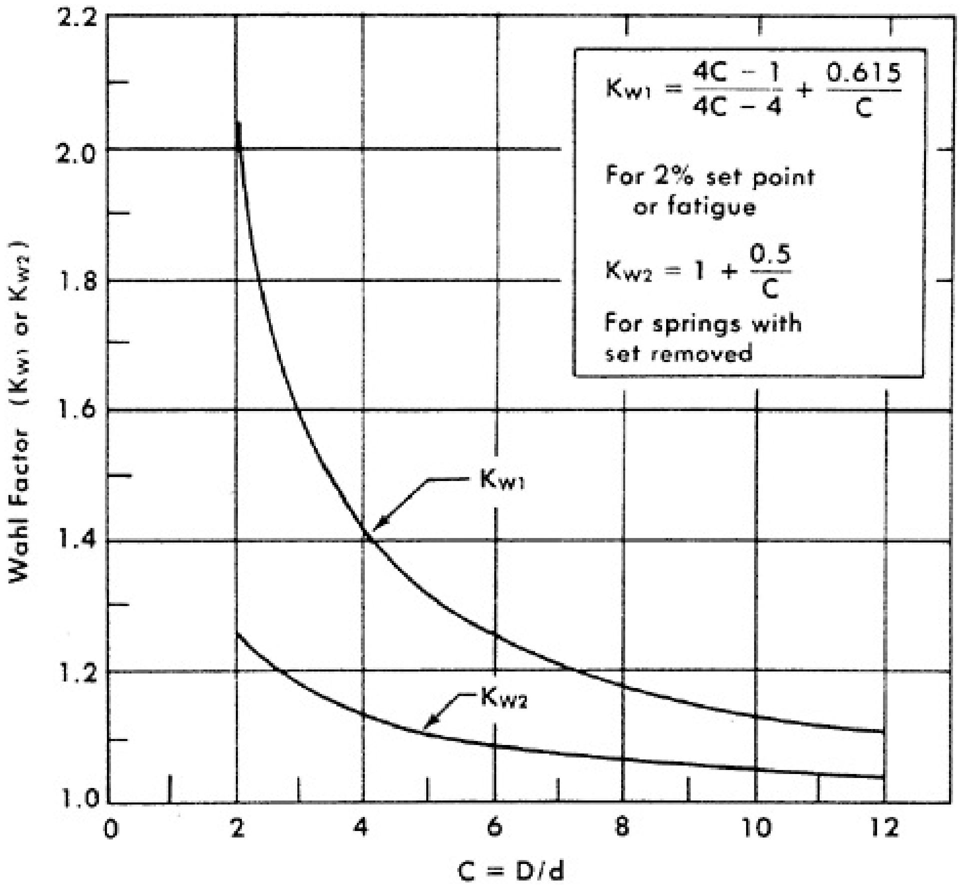

Figure 12-11. Wahl stress-correction factors for round wire helical compression and extension springs. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group,

Inc., Dallas, TX.)

Maximum allowable torsional stresses for helical compression springs in static applications are as listed in Table 12-7.

12.4.2. Diameter of Wire *d*

To choose the proper wire diameter for a given load, at an assumed stress, Eq. (12-8*a*) may be used.

$$\text{If } S = \frac{2.546PD}{d^3} \qquad \text{then} \qquad d = \sqrt[3]{\frac{2.546PD}{S}}$$

(12-8 a)

Knowing other pertinent values, we may calculate the spring diameter by using the formula for a round wire:

$$d = \frac{\pi D^2 S N_a}{Gf K_W}$$

(12-8 b)

where N_a = number of active coils

G = modulus of rigidity

F = deflection, in

K_W = Wahl's correction factor

whereas a wire of square cross section can be figured out as

$$d = \frac{2.32 SD^2 N_a}{Gf K_W}$$

(12-8 c)

where d = length of square side of coil

12.4.3. Deflection *f*

Accordingly, the deflection can be assessed on the basis of previous information by using the stress formula described previously:

$$f = \frac{8PD^3N_a}{Gd^4}$$

(12-9)

where N_a = number of active coils

G = modulus of rigidity, usually around 11,500,000 lb/in ^{2}for steel wire. For typical

properties of other materials see Tables 12-1 and 12-8

The modulus of rigidity, also called the modulus of shear, differs from the modulus of elasticity in that it produces an angular shift in the material's atomic structure. The modulus of rigidity and modulus of elasticity are related as follows:

$$E = 2G(1 + \mu)$$

(12-10)

where μ = Poisson's ratio

12.4.4. Spring Rate *k*

For helical compression springs, the spring rate is the change in load per unit of deflection:

$$k = \frac{P}{f} = \frac{Gd^4}{8D^3N_a}$$

(12-11)

where P = load on spring, lb

Where compression springs are used in parallel, the total rate is equal to the sum of the rates of individual springs. The sum of the rates of compression springs in series is calculable as

$$k = \frac{1}{1/k_1 + 1/k_2 + 1/k_3 + \cdots + 1/k_n}$$

12.4.5. Dynamic Loading, Suddenly Applied Load

Often, not only the influence of slowly applied loads should be figured out with springs. Suddenly applied loads too can have a tremendous impact on the life and performance of a spring. Since the load velocity is usually not exactly known, springs may end up retaining an unknown amount of kinetic energy.

Table 12-8. Typical Properties of Spring Temper Alloy Steel

Material	Tensile strength MPa (10^3 lb/in.^{2})	Rockwell hardness	Elongation,* %	Bend factor* ($2r/t$ transverse bends)	Modulus of elasticity, 10^4 MPa (10^6 lb/in.^{2})	Poisson's ratio
Steel, spring temper	1700 (246)	C50	2	5	20.7 (30)	0.30
Stainless 301	1300 (189)	C40	8	3	19.3 (28)	0.31
Stainless 302	1300 (189)	C40	5	4	19.3 (28)	0.31
Monel 400	690 (100)	B95	2	5	17.9 (26)	0.32
Monel K500	1200 (174)	C34	40	5	17.9 (26)	0.29
Inconel 600	1040 (151)	C30	2	2	21.4 (31)	0.29
Inconel X-750	1050 (152)	C35	20	3	21.4 (31)	0.29
Copper-beryllium	1300 (189)	C40	2	5	12.8 (18.5)	0.33
Ni-span-C	1400 (203)	C42	6	2	18.6 (27)	
Brass CA 260	620 (90)	B90	3	3	11.0 (16)	0.33
Phosphor bronze	690 (100)	B90	3	2.5	10.3 (15)	0.20
17-7 PH RH950	1450 (210)	C44	6	Flat	20.3 (29.5)	0.34
17-7 PH condition C	1650 (239)	C46	1	2.5	20.3 (29.5)	0.34

*Before heat treatment.

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Considering that work done on springs (= force × space) equals the energy absorbed by the spring when neglecting the hysteresis, the solution is as follows:

For loads applied very slowly:

$$f = \frac{P}{k}$$

(12-12)

For loads applied suddenly:

$$f = \frac{2P}{k}$$

(12-13)

For loads dropped from a certain height:

$$f = \sqrt{\frac{2P(s+f)}{k}}$$

(12-14)

where f = deflection, in

k = spring rate

P = load on the spring, lb

s = height from which the load was dropped, in

12.4.6. Dynamic Loading, Impact

With a spring being cyclically loaded and unloaded, an emergence of surge wave provides for the transmission of torsional stress from the point of loading application to the point of restraint. This surge wave travels at a velocity one-tenth the velocity of a normal torsion stress wave. Velocity of the torsion stress wave V_T can be calculated:

$$V_T = \sqrt{\frac{Gg}{\rho}} \quad \text{in./s}$$

(12-15 a)

which in metric version becomes

$$V_T = 10.1\sqrt{\frac{Gg}{\rho}} \quad \text{m/s}$$

(12-15 b)

where ρ = density, 1/1365 for steel

g = acceleration due to gravity, 32 ft/s ^{2}, or 9.8 m/s ^{2}

This surge wave limits the springs' absorption and release of energy by restricting its impact velocity V, which is a function of stress and material constants, applied in parallel with the spring axis.

