1.2 MATERIALS

Before discussing the properties of the materials available to the designer of scientific apparatus, we shall first define the parameters used to specify these properties. In Table 1.4 the properties of many useful materials are tabulated in terms of these parameters.

1.2.1 Parameters to Specify Properties of Materials

The strength and elasticity of a metal are best understood in terms of a stress-strain curve such as is shown in Figure 1.9. *Stress* is the force applied to the material per unit of cross-sectional area. This may be a stretching, compressing, or shearing force. *Strain* is the deflection per unit length which occurs in response to a stress. Most metals deform in a similar way under compression or elongation, but their response to shear is different. When a stress is applied to a metal the initial strain is *elastic* and the metal will return to its original dimensions if the stress is removed. Beyond a certain stress however, *plastic* strain occurs and the metal is permanently deformed.

The *tensile strength* or ultimate strength of a metal is the stress applied at the maximum of the stress-strain curve. The metal is very much deformed at this point, so that in most cases it is impractical to work at such a high load.

A more important parameter for design work is the *yield strength*. This is the stress required to produce a stated, small plastic strain in the metal—usually 0.2% permanent deformation. For some materials the *elastic limit* is specified. This is the maximum stress which the material can withstand without a permanent deformation occurring.

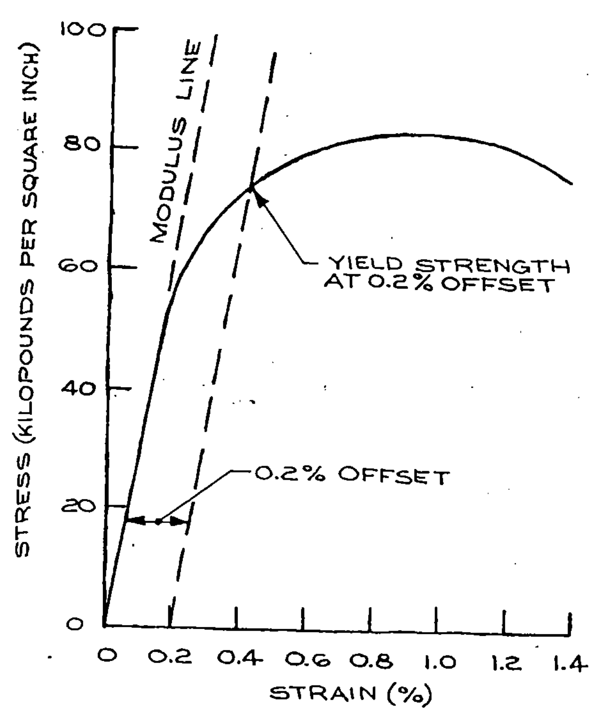


Figure 1.9 Typical stress-strain curve for a metal.

Table 1.4 TYPICAL PROPERTIES OF SOME RESEARCH MATERIALS

Material	Density (lb/in.^{3})	Yield Strength (10^3 psi)	Tensile Strength (10^3 psi)	Modulus of Elasticity (10^6 psi)	Shear Modulus (10^6 psi)	Hardness[a]	Coefficient of Thermal Expansion (10^{-6} °C^{-1})	Comments
Ferrous Metals								
Cast gray iron (ASTM 30)	0.253	25	30	13	5.2	200 BHN	11	
1015 steel (hot-finished)	0.284	27	50	29	11	100 BHN	15	Low carbon (0.15% C)
1030 steel (hot-finished)	0.284	37	68	29	11	137 BHN	15	Medium carbon (0.30% C)
1050 steel (hot-finished)	0.284	50	90	29	11	180 BHN	15	High carbon (0.50% C)
Type 304 stainless steel (annealed)	0.286	35	85	28	10	150 BHN	17	Austenitic
Type 304 stainless steel (cold-worked)	0.286	75	110	28	10	240 BHN	17	Austenitic
Type 316 stainless steel (cold-worked)	0.286	60	90	28	10	190 BHN	16	Austenitic
Nickel Alloys								
Monel 400: 25°C	0.319	25	70	26		110 BHN	14	Slightly magnetic
500°C		22	45				16	
Monel 500: 25°C	0.306	40–150	90–180	26		140 BHN	14	Nonmagnetic
500°C		90	95				16	
Inconel 600: 25°C	0.304	36	90	31		120 BHN	12	Strong, resists high-temperature oxidation
650°C		26	65	20			16	
Invar (0–100°C)	0.294	24	70	21		140 BHN	1.3	
Aluminum Alloys								
1100-0	0.100	5	13	10	4	23 BHN	23	99% Al
2024-T4	0.098	47	68	10.6	4	120 BHN	23	3.8% Cu, 1.2% Mg, 0.3% Mn
2024-T4 (200°C)	0.098	12	18					
6061-T6	0.100	40	45	10	4	95 BHN	23	0.15% Cu, 0.8% Mg, 0.4% Si
7075-T6	0.101	73	83	10.4	4	150 BHN	23	5.1% Zn, 2.1% Mg, 1.2% Cu
Copper Alloys								
Yellow brass (annealed)	0.306	14	46	15	6		20	65% Cu, 35% Zn
Yellow brass (cold-worked)	0.306	60	74	15	6	150 BHN	20	65% Cu, 35% Zn
Cartridge brass (½ hard)	0.308	52	70	16	6	145 BHN	20	70% Cu, 30% Zn
Beryllium copper (precipitation-hardened)	0.297	140	175	19	7	380 BHN	17	98% Cu, 2% Be
Unalloyed Metals								
Copper	0.323	10	32	17	6.5	44 BHN	17	
Molybdenum	0.369	82	95	47	17	190 VHN	5.4	Refractory, nonmagnetic
Tantalum	0.600	48	67	27	10	80 VHN	6.5	Refractory, somewhat ductile
Tungsten	0.697	220	220	59	22	350 VHN	5	Refractory, very dense
Plastics								
Phenolics	0.049		7.5	1		125 R_M	81	Bakelite®, Formica®
Polyethylene (low density)	0.033		2	0.025		10 R_R	180	
Polyethylene (high density)	0.034		4	0.12		40 R_R	216	
Polyamide	0.040	11.8		0.410		118 R_R	90	Nylon®
Polymethylmethacrylate	0.043		8	0.42		90 R_M	72	Lucite®, Plexiglas®
Polytetrafluoroethylene	0.077		2.5	0.060		60 R_R	99	Teflon®
Polychlorotrifluoroethylene	0.076		6	0.25		110 R_R	70	Kel-F®
Polyimide	0.052		12	0.37		45 R_R	60	
Polycarbonate	0.043		9	0.34		118 R_R	70	ABS®
Ceramics								
Alumina (polycrystalline)	0.141		35	48		9 Mohs	8	99% alumina
Macor®	0.091		4	9.3	3.6	250 Knoop	9.4	Corning machinable ceramic
Wood								
Douglas fir (air-dried)	0.011		9.5, 0.4[b]	1.4			6,35[b]	Typical of softwoods
Oregon white oak (air-dried)	0.026		10.2, 0.8[b]	2.3			5,55[b]	Typical of hardwoods

[a] BHN = Brinell hardness number; R = Rockwell hardness; VHN = Vickers hardness.
[b] Parallel to fiber, across fiber.

The slope of the straight-line portion of the stress-strain curve is called the *modulus of elasticity* E. It is a measure of the stiffness of the material. It is valuable to note that E is about the same for all grades of steel (about 30×10^6 psi) and about the same for all aluminum alloys (about 10×10^6 psi), regardless of the strength or hardness of the alloy. The effect of elastic deformation is specified by *Poisson's ratio* μ, which is the ratio of transverse contraction per unit dimension of a bar of unit cross section to its elongation per unit length, when subjected to a tensile stress. For most metals $\mu \simeq 0.3$.

The *hardness* of a material is a measure of its resistance to indentation and is usually determined from the force required to drive a standard indenter into the surface of the material or from the depth of penetration of an indenter under a standardized force. The common hardness scales used for metals are the Brinell hardness number (BHN), the Vickers scale (VHN), and the Rockwell C scale. Type 304 stainless steel is about 150 BHN, 160 Vickers, or 0 Rockwell C. A file is 600 BHN, 650 Vickers, or 60 Rockwell C. The Shore hardness scale is determined by the height of rebound of a steel ball dropped from a specified distance above the surface of a material under test. The Mohs hardness scale is used by mineralogists and ceramic engineers. It is based upon standard materials each of which will scratch all materials below it on the scale. Figure 1.10 shows the approximate relation of the various hardness scales.

