UNITED STATES PATENT AND TRADEMARK OFFICE

BEFORE THE PATENT TRIAL AND APPEAL BOARD

NORMAN INTERNATIONAL, INC.

Petitioner

v.

HUNTER DOUGLAS, INC.

Patent Owner

CASE: To Be Assigned

Patent No. 6,968,884 B2

DECLARATION OF LAWRENCE E. CARLSON IN SUPPORT OF PETITION FOR *INTER PARTES* REVIEW OF U.S. PATENT NO. 6,968,884 B2

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I. INTRODUCTION

1. My name is Lawrence E. Carlson, and I am a Professor Emeritus of Mechanical Engineering at the University of Colorado in Boulder, Colorado. I am also an independent consultant on various matters involving mechanical engineering.

2. I have been engaged by Norman International, Inc. ("Norman") to investigate and opine on certain issues relating to U.S. Patent No. 6,968,884 B2 entitled "MODULAR TRANSPORT SYSTEM FOR COVERINGS FOR ARCHITECTURAL OPENINGS" ("884 patent"), and U.S. Patent Nos. 6,283,192 B2 ("the 192 patent"), 6,648,050 B1 ("the 050 patent") and 8,230,896 B2 ("the 896 patent") in connection with Norman's petitions of *inter partes* review of those patents.

3. I understand that, according to the first page of the 884 patent, the 884 patent was assigned to Hunter Douglas Inc. Hunter Douglas Inc. is therefore referred to as the "Patent Owner" in this document.

4. In this declaration, I will discuss the technology related to the 884 patent, including an overview of that technology as it was known at the time of the earliest date to which the 884 patent may claim priority—March 23, 1999.

5. This declaration is based on the information currently available to me. To the extent that additional information becomes available, I reserve the right to supplement my opinions following further investigation and study, which may include a review of documents and information that may be produced, as well as testimony from depositions that may not yet be taken.

II. SUMMARY OF OPINIONS

6. The 884 patent describes a modular system of components to retract and extend a window covering. I have been asked by Norman's counsel to analyze claims 5-7 of the 884 patent, out of a total of 14 claims issued by the Patent Office. Claims 5-7 recite systems for covering an architectural opening, including a spring motor, a transmission, and a one-way friction brake.

7. Based on my review of the evidence and facts, it is my opinion that the claimed combination in each of claims 5-7 contains nothing novel or inventive, and, under the patentability standard of 35 U.S.C. § 103(a) explained to me by Norman's counsel as stated below, claims 5-7 are unpatentable and invalid.

8. Specifically, the components, their functions, and interconnections within the claimed systems are well-known mechanical components and are based on routine mechanical engineering designs that were documented before the earliest priority date of the 884 patent. Claims 5-7 are mere obvious and routine combinations of components and features that were known in the window

coverings industry and mechanical engineering in general. More specifically with respect to claims 5 and 6, each of the following features of (1) the recited spring motor including "a coil spring and a power spool, wherein said coil spring wraps onto and off of said power spool," and (2) the recited rotating output "operatively connected to the power spool of the spring motor," and (3) the recited one-way friction brake to provide "a braking force that stops the rotation of the rotating output" is not novel and their combinations with other features in claims 5 and 6 are not novel. With respect to claim 7, the recited "one-way friction brake operatively connected to said rotating output" is not novel nor are the features of "said one-way friction brake providing braking force opposing the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions" and "wherein said one-way brake applies a braking force opposing rotation of the rotating output for movement of the covering to the extended position while permitting free rotation for movement of the covering to the retracted position."

9. The prior art references cited below disclose the spring motor, transmission, and one-way friction brake, among other claimed elements in claims 5-7, either individually or in combination.

10. As described in further detail below, it is my opinion that claims 5 and 7 are rendered obvious by Japanese Unexamined Patent Application Publication

S54-38648 ("Tachikawa") in view of U.S. Patent No. 3,327,765 ("Strahm"). It is further my opinion that claim 6 is rendered obvious by Tachikawa in view of Strahm, and further in view of U.S. Patent No. 6,293,329 ("Toti").

11. Also, as described in further detail below, it is my opinion that claims 5 and 7 are rendered obvious by Tachikawa in view of G.B. Patent No. 1,174,127 ("Skidmore") and further in view of U.S. Patent No. 1,870,532 ("Schuetz").

12. Also, as described in further detail below, it is my opinion that claim 5 is rendered obvious by U.S. Patent No. 2,390,826 ("Cohn") in view of Strahm, and further in view of U.S. Patent No. 6,056,036 ("Todd"). It is further my opinion that claim 6 is rendered obvious by Cohn in view of Strahm, and further in view of Todd and U.S. Patent No. 6,293,329 ("Toti").

13. Also, as described in further detail below, it is my opinion that claim 7 is rendered obvious by Cohn in view of Strahm.

14. For purpose of my analysis in this declaration only and based on the disclosure and file history of the 884 patent, and under the Patent Office's standard of "broadest reasonable construction in light of the specification of the patent" to one of ordinary skill in the art, I provide my proposed construction of certain terms in claims 5-7 in a later part of this declaration.

15. The subsequent sections of this declaration will first provide my qualifications and experience and then describe details of my analysis and observations.

III. QUALIFICATIONS AND EXPERIENCE

A. Education and Work Experience

16. I received my Doctorate (D.Eng.) and Masters (M.S.) Degrees in Mechanical Engineering from the University of California at Berkeley in 1971 and 1968, respectively. I also received a Bachelor of Science in Mechanical Engineering from the University of Wisconsin in 1967.

17. I have spent nearly 40 years educating engineering students on mechanical and component design, primarily in the Department of Mechanical Engineering at the University of Colorado at Boulder. I was an Assistant Professor from 1974 to 1978, a tenured Associate Professor from 1978 to 1994, and a tenured Professor from 1994 to 2010, when I became a Professor Emeritus. Prior to joining the faculty of the University of Colorado, I was an Assistant Professor of Mechanical Design in the Materials Engineering Department at the University of Illinois at Chicago from 1971 to 1974.

18. I was also a founding co-director of the Integrated Teaching and Learning Laboratory and Program for the College of Engineering and Applied Science at the University of Colorado and have received several teaching awards

for my work at the University of Colorado, including the Bernard M. Gordon Prize for Innovation in Engineering and Technology Education from the National Academy of Engineering in 2008. A copy of my CV is included in Attachment A.

19. As a Professor of Mechanical Engineering, I regularly taught mechanical design courses at the University of Colorado beginning in the 1970's, including Component Design, Design for Manufacturability, Invention and Innovation, and hands-on design project courses at the undergraduate and graduate levels. The catalog description for the Component Design course (MCEN-3025) is the "[a]pplication of mechanics and materials science to the detailed design of various machine elements including shafts, bearings, gears, brakes, springs, and fasteners." It was my responsibility to teach engineering students how to describe and apply these fundamental machine elements to many types of mechanical systems. I have also reviewed several textbooks relating to component design during the course of my career.

20. In addition to my extensive teaching experience, I also have more than 40 years of practical experience in mechanical design and research in numerous fields, including rehabilitation engineering, upper-limb prosthetics, consumer products, sculptures, and products to help developing countries. This includes the supervision of undergraduate and graduate research projects, most of which involved hands-on mechanical design in countless areas, including

interactive learning exhibits, sporting equipment, and consumer products. My personal design efforts include a turbine-based flowmeter, a human-powered water pump, and a counterbalance mechanism for a computer monitor that allows it to float in space. I have had a supervisory and collaborative role in many other mechanisms, including a patented releasable ski binding, an improved spring-loaded rock climbing cam, and an automatic drywall screw gun. Many of these designs and design tests have been described in two dozen of my publications, which are listed in my CV (Attachment A).

21. For my doctoral research project, I designed, built, and tested a pneumatically-powered above-elbow prosthesis. This complex mechanical design utilized a variety of relevant mechanical components including bevel and spur gears, springs, cams, shafts, a clutch, pulleys, pneumatic cylinders, and other components to coordinate wrist and elbow rotation in various directions.

22. I am also a named inventor of five United States patents: (1) Patent No. 4,461,085 issued July 24, 1984, entitled "Goniometer"; (2) Patent No. 4,990,162 issued February 5, 1991, entitled "Rotary hand prosthesis"; (3) Patent No. 5,800,571 issued September 1, 1998, entitled "Locking mechanism for voluntary closing prosthetic prehensor"; (4) Patent No. 7,458,598 issued December 2, 2008, entitled "Telemark binding with releasable riser plate assembly"; and (5)

Patent No. 8,560,031 issued October 15, 2013, entitled "Extending socket for portable media player."

23. A true and accurate copy of my CV is included in Attachment A, which will supplement the additional details about my education and experience above.

B. Compensation

24. I am being compensated at the rate of $200 per hour for the services I am providing in this case. The compensation is not contingent upon my performance, the outcome of this *inter partes* review or any other proceeding, or any issues involved in or related to this *inter partes* review.

C. Documents and Other Materials Relied Upon

25. The documents on which I rely for the opinions expressed in this declaration are the 884 patent, the prosecution history for the 884 patent, the prior art references and information discussed in this declaration, and any other references specifically identified in this declaration, in their entirety, even if only portions of these documents are discussed here in an exemplary fashion. I also relied on my own experience and expertise in the relevant technologies and systems that were already in use prior to, and within the timeframe of the earliest potential priority date of the claimed subject matter in the 884 patent—March 23, 1999.

IV. STATEMENT OF LEGAL PRINCIPLES

A. Claim Construction

26. Norman's counsel has advised that, when construing claim terms, a claim subject to *inter partes* review receives the "broadest reasonable construction in light of the specification of the patent in which it appears." Norman's counsel has further informed me that the broadest reasonable construction is the broadest reasonable interpretation of the claim language, and that any term that lacks a definition in the specification is also given a reasonably broad interpretation.

B. Obviousness

27. Norman's counsel has advised that obviousness under pre-AIA 35 U.S.C. § 103 effective before March 16, 2013 is a basis for invalidity. I understand that where a prior art reference discloses less than all of the limitations of a given patent claim, that patent claim is invalid if the differences between the claimed subject matter and the prior art reference are such that the claimed subject matter as a whole would have been obvious at the time the invention was made to a person having ordinary skill in the relevant art. I understand that obviousness can be based on a single prior art reference or a combination of references that either expressly or inherently discloses all limitations of the claimed invention.

28. Norman's counsel has explained that prior art needs to be either (a) in the same field of endeavor as the claimed invention, even if it addresses a different

problem than the claimed invention, or (b) reasonably pertinent to the problem faced by the inventor, even if it is not in the same field of endeavor as the claimed invention. I understand that prior art is reasonably pertinent to the problem when it would have logically presented itself to an inventor's attention in considering the problem. Norman's counsel has also explained that in a simple mechanical invention, a broad spectrum of prior art must be explored, and it is reasonable to inquire into other areas where one of ordinary skill in the art would be aware that similar problems exist, including where other areas have inventions with similar structure and function.

29. Norman's counsel has also explained that a conclusion of obviousness can be supported by a number of reasons. Obviousness can be based on inferences, creative steps, and even routine steps and ordinary ingenuity that an inventor would employ. A conclusion of obviousness can be supported by combining or substituting known elements according to known methods to yield predictable results, or by using known techniques to improve similar devices in the same way, or by trying predictable solutions with a reasonable expectation of success, among other reasons.

V. LEVEL OF ORDINARY SKILL IN THE ART

30. I understand from Norman's counsel that the claims and specification of a patent must be read and construed through the eyes of a person of ordinary

skill in the art at the time of the priority date of the claims. I have also been advised that to determine the appropriate level of a person having ordinary skill in the art, the following factors may be considered: (a) the types of problems encountered by those working in the field and prior art solutions thereto; (b) the sophistication of the technology in question, and the rapidity with which innovations occur in the field; (c) the educational level of active workers in the field; and (d) the educational level of the inventor.

31. The relevant technologies to the 884 patent are mechanical design components used for spring motors and friction brakes. The 884 patent discloses the use of these spring motors and friction brakes in systems for covering an architectural opening, such as a window covering, although there are numerous potential and known applications for spring motors, one-way braking mechanisms, and friction brakes.

32. The technical problems encountered in these types of systems, and specifically the use of spring motors and friction brakes in systems for covering an architectural opening, involve basic, straight-forward, routine and well-known mechanical device solutions. This technology is not sophisticated, and the components of this technology—spring motors, one-way friction brakes, lift cords, and transmissions—are basic design components that have been in use long before the earliest potential priority date of the 884 patent, which is March 23, 1999.

33. Of note, the 884 patent's recitation of a one-way friction brake in the claims is a restatement of what the 884 patent actually discloses. The 884 patent instead describes and schematically illustrates *a one-way clutch mechanism that is in series with a friction brake mechanism*,and terms this combination as a one-way friction brake module (e.g., variable or manually adjustable).

34. Based on the above considerations and factors, it is my opinion that a person having ordinary skill in the art would have an associate's degree or a bachelor's degree in mechanical engineering or a related field involving mechanical design coursework and a few years of working experience in the area of mechanical design. This description is approximate and additional educational experience in mechanical design could make up for less work experience in mechanical design and vice versa.

VI. TECHNOLOGY BACKGROUND OF CLAIMED SUBJECT MATTER OF THE 884 PATENT

35. Technology related to window covers—including spring motors and friction brakes for window covers— involves basic mechanical design components. The components disclosed in the 884 patent, including spring motors, one-way friction brakes, lift cords, and transmissions, have been well known individually and in various combinations long before the 884 patent was filed.

A. Spring Motors

36. Spring motors (which can also be referred to as spring drives) are basic mechanical devices with numerous applications. At its most fundamental level, a spring is a mechanical element that exerts a force when deformed. Mechanical springs are used in machines to exert force, to provide flexibility, and to store or absorb energy. There are several types of springs. In general, springs can be classified as wire springs, flat springs, or special-shaped springs, although there are variations within these classifications. Flat springs include, for example, cantilever springs, elliptical springs, wound motor- or clock-type power springs, and Belleville springs. Attachment B to this declaration is a true and accurate copy of a chapter entitled "Mechanical Springs" from a mechanical engineering textbook that I regularly required when I taught the junior-level Component Design course, which is required of all mechanical engineering students. It was published prior to the relevant priority date and provides additional background information on springs known to one of ordinary skill in the art.

37. The particular spring disclosed in the 884 patent is termed a "coiled spring" or "coil spring." The term "coiled spring" or "coil spring" more commonly refers to helical extension or compression springs, or clock springs, for example. In my experience, the type of spring shown and described in the 884 patent is more properly termed a "constant-force spring" or a "flat spiral spring."

This type of spring is made from a strip of flat spring material (e.g., usually steel) that has been wound to a given curvature so that in its relaxed condition it is in the form of a tightly wound coil. (Attachment B at 443.) The unique characteristic of this type of spring is that the force exerted is independent of the deflection. In other words, the force required to uncoil a "constant-force spring" remains approximately constant, which is why it is called a "constant-force spring." (*Id.*) In reality, the force required to uncoil the spring actually has slight variations, but "constant-force" is generally understood to be the best word available to describe the force-deflection characteristics of this type of spring. It is also the term used by manufacturers who produce and sell this type of spring. A common example of this type of spring is the tape measure.

38. Many springs, such as the helical extension spring used to close screen doors, have a positive spring rate; i.e., the force increases linearly with deflection. Constant-force springs, on the other hand, generally have a zero spring rate, although it was well-known before the relevant date for the 884 patent that constant-force springs can also be manufactured to have either a positive or a negative spring rate, meaning that the force required to uncoil the spring can either increase or decrease with deflection. (*See, e.g.*, Attachment B at 443.) Based on my experience as an educator in mechanical design, this is all basic knowledge that has been taught to engineering students for decades and is widely available in

textbooks like Mechanical Engineering Design. This is also consistent with the 884 patent, which discloses that the spring motor is "preferably" a "constant force" motor. (884 patent, 5:5-16.) A person of ordinary skill in the art would have been knowledgeable about this known element.

39. When a constant-force spring is mounted on two drums, as is disclosed in the 884 patent, the result is a constant-force spring motor. Constant-force spring motors were well understood in the art long before the 884 patent, including design formulas and suggestions. For example, Attachment C to this declaration is a true and accurate copy of a chapter entitled "Springs" from a mechanical engineering reference text published prior to the relevant priority date. (Shigley, J. & C. Mischke, Standard Handbook of Machine Design (1986) in Attachment C.) This text provides design formulas and suggestions for constant-force spring motors. (*See, e.g.*, *id.* at 24-10 - 20-10-4.)

B. One-Way Friction Brake

40. A brake is a device usually associated with rotation that absorbs or transfers the energy of rotation to slow or stop a machine or an individual component. In a friction brake, the brake absorbs or transfers that energy through surface resistance, which depends on the coefficient of friction between the two contacting surfaces. The resistance force opposes the direction of motion, and is equal to the contact force between the two surfaces multiplied by the coefficient of

friction. If a friction brake only absorbs or transfers the energy of rotation when the machine or individual component is rotating in one direction, that friction brake can be described as a one-way friction brake. Rotation in the opposite direction is relatively free, hence the term “free wheeling” or “overrunning”.

41. Brakes generally, and more specifically the one-way friction brake disclosed in the 884 patent, were widely known and used in mechanical design long before the relevant date for the 884 patent in a host of applications. One common example is a fishing reel, which allows free rotation in one direction and a controlled drag torque in the opposite direction. Such a one-way braking mechanism in the fishing reel was commercially available many years before the priority date of the 884 patent.

C. Combinations of Design Components

42. All engineers, including mechanical engineers, are taught the design process, which is a general method for solving a wide variety of problems ranging from dams to electronic circuits to mechanical devices. Once functional design requirements have been specified, students are taught to generate as many alternate design concepts as possible for each component of the system, and to explore various combinations of the individual elements. For most basic and ordinary mechanical designs, such as the designs in the 884 patent, individual elements are chosen from a finite group of ordinary components and predictable solutions

(across a range of mechanical applications), and combinations and arrangements are chosen with a reasonable expectation of success.

43. A person of ordinary skill in the art of mechanical design would be educated and experienced in the various advantages and disadvantages of combining mechanical design components, such as spring motors, friction brakes, lift cords, and transmissions. For example, Mechanical Engineering Design (1985) is a widely known and respected textbook from which I taught engineering students about mechanical design. This textbook is a revised version of the same text I studied as an undergraduate in the 1960's. The textbook specifically addresses constant-force springs (Chapter 10), gear transmissions (Chapter 13), and bevel gears (Chapter 15) for purposes of mechanical design.

VII. OVERVIEW OF THE 884 PATENT

44. The 884 patent is directed to several individual functional modules that are used in modular transport system to retract and extend a window covering. For example, these modules include motor modules, transmission modules, brake modules, etc. that are categorized in a group based on function, such as a power and power transmission group, lift and/or tilt stations group, tilt mechanisms group, or the rest of the blind (see 884 patent at 3:10-4:16). One of the key points of emphasis in the 884 patent is the importance of modularity and interchangeability of these individual modules, which can form various transport

systems to retract and extend a window covering that satisfies a multitude of different scenarios for covering an architectural opening, e.g., including various sizes and weights of window coverings.

45. Ironically, while the 884 patent provides a voluminous description describing many embodiments of these individual modular components, the claims (i.e., claims 5-7) are vaguely and confusingly worded with terminology not expressly described in the specification, including the use of structural elements with unclear and imprecise connections and functions implemented by the structural elements to perform the operation claimed.

46. As an example, claims 5 and 7 recite a “one-way friction brake” that “provid[es] a braking force that stops” [claim 5] or “opposing” [claim 7] “the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions.” Neither claim 5 nor claim 7 defines the structure of how the one-way friction brake provides such braking force, and neither claim 5 nor claim 7 recites the structure of the one-way friction brake to enable the functional limitation of stopping or opposing one rotational direction while permitting free rotation in the other rotation, which is claimed in claim 5 and claim 7, respectively.

47. The relevant part of the description and drawings of the 884 patent on a “one-way friction brake” discloses a one-way clutch mechanism that is in series

with a "friction brake." Devices that allow rotation freely in one direction while preventing rotation in the opposite direction are more commonly known as "overrunning clutches." Overrunning clutches may take many forms, and have been widely used in diverse mechanical applications. Similarly, friction brakes may take many forms and have been used in very many applications. Friction brakes oppose rotation in either direction, but combining an overrunning clutch with a brake, such as a friction brake, will create a "one-way brake" that opposes rotation in one direction but not the other. Whether or not it actually stops rotation depends on the specific design of the brake, as well as the force applied to the brake.

48. The 884 patent attempts to address the problem in window coverings where the force required to raise the blind varies with the raising of the blind, as slats or cells stack on a moving rail. This problem, and the solution posed in the 884 patent, was not new, and was specifically addressed in the prior art, for example, in Tachikawa and others described in detail below. It is my opinion that the 884 patent merely restates this known problem and attempts to solve the problem with an obvious arrangement of well-known features published in the prior art below, and known in the mechanical arts.

49. The 884 patent and the related U.S. Patent No. 8,230,896 B2 ("896 patent") both claim priority to the same Provisional Application No. 60/125,776,

filed on March 23, 1999. I have also been engaged by Norman to investigate and opine on the 896 patent.

VIII. IDENTIFICATION OF THE PRIOR ART

50. I have been advised by Norman's counsel that the earliest potential priority date for the claims of the 884 patent is the filing date of the earliest application to which the 884 patent claims priority, which I understand is March 23, 1999.

51. As explained below, it is my opinion that the following prior art references, which are listed as Exhibits to the Petition for Inter Partes Review of the 884 patent, disclose all technical features in the challenged claims of the 884 patent by rendering them obvious. Therefore, at least claims 5, 6, and 7 of the 884 patent contain nothing novel or inventive and thus are unpatentable.

- Japanese Unexamined Patent Application Publication S54-38648 to Tachikawa ("Tachikawa") (published March 23, 1979);
- U.S. Patent No. 3,327,765 to Strahm ("Strahm") (issued June 27, 1967);
- G.B. Patent No. 1,174,127 to Skidmore ("Skidmore") (published December 10, 1969);
- U.S. Patent No. 1,870,532 to Schuetz ("Schuetz") (issued August 9, 1932);

- U.S. Patent No. 2,390,826 to Cohn ("Cohn") (issued December 11, 1945);
- U.S. Patent No. 6,056,036 to Todd ("Todd") (filed May 1, 1997; issued May 2, 2000); and
- U.S. Patent No. 6,293,329 to Toti ("Toti") (filed December 11, 1997, issued September 25, 2001)

52. Each prior art reference cited above constitutes analogous art for the purpose of an obviousness analysis under 35 U.S.C. §103. I understand that another expert retained for this *inter partes* review, Patrick Foley, has concluded that each prior art reference is analogous art, and I agree with that conclusion and his reasoning.

IX. CLAIM CONSTRUCTION

53. In conducting my analysis of the asserted claims of the 884 patent, I have applied the legal understandings I set out below regarding claim constructions consistent with the "broadest reasonable construction" ("BRI") standard described above, and offer them only for this *inter partes* review. The claim constructions do not necessary reflect the appropriate claim constructions to be used in litigation proceedings, such as litigation in a district court, where a different standard applies.

54. I understand that claims of a patent are interpreted from the perspective of one of ordinary skill in the art at the time of the invention in light of the intrinsic evidence, which includes the language of the claim itself, the detailed description and figures of the patent and the relevant prosecution history from the United States Patent and Trademark Office. Other evidence (such as dictionaries and textbooks) not in the written record of the patent, and other extrinsic evidence also may be considered if it is consistent with (not contradictory to) the intrinsic evidence. I also understand that, as a general matter, a claim should not be limited to a preferred embodiment, in that in certain cases, the scope of the right to exclude may be limited by a narrow disclosure. I also understand that the full scope of the claims must be supported by the specification.

A. "System for Covering an Architectural Opening"

55. The proposed BRI claim construction for the term is "a group of functional components that retract and extend a window blind or shade."

56. The basis in the 884 patent for the BRI claim construction includes, e.g., Abstract; 1:14-18; and FIGS. 1-13C and associated textual description. The 884 patent describes systems of window blinds or shades, including Venetian blinds, pleated shades, or other horizontal or vertical blinds or shades. As best discerned from the specification, the group of functional components are interchangeable, modular components that are easily configured to (ideally) any

size or configuration of architectural opening (e.g., window) to enable the retraction and extension of the window blind or shade.

B. “Covering”

57. The proposed BRI claim construction for the term is “a group of slats, a pleated fabric shade, or a roller shade.”

58. The basis in the 884 patent for the BRI claim construction includes: blind slats (FIGS. 1-7, 11-13, and 13B-13C), pleated fabric shades (FIGS. 8-10, 13A, and 214-216), and roller shades (FIGS. 217-220) and associated textual description. The 884 patent specifically discloses Venetian blinds, pleated shades, or other horizontal or vertical blinds or shades (1:15-18).

C. “Power Spool”

59. The proposed BRI claim construction for the term is “a component to hold and wind a spring.” The basis in the 884 patent for the BRI claim construction includes, e.g., 17:55-56; 19:5-6; 17:49-51, and FIGS. 16, 21-25, 28, and 30-38 and associated textual description.

D. “Spring Motor”

60. The proposed BRI claim construction for the term is “a mechanism that uses a spring to output mechanical power to another component or components.”

61. The 884 patent discloses several spring motors under various terms including "spring motor," "spring motor module," "coiled spring motor," "coiled spring motor power unit," "coil spring motor," "coiled spring motor module," "coil spring motor power module," "spring motor power module" or "spring motor power unit." The disclosed spring motors include, at least, a spring and a spool that winds or unwinds the spring. The disclosed spring motors are in coaxial and transaxial arrangements of the spring and its spools with respect to a rotating rod in which the winding or unwinding of the spring motor causes the rotating rod to rotate in raising or lowering the covering. Examples of support in the 884 patent include 17:32-24:34; FIGS. 14-63 and associated textual description.

E. "Rotating Output"

62. The proposed BRI claim construction for the term is "a component to rotate upon actuation by another component."

63. The 884 patent describes a lifting rod that provides the function and structure of this "rotating output" such as the lift rod 26 featured in several embodiments of a blind, e.g., FIGS. 1-13C and 214-216, or elongated spool 1070, e.g., FIGS. 217-220 and associated textual description, therefore providing the basis of the BRI claim construction. The claim term "rotating output" is not explicitly used in the description or drawings of the 884 patent outside the claims, but the 884 patent describes a lifting rod that provides the function and structure of

the claimed "rotating output" such as the lift rod 26 featured in several embodiments of a blind.

F. "Lift Cord"

64. The proposed BRI claim construction for the term is "a flexible line or cord capable of being wound or unwound, to cause lifting or lowering of the covering of a window blind or shade."

65. The basis in the 884 patent for the BRI claim construction includes description and figures of a lift cord 16 featured in several embodiments of a blind, e.g., FIGS. 1-13C and 214-216 and associated textual description.

G. "One-Way Friction Brake"

66. The proposed BRI claim construction for the term is "a mechanism that applies a frictional braking force against a rotational motion of the rotating output in one rotational direction and insubstantial frictional braking force in the other rotational direction."

67. The claim term "one-way friction brake" is not found in the description or drawings of the 884 patent outside the claims. In addition, there is not sufficient support in the original specification for the following claimed features for "one-way friction brake" as recited in claim 5: "said one-way friction brake providing a braking force that stops the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of

said directions" Neither the original specification of the 884 patent provides sufficient support for claimed features for "one-way friction brake" as recited in claim 7: "said one-way brake applies a braking force opposing rotation of the rotating output for movement of the covering to the extended position while permitting free rotation for movement of the covering to the retracted position."

68. The basis in the 884 patent for the BRI claim construction for "one-way friction brake" is based on the claim language in the claims of the "one-way friction brake." The specification of the 884 patent includes description and figures of a variable brake module 900, e.g., [58:51-59:64] and FIGS. 175-182, and an adjustable brake module 900A, e.g., [59:65-60:21] and FIGS. 183A-190. The above claim term "one-way friction brake" is not found in the description or drawings of the 884 patent outside the claims.

69. The only examples provided in the 884 patent that correspond, at least somewhat, to what claims 5-7 refer to as a "one-way friction brake" are found in FIGS. 175-190 and associated textual description (e.g., 58:51-59:64 and 59:65-60:21) for the variable brake 900 and the adjustable brake 900A. The variable brake 900 and the adjustable brake 900A are actually a friction brake that is coupled in series to a one-way clutch. Notably, there are separate components, albeit nested together and connected in series. As shown in FIGS. 175-190 and associated textual description, the brake includes brake drum 926, which rotates,

and brake shoe 928, which is stationary. Spring 942 generates the contact force between the two. Being a friction brake, it can rotate continuously with a drag force dependent on coefficient of friction of the two parts in contact and the contact force. Taken by itself, the brake as shown would generate a resistance to rotation in either direction of drum 926 – although it may not be the same in each direction based on the specific geometry of how the shoe contacts the drum. What the 884 patent refers to as the "toothed drive 932" functions as a one-way clutch because of the teeth 940, 940A. They are engaged by the inclined planes 936 – but only when the input shaft 914 is rotated clockwise (CW), as indicated by arrow 930. Put another way, when shaft 914 is rotated CW, toothed drive 932 engages positively (no slipping) to turn the brake drum 926 CW. There is a resistance torque generated by the brake, presumably in a direction to maintain the position of the blinds. When shaft 914 rotates counterclockwise (CCW), the teeth are disengaged, so that shaft 914 and the brake drum are uncoupled. The brake drum remains stationary and shaft 914 freewheels – with, of course, a small amount of "inherent" friction.

70. Regarding the limitation of the one-way friction brake as recited in claim 5, "said one-way friction brake providing a braking force that stops the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions," the BRI claim construction is

inclusive to enable the one-way friction brake to be capable of applying a frictional braking force sufficient to stop the rotating output's rotational motion. Presumably, that is the intent of a brake, but whether or not that function is actually achievable depends on the design. For example, if the spring 942 does not generate sufficient contact pressure between the brake shoe 928 and brake drum 926, it could conceivably slip if the window covering (e.g., blind) were too heavy.

71. Regarding the limitation of the one-way friction brake as recited in claim 7, "said one-way brake applies a braking force opposing rotation of the rotating output for movement of the covering to the extended position while permitting free rotation for movement of the covering to the retracted position," the BRI claim construction is inclusive to enable the one-way friction brake to be capable of applying a frictional braking force that opposes the rotating output's rotational motion, regardless if the braking force is sufficient or insufficient to stop the rotational motion, for example. The basis in the 884 patent for the BRI claim construction of this limitation includes description and figures of the variable brake module 900, e.g., 58:51-59:64.

H. "Transmission"

72. The proposed BRI claim construction for the term "transmission" used in claim 6 is "a mechanism coupled to the spring motor and the rotating

output to transmit motion between the rotation of the spring motor and the rotation of the rotating output."

73. The basis in the 884 patent for the BRI claim construction includes the embodiments of transmission modules described in 24:35-33:63 and FIGS. 64-90B, 208A, 208B, and 210.

I. Other Terms

74. I understand that other terms in claims 5-7 will be given their customary and ordinary meaning.

X. UNPATENTABILITY OF THE 884 PATENT CLAIMS

75. I reviewed each of the Tachikawa, Strahm, Toti, Skidmore, Schuetz, Cohn, and Todd references. In the following, I explain my opinion that any alleged invention in claims 5-7 would have been rendered obvious to one of ordinary skill in the art at the time of the invention by the following obvious combinations.

A. Tachikawa In View Of Strahm

1. Reasons To Combine Tachikawa And Strahm

75. Claims 5 and 7 of the 884 patent are unpatentable as being obvious over Tachikawa in view of Strahm. The disclosures of Tachikawa and Strahm are in the same technical field of window coverings of the 884 patent. Like the 884

patent, Tachikawa and Strahm relate to mechanisms that extend and retract a window covering based on a lifting mechanism having a spring motor.

76. Specifically, Tachikawa teaches systems for covering an architectural opening with a Venetian blinds roll-up device powered by a spring motor. With a filing date in September of 1977 and a publication date in March of 1979, Tachikawa predates the earliest possible priority date of March 1999 by the 884 patent by more than 20 years. Tachikawa is part of a body of window covering prior art work long before the filing of the 884 patent and reflects the pre-1999 state of the technology development in the spring-based window covering systems that were specifically designed and engineered for producing variable lifting forces based on substantially constant force springs in spring motors to balance and to hold window blinds against varying weights of the window blinds at different positions between the fully roll-up position and fully roll-down position.

77. Tachikawa discloses the key mechanisms and features of the 884 patent, including substantially identical structures that operate to raise and lower a window covering by providing sufficient lifting force, where the user need only aapply small force for extending and retracting the window covering, and cause the window covering to remain stationary at the user's desired extended or retracted position. See, e.g., Tachikawa 1:22-2:10, and 3:15-4:6. The graph of FIG. 9 in Tachikawa shows the load versus elongation relationship of Tachikawa's spring

motor, which demonstrates that the spring force can be tailored to the specific load requirements of the blind. Comparisons between the claimed subject matter in the claims 5-7 of the 884 patent and the disclosures in Tachikawa are provided in detail in subsequent sections of this Declaration.

78. Moreover, beyond disclosing the features, mechanisms, and structures found in claims 5-7 of the 884 patent, Tachikawa recognized and specifically addressed the same technical issues that are identified by the 884 patent more than 20 years before the 884 patent. For example, Tachikawa states, "*[w]hen venetians blinds are rolled up by turning the operating shaft of the blinds by means of a gear mechanism or the like, the load increases gradually as the roll-up progresses, and conversely when the blinds are rolled down, the load becomes smaller as the roll-down progresses. Therefore, the force necessary to manipulate the operating shaft of the blinds is not constant and changes ceaselessly*" (Tachikawa, 1:14-21) and "*at the start of the roll-up, there is only the load of the lower case 6, but as the roll-up progresses, the load of the slats 5 is applied, so the torque to be applied in order to tum the operating shaft 2 is not constant and changes ceaselessly*" (*Id.*, 3:9-14). Tachikawa offers a venetian blinds roll-up device to address this technical problem that includes: "*a constant force spring [that] is mounted on a drive shaft or operating shaft for performing the rolling up of venetian blinds, and the radius of curvature of the constant force spring is changed in response to the gradually*

increasing change in the gradually increasing load as the blinds are rolled up, so as to constantly generate a spring torque corresponding to the load of the blinds" (*Id.*, 1:5-12, and FIG. 9).

79. In comparison, the 884 patent repeats or restates the same issue in the "Background of the Invention": "*[t]he force required to raise the blind varies directly and approximately linearly with the raising of the blind, increasing from a minimum when the blind is fully lowered to a maximum when the blind is fully raised. This same force also varies directly and approximately linearly with the size and weight of the window covering*" (the 884 patent at 1:59-64).

80. The 884 patent purports to claim a transport mechanism using known mechanisms and components, like those disclosed in Tachikawa and other prior art cited in this Declaration. Specifically, the 884 patent states, "*The primary objective of the present invention is to provide a modular blind transport system which overcomes the shortcomings of prior blind transport systems. Rather than having to design a completely new system for each size and weight of blind, the designs of the present invention provide a system comprised of individual modules which are readily interconnected to satisfy the requirements of a multitude of different blind systems, it also includes the individual modules which make the overall system possible. Accordingly, modularity is an important feature of the present invention. The individual modules in the present invention are contained in housings which*

make each element an independent and self contained module. Each module is easily and readily installed, mounted, replaced, removed, and interconnected within the blind transport system with an absolute minimum of time and expense. Each housing provides the mounting mechanism for its module onto the blind transport system, and removal of the housing also removes all the individual components which make up the module, leaving the balance of the blind transport system essentially unaffected except perhaps for the need to use a longer or shorter connecting rod." (the 884 patent, 3:10-32).

81. The mechanisms and features in claims 5-7 of the 884 patent are simply a belated repeat of the teachings in Tachikawa and other references long before the earliest priority date of the 884 patent. As shown by this Declaration, claims 5-7 merely recite the same fundamental technical teachings and designs in Tachikawa and other prior art cited in this Declaration in somewhat different language recitations. Simply put, claims 5-7 contain nothing but mere well-known features published in the window covering industry decades before the 884 patent, and there is no invention in any of the claims in the 884 patent.