Impact velocity may be calculated as follows:

$$V \simeq S\sqrt{\frac{g}{2\rho G}} \quad \text{in./S}$$

(12-16 a)

and in metric:

$$V \simeq 10.1S\sqrt{\frac{g}{2\rho G}} \quad \text{m/s}$$

(12-16 b)

Impact velocity and stress are actually independent of the configuration of the spring. For steel materials, impact velocity should be in the range of

$$V = \frac{S}{131} \quad \text{in./s} \qquad \text{or} \qquad V = \frac{S}{35.5} \quad \text{m/s}$$

(12-17)

12.4.7. Dynamic Loading, Resonance

Springs have a natural inclination to vibration, creating a resonance within their mass. Resonance occurs where the cyclic loading approaches the natural frequency of the spring or its multiples. Resonance may increase the coil deflection and stress level, exceeding all assumed amounts. It can cause the spring to shiver and bounce, with subsequent alteration of its load-carrying capacity and other values.

A natural spring's frequency must be at least 13 times greater than its operating frequency to prevent the emergence of resonance.

The compression spring's natural frequency is inversely proportional to the amount of time needed for a surge wave to traverse the spring. For a spring which has no damping and has both ends fixed, this amounts to

$$n = \frac{d}{9D^2N_a}\sqrt{\frac{Gg}{\rho}}$$

(12-18 a)

where the value of n for steel is

$$n = \frac{14{,}000d}{N_a D^2}$$

(12-19 a)

In metric translation, this calculation becomes

$$n = \frac{1.12\times 10^3 d}{D^2N_a}\sqrt{\frac{Gg}{\rho}}$$

(12-18 b)

where the value of n for steel is

$$n = \frac{3.5\times 10^5 d}{D^2N_a}$$

(12-19 b)

where n = natural frequency, Hz

ρ = density, 1/1365 for steel

g = acceleration due to gravity, 32 ft/s ^{2}, or 9.8 m/s ^{2}

12.5. *SPECIAL CROSS SECTIONS OF THE WIRE*

Springs whose cross section is rectangular in shape and oriented with the width of the rectangle perpendicular to the spring axis have a capacity to absorb more work energy in smaller space than equivalent round wire. This is true despite the fact that the distribution of stress around the rectangular section may not be quite as uniform as that of the round wire. Rectangular-shaped wire is also more costly than round wire, with the keystoned wire being the most expensive of the three (Fig. 12-12).

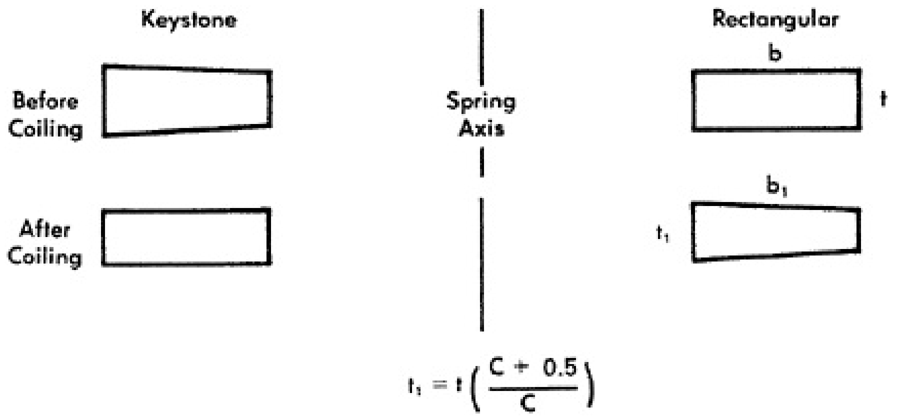


Figure 12-12. Wire cross section before and after coiling. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

The coiling operation, when applied to a rectangular wire, alters its shape, slanting the rectangle against one of the sides. Keystone wire is manufactured to come out from coiling rectangular.

Rectangular-shaped wire springs can be calculated with slightly altered round-wire formulas. The rate for such a compression spring is

$$k = \frac{P}{f} = \frac{Gbh^3}{N_a D^3} K_2$$

(12-20)

Since the wire is torsionally loaded, the rate is equal if the wire is wound on flat or on edge (see Fig. 12-8). Values of constants K_1 and K_2 are as shown in Fig. 12-13.

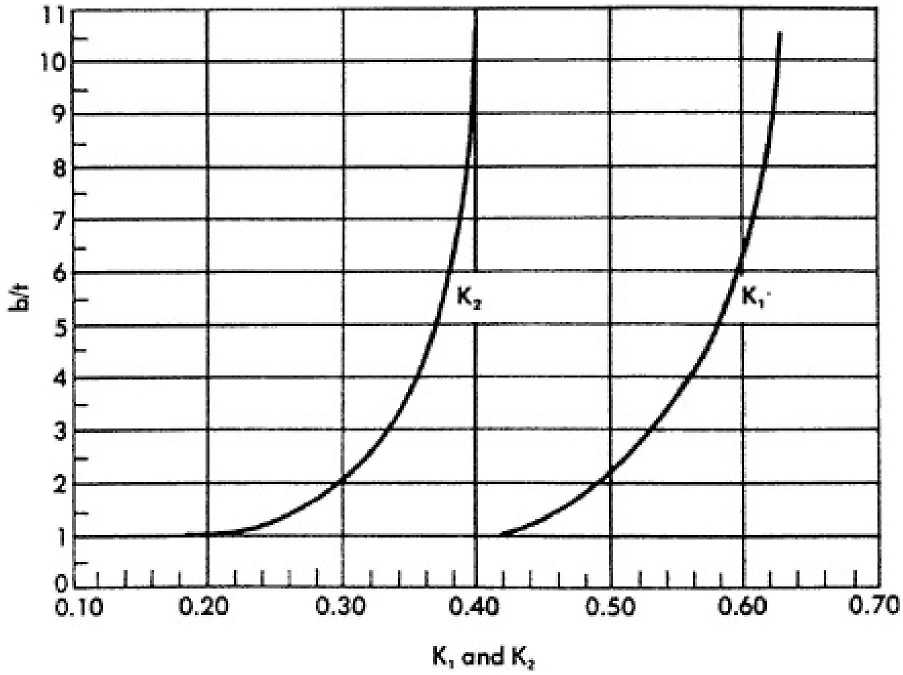


Figure 12-13. Constants for rectangular wire in torsion. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

The stress in such a spring can be obtained by using the formula given in Eq. (12-21).

$$S = \frac{PD}{K_1 bh^2} K_E \qquad \text{or} \qquad \frac{PD}{K_1 bh^2} K_F$$

(12-21)

where K_E = stress-correcting factor for springs wound on edge, shown in Fig. 12-14

K_F = stress-correcting factor for springs wound on flat, shown in Fig. 12-15

Where an attempt is made to produce a rectangular-shaped wire by rolling a round material, or where the cross-sectional shape of the wire is not quite round, a correction factor h' should be added to the stress formulas above, to replace h:

$$h' = \frac{2d}{1 + b/h}$$

(12-22)