A designer must consider the machinability of a material before specifying its use in the fabrication of a part that must be lathe-turned or milled. In general, the harder and stronger a material, the more difficult it is to machine. On the other hand, very soft materials such as copper and some nearly pure aluminums are also difficult to machine because the metal tends to adhere to the cutting tool and produce a ragged cut. Some metals are alloyed with other elements to improve their machinability. Free-machining steels and brass contain a small percentage of lead or sulfur. These additives do not usually affect the mechanical properties of the metal, but since they have a relatively high vapor pressure, their outgassing at high temperature can pose a problem in some applications.

1.2.2 Heat Treating and Cold Working

The properties of many metals and metal alloys may be considerably changed by heat treating or cold working, which changes the chemical or mechanical nature of the granular structure of the metal. The two classes of heat treatment are quenching and annealing.[1] In *quenching*, the metal is heated above a transition point and then quickly cooled in order to freeze in the granular properties possessed by the metal above the transition. The cooling is accomplished by plunging the heated part into water or oil. Quenching is usually performed to harden a metal. Hardened metals may be softened by *annealing*, wherein the metal is heated above the transition temperature and then slowly cooled. It is frequently desirable to anneal hardened metals before machining and reharden them after working. *Tempering* is an intermediate heat treatment wherein previously hardened metal is reheated to a temperature below the transition point in order to relieve stresses and then cooled at a rate that preserves the desired properties of the hardened material.

Cold working consists of rolling or otherwise plastically deforming a metal to reduce the grain size. Not all metals benefit from cold working, but for some the strength is greatly increased. Because of the annealing effect, the strength and hardness derived from cold working begin to disappear as a metal is heated. This occurs above 250°C for steel and above 125°C for aluminum. In rolling or spinning operations, some metals will work-harden to such an extent that the material must be periodically annealed during fabrication to retain its workability. Cold working reduces the toughness of metal. The surface of a metal part can be work-hardened, without modifying the internal structure, by peening or shot blasting and by some rolling operations. The strength of metal stock may depend upon the method of manufacture. For example, sheet metal and metal wire are usually much stronger than the bulk metal because of the work hardening produced by rolling or drawing.

The surface of a metal part may be chemically modified and then heat-treated to increase its hardness while retaining the toughness of the bulk of the material welded because their electrical resistance is so low that a current dissipates little power. Small spot-welding units with hand-held electrodes are available from commercial sources and are quite useful in the lab.

1.3.4 Threaded Fasteners

Threaded fasteners are used to join parts that must be frequently disassembled. A thread on the outside of a cylinder, such as the thread on a bolt, is referred to as **an external or male thread. The thread in a nut or a** tapped hole is referred to as an internal or female thread. The terminology used to specify a screw thread is illustrated in Figure 1.13. The *pitch* is the distance between successive crests of the thread. The pitch in inches is the reciprocal of the number of threads per inch (tpi). The *major diameter* is the largest diameter of either an external or an internal thread. The *minor diameter* is the smallest diameter. In the United States, Britain, and Canada the form of a thread is specified by the Unified Standard. Both the *crest* and *root* of these threads are flat, as shown in Figure 1.13, or else slightly rounded. The *thread angle* is always 60°.

There are two thread series commonly used for instrument work. The *coarse-thread series*, designated UNC for Unified National Coarse, is for general use. Coarse threads provide maximum strength. The *fine-thread series*, designated UNF, is for use on parts subject to shock or vibration, since a tightened fine-thread nut and bolt are less likely to shake loose than a coarse-thread nut and bolt. The UNF thread is also used where fine adjustment is necessary.

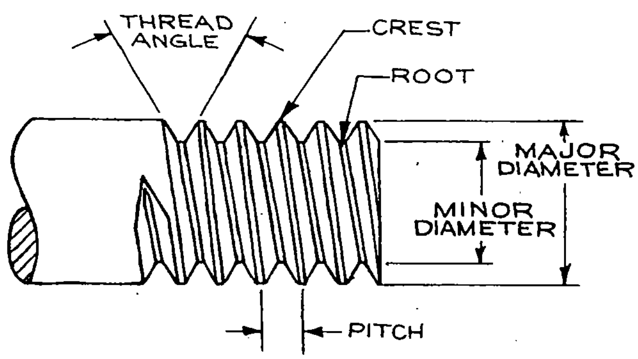


Figure 1.13 Screw-thread terminology.

The fit of threaded fasteners is specified by tolerances designated as 1A, 2A, 3A, and 5A for external threads and 1B, 2B, 3B, and 5B for internal threads. The fit of 2A and 2B threads is adequate for most applications, and such threads are usually provided if a tolerance is not specified. Many types of machine screws are available only with 2A threads. The 2A and 2B fits allow **sufficient clearance for plating. 1A and 1B fits leave** sufficient clearance that dirty and scratched parts can be easily assembled. 3A and 3B fits are for very precise work. 5A and 5B are interference fits such as are used on studs that are to be installed semipermanently.

The specification of threaded parts is illustrated by the following examples:

1. An externally threaded part with a nominal major diameter of $\frac{3}{8}$ in., a coarse thread of 16 threads per inch, and a 2A tolerance is

   $\frac{3}{8}$-16 UNC-2A.

2. An internally threaded part with a nominal major diameter of $\frac{5}{8}$ in., a fine thread of 18 tpi, and a 2B tolerance is designated

   $\frac{5}{8}$-18 UNF-2B.

3. Major diameters less than $\frac{1}{4}$ in. are specifed by a gauge number; thus a thread with a nominal major diameter of 0.164 in. and a coarse thread of 32 tpi is designated

   8-32 UNC.

   The specifications of the UNC and UNF thread forms are listed in Table 1.5.

Pipes and pipe fittings are threaded together. Pipe threads are tapered so that a seal is formed when an externally threaded pipe is screwed into an internally threaded fitting. The American Standard Pipe Thread, designated NPT for National Pipe Thread, has a taper of 1 in 16. The diameter of a pipe thread is specified by stating the nominal internal diameter of a pipe that will accept that thread on its outside. For example, a pipe with a nominal internal diameter of $\frac{1}{4}$ in. and a standard thread of 18 tpi is designated

$$\tfrac{1}{4}\text{-18 NPT}$$

or simply

$$\tfrac{1}{4}\text{-NPT.}$$

The American Standard Pipe Thread specifications are listed in Table 1.6.

The common forms of machine screws are illustrated in Figure 1.14. *Hex-head* cap screws are ordinarily available in sizes larger than $\frac{1}{4}$ in. They have a large bearing surface and thus cause less damage to the surface under the head than other types of screws. A large torque can be applied to a hex-head screw, since it is tightened with a wrench.

Slotted-head screws are available with *round*, *flat*, and *fillister* heads. The flat head is countersunk so that the top of the head is flush. Fillister screws are preferred to round-head screws, since the square shoulders of the head provide better support for the blade of a screwdriver.

Socket-head cap screws with a hexagonal recess are preferred for instrument work. These are also known as *Allen* screws. L-shaped, straight, and ball-pointed hex drivers are available that permit Allen screws to be installed in locations inaccessible to a wrench or screwdriver. An Allen screw can only be driven by a

Table 1.5 AMERICAN STANDARD UNIFIED AND AMERICAN NATIONAL THREADS

Size	*Coarse* (NC, UNC)		*Fine* (NF, UNF)	
(nominal diameter)	*Threads per Inch*	*Tap Drill*[a]	*Threads per Inch*	*Tap Drill*[a]
0 (0.060)			80	$\frac{3}{64}$
1 (0.073)	64	No. 53	72	No. 53
2 (0.086)	56	No. 50	64	No. 50
3 (0.099)	48	No. 47	56	No. 45
4 (0.112)	40	No. 43	48	No. 42
5 (0.125)	40	No. 38	44	No. 37
6 (0.138)	32	No. 36	40	No. 33
8 (0.164)	32	No. 29	36	No. 29
10 (0.190)	24	No. 25	32	No. 21
12 (0.216)	24	No. 16	28	No. 14
$\frac{1}{4}$	20	No. 7	28	No. 3
$\frac{5}{16}$	18	Let. F	24	Let. I
$\frac{3}{8}$	16	$\frac{5}{16}$	24	Let. Q
$\frac{7}{16}$	14	Let. U	20	$\frac{25}{64}$
$\frac{1}{2}$	13	$\frac{27}{64}$	20	$\frac{29}{64}$
$\frac{9}{16}$	12	$\frac{31}{64}$	18	$\frac{33}{64}$
$\frac{5}{8}$	11	$\frac{17}{32}$	18	$\frac{37}{64}$
$\frac{3}{4}$	10	$\frac{21}{32}$	16	$\frac{11}{16}$
$\frac{7}{8}$	9	$\frac{49}{64}$	14	$\frac{13}{16}$
1	8	$\frac{7}{8}$	12	$\frac{59}{64}$

Note: ASA B1.1-1960.

