31 Mechanical Assembly

Chapter Contents

Mechanical assembly uses various methods to mechanically attach two (or more) parts together. In most cases, the method involves the use of discrete hardware components, called ***fasteners***, that are added to the parts during the assembly operation. In other cases, the method involves the shaping or reshaping of one of the components being assembled, and no separate fasteners are required. Many consumer products are produced using mechanical assembly: automobiles, large and small appliances, telephones, furniture, utensils—even wearing apparel is "assembled" by mechanical means. In addition, industrial products such as airplanes, machine tools, and construction equipment almost always involve mechanical assembly.

Mechanical fastening methods can be divided into two major classes: (1) those that allow for disassembly, and (2) those that create a permanent joint. Threaded fasteners (e.g., screws, bolts, and nuts) are examples of the first class, and rivets illustrate the second. There are good reasons why mechanical assembly is often preferred over other joining processes discussed in previous chapters. The main reasons are (1) ease of assembly and (2) ease of disassembly (for the fastening methods that permit disassembly).

Mechanical assembly is usually accomplished by unskilled workers with a minimum of special tooling and in a relatively short time. The technology is simple, and the results are easily inspected. These factors are advantageous not only in the factory, but also during field installation. Large products that are too big and heavy to be transported completely assembled can be shipped in smaller subassemblies and then put together at the customer's site.

Ease of disassembly applies, of course, only to the mechanical fastening methods that permit disassembly. Periodic disassembly is required for many products so that maintenance and repair can be performed; for example, to replace worn-out components, to make adjustments, and so forth. Permanent joining techniques such as welding do not allow for disassembly.

For purposes of organization, mechanical assembly methods are divided into the following categories: (1) threaded fasteners, (2) rivets, (3) interference fits, (4) other mechanical fastening methods, and (5) molded-in inserts and integral fasteners. These categories are described in Sections 31.1 through 31.5. Section 31.6 discusses the important topic of design for assembly. Assembly of electronic products includes mechanical techniques. However, electronics assembly represents a specialized field, which is covered in Chapter 34.

31.1 Threaded Fasteners

Threaded fasteners are discrete hardware components that have external or internal threads for assembly of parts. In nearly all cases, they permit disassembly. Threaded fasteners are the most important category of mechanical assembly; the common threaded fastener types are screws, bolts, and nuts.

31.1.1 SCREWS, BOLTS, AND NUTS

Screws and bolts are threaded fasteners that have external threads. There is a technical distinction between a screw and a bolt that is often blurred in popular usage. A ***screw*** is an externally threaded fastener that is generally assembled into a blind threaded hole. Some types, called ***self-tapping screws***, possess geometries that permit them to form or cut the matching threads in the hole. A ***bolt*** is an externally threaded fastener that is inserted through holes in the parts and "screwed" into a nut on the opposite side. A ***nut*** is an internally threaded fastener having standard threads that match those on bolts of the same diameter, pitch, and thread form. The typical assemblies that result from the use of screws and bolts are illustrated in Figure 31.1.

Screws and bolts come in a variety of standard sizes, threads, and shapes. Table 31.1 provides a selection of common threaded fastener sizes in metric units (ISO standard) and U.S. customary units (ANSI standard).[1] The metric specification consists of the nominal major diameter, mm, followed by the pitch, mm. For example, a specification of 4×0.7 means a 4.0-mm major diameter and a pitch of 0.7 mm. The U.S. standard specifies either a number designating the major diameter (up to 0.2160 in) or the nominal major diameter, in, followed by the number of threads per inch. For example, the specification 1/4-20 indicates a major diameter of 0.25 in and 20 threads per inch. Both coarse pitch and fine pitch standards are given in Table 31.1.

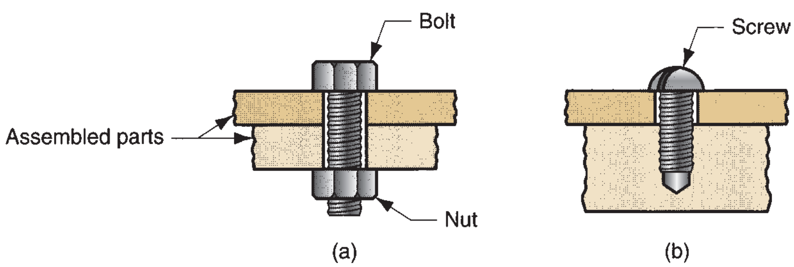

FIGURE 31.1 Typical assemblies using (a) bolt and nut, and (b) screw.

[1] ISO is the abbreviation for the International Standards Organization. ANSI is the abbreviation for the American National Standards Institute.