82. Strahm is another example of prior art in the industry of window covering in the public domain long before the filing of the 884 patent by over 30 years. Like Tachikawa, Strahm also addresses the same and other similar technical issues in the 884 patent for covering an architectural opening. The technology

disclosed in Strahm addresses the technical challenge of controlling the rotation of the rotating output that raises and lowers the window covering so that the blind can be reliably raised and lowered by a user to remain in the intended position with minimum user effort.

83. Specifically, Strahm teaches a raising and lowering mechanism for a window covering (e.g., Venetian blind) that includes a one-way brake mechanism that operates a brake against lowering of the covering and releases when the covering is being raised. For example, Strahm discloses "*[i]t is another object of the present invention to provide a raising and lowering mechanism for a blind and including a brake which operates to brake the rate of descent of the blind, so that it can be lowered in a controlled manner, but which is automatically released during raising of the blind so that raising can be performed with the minimum of effort*" (Strahm, at 1:28-34).

84. Both Tachikawa and Strahm employ comparable and commonly known mechanical components and mechanisms that could have been easily combined or interchanged by a person of ordinary skill in the art at the time of the 884 patent. Like Tachikawa, Strahm discloses transport mechanisms and systems for a covering an architectural opening having a covering (e.g., slats), lift cords that extend and retract the slats and wraps onto/off of lift spools, a rotating shaft that causes the lift spools to take up/down the lift cords, and a drive mechanism (e.g.,

motor) that rotates the rotating shaft. These components and mechanisms are structurally comparable and functionally and operationally the same as the components and mechanisms claimed in claims 5-7 of the 884 patent, as well as interchangeable and combinable with the window covering system of Tachikawa and other prior art cited in this Declaration. Comparisons between the claimed subject matter in claims 5-7 of the 884 patent and the disclosures in Strahm are provided in detail in subsequent sections of this Declaration.

85. Because of the close linkages amongst Tachikawa and Strahm with respect to the subject matter in the 884 patent, there is a motivation or suggestion in the teachings by Tachikawa and Strahm to enable a person having ordinary skill in the art to combine the teachings of Tachikawa and Strahm. Such combinations render claims 5-7 of the 884 patent unpatentable.

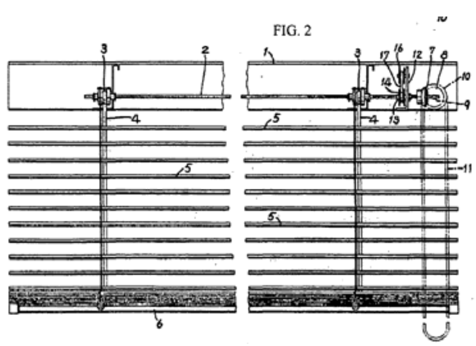
86. Therefore, in light of the entire disclosures in Tachikawa and Strahm, and in view of the common recognition of technical issues in window blinds against raising/lowering the blind to a desired location to remain stationary despite varying blind weights at different raised blind positions by Tachikawa and Strahm, and in recognition of the substantially similar designs of components and mechanisms disclosed by Tachikawa and Strahm, Petitioner respectively submits to the Board that it is obvious for a person of ordinary skill in the art of the window covering to combine Tachikawa and Strahm for their teachings published at a time

before the earliest priority date of 1999 for the 884 patent. In fact, both Tachikawa and Strahm provide teachings that would motivate a person of ordinary skill in the art of the window covering to make the combining, including the specific combinations as provided in this Declaration.

87. With the above background information on Tachikawa and Strahm with respect to the 884 patent, the following sections provide the detailed analysis of how Tachikawa and Strahm collectively render claims 5-7 obvious and unpatentable.

2. Claim 5 Is Rendered Obvious By Tachikawa In View Of Strahm

88. **Preamble**: <u>A system for covering an architectural opening, comprising:</u> Tachikawa discloses a venetian blinds roll-up device, shown by venetian blind in FIG. 2 of Tachikawa (reproduced here). Therefore, Tachikawa teaches “a system for covering an architectural opening” in claim 5.



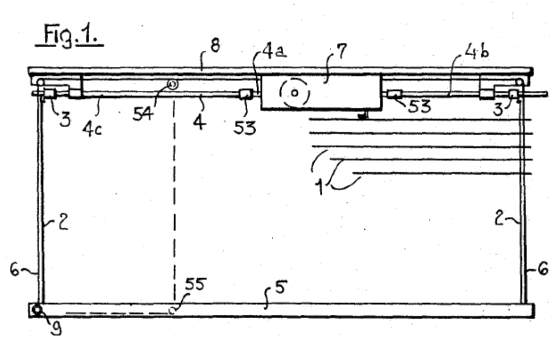


FIG. 2 of Tachikawa

FIG. 1 of Strahm

89. Like Tachikawa, Strahm also discloses “a system for covering an architectural opening” of Claim 5 in its FIG. 1 (reproduced here) and associated text.

90. **Element [5A]**: <u>a covering movable between an extended position for covering the opening and a retracted position for uncovering the opening;</u> Tachikawa discloses a plurality of slats 5 of the venetian blind that can be retracted and extended about a window opening, as discussed at 2:11-18 and shown in FIG. 2 of Tachikawa. Accordingly, the slats 5 of Tachikawa corresponds to and discloses the covering of Claim 5.

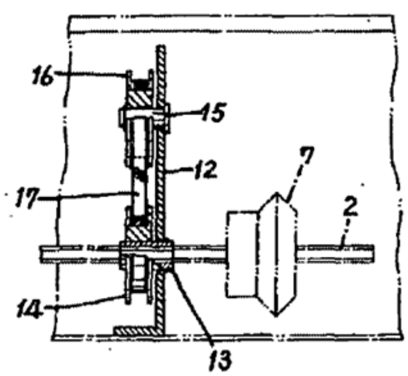
91. Like Tachikawa, Strahm also discloses the window covering of Claim 5 in its FIG. 1 shown as parallel slats 1 of the blind.

92. **Element [5B]**: <u>a spring motor including a coil spring and a power spool, wherein said coil spring wraps onto and off of said power spool;</u> Tachikawa discloses a spring 17 (i.e., “coil spring”) wound diagonally between drum 14 (i.e., “power spool”) and drum 16, forming a spring motor, shown in FIG. 5 (reproduced here). The spring 17 wraps onto and off of drum 14. See, e.g., Tachikawa at 1:22-2:10, 3:15-4:6 and FIGS. 5-8. Accordingly, the spring 17 wound between drums 14 and 16 of Tachikawa corresponds to and discloses the spring motor of Claim 5.

93. Similar to Tachikawa, Strahm also discloses a drive for raising and lowering the blind that can include a motor (e.g., see Strahm at 4:10-11), which corresponds to the spring motor of Claim 5.

94. **Element [5C]**: <u>a rotating output operatively connected to the power spool of the spring motor;</u> Tachikawa discloses an operating shaft 2, operatively connected to the drum 14 of the spring 17. See, e.g., Tachikawa at 2:11-22, 3:15-22 and FIGS. 2, 5 and 6 (reproduced here). Accordingly, the shaft 2 of Tachikawa corresponds to and discloses the rotating output of Claim 5.

FIG. 6



95. Like Tachikawa, Strahm also discloses the rotating output of Claim 5 in its FIG. 1 shown as operating shaft 4 of the blind.

96. **Element [5D]**: <u>a lift cord operatively connected to the rotating output and to the covering;</u> Tachikawa discloses a tape 4, operatively connected to shaft 2 and slats 5, where tape 4 is wound onto drum 3 attached to shaft 2 and is passed through slats 5 and coupled to lower case 6. See, e.g., Tachikawa at 2:11-22, and

FIGS. 2 and 4. Accordingly, the tape 4 of Tachikawa corresponds to and discloses the lift cord of Claim 5.

97. Like Tachikawa, Strahm also discloses pull tapes 6 (e.g., see Strahm at 2:49-52 and FIG. 1), which corresponds to the lift cord of Claim 1.

98. **Element [5E]**: said rotating output being rotatable in clockwise and counterclockwise directions to move the covering between its extended and retracted positions; Tachikawa discloses that shaft 2 is rotatable in both directions (clockwise and counterclockwise) to move the slats 5 between extended and retracted positions. Specifically, Tachikawa discloses that "*The present invention relates to a venetian blinds roll-up device characterized in that a constant force spring is mounted on the operating shaft of the venetian blinds, and the radius of curvature of the constant force spring is changed in response to the change in load as the blinds are rolled up, so as to constantly generate a spring torque in the opposite direction and identical to the torque due to the load of the blinds acting upon the operating shaft, which has the effect that the force for manipulating the operating shaft in order to perform roll-up and roll-down of the blinds can be a: small, constant force regardless of the position of the blinds, and that the blinds do not fall spontaneously due to the weight of the blinds if roll-up is stopped mid-way, but are rather stopped at that position by the spring torque*" (*Id.*, 1:22-2:10). Therefore, Tachikawa discloses Element [E] in Claim 5.

99. **Element [5F]**: a one-way friction brake operatively connected to said rotating output, said one-way friction brake providing a braking force that stops the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions. The Tachikawa system for covering an architectural opening is designed to raise and lower the window covering to a desired position and to balance the blind to remain stationary. Tachikawa provides a blinds roll-up device including a spring motor. Another way to achieve this is to also include a brake to further assist in balancing the window covering at the desired position.

100. Strahm discloses a one-way friction brake mechanism for window blinds. The disclosed one-way friction brake has conical washers 33 and 36 that contact wall 34 when sleeve 32 is rotated, thus forming a friction brake. The "hand" (direction of winding) of spring 21 which contacts sleeve 19 allows rotation in one direction but not the other. This combination creates a one-way friction brake. Strahm discloses that the one-way friction brake mechanism is operatively connected to rotating shaft 4 (i.e., "rotating output").

101. Strahm's one-way friction brake mechanism is capable of providing a braking force that stops the rotation of the rotating shaft 4 in one direction and permits the rotating shaft 4 to rotate freely in the other direction. See, e.g., Strahm at 3:11-35, 4:31-33, 1:28-34, and FIG. 6 (reproduced here). Accordingly, Strahm's

one-way friction brake corresponds to and discloses the one-way friction brake of Claim 5.

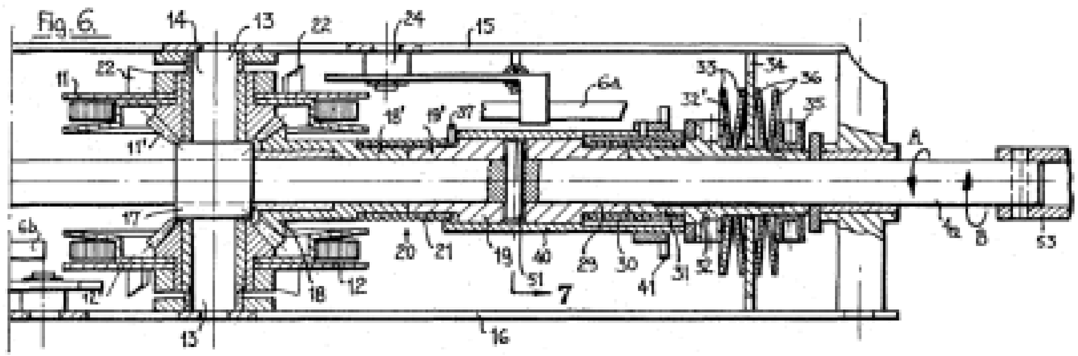


FIG. 6 of Strahm

102. One of ordinary skill in the art at the time of the 884 patent could easily have combined the one-way friction brake of Strahm with the venetian blind of Tachikawa. Notably, the operating shaft 4 of Strahm is substantially the same, both structurally and operationally, as the operating shaft 2 of Tachikawa, allowing for the one-way friction brake of Strahm to easily be operatively coupled to the shaft 2 of Tachikawa.

103. Therefore, the combination of Tachikawa and Strahm teaches each feature recited in claim 5 and renders the subject matter of claim 5 as a whole obvious and unpatentable.

3. Claim 7 Is Rendered Obvious By Tachikawa In View Of Strahm

104. **Preamble**: A system for covering an architectural opening, comprising:

See discussion of relevant teachings in Tachikawa for **Preamble** of claim 5.

105. **Element [7A]**: a covering movable between an extended position for covering the opening and a retracted position for uncovering the opening;

See discussion of relevant teachings in Tachikawa for **Element [5A]** of claim 5.

106. **Element [7B]**: a spring motor; See discussion of relevant teachings in Tachikawa for **Element [5B]** of claim 5.

107. **Element [7C]**: a rotating output operatively connected to the spring motor; See discussion of relevant teachings in Tachikawa for **Element [5C]** of claim 5.

108. **Element [7D]**: a lift cord operatively connected to the rotating output and to the covering; See discussion of relevant teachings in Tachikawa for **Element [5D]** of claim 5.

109. **Element [7E]**: said rotating output being rotatable in clockwise and counterclockwise directions to move the covering between its extended and retracted positions; See discussion of relevant teachings in Tachikawa for **Element [5E]** of claim 5.

110. **Element [7F]**: <u>a one-way friction brake operatively connected to said rotating output, said one-way friction brake providing braking force opposing the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions;</u> Similarly as discussed in **Element [5F]** of claim 5 above with respect to the combination of Tachikawa and Strahm, Strahm discloses a one-way friction brake mechanism, which has conical washers 33 and 36 that contact wall 34 when sleeve 32 is rotated, thus forming a friction brake. The "hand" (direction of winding) of spring 21 which contacts sleeve 19 allows rotation in one direction but not the other, commonly known as a wrap-spring clutch. This combination creates a one-way friction brake. Strahm discloses that the one-way friction brake mechanism is operatively connected to rotating shaft 4 (i.e., "rotating output"). The one-way friction brake mechanism of Strahm provides a braking force that opposes the rotation of the rotating shaft 4 in one direction and permits the rotating shaft 4 to rotate freely in the other direction. See, e.g., 3:11-35, 4:31-33, 1:28-34 and FIG. 6 of Strahm.

111. One of ordinary skill in the art could have easily configured the one-way friction brake of Strahm operatively coupled to the shaft 2 of Tachikawa. Accordingly, the one-way friction brake of Strahm corresponds to the one-way friction brake of claim 7.

112. **Element [7G]**: wherein said one-way brake applies a braking force opposing rotation of the rotating output for movement of the covering to the extended position while permitting free rotation for movement of the covering to the retracted position. Strahm discloses this feature of claim 7. Specifically, the one-way friction brake mechanism of Strahm applies a braking force that opposes the rotation of the rotating shaft 4 when the rotating shaft 4 rotates in a direction to lower parallel slats 1 of the blind, and permits the rotating shaft 4 to rotate freely in the other direction to raise the blind, as discussed in Strahm at 1:28-34.

113. Therefore, the combination of Tachikawa and Strahm teaches each feature recited in claim 7 and renders the subject matter of claim 7 as a whole obvious and unpatentable.

B. Tachikawa In View Of Strahm, And Further In View Of Toti

1. Reasons To Combine Tachikawa, Strahm, and Toti

102. Claim 6 is a dependent claim of claim 5. Claim 6 is unpatentable as being obvious over Tachikawa in view of Strahm and in further view of Toti. Toti discloses such a transmission, and thus the combination of Tachikawa, Strahm and Toti renders claim 6 unpatentable.

103. Toti teaches a system for covering an architectural opening using a coil spring drive. See, e.g., Toti at Title, and 1:11-15. In one example, Toti discloses "*a window cover system of the type comprising an extendible window*

cover and lift means including lift cords attached to the cover for assisting extending and retracting the window cover" that comprises "*a spring drive unit connected to the lift cords for assisting extending and retracting the cover to selected positions*", in which the spring drive unit includes a rotating shaft, a coil spring mounted along the shaft and having a fixed and a rotatable end, and a transmission operatively connected to the coil spring via the rotatable end and to lift spools at the other to alter the rotation between the spring drive unit and the lift spools (*Id.*, claim 1 at 9:4-23). Toti discloses that the transmission of the spring drive unit can include a gear transmission, band transmission, or combination thereof to regulate rotation rate between a rotating body of a coil spring drive and a rotating body coupled to a lifting mechanism of the window covering (*Id.*, 2:26-65).

104. The disclosures of Toti, Tachikawa and Strahm are in the same technical field of window blinds and shades of the 884 patent. Like Tachikawa and Strahm, Toti employs comparable and commonly known mechanical components and mechanisms that could have been easily combined or interchanged by a person of ordinary skill in the art at the time of the 884 patent. Comparisons between the claimed subject matter in claim 6 of the 884 patent and the disclosures in Toti, Tachikawa and Strahm are provided in detail in subsequent sections of this Declaration.

105. For example, Toti states that the various components including the gear and band transmissions can be used "*alone or in essentially any combination*" to accommodate a given blind or cover, and that one familiar with the art will appreciate that the components and arrangements of Toti are applicable in general to window covers that use spring drive mechanism. See, e.g., Toti at 8:55-9:1. Particularly, Toti provides motivation to combine such components and mechanisms with others: "*Adaptation of the system to other articles, objects and systems, including other blinds will be readily done by those of usual skill in the art*" (*Id.*, 8:66-9:1).

106. Moreover, beyond disclosing the features, mechanisms, and structures purported as an invention in claims 5-7 of the 884 patent, Toti recognized and specifically addressed the same technical issues that are identified by Tachikawa and Strahm and the 884 patent before the 884 patent. For example, Toti states, "*as the blind is lowered, the slat weight supported by the lift cords decreases and the compression force of the pleats decreases. However, as the blind is lowered, the spring is wound and the energy stored in the spring increases, such that the increasing torque or force of the spring may then raise the blind in fast, uncontrolled fashion. Also, it may be difficult to keep the blind at a selected position. Furthermore, if the blind is heavy, and requires a strong spring to maintain the blind open, the blind is particularly susceptible to instability and*

uncontrolled raising operation when partially or fully extended or closed. Conversely, when the blind is at or near the upper limit of its travel (i.e., is open), the slat weight supported by the lift cords and the pleat compression is at or near maximum, while the spring torque is at or near minimum." (*Id.*, 2:3-17). Toti offers the spring drive unit to address this technical problem and suggests that: "*[t]he combination of the coil spring, transmission fixed gear ratio, gear friction and the spring buckling braking action allows the spring drive unit 15 to hold the blind 10, 20 in position at even the "heaviest" (uppermost) blind positions, prevents the spring from overpowering the blind, especially when the spring is wound (at the lower blind positions), and allows the blind to be pulled downward to any selected position by gently pulling the blind to that position and, conversely, to be pushed upward to any selected position by gently pushing upward to that position. Little force is required to move the blind up and down, the blind stops accurately at any selected position between and including the fully opened and fully closed positions, and the blind remains at the selected positions.*" (*Id.*, 6:52-65).

107. Because Toti, Tachikawa, and Strahm share common recognition of technical issues in window blinds for raising/lowering the blind to a desired location and remain stationary despite varying blind weights at different raised blind positions, because Toti, Tachikawa, and Strahm disclose substantially similar designs of components and mechanisms in their respective window covering

systems, and because Toti, Tachikawa, and Strahm suggest adaptations to their respective systems using other components and mechanisms known in the art by a person of ordinary skill, there is a motivation or suggestion in the teachings by Toti, Tachikawa, and Strahm to enable a person having ordinary skill in the art to combine the teachings of these references. Such combinations render claim 6 of the 884 patent unpatentable.

2. Claim 6 Is Rendered Obvious By Tachikawa In View Of Strahm And In Further View Of Toti

108. **Preamble and Element [6A]**: A system for covering an architectural opening as recited in claim 5, and further comprising a transmission operatively connected to the spring motor and to the rotating output. In one example of a transmission of Toti, Toti discloses a spring drive unit for window covers including a transmission 50 operatively coupled to a coil spring 47 wrapped around middle shaft 31 (i.e., "spring motor") and to an adjacent shaft 35 (i.e., "rotating output") that rotates the lift cord pulley 18. The transmission 50 causes the coil spring 47 to rotate at one rate and the pulley 18 to rotate at another rate. See, e.g., Toti at 4:29-33 and 2:26-37, and FIGS. 5 and 6. One of ordinary skill in the art could have easily configured the transmission 50 of Toti operatively coupled to the spring motor and shaft of Tachikawa. Accordingly, the transmission of Toti corresponds to the transmission of claim 6.

109. Therefore, the combination of Tachikawa, Strahm, and Toti teaches each feature recited in claim 6 and renders the subject matter of claim 6 as a whole obvious and unpatentable.

C. Tachikawa In View Of Skidmore And Further In View Of Schuetz

1. Reasons To Combine Tachikawa, Skidmore, and Schuetz

110. Claims 5 and 7 of the 884 patent are unpatentable as being obvious over Tachikawa in view of Skidmore and further in view of Schuetz. The disclosures of Tachikawa, Skidmore, and Schuetz pertain to the subject matter of the 884 patent. Like the 884 patent, Tachikawa relates to lifting mechanisms that extend and retract a window covering via a rotating shaft driven by a spring drive (as shown in FIGS. 3 and 4 of Tachikawa). Skidmore discloses a gear box 12 with geared transmissions as a lifting mechanism for raising or lowering venetian blinds against gravity (Skidmore, pg. 2, lines 112-120, and FIGS. 1, 3, and 4) in a general configuration similar to Tachikawa's design and further teaches in its FIG. 2 a friction brake inside the gear box 12 for holding the raised or partially raised blinds in position. Schuetz relates to lifting mechanisms and brakes employed in a hoisting apparatus to raise and lower a load while preventing undesired reverse-rotation of a rotating shaft.

111. In raising and lowering window blinds, one common technical issue is, as stated by Tachikawa, using a lifting mechanism to balance and to hold window blinds against varying weights of the window blinds against the gravity at different positions between the fully roll-up position and fully roll-down position. Specifically, Tachikawa discloses that, under its lifting mechanism design, the user need only apply small force for extending and retracting the window covering, and cause the window covering to remain stationary at the user's desired extended or retracted position. See, e.g., Tachikawa 1:22-2:10 and 3:15-4:6 The graph of FIG. 9 in Tachikawa shows the load versus elongation relationship of Tachikawa's spring motor, which demonstrates that the spring force can be tailored to the specific load requirements of the blind.

112. Like Tachikawa, Skidmore teaches a lifting mechanism for raising or lowering window blinds. In addition, Skidmore teaches a friction brake in FIG. 2 to hold the blind in a raised or partially raised position against the gravity in a venetian blind covering. Due to the operation of the gravity on the blinds, "*there is a tendency for the weight of the bottom rail 34 and slats 41 to rotate the winding drum 16 and other components of the gear box due to tension 125 in the tapes 27 and 28. In order to overcome this tendency a friction brake is provided as shown in Figure 2*" (Skidmore, pg. 2, lines 121-128).

113. This balancing against the weight to be lifted in window blinds in Tachikawa and Skidmore is a common technical issue in a range of lifting mechanisms including lifting window blinds and the hoisting apparatus disclosed in Schuetz. In this regard, Schuetz discloses a one-way friction brake for a rotational shaft that resists the reverse rotation of the shaft when under a load. An embodiment of the one-way friction brake of Schuetz is described to be connected to a rotating shaft of the Schuetz hoisting apparatus, where the one-way friction brake is not engaged when a user is hoisting a load (e.g., analogously, the one-way friction brake is not engaged while raising (i.e. retracting a window blind), and the one-way friction brake is engaged against rotation in the opposite direction to prevent retrograde movement of the shaft (e.g., analogously, the one-way friction brake is engaged against lowering (i.e. extending a window blind). See, e.g., Schuetz at pg. 2, lines 71-85 and 20-50.

114. Tachikawa and Skidmore recognize the technical challenge of holding a raised or partially raised blinds in a desired position. Skidmore provides a brake to provide a solution. In a similar approach to Skidmore's brake design for window blinds, Schuetz addresses this technical challenge for raising a load to a precise position by preventing retrograde motion of a rotatable shaft using a one-way brake that is analogous to similar technical challenges faced in the window covering industry for controlling the positioning of a blind and holding the blind

stationary against gravity in that intended position. Schuetz' one-way brake is a better version of the simple friction brake in FIG. 2 of Skidmore and provides better braking control. Specifically, Schuetz disclosed the one-way brake that is recited in claims 5-7 of the 884 patent 70 years before the 884 patent. One of ordinary skill in the art at the time of the 884 patent would have been familiar with common mechanical components and mechanisms, such as the one-way brake of Schuetz, that have been initially disclosed in other mechanical systems and available to the public, like Schuetz. Such common mechanical components and mechanisms have been employed in the development of window coverings.Moreover, the particular components and mechanisms employed in the one-way friction brake of Schuetz could have been easily combined with the known mechanical components and mechanisms of Tachikawa (i.e., the shaft 2 of Tachikawa) by a person of ordinary skill in the art at the time of the 884 patent.

115. Because of the close linkages amongst Tachikawa, Skidmore, and Schuetz in controlling rotation when raising and lowering a load, such as a window covering, with respect to the subject matter in the 884 patent, there is a motivation or suggestion in the general teachings by Tachikawa, Skidmore, and Schuetz to enable a person having ordinary skill in the art to combine the teachings of these references. Such combinations render claims 5 and 7 of the 884 patent unpatentable.

2. Claim 5 Is Rendered Obvious By Tachikawa In View Of Skidmore And In Further View Of Schuetz

116. **Preamble**: A system for covering an architectural opening, comprising:

See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Preamble** of claim 5.

117. **Element [5A]**: a covering movable between an extended position for covering the opening and a retracted position for uncovering the opening;

See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [5A]** of claim 5.

118. **Element [5B]**: a spring motor including a coil spring and a power spool, wherein said coil spring wraps onto and off of said power spool; See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [5B]** of claim 5.

119. **Element [5C]**: a rotating output operatively connected to the power spool of the spring motor; See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [5C]** of claim 5.

120. **Element [5D]**: a lift cord operatively connected to the rotating output and to the covering; See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [5D]** of claim 5.

121. **Element [5E]**: <u>said rotating output being rotatable in clockwise and counterclockwise directions to move the covering between its extended and retracted positions;</u> See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [5E]** of claim 5.

122. **Element [5F]**: <u>a one-way friction brake operatively connected to said rotating output, said one-way friction brake providing a braking force that stops the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions.</u> The Tachikawa system for covering an architectural opening is designed to raise and lower the window covering to a desired position and balance the blind to remain stationary. Tachikawa provides a blinds roll-up device including a spring motor. Another way to achieve this is to also include a brake to further assist in balancing the window covering at the desired position.

123. Skidmore discloses a friction brake shown in FIG. 2 that assists in holding the window covering (e.g., venetian blind) in a desired position against gravity (Skidmore, pg. 2, lines 121-128) and allowing the blind to be raised and lowered with “very low operating effort” (Skidmore, pg. 4, lines 14-21).

124. In a similar approach to Skidmore’s brake design for window blinds, Schuetz addresses the same technical issue for raising a load to a precise position by preventing retrograde motion of a rotatable shaft using a one-way brake that is

analogous to similar technical challenges faced in the window covering industry for controlling the positioning of a blind and holding the blind stationary against gravity in that intended position.

125. Schuetz discloses such a one-way friction brake. As described in claim 1 of Schuetz, Schuetz teaches "*a friction brake for a rotary shaft, comprising a housing adapted to receive a portion of said shaft, a friction plate unit in said housing surrounding said shaft and comprising a plurality of friction plates secured together with a clutch casing interposed therebetween, a clutch member fixed on said shaft and rotatable therewith, within said clutch casing, and a pressure member adapted to normally prevent the rotation of said [friction plate] unit, said clutch being adapted to rotate freely in one direction and engage said friction plate unit when force is applied in the reverse direction, whereby retrograde rotation of said shaft is prevented*." (Schuetz, pg. 2, lines 71-85). Schuetz's one-way friction brake mechanism is capable of providing a braking force that stops the rotation of the rotary shaft (e.g., shaft 2 in FIG. 1 of Schuetz) in one direction and permits the shaft to rotate freely in the other direction. See, e.g., as Schuetz at 71-85 on pg. 2, 20, 50, and FIG. 1 (reproduced here) and FIG. 3. Accordingly, Schuetz's one-way friction brake corresponds to and discloses the one-way friction brake of claim 5.

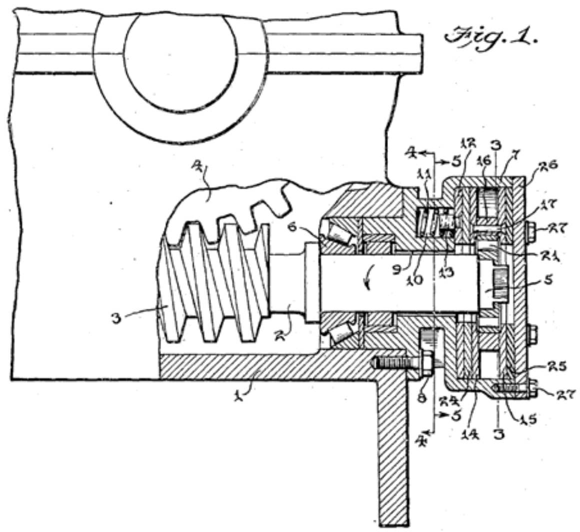


FIG. 1 of Schuetz

126. One of ordinary skill in the art at the time of the 884 patent could easily have combined the one-way friction brake of Schuetz with the venetian blind of Tachikawa. Notably, the shaft 2 of Schuetz is substantially the same, both structurally and operationally, as the operating shaft 2 of Tachikawa, allowing for the one-way friction brake of Schuetz to easily be operatively coupled to the shaft 2 of Tachikawa.

127. Therefore, the combination of Tachikawa, Skidmore, and Schuetz teaches each feature recited in claim 5 and renders the subject matter of claim 5 as a whole obvious and unpatentable.

3. Claim 7 Is Rendered Obvious By Tachikawa In View Of Skidmore And In Further View Of Schuetz

128. **Preamble**: <u>A system for covering an architectural opening, comprising:</u>

See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Preamble** of claim 7.

129. **Element [7A]**: a covering movable between an extended position for covering the opening and a retracted position for uncovering the opening; See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [7A]** of claim 7.

130. **Element [7B]**: a spring motor; See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [7B]** of claim 7.

131. **Element [7C]**: a rotating output operatively connected to the spring motor; See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [7C]** of claim 7.

132. **Element [7D]**: a lift cord operatively connected to the rotating output and to the covering; See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [7D]** of claim 7.

133. **Element [7E]**: said rotating output being rotatable in clockwise and counterclockwise directions to move the covering between its extended and retracted positions; See discussion of relevant teachings in Tachikawa as discussed above in Ground 1 for **Element [7E]** of claim 7.

134. **Element [7F]**: a one-way friction brake operatively connected to said rotating output, said one-way friction brake providing braking force opposing the

rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions; Similarly as discussed in **Element [5F]** of claim 5 in Ground 3 above with respect to the combination of Tachikawa, Skidmore, and Schuetz, Schuetz discloses such a one-way friction brake. See, e.g., claim 1 of Schuetz at pg. 2, lines 71-85. Schuetz discloses that the one-way friction brake is operatively connected to a rotary shaft (i.e., "rotating output"). The one-way friction brake of Schuetz provides a braking force that opposes the rotation of the rotary shaft (e.g. ,shaft 2 in FIG. 1 of Schuetz) in one direction and permits the rotary shaft 2 to rotate freely in the other direction. See, e.g., Schuetz at 71-85 on pg. 2, 20-50 of pg. 2, and FIGS. 1 and 3.

135. One of ordinary skill in the art could have easily configured the one-way friction brake of Schuetz operatively coupled to the shaft 2 of Tachikawa. Accordingly, the one-way friction brake of Schuetz corresponds to the one-way friction brake of claim 7.

136. **Element [7G]**: wherein said one-way brake applies a braking force opposing rotation of the rotating output for movement of the covering to the extended position while permitting free rotation for movement of the covering to the retracted position. Schuetz discloses this feature of claim 7. Specifically, as discussed at pg. 2, lines 23-45 of Schuetz, the one-way friction brake mechanism of Schuetz applies a braking force that opposes the rotation of the shaft 2 in a

direction against the lowering of a load, and permits the shaft 2 to rotate freely in the other direction to not impede the raising of the load.

137. Therefore, the combination of Tachikawa, Skidmore, and Schuetz teaches each feature recited in claim 7 and renders the subject matter of claim 7 as a whole obvious and unpatentable.

D. Cohn In View Of Strahm And Further In View Of Todd

1. Reasons To Combine Cohn, Strahm, and Todd

138. Claim 5 of the 884 patent is unpatentable as being obvious over Cohn in view of Strahm and in further view of Todd. The disclosures of Cohn, Strahm and Todd are in the same technical field of window coverings as the 884 patent. Like the 884 patent, Cohn, Strahm and Todd relate to mechanisms that extend and retract a window covering using spring motors.

139. Cohn teaches a system for covering an architectural opening with a cordless Venetian blind. One objective of Cohn's invention is to provide a cordless blind that is "universally adaptable to all sizes of blind, and which can be installed in windows of various widths by the simple expedient of varying the length of the drive shaft" (see 1:37-40 on pg. 1 of Cohn), which is similar to the primary objective of the 884 patent, as described at 3:10-19 in the 884 patent.

140. Furthermore, Cohn discloses the same key features and mechanisms of the systems described in the 884 patent. Cohn discloses transport mechanisms

and systems for a covering an architectural opening in the form of a cordless Venetian blind having a covering (e.g., slats), lift cords that extend and retract the slats and wraps onto/off of lift spools (e.g., drums or reels), a rotating shaft that causes the lift spools to take up/down the lift cords, and a spring motor that drives rotation the rotating shaft. These components and mechanisms are structurally comparable and functionally and operationally the same as the components and mechanisms claimed in claims 5-7 of the 884 patent, as well as interchangeable and combinable with other components and mechanisms in other window covering systems, such as the one-way friction brake of Strahm and the spring motor unit of Todd or Tachikawa, as well as other prior art cited in this Declaration.

141. For example, Cohn's cordless Venetian blind includes a horizontal, rotatable shaft that operatively couples to other independent mechanical components to raise and lower the covering via a lifting mechanism including one or more spring motors. Moreover, Cohn's cordless blind includes a brake mechanism operatively coupled to the horizontal shaft. Cohn teaches that "*while the form of my invention illustrated and described herein is now deemed to be the preferred form thereof, I do not mean to limit myself to that particular form, but intend to include all equivalents thereof as defined by the appended claims*." See, e.g., Cohn at pg. 4, left column, lines 12-17. Furthermore, claim 1 in Cohn discloses a "*releasable means associated with said shaft adapted to lock said shaft*

against rotation." Therefore, Cohn suggests any suitable brake that can stop and maintain the window casing at a desired height. This suitable brake can be a one-way friction brake as in Strahm. Strahm teaches a one-way brake mechanism for a window covering that applies frictional braking against lowering of the covering and releases when the covering is being raised.

142. The technology disclosed in Cohn and Strahm addresses the same technical challenges of controlling the rotation of a rotating shaft that raises and lowers a window covering (e.g., a blind or shade) so that the window covering can be reliably raised and lowered by a user to remain in the intended position. Cohn and Strahm employ comparable and commonly known mechanical components and mechanisms that could have been easily combined or interchanged by a person of ordinary skill in the art at the time of the 884 patent. Comparisons between the claimed subject matter in claims 5-7 of the 884 patent and the disclosures in Cohn and Strahm are provided in detail in subsequent sections of this Declaration.