[a] For approximately 75% thread depth.

Table 1.6 AMERICAN STANDARD TAPER PIPE THREADS

Nominal Pipe Size	*Actual O.D. of Pipe*	*Threads per Inch*	*Normal Length of Engagement by Hand*	*Length of Effective Thread*
$\frac{1}{8}$	0.405	27	0.180	0.260
$\frac{1}{4}$	0.540	18	0.200	0.401
$\frac{3}{8}$	0.675	18	0.240	0.408
$\frac{1}{2}$	0.840	14	0.320	0.534
$\frac{3}{4}$	1.050	14	0.340	0.546
1	1.315	$11\frac{1}{2}$	0.400	0.682
$1\frac{1}{4}$	1.660	$11\frac{1}{2}$	0.420	0.707
$1\frac{1}{2}$	1.900	$11\frac{1}{2}$	0.420	0.724
2	2.375	$11\frac{1}{2}$	0.436	0.756
$2\frac{1}{2}$	2.875	8	0.682	1.136
3	3.500	8	0.766	1.200

Note: ASA B2.1-1960.

wrench of the correct size, so the socket does not wear so fast as the slot in a slotted-head screw. These screws have a relatively small bearing surface under the head and thus should be used with a washer.

Setscrews are used to fix one part in relation to another. They are often used to secure a hub to a shaft. In this application it is wise to put a flat on the shaft where the setscrew is to bear; otherwise the screw may mar the shaft, making it impossible for it to be withdrawn from the hub. Setscrews should not be used to lock a hub to a hollow shaft, since the force exerted by the screw will deform the shaft. In general, setscrews are suitable only for the transmission of small torque, and their use should be avoided if possible.

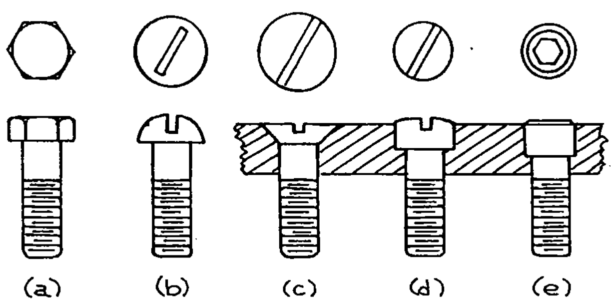


Figure 1.14 Common machine screws: (*a*) hex head; (*b*) round head; (*c*) flat head; (*d*) fillister head; (*e*) socket head.

Machine screws shorter than 2 in. are threaded their entire length. Longer screws are only threaded for part of their length. Screws are usually only available with class 2A coarse or fine threads.

The type of head on a screw is usually designated by an abbreviation such as "HEX HD CP SCR" or "FILL HD MACH SCR." For example, a socket-head screw $1\frac{1}{2}$ in. long with an 8-32 thread is designated

$$8\text{-}32\ \mathrm{UNC} \times 1\tfrac{1}{2}\mathrm{SOC\ HD\ CAP\ SCR.}$$

The torque T required to produce a tension load F in a bolt of diameter D is

$$T = CDF,$$

where the coefficient C depends upon the state of lubrication of the threads. In general, C may be taken as 0.2. If the threads are oiled or coated with molybdenum disulfide, a value of 0.15 may be more accurate. If the threads are very clean, C may be as large as 0.4. The tension load on the bolt is equal to the compressive force exerted by the underside of the bolt head.

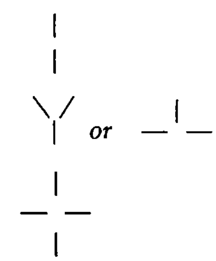
Steel bolts meeting SAE specifications are identified by markings on the head:

SAE grades 0, 1, 2: no mark

SAE grade 3:

SAE grade 5:

SAE grade 6:



The strength of the bolt increases with the SAE grade number. Most common steel bolts are SAE grade 2. In sizes up to 1 in., these bolts have a yield strength of about 50,000 psi. SAE grade 5 bolts have a yield strength of about 80,000 psi.

When a bolt is to be tightened to a specified torque, the threads should be first seated by an initial tightening. It should then be loosened and retightened to the computed torque with a torque wrench. The torque corresponding to the yield strength should not be exceeded during this operation.

Because a bolt is usually made of a fairly hard material, excessive wear may result if it is frequently screwed into a threaded (tapped) hole in a soft material such as plastic or aluminum. Threaded inserts (Heli-Coils) are made to alleviate this problem. These inserts are tightly wound helices of stainless steel or phosphor-bronze wire that have a diamond-shaped cross section. The insert is placed in a tapped oversize hole and the mating bolt is screwed into the insert. Special taps are required to prepare a hole for the insert, and a special tool is required to drive the insert into the hole.

To obtain maximum load-carrying strength, a steel bolt engaging an internal thread in a steel part should enter the thread to a distance equal to at least one bolt diameter. For a steel bolt entering an internal thread in brass or aluminum, the length of engagement should be closer to two bolt diameters.

1.3.5 Rivets

Rivets are used to permanently join sheet-metal or sheet-plastic parts together. They are frequently used when some degrees of flexibility is desired in a joint, as when joining the ends of a belt to give a continuous loop. The most common rivet shapes are shown in Figure 1.15. Rivets are made of soft copper, aluminum, or steel. To join two pieces, a hole, slightly larger than the body of the rivet, is drilled or punched in each piece. A rivet is inserted through the holes, and a head is formed on the plain end of the rivet using a hammer or, preferably, a riveting machine. The hammering action swells the body of the rivet to fill the hole.

"Pop" rivets, illustrated in Figure 1.15, are useful in the lab. These can be installed without access to the back side of the joint. The mandrel is grasped by a special rivet gun, the rivet is inserted into the hole, and the mandrel is pulled back until it breaks. The head of the mandrel rolls the stem of the rivet over to form a head, and the remaining broken portion of the mandrel seals the center hole of the rivet. The installation of pop rivets in carefully positioned, reamed holes is an excellent means of aligning two thin pieces with respect to one another. Such joints are only semipermanent. A pop rivet is easily removed by drilling into its center with an oversize drill until the head separates from the body. With care, the drill will never touch the parts joined by the rivet.

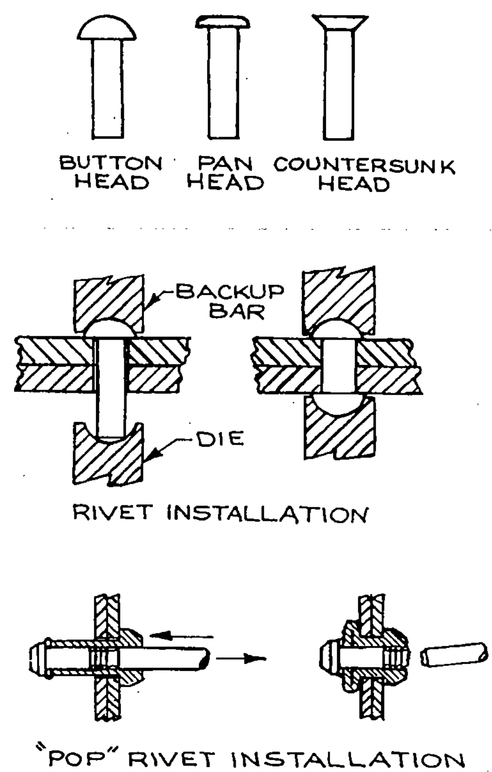


Figure 1.15 Rivets and riveted joints.