TABLE • 31.1 Selected standard threaded fastener sizes in metric and U.S. customary units.

ISO (Metric) Standard			ANSI (U.S.C.S) Standard			
Nominal Diameter, mm	Coarse Pitch, mm	Fine Pitch, mm	Nominal Size	Major Diameter, in	Threads/in, Coarse (UNC)[a]	Threads/in, Fine (UNF)[a]
2	0.4		2	0.086	56	64
3	0.5		4	0.112	40	48
4	0.7		6	0.138	32	40
5	0.8		8	0.164	32	36
6	1.0		10	0.190	24	32
8	1.25		12	0.216	24	28
10	1.5	1.25	1/4	0.250	20	28
12	1.75	1.25	3/8	0.375	16	24
16	2.0	1.5	1/2	0.500	13	20
20	2.5	1.5	5/8	0.625	11	18
24	3.0	2.0	3/4	0.750	10	16
30	3.5	2.0	1	1.000	8	12

[a] UNC stands for unified coarse in the ANSI standard, and UNF stands for unified fine.

Additional technical data on these and other standard threaded fastener sizes can be found in design texts and handbooks. The United States has been gradually converting to metric thread sizes, which will reduce proliferation of specifications. It should be noted that differences among threaded fasteners have tooling implications in manufacturing. To use a particular type of screw or bolt, the assembly worker must have tools that are designed for that fastener type. For example, there are numerous head styles available on bolts and screws, the most common of which are shown in Figure 31.2. The geometries of these heads, as well as the variety of sizes available, require different hand tools (e.g., screwdrivers) for the worker. One cannot turn a hexhead bolt with a conventional flat-blade screwdriver.

Screws come in a greater variety of configurations than bolts, since their functions vary more. The types include machine screws, capscrews, setscrews, and self-tapping screws. ***Machine screws*** are the generic type, designed for assembly into tapped holes. They are sometimes assembled to nuts, and in this usage they overlap with bolts. ***Capscrews*** have the same geometry as machine screws but are made of higher strength metals and to closer tolerances. ***Setscrews*** are hardened and designed for

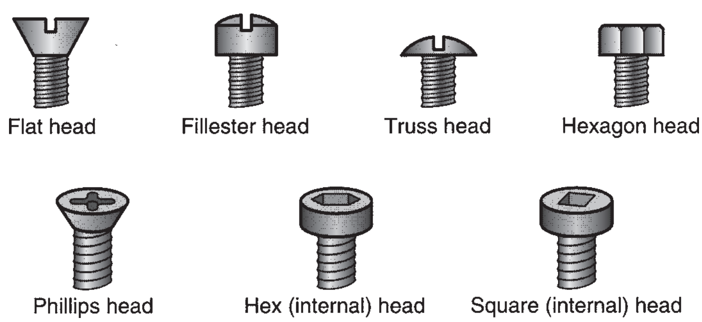

FIGURE 31.2 Various head styles available on screws and bolts. There are additional head styles not shown.

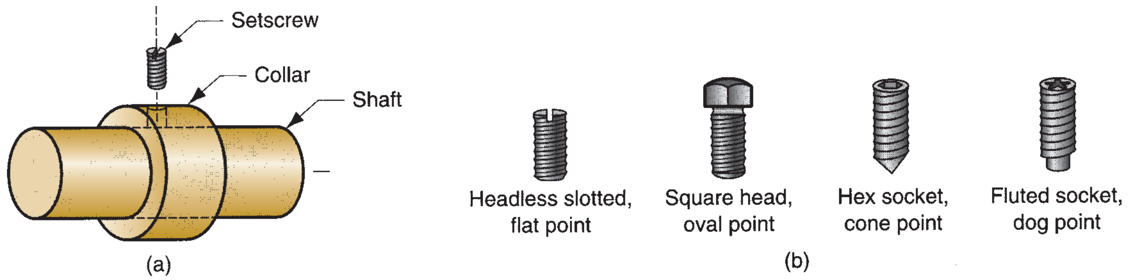

FIGURE 31.3 (a) Assembly of collar to shaft using a setscrew; (b) various setscrew geometries (head types and points).

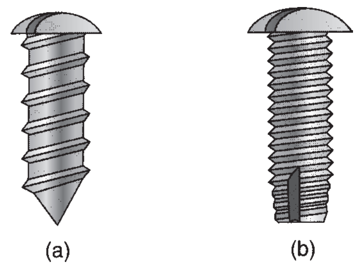

FIGURE 31.4 Self-tapping screws: (a) thread-forming and (b) thread-cutting.

assembly functions such as fastening collars, gears, and pulleys to shafts as shown in Figure 31.3(a). They come in various geometries, some of which are illustrated in Figure 31.3(b). A ***self-tapping screw*** (also called a ***tapping screw***) is designed to form or cut threads in a preexisting hole into which it is being turned. Figure 31.4 shows two of the typical thread geometries for self-tapping screws.