143. Addressing the same or similar technical issues in Cohn, Strahm, and the 884 patent for a system for covering an architectural opening, Todd teaches a modular drive mechanism for a window covering (e.g., cordless blind or shade) that includes a spring assembly that drives a horizontal shaft (like Cohn's shaft) to raise and lower a shade via a lift spool, and a brake/clutch mechanism to regulate braking force and speed of travel of the shaft to brake against lowering and raising

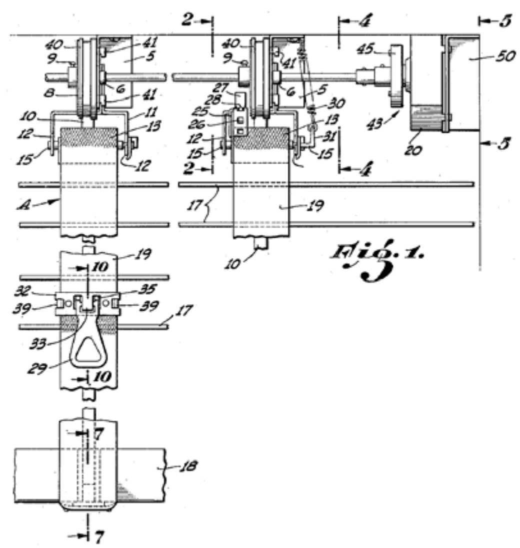
of the shade. Todd's spring drive assembly includes a spring that wraps and unwraps on/off a spool that drives rotation of Todd's shaft.

144. Like, Cohn and Strahm, Todd also discloses window covering systems configured using mechanism designed to control the rotation of a rotating shaft for raising and lowering the window blind or shade so that the window covering can be reliably raised and lowered by a user to remain in the intended position (Todd, 3:10-21). Cohn, Strahm, and Todd all employ comparable and commonly known mechanical components and mechanisms that could have been easily combined or interchanged by a person of ordinary skill in the art at the time of the 884 patent.

145. Because of the close linkages amongst Cohn ,Strahm, and Todd with respect to the subject matter in the 884 patent, there is a motivation or suggestion in the teachings by Cohn, Strahm, and Todd to enable a person having ordinary skill in the art to combine the teachings of these references. Such combinations render claim 5, 6, and 7 of the 884 patent unpatentable.

2. Claim 5 Is Rendered Obvious By Cohn In View Of Strahm And In Further View Of Todd

146. **Preamble**: A system for covering an architectural opening, comprising: Cohn discloses a cordless Venetian blind including mechanisms adaptable to all sizes of blinds for installing in various window configurations, shown by the Venetian blind in FIG. 1 of Cohn (reproduced here). Therefore, Cohn teaches "a system for covering an architectural opening" in claim 5.



147. Similar to Cohn, Todd discloses a cordless window shade, shown by cordless window shade assembly 10 in FIG. 1 of Todd, which teaches "a system for covering an architectural opening" in claim 5.

148. **Element [5A]**: a covering movable between an extended position for covering the opening and a retracted position for uncovering the opening; Cohn discloses a plurality of slats 17 of the cordless Venetian blind that can be retracted and extended about a window opening. Accordingly, the slats 17 of Cohn corresponds to and discloses the covering of claim 5.

149. Like Cohn, Todd also discloses shade 14 of the cordless window shade assembly 10 that can be retracted and extended about a window opening, which corresponds to the covering of claim 5.

150. **Element [5B]**: <u>a spring motor including a coil spring and a power spool, wherein said coil spring wraps onto and off of said power spool;</u> Cohn discloses a spring motor 20 "*disposed in operative relation with shaft 7*", at pg. 3, left column (LC) lines 16-75, and shown in FIGS. 1, 5 and 6 of Cohn. And, Todd discloses spring assembly 26 of drive mechanism 20 of the cordless window shade assembly 10 that includes spring 58 (i.e., "coil spring") and take-up spool 56 (i.e., "power spool"), in which spring 58 wraps onto and off of take-up spool 56. See, e.g., 5:40-46 and FIGS. 3 and 6 of Todd (reproduced here).

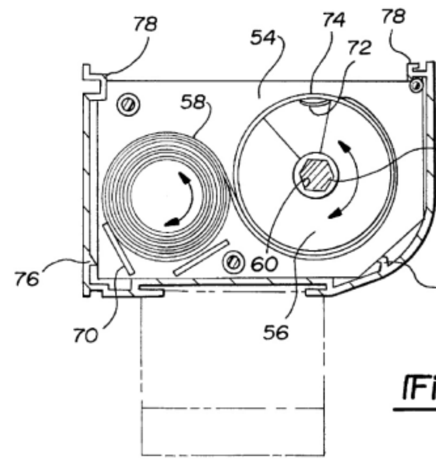


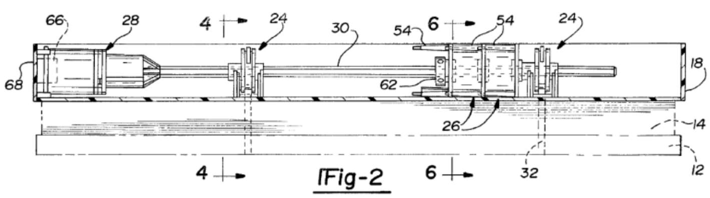
Fig-6

151. The spring motor 20 of Cohn is a clock spring motor that is operationally equivalent to the spring drive mechanism 20 of Todd. It would be obvious to one of ordinary skill in the art to employ the spring drive mechanism of Todd in the cordless Venetian blind of Cohn if one desired to such a spring motor design having a power spool, as that of Todd, and the system in claim 5 of the 884 patent. Accordingly, the spring assembly 26 of Todd corresponds to and discloses the spring motor of claim 5.

152. **Element [5C]**: <u>a rotating output operatively connected to the power spool of the spring motor;</u> Cohn discloses a shaft 7 operatively connected to the

spring motor 20. See, e.g., pg. 1, right column (RC) lines 33-34 and pg. 2 LC lines 8-11, and FIGS. 1, 5 and 6 of Cohn. Accordingly, the shaft 7 of Cohn corresponds to and discloses the rotating output of claim 5.

153. Like Cohn, Todd also discloses a rotating output by a shaft 30, operatively connected to spring assembly 26 via hole 60 of take-up spool 56. See, e.g., 5:40-46 and FIGS. 2 and 3 of Todd (reproduced here). Also accordingly, the shaft 30 of Todd corresponds to and discloses the rotating output of claim 5.



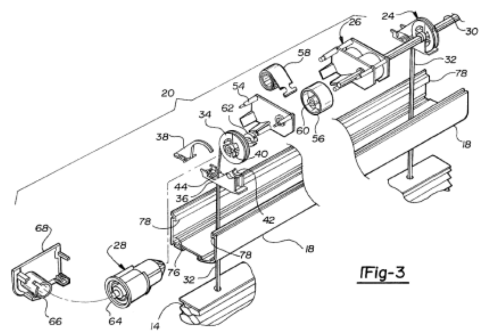


FIG. 2 of Todd FIG. 3 of Todd

154. **Element [5D]**: <u>a lift cord operatively connected to the rotating output and to the covering;</u> Cohn discloses lifting tape 10 operatively connected to the shaft 7 (via the drums/reels 8) and to the slats 17(via fabric rungs 16 of ladder tape 19). See, e.g., pg. 1, RC lines 33-43 and pg. 2, LC lines 3-7, and FIGS. 1 and 4 of Cohn. Accordingly, the lifting tape 10 of Cohn corresponds to and discloses the lift cord of claim 5.

155. Similar to Cohn, Todd also discloses a lift tape 32 of spool assembly 24 operatively coupled to the shaft 30. See, e.g., 4:33-37 and FIG. 4 of Todd. Also accordingly, the shaft 30 of Todd corresponds to and discloses the rotating output of claim 5.

156. **Element [5E]**: <u>said rotating output being rotatable in clockwise and counterclockwise directions to move the covering between its extended and retracted positions;</u> Cohn discloses, "*In order to lower the blind, a person merely grasps the bottom rail 18 and pulls it downwardly. This downward movement of the bottom rail will effect an unreeling of the lift tapes 10 from the drums 8 and cause rotation of said drums and the shaft 7, upon which they are mounted.*" (Cohn, at pg. 3, LC lines 43-48). Cohn discloses, "*energy stored in the springs 54 and 59 causes rotation of the shaft 7 with the consequent raising of the blind by means of the lift tapes 10.*" (Cohn, at pg. 3, RC lines 25-28). Therefore, Cohn discloses Element [E] in claim 5.

157. **Element [5F]**: <u>a one-way friction brake operatively connected to said rotating output, said one-way friction brake providing a braking force that stops the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions.</u> Cohn teaches at pg. 3, RC lines 1-3, "*The blind may be stopped and maintained at any desired height relative to the window casing by suitable means[.]*" Strahm discloses a suitable means to

provide braking force that stops the rotation of the rotating output in one direction while permitting free rotation in the other direction. Specifically, Strahm discloses a one-way friction brake mechanism for window blinds. The disclosed one-way friction brake has conical washers 33 and 36 that contact wall 34 when sleeve 32 is rotated, thus forming a friction brake. The "hand" (direction of winding) of spring 21 which contacts sleeve 19 allows rotation in one direction but not the other. This combination creates a one-way friction brake. Strahm discloses that the one-way friction brake mechanism is operatively connected to rotating shaft 4 (i.e., "rotating output"). Strahm's one-way friction brake mechanism is capable of providing a braking force that stops the rotation of the rotating shaft 4 in one direction and permits the rotating shaft 4 to rotate freely in the other direction. See, e.g., as Strahm at 3:11-35, 4:31-33, 1:28-34, and FIG. 6 (previously reproduced). Accordingly, Strahm's one-way friction brake corresponds to and discloses the one-way friction brake of claim 5.

158. As suggested by Cohn, one of ordinary skill in the art at the time of the 884 patent could have easily combined the one-way friction brake of Strahm with the cordless Venetian blind of Cohn. Notably, the operating shaft 4 of Strahm is substantially the same, both structurally and operationally, as the shaft 7 of Cohn, as well as the shaft 30 of Todd.

159. Therefore, the combination of Cohn, Strahm, and Todd teaches each feature recited in claim 5 and renders the subject matter of claim 5 as a whole obvious and unpatentable.

E. Cohn In View Of Strahm And Further In View Of Todd And Toti

1. Reasons To Combine Cohn, Strahm, Todd, And Toti

160. Claim 6 is a dependent claim of claim 5. Claim 6 is unpatentable as being obvious over Cohn in view of Strahm, in further view of Todd, and in further view of Toti. Toti discloses such a transmission, and thus the combination of Cohn, Strahm, Todd, and Toti renders claim 6 unpatentable.

161. The disclosures of Cohn, Strahm, Todd and Toti are in the same technical field of window blinds and shades and address the same technical challenges of the 884 patent. As discussed before in Ground 2 above, Toti suggests that its components such as the gear and band transmissions can be used "alone or in essentially any combination" to accommodate a given blind or cover, and that one familiar with the art will appreciate that the components and arrangements of Toti are applicable in general to window covers that use spring drive mechanism. See 8:55-9:1 of Toti. Moreover, Toti provides motivation to combine the technology with others. See 8:66-9:1 of Toti. Furthermore, like Cohn, Strahm, Todd, and the 884 patent, Toti employs comparable and commonly known mechanical components and mechanisms that could have been easily combined or

interchanged by a person of ordinary skill in the art at the time of the 884 patent, as stated by Toti.

2. Claim 6 Is Rendered Obvious By Cohn In View Of Strahm And In Further View Of Todd And Toti

162. **Preamble and Element [6A]**: <u>A system for covering an architectural opening as recited in claim 5, and further comprising a transmission operatively connected to the spring motor and to the rotating output.</u> See discussion of relevant teachings in Toti as discussed above in Ground 2 for **Element [6A]** of claim 6.

163. Therefore, the combination of Cohn, Strahm, Todd, and Toti teaches each feature recited in claim 6 and renders the subject matter of claim 6 as a whole obvious and unpatentable.

F. Cohn In View Of Strahm

1. Reasons To Combine Cohn And Strahm

164. Claim 7 of the 884 patent is unpatentable as being obvious over Cohn in view of Strahm. The disclosures of Cohn and Strahm are in the same technical field of window blinds and shades of the 884 patent. Like the 884 patent, Cohn and Strahm relate to mechanisms that extend and retract a window covering.

2. Claim 7 Is Rendered Obvious By Cohn In View Of Strahm

165. **Preamble**: <u>A system for covering an architectural opening, comprising:</u>

See discussion of relevant teachings in Cohn as discussed above in Ground 4 for **Preamble** of claim 5.

166. **Element [7A]**: <u>a covering movable between an extended position for covering the opening and a retracted position for uncovering the opening;</u> See discussion of relevant teachings in Cohn as discussed above in Ground 4 for **Element [5A]** of claim 5.

167. **Element [7B]**: <u>a spring motor;</u> Cohn discloses a spring motor 20 "*disposed in operative relation with shaft 7*", at pg. 2, LC lines 8-11, and pg. 3, LC lines 16-75, and shown in FIGS. 1, 5 and 6 of Cohn (reproduced here). Accordingly, the spring motor 20 of Cohn corresponds to and discloses the spring motor of claim 7.

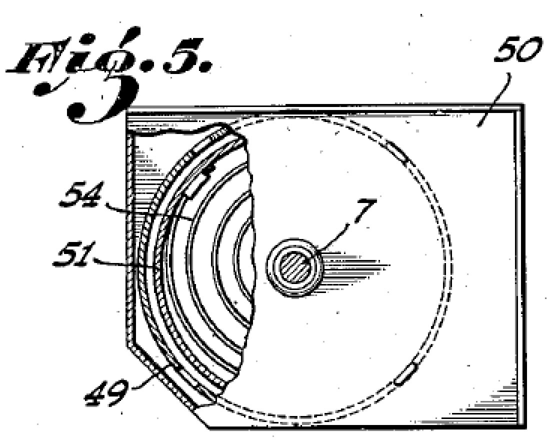


FIG. 5 of Cohn

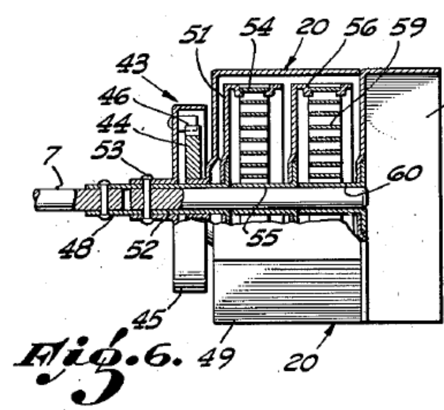


FIG. 6 of Cohn

168. **Element [7C]**: <u>a rotating output operatively connected to the spring motor;</u> Cohn discloses a shaft 7 operatively connected to the spring motor 20, at

pg. 2, LC lines 8-11, and shown in FIGS. 1, 5 and 6 of Cohn. Accordingly, the shaft 7 of Cohn corresponds to and discloses the rotating output of claim 7.

169. **Element [7D]**: <u>a lift cord operatively connected to the rotating output and to the covering;</u> See discussion of relevant teachings in Cohn as discussed above in Ground 4 for **Element [5D]** of claim 5.

170. **Element [7E]**: <u>said rotating output being rotatable in clockwise and counterclockwise directions to move the covering between its extended and retracted positions;</u> See discussion of relevant teachings in Cohn as discussed above in Ground 4 for **Element [5E]** of claim 5.

171. **Element [7F]**: <u>a one-way friction brake operatively connected to said rotating output, said one-way friction brake providing braking force opposing the rotation of the rotating output in one of the directions while permitting the rotating output to rotate freely in the other of said directions;</u> Cohn teaches at pg. 3, right column lines 1-3, "*The blind may be stopped and maintained at any desired height relative to the window casing by suitable means[.]*" Strahm discloses a suitable means to provide braking force that opposes the rotation of the rotating output in one direction while permitting free rotation in the other direction. See discussion of relevant teachings of the one-way friction brake in Strahm as discussed above in Ground 1 for **Element [7E]** of claim 7.

172. Accordingly, the one-way friction brake of Strahm corresponds to the one-way friction brake of claim 7.

173. **Element [7G]**: <u>wherein said one-way brake applies a braking force opposing rotation of the rotating output for movement of the covering to the extended position while permitting free rotation for movement of the covering to the retracted position.</u> See discussion of relevant teachings of the one-way friction brake in Strahm as discussed above in Ground 1 for **Element [7F]** of claim 7.

174. Therefore, the combination of Cohn and Strahm teaches each feature recited in claim 7 and renders the subject matter of claim 7 as a whole obvious and unpatentable.

I hereby declare that all statements made in this declaration are based on my own knowledge and are true based on information and belief, and that all statements were made with the knowledge that willful false statements and the like are punishable by fine or imprisonment, or both under 18 U.S.C. 1001 and such willful false statements may jeopardize the validity of the application or any patent issuing thereon.

The contents of this declaration are true under penalty of perjury of the laws of the United States.

Executed July 16, 2014 in Boulder, Colorado.

/ Lawrence E. Carlson /

LAWRENCE E. CARLSON

ATTACHMENT A: CURRICULUM VITAE OF LAWRENCE E. CARLSON

ATTACHMENT B: MECHANICAL ENGINEERING DESIGN

ATTACHMENT C: STANDARD HANDBOOK OF MACHINE DESIGN

CURRICULUM VITAE

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M.S., Mechanical Engineering, University of California at Berkeley, Jun. 1968
D.Eng., Mechanical Engineering, University of California at Berkeley, Sep. 1971
Engineer-in-Training (EIT), Wisconsin

Professional Experience

IDEO Fellow, IDEO Product Design and Development, Palo Alto, CA, Spring 2001
Professor, Dept. of Mechanical Engineering, University of Colorado, Boulder, 1994-present
Founding Co-Director, Integrated Teaching and Learning Program and Laboratory, College of Engineering and Applied Science, University of Colorado, Boulder, 1992-2007
Visiting Associate Professor, Dept. of Mechanical Engineering, Stanford University, 1990-91
Associate Professor, Dept. of Mechanical Engineering, University of Colorado, Boulder, 1978-1994
Engineering Consultant, Ponderosa Associates, Lafayette, CO, 1983-present
Principal Research Fellow, Biomechanical Research & Development Unit, Dept. of Health and Social Security, London, 1979-80
Assistant Professor, Dept. of Engineering Design and Economic Evaluation, University of Colorado, Boulder, 1974-78
Assistant Professor of Mechanical Design, Materials Engineering Dept., University of Illinois at Chicago Circle, 1971-74
Research Assistant, Biomechanics Laboratory, University of California at Berkeley, 1970-71

Society Membership

Founding Member, Rehabilitation Engineering Society of North America
Member, American Society of Mechanical Engineers
Member, Phi Eta Sigma
Member, Pi Tau Sigma
Member, Sigma Xi
Member, International Society of Prosthetics and Orthotics
Member, American Society for Engineering Education

Grants

National Institutes of Health, Bioengineering Traineeship, 1967-70
National Science Foundation International Travel Grants to Yugoslavia, 1972 & 1975
National Science Foundation Research Initiation Grant, 1973-74
Veterans Administration grant, "Mobility system for adult paraplegics", 1976-77
Veterans Administration grant, "Position control of above-elbow prostheses", 1977-79
National Institutes of Health, Research Career Development Award, 1976-81
IBM Grant, "Mechanical design for robotic assembly", 1984-87
Veterans Administration grant, "Implementation of extended physiological proprioception for prosthesis control", 1985-88
National Institute on Disability and Rehabilitation Research, Field-Initiated Research Grant, "Improved actuation of body-powered prostheses", 1987-90
National Center for Medical Rehabilitation Research (within NIH), Grant, "Improving prosthetic prehension", 1992-94
Colorado Commission on Higher Education Program of Excellence, "K-16 integrated engineering outreach", co-PI, 1998-2003
National Science Foundation, GK-12 Graduate Teaching Fellows Grant, "Creating an integrated engineering and technology education continuum", co-PI, 1999-2002

Honors and Awards

Bronze Award, Lincoln Arc Welding Design Competition, 1981
American Men and Women in Science
Vince Kontny Award - Outstanding Undergraduate Advisor, University of Colorado, College of Engineering and Applied Science, 1990
Outstanding Undergraduate Advisor, Council on Academic Advising, University of Colorado, 1990
National Institute on Disability and Rehabilitation Research, Mary E. Switzer Distinguished Research Fellowship, 1990-91
Teaching Award, Mechanical Engineering Department, 1995
Sullivan-Carlson Innovation in Education Award – Annual award to honor an engineering faculty member, nominated by engineering students, endowed by the student-run Engineering Excellence Fund in the names of Jacquelyn F. Sullivan and Lawrence E. Carlson at the dedication of the ITL Laboratory, 1997
IDEO Fellow, IDEO Product Design and Development, Palo Alto, CA, 2001
Charles Hutchinson Outstanding Teaching Award, College of Engineering and Applied Science, 2001
John and Mercedes Peebles Innovation in Education Award, College of Engineering and Applied Science, 2004
Excellence in Teaching Award, Boulder Faculty Assembly, University of Colorado, 2008
Bernard M. Gordon Prize for Innovation in Engineering and Technology Education, National Academy of Engineering (co-recipient with Jacquelyn Sullivan), 2008

Patents

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Mechanical Engineering Design

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ATTACHMENT B

10 Mechanical Springs

Mechanical springs are used in machines to exert force, to provide flexibility, and to store or absorb energy. In general, springs may be classified as either wire springs, flat springs, or special-shaped springs, and there are variations within these divisions. Wire springs include helical springs of round or square wire and are made to resist tensile, compressive, or torsional loads. Under flat springs are included the cantilever and elliptical types, the wound motor- or clock-type power springs, and the flat spring washers, usually called Belleville springs.

10-1 STRESSES IN HELICAL SPRINGS

Figure 10-1*a* shows a round-wire helical compression spring loaded by the axial force F. We designate D as the *mean spring diameter* and d as the *wire diameter*. Now imagine that the spring is cut at some point (Fig. 10-1*b*), a portion of it removed, and the effect of the removed portion replaced by the internal forces. Then, as shown in the figure, the cut portion would exert a direct shear force F and a torsion T on the remaining part of the spring.

To visualize the torsion, picture a coiled garden hose. Now pull one end of the hose in a straight line perpendicular to the plane of the coil. As each turn of hose is pulled off the coil, the hose twists or turns about its own axis. The flexing of a helical spring creates a torsion in the wire in a similar manner.

The maximum stress in the wire may be computed by superposition of Eqs. (2-15) and (2-41). The result is

$$\tau_{\max} = \pm\frac{Tr}{J} + \frac{F}{A} \tag{a}$$

where the term Tr/J is the torsion formula and F/A is the direct (not flexural) shear stress. Replacing the terms by $T = FD/2$, $r = d/2$, $J = \pi d^4/32$, and $A = \pi d^2/4$ gives

$$\tau = \frac{8FD}{\pi d^3} + \frac{4F}{\pi d^2} \tag{10-1}$$

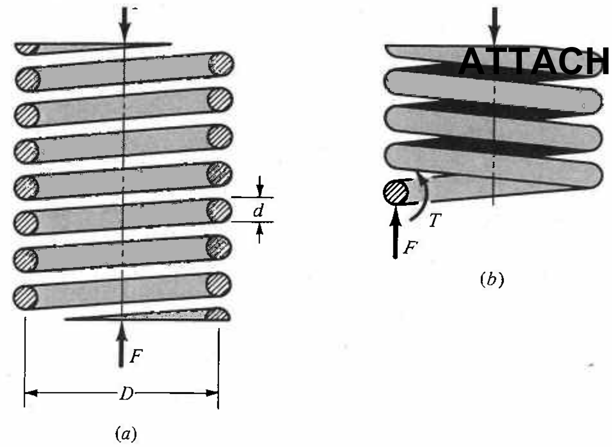

FIGURE 10-1
(*a*) Axially loaded helical spring; (*b*) free-body diagram showing that the wire is subjected to a direct shear and a torsional shear.

In this equation the subscript indicating maximum shear stress has been omitted as unnecessary. The positive signs of Eq. (*a*) have been retained, and hence Eq. (10-1) gives the shear stress at the *inside* fiber of the spring.

Now we define the *spring index*

$$C = \frac{D}{d} \tag{10-2}$$

as a measure of coil curvature. With this relation, Eq. (10-1) can be rearranged to give

$$S_{sy} = \tau = K_s \frac{8FD}{\pi d^3} \qquad F = \frac{S_{sy}\,\pi d^3}{8\,K_s} \tag{10-3}$$

where K_s is a *shear-stress correction factor* and is defined by the equation

$$K_s = \frac{2C + 1}{2C} \tag{10-4}$$

For most springs, C will range from about 6 to 12. Equation (10-3) is quite general and applies for both static and dynamic loads.

The use of square or rectangular wire is not recommended for springs unless space limitations make it necessary. Springs of special wire shapes are not made in large quantities, as are those of round wire; they have not had the benefit of refining development and hence may not be as strong as springs made from round wire. When space is severely limited, the use of nested round-wire springs should always be considered. They may have an economical advantage over the special-section springs, as well as a strength advantage.

10-2 THE CURVATURE EFFECT

An effect very similar to stress concentration occurs at the inside surface of a helical spring. The curvature of the wire increases the stress on the inside of the spring but fe. The corre-

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the springs in parallel. On the other hand, stacking in series provides a larger deflection for the same load, but in this case there is danger of instability.

10-14 MISCELLANEOUS SPRINGS

The extension spring shown in Fig. 10-12 is made of slightly curved strip steel, not flat, so that the force required to uncoil it remains constant; thus it is called a *constant-force spring*. This is equivalent to a zero spring rate. Such springs can also be manufactured having either a positive or a negative spring rate.

A *volute spring* is a wide, thin strip, or "flat," of steel wound on the flat so that the coils fit inside one another. Since the coils do not stack, the solid height of the spring is the width of the strip. A variable-spring scale, in a compression volute spring, is obtained by permitting the coils to contact the support. Thus, as the deflection increases, the number of active coils decreases. The volute spring, shown in Fig. 10-13*a*, has another important advantage which cannot be obtained with round-wire springs: if the coils are wound so as to contact or slide on one another during action, the sliding friction will serve to damp out vibrations or other unwanted transient disturbances.

A *conical spring*, as the name implies, is a coil spring wound in the shape of a cone. Most conical springs are compression springs and are wound with round wire. But a

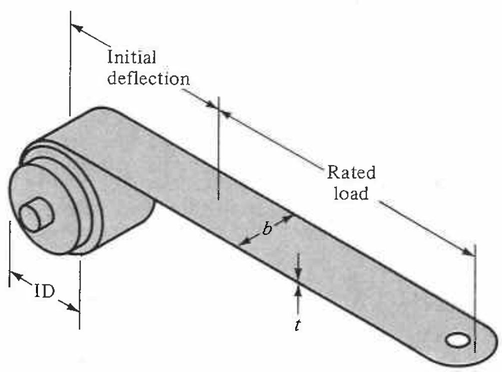

FIGURE 10-12
Constant-force spring. *(Courtesy of Vulcan Spring & Mfg. Co., Huntingdon Valley, Pa.)*

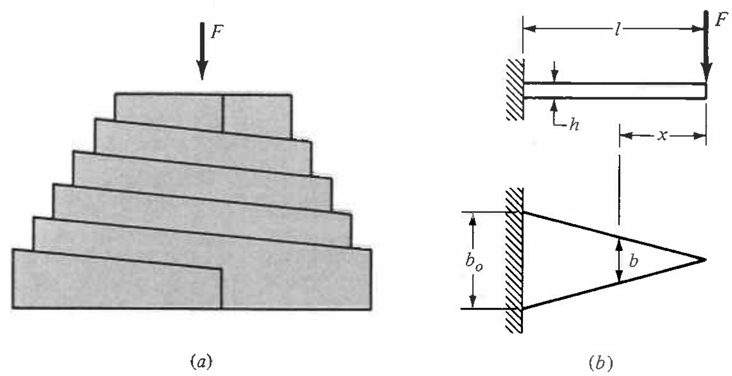

FIGURE 10-13
(*a*) A volute spring; (*b*) a flat triangular spring.

volute spring is a conical spring too. Probably the principal advantage of this type of spring is that it can be wound so that the solid height is only a single wire diameter.

Flat stock is used for a great variety of springs, such as clock springs, power springs, torsion springs, cantilever springs, and hair springs; frequently it is specially shaped to create certain spring actions for fuse clips, relay springs, spring washers, snap rings, and retainers.

In designing many springs of flat stock or strip material, it is often economical and of value to proportion the material so as to obtain a constant stress throughout the spring material. A uniform-section cantilever spring has a stress

$$\sigma = \frac{M}{I/c} = \frac{Fx}{I/c} \tag{a}$$

which is proportional to the distance x if I/c is a constant. But there is no reason why I/c need be a constant. For example, one might design such a spring as that shown in Fig. 10-13b, in which the thickness h is constant but the width b is permitted to vary. Since, for a rectangular section, $I/c = bh^2/6$, we have, from Eq. (a),

$$\frac{bh^2}{6} = \frac{Fx}{\sigma}$$

or

$$b = \frac{6Fx}{h^2\sigma} \tag{b}$$

Since b is linearly related to x, the width b_σ at the base of the spring is

$$b_\sigma = \frac{6Fl}{h^2\sigma} \tag{10-41}$$

But the deflection of this triangular flat spring is more difficult to obtain, because the second moment of area is now a variable. Probably the quickest solution could be obtained by using singularity functions or the method of numerical integration.

The methods of stress and deflection analysis illustrated in previous sections of this chapter have served to illustrate that springs may be analyzed and designed by using the fundamentals discussed in the earlier chapters of this book. This is also true for most of the miscellaneous springs mentioned in this section, and you should now experience no difficulty in reading and understanding the literature of such springs.

PROBLEMS*

10-1 Make a mechanical drawing using two views, or a good freehand sketch, of a helical compression spring closed to its solid length and having a wire diameter of $\frac{1}{2}$ in, an outside diameter of 4 in, and one active coil. The spring is to have plain ends.

10-2 The same as Prob. 10-1, except that the ends are plain and ground.

10-3 A helical compression spring is wound using 0.105-in-diameter music wire. The spring has an outside diameter of 1.225 in with plain ground ends, and 12 total coils.

*An asterisk indicates a design-type problem.

13
Gearing—General

This chapter deals with the geometry, the kinematic relations, and the force analysis of the four principal types of gears. The two chapters that follow deal with other design considerations, such as stress, strength, safety, and reliability.

13-1 TYPES OF GEARS

Spur gears, illustrated in Fig. 13-1, have teeth parallel to the axis of rotation and are used to transmit motion from one shaft to another, parallel, shaft. Of all types, the spur gear is the simplest and, for this reason, will be used to develop the primary kinematic relationships of the tooth form.

Helical gears, shown in Fig. 13-2, have teeth inclined to the axis of rotation. Helical gears can be used for the same applications as spur gears and, when so used, are not as noisy, because of the more gradual engagement of the teeth during meshing. The inclined tooth also develops thrust loads and bending couples, which are not present with spur gearing. Sometimes helical gears are used to transmit motion between nonparallel shafts.

Bevel gears, shown in Fig. 13-3, have teeth formed on conical surfaces and are used mostly for transmitting motion between intersecting shafts. The figure actually illustrates *straight-tooth bevel gears*. *Spiral bevel gears* are cut so that the tooth is no longer straight, but forms a circular arc. *Hypoid gears* are quite similar to spiral bevel gears except that the shafts are offset and nonintersecting.

Shown in Fig. 13-4 is the fourth basic gear type, the *worm* and *worm gear*. As shown, the worm resembles a screw. The direction of rotation of the worm gear, also called the worm wheel, depends upon the direction of rotation of the worm and upon whether the worm teeth are cut right-hand or left-hand. Worm-gear sets are also made so that the teeth of one or both wrap partly around the other. Such sets are called *single-enveloping* and *double-enveloping* worm-gear sets. Worm-gear sets are mostly used when the speed ratios of the two shafts are quite high, say, 3 or more.

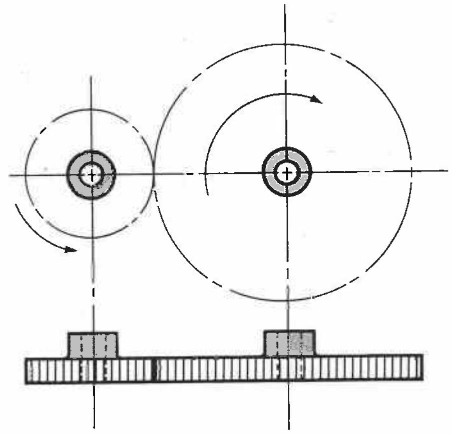

FIGURE 13-1

Spur gears are used to transmit rotary motion between parallel shafts.

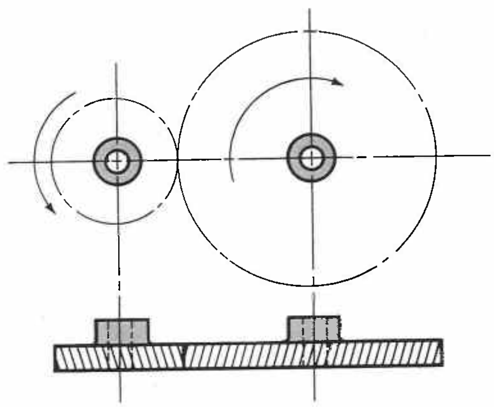

FIGURE 13-2

Helical gears are used to transmit motion between parallel or non-parallel shafts.

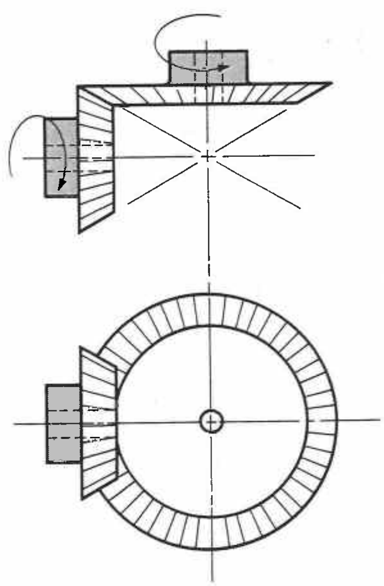

FIGURE 13-3

Bevel gears are used to transmit motion between intersecting shafts.

what by finishing the tooth profiles. The teeth may be finished, after cutting, by either shaving or burnishing. Several shaving machines are available which cut off a minute amount of metal, bringing the accuracy of the tooth profile within the limits of 250 μin.

Burnishing, like shaving, is used with gears which have been cut but not heat-treated. In burnishing, hardened gears with slightly oversize teeth are run in mesh with the gear until the surfaces become smooth.

Grinding and lapping are used for hardened gear teeth after heat treatment. The grinding operation employs the generating principle and produces very accurate teeth. In lapping, the teeth of the gear and lap slide axially so that the whole surface of the teeth is abraded equally.

13-9 STRAIGHT BEVEL GEARS

X

When gears are to be used to transmit motion between intersecting shafts, some form of bevel gear is required. A bevel gearset is shown in Fig. 13-20. Although bevel gears are usually made for a shaft angle of 90°, they may be produced for almost any angle. The teeth may be cast, milled, or generated. Only the generated teeth may be classed as accurate.

The terminology of bevel gears is illustrated in Fig. 13-20. The pitch of bevel gear is measured at the large end of the tooth, and both the circular pitch and the pitch diameter are calculated in the same manner as for spur gears. It should be noted that the clearance is uniform. The pitch angles are defined by the pitch cones meeting at the apex, as shown in the figure. They are related to the tooth numbers as follows

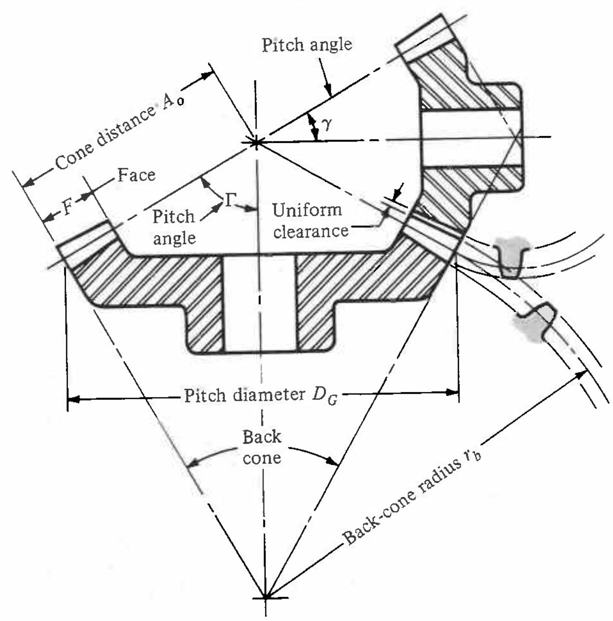

FIGURE 13-20
Terminology of bevel gears.

$$\tan \gamma = \frac{N_P}{N_G} \qquad \tan \Gamma = \frac{N_G}{N_P} \tag{13-10}$$

where the subscripts P and G refer to the pinion and gear, respectively, and where γ and Γ are, respectively, the pitch angles of the pinion and gear.