To figure out the amount of stress and deflection, a triangular cross section of the wire would utilize the formulas

$$S = \frac{20PR}{l^3}$$

(12-23)

where l = length of each side of the triangle (see Fig. 12-16)

R = mean radius of the coil

$$f = \frac{290.3PN_aR^3}{Gl^4}$$

(12-24)

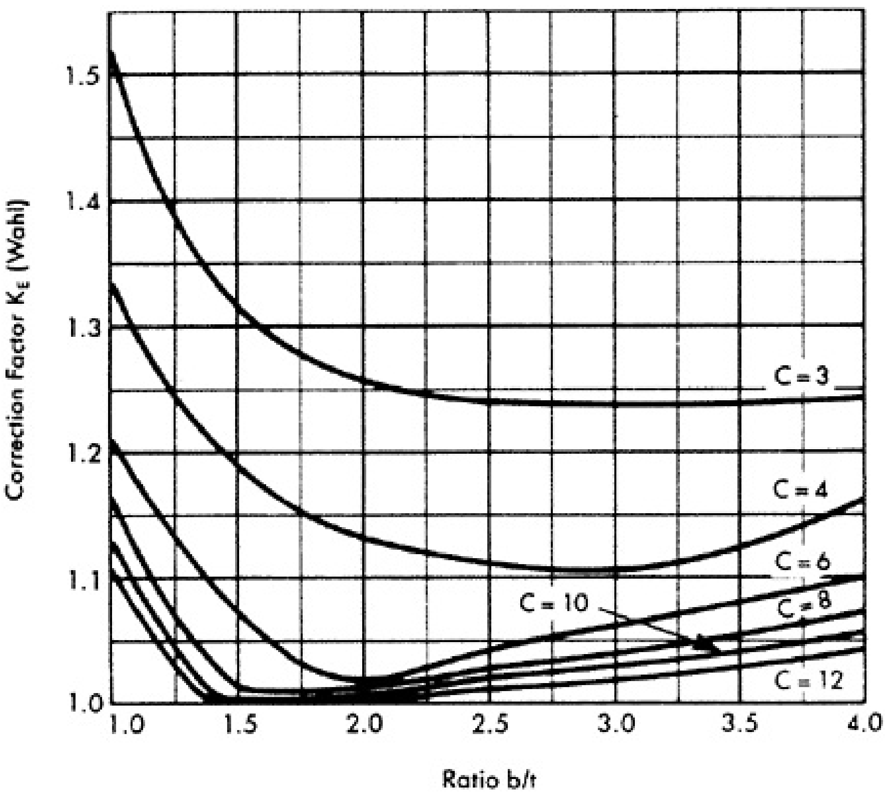


Figure 12-14. Stress-correction factors for rectangular wire compression springs wound on edge. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

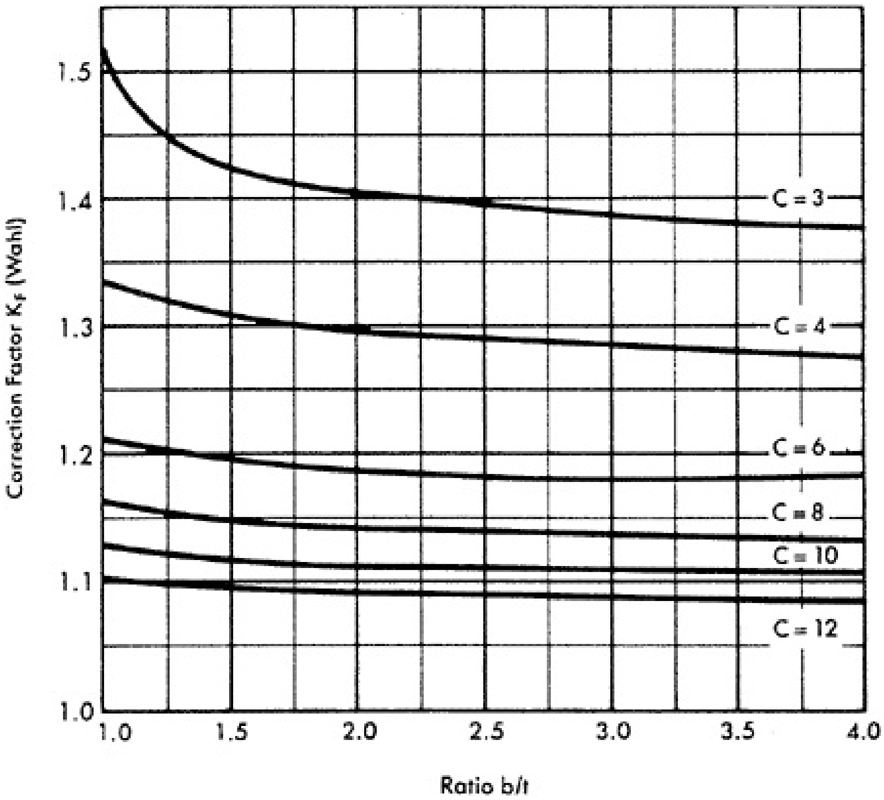


Figure 12-15. Stress-correction factors for rectangular wire compression springs wound on flat. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

Hexagonal cross sections, where the inscribed circle's diameter (as shown in Fig. 12-16) is v and the area of the cross section is A, can be calculated by using the following formulas:

$$S = \frac{PR}{0.217Av}$$

(12-25)

and

$$f = \frac{47.24PN_aR^3}{Gv^2A}$$

(12-26)

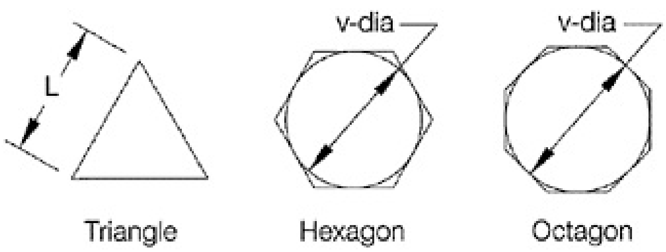


Figure 12-16. Cross-sectional shapes.

For octagonal sections (Fig. 12-16), where the inscribed circle's diameter is v

and the area of the cross section is A, the formula is

$$S = \frac{PR}{0.223Av}$$
(12-27)

and

$$f = \frac{48.33PN_aR^3}{Gv^2A}$$
(12-28)

Figure 12-16 shows triangular, hexagonal, and octagonal cross sections. For a regular elliptical section, the following formulas can be used:

$$S = \frac{16PR}{\pi x^2 y}$$
(12-29)

and

$$f = \frac{248.1PN_aR^3J}{GA^4}$$
(12-30)

where x = minor axis of ellipse

y = major axis of ellipse

A = cross-sectional area of ellipse

J = polar moment of inertia of the section, which can be calculated as

$$J = \frac{\pi(xy^3 + x^3y)}{64}$$
(12-31)

and subsequently,

$$A = \frac{\pi xy}{4}$$
(12-32)

12.6. *HOT-WOUND SPRINGS*

Most often, springs are cold-formed up to 3/8 in. (10 mm) diameter of the wire or bar size. After this dimension, cold forming becomes difficult, and hot winding of springs is used instead. This type of spring manufacture involves heating of the steel up to the austenitic range, winding it, quenching down to martensitic structure, and tempering to arrive at required properties.