Most threaded fasteners are produced by cold forming (Section 18.2). Some are machined (Sections 21.2.2 and 21.7.1), but this is usually a more expensive thread-making process. A variety of materials are used to make threaded fasteners, steels being the most common because of their good strength and low cost. These include low and medium carbon as well as alloy steels. Fasteners made of steel are usually plated or coated for superficial resistance to corrosion. Nickel, chromium, zinc, black oxide, and similar coatings are used for this purpose. When corrosion or other factors deny the use of steel fasteners, other materials must be used, including stainless steels, aluminum alloys, nickel alloys, and plastics (however, plastics are suited to low stress applications only).

31.1.2 OTHER THREADED FASTENERS AND RELATED HARDWARE

Additional threaded fasteners and related hardware include studs, screw thread inserts, captive threaded fasteners, and washers. A ***stud*** (in the context of fasteners) is an externally threaded fastener, but without the usual head possessed by a bolt. Studs can be used to assemble two parts using two nuts as shown in Figure 31.5(a). They are available with threads on one end or both as in Figure 31.5(b) and (c).

Screw thread inserts are internally threaded plugs or wire coils made to be inserted into an unthreaded hole and to accept an externally threaded fastener. They are assembled into weaker materials (e.g., plastic, wood, and light-weight metals such as magnesium) to provide strong threads. There are many designs of screw thread inserts, one example of which is illustrated in Figure 31.6. Upon subsequent assembly

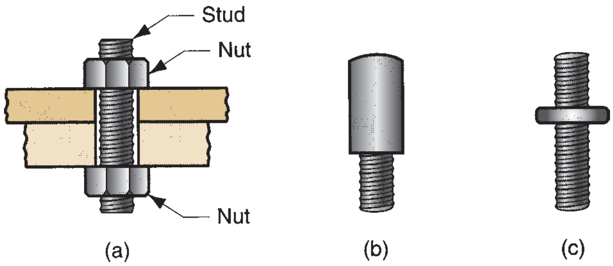

FIGURE 31.5 (a) Stud and nuts used for assembly. Other stud types: (b) threads on one end only, and (c) double-end stud.

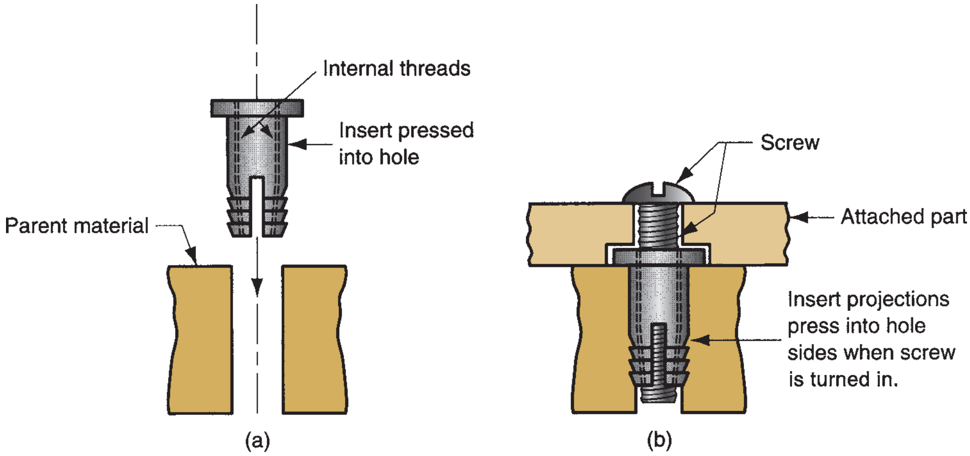

FIGURE 31.6 Screw thread inserts: (a) before insertion, and (b) after insertion into hole and screw is turned into the insert.

of the screw into the insert, the insert barrel expands into the sides of the hole, securing the assembly.

Captive threaded fasteners are threaded fasteners that have been permanently preassembled to one of the parts to be joined. Possible preassembly processes include welding, brazing, press fitting, or cold forming. Two types of captive threaded fasteners are illustrated in Figure 31.7.