1.3.6 Pins

Pins such as are shown in Figure 1.16 are used to precisely locate one part with respect to another or to fix a point of rotation. To install a pin, the two pieces to be joined are clamped together and the hole for the pin is drilled and reamed.

Straight pins are made of steel that has been hardened and ground to a diameter tolerance of 0.0001 in. They

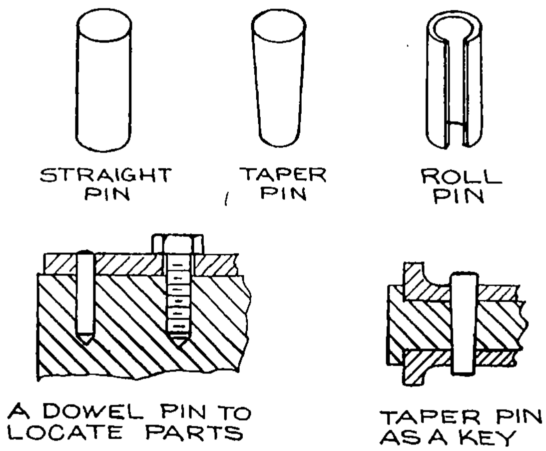


Figure 1.16 Pins.

require a hole that has been reamed to a close tolerance, since they are intended to be pressed into place. If a straight pin is to serve as a pivot or if it is to locate a part that is to be removable, the hole in the movable part is reamed slightly oversize.

A *roll pin* has the advantage of ease of installation and removal and does not require a precision-reamed hole. The spring action of the walls of the pin hold it in place.

Taper pins are installed in holes that are shaped with a special reamer. The taper is 0.250 in. per ft. Plain taper pins are driven into place. Taper pins with the small end threaded are drawn into place with a nut, which then secures the installed pin.

A *shoulder screw* is used as a pivot pin. These screws are hardened, and the shoulder is ground to a diameter tolerance of 0.0001 in.

1.3.7 Retaining Rings

A retaining ring serves as a removable shoulder on a shaft or in a hole to position parts assembled on the shaft or in the hole. An axially assembled external retaining ring is expanded slightly with a special pair of pliers, then slipped over the end of a shaft and allowed to spring shut in a groove on the shaft. An axially assembled internal ring is compressed, inserted in a hole and permitted to spring open into a groove. A radially assembled external retaining ring is forced onto a shaft from the side. It springs open as it slides over the diameter and then closes around the shaft.

Some retaining rings have a beveled edge, as shown in Figure 1.17, so that the spring action of the ring on the edge of its groove produces an axial load to take up unwanted clearances between assembled parts. This scheme is frequently used to take up the end play in a ball bearing. Another solution to end-play take-up is the use of a bowed retaining ring as shown in Figure 1.17.

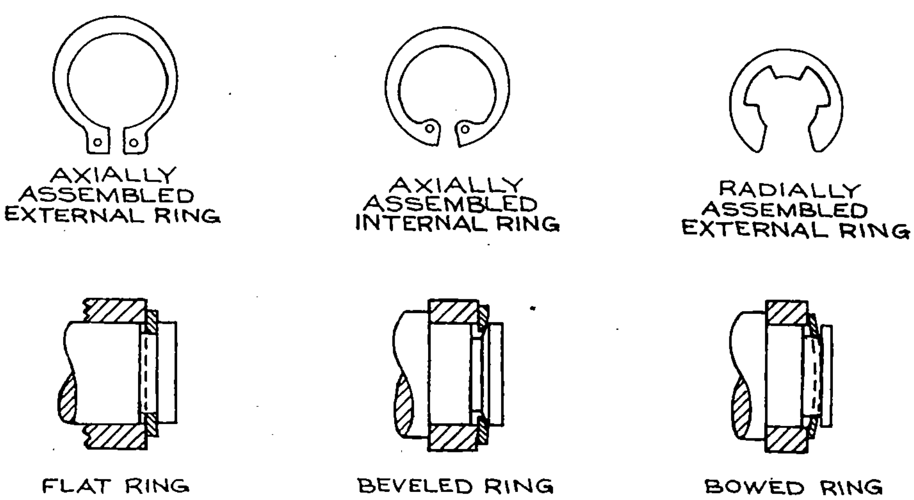


Figure 1.17 Retaining rings and retaining-ring installations.

1.3.8 Adhesives

Epoxy resins are the most universally applicable adhesives for the laboratory. They are available in a variety of strengths and hardnesses. Epoxies that are either thermally or electrically conducting are also available. Epoxy adhesives consist of a resin and a hardener, which are mixed just prior to use. The mixing proportions and curing schedule must be controlled to achieve the desired properties. Tra-Con and Devcon manufacture a variety of epoxy adhesives that are prepackaged in small quantities in the correct proportions.

Epoxies will adhere to metals, glass, and some plastics. Hard, smooth surfaces should be roughened prior to the application of the adhesive. Sandblasting is a convenient method. Parts of a surface that are not to receive the adhesive can be masked with tape during this operation. To obtain maximum strength in an epoxied joint, the gap filled by the epoxy should be 0.002–0.006 in. wide. To maintain this gap between flat smooth surfaces, shims can be inserted to hold them apart.

Self-curing silicone rubber such as General Electric RTV can be used as an adhesive. RTV is chemically stable and will stick to most surfaces. The cured rubber is not mechanically strong; however, this can be an advantage when making a joint that may occasionally have to be broken.

Cyanoacrylate contact adhesives have found wide use in instrument construction. Eastman 910 and Techni-Tool Permabond are representative of this type of adhesive. These adhesives are monomers that polymerize rapidly when pressed into a thin film between two surfaces. They will adhere to most materials, including metal, rubber, and nylon. Cyanoacrylate adhesives are not void-filling and only work to bond surfaces where contours are well matched. An adhesive film of about 0.001 in. gives best results. A firm set is achieved in about a minute, and maximum strength is usually reached within a day. When joining metals and plastics, a shear strength in excess of 1000 psi is possible.

Sauereisen manufactures a line of ceramic cements. These are inorganic materials in powder, paste, or liquid form that may be used to join pieces of ceramic or to bond ceramic to metal. Some are designed to be used as electrically insulating coatings and others may be used for casting small ceramic parts. The tensile strength of these materials is only of the order of 500 psi, but the materials are serviceable up to at least 1100°C.

1.4 MECHANICAL DRAWING

Mechanical drawing is the language of the scientific designer. Initial ideas for a design are best expressed in terms of simple, full-scale drawings that develop in complexity and detail as the design matures. The construction of an apparatus is realized through communications to shop personnel in the form of working drawings of each part of the device. Success for the designer depends in large part on his command of the language.

1.4.1 Drawing Tools

A scientist who spends any more than five or ten percent of his time on apparatus design should acquire the set of tools listed in Table 1.7. The drawing board should be located in a well-lighted corner of an office or laboratory. Light from a north-facing window is most desirable. The drawing tools should be stored nearby. A scientist who does a considerable amount of work at the drawing board should consider acquiring a regular drafting table and draftsman's stool. A drafting machine for the drawing board instead of a T-square will improve efficiency considerably. It is also helpful to cover the drawing surface with graph sheet ruled in 0.1-in. squares or with a ruled surface sheet such as K&E Laminene.

1.4.2 Basic Principles of Mechanical Drawing

If mechanical drawing is the designer's language, then the line is the alphabet of this language. Some of the basic lines used in pencil drawing are illustrated in the

In the above discussion we have assumed that the scientific designer has the services of a model shop. This is the case in industrial laboratories and in many university laboratories. However, a designer should follow the same procedure when he fabricates his own apparatus. All questions of sizes, tolerances, and fits must be answered before construction. When a scientist attempts to make these decisions as he proceeds with construction, the results are inevitably poor.

1.5 PHYSICAL PRINCIPLES OF MECHANICAL DESIGN

No material is perfectly rigid. When any member of a machine is subjected to a force, no matter how small, it will bend or twist to some extent. A member subjected to a force that varies in time will vibrate. A designer must appreciate the extent of deflection of mechanical parts under load.