The most often used type of hot-wound spring is the compression spring, utilized as a part of an automobile suspension system or as springs used in rail cars.

Marginally some extension, torsion, and volute springs may be hot-wound as well.

12.6.1. Design and Calculations

Design parameters and calculations for this type of spring are the same as those of other springs. The only exception is that of the spring rate calculation, which here includes an empirical factor K_H, which is providing for the adjustment due to scaling-caused complications.

$$k = \frac{P}{f} = \frac{Gd^4 K_H}{8D^3 N_a}$$

(12-33)

where k = spring rate

P = load, lb

f = deflection, in

G = modulus of rigidity

N_a = number of active coils

K_H = 0.91 for hot-rolled carbon or low-alloy steel materials, which are not centerless ground

= 0.96 for hot-rolled carbon or low-alloy steel materials, centerless ground

= 0.95 for carbon or low-alloy steel material on torsion springs

12.6.2. Types of Spring Ends

The ends of hot-wound springs may be ground, tapered, tangent, or pigtailed, as shown in Fig. 12-17.

Ground ends provide the spring with a good bearing surface and unsurpassed squareness.

Tapered ends are produced by rolling a taper on the bar. During hot winding, these ends must be guided to provide for their proper orientation. Additional grinding improves the spring's squareness and bearing surface quality.

Tangent ends are standard, with no secondary manufacturing procedures involved. Because of the hot-winding process, springs with tangent ends have a straight portion approximately two wire diameters in size at each end. Their bearing surface must be designed in accordance with the requirements of such a shape, as it tends to exceed the outline of the spring.

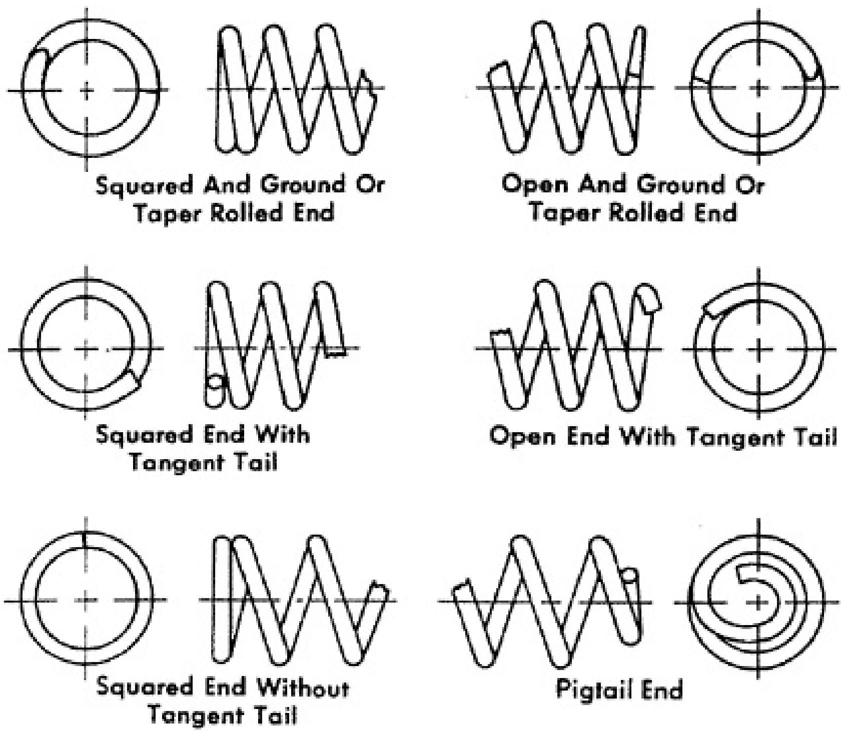


Figure 12-17. Typical ends of hot-wound compression springs. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

Pigtailed ends are formed along with the hot-winding process. These ends are popular in situations where the spring must be clamped or bolted to its seat.

12.6.3. Hot-Wound, Noncompression Springs

Extension and torsion springs, utilizing loops and legs, have these additions to their shapes formed at the same time the spring is wound. For that reason, such shapes should be kept as simple as possible. All elaborate designs that are difficult to achieve involve reaustenitizing of the spring, with subsequent increase in manufacturing costs.

12.7. *HELICAL EXTENSION SPRINGS*

Helical extension springs are used where a pulling force is needed. They are most often made from round wire, closely wound, with initial tension obtained through stressing them in torsion.

12.7.1. Design of Extension Springs

Design procedures are the same as those of compression springs. It should be remembered, though, that extension springs, when compared to compression-type springs, are slightly different in that they are

- Coiled with initial tension, equal to the minimal amount of force needed to separate adjacent coils
- Coiled (usually) without their set being removed
- Equipped with no fixed stop to prevent their overloading

Figure 12-18 shows typical extension spring dimensions.

12.7.2. Spring Rate of Extension Springs k

Same as for helical compression springs, the spring rate of the extension spring is the change in load per unit of deflection. The formula to be used with extension springs is, however, slightly different:

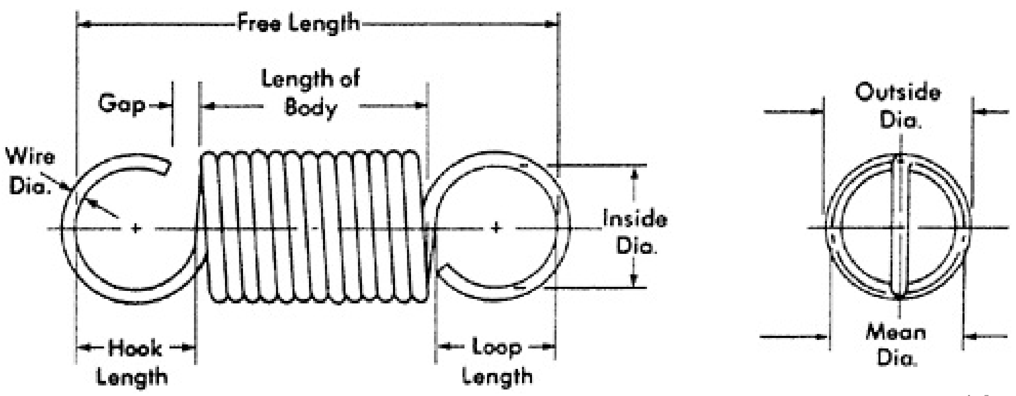


Figure 12-18. Typical extension spring dimensions. (From “Design Handbook,” 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

$$k = \frac{P - P_I}{f} = \frac{Gd^4}{8D^3 N_a}$$

(12-34)

where P_I is the initial tension. The applicable stress is given by the formula

$$S = \frac{8PD}{\pi d^3} K_W$$

(12-35)

12.7.3. Types of Spring Ends

A wide variety of ends is utilized with this type of spring as a provision for their attachment to other parts of the assembly. There are twist loops, side loops, cross-center loops, and extended hooks. Loops differ from hooks in that their shape will form small gaps, whereas hooks are actually loops with large gaps.

Special types of ends are formed from straight sections of wire tangent to the spring body shape.

Naturally, common loops of standard lengths are most economical to obtain. Figure 12-19 shows common end configurations for helical extension springs.

Stresses in loops or hooks are often higher than those within the spring wound-up body itself. For that reason, liberal bend radii in loops, combined with reduced end coil diameters, should be used to alleviate this problem.

The stress encountered in a full twist loop may reach its maximal value in bending at point A (shown in Fig. 12-20), while the value of the maximum stress in torsion is at its highest at point B.