Figure 13-20 shows that the shape of the teeth, when projected on the back cone, is the same as in a spur gear having a radius equal to the back-cone distance r_b. This is called Tredgold's approximation. The number of teeth in this imaginary gear is

$$N' = \frac{2\pi r_b}{p} \tag{13-11}$$

where N' is the *virtual number of teeth* and p is the circular pitch measured at the large end of the teeth.

Standard straight-tooth bevel gears are cut by using a 20° pressure angle, unequal addenda and dedenda, and full-depth teeth. This increases the contact ratio, avoids undercut, and increases the strength of the pinion.

13-10 PARALLEL HELICAL GEARS

Helical gears, used to transmit motion between parallel shafts, are shown in Fig. 13-2. The helix angle is the same on each gear, but one gear must have a right-hand helix and the other a left-hand helix. The shape of the tooth is an involute helicoid and is illustrated in Fig. 13-21. If a piece of paper cut in the shape of a parallelogram is wrapped around a cylinder, the angular edge of the paper becomes a helix. If we unwind this paper, each point on the angular edge generates an involute curve. This surface obtained when every point on the edge generates an involute is called an *involute helicoid*.

The initial contact of spur-gear teeth is a line extending all the way across the face of the tooth. The initial contact of helical-gear teeth is a point which extends into a line as the teeth come into more engagement. In spur gears the line of contact is parallel to the axis of rotation; in helical gears the line is diagonal across the face of the tooth. It is this gradual engagement of the teeth and the smooth transfer of load from one tooth to another which give helical gears the ability to transmit heavy loads at high speeds. Because of the nature of contact between helical gears, the contact ratio is of only

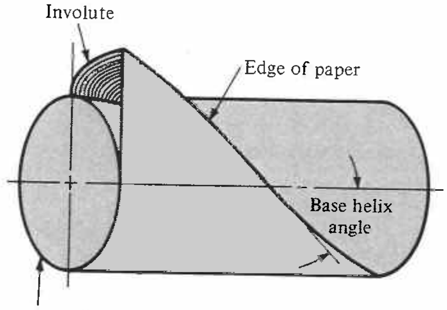

FIGURE 13-21
An involute helicoid.

Shigley &
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Standard HANDBOOK of Machine Design

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ATTACHMENT C

chapter **24**
SPRINGS

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GENERAL NOMENCLATURE†

A	Area, mm² (in²)
b	Width, mm (in)
C	Spring index, D/d
d	Wire diameter, mm (in)
D	Mean diameter (OD minus wire diameter), mm (in)
E	Modulus of elasticity in tension or Young's modulus, MPa (psi)
f	Deflection, mm (in)
g	Gravitational constant, 9.807 m/s² (386.4 in/s²)
G	Shear modulus or modulus of rigidity, MPa (psi)
I	Moment of inertia, mm⁴ (in⁴)
ID	Inside diameter, mm (in)
k	Spring rate, N/mm (lb/in) or N·mm/r (lb·in/r)
K	Design constant
K_w	Stress correction factor for helical springs
L	Length, mm (in)
L_f	Free length, mm (in)
L_s	Length at solid, mm (in)
M	Moment or torque, N·mm (lb·in)

†The symbols presented here are used extensively in the spring industry. They may differ from those used elsewhere in this Handbook.

n	Frequency, Hz
N_a	Number of active coils or waves
N_t	Total number of coils
OD	Outside diameter, mm (in)
P	Load, N (lbf)
r	Radius, mm (in)
S	Stress, MPa (psi)
TS	Tensile strength, MPa (psi)
t	Thickness, mm (in)
YS	Yield strength, MPa (psi)
ρ	Density, g/cm^3 (lb/in^3)
θ	Angular deflection, expressed in number of revolutions
μ	Poisson's ratio

24-1 INTRODUCTION

Spring designing is a complex process. It is an interactive process which may require several iterations before the best design is achieved. Many simplifying assumptions have been made in the design equations, and yet they have proved reliable over the years. When more unusual or complex designs are required, designers should rely on the experience of a spring manufacturer.

The information in this chapter is offered for its theoretical value and should be used accordingly.

24-2 GLOSSARY OF SPRING TERMINOLOGY

active coils: those coils which are free to deflect under load.

baking: heating of electroplated springs to relieve hydrogen embrittlement.

buckling: bowing or lateral displacement of a compression spring; this effect is related to slenderness ratio L/D.

closed and ground ends: same as *closed ends,* except the first and last coils are ground to provide a flat bearing surface.

closed ends: compression spring ends with coil pitch angle reduced so they are square with the spring axis and touch the adjacent coils.

close-wound: wound so that adjacent coils are touching.

deflection: motion imparted to a spring by application or removal of an external load.

elastic limit: maximum stress to which a material may be subjected without permanent set.

endurance limit: maximum stress, at a given stress ratio, at which material will operate in a given environment for a stated number of cycles without failure.

free angle: angular relationship between arms of a helical torsion spring which is not under load.

free length: overall length of a spring which is not under load.

gradient: see *rate.*

heat setting: a process to prerelax a spring in order to improve stress-relaxation resistance in service.

helical springs: springs made of bar stock or wire coiled into a helical form; this category includes compression, extension, and torsion springs.

hooks: open loops or ends of extension springs.

hysteresis: mechanical energy loss occurring during loading and unloading of a spring within the elastic range. It is illustrated by the area between load-deflection curves.

initial tension: a force that tends to keep coils of a close-wound extension spring closed and which must be overcome before the coils start to open.

loops: formed ends with minimal gaps at the ends of extension springs.

mean diameter: in a helical spring, the outside diameter minus one wire diameter.

modulus in shear or torsion (modulus of rigidity *G*): coefficient of stiffness used for compression and extension springs.

modulus in tension or bending (Young's modulus *E*): coefficient of stiffness used for torsion or flat springs.

moment: a product of the distance from the spring axis to the point of load application and the force component normal to the distance line.

natural frequency: lowest inherent rate of free vibration of a spring vibrating between its own ends.

pitch: distance from center to center of wire in adjacent coils in an open-wound spring.

plain ends: end coils of a helical spring having a constant pitch and with the ends not squared.

plain ends, ground: same as *plain ends,* except wire ends are ground square with the axis.

rate: spring gradient, or change in load per unit of deflection.

residual stress: stress mechanically induced by such means as set removal, shot peening, cold working, or forming; it may be beneficial or not, depending on the spring application.

set: permanent change of length, height, or position after a spring is stressed beyond material's elastic limit.

set point: stress at which some arbitrarily chosen amount of set (usually 2 percent) occurs; set percentage is the set divided by the deflection which produced it.

set removal: an operation which causes a permanent loss of length or height because of spring deflection.

solid height: length of a compression spring when deflected under load sufficient to bring all adjacent coils into contact.

spiral springs: springs formed from flat strip or wire wound in the form of a spiral, loaded by torque about an axis normal to the plane of the spiral.

spring index: ratio of mean diameter to wire diameter.

squared and ground ends: see *closed* and *ground ends.*

squared ends: see *closed ends.*

squareness: angular deviation between the axis of a compression spring in a free state and a line normal to the end planes.

stress range: difference in operating stresses at minimum and maximum loads.

stress ratio: minimum stress divided by maximum stress.

stress relief: a low-temperature heat treatment given springs to relieve residual stresses produced by prior cold forming.

torque: see *moment.*

total number of coils: the sum of the number of active and inactive coils in a spring body.

24-3 SELECTION OF SPRING MATERIALS

24-3-1 Chemical and Physical Characteristics

Springs are resilient structures designed to undergo large deflections within their elastic range. It follows that the materials used in springs must have an extensive elastic range.

Some materials are well known as spring materials. Although they are not specifically designed alloys, they do have the elastic range required. In steels, the medium- and high-carbon grades are suitable for springs. Beryllium copper and phosphor bronze are used when a copper-base alloy is required. The high-nickel alloys are used when high strength must be maintained in an elevated-temperature environment.

The selection of material is always a cost-benefit decision. Some factors to be considered are costs, availability, formability, fatigue strength, corrosion resistance, stress relaxation, and electric conductivity. The right selection is usually a compromise among these factors. Table 24-1 lists some of the more commonly used metal alloys and includes data which are useful in material selection.

Surface quality has a major influence on fatigue strength. This surface quality is a function of the control of the material manufacturing process. Materials with high surface integrity cost more than commercial grades but must be used for fatigue applications, particularly in the high cycle region.

24-3-2 Heat Treatment of Springs

Heat treatment is a term used in the spring industry to describe both low- and high-temperature heat treatments. Low-temperature heat treatment, from 350 to 950°F (175 to 510°C), is applied to springs after forming to reduce unfavorable residual stresses and to stabilize parts dimensionally.

When steel materials are worked in the spring manufacturing process, the yield point is lowered by the unfavorable residual stresses. A low-temperature heat treatment restores the yield point. Most heat treatment is done in air, and the minor oxide that is formed does not impair the performance of the springs.

When hardened high-carbon-steel parts are electroplated, a phenomenon known as *hydrogen embrittlement* occurs, in which hydrogen atoms diffuse into the metallic lattice, causing previously sound material to crack under sustained stress. Low-temperature baking in the range of 375 to 450°F (190 to 230°C) for times ranging from 0.5 to 3 h, depending on the type of plating and the degree of embrittlement, will reduce the concentration of hydrogen to acceptable levels.

TABLE 24-1 Typical Properties of Common Spring Materials

Common Name	Young's Modulus E (1) MPa 10^3	Young's Modulus E (1) psi 10^6	Modulus of Rigidity G (1) MPa 10^3	Modulus of Rigidity G (1) psi 10^6	Density (1) g/cm^3 (lb/in^3)	Electrical Conductivity (1) % IACS	Sizes Normally Available (2) Min. mm (in.)	Sizes Normally Available (2) Max. mm (in.)	Typical Surface Quality (3)	Maximum Service Temperature (4) °C	Maximum Service Temperature (4) °F
Carbon Steel Wires:											
Music (5)	207	(30)	79.3	(11.5)	7.86 (0.284)	7	0.10 (0.004)	6.35 (0.250)	a	120	250
Hard Drawn (5)	207	(30)	79.3	(11.5)	7.86 (0.284)	7	0.13 (0.005)	16 (0.625)	c	150	250
Oil Tempered	207	(30)	79.3	(11.5)	7.86 (0.284)	7	0.50 (0.020)	16 (0.625)	c	150	300
Valve Spring	207	(30)	79.3	(11.5)	7.86 (0.284)	7	1.3 (0.050)	6.35 (0.250)	a	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)	a,b	220	425
Chrome Silicon	207	(30)	79.3	(11.5)	7.86 (0.284)	5	0.50 (0.020)	9.5 (0.375)	a,b	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)	b	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)	b	315	600
NiCr A286	200	(29)	71.7	(10.4)	8.03 (0.290)	2	0.40 (0.016)	5 (0.200)	b	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)	b	95	200
Silicon Bronze (A)	103	(15)	38.6	(5.6)	8.53 (0.308)	7	0.10 (0.004)	12.5 (0.500)	b	95	200
Silicon Bronze (B)	117	(17)	44.1	(6.4)	8.75 (0.316)	12	0.10 (0.004)	12.5 (0.500)	b	95	200
Beryllium Copper	128	(18.5)	48.3	(7.0)	8.26 (0.298)	21	0.08 (0.003)	12.5 (0.500)	b	205	400
Spring Brass, CA260	110	(16)	42.0	(6.0)	8.53 (0.308)	17	0.10 (0.004)	12.5 (0.500)	b	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)	b	320	700
Inconel Alloy X750	214	(31)	79.3	(11.5)	8.25 (0.298)	1	0.10 (0.004)	12.5 (0.500)	b	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)	b	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)	b	230	450
Monel Alloy K500	179	(26)	66.2	(9.6)	8.46 (0.306)	3	0.05 (0.002)	9.5 (0.375)	b	260	500

Monel Alloy 400
Monel Alloy K500 | 179 | (26) | 66.2 | (9.6) | 8.46 (0.306)

Carbon Steel Strip:											
AISI 1050	207	(30)	79.3	(11.5)	7.86 (0.284)	7	0.25 (0.010)	3 (0.125)	b	95	200
1065	207	(30)	79.3	(11.5)	7.86 (0.284)	7	0.08 (0.003)	3 (0.125)	b	95	200
1074, 1075	207	(30)	79.3	(11.5)	7.86 (0.284)	7	0.08 (0.003)	3 (0.125)	b	120	250
1095	207	(30)	79.3	(11.5)	7.86 (0.284)	7	0.08 (0.003)	3 (0.125)	b	120	250
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)	b	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)	b	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)	b	95	200
Beryllium Copper	128	(18.5)	48	(7.0)	8.26 (0.298)	21	0.08 (0.003)	9.5 (0.375)	b	205	400

(1) Elastic moduli, density and electrical conductivity can vary with cold work, heat treatment and operating stress. These variations are usually minor but should be considered if one or more of these properties is critical.
(2) Sizes normally available are diameters for wire; thicknesses for strip.
(3) Typical surface quality ratings. (For most materials, special processes can be specified to upgrade typical values.)
a. Maximum defect depth: 0 to 0.5% of d or t.
b. Maximum defect depth: 1.0% of d or t.
c. Defect depth: less than 3.5% of d or t.
(4) Maximum service temperatures are guidelines and may vary due to operating stress and allowable relaxation.
(5) Music and hard drawn are commercial terms for patented and cold-drawn carbon steel spring wire.

SOURCE: Associated Spring, Barnes Group Inc.

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TABLE 24-2 Typical Heat Treatments for Springs after Forming

Materials	Heat Treatment °C	Heat Treatment °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 hour	900/1 hour
Condition A to TH 1050	760/1 hour cool to 15°C followed by 565/1 hour	1400/1 hour, cool to 60°F followed by 1050/1 hour
Monel:		
Alloy 400	300–315	575–600
Alloy K500, Spring Temper	525/4 hours	980/4 hours
Inconel:		
Alloy 600	400–510	750–950
Alloy X-750:		
#1 Temper	730/16 hours	1350/16 hours
Spring Temper	650/4 hours	1200/4 hours
Copper Base, Cold Worked (Brass, Phosphor Bronze, etc.)	175–205	350–400
Beryllium Copper:		
Pretempered (Mill Hardened)	205	400
Solution Annealed, Temper Rolled or Drawn	315/2-3 hours	600/2–3 hours
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.

SOURCE: Associated Spring, Barnes Group Inc.

High-temperature heat treatments are used to strengthen annealed material after spring forming. High-carbon steels are austenitized at 1480 to 1652°F (760 to 900°C), quenched to form martensite, and then tempered to final hardness. Some nickel-base alloys are strengthened by high-temperature aging. Oxidation will occcur at these temperatures, and it is advisable to use a protective atmosphere in the furnace.

Heat treatments for many common materials are listed in Table 24-2. Unless otherwise noted, 20 to 30 min at the specified temperature is sufficient. Thin, flimsy cross-sectional springs can be distorted by the heat treatment operation. Pretempered materials are available for use in such cases.

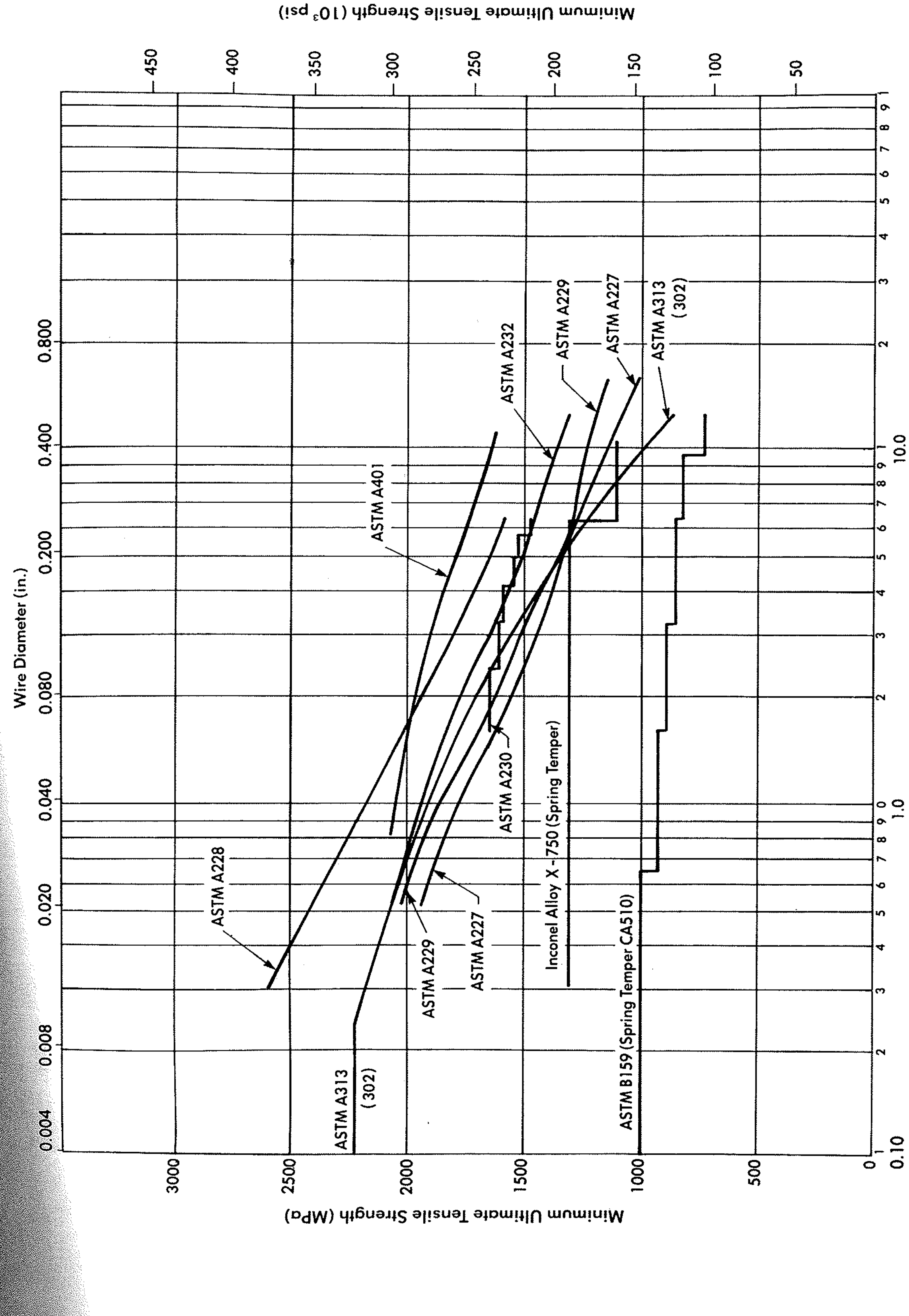

FIG. 24-1 Minimum tensile strengths of spring wire. *(Associated Spring, Barnes Group Inc.)*

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24-3-3 Relaxation

The primary concern in elevated-temperature applications is stress relaxation. *Stress relaxation* is the loss of load or spring length that occurs when a spring is held at load or cycled under load. Heat affects modulus and tensile strength. In addition to the factors of stress, time, and temperature which affect relaxation, other controllable factors are

1. Alloy type—the highly alloyed materials are generally more temperature-resistant.
2. Residual stresses—such stresses remaining from forming operations are detrimental to relaxation resistance. Use the highest practical stress-relief temperature.
3. Heat setting—procedures employed to expose springs under some load to stress and heat to prepare them for a subsequent exposure. The effect is to remove the first stage of relaxation.

24-3-4 Corrosion

The specific effect of a corrosive environment on spring performance is difficult to predict. In general, if the environment causes damage to the spring surface, the life and the load-carrying ability of the spring will be reduced.

The most common methods of combating corrosion are to use materials resistant or inert to the particular corrosive environment or to use coatings that slow down the rate of corrosion attack on the base metal. The latter approach is most often the most cost-effective method.

TABLE 24-3 Ranking of Relative Costs of Common Spring Wires

Wire	Specification	Relative Cost of 2 mm (0.079″) Dia. Mill Quantities	Ware-House Lots
Patented and Cold Drawn Oil Tempered	ASTM A227 ASTM A229	1.0 1.3	1.0 1.3
Music Carbon Valve Spring	ASTM A228 ASTM A230	2.6 3.1	1.4 1.9
Chrome Silicon Valve Stainless Steel (Type 302)	ASTM A401 ASTM A313 (302)	4.0 7.6	3.9 4.7
Phosphor Bronze Stainless Steel (Type 631) (17–7 PH)	ASTM ASTM A 313 (631)	8.0 11	6.7 8.7
Beryllium Copper Inconel Alloy X–750	ASTM B197	27 44	17 31

SOURCE: Associated Spring, Barnes Group Inc.

SPRING WIRE. The tensile strength of spring wire varies inversely with the wire diameter (Fig. 24-1).

Common spring wires with the highest strengths are ASTM A228 (music wire) and ASTM A401 (oil-tempered chrome silicon). Wires having slightly lower tensile strength and with surface quality suitable for fatigue applications are ASTM A313 type 302 (stainless steel), ASTM A230 (oil-tempered carbon valve-spring-quality steel), and ASTM A232 (oil-tempered chrome vanadium). For most static applications ASTM A227 (hard-drawn carbon steel) and ASTM A229 (oil-tempered carbon steel) are available at lower strength levels. Table 24-3 ranks the relative costs of common spring materials based on hard-drawn carbon steel as 1.0.

SPRING STRIP. Most "flat" springs are made from AISI grades 1050, 1065, 1074, and 1095 steel strip. Strength and formability characteristics are shown in Fig. 24-2, covering the range of carbon content from 1050 to 1095. Since all carbon levels can be obtained at all strength levels, the curves are not identified by composition. Figure 24-3 shows the tensile strength versus Rockwell hardness for tempered carbon-steel strip. Edge configurations for steel strip are shown in Fig. 24-4.

Formability of annealed spring steels is shown in Table 24-4, and typical properties of various spring-tempered alloy strip materials are shown in Table 24-5.

24-4 HELICAL COMPRESSION SPRINGS

24-4-1 General

A helical compression spring is an open-pitch spring which is used to resist applied compression forces or to store energy. It can be made in a variety of configurations and from different shapes of wire, depending on the application. Round, high-car-

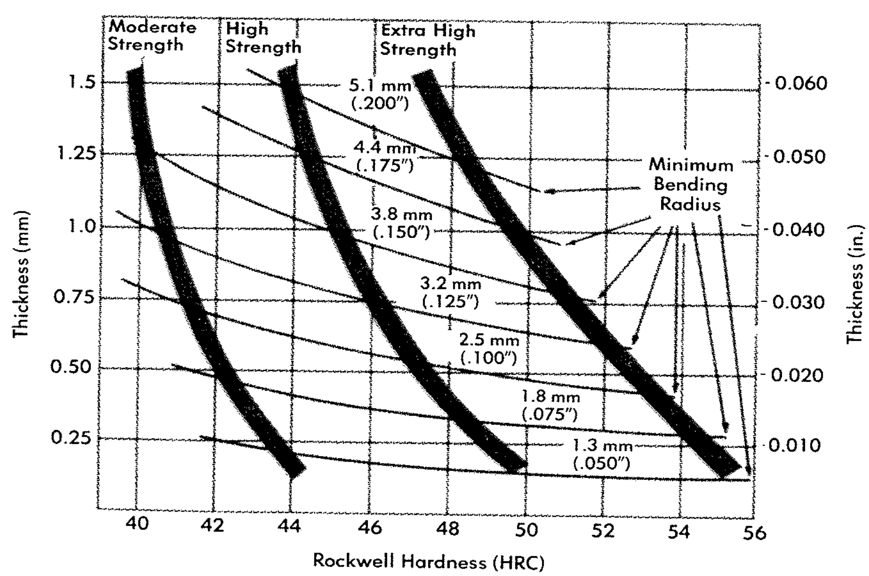

FIG. 24-2 Minimum transverse bending radii for various tempers and thicknesses of tempered spring steel. *(Associated Spring, Barnes Group Inc.)*

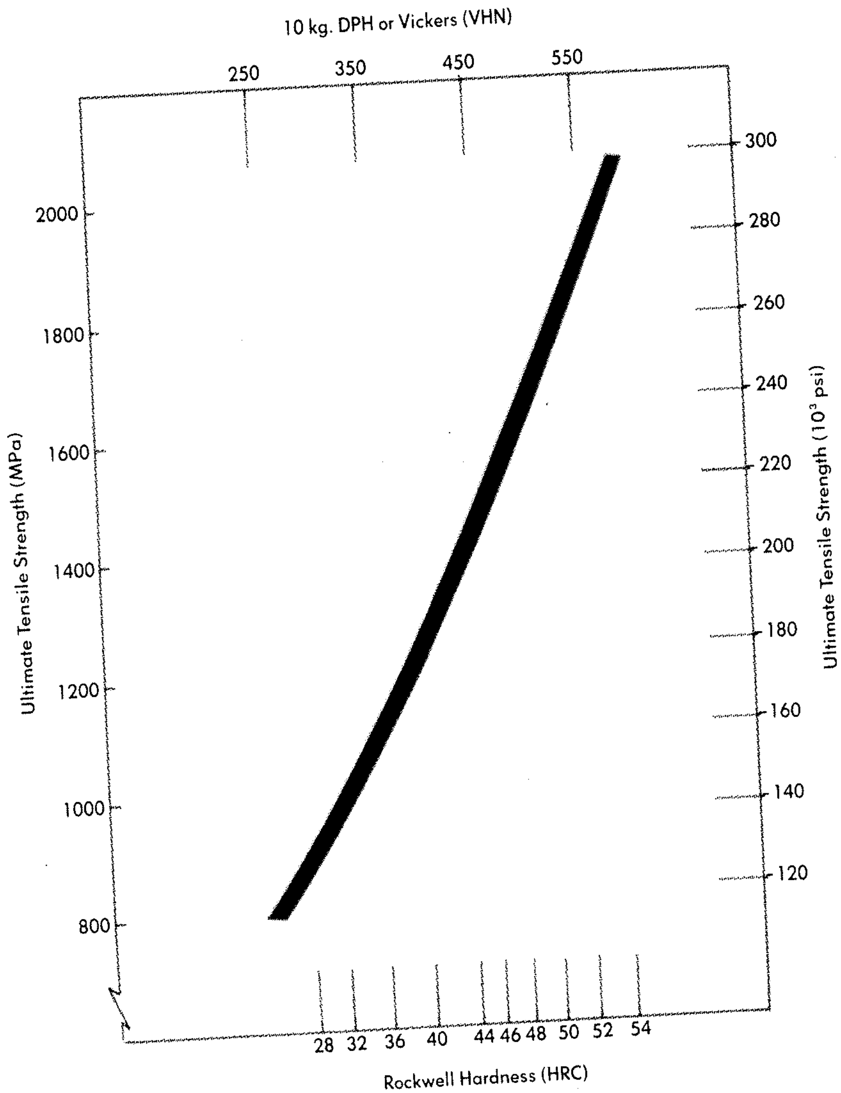

FIG. 24-3 Tensile strength versus hardness of quenched and tempered spring steel. *(Associated Spring, Barnes Group Inc.)*

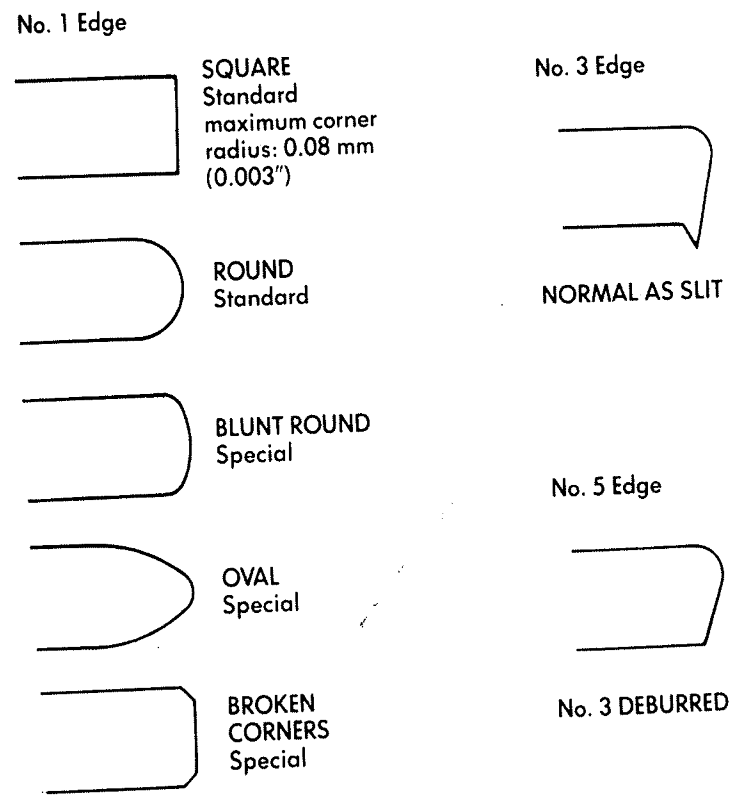

FIG. 24-4 Edges available on steel strip. *(Associated Spring, Barnes Group Inc.)*

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spring steel. (Associated Spring, Barnes Group Inc.)

TABLE 24-4 Formability of Annealed Spring Steels

Thickness (t) mm (in.)	Direction of Bend	AISI 1050 N_t/t		AISI 1065 N_t/t		AISI 1074 N_t/t	AISI 1095 N_t/t
		Annealed (standard lowest max.)	WBS* Barco-Form®	Annealed (standard lowest max.)	WBS Barco-Form	Annealed (standard lowest max.)	Annealed (standard lowest max.)
1.9 mm (0.076)-over	⊥ ∥	2 4	0 3	2 4	0 3	2 4	3 5
0.9–1.89 mm (0.036–0.075″)	⊥ ∥	1 2	0 1	1 2	0 1	1 2	2 3
0.37–0.89 mm (0.015–0.035″)	⊥ ∥	0 1	0 0	0 1 1/2	0 1	1 1 1/2	1 2
0.2–0.36 mm (0.008–0.014″)	⊥ ∥	0 0	0 0	0 0	0 0	1 1	1 1

Formability is determined by slowly bending a sample over 180° until its ends are parallel. The measured distance between the ends is N_t. For example, if $N_t = 4$ and $t = 2$, then $N_t/t = 2$

*Wallace Barnes Steel.

SOURCE: Associated Spring, Barnes Group Inc.

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TABLE 24-5 Typical Properties of Spring-Tempered Alloy Strip

Material	Tensile Strength MPa (10^3 psi)	Rockwell Hardness	Elongation(1) Percent	Bend Factor (1) (2r/t trans. bends)	Modulus of Elasticity 10^4 MPa (10^6 psi)	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 (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

(1) Before heat treatment.

SOURCE: Associated Spring, Barnes Group Inc.

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bon-steel wire is the most common spring material, but other shapes and compositions may be required by space and environmental conditions.

Usually the spring has a uniform coil diameter for its entire length. Conical, barrel, and hourglass shapes are a few of the special shapes used to meet particular load-deflection requirements.

Helical compression springs are stressed in the torsional mode. The stresses, in the elastic range, are not uniform about the wire's cross section. The stress is greatest at the surface of the wire and, in particular, at the inside diameter (ID) of the spring.

In some circumstances, residual bending stresses are present as well. In such cases, the bending stresses become negligible after set is removed (or the elastic limit is exceeded) and the stresses are redistributed more uniformly about the cross section.

24-4-2 Compression Spring Terminology

The definitions that follow are for terms which have evolved and are commonly used in the spring industry. Figure 24-5 shows the relationships among the characteristics.

WIRE DIAMETER d. Round wire is the most economical form. Rectangular wire is used in situations where space is limited, usually to reduce solid height.

COIL DIAMETER. The outside diameter (OD) is specified when a spring operates in a cavity. The inside diameter is specified when the spring is to operate over a rod. The mean diameter D is either OD minus the wire size, or ID plus the wire size.

The coil diameter increases when a spring is compressed. The increase, though small, must be considered whenever clearances could be a problem. The diameter increase is a function of the spring pitch and follows the equation

$$\mathrm{OD}_{\text{at solid}} = \sqrt{D^2 + \frac{p^2 - d^2}{\pi^2}} + d \qquad (24\text{-}1)$$

where p = pitch and d = wire size.

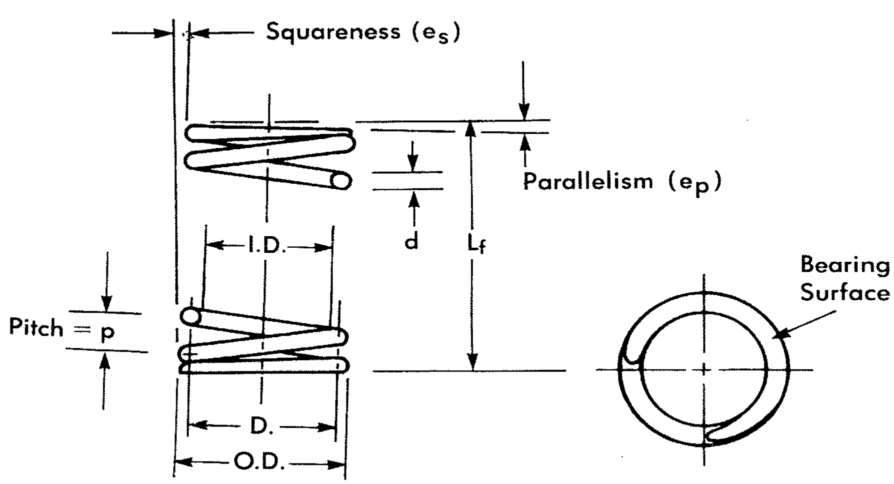


FIG. 24-5 Dimensional terminology for helical compression springs. *(Associated Spring, Barnes Group Inc.)*

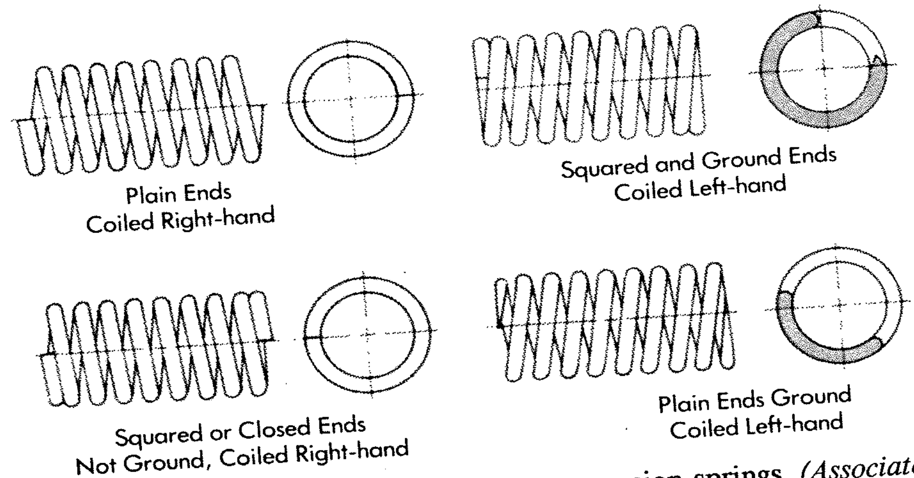


FIG. 24-6 Types of ends for helical compression springs. *(Associated Spring, Barnes Group Inc.)*

SPRING INDEX. Spring index C is the ratio of the mean diameter to the wire diameter (or to the radial dimension if the wire is rectangular). The preferred range of index is 5 to 9, but ranges as low as 3 and as high as 15 are commercially feasible. The very low indices are hard to produce and require special setup techniques. High indices are difficult to control and can lead to spring tangling.

FREE LENGTH. Free length L_f is the overall length measured parallel to the axis when the spring is in a free, or unloaded, state. If loads are not given, the free length should be specified. If they are given, then free length should be a reference dimension which can be varied to meet the load requirements.