A ***washer*** is a hardware component often used with threaded fasteners to ensure tightness of the mechanical joint; in its simplest form, it is a flat, thin ring of sheet metal. Washers serve various functions. They (1) distribute stresses that might otherwise be concentrated at the bolt or screw head and nut, (2) provide support for large clearance holes in the assembled parts, (3) increase spring tension, (4) protect part surfaces, (5) seal the joint, and (6) resist inadvertent unfastening [13]. Three washer types are illustrated in Figure 31.8.

31.1.3 STRESSES AND STRENGTHS IN BOLTED JOINTS

Typical stresses acting on a bolted or screwed joint include both tensile and shear, as depicted in Figure 31.9. Shown in the figure is a bolt-and-nut assembly. Once tightened, the bolt is loaded in tension, and the parts are loaded in compression. In addition, forces may be acting in opposite directions on the parts, which results in a shear stress on the bolt cross section. Finally, there are stresses applied on the threads throughout their engagement length with the nut in a direction parallel to

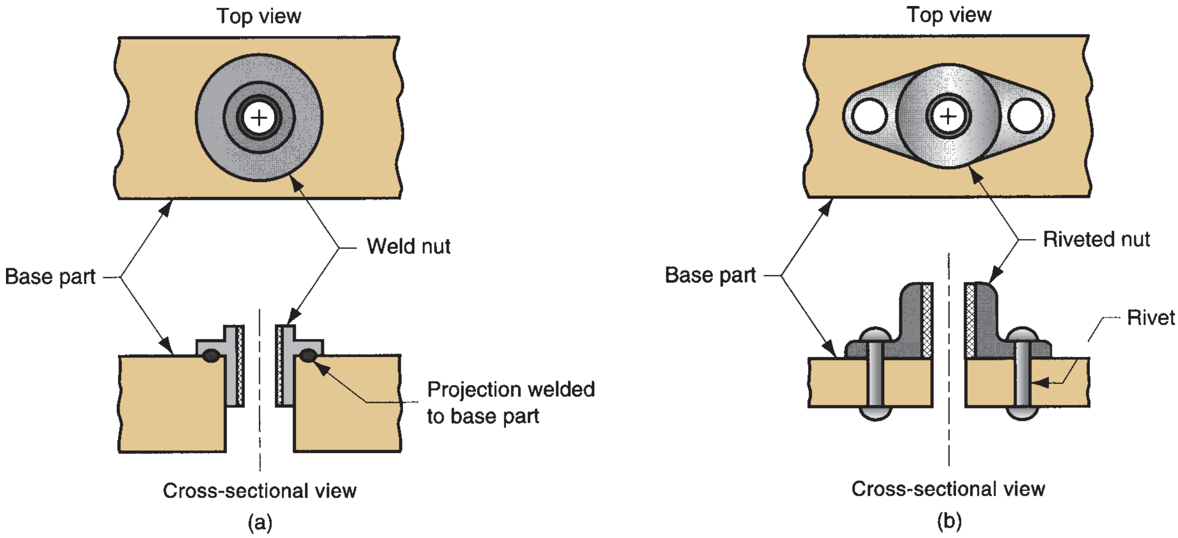

FIGURE 31.7 Captive threaded fasteners: (a) weld nut, and (b) riveted nut.

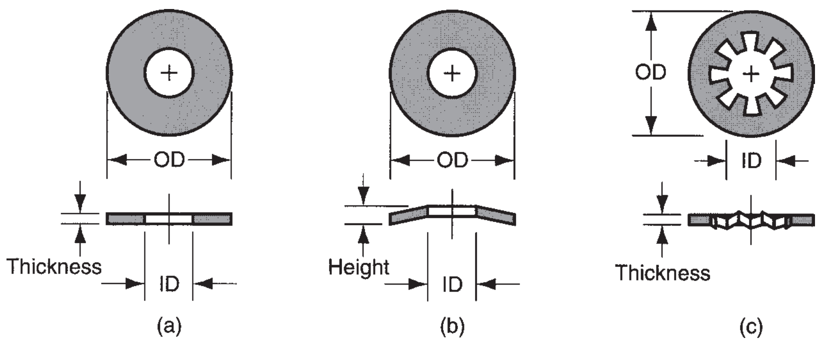

FIGURE 31.8 Types of washers: (a) plain (flat) washers; (b) spring washers, used to dampen vibration or compensate for wear; and (c) lockwasher designed to resist loosening of the bolt or screw.