1.5.1 Bending of a Beam or Shaft

When a beam bends, one side of the beam experiences a tensile load and the other a compressive load. Consider a point in the flexed beam in Figure 1.36. The stress at this point depends upon its distance, c, from the centroid of the beam in the direction of the applied force. The *centroid* is the center of gravity of the cross section of the beam [see Figure 1.36(b)]. The stress is given by

$$s = \frac{Mc}{I},$$

where M is the bending moment at the point of interest and I is the centroidal moment of inertia of the section of the beam. Clearly the stress in a flexed beam is greatest at the outer surface of the beam.

The *centroidal moment of inertia* of the section (more correctly known as the second moment of the area of the cross section) is taken about an axis that passes through the centroid and in the direction perpendicular to the applied force. For the rectangular section in Figure 1.36(b)

$$I = \int_{-B/2}^{B/2} \int_{-H/2}^{H/2} h^2\, dh\, db = \frac{BH^3}{12},$$

where B is the width of the section and H is the height. The centroidal moments of inertia of some common symmetrical sections are given in Figure 1.37.

The *bending moment* is proportional to the curvature produced in a beam by the applied force:

$$M = EI\frac{d^2y}{dx^2},$$

where y is the deflection produced by the force and E is the modulus of elasticity of the beam. For the example given in Figure 1.36, assuming the weight of the shaft can be ignored, the bending moment is simply

$$M = -F(L - x).$$

The bending moments for this and other common systems are given in Figure 1.38.

The deflection of a stressed beam depends upon the modulus of elasticity of the material. Expressions for the deflection of both point-loaded and uniformly loaded beams are given in Figure 1.38.

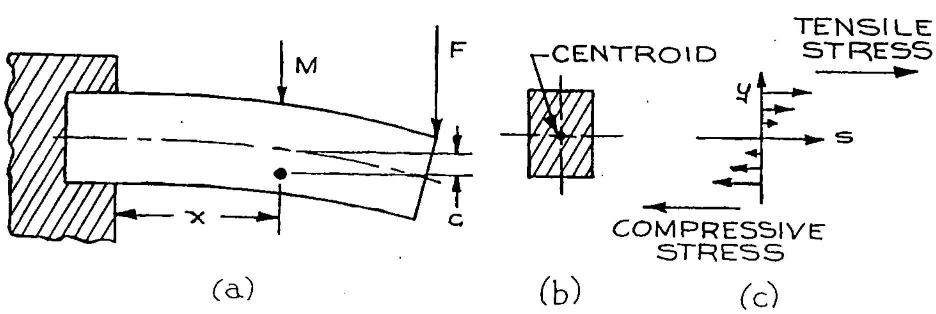
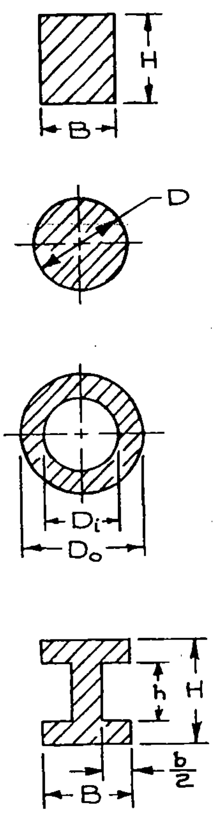


Figure 1.36 A flexed beam: (a) the beam bending under a load; (b) a cross section of the beam showing the centroid; (c) the distribution of shear forces along a vertical line through the centroid of the beam.



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

$$I = \frac{\pi D^4}{64}$$

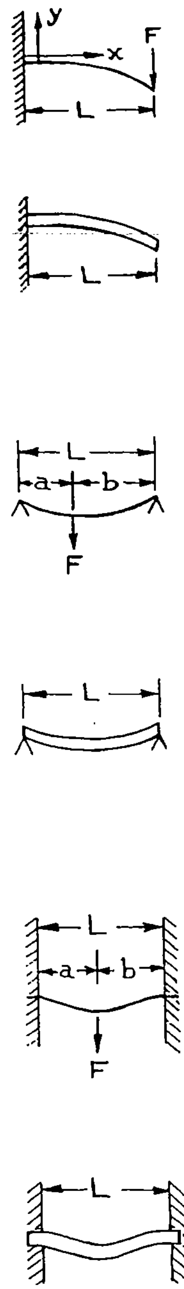
$$I = \frac{\pi}{64}\left(D_o^4 - D_i^4\right)$$

$$I = \frac{1}{12}\left(BH^3 - bh^3\right)$$

Figure 1.37 Centroidal moments of inertia.

If the bending moment on a beam is sufficiently large, it will be permanently deformed. A conservative relation between the bending moment at the yield point, M_y, and the yield strength of the material, s_y, is given by $M_y = s_y I / c_{\max}$. Actually, the yield strength for a beam in bending is greater than for a beam in tension, since the inner fibers of the material must be loaded to their yield point to a considerable depth before the beam is permanently deformed. To determine the maximum allowable force on a beam or shaft one simply sets M_y equal to the expression for the maximum value of the bending moment on the beam and solves for the force.

The preceding discussion assumes that the tensile strength of a material is the same as the compressive



$$M = -F(L-x)$$
$$y = -\frac{F}{6EI}\left(3Lx^2 - x^3\right)$$

$$M = -\frac{w}{L}(L-x)^2$$
$$y = -\frac{w}{24EI}\left(6L^2x^2 - 4Lx^3 + x^4\right)$$

in length a

$$M = \frac{Fbx}{L}$$
$$y = -\frac{Fbx}{6EIL}\left(L^2 - b^2 - x^2\right)$$

in length b

$$M = \frac{Fa}{L}(L-x)$$
$$y = -\frac{Fa(L-x)}{6EIL}\left[L^2 - a^2 - (L-x)^2\right]$$

$$M = \frac{wL}{2}\left(x - \frac{x^2}{L}\right)$$
$$y = -\frac{wx}{24EI}\left(L^3 - 2Lx^2 + x^3\right)$$

in length a

$$M = \frac{Fb^2}{L^3}(bx + 3ax - aL)$$
$$y = -\frac{Fb^2}{6EIL^3}\left(3aLx^2 - 3ax^3 - bx^3\right)$$

in length b

$$M = \frac{Fa^2}{L^3}(2bL - 3bx + aL - ax)$$
$$y = -\frac{Fa^2(L-x)^2}{6EIL^3}\left[3bL - (3b+a)(L-x)\right]$$

$$M = -\frac{wL^2}{12} + \frac{wLx}{2} - \frac{wx^2}{2}$$
$$y = -\frac{wx^2}{24EI}\left(L^2 + x^2 - 2Lx\right)$$

Figure 1.38 Bending formulae: M = bending moment; E = modulus of elasticity; I = centroidal moment of inertia; w = weight per unit length.

strength. For materials such as cast iron, where this is not true, the bending equations are much more complicated. The foregoing also ignores shear stresses in a loaded beam. For very short beams, shear stresses become important and the shear strength of the material must be considered.[4]

1.5.2 Twisting of a Shaft

The stress at a point in a round shaft subjected to a torsional load is

$$s = \frac{Tc}{J}, \qquad J = \frac{\pi}{32}\left(D_o^4 - D_i^4\right),$$

where c is the distance of the point of interest from the center of the shaft and T is the applied torque. J is the centroidal polar moment of inertia of the section of the shaft, and D_o and D_i are the outer and inner diameters of the shaft.

The total angle of twist in a shaft of length L is

$$\theta = \frac{57LT}{GJ} \text{ degrees,}$$

where G is the modulus of elasticity in shear or *shear modulus* of the material of the shaft. For metals, the shear modulus is about $\frac{1}{3}$ the elastic modulus.

1.5.3 Stress Relief

An abrupt change in the cross section of a shaft produces a concentration of stresses as shown in Figure 1.39(a). Stresses are increased at steps, grooves, keyways, holes, dents, and scratches. A stress concentration is an area where there is a large gradient in the stress. Material tends to fail because of the shear forces at a stress concentration. Anyone who has ever broken a bolt is familiar with this effect: the bolt invariably fails just where the head joins the shaft. Methods of relieving stresses at a step in a shaft are suggested in Figure 1.39(b), (c), and (d).