To assess the actual amount of stress at these two locations, the following formulas should be used:

$$S_A = \frac{16DP}{\pi d^3} K_1 + \frac{4P}{\pi d^2} \text{ bending}$$

(12-36 a)

where

$$K_1 = \frac{4C_1^2 - C_1 - 1}{4C_1(C_1 - 1)}$$

(12-37)

and

$$C_1 = \frac{2R_1}{d}$$
(12-38 a)

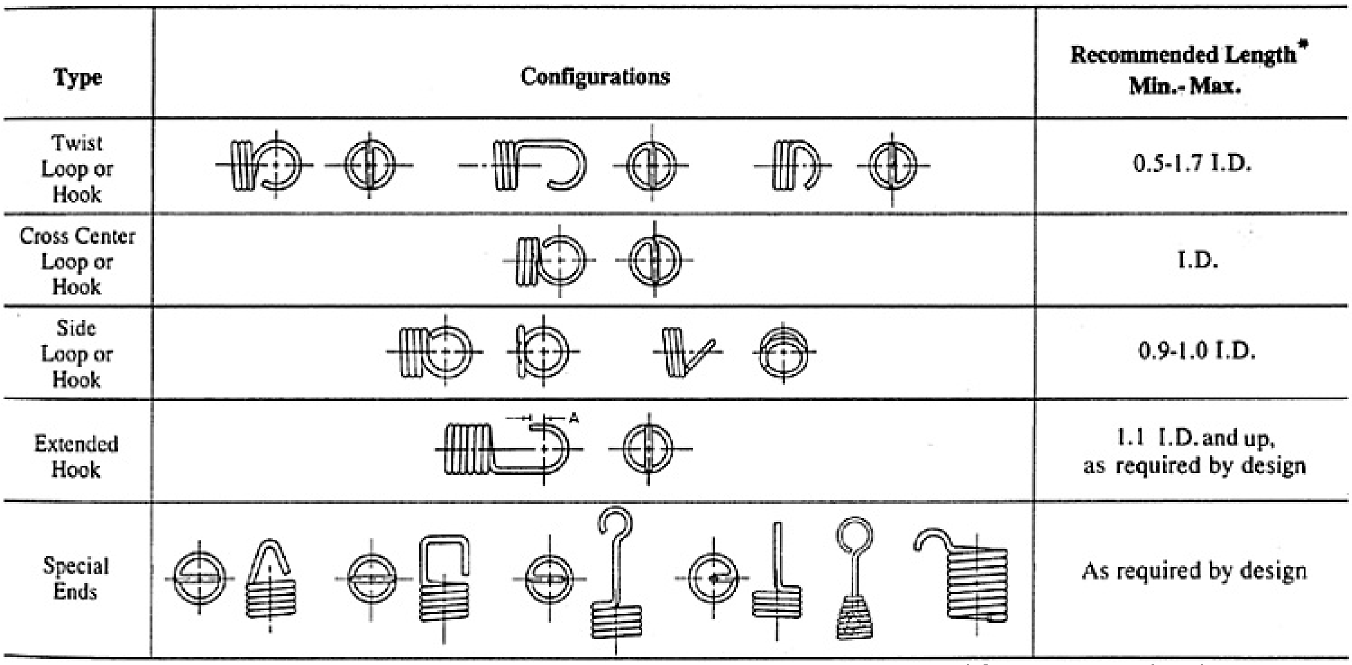
and

$$S_B = \frac{8DP}{\pi d^3}\frac{4C_2 - 1}{4C_2 - 4}$$
(12-36 b)

and

$$C_2 = \frac{2R_2}{d} \text{ torsion}$$
(12-38 b)

It is recommended that the value of C_2 be greater than 4.



Type	Configurations	Recommended Length* Min.- Max.
Twist Loop or Hook		0.5-1.7 I.D.
Cross Center Loop or Hook		I.D.
Side Loop or Hook		0.9-1.0 I.D.
Extended Hook		1.1 I.D. and up, as required by design
Special Ends		As required by design

Figure 12-19. Common end configurations for helical extension springs. (From “Design Handbook,” 1987. Reprinted with permission from Associated) Spring, Barnes Group, Inc., Dallas, TX.

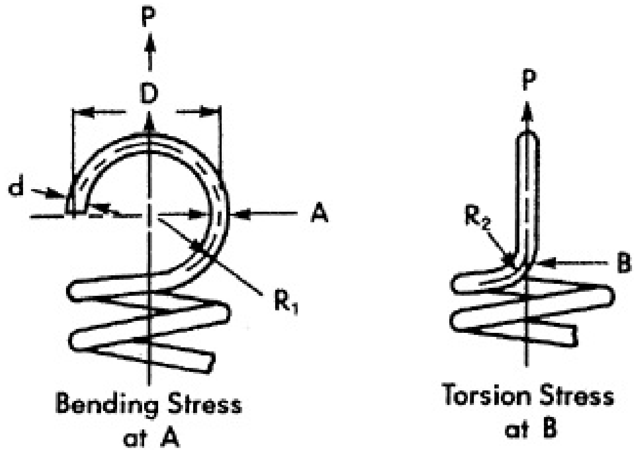


Figure 12-20. Location of maximum bending and torsion stresses in twist loops. (From "Design Handbook," 1987. Reprinted with permission from Associated Spring, Barnes Group, Inc., Dallas, TX.)

12.7.4. Specific Recommended Dimensions

With the free length of an extension spring being measured between the inner surfaces of both ends (per Fig. 12-18), this dimension should be made equal to the length of the spring body, plus its ends. The spring body can be calculated as

$$L_{\text{body}} = d(N + 1)$$

(12-39)

where N = number of coils

The gap in the loop (or hook) opening can vary with the manufacturer. Generally, this gap should not be specified smaller than one-half the wire diameter. If a gap smaller than that is desired, the designer should consult the spring manufacturer about its feasibility.

The number of active extension spring coils is equal to the number of coils it contains. In springs used with threaded inserts and swivel hooks, the number of active coils is lower.

Loops and hooks account for approximately $0.1N_a$ of active coils with one-half twist loops. Up to $0.5N_a$ can be used with some cross-center, full-twist loops, or extended loops.

12.8. *FLAT SPRINGS*

Flat springs are those made from strip or sheet material. They may contain bends and sometimes their shapes may be quite complex. Often, they perform additional functions in an assembly, such as locating the opening or a part, retaining other parts in an assembly, or banking on them, acting as a latch, or conducting electricity.

The most commonly used flat springs are of a cantilever type (Fig. 12-21). When calculating such springs, all cantilever and simple-beam equations may be used. However, to obtain more accurate results, calculations based on curved beam theory are recommended by Associated Spring's experts. Where the amount of elastic deflection is of importance, Castigliano's method is their additional choice.