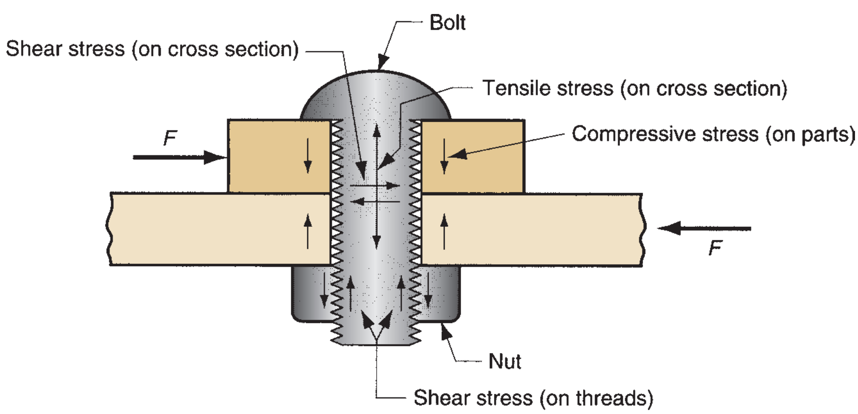

FIGURE 31.9 Typical stresses acting on a bolted joint.

TABLE • 31.2 Typical values of tensile and proof strengths for steel bolts and screws, diameters range from 6.4 mm (0.25 in) to 38 mm (1.50 in).

	Proof Stress		Tensile Stress	
Material	**MPa**	**lb/in²**	**MPa**	**lb/in²**
Low/medium C steel	228	33,000	414	60,000
Alloy steel	830	120,000	1030	150,000

Source: [13].

the axis of the bolt. These shear stresses can cause ***stripping*** of the threads, which can also occur on the internal threads of the nut.

The strength of a threaded fastener is generally specified by two measures: (1) tensile strength, which has the traditional definition (Section 3.1.1), and (2) proof strength. ***Proof strength*** is roughly equivalent to yield strength; specifically, it is the maximum tensile stress to which an externally threaded fastener can be subjected without permanent deformation. Typical values of tensile and proof strength for steel bolts are given in Table 31.2.

The problem that can arise during assembly is that the threaded fasteners are overtightened, causing stresses that exceed the strength of the fastener material. Assuming a bolt-and-nut assembly as shown in Figure 31.9, failure can occur in one of the following ways: (1) external threads (e.g., bolt or screw) can strip, (2) internal threads (e.g., nut) can strip, or (3) the bolt can break because of excessive tensile stresses on its cross-sectional area. Thread stripping, failures (1) and (2), is a shear failure and occurs when the length of engagement is too short (less than about 60% of the nominal bolt diameter). This can be avoided by providing adequate thread engagement in the fastener design. Tensile failure (3) is the most common problem. The bolt breaks at about 85% of its rated tensile strength because of combined tensile and torsion stresses during tightening [2].

The tensile stress to which a bolt is subjected can be calculated as the tensile load applied to the joint divided by the applicable area:

$$\sigma = \frac{F}{A_s} \tag{31.1}$$

where σ = stress, MPa (lb/in^2); F = load, N (lb); and A_s = tensile stress area, mm^2 (in^2). This stress is compared to the bolt strength values listed in Table 31.2. The tensile stress area for a threaded fastener is the cross-sectional area of the minor diameter. This area can be calculated directly from one of the following equations [2], depending on whether the bolt is metric standard or American standard. For the metric standard (ISO), the formula is

$$A_s = \frac{\pi}{4}(D - 0.9382p)^2 \tag{31.2}$$

where D = nominal size (basic major diameter) of the bolt or screw, mm; and p = thread pitch, mm. For the American standard (ANSI), the formula is

$$A_s = \frac{\pi}{4}\left(D - \frac{0.9743}{n}\right)^2 \tag{31.3}$$

where D = nominal size (basic major diameter) of the bolt or screw, in; and n = the number of threads per inch.

31.1.4 TOOLS AND METHODS FOR THREADED FASTENERS

The basic function of the tools and methods for assembling threaded fasteners is to provide relative rotation between the external and internal threads, and to apply sufficient torque to secure the assembly. Available tools range from simple hand-held screwdrivers or wrenches to powered tools with sophisticated electronic sensors to ensure proper tightening. It is important that the tool match the screw or bolt and/or the nut in style and size, since there are so many heads available. Hand tools are usually made with a single point or blade, but powered tools are generally designed to use interchangeable bits. The powered tools operate by pneumatic, hydraulic, or electric power.