There are a number of means of detecting stress in a part. One of the most useful is the *photoelastic method*.[5]

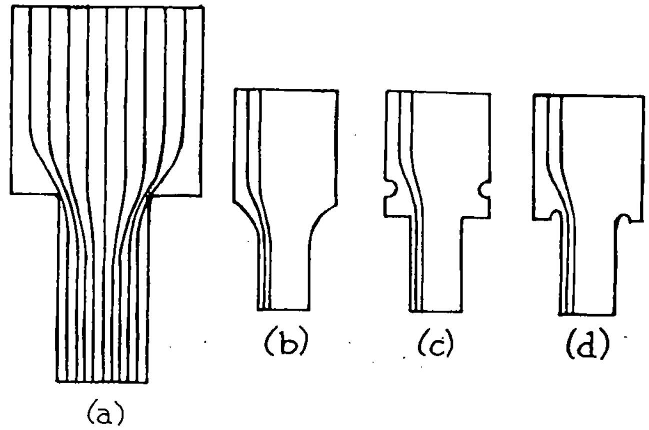


Figure 1.39 (a) Stress concentrations in a shaft. Lines indicate surfaces of constant stress. Parts (b), (c), and (d) show methods of relieving stresses at a step in a shaft; (c) and (d) are useful when a shoulder is required to locate a bearing.

In this technique a transparent plastic model is stressed in the same way as the element of interest without necessarily duplicating the magnitude of the stress. The model is illuminated with polarized, monochromatic light and viewed through a polarizer. As shown in Figure 1.40, the stress distribution appears as fringes in

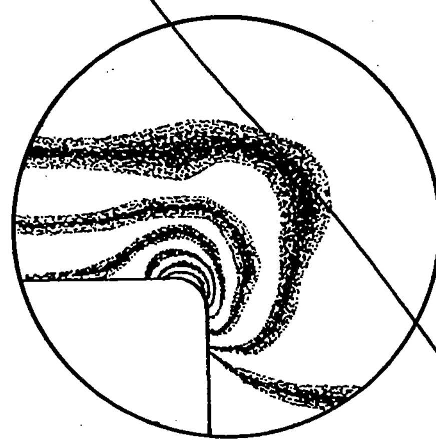


Figure 1.40 Stress distribution in a plastic model as observed by the photoelastic method.

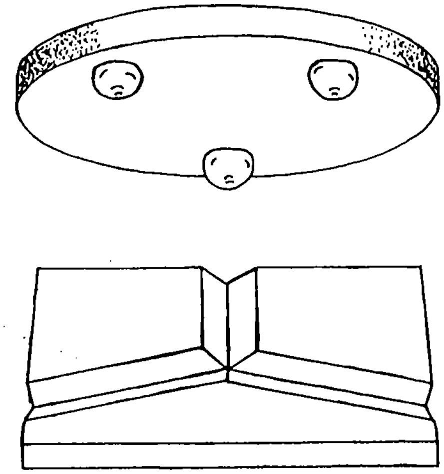


Figure 1.44 Kinematic design that permits an accurately located part to be removed and replaced in the same position.

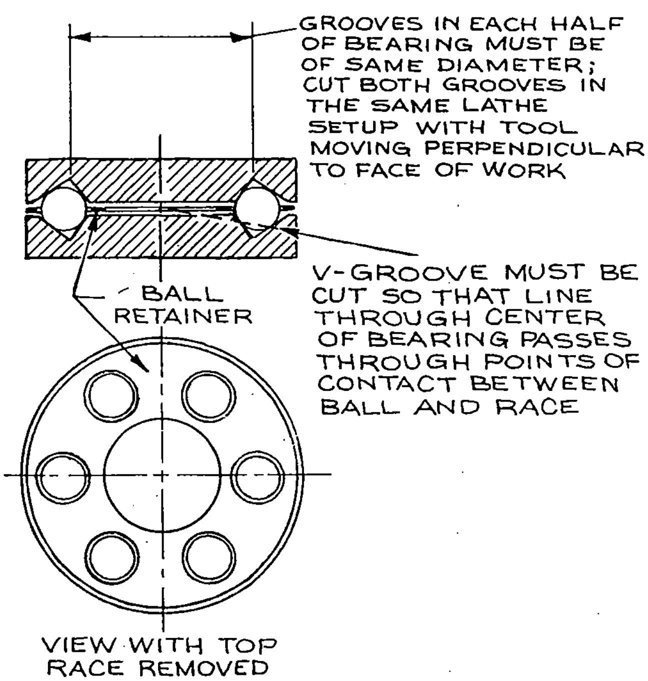


Figure 1.45 Semikinematic design that allows rotation about a single axis. If the grooves are cut as indicated, the balls will always be in rolling contact with the grooves.

tolerance of ± 0.0001 in. on the locat
part.

The design of Figure 1.45 can l
bearing must be constructed of exo
example, in high-vacuum work, ball bearings
ceramic or sapphire balls and stainless-steel races are used because steel balls in a steel race tend to cold-weld when they are very clean, as they would be for a vacuum application. In practice, more than three balls can be used. Since this is a semikinematic design, the addition of more balls will increase the bearing strength without significantly decreasing accuracy.

1.6.2 Plain Bearings

A bearing is a stationary element that locates and carries the load of a moving part. Bearings can be divided into two categories depending upon whether there is sliding or rolling contact between the moving and stationary parts. Sliding-contact bearings are called *plain bearings*. Rolling bearings will be discussed in the next section.

Plain bearings may be designed to carry a radial load, an axial load, or both. Different types are illustrated in Figure 1.46. A radial bearing consists of a cylindrical shaft or *journal* rotating or sliding within a shell, which is the *bearing* proper. The entire assembly is referred to as a *journal bearing*. An axial bearing consists of a flat bearing surface, like a washer, against which the end of the shaft rests. These are called *thrust bearings*. A journal bearing may incorporate a *flanged journal*, in

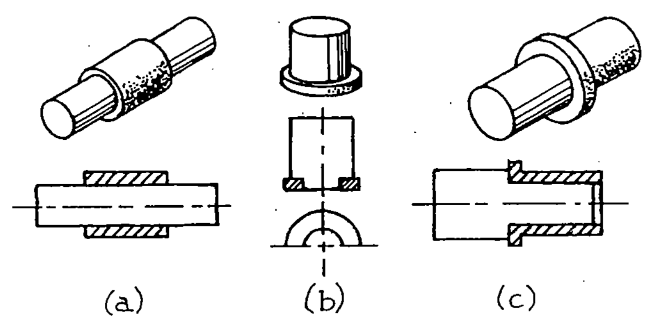


Figure 1.46 Different types of plain bearings: (*a*) a journal bearing; (*b*) a thrust bearing; (*c*) a flanged journal bearing.

which case it will support a radial load as well as an axial load.

A journal is usually hardened steel or stainless steel. Precision-ground shafts and shafts with precision-ground journals are available in diameters from $\frac{1}{32}$ to 1 in. Bearing shells of bronze or oil-impregnated bronze are available to fit. Nylon bearings for light loads and oil-free applications are also available. Commercial shafting and bearings are manufactured to provide a clearance of 0.0002–0.0010 in.

Many different methods of lubrication can be employed instead of oil impregnation. The inner surface of the bearing can be grooved, and oil or grease can be forced into the groove through a hole in the shell. If oil is objectionable, a groove on the inner surface of the bearing can be packed with molybdenum disulfide or other dry lubricant.

A plain bearing is installed by pressing it into a hole in the supporting structure. An interference of about 0.001 in. is desirable for bearings up to an inch in diameter. That is, the outer diameter of the bearing shell should be about 0.001 in. larger than the hole into which it is pressed. If the interference is too great, the inner diameter of the bearing may become significantly reduced.

A variety of *bearing housings* and *pillow blocks* (Figure 1.47) are available. These mountings are bored to accept standard bearings and in many instances the bearing is premounted. These mounts replace precision-bored bearing mounts.

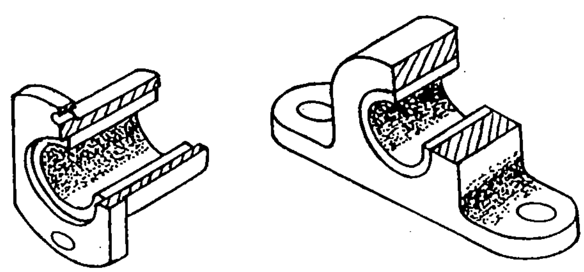
It is usually necessary to provide axial location for a shaft in a journal bearing. This can be accomplished with a retaining ring in a groove on the shaft or a collar secured by a setscrew.