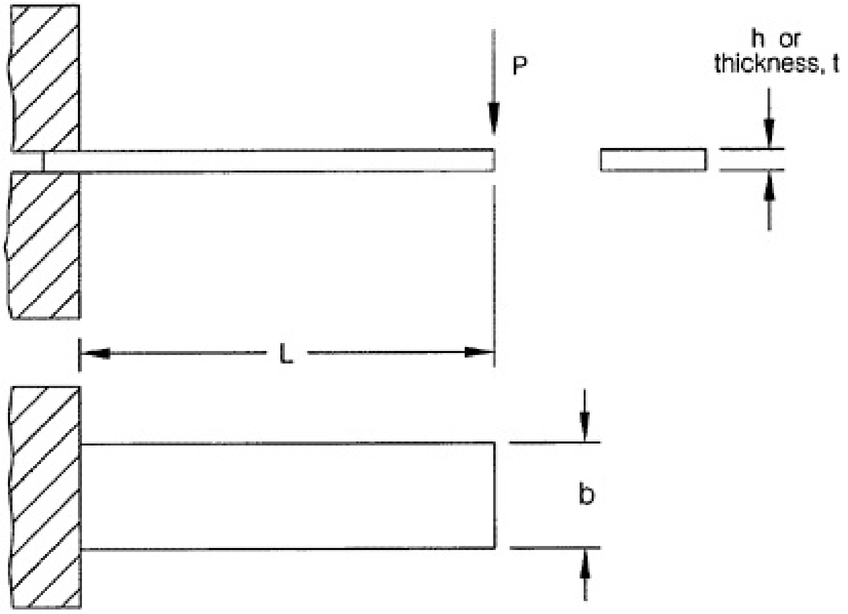


Figure 12-21. Cantilever spring.

12.8.1. Materials

Carbon steel materials, most often used in flat spring manufacture, usually belong in one of the following groups:

- 0.70 to 0.80 percent carbon content, which is a slightly less expensive material, more tolerant of sharper bends
- 0.90 to 1.05 percent carbon content with a higher elastic limit

Materials are used either in their annealed form or pretempered. Annealed materials must be heat-treated after forming.

The amount of distortion caused by a heat-treatment is difficult to assess or calculate. Rather, the designer should depend on a spring manufacturer's experience, while avoiding too precise tolerances, sharp corners, and edges on the spring. Parts with thin and wide cross sections will tend to be more distorted, sometimes requiring restriking or other adjusting operation.

Pretempered materials must be hard enough to possess a sufficient elastic limit for their function under desired loads. At the same time, this material should not be too hard, as it may fracture during forming or cause breakage and excessive wear to the tooling. The spring-back of pretempered material is greater, and an allowance for it must be made in the tool designing stage. Again, an experienced spring manufacturer may be the one to evaluate the amount of necessary alterations.

12.8.2. Design and Calculations

Mostly all flat springs are preloaded during the bending operation. The surface condition of the material must be smooth, possibly polished, with no dents or nicks. Sharp edges and burrs should be eliminated by design or by abrasive means. Bend radii should be liberal in size, since sharp radii become stress points, exerting damaging influence on the part. Naturally, all bends should be oriented across the material's grain line.

Based on Bernoulli-Euler beam theory for bending of beams, the maximum stress can be calculated as

$$S = \frac{Mc}{I}$$

(12-40)

where c = distance from neutral axis to outside or one-half of material thickness

M = moment, amounting to distance from support times the load, or $M = PL$

I = moment of inertia. For a rectangular section, moment of inertia can be calculated:

$$I = \frac{bh^3}{12}$$

(12-41)

Combining the above values into the single equation, we get the maximum stress expressed as

$$S = \frac{6PL}{bh^2}$$

(12-42)

where P = load on the spring, lb

L = length of lever arm, in

The load value may be calculated by using a formula

$$P = \frac{fEbh^3}{4L^3}$$

(12-43)

where E = modulus of elasticity

f = deflection, in

For flat springs, where the width to thickness ratio is relatively small, the maximum stress and deflection formulas are reasonably accurate. Higher width to thickness ratio increases the flexural rigidity of the spring, resulting in the modulus of elasticity E being replaced by E' as follows:

$$E' = \frac{E}{1-\mu^2}$$

(12-44)

where μ = Poisson's ratio

The deflection can be calculated using a standard formula

$$f = \frac{PL^3}{3EI}$$

(12-45)

where E = modulus of elasticity

I = moment of inertia; for a rectangular section the value can be calculated with

Eq. (12-41).

This formula, when applied to a rectangular section, changes to

$$f_{RT} = \frac{4PL^3}{Ebh^3} \quad \text{or} \quad \frac{2SL^2}{3Eh}$$

(12-46)

where S = stress

Since L and h values are raised to the third power, accurate measurements are vital.

All these equations were proved satisfactory where the ratio of deflection to cantilever length F/L was less than 0.3. For larger deflections, E should be replaced by E', as given by Eq. (12-44).

12.9. *GAS AND AIR SPRINGS AND THEIR APPLICATIONS*

Most probably, forced by an unending and never-resting competition, the industry had to come up with a different type of springs, to ease their installation, improve their function, and remove the “gray areas” of preload from the spring usage dictionary. With the wound springs, special spring pockets had to be milled into the blocks; the correct spring height was an unending problem; and the force buildup, also called preload, was always somewhat a mystery. During the operation, the force of wound springs started from zero and progressed upwards, sometimes becoming unpredictable and often even excessive.

Gas springs are different. They are capable of delivering much more force in lesser area than ordinary wound springs. They generate pressure on contact, eliminating the need for preload. This way the pressure pad can be smaller, the amount of cylinders diminished, the stroke shorter, while the force produced by springs is constant and unwavering alongside the stroke of a press. Their travel to length ratio is much larger and their pressure can be easily monitored.

Gas springs are also more balanced. Whereas in an assembly of several wound springs some may be cracked and the rest may not produce the pressure needed, gas springs are always there, always working. Should their pressure drop somehow, the gas springs can reclaim the gas needed and prop up the pressure to the demanded levels.

Out of all gasses, nitrogen springs gained the ground across the board. One of the reasons may be the low cost of nitrogen gas, but nitrogen is also nonflammable, inert, and tonnage resistant. This means that as the pressure against such spring rises, the force of its output increases in proportion to the volume of gas that was compressed.

Nitrogen springs should never be preloaded, and, actually, manufacturers caution against preload with determination. But at the same token, nitrogen charge should not be lowered in anticipation of extending the life expectancy of the seal. Such a precaution may actually harm the spring, as the modern seals are designed to operate at the full nitrogen charge.

Their loading is of concern though, as they are not to be used at the operating pressure exceeding 90 percent of their recommended maximum.

12.9.1. Nitrogen Springs and Their Types

There are several types of nitrogen spring systems available on the market today, the difference between each group being provided by the method of attachment and gas distribution. These types are as described below:

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12.9.1.1. *Manifold System.*

Manifold system is a closed system, embedded in a metal plate, which is cross-drilled to allow for the nitrogen gas distribution. The spring cylinders are attached to the channels through the tapped holes and may be positioned where needed. The whole assembly is connected to the control panel, which directs the volume of gas within the system (See Fig. 12-22).

Manifold systems require clearance between the die and the end of the cylinder rod, so that the rod does not touch the plate of the opposite die half (either upper or lower plate, with dependence on the type of mounting). The clearance is necessary for the piston to come to a full die-open position.

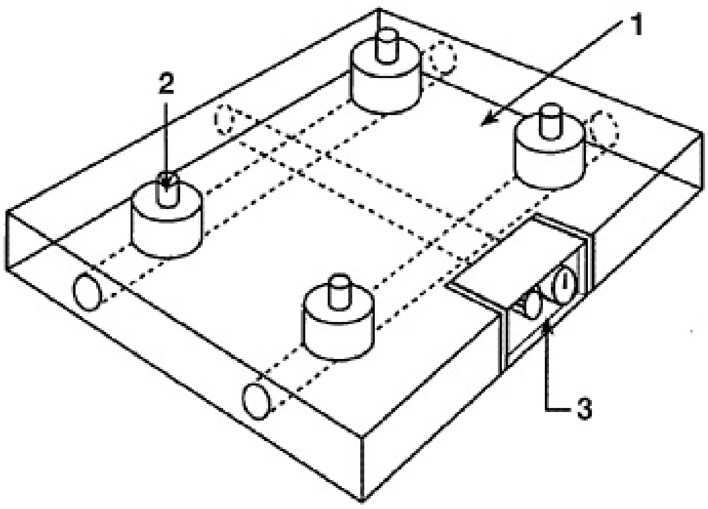

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Figure 12-22. Nitro Dyne ® XP manifold system of gas springs. (Reprinted with permission from Hyson ™ Products, Brecksville, OH.)