TYPES OF ENDS. Four basic type of ends are used: closed (squared) ends, closed (squared) ends ground, plain ends, and plain ends ground. Figure 24-6 illustrates the various end conditions. Closed and ground springs are normally supplied with a ground bearing surface of 270 to 330°.

NUMBER OF COILS. The number of coils is defined by either the total number of coils N_t or the number of active coils N_a. The difference between N_t and N_a equals the number of inactive coils, which are those end coils that do not deflect during service.

SOLID HEIGHT. The solid height L_s is the length of the spring when it is loaded with enough force to close all the coils. For ground springs, $L_s = N_t d$. For unground springs, $L_s = (N_t + 1)d$.

DIRECTION OF THE HELIX. Springs can be made with the helix direction either right or left hand. Figure 24-7 illustrates how to define the direction. Springs that are nested one inside the other should have opposite helix directions. If a spring is to be assembled onto a screw thread, the direction of the helix must be opposite to that of the thread.

SPRING RATE. Spring rate k is the change in load per unit deflection. It is expressed as

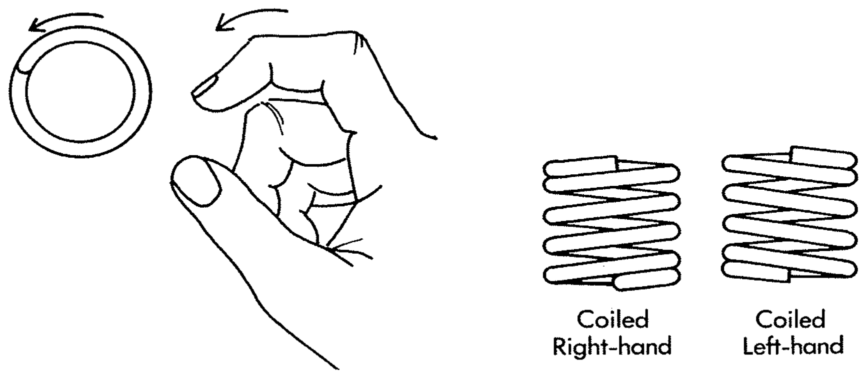


FIG. 24-7 Direction of coiling of helical compression springs. *(Associated Spring, Barnes Group Inc.)*

$$k = \frac{P}{f} = \frac{Gd^4}{8D^3N_a} \tag{24-2}$$

where G = shear modulus.

The rate equation is accurate for a deflection range between 15 and 85 percent of the maximum available deflection. When compression springs are loaded in parallel, the combined rate of all the springs is the sum of the individual rates. When the springs are loaded in series, the combined rate is

$$k = \frac{1}{1/k_1 + 1/k_2 + 1/k_3 + \cdots + 1/k_n} \tag{24-3}$$

This relationship can be used to design a spring with variable diameters. The design method is to divide the spring into many small increments and calculate the rate for each increment. The rate for the whole spring is calculated as in Eq. (24-3).

STRESS. Torsional stress S is expressed as

$$S = \frac{8K_wPD}{\pi d^3} \tag{24-4}$$

Under elastic conditions, torsional stress is not uniform around the wire's cross section because of the coil curvature and direct shear loading.

The highest stress occurs at the surface in the inside diameter of the spring, and it is computed by using the stress factor K_w. In most cases, the correction factor is expressed as

$$K_{w_1} = \frac{4C - 1}{4C - 4} + \frac{0.615}{C} \tag{24-5}$$

The stress-concentration factor K_{w_1} becomes K_{w_2} after a spring has been set out because stresses become more uniformly distributed after subjecting the cross section to plastic flow during set-out:

$$K_{w_2} = 1 + \frac{0.5}{C} \tag{24-6}$$

The appropriate stress correction factor is discussed in the Sec. 24-4-3.

LOADS. If deflection is known, the load is found by multiplying deflection by the spring rate. When the stress is either known or assumed, loads can be obtained from the stress equation.

Loads should be specified at a test height so that the spring manufacturer can control variations by adjustments of the free length. The load-deflection curve is not usually linear at the start of deflection from free position or when the load is very close to solid height. It is advisable to specify loads at test heights between 15 and 85 percent of the load-deflection range.

Loads can be conveniently classified as static, cyclic, and dynamic. In static loading, the spring will operate between specified loads only a few times. In other instances, the spring may remain under load for a long time. In cyclic applications, the spring may typically be required to cycle between load points from 10^4 to more than 10^9 times. During dynamic loading, the rate of load application is high and causes a surge wave in the spring which usually induces stresses higher than calculated from the standard stress equation.

BUCKLING. Compression springs with a free length more than 4 times the mean coil diameter may buckle when compressed. Guiding the spring, either in a tube or over a rod, can minimize the buckling but can result in additional friction which will affect loads, especially when the L_f/D ratio is high.

Buckling conditions are shown in Figs. 24-8 and 24-9 for springs loaded axially and with squared and ground ends. Buckling occurs at points above and to the right of the curves. Curve *A* is for the springs with one end on a fixed, flat surface and the other end free to tip. Curve *B* is for springs with both ends on fixed, flat surfaces. The tendency to buckle is clearly less for curve *B* springs.

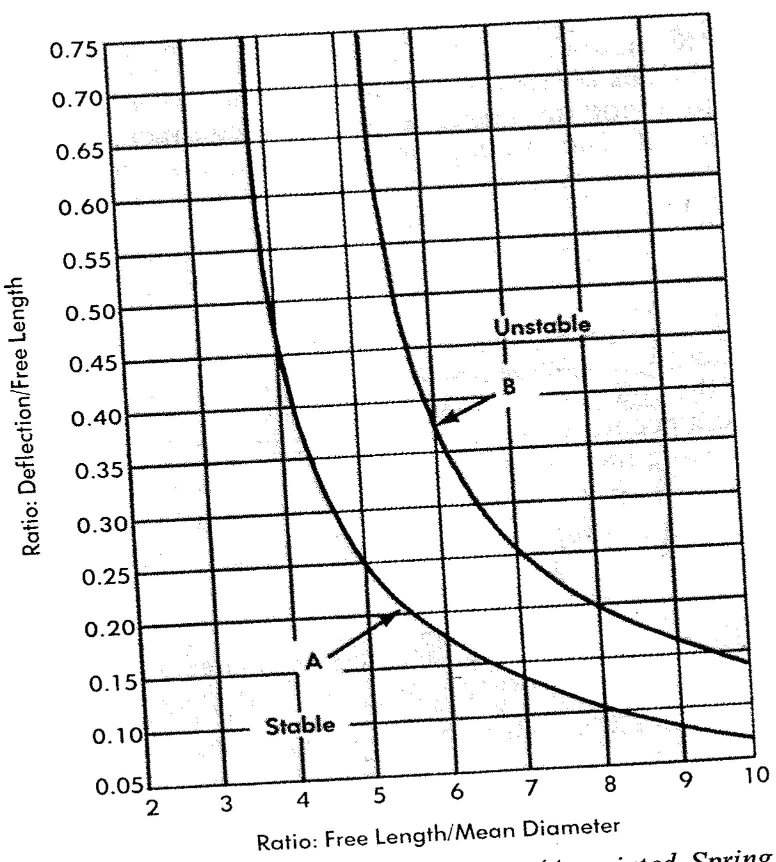


FIG. 24-8 Critical buckling curves. *(Associated Spring, Barnes Group Inc.)*

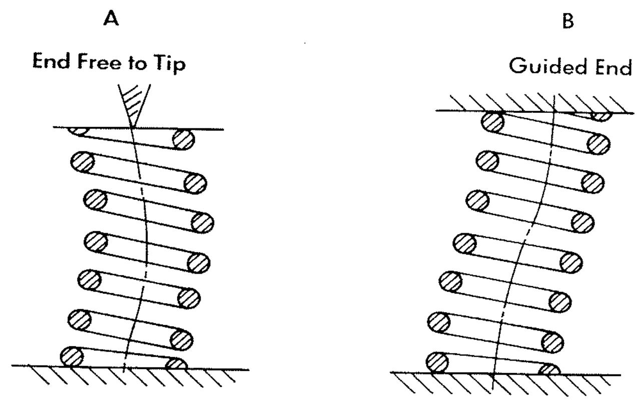


FIG. 24-9 End conditions used to determine critical buckling. *(Associated Spring, Barnes Group Inc.)*

24-4-3 Choice of Operating Stress

The choice of operating stress depends on whether the application is static or cyclic. For static applications, yield strength or stress-relaxation resistance of the material limits the load-carrying ability of the springs. The required cycles are few, if any, and the velocity of the end coils is so low as to preclude surging or impact conditions.

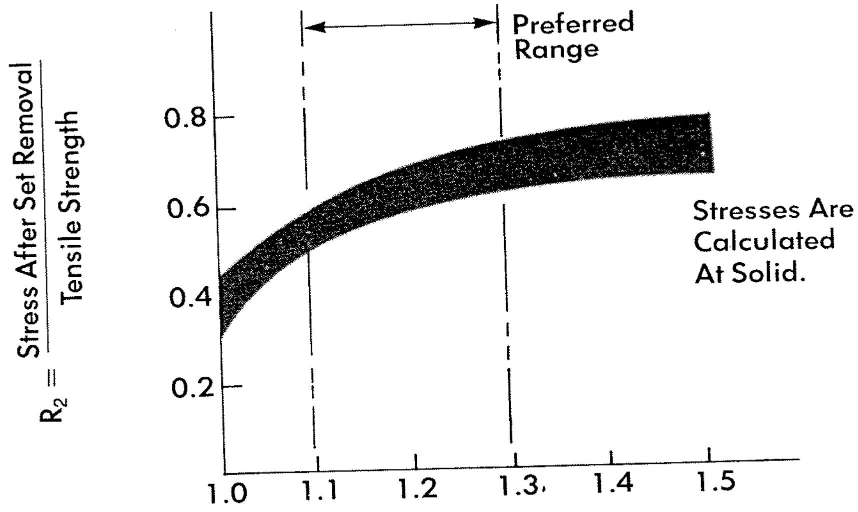
The maximum allowable torsional stresses for static applications are shown in Table 24-6 as percentages of tensile strengths for common spring materials. To calculate the stress before set removal, use the K_{w_1} correction factor. If the calculated stress is greater than the indicated percentage of the tensile strength, then the spring will take a permanent set when deflected to solid. The amount of set is a function of the amount by which the calculated stress exceeds the tabular percentage.

It is common practice, in static applications, to increase the load-carrying capability of a spring by making it longer than the desired free length and then compressing it to solid. The spring *sets* to its final desired length. This procedure is called *removing set.* It induces favorable residual stresses which allow for significantly higher stresses than in springs not having the set removed. The loss of the length should be at least 10 percent to be effective (see Fig. 24-10).

TABLE 24-6 Maximum Allowable Torsional Stresses for Helical Compression Springs in Static Applications

Materials	Maximum % of Tensile Strength	
	Before Set Removed (K_{W1})	After Set Removed (K_{W2})
Patented and cold drawn carbon steel	45%	65–75%
Hardened and tempered carbon and low alloy steel	50%	
Austenitic stainless steels	35%	
Nonferrous alloys	35%	

SOURCE: Associated Spring, Barnes Group Inc.



$$R_2 = \frac{\text{Stress After Set Removal}}{\text{Tensile Strength}}$$

$$R_1 = \frac{\text{Stress Before Set Removal}}{\text{Stress After Set Removal}} = \frac{S_1}{S_2}$$

FIG. 24-10 Spring load-carrying ability versus amount of set removed. *(Associated Spring, Barnes Group Inc.)*

Note that set removal causes stresses to be more uniformly distributed about the cross section. Therefore, stress after set removal is calculated by using the K_{w_2} correction factor. If the stress calculated by using the K_{w_2} correction factor exceeds the percentage of tensile strength shown in Table 24-6, the spring cannot be made. It is then necessary either to lower the design stress or to select a higher-strength material.

For cyclic applications, the load-carrying ability of the spring is limited by the fatigue strength of the material. To select the optimum stress level, spring costs must be balanced against reliability. The designer should know the operating environment, desired life, stress range, frequency of operation, speed of operation, and permissible levels of stress relaxation in order to make a cost-reliability decision.

Fatigue life can be severely reduced by pits, seams, or tool marks on the wire surface where stress is at a maximum. Shot peening improves fatigue life, in part, by minimizing the harmful effects of surface defects. It does not remove them. Additionally, shot peening imparts favorable compression stresses to the surface of the spring wire.

Maximum allowable stresses for fatigue applications should be calculated by using the K_{w_1} stress correction factor. Table 24-7 shows the estimated fatigue life for common spring materials. Note the significant increase in fatigue strength from shot peening.

The fatigue life estimates in Table 24-7 are guideline values which should be used only where specific data are unavailable. The values are conservative, and most springs designed using them will exceed the anticipated lives.

24-4-4 Dynamic Loading under Impact (24-11)

When a spring is loaded or unloaded, a surge wave is established which transmits torsional stress from the point of load along the spring's length to the point of restraint. The surge wave will travel at a velocity approximately one-tenth that of a normal, torsional-stress wave. The velocity of the torsional-stress wave V_T, in meters per second (m/s) [inches per second (in/s)], is given by

TABLE 24-7 Maximum Allowable Torsional Stress for Round-Wire Helical Compression Springs in Cyclic Applications

Fatigue Life (cycles)	Percent of Tensile Strength			
	ASTM A228, Austenitic Stainless Steel and Nonferrous		ASTM A230 and A232	
	Not Shot-Peened	Shot-Peened	Not Shot-Peened	Shot-Peened
10^5	36	42	42	49
10^6	33	39	40	47
10^7	30	36	38	46

This information is based on the following conditions: no surging, room temperature and noncorrosive environment.

$$\text{Stress ratio in fatigue} = \frac{\text{S minimum}}{\text{S maximum}} = 0$$

SOURCE: Associated Spring, Barnes Group Inc.

$$V_T = 10.1\sqrt{\frac{Gg}{\rho}}\ \text{m/s} \quad \text{or} \quad V_T = \sqrt{\frac{Gg}{\rho}}\ \text{in/s} \tag{24-7}$$

The velocity of the surge wave V_s varies with material and design but is usually in the range of 50 to 500 m/s. The surge wave limits the rate at which a spring can absorb or release energy by limiting the impact velocity V. *Impact velocity* is defined as the spring velocity parallel to the spring axis and is a function of stress and material as shown:

$$V \simeq 10.1S\sqrt{\frac{g}{2\rho G}}\ \text{m/s} \quad \text{or} \quad V \simeq S\sqrt{\frac{g}{2\rho G}}\ \text{in/s} \tag{24-8}$$

For steel, this reduces to

$$V = \frac{S}{35.5}\ \text{m/s} \quad \text{or} \quad V = \frac{S}{131}\ \text{in/s} \tag{24-9}$$

If a spring is compressed to a given stress level and released instantaneously, the maximum spring velocity is the stress divided by 35.5. Similarly, if the spring is loaded at known velocity, the instantaneous stress can be calculated. At very high load velocities, the instantaneous stress will exceed the stress calculated by the conventional equation. This will limit design performance. Since the surge wave travels the length of the spring, springs loaded at high velocity often are subject to resonance.

24-4-5 Dynamic Loading—Resonance

A spring experiences resonance when the frequency of cyclic loading is near the natural frequency or a multiple of it. Resonance can cause an individual coil to deflect to stress levels above those predicted by static stress analysis. Resonance can also cause the spring to bounce, resulting in loads lower than calculated. To avoid these effects, the natural frequency should be a minimum of 13 times the operating frequency.

For a compression spring with both ends fixed and no damper, the natural frequency in International System (SI) units is

$$n = \frac{1.12(10^3)d}{D^2N_a} \sqrt{\frac{Gg}{\rho}} \qquad (24\text{-}10)$$

For steel, this equation becomes

$$n = \frac{3.5(10^5)d}{D^2N_a} \qquad (24\text{-}11)$$

where n = frequency in hertz (Hz). The corresponding equation in U.S. Customary System (USCS) units is

$$n = \frac{d}{9D^2N_a} \sqrt{\frac{Gg}{\rho}} \qquad (24\text{-}12)$$

and for steel we have

$$n = \frac{14(10^3)d}{D^2N_a} \qquad (24\text{-}13)$$

If the spring cannot be designed to have a natural frequency more than 13 times the operating frequency, energy dampers may be employed. They are generally friction devices which rub against the coils. Often, variable-pitch springs are used to minimize resonance effects.

24-4-6 Rectangular-Wire Springs

In applications where high loads and relatively low stresses are required but solid height is also restricted, rectangular wire can be used to increase the material volume while maintaining the maximum solid-height limitation.

Springs made of rectangular wire with the long side of the wire cross section perpendicular to the axis of the coils can store more energy in a smaller space than an equivalent, round-wire spring.

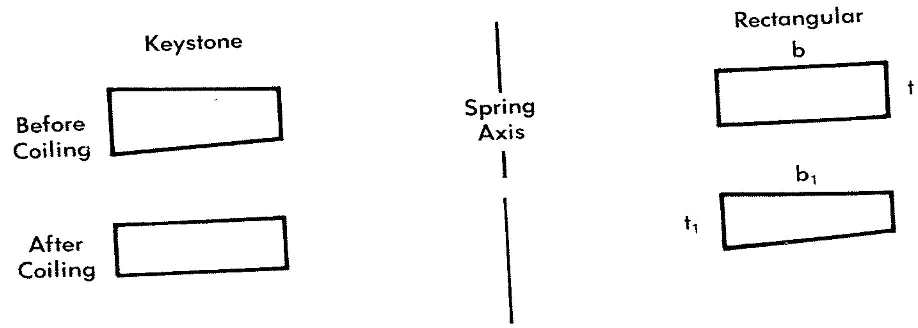


FIG. 24-11 Wire cross section before and after coiling. *(Associated Spring, Barnes Group Inc.)*

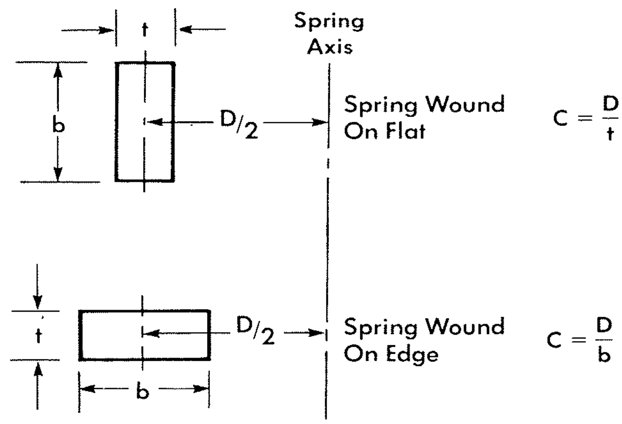


FIG. 24-12 Rectangular-wire compression spring wound on flat or edge. *(Associated Spring, Barnes Group Inc.)*

When rectangular wire is coiled, it changes from a rectangular to a keystone shape, as shown in Fig. 24-11. Similarly, if the wire is made to the keystone shape, it will become rectangular after coiling. The cross-sectional distortion can be approximated by

$$t_1 = t\frac{C + 0.5}{C} \tag{24-14}$$

where t_1 = wider end of keystone section and t = original, smaller dimension of rectangle.

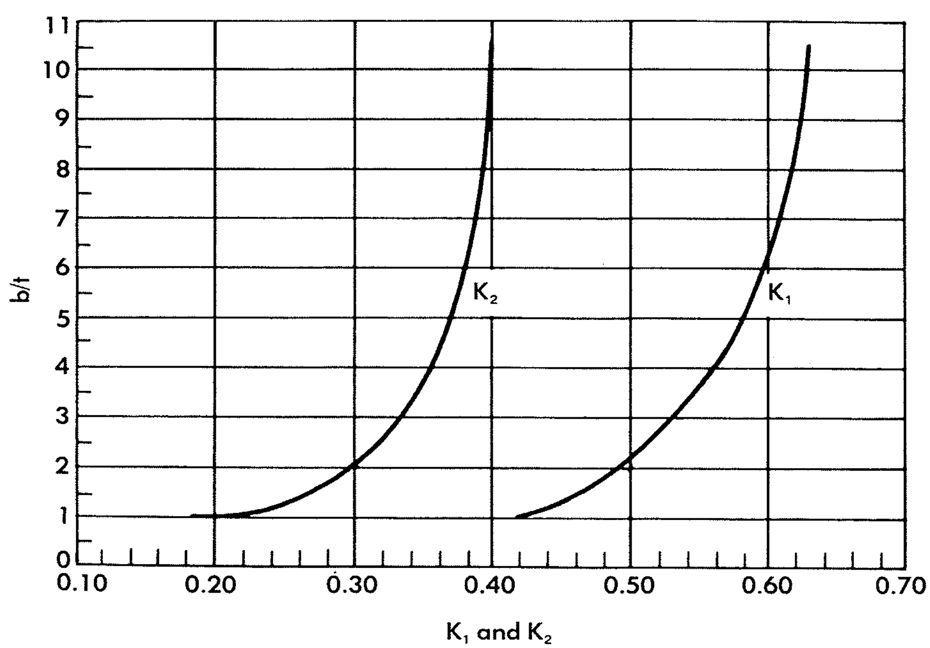


FIG. 24-13 Constants for rectangular wire in torsion. *(Associated Spring, Barnes Group Inc.)*

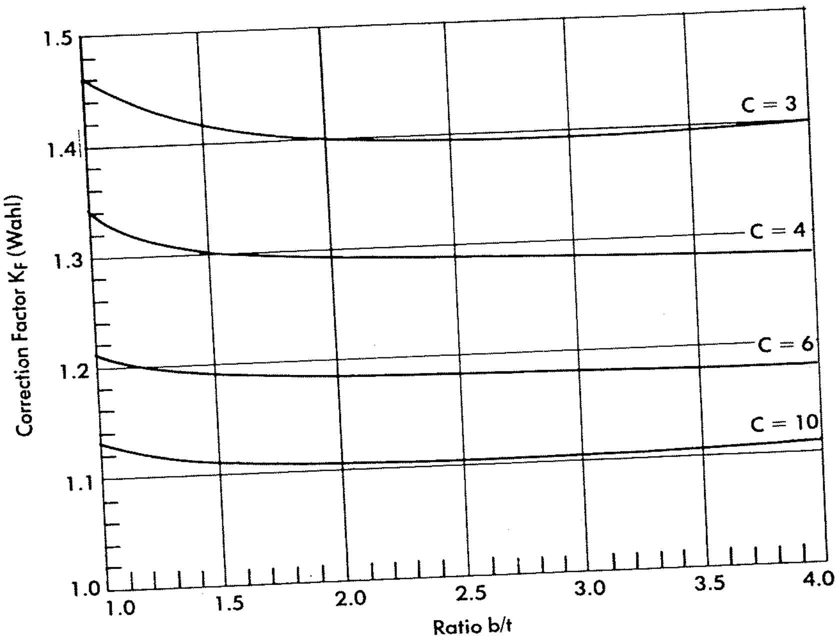


FIG. 24-14 Stress correction factors for rectangular-wire compression springs wound on flat. *(Associated Spring, Barnes Group Inc.)*

The spring rate for a rectangular-wire spring is

$$k = \frac{P}{f} = \frac{K_2 G b t^3}{N_a D^3} \tag{24-15}$$

Since the wire is loaded in torsion, it makes no difference whether the wire is wound on the flat or on edge. See Fig. 24-12.

Stress is calculated by

$$S = \frac{K_E P D}{K_1 b t^2} \quad \text{or} \quad S = \frac{K_F P D}{K_1 b t^2} \tag{24-16}$$

Values for K_1 and K_2 are found in Fig. 24-13, and those for K_E and K_F are found in Figs. 24-15 and 24-14, respectively.

When a round wire cannot be used because the solid height exceeds the specification, the approximate equivalent rectangular dimensions are found from

$$t = \frac{2d}{1 + b/t} \tag{24-17}$$

where d = round-wire diameter.

24-4-7 Variable-Diameter Springs

Conical, hourglass, and barrel-shaped springs, shown in Fig. 24-16, are used in applications requiring a low solid height and an increased lateral stability or resistance to surging. Conical springs can be designed so that each coil nests wholly or partly

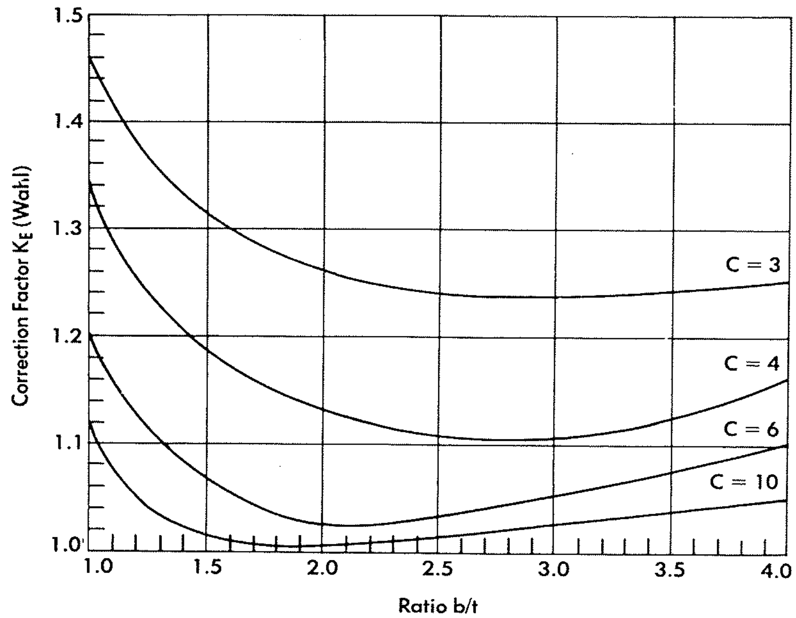


FIG. 24-15 Stress correction factors for rectangular-wire compression springs wound on edge. *(Associated Spring, Barnes Group Inc.)*

within an adjacent coil. Solid height can be as low as one wire diameter. The rate for conical springs usually increases with deflection (see Fig. 24-17) because the number of active coils decreases progressively as the spring approaches solid. By varying the pitch, conical springs can be designed to have a uniform rate. The rate for conical springs is calculated by considering the spring as many springs in series. The rate for each turn or fraction of a turn is calculated by using the standard rate equation. The rate for a complete spring is then determined, given that the spring rate follows the series relationship in Eq. (23-4).

To calculate the highest stress at a given load, the mean diameter of the largest active coil at load is used. The solid height of a uniformly tapered, but not telescoping, spring with squared and ground ends made from round wire can be estimated from

$$L_S = N_a\sqrt{d^2 - u^2} + 2d \quad (24\text{-}18)$$

where u = OD of large end minus OD of small end, divided by $2N_a$.

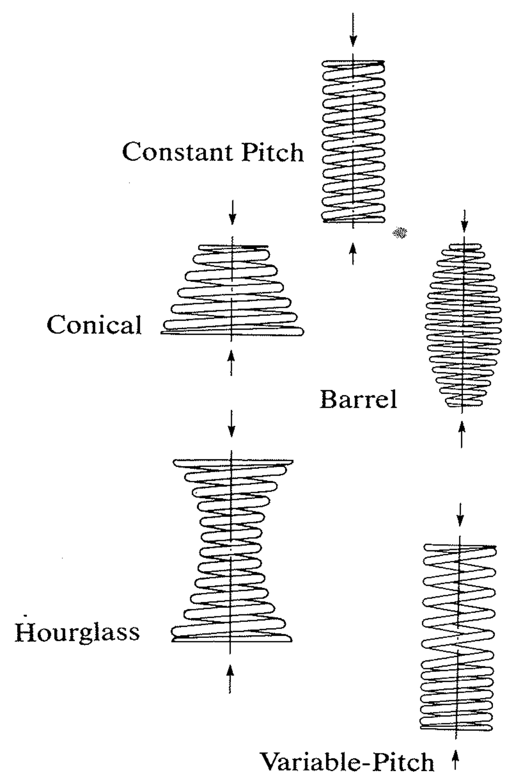


FIG. 24-16 Various compression-spring body shapes. *(Associated Spring, Barnes Group Inc.)*

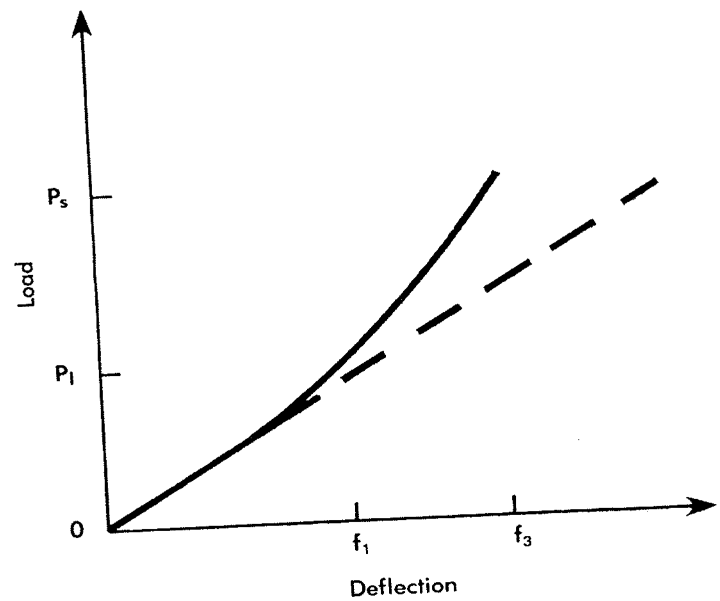


FIG. 24-17 Typical load-deflection curve for variable-diameter springs (solid line). (*Associated Spring, Barnes Group Inc.*)

Barrel- and hourglass-shaped springs are calculated as two conical springs in series.

24-4-8 Commercial Tolerances

Standard commercial tolerances are presented in Tables 24-8, 24-9, and 24-10 for free length, coil diameter, and load tolerances, respectively. These tolerances represent the best trade-offs between manufacturing costs and performance.

24-5 HELICAL EXTENSION SPRINGS

24-5-1 General

Helical extension springs store energy and exert a pulling force. They are usually made from round wire and are close-wound with initial tension. They have various types of end hooks or loops by which they are attached to the loads.

Like compression springs, extension springs are stressed in torsion in the body coils. The design procedures for the body coil are similar to those discussed in Sec. 24-4 except for the initial tension and the hook stresses.

Most extension springs are made with the body coils held tightly together by a force called *initial tension*. The measure of initial tension is the load required to overcome the internal force and start coil separation.

Extension springs, unlike compression springs, seldom have set removed. Furthermore, they have no solid stop to prevent overloading. For these reasons, the design stresses are normally held to lower values than those for compression springs.

TABLE 24-8 Free-Length Tolerances of Squared and Ground Helical Compression Springs

Number of Active coils per mm(in.)	Tolerances: ±mm/mm (in./in.) of Free Length						
	Spring Index (D/d)						
	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 (4)	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 (16)	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″) long, use the tolerances for 12.7 mm (0.500″). For closed ends not ground, multiply above values by 1.7.

SOURCE: Associated Spring, Barnes Group Inc.

TABLE 24-9 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 (0.015)	0.05 (0.002)	0.05 (0.002)	0.08 (0.003)	0.10 (0.004)	0.13 (0.005)	0.15 (0.006)	0.18 (0.007)
0.58 (0.023)	0.05 (0.002)	0.08 (0.003)	0.10 (0.004)	0.15 (0.006)	0.18 (0.007)	0.20 (0.008)	0.25 (0.010)
0.89 (0.035)	0.05 (0.002)	0.10 (0.004)	0.15 (0.006)	0.18 (0.007)	0.23 (0.009)	0.28 (0.011)	0.33 (0.013)
1.30 (0.051)	0.08 (0.003)	0.13 (0.005)	0.18 (0.007)	0.25 (0.010)	0.30 (0.012)	0.38 (0.015)	0.43 (0.017)
1.93 (0.076)	0.10 (0.004)	0.18 (0.007)	0.25 (0.010)	0.33 (0.013)	0.41 (0.016)	0.48 (0.019)	0.53 (0.021)
2.90 (0.114)	0.15 (0.006)	0.23 (0.009)	0.33 (0.013)	0.46 (0.018)	0.53 (0.021)	0.64 (0.025)	0.74 (0.029)
4.34 (0.171)	0.20 (0.008)	0.30 (0.012)	0.43 (0.017)	0.58 (0.023)	0.71 (0.028)	0.84 (0.033)	0.97 (0.038)
6.35 (0.250)	0.28 (0.011)	0.38 (0.015)	0.53 (0.021)	0.71 (0.028)	0.90 (0.035)	1.07 (0.042)	1.24 (0.049)
9.53 (0.375)	0.41 (0.016)	0.51 (0.020)	0.66 (0.026)	0.94 (0.037)	1.17 (0.046)	1.37 (0.054)	1.63 (0.064)
12.70 (0.500)	0.53 (0.021)	0.76 (0.030)	1.02 (0.040)	1.57 (0.062)	2.03 (0.080)	2.54 (0.100)	3.18 (0.125)

SOURCE: Associated Spring, Barnes Group Inc.

TABLE 24-10 Load Tolerances of Helical Compression Springs

Length Tolerance ± mm (in.)	Tolerances: ±% of Load. Start with Tolerance from Table 24-8 Multiplied by L_F. Deflection from Free Length to Load, 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.

SOURCE: Associated Spring, Barnes Group Inc.

The pulling force exerted by an extension spring is transmitted to the body coils through hooks or loops. Careful attention must be given to the stresses in the hooks. The hook ends must be free of damaging tool marks so that spring performance will not be limited by hook failure.

24-5-2 Initial Tension

Initial tension is illustrated in Fig. 24-18. The point of intersection on the ordinate is initial tension P_I. The amount of initial tension is governed by the spring index, material, method of manufacture, and the post stress-relief heat treatment temperature. Note that a high stress-relief temperature can reduce the initial tension. This is sometimes used as a means to control initial tension in low-stress, low-index springs. It follows that an extension spring requiring no initial tension can be made either by removing the initial tension with heat treatment or by keeping the coils open during coiling. The levels of initial tension obtainable are shown in Fig. 24-19.

24-5-3 Types of Ends

Extension springs require a means of attachment to the system which is to be loaded. A variety of end configurations have been developed over the years. The configurations most commonly used are shown in Fig. 24-20. Loops or hooks longer than recommended will require special setup and are more expensive. Specifying an angular relationship for the loops may also add to the cost. Allow a random relationship of loops whenever possible.