Whether a threaded fastener serves its intended purpose depends to a large degree on the amount of torque applied to tighten it. Once the bolt or screw (or nut) has been rotated until it is seated against the part surface, additional tightening will increase the tension in the fastener (and simultaneously the compression in the parts being held together); and the tightening will be resisted by an increasing torque. Thus, there is a correlation between the torque required to tighten the fastener and the tensile stress experienced by it. To achieve the desired function in the assembled joint (e.g., to improve fatigue resistance) and to lock the threaded fasteners, the product designer will often specify the tension force that should be applied. This force is called the ***preload***. The following relationship can be used to determine the required torque to obtain a specified preload [13]:

$$T = C_t DF \tag{31.4}$$

where T = torque, N-mm (lb-in); C_t = the torque coefficient whose value typically ranges between 0.15 and 0.25, depending on the thread surface conditions; D = nominal bolt or screw diameter, mm (in); and F = specified preload tension force, N (lb).

Example 31.1 Threaded fasteners

A Metric 8 × 1.25 bolt must be tightened to a preload of 275 N. The torque coefficient = 0.22. Determine (a) the required torque that will achieve the specified preload, and (b) the stress on the bolt when preloaded.

Solution: Using Equation 31.4, the required torque $T = 0.22(8)(275) = 484$ N-mm = **0.484 N-m**

(b) The area of the minor diameter of the bolt is obtained using Equation 31.2:

$$A_s = \frac{\pi}{4}(D - 0.9382p)^2 = \frac{\pi}{4}(8 - 0.9382(1.25))^2 = 36.6 \text{ mm}^2$$

$$\sigma = F/A_s = \frac{275}{36.6} = 7.51 \text{ N/mm}^2 = \mathbf{7.51 \text{ MPa}}$$

Various methods are used to apply the required torque, including (1) operator feel—not very accurate, but adequate for most assemblies; (2) torque wrenches, which measure the torque as the fastener is being turned; (3) stall-motors, which are motorized wrenches designed to stall when the required torque is reached; and (4) torque-turn tightening, in which the fastener is initially tightened to a low torque level and then rotated a specified additional amount (e.g., a quarter turn).

31.2 Rivets and Eyelets

Rivets are widely used for achieving a permanent mechanically fastened joint. Riveting is a fastening method that offers high production rates, simplicity, dependability, and low cost. Despite these apparent advantages, its applications have declined in recent decades in favor of threaded fasteners, welding, and adhesive bonding. Riveting is one of the primary fastening processes in the aircraft and aerospace industries for joining skins to channels and other structural members.

A ***rivet*** is an unthreaded, headed pin used to join two (or more) parts by passing the pin through holes in the parts and then forming (upsetting) a second head in the pin on the opposite side. The deforming operation can be performed hot or cold (hot working or cold working), and by hammering or steady pressing. Once the rivet has been deformed, it cannot be removed except by breaking one of the heads. Rivets are specified by their length, diameter, head, and type. Rivet type refers to five basic geometries that affect how the rivet will be upset to form the second head. The five types are defined in Figure 31.10. In addition, there are special rivets for special applications.

Rivets are used primarily for lap joints. The clearance hole into which the rivet is inserted must be close to the diameter of the rivet. If the hole is too small, rivet insertion will be difficult, thus reducing production rate. If the hole is too large, the rivet will not fill the hole and may bend or compress during formation of the opposite head. Rivet design tables are available to specify the optimum hole sizes.

The tooling and methods used in riveting can be divided into the following categories: (1) impact, in which a pneumatic hammer delivers a succession of blows to upset the rivet; (2) steady compression, in which the riveting tool applies a continuous squeezing pressure to upset the rivet; and (3) a combination of impact and compression. Much of the equipment used in riveting is portable and manually operated. Automatic drilling-and-riveting machines are available for drilling the holes and then inserting and upsetting the rivets.

Eyelets are thin-walled, tubular fasteners with a flange on one end, usually made from sheet metal, as in Figure 31.11(a). They are used to produce a permanent lap joint between two (or more) flat parts. Eyelets are substituted for rivets in low-stress applications to save material, weight, and cost. During fastening, the eyelet is inserted through the part holes, and the straight end is formed over to secure the assembly.