Nitrogen reservoirs can also be added to the bottom die shoe (see Fig. 12-23), if the shut height of the press is too limited. The details of such an arrangement are the same as those of the regular manifold system. The nitrogen reservoir interconnects the cylinders via the holes drilled through the shoe. As can be expected, a demand such as this will weaken the die shoe somewhat.

The compression tank retains the excessive nitrogen, which the springs leak when being pressed down. To determine the tank size, the amount of excessive nitrogen (also called “swept volume”) has to be determined first. This can be calculated as:

$$V_{SW} = A_P \times L_{WK} \times \text{No. of Cylinders}$$
(12-47)

where V_{SW} = swept volume
A_P = area, piston
L_{WK} = working stroke

From the result, the volume of the tank can be determined as:

$$V_T = V_{SW}(100 : R_P)$$
(12-48)

where V_T is the required volume of the tank, and R_P is the percentage of desired pressure rise, or increase.

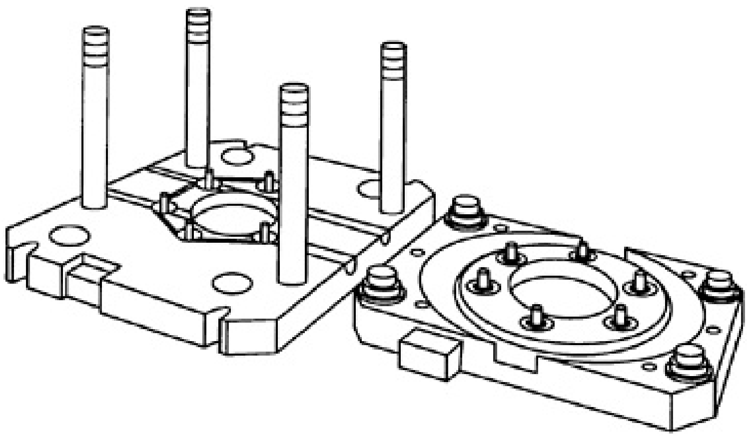

Figure 12-23. Nitrogen cylinders as installed in the die shoe. (Reprinted with permission from Hyson ™ Products, Brecksville, OH.)

Pressure increase, also called pressure rise, is generally recommended at 15–20 percent for draw dies and 30–40 percent for strippers, form pads, and cam returns.

12.9.1.2. *12-9-1-2 Hose and Tank System.*

In this design, a reservoir tank is connected with cylinders via high-pressure hoses. The whole assembly is wired to the control panel for the balance of pressure between cylinders. There is no fixed mounting and the cylinders can be bolted exactly where needed, as shown in Fig. 12-24.

12.9.1.3. *12-9-1-3 Self-Contained Cylinders.*

Self-contained cylinders (see Fig. 12-25) are isolated springs, which already contain the amount nitrogen needed for their function and do not need any additional supply of it. Where balanced force is necessary, several cylinders can be connected together with pressure hoses, as shown in Fig. 12-25.

As with all other springs/cylinders, self-contained cylinders should be protected from the contact with any fluids, be it die lubricants, cleaners, water, or any other liquids. For this reason, their retaining pockets should be provided with adequate draining channels. The spring should always be attached to the bottom surface of its retaining pocket with bolts. This precaution not only prevents the cylinder from being swayed aside during the die function; it also does not allow for a gap to retain metal chips, lubricants, grime, and other debris underneath it.

The piston contact surface should be straight and perpendicular to the die surface. Where a slanted contact surface may be used, side loading will result, which may sway the piston aside and eventually ruin it. Same with surfaces containing pockets or screw heads—these may produce an uneven pattern of wear on the piston rod (see Fig. 12-26).

The disadvantage of single, self-contained cylinders is their height: these types of gas springs are always higher than other cylinder types (see Fig. 12-27).

12.9.1.4. *12-9-1-4 Spring Cushions.*

These are small assemblies of cylinders under a common pressure pad as shown in Fig. 12-28. As these are very powerful devices, such cushions are useful in aiding the press function and can be installed either attached to the ram, or under the bolster, or just about everywhere. The advantage is in their nearly constant force throughout the stroke.

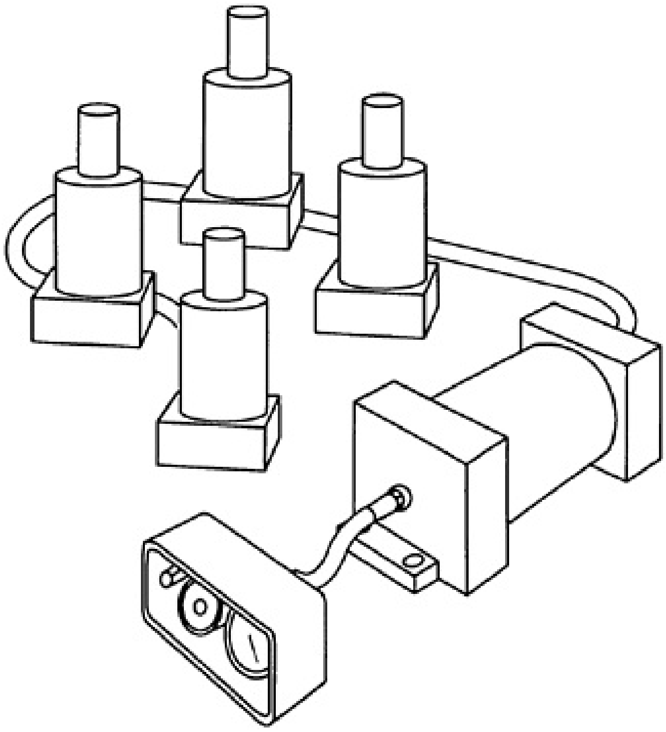

Figure 12-24. Hose and tank system. (Reprinted with permission from Hyson ™ Products, Brecksville, OH.)

Figure 12-25. Super Tanker ® Cylinders. (Reprinted with permission from Hyson ™ Products, Brecksville, OH.)

Self-contained pressure pads are also used as cam-driving devices, in which case they can be provided with a cam-driving block or with a roller. Stripper springs for the return pressure must be used in conjunction with the cam pads. The cam-driving cushions also serve well in delayed piercing.

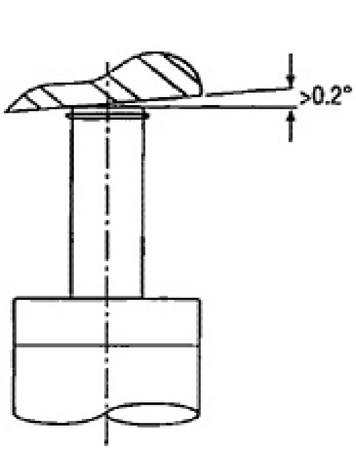

Figure 12-26. Inclined or uneven surface produces damage to the piston rod. (Reprinted with permission from Hyson ™ Products, Brecksville, OH.)