Stresses in the loops are often higher than those in the body coils. In such cases, the loops are the performance limiters, particularly in cyclic applications. Generous

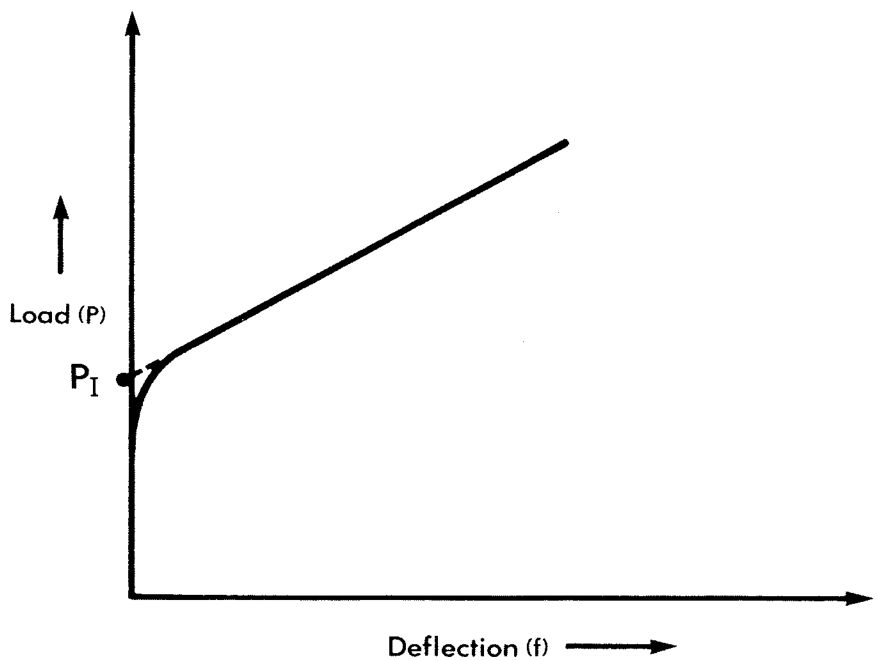


FIG. 24-18 Load-deflection curve for a helical extension spring with initial tension. *(Associated Spring, Barnes Group Inc.)*

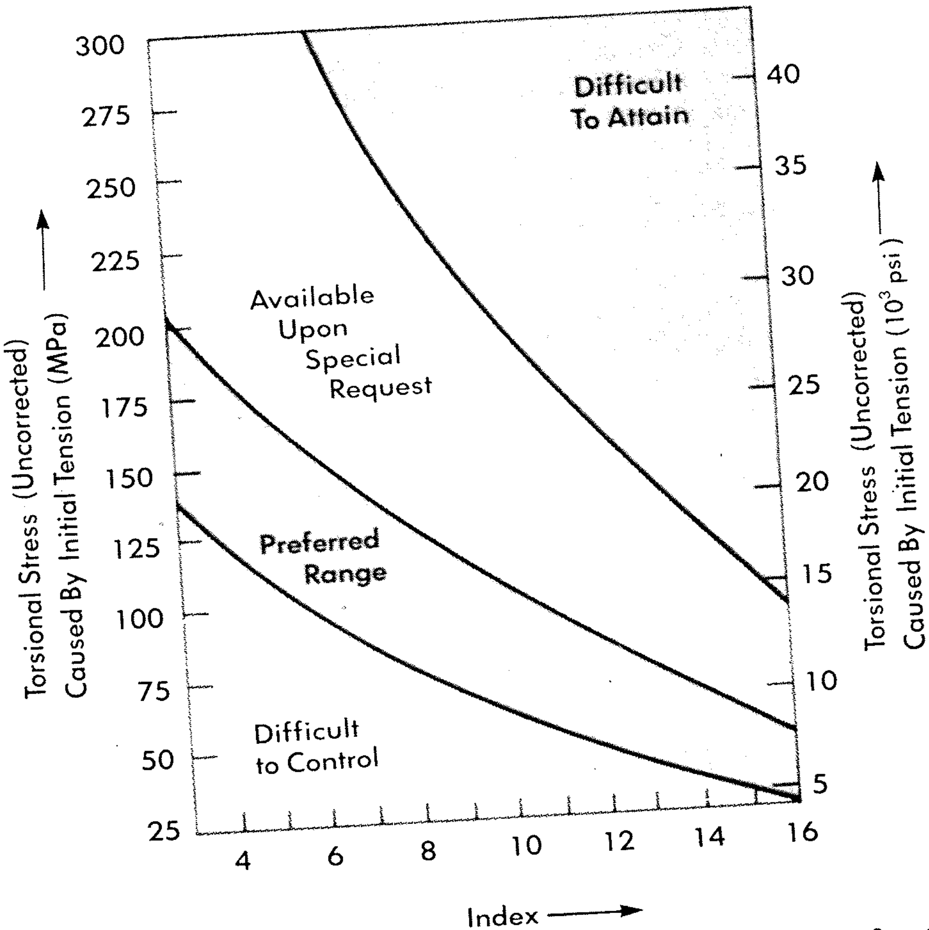


FIG. 24-19 Torsional stress resulting from initial tension as a function of index in helical extension springs. *(Associated Spring, Barnes Group Inc.)*

bend radii, elimination of tool marks, and a reduced diameter of end coils are methods used to reduce loop stresses. In a full-twist loop, stress reaches a maximum in bending at point A (Fig. 24-21) and a maximum in torsion at point B. The stresses at these locations are complex, but useful approximations are, for bending,

$$S_A = \frac{16K_1DP}{\pi d^3} + \frac{4P}{\pi d^2} \qquad (24\text{-}19)$$

where the constants are

$$K_1 = \frac{4C^2 - C_1 - 1}{4C_1(C_1 - 1)} \qquad (24\text{-}20)$$

and

$$C_1 = \frac{2R_1}{d} \qquad (24\text{-}21)$$

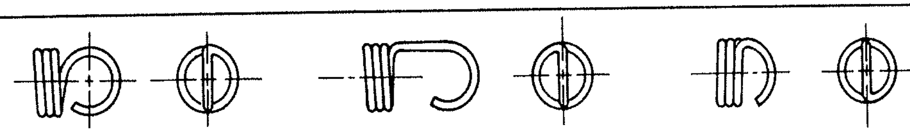
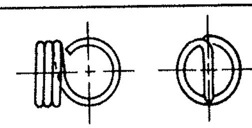
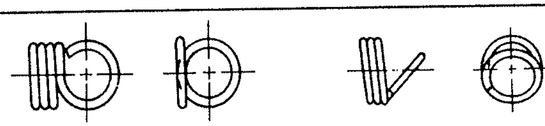
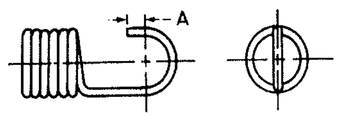
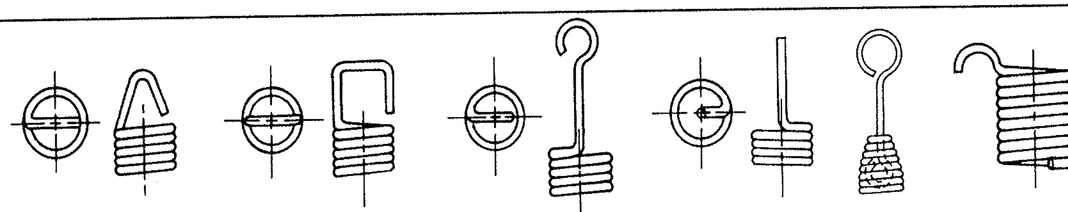
Type	Configurations	Recommended Length Min.-Max.
Twist Loop or Hook		0.5-1.7 I.D.
Cross Center Loop or Hook		I.D.
Side Loop or Hook		0.9-1.0 I.D.
Extended Hook		1.1 I.D. and up, as required by design
Special Ends		As required by design

FIG. 24-20 Common end configurations for helical extension springs. Recommended length is distance from last body coil to inside of end. ID is inside diameter of adjacent coil in spring body. *(Associated Spring, Barnes Group Inc.)*

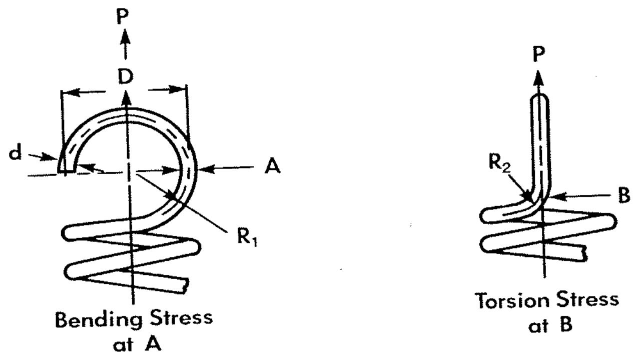


FIG. 24-21 Location of maximum bending and torsional stresses in twist loops. *(Associated Spring, Barnes Group Inc.)*

The torsional stresses are

$$S_B = \frac{8DP}{\pi d^3}\frac{4C_2 - 1}{4C_2 - 4} \tag{24-22}$$

where

$$C_2 = \frac{2R_2}{d} \tag{24-23}$$

General practice is to make C_2 greater than 4.

24-5-4 Extension Spring Dimensioning

The dimensioning shown in Fig. 24-22 is generally accepted for extension springs. The free length is the distance between the inside surfaces of the loops. The body length is $L_B = d(N + 1)$. The loop opening, or gap, can be varied. The number of active coils is equal to the number of coils in the body of the spring. However, with special ends such as threaded plugs or swivel hooks, the number of active coils will be less than the number of body coils.

24-5-5 Design Equations

The design equations are similar to those for compression springs with the exception of initial tension and loop stresses. The rate is given by

$$k = \frac{P - P_I}{f} = \frac{Gd^4}{8D^3N_a} \tag{24-24}$$

where P_I is initial tension. Stress is given by

$$S = \frac{K_w 8PD}{\pi d^3} \tag{24-25}$$

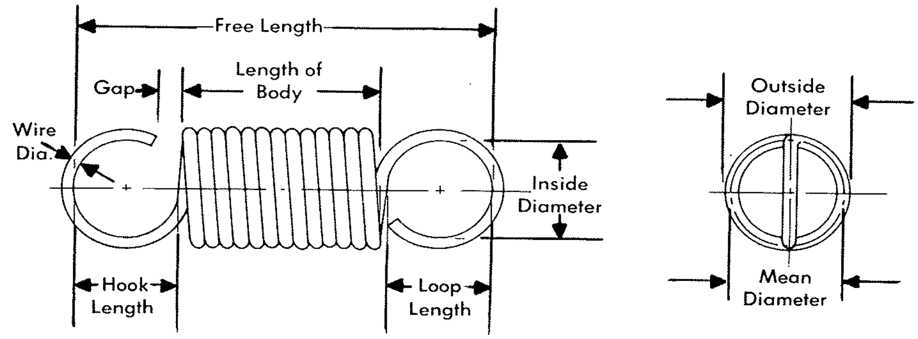


FIG. 24-22 Typical extension-spring dimensions. *(Associated Spring, Barnes Group Inc.)*

Dynamic considerations discussed previously are generally applicable to extension springs. Natural frequency with one end fixed, in SI units, is

$$n = \frac{560d}{D^2N_a}\sqrt{\frac{Gg}{\rho}} \tag{24-26}$$

For steel, this equation becomes

$$n = \frac{176\,000d}{N_aD^2} \tag{24-27}$$

where n = frequency in hertz. The corresponding equation in USCS units is

$$n = \frac{d}{18D^2N_a}\sqrt{\frac{Gg}{\rho}} \tag{24-28}$$

And for steel we have

$$n = \frac{7000d}{N_aD^2} \tag{24-29}$$

24-5-6 Choice of Operating Stress—Static

The maximum stresses recommended for extension springs in static applications are given in Table 24-11. Note that extension springs are similar to compression springs without set removed. For body coil stresses in springs that cannot be adequately stress-relieved because of very high initial-tension requirements, use the maximum recommended stress in torsion, given for the end loops.

24-5-7 Choice of Operating Stress—Cyclic

Table 24-12 presents the maximum stresses for extension springs used in cyclic applications. The data are for stress-relieved springs with initial tension in the preferred range.

TABLE 24-11 Maximum Allowable Stresses (K_{w_1} Corrected) for Helical Extension Springs in Static Applications

Materials	Percent of Tensile Strength		
	In Torsion		In Bending
	Body	End	End
Patented, cold-drawn or hardened and tempered carbon and low alloy steels	45–50	40	75
Austenitic stainless steel and nonferrous alloys	35	30	55

This information is based on the following conditions: set not removed and low temperature heat treatment applied.
For springs that require high initial tension, use the same percent of tensile strength as for end.

SOURCE: Associated Spring, Barnes Group Inc.

TABLE 24-12 Maximum Allowable Stresses for ASTM A228 and Type 302 Stainless-Steel Helical Extension Springs in Cyclic Applications

Number of Cycles	Percent of Tensile Strength		
	In Torsion		In Bending
	Body	End	End
10^5	36	34	51
10^6	33	30	47
10^7	30	28	45

This information is based on the following conditions: not shot-peened, no surging and ambient environment with a low temperature heat treatment applied. Stress ratio = 0.

SOURCE: Associated Spring, Barnes Group Inc.

TABLE 24-13 Commerical Free-Length Tolerances for Helical Extension Springs with Initial Tension

Spring Free Length (inside hooks) mm (in.)	Tolerance ± mm (in.)
Up to 12.7 (0.500)	0.51 (0.020)
Over 12.7 to 25.4 (0.500 to 1.00)	0.76 (0.030)
Over 25.4 to 50.8 (1.00 to 2.00)	1.0 (0.040)
Over 50.8 to 102 (2.00 to 4.00)	1.5 (0.060)
Over 102 to 203 (4.00 to 8.00)	2.4 (0.093)
Over 203 to 406 (8.00 to 16.0)	4.0 (0.156)
Over 406 to 610 (16.0 to 24.0)	5.5 (0.218)

SOURCE: Associated Spring, Barnes Group Inc.

TABLE 24-14 Tolerances on Angular Relationship of Extension Spring Ends

Angular Tolerance per Coil: ± Degrees									
Index									
4	5	6	7	8	9	10	12	14	16
0.75	0.9	1.1	1.3	1.5	1.7	1.9	2.3	2.6	3

For example, tolerance for a 10-coil spring with an index of 8 is $10 \times \pm 1.5 = \pm 15°$.
If angular tolerance is greater than ± 45°, or if closer tolerances than indicated must be held, consult with Associated Spring.

SOURCE: Associated Spring, Barnes Group Inc.

24-5-8 Tolerances

Extension springs do not buckle or require guide pins when they are deflected, but they may vibrate laterally if loaded or unloaded suddenly. Clearance should be allowed in these cases to eliminate the potential for noise or premature failure. The load tolerances are the same as those given for compression springs. Tolerances for free length and for angular relationship of ends are given in Tables 24-13 and 24-14.

24-6 HELICAL TORSION SPRINGS

Helical springs that exert a torque or store rotational energy are known as *torsion springs.* The most frequently used configuration of a torsion spring is the single-body type (Fig. 24-23). Double-bodied springs, known as double-torsion springs, are sometimes used where dictated by restrictive torque, stress, and space requirements. It is often less costly to make a pair of single-torsion springs than a double-torsion type.

Torsion springs are used in spring-loaded hinges, oven doors, clothes pins, window shades, ratchets, counterbalances, cameras, door locks, door checks, and many other applications. Torsion springs are almost always mounted on a shaft or arbor with one end fixed. They can be wound either right or left hand.

In most cases the springs are not stress-relieved and are loaded in the direction that winds them up or causes a decrease in body diameter. The residual forming stresses which remain are favorable in that direction. Although it is possible to load a torsion spring in the direction to unwind and enlarge the body coils, ordinarily it is not good design practice and should be avoided. Residual stresses in the unwind direction are unfavorable. Torsion springs which are plated or painted and subsequently baked or are stress-relieved will have essentially no residual stresses and can be loaded in either direction, but at lower stress levels than springs which are not heat-treated.

Correlation of test results between manufacturer and user may be difficult because there are few, if any, standardized torsion-spring testing machines. The springs will have varying degrees of intercoil friction and friction between the mounting arbor and the body coils. Often duplicate test fixtures must be made and test methods coordinated.

Spring ends most commonly used are shown in Fig. 24-24, although the possible variations are unlimited. In considering spring mounting, it must be recognized that for each turn of windup, the overall length L of the spring body will increase as

$$L_1 = d(N_a + 1 + \theta) \qquad (24\text{-}30)$$

where θ = deflection in revolutions.

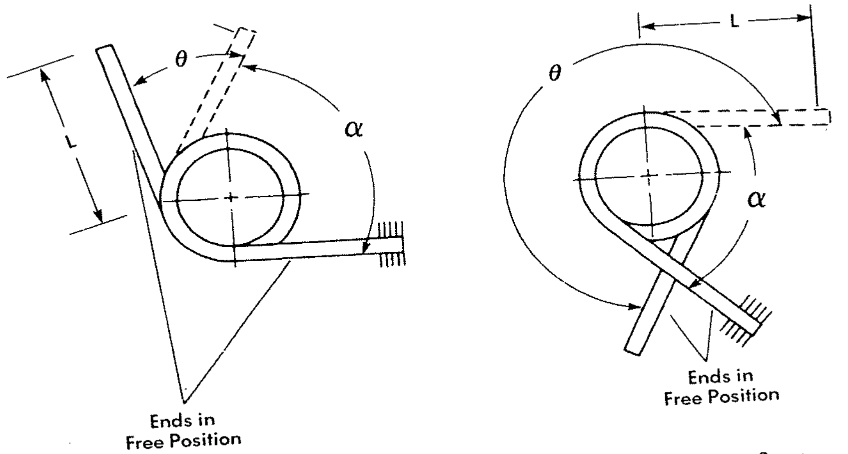


FIG. 24-23 Specifying load and deflection requirements for torsion spring: α = angle between ends; P = load on ends at α; L = moment arm; θ = angular deflection from free position. *(Associated Spring, Barnes Group Inc.)*

Also note that the body coil diameter will be reduced to

$$D = \frac{D_I N_a}{Na + \theta} \tag{24-31}$$

where D_I = initial mean coil diameter. Experience indicates that the diameter of the arbor over which the spring operates should be approximately 90 percent of the

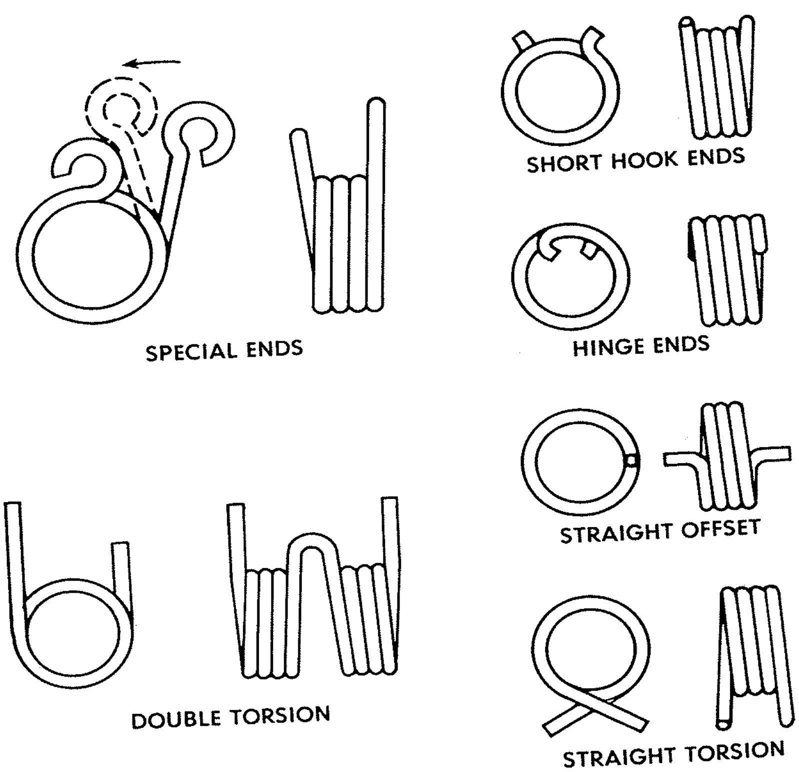


FIG. 24-24 Common helical torsion-spring end configurations. *(Associated Spring, Barnes Group Inc.)*

smallest inside diameter to which the spring is reduced under maximum load. Too large an arbor will interfere with deflection, while too small an arbor will provide too little support. Both conditions lead to unexpectedly early failure. Coil diameter tolerances are given in Table 24-17.

24-6-1 Spring Rate

The spring rate, or moment per turn, is given by

$$k = \frac{M}{\theta} = \frac{Ed^4}{10.8DN_a} \tag{24-32}$$

The number of coils is equal to the number of body coils plus a contribution from the ends. The effect is more pronounced when the ends are long. The number of equivalent coils in the ends is

$$N_e = \frac{L_1 + L_2}{3\pi D} \tag{24-33}$$

where L_1 and L_2 = lengths of ends, so $N_a = N_b + N_e$, where N_b = number of body coils.

The load should be specified at a fixed angular relationship of the spring ends rather than at a specific angular deflection from free or load positions. Helical torsion springs are stressed in bending. Rectangular sections are more efficient than round sections, but round sections are normally used because there is usually a premium cost for rectangular wire.

24-6-2 Stresses

Stress in round-wire torsion springs is given by

$$S = \frac{32K_BM}{\pi d^3} \tag{24-34}$$

where K_B = a stress correction factor. Stress is higher on the inner surface of the coil. A useful approximation of this factor is

$$K_B = \frac{4C - 1}{4C - 4} \tag{24-35}$$

24-6-3 Rectangular-Wire Torsion Springs

When rectangular wire is formed into coils, it approaches a keystone according to the relation

$$b_I = b\frac{C + 0.5}{C} \tag{24-36}$$

where b_I = axial dimension b after keystoning. The radial dimension is always t.

TABLE 24-15 Maximum Recommended Bending Stresses for Helical Torsion Springs in Static Applications

Material	Percent of Tensile Strength	
	Stress-Relieved (1) (K_B Corrected)	With Favorable Residual Stress (2) (No Correction Factor)
Patented and Cold Drawn	80	100
Hardened and Tempered Carbon and Low Alloy Steels	85	100
Austenitic Stainless Steels and Non-Ferrous Alloys	60	80

(1) Also for springs without residual stresses.
(2) Springs that have not been stress-relieved and which have bodies and ends loaded in a direction that decreases the radius of curvature.

SOURCE: Associated Spring, Barnes Group Inc.

The rate equation is

$$k = \frac{M}{\theta} = \frac{Ebt^3}{6.6DN_a} \tag{24-37}$$

Stress in rectangular-wire torsion springs is given by

$$S = \frac{6K_BM}{bt^2} \tag{24-38}$$

where $K_{B_{ID}} = 4C/(4C - 3)$ and b = axial dimension of rectangular cross section. Maximum recommended stresses are given in Table 24-15 for static applications and in Table 24-16 for cyclic applications.

24-6-4 Tolerances

The tolerances for coil diameter and end position are given in Tables 24-17 and 24-18, respectively. Use them as guides.

TABLE 24-16 Maximum Recommended Bending Stresses (K_B Corrected) for Helical Torsion Springs in Cyclic Applications

Fatigue Life (cycles)	ASTM A228 and Type 302 Stainless Steel		ASTM A230 and A232	
	Not Shot-Peened	Shot-Peened*	Not Shot-Peened	Shot-Peened*
10^5	53	62	55	64
10^6	50	60	53	62

This information is based on the following conditions: no surging, springs are in the "as-stress-relieved" condition
*Not always possible.

SOURCE: Associated Spring, Barnes Group Inc.

TABLE 24-17 Commercial Tolerances for Torsion-Spring Coil Diameters

Wire Diameter mm (in.)	Tolerance: ±mm (in.) Spring Index D/d						
	4	6	8	10	12	14	16
0.38 (0.015)	0.05 (0.002)	0.05 (0.002)	0.05 (0.002)	0.05 (0.002)	0.08 (0.003)	0.08 (0.003)	0.10 (0.004)
0.58 (0.023)	0.05 (0.002)	0.05 (0.002)	0.05 (0.002)	0.08 (0.003)	0.10 (0.004)	0.13 (0.005)	0.15 (0.006)
0.89 (0.035)	0.05 (0.002)	0.05 (0.002)	0.08 (0.003)	0.10 (0.004)	0.15 (0.006)	0.18 (0.007)	0.23 (0.009)
1.30 (0.051)	0.05 (0.002)	0.08 (0.003)	0.13 (0.005)	0.18 (0.007)	0.20 (0.008)	0.25 (0.010)	0.31 (0.012)
1.93 (0.076)	0.08 (0.003)	0.13 (0.005)	0.18 (0.007)	0.23 (0.009)	0.31 (0.012)	0.38 (0.015)	0.46 (0.018)
2.90 (0.114)	0.10 (0.004)	0.18 (0.007)	0.25 (0.010)	0.33 (0.013)	0.46 (0.018)	0.56 (0.022)	0.71 (0.028)
4.37 (0.172)	0.15 (0.006)	0.25 (0.010)	0.33 (0.013)	0.51 (0.020)	0.69 (0.027)	0.86 (0.034)	1.07 (0.042)
6.35 (0.250)	0.20 (0.008)	0.36 (0.014)	0.56 (0.022)	0.76 (0.030)	1.02 (0.040)	1.27 (0.050)	1.52 (0.060)

SOURCE: Associated Spring, Barnes Group Inc.

24-7 BELLEVILLE SPRING WASHER

Belleville washers, also known as *coned-disk springs,* take their name from their inventor, Julian F. Belleville. They are essentially circular disks formed to a conical shape, as shown in Fig. 24-25. When load is applied, the disk tends to flatten. This elastic deformation constitutes the spring action.

Belleville springs are used in two broad types of applications. First, they are used to provide very high loads with small deflections, as in stripper springs for punch-press dies, recoil mechanisms, and pressure-relief valves. Second, they are used for their special nonlinear load-deflection curves, particularly those with a constant-load portion. In loading a packing seal or a live center for a lathe, or in injection molding machines, Belleville washers can maintain a constant force throughout dimensional changes in the mechanical system resulting from wear, relaxation, or thermal change.

TABLE 24-18 End-Position Tolerances (for D/d Ratios up to and Including 16)

Total Coils	Tolerance: ± Degrees*
Up to 3	8
Over 3–10	10
Over 10–20	15
Over 20–30	20
Over 30	25

*Closer tolerances available

SOURCE: Associated Spring, Barnes Group Inc.

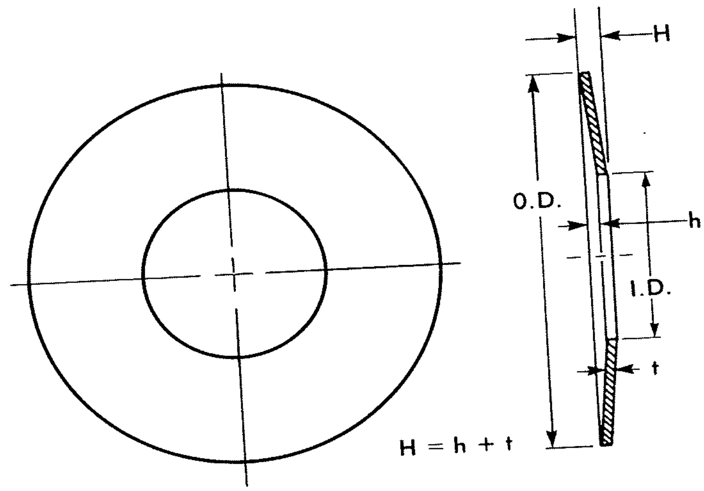


FIG. 24-25 Belleville washer. *(Associated Spring, Barnes Group Inc.)*

The two types of performance depend on the ratio of height to thickness. Typical load-deflection curves for various height-thickness ratios are shown in Fig. 24-26. Note that the curve for a small h/t ratio is nearly a straight line. At $h/t = 1.41$ the curve shows a nearly constant load for approximately the last 50 percent of deflection before the flat position. Above $h/t = 1.41$ the load decreases after reaching a peak. When h/t is 2.83 or more, the load will go negative at some point beyond flat and will require some force to be restored to its free position. In other words, the washer will turn inside out.

The design equations given here are complex and may present a difficult challenge to the occasional designer. Use of charts and the equation transpositions presented here has proved helpful. Note that these equations are taken from the mathematical

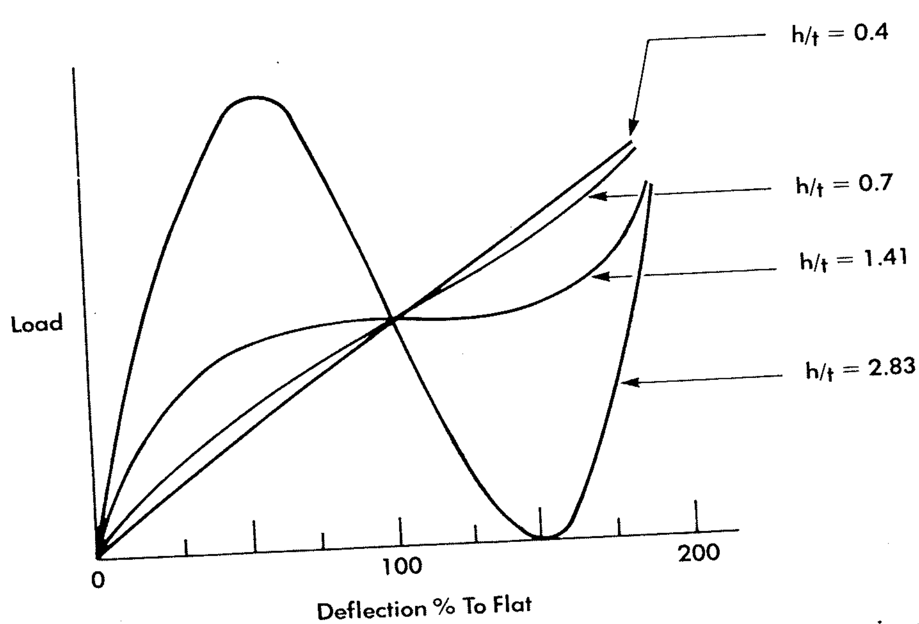


FIG. 24-26 Load-deflection curves for Belleville washers with various h/t ratios. *(Associated Spring, Barnes Group Inc.)*

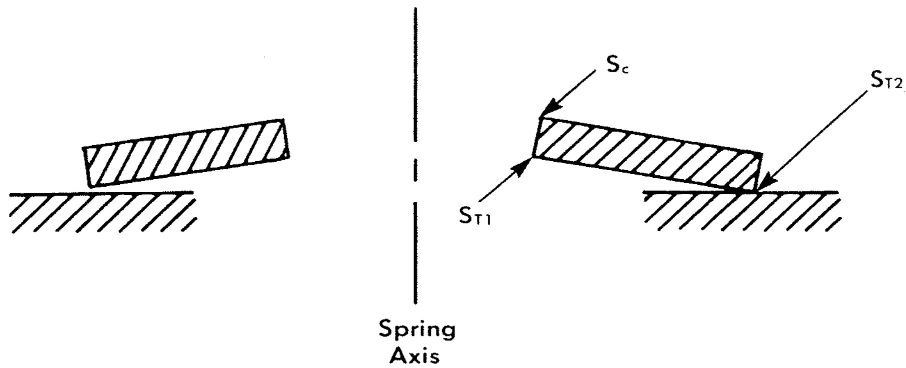


FIG. 24-27 Highest stressed regions in Belleville washers. *(Associated Spring, Barnes Group Inc.)*

analysis by Almen and Laszlo [24-5]. The symbols used here are those originally used by the authors and may not necessarily agree with those used elsewhere in the text.

24-7-1 Nomenclature

a	OD/2, mm (in)
C_1	Compressive stress constant (see formula and Fig. 24-28)
C_2	Compressive stress constant (see formula and Fig. 24-28)
E	Modulus of elasticity (see Table 24-19), MPa (psi)
f	Deflection, mm (in)
h	Inside height, mm (in)
ID	Inside diameter, mm (in)
M	Constant
OD	Outside diameter, mm (in)
P	Load, N (lb)
P_f	Load at flat position, N (lb)
R	OD/ID
S_c	Compressive stress (Fig. 24-27), MPa (psi)
S_{T_1}	Tensile stress (Fig. 24-27), MPa (psi)
S_{T_2}	Tensile stress (Fig. 24-27), MPa (psi)
t	Thickness, mm (in)
T_1	Tensile stress constant (see formula and Fig. 24-29)
T_2	Tensile stress constant (see formula and Fig. 24-29)
μ	Poisson's ratio (Table 24-19)

24-7-2 Basic Equations

$$P = \frac{Ef}{(1-\mu^2)Ma^2}\left[(h-f)\left(h-\frac{f}{2}\right)t + t^3\right] \qquad (24\text{-}39)$$

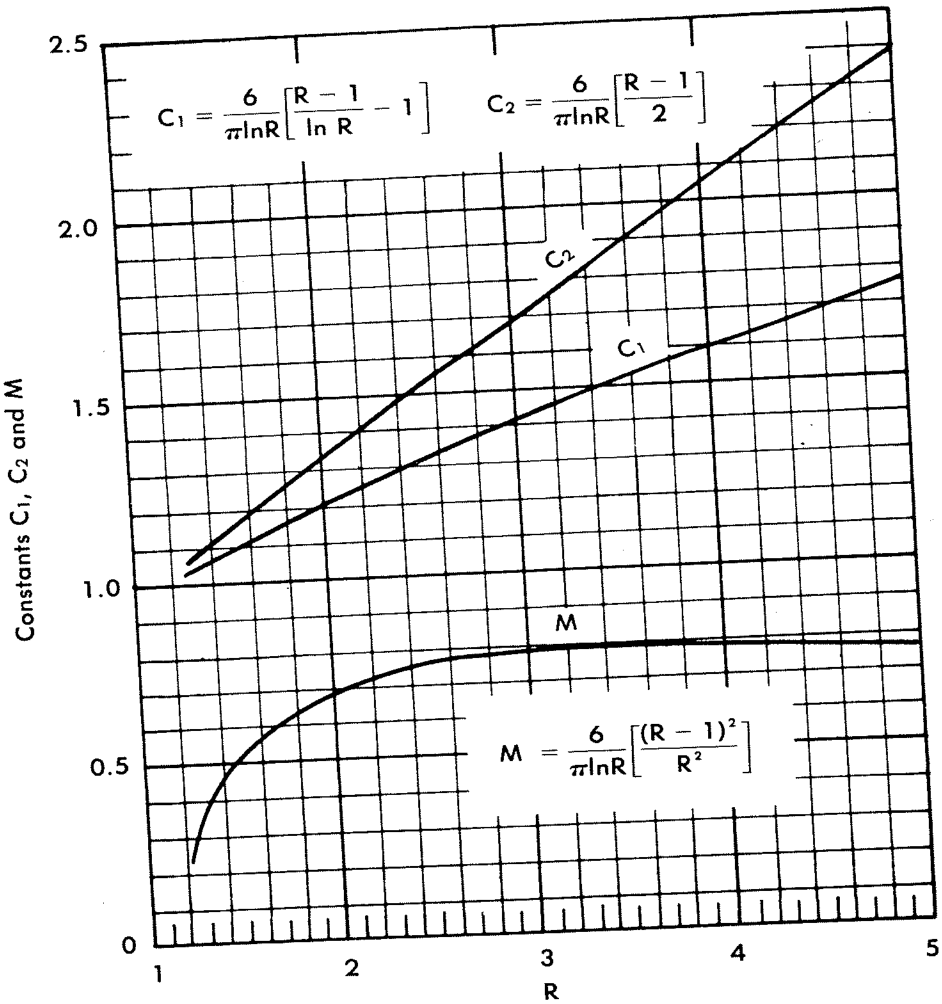


FIG. 24-28 Compressive stress constants for Belleville washers. *(Associated Spring, Barnes Group Inc.)*

TABLE 24-19 Elastic Constants of Common Spring Materials

Material	Modulus of Elasticity E		Poisson's ratio μ
	Mpsi	GPa	
Steel	30	207	0.30
Phosphor bronze	15	103	0.20
17-7 PH stainless	29	200	0.34
302 stainless	28	193	0.30
Beryllium copper	18.5	128	0.33
Inconel	31	214	0.29
Inconel X	31	214	0.29

SOURCE: Associated Spring, Barnes Group Inc.