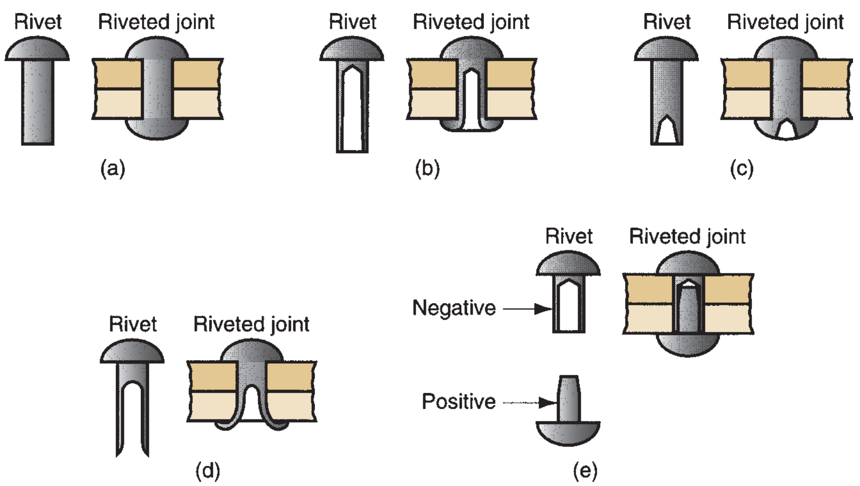

FIGURE 31.10 Five basic rivet types, also shown in assembled configuration: (a) solid, (b) tubular, (c) semitubular, (d) bifurcated, and (e) compression.

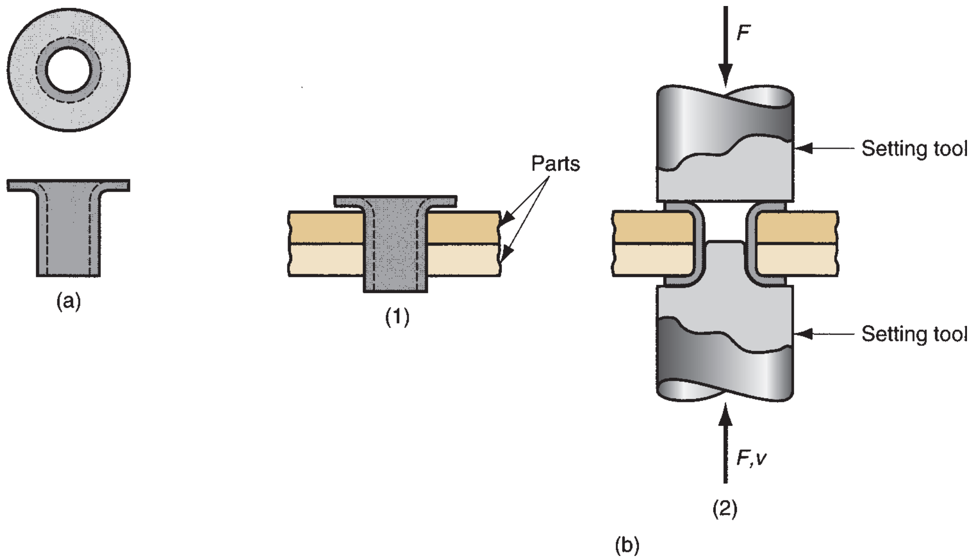

FIGURE 31.11
Fastening with an eyelet: (a) the eyelet, and (b) assembly sequence: (1) inserting the eyelet through the hole, and (2) setting operation.

The forming operation is called ***setting*** and is performed by opposing tools that hold the eyelet in position and curl the extended portion of its barrel. Figure 31.11(b) illustrates the sequence for a typical eyelet design. Applications of this fastening method include automotive subassemblies, electrical components, toys, and apparel.

31.3 Assembly Methods Based on Interference Fits

Several assembly methods are based on mechanical interference between the two mating parts being joined. This interference, which occurs either during assembly or after the parts are joined, holds the parts together. The methods include press fitting, shrink and expansion fits, snap fits, and retaining rings.

Press Fitting A press fit assembly is one in which the two components have an interference fit between them. The typical case is where a pin (e.g., a straight cylindrical pin) of a certain diameter is pressed into a hole of a slightly smaller diameter. Standard pin sizes are commercially available to accomplish a variety of functions, such as (1) locating and locking the components—used to augment threaded fasteners by holding two (or more) parts in fixed alignment with each other; (2) pivot points, to permit rotation of one component about the other; and (3) shear pins. Except for (3), the pins are normally hardened. Shear pins are made of softer metals so as to break under a sudden or severe shearing load to save the rest of the assembly. Other applications of press fitting include assembly of collars, gears, pulleys, and similar components onto shafts.

The pressures and stresses in an interference fit can be estimated using several applicable formulas. If the fit consists of a round solid pin or shaft inside a collar (or similar component), as depicted in Figure 31.12, and the components are made of the same material, the radial pressure between the pin and the collar can be determined by [13]:

$$P_f = \frac{Ei(D_c^2 - D_p^2)}{D_p D_c^2} \tag{31.5}$$

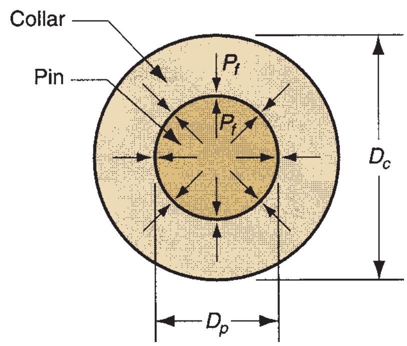

FIGURE 31.12 Cross section of a solid pin or shaft assembled to a collar by interference fit.

where p_f = radial or interference fit pressure, MPa (lb/in^2); E = modulus of elasticity for the material; i = interference between the pin (or shaft) and the collar; that is, the starting difference between the inside diameter of the collar hole and the outside diameter of the pin, mm (in); D_c = outside diameter of the collar, mm (in); and D_p = pin or shaft diameter, mm (in).