Plain bearings run smoothly and quietly, and have a high load-carrying ability. Properly installed and lubricated, they have a very long life. Because of the close clearances between parts, they are not easily fouled by dirt in their environment. However, they are limited to low-speed operation. Speeds in excess of a few hundred rpm are not practical without forced lubrication. The primary disadvantage of plain bearings is their high starting friction, although, when properly installed and lubricated, their running friction can be very low.

1.6.3 Ball Bearings

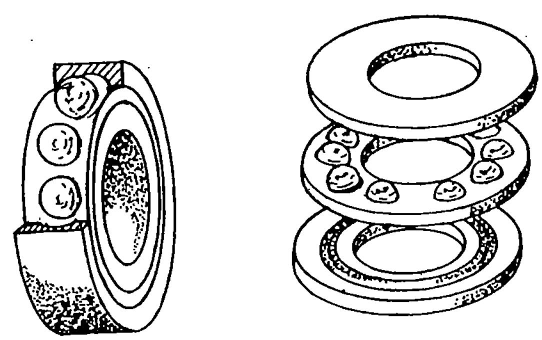
The rolling element in a rolling contact bearing may be a ball, cylinder, or cone. Ball bearings are used for light loads and high speeds. Roller bearings, which employ cylindrical or conical rollers, are suitable for very heavy loads and are not often used for instrument work. We shall discuss only ball bearings.

As with plain bearings, there are both radial and thrust ball bearings (Figure 1.48). A *radial ball bearing* consists of an inner and an outer *race* with a row of balls between. The grooves in each race have a radius slightly larger than the radius of the balls so that there is only point contact between the balls and the race. The balls are separated by a *retainer* which prevents the balls from rubbing against one another and keeps them uni-



(a) (b)

Figure 1.47 (*a*) A bearing housing; (*b*) bearing mounted in a pillow block.



(a) (b)

Figure 1.48 Ball bearings: (*a*) a radial ball bearing; (*b*) a thrust ball bearing.

formly spaced around the bearing. A radial ball bearing can tolerate a substantial thrust load, but for pure axial loads a thrust bearing should be used. A *thrust bearing* is similar to an axial bearing except that it has upper and lower races rather than inner and outer.

Ball bearings are made of steel or stainless steel. They are graded 1, 3, 5, 7 or 9 depending on manufacturing tolerances. Grades 7 and 9 have ground races and are made to the closest tolerances. They cost little more than lower-grade ones, and they should be specified for instrument applications.

Proper installation is required to obtain good performance from a ball bearing. The rotating race should be given a firm interference fit, and the stationary race given a light "push fit" to permit some rotational creep. This slight movement of the stationary race helps prevent the maximum load from always bearing on the same spot.

Press fitting changes the internal clearances in a bearing. Bearing manufacturers specify the amount of interference that should be used. As shown in Figure 1.49, the press arbor used to drive a bearing onto a shaft or into a housing should be designed so that the thrust is not transmitted through the balls. Never hammer a bearing into place.

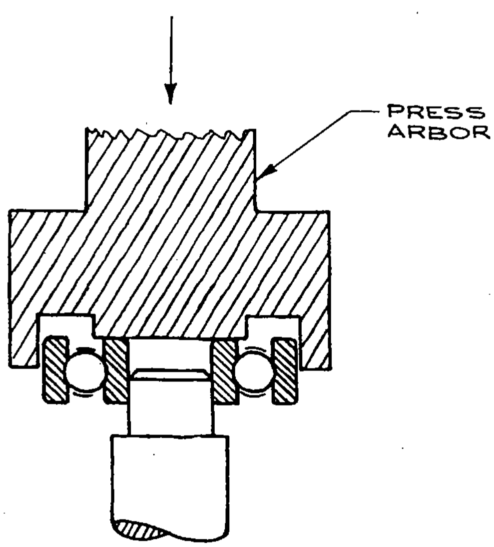


Figure 1.49 Installation of a bearing. The press arbor should bear on the race that is being fitted.

The surface quality and diameter tolerance of the shaft that is to be fitted into a bearing, or the hole that is to house a bearing, must be of the same quality as the bearing. For high-quality bearings the mounting surfaces should be ground. Fortunately for the builder of scientific apparatus, centerless-ground precision shafting is available to fit all standard bearings. Bearing mounts and pillow blocks with premounted bearings are similarly available to eliminate the need for precision machine work when installing ball bearings.

Ball bearings are designed with both radial and axial clearances. This play is intended to allow for axial misalignment and for dimensional changes that occur upon installation or because of thermal stresses. For the best locational precision and to obtain smooth, vibrationless operation, a ball bearing should be *preloaded* to remove most of this play. A preloading force that displaces one race axially with respect to the other will remove both radial and axial play by causing the balls to roll up the sides of their grooves. The use of a shim to take up the play in a bearing is illustrated in Figure 1.50. For light-duty applications, the end play can also be taken up by installing a spring washer (Section 1.6.5) instead of a shim. With proper installation, a ball bearing will locate a revolving axis to within a few ten-thousandths of an inch.

Bearings must be protected from effects that will damage the race surface on which the balls roll. Ball bearings are most likely to fail because of large static loads, which produce an indentation in the race. Such a

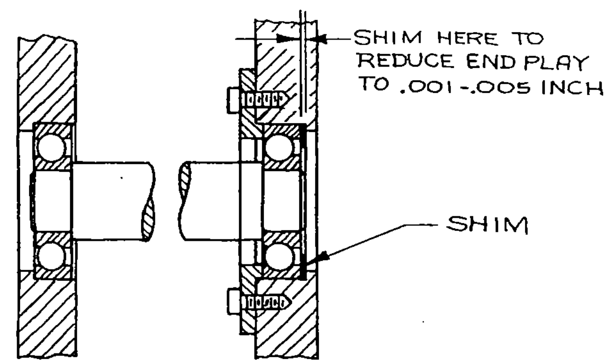


Figure 1.50 Installation of a shim to remove play in a shaft mounted in ball bearings.

dent is called a *brinell*. Dynamic loads are distributed around the race and are thus less likely to cause damage; however, a hard vibration can cause brinelling in the form of a series of dents or waves on the surface of a race. Of course, a bearing is also damaged by the introduction of foreign matter which abrades or corrodes the bearing surfaces.

Bearings should be lubricated with petroleum oils or greases. For high speeds and light loads the lightest, finest grades of machine oil can be used. Ball bearings require very little oil. Lubrication is sufficient if there is enough oil to produce an observable meniscus at the point where each ball contacts a race.

Cleanliness is important. In a dirty environment, bearings with built-in side shields should be used. It is probably wise to use enclosed bearings in all instrument applications to keep the bearings clean and to prevent oil from contaminating the environment.

The chief advantages of ball bearings are their low starting friction and very low running friction. They are well suited to high-speed, low-load operation. Relative to plain bearings, ball bearings are noisy and occupy a large volume. The cost of quality ball bearings is so low that economic considerations are usually not important in choosing between rolling bearings and plain bearings for instrument use.

1.6.4 Linear-Motion Bearings

A linear-motion bearing may be of either the sliding or rolling type. A plain journal bearing can be used to locate a shaft that is to move axially. A V-shaped or dovetail groove sliding over a mating rail can serve as a bearing between a heavily loaded, slowly moving carriage and a stationary platform. This is the type of bearing used between the carriage and bed of a lathe.

Recently, linear ball bearings have become commercially available. In a bearing for use with an axially moving shaft, the balls that carry the load between the outer race and the shaft move in grooves that run parallel to the axis of the shaft. The balls are recirculated through a return track when they roll to the end of a groove. Linear-motion ball bearings will locate a shaft to within ± 0.0002 in. of a reference axis. Their cost is comparable to conventional rotating ball bearings. Complete roller-slide assemblies are also available. These employ balls rolling in V-grooves. Roller slides will maintain straight-line motion to within 0.0002 in. per inch of travel.

1.6.5 Springs

In many instances it is desirable for a motion to be constrained by a flexible element such as a spring. Springs are used to hold two parts in contact when zero clearance is required, to absorb shock loads, to damp vibrations, and to measure forces.