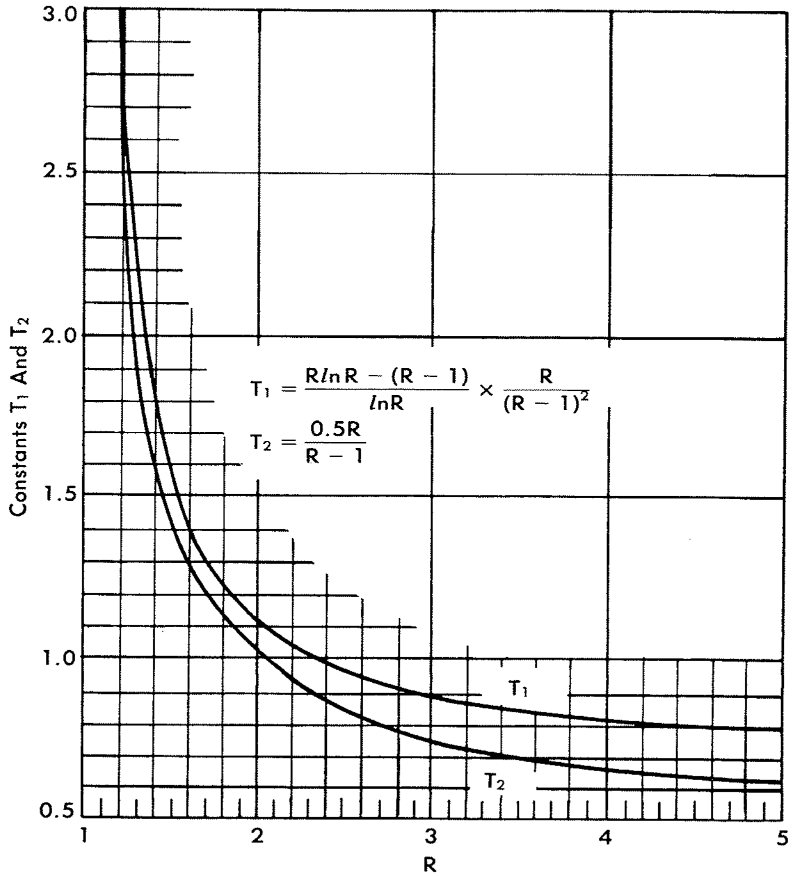


FIG. 24-29 Tensile stress constants for Belleville washers. *(Associated Spring, Barnes Group Inc.)*

$$P_F = \frac{Eht^3}{(1-\mu^2)Ma^2} \tag{24-40}$$

$$S_c = \frac{Ef}{(1-\mu^2)Ma^2}\left[C_1\left(h - \frac{f}{2}\right) + C_2 t\right] \tag{24-41}$$

$$S_{T_1} = \frac{Ef}{(1-\mu^2)Ma^2}\left[C_1\left(h - \frac{f}{2}\right) - C_2 t\right] \tag{24-42}$$

$$S_{T_2} = \frac{Ef}{(1-\mu^2)a^2}\left[T_1\left(h - \frac{f}{2}\right) + T_2 t\right] \tag{24-43}$$

The design approach recommended here depends on first determining the loads and stresses at flat position, as shown in Fig. 24-30. Intermediate loads are determined from the curves in Fig. 24-31.

Figure 24-30 gives the values graphically for compressive stresses S_c at flat position. The stress at intermediate stages is approximately proportional to the deflection. For critical applications involving close tolerances or unusual proportions, stresses should be checked by using the equation before the design is finalized.

The stress level for static applications is evaluated in accordance with Eq. (24-41). This equation has been used most commonly for appraising the design of a Belleville

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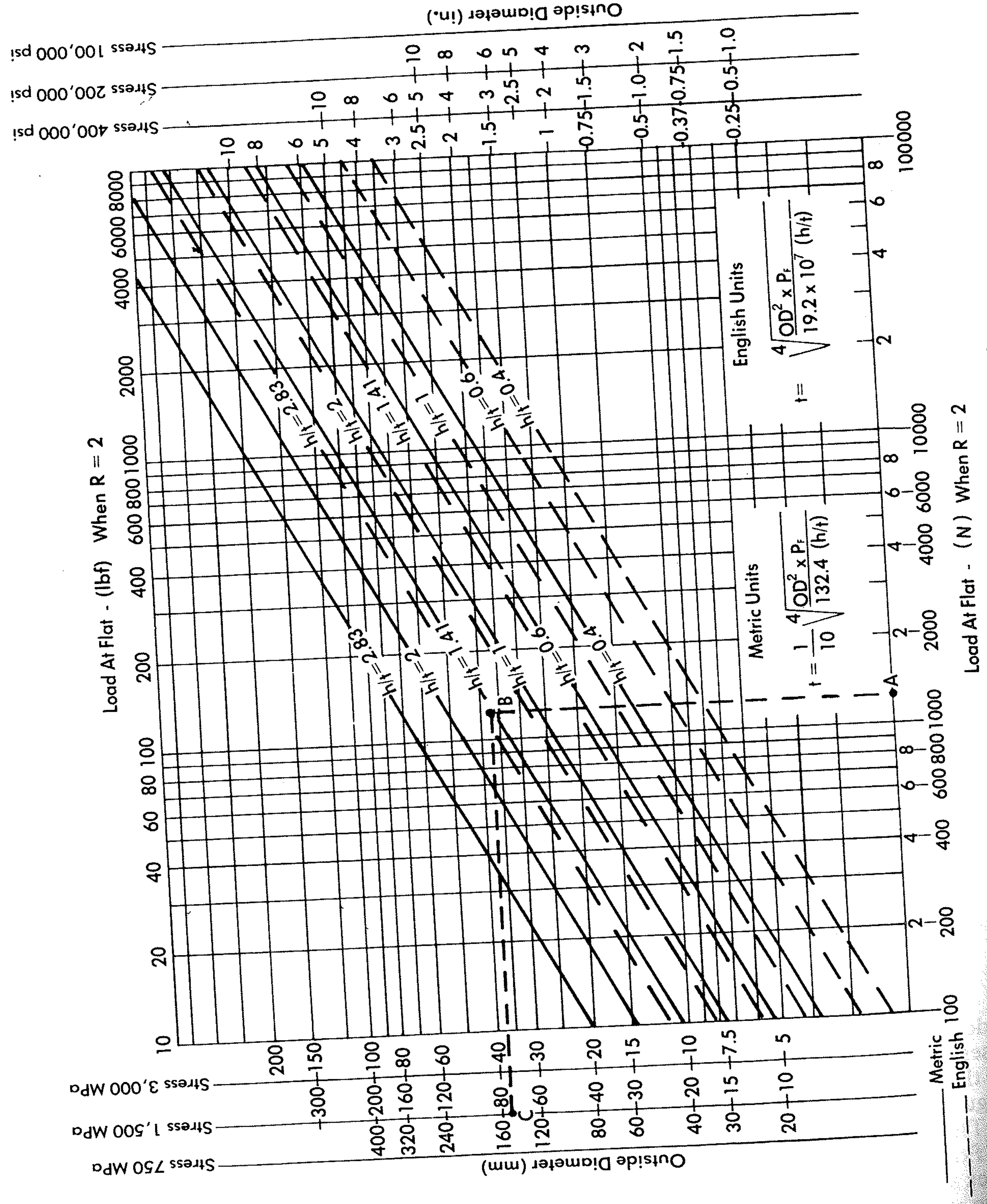

FIG. 24-30 Loads and compressive stresses S_c for Belleville washers wth various outside diameters and h/t ratios. (*Associated Spring, Barnes Group Inc.*)

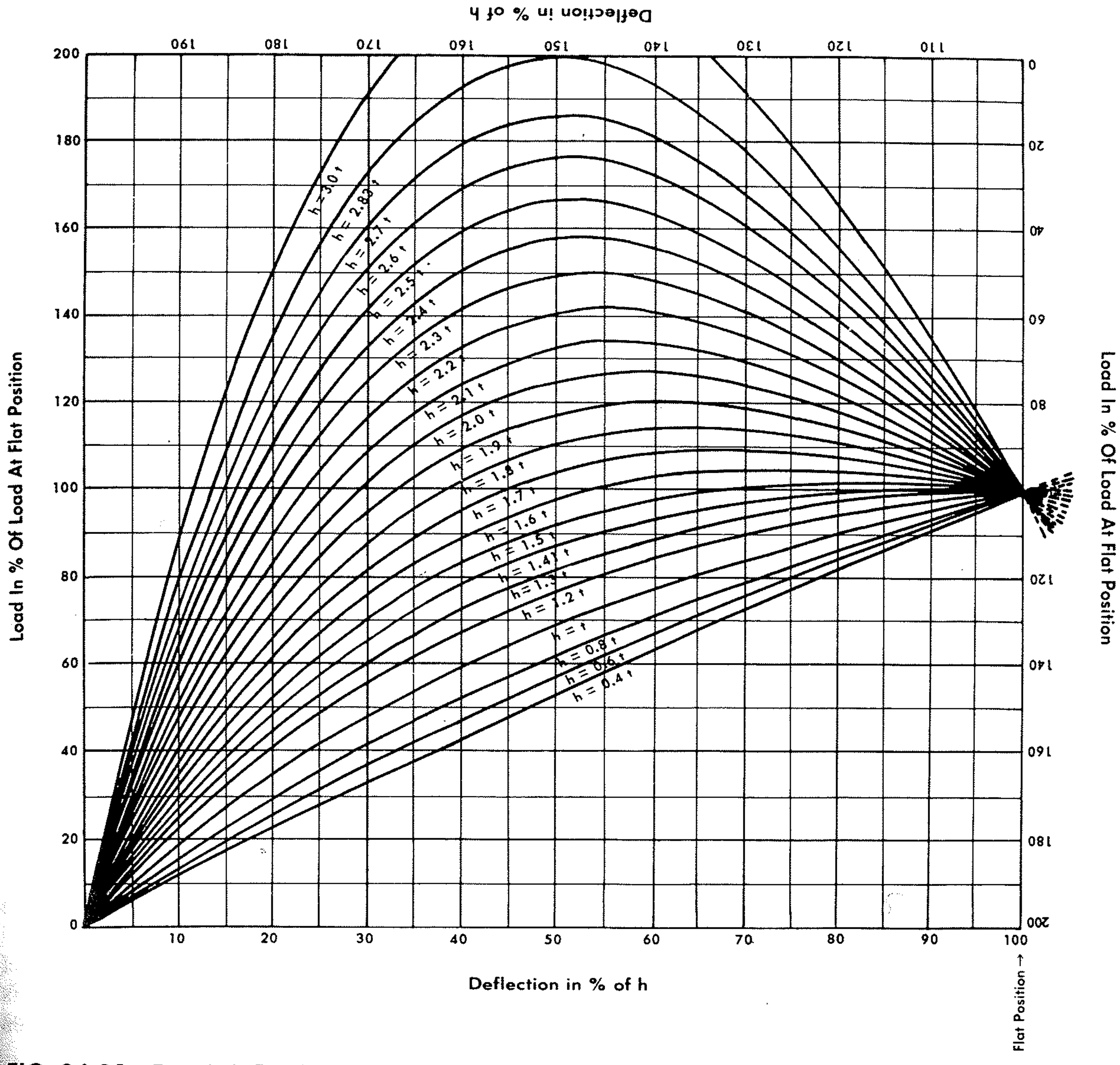

FIG. 24-31 Load-deflection characteristics for Belleville washers. If a washer is supported and loaded at its edges so that it is deflected beyond the flat position, then the greatest possible deflection can be utilized. Since the load-deflection curve beyond the horizontal position is symmetric with the first part of the curve, this chart has been labeled at the right and top to be read upside down for deflection beyond horizontal. Dotted lines extending beyond the chart indicate continuation of curves beyond flat. *(Associated Spring, Barnes Group Inc.)*

spring because it gives the highest numerical value. It gives the compressive stress at the point shown in Fig. 24-27.

A Belleville spring washer should be designed so that it can be compressed flat by accidental overloading, without setting. This can be accomplished either by using a stress so low that the spring will not set or by forming the spring higher than the design height and removing set by compressing flat or beyond flat (see Table 24-21). The table values should be reduced if the washers are plated or used at elevated temperatures.

For fatigue applications it is necessary to consider the tensile stresses at the points marked S_{T_1} and S_{T_2} in Fig. 24-27. The higher value of the two can occur at either the

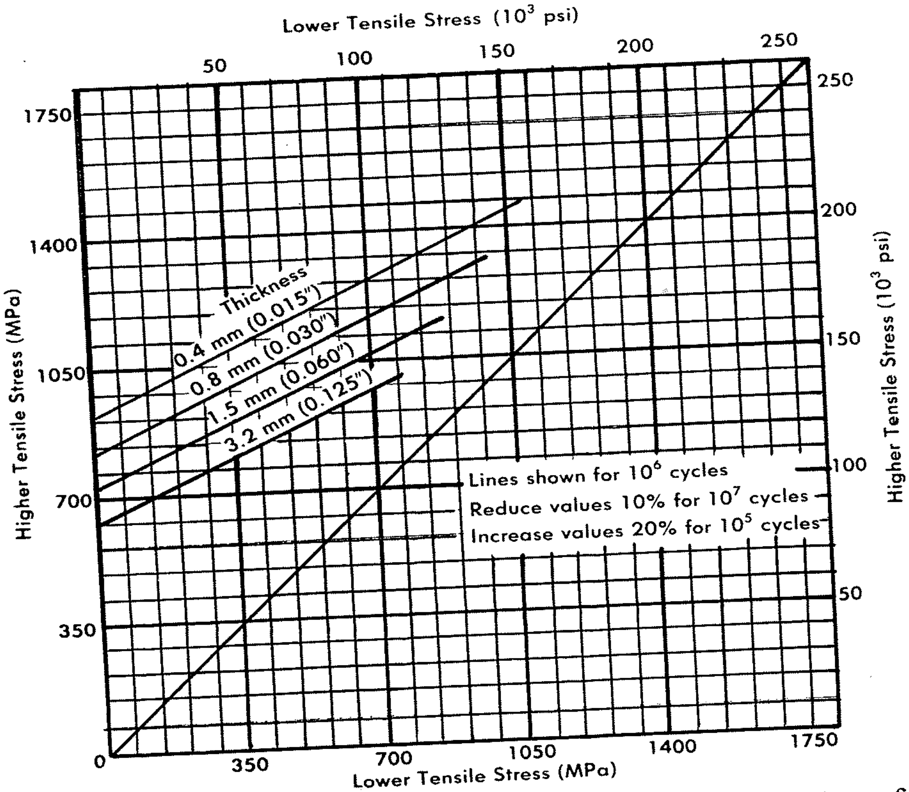

FIG. 24-32 Modified Goodman diagram for Belleville washers; for carbon and alloy steels at 47 to 49 R_c with set removed, but not shot-peened. *(Associated Spring, Barnes Group Inc.)*

ID or OD, depending on the proportions of the spring. Therefore, it is necessary to compute both values.

Fatigue life depends on the stress range as well as the maximum stress value. Figure 24-32 predicts the endurance limits based on either S_{T_1} or S_{T_2}, whichever is higher. Fatigue life is adversely affected by surface imperfections and edge fractures and can be improved by shot peening.

Since the deflection in a single Belleville washer is relatively small, it is often necessary to combine a number of washers. Such a combination is called a *stack*.

The deflection of a series stack (Fig. 24-33) is equal to the number of washers times the deflection of one washer, and the load of the stack is equal to that of one washer. The load of a parallel stack is equal to the load of one washer times the number of washers, and the deflection of the stack is that of one washer.

Because of production variations in washer parameters both the foregoing state-

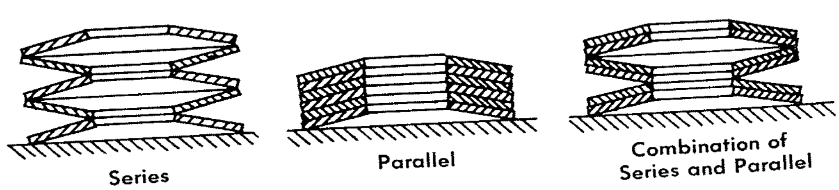

FIG. 24-33 Stacks of Belleville washers. *(Associated Spring, Barnes Group Inc.)*

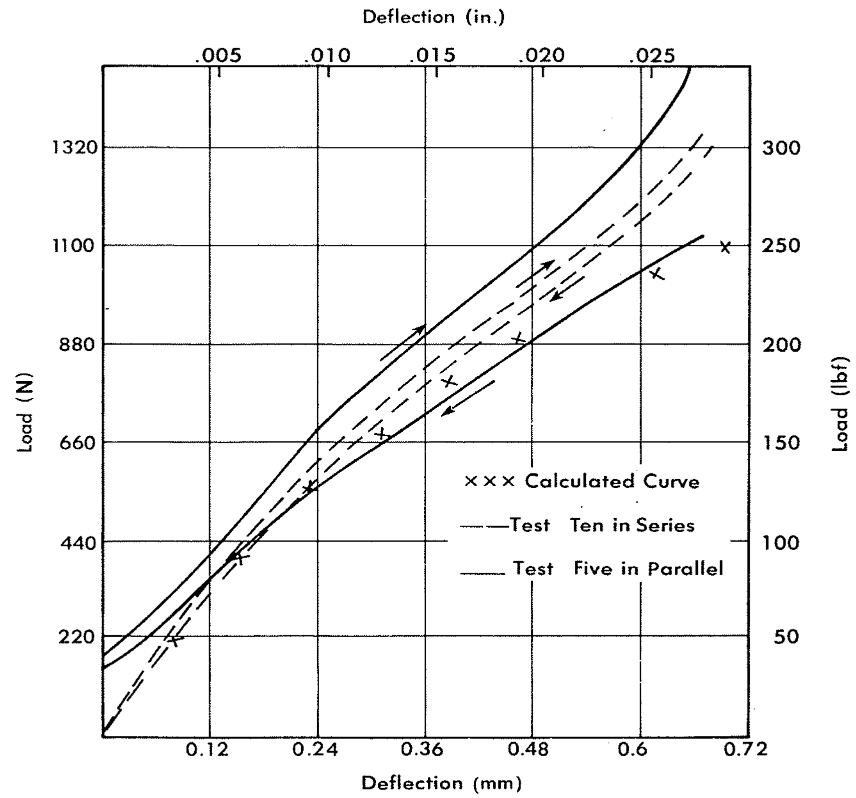

FIG. 24-34 Hysteresis in stacked Belleville washers. *(Associated Spring, Barnes Group Inc.)*

ments carry cautionary notes. In the series stack, springs of the constant-load type ($h/t = 1.41$) may actually have a negative rate in some portion of their deflection range. When such a series stack is deflected, some washers will snap through, producing jumps in the load-deflection curve. To avoid this problem, the h/t ratio in a series stack design should not exceed 1.3.

In the parallel stack, friction between the washers causes a hysteresis loop in the load-deflection curve (Fig. 24-34). The width of the loop increases with each washer added to the stack but may be reduced by adding lubrication as the washers burnish each other during use.

Stacked washers normally require guide pins or sleeves to keep them in proper alignment. These guides should be hardened steel at HRC 48 minimum hardness. Clearance between the washer and the guide pin or sleeve should be about 1.5 percent of the appropriate diameter.

24-7-3 Tolerances

Load tolerances should be specified at test height. For carbon-steel washers with $h/t < 0.25$, use load tolerance of ± 15 percent. For washers with $h/t > 0.25$, use ± 10 percent. The recommended load tolerance for stainless steel and nonferrous washers is ± 15 percent. See Table 24-20 for outside- and inside-diameter tolerances.

EXAMPLE. In a clutch, a minimum pressure of 202 lb (900 N) is required. This pressure must be held nearly constant as the clutch facing wears down 0.31 in (7.9

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TABLE 24-20 Belleville Washer Diameter Tolerances

Diameter, mm (in.)	O.D. mm (in.) +0.00	I.D. mm (in.) −0.00
Up to 5 (0.197)	−0.20 (−0.008)	+0.20 (+0.008)
5–10 (0.197–0.394)	−0.25 (−0.010)	+0.25 (+0.010)
10–25 (0.394–0.984)	−0.30 (−0.012)	+0.30 (+0.012)
25–50 (0.984–1.969)	−0.40 (−0.016)	+0.40 (+0.016)
50–100 (1.969–3.937)	−0.50 (−0.020)	+0.50 (+0.020)

Based on R = 2, increased tolerances are required for lower R ratios.
SOURCE: Associated Spring, Barnes Group Inc.

mm). The washer OD is 2.99 in (76 mm). The material washer OD is 2.99 in (76 mm). The material selected for the application is spring steel HRC 47-50.

Solution

1. Base the load on a value 10 percent above the minimum load, or 202 + 10 percent = 223 lb (998 N). Assume OD/ID = 2. From Fig. 24-31, select a load-deflection curve which gives approximately constant load between 50 and 100 percent of deflection to flat. Choose the h/t = 1.41 curve.
2. From Fig. 24-31, the load at 50 percent of deflection to flat is 88 percent of the flat load.
3. Flat load is $P_F = 223/0.88 = 252$ lb (1125 N).
4. From Fig. 24-30 [follow line AB from 1125 N to h/t = 1.41 and line BC to approximately 76-mm (2.99-in) OD], the estimated stress is 1500 MPa [218 kilopounds per square inch (kpsi)].
5. From Table 24-21 maximum stress without set removed is 120 percent of tensile strength. From Fig. 24-3, the tensile strength at HRC 48 will be approximately 239 kpsi (1650 MPa). Yield point without residual stress will be (239 kpsi)(1.20) = 287 kpsi. Therefore 218 kpsi stress is less than the maximum stress of 287 kpsi.
6. Stock thickness is

$$t = \sqrt[4]{\frac{\mathrm{OD}^2(P_F)}{19.2(10^7)(h/t)}} = 0.054 \text{ in (1.37 mm)}$$

7. $$h = 1.41t = 1.41(0.054) = 0.076 \text{ in}$$
$$H = h + t = 0.076 + 0.054 = 0.130 \text{ in}$$

8. Refer to Fig. 24-31. The load of 202 lb will be reached at f_1 = 50 percent of maximum available deflection. And $f_1 = 0.50(0.076) = 0.038$ in deflection, or the load of 223 lb will be reached at $H_1 = H - f_1 = 0.130 - 0.038 = 0.092$-in height at load. To allow for wear, the spring should be preloaded at $H_2 = H_1 - f(\text{wear}) = 0.092 - 0.032 = 0.060$-in height. This preload corresponds to a deflection $f_2 = H - H_2 = 0.130 - 0.060 = 0.070$ in. Then $f_2/h = 0.070/0.076 = 0.92$, or 92 percent of h.
9. Because 92 percent of h exceeds the recommended 85 percent (the load-deflection curve is not reliable beyond 85 percent deflection when the washer is compressed between flat surfaces), increase the deflection range to 40 to 85 percent. From Fig. 24-31, the load at 40 percent deflection is 78.5 percent, and $P_F = 223/0.785 =$

TABLE 24-21 Maximum Recommended Stress Levels for Belleville Washers in Static Applications

Material	Percent of Tensile Strength	
	Set Not Removed	Set Removed
Carbon or Alloy Steel	120	275
Nonferrous and Austenitic Stainless Steel	95	160

SOURCE: Associated Spring, Barnes Group Inc.

284 lb. Repeat previous procedures 4, 5, 6, 7, and 8, and find that $100(f_2/h) = 81$ percent of h. The final design is as follows:

Material: AISI 1074

OD = 2.99 in (76 mm)

ID = 1.50 in (38 mm)

t = 0.055 in (1.40 mm) nominal

h = 0.078 in (1.95 mm) nominal

Tensile stress $S_{T_1} = -29.5$ kpsi (-203 MPa) at $f_2 = 85$ percent of h

Tensile stress $S_{T_2} = 103$ kpsi (710 MPa) at $f_2 = 85$ percent of h

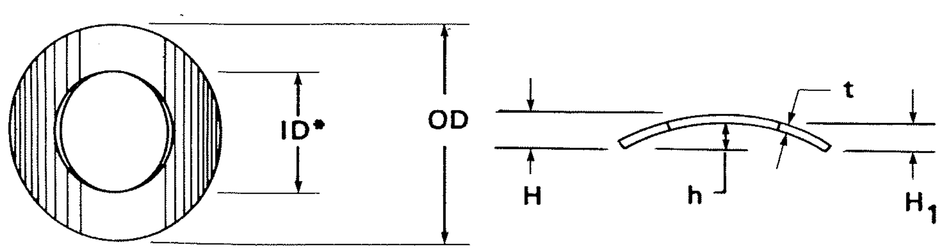
24-8 SPECIAL SPRING WASHERS

Spring washers are being used increasingly in applications where there is a requirement for miniaturization and compactness of design. They are used to absorb vibrations and both side and end play, to distribute loads, and to control end pressure.

Design equations have been developed for determining the spring characteristics of curved, wave, and Belleville washers. There are no special design equations for slotted and finger washers. They are approximated by using Belleville and cantilever equations and then are refined through sampling and testing.

24-8 Curved Washers

These springs (Fig. 24-35) exert relatively light thrust loads and are often used to absorb axial end play. The designer must provide space for diametral expansion



*Long axis of the washer in free position

FIG. 24-35 Curved washer. *(Associated Spring, Barnes Group Inc.)*

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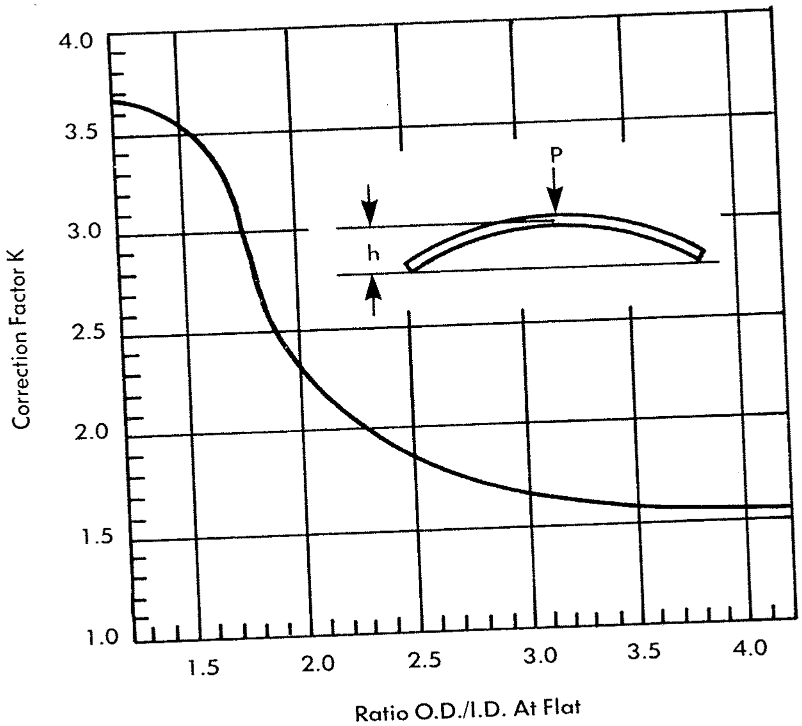

FIG. 24-36 Empirical correction factor K for curved spring washers. *(Associated Spring, Barnes Group Inc.)*

which occurs as the washer is compressed during loading. Bearing surfaces should be hard, since the washer edges tend to dig in. The spring rate is approximately linear up to 80 percent of the available deflection. Beyond that the rate will be much higher than calculated. Load tolerance should not be specified closer than ± 20 percent.

Approximate equations are

$$P = \frac{4fEt^3}{\mathrm{OD}^2(K)} \tag{24-44}$$

and

$$S = \frac{1.5KP}{t^2} \tag{24-45}$$

where K is given in Fig. 24-36 and f is 80 percent of h or less.

Maximum recommended stress levels for static operations are given in Table 24-22. Favorable residual stresses can be induced by shot peening and, to a lesser extent, by removing set. The maximum recommended stresses for cyclic applications are given in Table 24-23.

Tensile strengths for carbon steel are obtained from Fig. 24-3.

24-8-2 Wave Washer

These spring washers (Fig. 24-37) are regularly used in thrust loading applications, for small deflections, and for light to medium loads. The rate is linear between 20

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TABLE 24-22 Maximum Recommended Operating Stress Levels for Special Spring Washers in Static Applications

Material	Percent of Tensile Strength: Stress-Relieved	Percent of Tensile Strength: With Favorable Residual Stresses
Steels, Alloy Steels	80	100
Nonferrous Alloys and Austenitic Steel	—	80

Finger washers are not generally supplied with favorable residual stresses.
SOURCE: Associated Spring, Barnes Group Inc.

and 80 percent of available deflection. Load tolerances should be no less than ±20 percent. In the most commonly used range of sizes, these washers can have three, four, or six waves.

Design equations are

$$\frac{P}{f} = \frac{Ebt^3N^4(\text{OD})}{2.4D^3(\text{ID})} \tag{24-46}$$

and

$$S = \frac{3\pi PD}{4bt^2N^2} \tag{24-47}$$

where $D = \text{OD} - b$. The washer expands in diameter when compressed, according to the formula

$$D' = \sqrt{D^2 + 0.458h^2N^2} \tag{24-48}$$

Maximum recommended stress levels for static applications are given in Table 24-22. Favorable residual stresses are induced by shot peening or removing set. Table

TABLE 24-23 Maximum Recommended Operating Stress Levels for Steel Curved and Wave Washers in Cyclic Applications

Life (Cycles)	Percent of Tensile Strength: Maximum Stress
10^4	80
10^5	53
10^6	50

This information is based on the following conditions: ambient environment, free from sharp bends, burrs, and other stress concentrations. AISI 1075
SOURCE: Associated Spring, Barnes Group Inc.

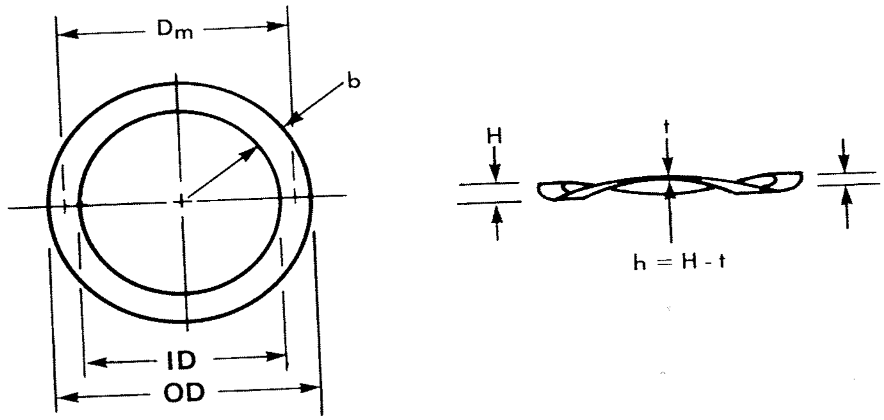

FIG. 24-37 Typical wave spring washer. *(Associated Spring, Barnes Group Inc.)*

24-23 gives the maximum recommended stress levels for cyclic applications. Figure 24-3 provides tensile strengths for carbon steel.

24-8-3 Finger Washers

Finger washers (Fig. 24-38) have both the flexibility of curved washers and the distributed points of loading of wave washers. They are calculated, approximately, as groups of cantilever springs; then samples are made and tested to prove the design. They are most frequently used in static applications such as applying axial load to ball-bearing races to reduce vibration and noise. These washers are not used in cyclic applications because of the shear cuts.

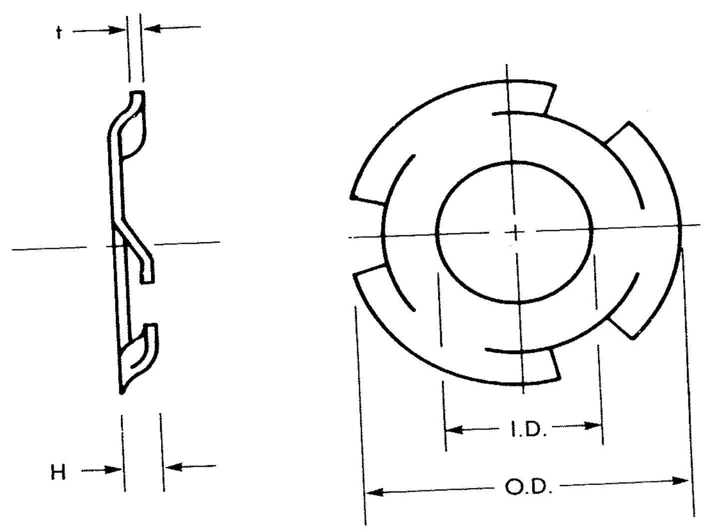

FIG. 24-38 Finger washer. *(Associated Spring, Barnes Group Inc.)*

24-8-4 Slotted Washers

These are more flexible than plain dished washers but should be designed to maintain a constant pressure rather than to operate through a deflection range (see Fig. 24-39).

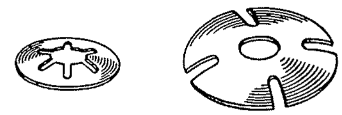

FIG. 24-39 Slotted washers. *(Associated Spring, Barnes Group Inc.)*

24-8-5 Special Considerations

Load specification in flat springs is closely connected with the dimensioning of the form of the spring. From the equations it can be seen that the deflection and load vary in proportion to the third power of the material thickness. The important factors in load control are, first, the material thickness and, second, the deflection. Where close load control is required, the material may have to be selected to restricted thickness tolerance, and/or the free shape may be trued.

24-9 FLAT SPRINGS

24-9-1 Introduction

The classification *flat springs* applies to a wide range of springs made from sheet, strip, or plate material. Exceptions to this classification are power springs and washers. Flat springs may contain bends and forms. Thus the classification refers to the raw material and not to the spring itself.

Flat springs can perform functions beyond normal spring functions. A flat spring may conduct electricity, act as a latch, or hold a part in position. In some flat springs, only a portion of the part may have a spring function.

Most flat springs are custom designs, and the tooling is often a major cost consideration. Flat springs can be cantilever or simple elliptical beams or combinations of both. These two elementary forms are discussed in this section. For a description of the methods used to compute complex flat-spring designs, see [24-6].

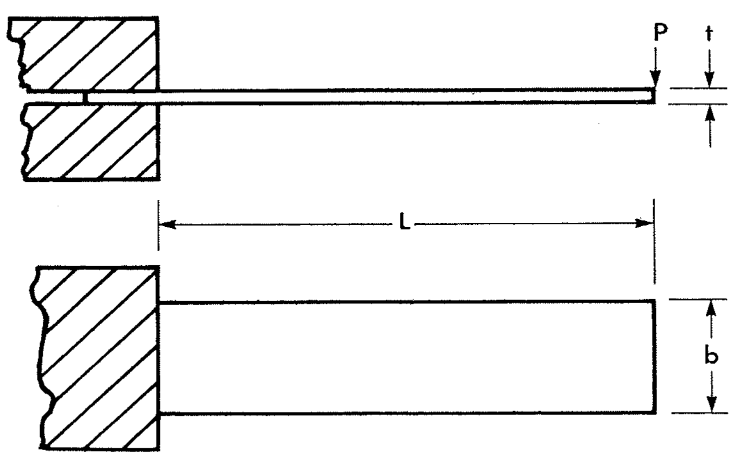

FIG. 24-40 Rectangular cantilever spring. *(Associated Spring, Barnes Group Inc.)*

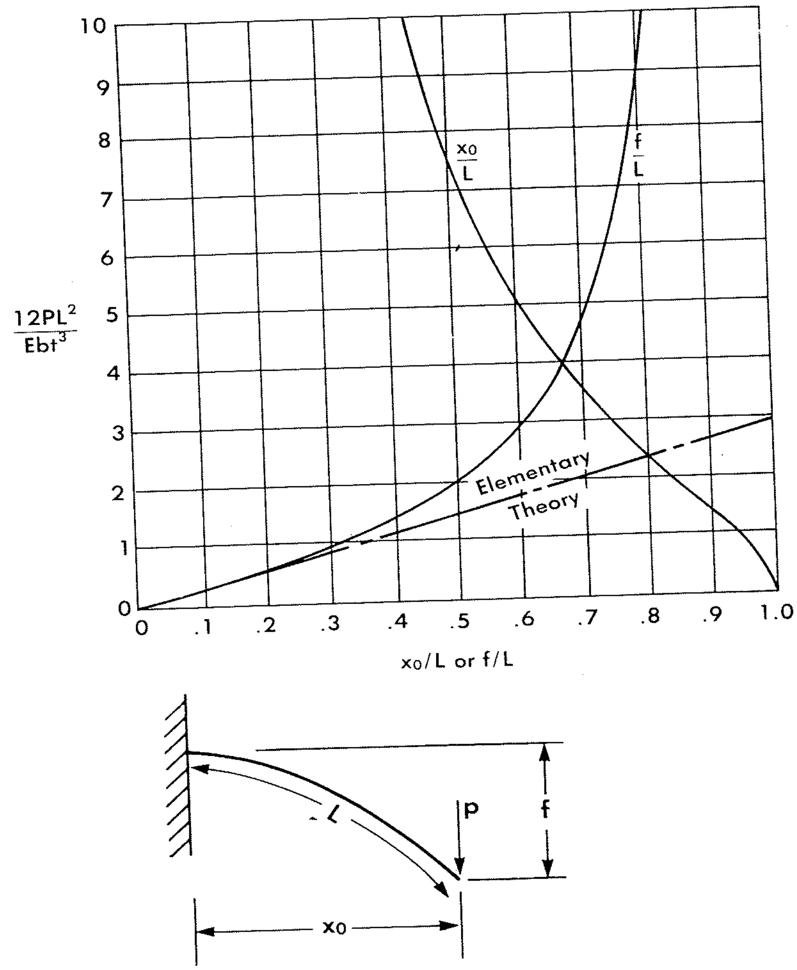

FIG. 24-41 Calculating large deflection in cantilever beams [24-7]. To utilize this figure for any load P, first calculate the quantity $12PL^3/Ebt^3$. Using this value, from the curves find f/L and x_o/L, where x_o is the moment arm of the load P. Deflection then equals L multiplied by f/L. The maximum stress is reduced in the ratio x_o/L. *(Associated Spring, Barnes Group Inc.)*

24-9-2 Cantilever Springs

The basic type of cantilever is a rectangular spring as shown in Fig. 24-40. The maximum bending stress occurs at the clamping point, and the stress is not uniform through the section. This stress is

$$S = \frac{6PL}{bt^2} \tag{24-49}$$

The load is given by

$$P = \frac{fEbt^3}{4L^3} \tag{24-50}$$

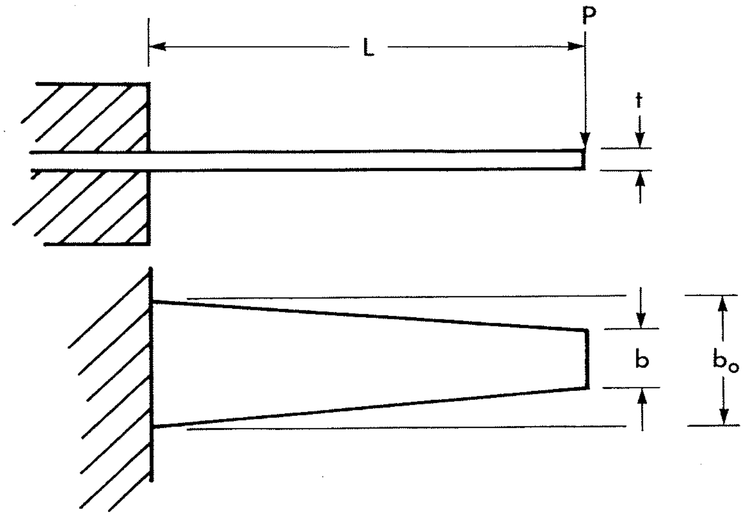

FIG. 24-42 Trapezoidal cantilever spring. *(Associated Spring, Barnes Group Inc.)*

These equations are satisfactory when the ratio of deflection to length f/L is less than 0.3. For larger deflections, use the method described in Fig. 24-41.