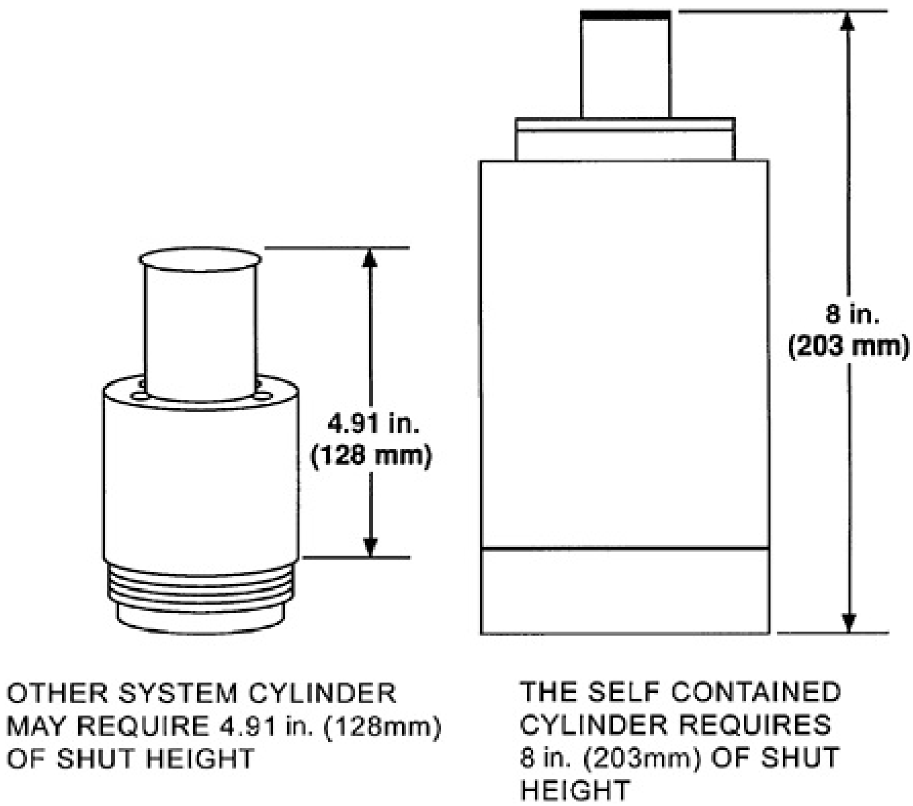

Figure 12-27. Height of cylinders. (Reprinted with permission from Hyson ™ Products, Brecksville, OH.)

12.9.2. 12-9-2

Air springs, also called pneumatic cylinders (or air pistons) are preferred where long press strokes and adjustable forces of their application are required. Often, these types of springs contain a hollow cylinder, which at the same time acts as a surge tanks for air. The pistons operate on shop air, the pressure of which can be increased at the die closure by implementing one-way check lines at the air inlet.

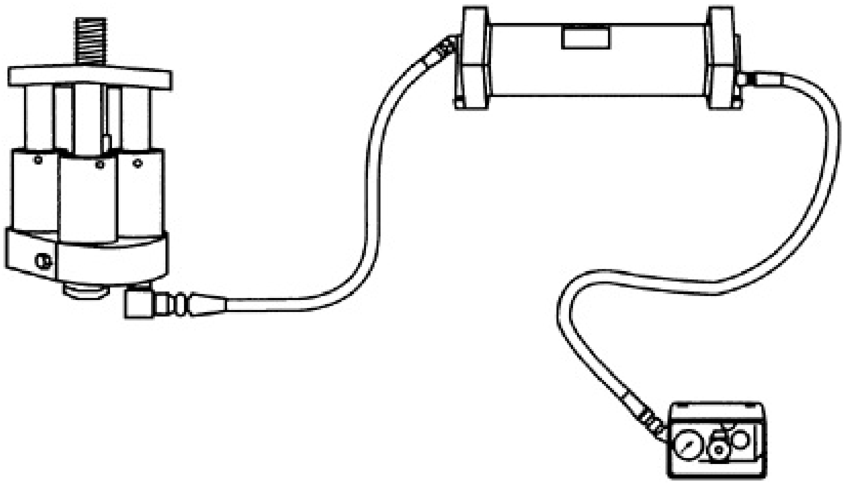

Figure 12-28. Spring cushion. (Reprinted with permission from Hyson ™ Products, Brecksville, OH.)

Hydraulic cylinders are slower in response, for which reason they are not used in dies too often. However, their usage with some forming applications brings about definitive advantages. First of all, by adding an accumulator of fluid, their force can be adjustable. A press operator can monitor this force by himself or herself and change it on demand, either up or down. Additionally, a typical lifetime of a hydraulic cylinder system is up to 2.5 million cycles, with dependence on other variables, such as the severity of manufacturing operation.

12.9.3. Calculation of Resistant Tonnage for Nitrogen Springs

The tonnage, also called the resistant tonnage, is an important element in the selection of proper nitrogen springs. Resistant tonnage is the force needed to maintain the given pad pressure.

The calculation of tonnage, meaning any tonnage at all, is mostly an open guess and subject to variations due to friction, heat, galling, to name but few invasive effects on the metal stamping field that do not hesitate to exert their influence when the least appropriate.

The resistant tonnage calculation of nitrogen springs is probably bound to change over time, as new formulas come up quickly and the old ones do not die easily. But the basic formula for the drawing tonnage is similar to the calculation of the blanking or piercing pressure, and it can be calculated as follows:

$$V_{TON} = L \cdot t \cdot c \cdot F_s$$

(12-49)

where L = length of the line, linear

t = material thickness

F_S = safety factor, 1.2 to 1.25 (i.e., 20 to 25%)

c = coefficient, per values below

The values of the above coefficient c are approximately

23 = for cold rolled steel

18–20 = for aluminum

28–35 = for stainless steel

12.9.4. Comparison of Different Types of Springs

Each of the spring systems described earlier has its advantages and disadvantages. For example, coil springs may still be needed in great quantities for shorter die runs, or where the life of a spring is of no concern. The cost of wound springs is low and their exchange is most often quick and cost-efficient, when not counting the time a die needs for its removal from the press.

Air, gas, and pneumatic springs’ performance suffers where a short stroke, high-speed application with many million of cycles are expected. At the same time, longer spring travels with high speed applications can incredibly weaken wound springs.

A nitrogen cushion does not require a compressor; an air spring does need one. The nitrogen spring cushion may cost half of the air cushion’s price tag. The force exerted by nitrogen cushion is constant, unwavering. An air cushion’s force varies, since the build up of pressure is controlled by the air valves, expansion tanks, and compressors, all attached to the spring. The cost of maintenance for the nitrogen spring is also lower.

If the parts do not come out from the die as they should and a wrong amount of spring force is suspected, it is quite a task to determine the spring pressure received from wound springs. Where an increase in force is needed, wound spring's pressure cannot be easily adjusted.

Nitrogen gas is nontoxic, colorless, odorless, and inert. It will not ignite under high pressures or in the close proximity of an open flame. Nitrogen gas additionally does not support combustion. Air and other gases may be flammable in some environments. For this reason, their application with dies producing heat, should be limited. Their maximum operating temperature is approximately 170°F, and unless the heat-producing dies are cooled, coil springs should be opted for.

Hose-and-tank system of nitrogen springs is more costly, but it allows for controlling and adjusting the spring pad balance. Their safety factor is greater than that of other spring types.

As seen earlier, each type of a spring has its application range and subsequently, its usage. The proper decision depends on each particular situation, on the die function, and on the environment where it operates. Naturally, the cost factor exerts its influence here as well.