A spring is characterized by the ratio of the magnitude of an applied force to the resulting deflection. This is the *spring rate*

$$k = \frac{F}{\delta}.$$

As is the case for any flexible system, an assembly consisting of a spring and an *attached* load has a natural frequency of vibration

$$f_n = \frac{1}{2\pi}\left(\frac{k}{m}\right)^{1/2},$$

where m includes the mass of both the spring and any load that is affixed to it. Since $m = W/g = F/a$, we have upon substituting for the spring rate in this equation

$$f_n = \frac{1}{2\pi}\left(\frac{g}{\delta_{st}}\right)^{1/2},$$

where δ_{st} is the static deflection produced by the weight of the spring plus the attached load.

In most applications it is desirable to choose a spring that will not resonate with any other part of the apparatus in which it is installed. If the spring is expected to damp a vibratory motion, its natural frequency should differ from that of the disturbance by more than an order of magnitude.

A spring will exert an uneven force when it is subjected to a periodically varying load whose frequency is close to the natural frequency of the spring. When the vibration of the spring is in phase with the load, the reactive force of the spring will be less than its static force for any given deflection. When the spring is out of phase with the load, it will exert a greater force than expected. This phenomenon is called *surge*. Surge can be reduced or eliminated by using two springs with different natural frequencies. In the case of helical springs, they can be placed one inside the other.

In instrument work, helical springs are most often used. *Helical compression, extension,* and *torsion* springs are illustrated in Figure 1.51. The number of coils in a helical spring must be sufficient to ensure that the spring wire remains within its elastic limit when the spring is at maximum deflection. The number of coils in a compression spring also determines the minimum length, which is realized when successive coils come into contact. A long compression spring may buckle under stress. This tendency is discouraged if the ends of the spring are square. It can be prevented by placing a rod through the center or by installing the spring in a hole. In general, a compression spring must be supported by some means if its length exceeds its diameter by more than a factor of five.

The spring rate of a helical spring made of round wire is

$$k = \frac{Gd^4}{8D^3N},$$

where G is the shear modulus of the spring material (about one-third of the elastic modulus), d the diameter of the wire, N the number of coils, and D the mean diameter (the average of the inner and outer diameters

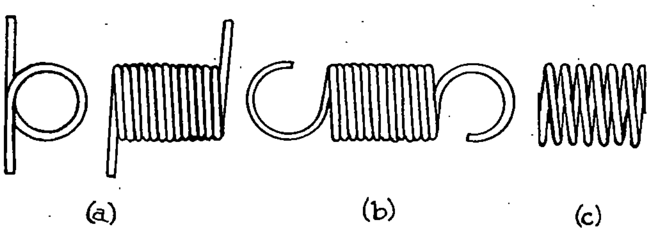


Figure 1.51 Springs: (*a*) helical compression spring; (*b*) extension spring; (*c*) torsion spring.

of the spring). The natural frequencies for free vibration are

$$f_n = \frac{nd}{4\pi D^2 N}\left(\frac{G}{2\rho}\right)^{1/2} \text{ Hz},$$

where ρ is the mass density of the spring wire and n is an integer. For steel wire

$$f_n = \frac{14000\,nd}{ND^2} \text{ Hz},$$

when d and D are in inches.

A variety of steels and bronzes are used for spring manufacture. High-carbon steel wire works well. If necessary the spring can be wound in the annealed state and hardened after forming. Music wire, or piano wire, is one of the best materials for one-off construction of small springs. It is available in diameters of 0.004 to 0.103 in. This wire is very strong and hard, because of the drawing process used in its production, and does not need to be hardened after forming. Type-302 stainless-steel wire is useful for springs that are subject to a corrosive environment. Springs of beryllium-copper wire are especially useful in applications where spring deflection is used to gauge a force, since this material maintains a linear stress-strain relation almost to the point of permanent deformation. As mentioned earlier, quartz fiber torsion springs can also be used in these applications.

Helical coil springs are conveniently formed by winding wire on a mandrel, which is rotated in a lathe. The wire must be kept under tension as it is pulled onto the mandrel, and when this tension is released the formed spring will expand. Thus the mandrel must be somewhat smaller than the desired inner diameter of the finished spring. The production of a small number of springs of a given size and spring rate is probably best carried out by cut and try.

Commercially manufactured springs are readily available. They are convenient to use because such properties as the spring rate and free length are specified by the supplier.

There are hundreds of possible spring configurations; however, *disc* springs are the only form that we shall

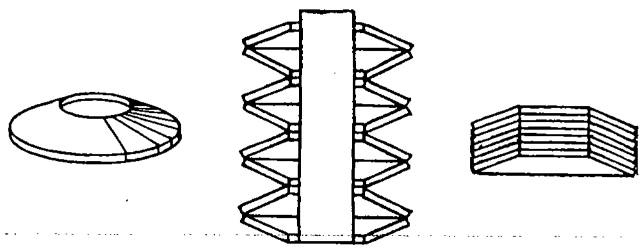


Figure 1.52 Belleville spring washers: (*a*) a disc spring; (*b*) stacked disc springs; (*c*) disc springs stacked in parallel.

mention other than coil springs. A disc spring, also known as a Belleville spring washer, is a cone-shaped disc with a hole in the center [Figure 1.52(*a*)]. When loaded, the cone flattens. This is a very stiff spring and thus can absorb a large amount of energy per unit length. A spring of any desired travel can be created by stacking disc springs as in Figure 1.52(*b*). They must be aligned by a rod passing through their centers or else by stacking them in a hole slightly larger than the outer diameter of the discs. If the discs are stacked in parallel as shown in Figure 1.52(*c*), they will provide a great deal of damping owing to friction between the faces of the discs.

CITED REFERENCES

1. *Metals Handbook*, *Heating Treating, Cleaning and Finishing*, Vol. 2, 8th edition, American Society for Metals, Metals Park, Ohio, 1964.
2. F. Rosebury, *Handbook of Electron Tube and Vacuum Techniques*, Addison-Wesley, Reading, Mass., 1965.
3. A comprehensive list of brazing alloys has been compiled by W.H. Kohl, in *Handbook of Vacuum Physics*, Vol. 3, *Technology*, A.H. Beck, Ed., Pergamon Press, Elmsford, N.Y., 1964; *Materials and Techniques for Electron Tubes*, Reinhold, New York, 1959.
4. R.J. Roark, *Formulas for Stress and Strain*, 4th edition, McGraw-Hill, New York, 1965.
5. J.W. Dally and W.F. Riley, *Experimental Stress Analysis*, McGraw-Hill, New York, 1965, pp. 165–221.
6. H.H. Mobie and F.W. Ocvirk, *Mechanisms and Dynamics of Machinery*, 2nd edition, Wiley, New York, 1963, pp. 489–491.
7. A useful review of kinematic design is given by J.E. Furse, J. Phys. E., 14, 264, 1981.
8. Carl Pelander of the JILA Instrument Shop of the University of Colorado is responsible for this groove design.

GENERAL REFERENCES

Design of Moving and Rotating Machinery

R. M. Phelan, *Dynamics of Machinery*, McGraw-Hill, New York, 1967.

Machine-Shop Manual

J. L. Feirer and E. A. Tatro, *Machine Tool Metalworking*, McGraw-Hill, New York, 1961.

Mechanical-Drawing Texts

T. E. French and C. J. Vierck, *Engineering Drawing and Graphic Technology*, 11th edition, McGraw-Hill, New York, 1972; (entitled *A Manual of Engineering Drawing* in its 1st–10th editions).

F. E. Giesecke, A. Mitchell, H. C. Spencer, and I. L. Hill, *Technical Drawing*, 5th edition, Macmillan, New York, 1967.

Practical Mechanical-Engineering Texts

A. D. Deutschman, W. J. Michels, and C. E. Wilson, *Machine Design*, Macmillan, New York, 1975.

V. M. Faires, *Design of Machine Elements*, 4th edition, Macmillan, New York, 1965.

R. E. Parr, *Principles of Mechanical Design*, McGraw-Hill, New York, 1970.

R. M. Phelan, *Fundamentals of Mechanical Design*, 3rd edition, McGraw-Hill, New York, 1970.