In cantilever springs with a trapezoidal or triangular configuration (Fig. 24-42), the stress is uniform throughout and is

$$S = \frac{6PL}{b_o t^2} \tag{24-51}$$

The corresponding load is

$$P = \frac{fEb_o t^3}{4L^3K} \tag{24-52}$$

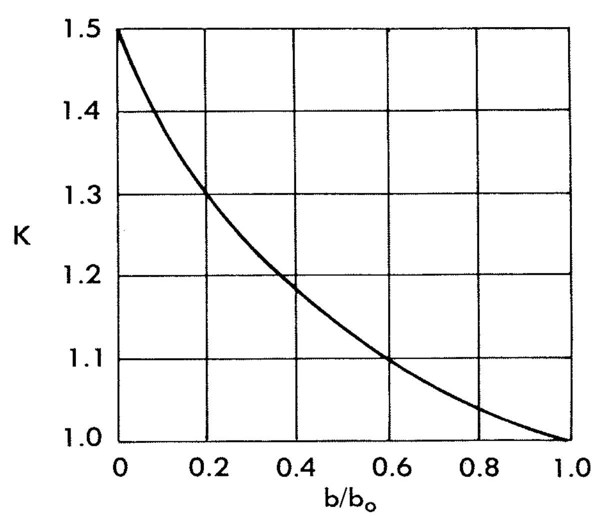

FIG. 24-43 Correction factor for trapezoidal beam-load equation. *(Associated Spring, Barnes Group Inc.)*

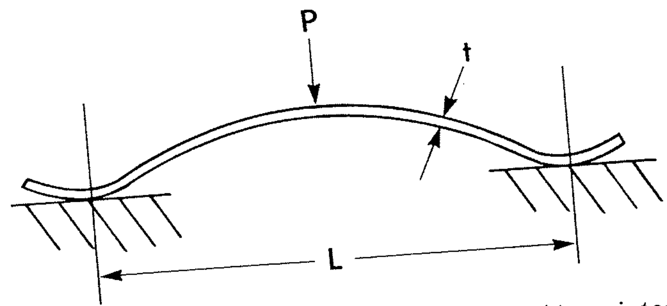

FIG. 24-44 Simple beam spring. *(Associated Spring, Barnes Group Inc.)*

where K = constant based on the ratio b/b_o (Fig. 24-43). These equations are valid for f/L ratios of less than 0.3.

24-9-3 Simple Beams or Elliptical Springs

Simple beams are usually rectangular and are formed into an arc as in Fig. 24-44. If holes are introduced for clamping purposes, stress will increase at the hole and at the clamping point owing to stress concentration.

When ends are free to move laterally, the equation for load is

$$P = \frac{4fEbt^3}{L^3} \qquad (24\text{-}53)$$

and stress is given by

$$S = \frac{1.5PL}{bt^2} \qquad (24\text{-}54)$$

These equations apply when the ratio f/L is less than 0.15.

STRESS CONSIDERATIONS. The maximum design stresses for cantilevers and simple beams are given in Table 24-24 for static applications and in Table 24-25 for cyclic applications. These recommendations do not apply when holes, sharp corners, notches, or abrupt changes in cross section are incorporated in the design, and should be used for guidance only.

TABLE 24-24 Maximum Design Stresses for Cantilever and Simple Beam Springs in Static Applications

Percent of Tensile Stength			
Ferrous Material		Nonferrous Material	
No Residual Stress	Maximum Residual Stress	No Residual Stress	Maximum Residual Stress
80	100	75	80

SOURCE: Associated Spring, Barnes Group Inc.

TABLE 24-25 Maximum Design Stresses for Carbon-Steel Cantilever and Simple Beam Springs in Cyclic Applications

Number of Cycles	Percent of Tensile Strength	
	Not Shot-Peened	Shot-Peened*
10^5	53	62
10^6	50	60
10^7	48	58

*Shot peening is not recommended for thin materials and complex shapes. This information is based on an ambient environment. Stress ratio = 0.

SOURCE: Associated Spring, Barnes Group Inc.

24-10 CONSTANT-FORCE SPRINGS

A constant-force spring is a roll of prestressed material which exerts a nearly constant restraining force to resist uncoiling. Its unique characteristic is *force independent of deflection.* The force required to produce a unit deflection is the same for each increment of coil because the radius of curvature of each increment is the same as any other.

Although these springs are not constant-load or constant-torque springs in the precise meaning of those terms, they produce a more nearly constant load over a greater deflection than any other spring design covered here. See Fig. 24-45. Constant-force springs are made of both type 301 stainless steel and ultra-high-strength

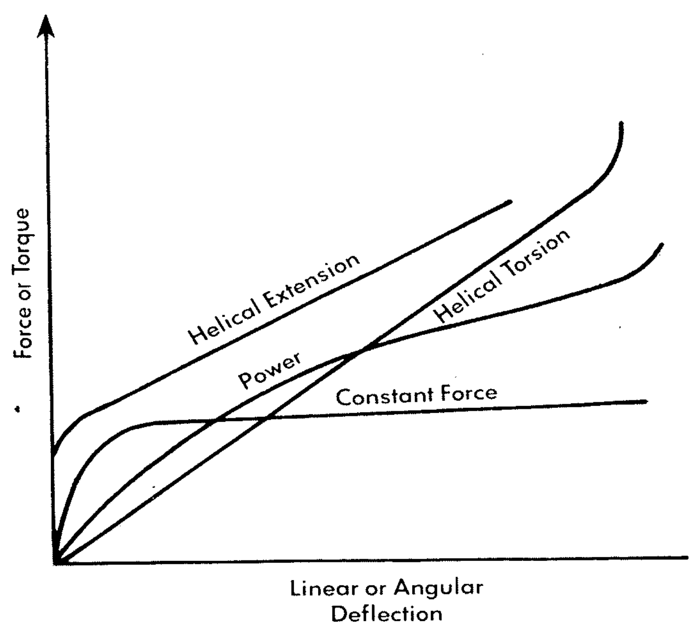

FIG. 24-45 Load-deflection curves for various spring configurations. *(Associated Spring, Barnes Group Inc.)*

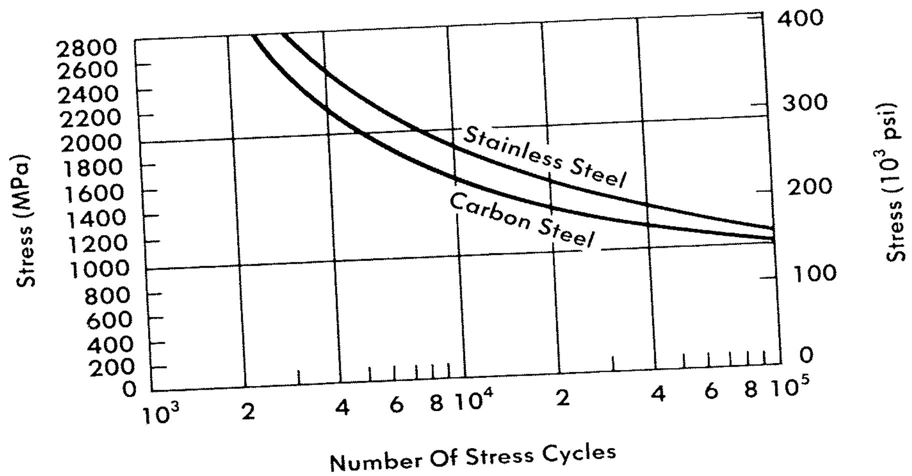

FIG. 24-46 Maximum bending stress versus number of stress cycles for constant-force springs. These curves are based on no. 1 round-edge strip. *(Associated Spring, Barnes Group Inc.)*

high-carbon steels, with many of the applications using stainless steel because of its inherent resistance to corrosion.

One of the most severe limitations on the use of constant-force springs is their relatively short operating life. The most efficient use of material will produce a life of about 3000 cycles. Although life of hundreds of thousands of cycles is possible, most applications fall into the range of 3000 to 30 000 cycles. Figure 24-46 shows the relationship between stress and fatigue life. These curves are derived from experimentally obtained data.

Some applications involving constant-force and constant-torque springs are simple extension springs, window sash counterbalances, camera motors, toys, machine carriage returns, constant-pressure electric-motor brush springs, space vehicle applications, and retraction devices.

24-10-1 Extension Type

This type of spring is a spiral spring made of strip material wound on the flat with an inherent curvature such that, in repose, each coil wraps tightly on its inner neigh-

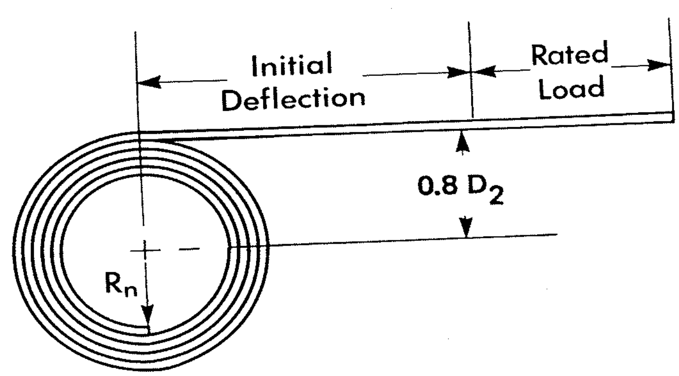

FIG. 24-47 Typical constant-force extension spring (extension form). *(Associated Spring, Barnes Group Inc.)*

bor. In use the strip is extended with the free end loaded and the inner end supported on a drum or arbor. Very long deflections are possible, but the strip becomes unstable in long deflections and must be guided or supported to avoid kinking or snarling on the return stroke.

The rated load is not reached until after an initial deflection of 1.25 times the drum diameter, as shown in Fig. 24-47. Idler pulleys can be used but should be no smaller in diameter than the natural diameter of the coils and should never be used in a direction to cause backbending against the strip curvature.

24-10-2 Design Equations

$$P = \frac{Ebt^3}{6.5D_n^2} \qquad \text{for } N \leq 10 \tag{24-55}$$

$$P = \frac{Ebt^3}{6.5D_1}\left(\frac{2}{D_n} - \frac{1}{D_1}\right) \qquad \text{for } N > 10 \tag{24-56}$$

If unknown, let $b/t = 100/1$, $D_2 = 1.2\ D_n$,

$$S = \frac{Et}{D_n} \tag{24-57}$$

and

$$L = 1.57N(D_1 + D_2) \qquad \text{or} \qquad L \simeq f + 5D_2$$

where N = number of turns
D_1 = outside coil diameter
D_2 = drum (arbor) diameter
D_n = natural diameter
E = modulus of elasticity

24-10-3 Spring Motor Type

When a constant-force spring is mounted on two drums of different diameters and the spring is backbent onto the larger diameter, the result is a constant-force spring motor. The strip is in repose on the smaller (storage) drum and is backbent onto the larger (output) drum. Torque is taken from the output drum shaft as shown in Fig. 24-48.

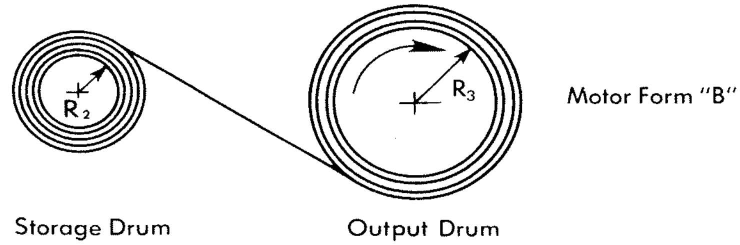

FIG. 24-48 Typical constant-torque motor spring. *(Associated Spring, Barnes Group Inc.)*

ATTACHMENT C

Note here that constant torque does not mean constant speed. Constant torque implies uniform acceleration, and the mechanism so driven will continue to speed up unless restrained by a governor mechanism. Load tolerances are normally held within ±10 percent.

24-10-4 Design Equations

$$M = \frac{Ebt^3 D_3}{13}\left(\frac{1}{D_n} + \frac{1}{D_3}\right)^2 \tag{24-58}$$

$$S = Et\left(\frac{1}{D_n} + \frac{1}{D_3}\right) \tag{24-59}$$

$$L = \pi N(D_3 + Nt) + 10D_3 \tag{24-60}$$

$$R_c = R_n \sqrt{4 + \frac{4R_3}{R_n} + \frac{R_n}{R_3} + \left(\frac{R_3}{R_n}\right)^2} \tag{24-61}$$

Design Suggestions. Let

$$\frac{b}{t} = 100 \qquad \frac{D_n}{t} = 250 \qquad \frac{D_3}{D_n} = 2 \qquad \frac{D_3}{D_2} = 1.6$$

where D_n = natural diameter
R_n = natural radius
D_2 = storage-drum diameter
D_3 = output-drum diameter
R_3 = output-drum radius
N = number of revolutions
R_c = minimum center-to-center distance of drums

24-11 TORSION BARS

Torsion bars used as springs are usually straight bars of spring material to which a twisting couple is applied. The stressing mode is torsional. This type of spring is very efficient in its use of material to store energy. The major disadvantage with the torsion bar is that unfavorable stress concentrations occur at the point where the ends are fastened.

Although both round and rectangular bar sections are used, the round section is used more often.

24-11-1 Design Equations: Round Sections

$$\phi = \frac{584ML}{d^4 G} \tag{24-62}$$

$$S = \frac{16M}{\pi d^3} \tag{24-63}$$

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where ϕ = rotation angle in degrees
S = shear stress
L = active length

24-11-2 Design Equations: Rectangular Sections

$$\phi = \frac{57.3ML}{K_1bt^3G} \qquad (24\text{-}64)$$

$$S = \frac{M}{K_2bt^2} \qquad (24\text{-}65)$$

where factors K_1 and K_2 are taken from Table 24-26.

The assumptions used in deriving these equations are (1) the bar is straight, (2) the bar is solid, and (3) loading is in pure torsion.

Torsion-bar springs are often preset in the direction in which they are loaded by twisting the bar beyond the torsional elastic limit. Care must be taken in the use of a preset bar: It must be loaded in the same direction in which it was preset; otherwise, excessive set will occur.

24-12 POWER SPRINGS

Power springs, also known as clock, motor, or flat coil springs, are made of flat strip material which is wound on an arbor and confined in a case. Power springs store and release rotational energy through either the arbor or the case in which they are retained. They are unique among spring types in that they are almost always stored in a case or housing while unloaded. Figure 24-49 shows typical retainers, a case, and various ends.

24-12-1 Design Considerations

Power springs are stressed in bending, and stress is related to torque by

$$S = \frac{6M}{bt^2} \qquad (24\text{-}66)$$

TABLE 24-26 Factors for Computing Rectangular Bars in Torsion

b/t	K_1	K_2
1.0	0.140	0.208
1.5	0.196	0.231
2.0	0.229	0.246
2.5	0.249	0.258
3.0	0.263	0.267
5.0	0.291	0.291

SOURCE: A. M. Wahl, *Mechanical Springs,* 2d ed., McGraw-Hill Book Company, New York, 1963.

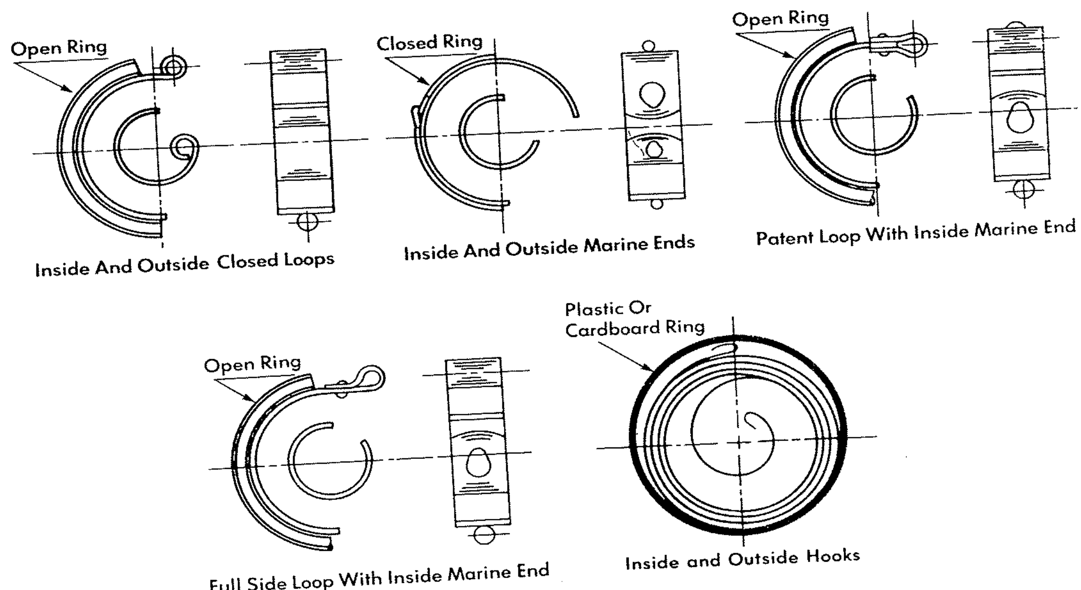

FIG. 24-49 Typical power spring retainers and ends. *(Associated Spring, Barnes Group Inc.)*

Load-deflection curves for power springs are difficult to predict. As a spring is wound up, material is wound onto the arbor. This material is drawn from that which was at rest against the case. Thus, the length of active material is constantly changing, which makes it difficult to develop a workable expression for the spring rate. For these reasons, ratios, tables, and graphical presentations are used to develop the design criteria.

The ratio of arbor diameter to thickness D_a/t is sometimes called the *life factor*. If it is too small, fatigue life will suffer. The life factor is usually maintained from 15 to 25. The ratio of active strip length to thickness L/t determines the flatness of the spring-gradient (torque-revolution) curve. The curve is flatter when L is longer. The usual range of the L/t ratio is from 5000 to 10 000. The ratio of the inside diameter of cup (case or housing) to thickness D_c/t is the *turns factor*. This determines the motion capability of the spring or indicates how much space is available between the arbor and the material lying against the inside of the case.

24-12-2 Design Procedure

In order to design a power spring that will deliver a given torque and number of turns, first determine its maximum torque in the fully wound condition. If a spring is required to deliver a minimum torque of 0.5 N·m for 10 revolutions (r) of windup and 10 r equals 80 percent unwound from solid, then from Fig. 24-50 we see that the torque at that point is 50 percent of the fully wound. Thus the fully wound torque is 1.0 N·m. Table 24-27 shows that a strip of steel 0.58 mm thick and 10 mm wide will provide 1.0 N·m of torque at the fully wound position per 10 mm of strip width.

Figure 24-51 shows that the average maximum solid stress for 0.58-mm-thick stock is about 1820 MPa. At the hardness normally supplied in steel strip for power springs, this is about 95 percent of tensile strength.

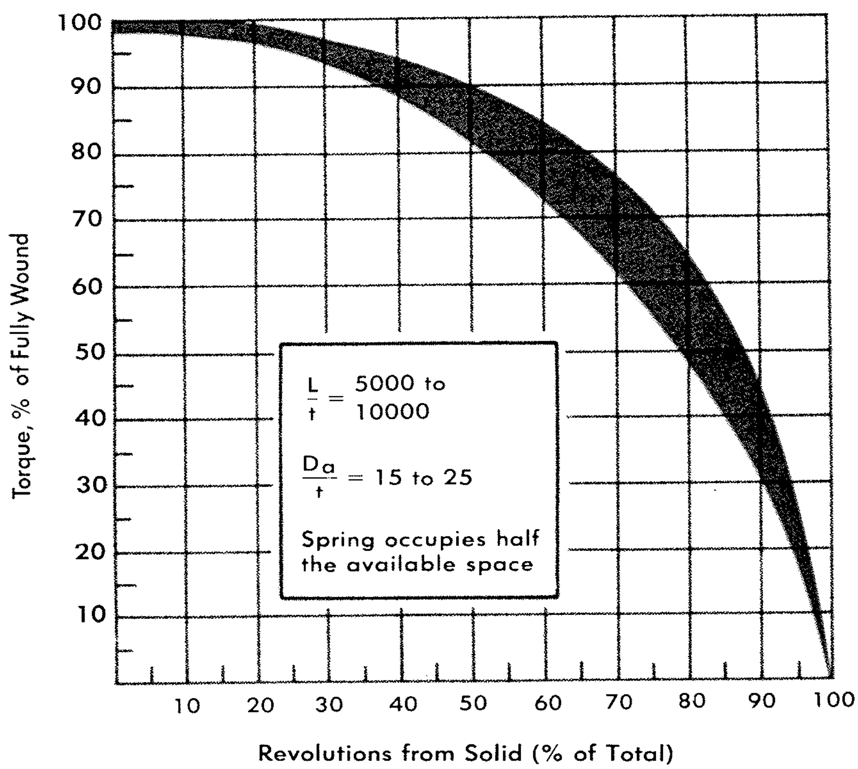

FIG. 24-50 Typical normalized torque-revolution curve for power springs. *(Assoicated Spring, Barnes Group Inc.)*

In Fig. 24-52, 10 turns relate to a length-to-thickness L/t ratio of 4300. With t = 0.58, L equals 2494 mm. Similarly, 4300 L/t relates to a D_c/t ratio of 107. Then D_c = 62.06 mm. If

$$L = \frac{D_c^2 - D_a^2}{2.55t} \tag{24-67}$$

then $D_a = \sqrt{D_c^2 - 2.55Lt} = 12.72$ mm and $D_a/t = 22$.

The equation for the number of turns a power spring will deliver, when it occupies half the space between arbor and case, is

$$\theta = \frac{\sqrt{2(D_c^2 + D_a^2)} - (D_c + D_a)}{2.55t} \tag{24-68}$$

In this example $\theta = 10$ r.

Experience shows that highly stressed power springs, made from pretempered AISI 1095 steel with a hardness of HRC 50 to 52 and stressed to 100 percent of tensile strength, could be expected to provide approximately 10 000 full-stroke life cycles. If the maximum stress were 50 percent of tensile strength at full stroke, then a life of about 100 000 cycles could be expected.

The final design is as follows:

t = 0.023 in (0.58 mm)

1095 carbon steel, HRC 51, no.1 round edge

TABLE 24-27 Torque per Unit of Width at Maximum Allowable Stress for Steel; L/t Range Is 5000 to 10 000

Thickness t		Unit Torque M		Thickness t		Unit Torque M	
mm	in	N·m/10 mm of width	lb·in/in of width	mm	in	N·m/10 mm of width	lb·in/in of width
0.127	0.005	0.0587	1.32	1.30	0.051	4.132	92.90
0.152	0.006	0.0841	1.89	1.37	0.054	4.541	102.1
0.178	0.007	0.1094	2.46	1.45	0.057	4.991	112.2
0.203	0.008	0.1419	3.19	1.60	0.063	5.947	133.7
0.229	0.009	0.1775	3.99	1.70	0.067	6.619	148.8
0.254	0.010	0.2171	4.88	1.83	0.072	7.504	168.7
0.279	0.011	0.2620	5.89	1.93	0.076	8.282	186.2
0.305	0.012	0.3074	6.91	2.03	0.080	8.981	201.9
0.330	0.013	0.3567	8.02	2.18	0.086	10.37	233.2
0.356	0.014	0.4101	9.22	2.34	0.092	11.74	264.0
0.381	0.015	0.4679	10.52	2.49	0.098	13.12	295
0.406	0.016	0.5271	11.85	2.67	0.105	14.86	334
0.432	0.017	0.5876	13.21	2.84	0.112	16.59	373
0.457	0.018	0.6530	14.68	3.05	0.120	18.82	423
0.483	0.019	0.7215	16.22	3.18	0.125	20.24	455
0.508	0.020	0.7953	17.88	3.43	0.135	23.35	525
0.584	0.023	1.025	23.05	3.58	0.141	25.35	570
0.635	0.025	1.189	26.72	3.76	0.148	27.76	624
0.711	0.028	1.452	32.65	3.96	0.156	30.69	690
0.813	0.032	1.841	41.40	4.11	0.162	33.00	742
0.889	0.035	2.144	48.20	4.50	0.177	39.23	882
1.041	0.041	2.824	63.50	4.75	0.187	43.81	985
1.19	0.047	3.585	80.60				

SOURCE: Associated Spring, Barnes Group Inc.

$$b = 0.394 \text{ in } (10 \text{ mm})$$
$$L = 98.188 \text{ in } (2494 \text{ mm})$$
$$D_a = 0.501 \text{ in } (12.72 \text{ mm})$$
$$D_c = 2.443 \text{ in } (62.06 \text{ mm})$$

24-13 HOT-WOUND SPRINGS

24-13-1 Introduction

Springs are usually cold-formed when bar or wire diameters are less than 10 mm (approximately $\frac{3}{8}$ in). When the bar diameter exceeds 16 mm (approximately $\frac{5}{8}$ in), cold forming becomes impractical and springs are hot-wound.

Hot winding involves heating the steel into the austenitic range, winding hot, quenching to form martensite, and then tempering to the required properties. Although the most common types of hot-wound springs are compression springs for

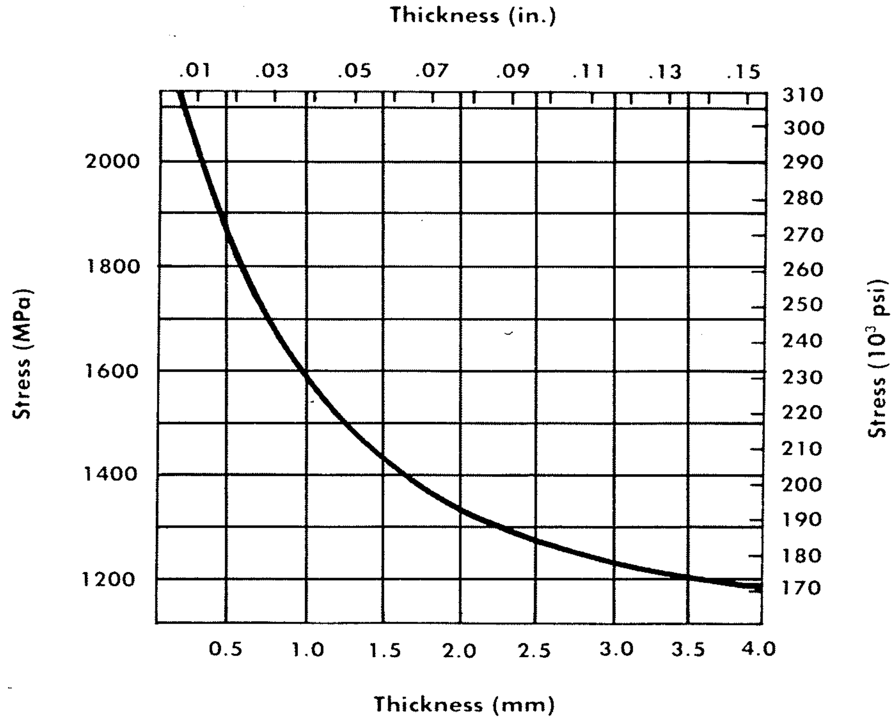

FIG. 24-51 Average maximum solid stress in carbon-steel power springs. *(Associated Spring, Barnes Group Inc.)*

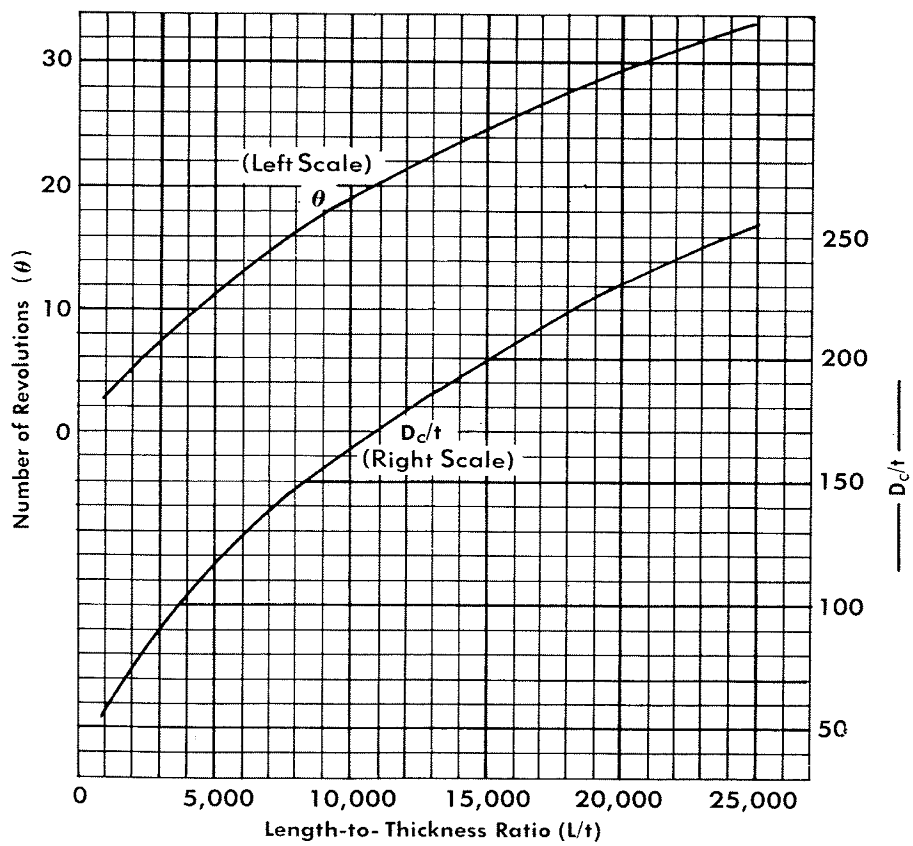

FIG. 24-52 Relationships among number of revolutions, case diameter, strip length, and thickness for power springs. *(Associated Spring, Barnes Group Inc.)*

TABLE 24-28 Diameter and Out-of-Roundness Tolerances for Hot-Rolled Carbon-Steel Bars

Diameter mm(in.) Over	Diameter mm(in.) Through	Tolerance ± mm(in.)	Out-of-Roundness mm(in.)
	8 (0.315)	0.13 (0.005)	0.20 (0.008)
8 (0.315)	10 (0.394)	0.15 (0.006)	0.22 (0.009)
10 (0.394)	15 (0.591)	0.18 (0.007)	0.27 (0.011)
15 (0.591)	20 (0.787)	0.20 (0.008)	0.30 (0.012)
20 (0.787)	25 (0.984)	0.23 (0.009)	0.34 (0.013)
25 (0.984)	30 (1.181)	0.25 (0.010)	0.38 (0.015)
30 (1.181)	35 (1.378)	0.30 (0.012)	0.45 (0.018)
35 (1.378)	40 (1.575)	0.35 (0.014)	0.52 (0.020)
40 (1.575)	60 (2.362)	0.40 (0.016)	0.60 (0.024)
60 (2.362)	80 (3.150)	0.60 (0.024)	0.90 (0.035)

SOURCE: Associated Spring, Barnes Group Inc.

highway, off-highway, and railroad-vehicle suspension applications, torsion and extension springs can also be hot-wound.

24-13-2 Special Design Considerations

Design equations for hot-wound springs are the same as those for cold-formed springs except for the use of an empirical factor K_H which adjusts for effects related to hot-winding springs. Multiply the spring rate by K_H.

The values for factor K_H are 0.91 for springs made from hot-rolled carbon or low-alloy steel, *not* centerless ground; 0.96 for springs made from hot-rolled carbon or low-alloy steel, centerless ground; and 0.95 for torsion springs made from carbon or low-alloy steel.

The ends of hot-wound springs can be open or squared or either ground or not ground. Solid height is calculated in the same way as for cold-wound springs; but when space is limited, L_s can be reduced to $(N_t - 0.5)d$ by using a heavy grind.

The end configurations of extension or torsion springs must be formed hot at the same time as the spring is wound. If the configuration is complex, they may become cool in the process and the whole spring may have to be reheated into the austenitic range. Note that hot-wound extension springs cannot have initial tension.

TABLE 24-29 Maximum Allowable Torsional Stress for Hot-Wound Helical Compression Springs in Static Applications

Before Set Removal	After Set Removal
50% of TS	65–75% of TS

Torsional stress after set removal depends on material size and amount of set removed.

SOURCE: Associated Spring, Barnes Group Inc.

TABLE 24-30 Maximum Allowable Torsional Stress for Hot-Wound Helical Compression Springs in Cyclic Applications

Fatigue Life (Cycles)	Percent of Tensile Strength	
	Not Shot-Peened	Shot-Peened
10^5	40	48
10^6	38	46
10^7	35	43

This information is based on **centerless ground** AISI 5160, 5160H and 1095, HRC 44 to 48, 25 mm (1″) diameter. Set has not been removed. Conditions are: no surging, room temperature and non-corrosive environment.

$$\text{Stress ratio in fatigue} = \frac{\text{S Minimum}}{\text{S Maximum}} = 0.$$

SOURCE: Associated Spring, Barnes Group Inc.

24-13-3 Materials

The common hot-wound alloys are AISI 5160, 5160H, and 1095 steels. The normal range of hardness is from HRC 44 to 48. Corresponding tensile strengths are 1430 to 1635 MPa.

The hot-rolled wire used in hot-wound springs is produced in standard sizes. Section 9-3 lists preferred bar diameters. Bar diameter variation and bar out-of-roundness tolerances are approximated in Table 24-28.

24-13-4 Choice of Operating Stress

Static Applications. The stress is calculated as in cold-wound springs. Use Table 24-29 for set-point information.

Cyclic Applications. Hot-wound springs are made from hot-rolled wire are used in cyclic applications because rolled bars are subject to a variety of characteristic material defects mostly related to the bar surface condition. Therefore Table 24-30 can be used only for centerless ground alloy bars. Practical manufacturing tolerances for hot-wound springs can be found in ASTM A125.

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