The maximum effective stress occurs in the collar at its inside diameter and can be calculated as

$$\text{Max } \sigma_e = \frac{2p_f D_c^2}{D_c^2 - D_p^2} \tag{31.6}$$

where Max σ_e = the maximum effective stress, MPa (lb/in^2), and p_f is the interference fit pressure computed from Equation (31.5).

In situations in which a straight pin or shaft is pressed into the hole of a large part with geometry other than that of a collar, the previous equations can be simplified by taking the outside diameter D_c to be infinite, thus reducing the equation for interference pressure to

$$p_f = \frac{Ei}{D_p} \tag{31.7}$$

and the corresponding maximum effective stress becomes

$$\text{Max } \sigma_e = 2p_f \tag{31.8}$$

In most cases, particularly for ductile metals, the maximum effective stress should be compared with the yield strength of the material, applying an appropriate safety factor, as in the following:

$$\text{Max } \sigma_e \leq \frac{Y}{SF} \tag{31.9}$$

where Y = yield strength of the material, and SF is the applicable safety factor.

Various pin geometries are available for interference fits. The basic type is a ***straight pin***, usually made from cold-drawn carbon steel wire or bar stock, ranging in diameter from 1.6 to 25 mm (1/16–1.0 in). They are unground, with chamfered or square ends (chamfered ends facilitate press fitting). ***Dowel pins*** are manufactured to more precise specifications than straight pins, and can be ground and hardened. They are used to fix the alignment of assembled components in dies, fixtures, and machinery. ***Taper pins*** possess a taper of 6.4 mm (0.25 in) per foot and are driven into the hole to establish a fixed relative position between the parts. Their advantage is that they can be driven readily back out of the hole.

Additional pin geometries are commercially available, including ***grooved pins***—solid straight pins with three longitudinal grooves in which the metal is raised on either side of each groove to cause interference when the pin is pressed into a hole; ***knurled pins***, pins with a knurled pattern that causes interference in the mating hole; and ***coiled pins***, also called ***spiral pins***, which are made by rolling strip stock into a coiled spring.

Shrink and Expansion Fits These terms refer to the assembly of two parts that have an interference fit at room temperature. The typical case is a cylindrical pin or shaft assembled into a collar. To assemble by ***shrink fitting***, the external part is heated to enlarge it by thermal expansion, and the internal part either remains at room temperature or is cooled to contract its size. The parts are then assembled and brought back to room temperature, so that the external part shrinks, and if previously cooled the internal part expands, to form a strong interference fit. An ***expansion fit*** is when only the internal part is cooled to contract it for assembly; once inserted into the mating component, it warms to room temperature, expanding to create the interference assembly. These assembly methods are used to fit gears, pulleys, sleeves, and other components onto solid and hollow shafts.

Various methods are used to heat and/or cool the work parts. Heating equipment includes torches, furnaces, electric resistance heaters, and electric induction heaters. Cooling methods include conventional refrigeration, packing in dry ice, and immersion in cold liquids, including liquid nitrogen. The resulting change in diameter depends on the coefficient of thermal expansion and the temperature difference that is applied to the part. If it is assumed that the heating or cooling has produced a uniform temperature throughout the work, then the change in diameter is given by

$$D_2 - D_1 = \alpha\, D_1(T_2 - T_1) \tag{31.10}$$

where α = the coefficient of linear thermal expansion, mm/mm-°C (in/in-°F) for the material (see Table 4.1); T_2 = the temperature to which the parts have been heated or cooled, °C (°F); T_1 = starting ambient temperature; D_2 = diameter of the part at T_2, mm (in); and D_1 = diameter of the part at T_1.

Equations (31.5) through (31.9) for computing interference pressures and effective stresses can be used to determine the corresponding values for shrink and expansion fits.

Example 31.2
Expansion fit

An alloy steel shaft is to be inserted into a collar of the same metal using an expansion fit. At room temperature (20°C), the outer and inner diameters of the collar = 50.00 mm and 30.00 mm, respectively, and the shaft has a diameter = 30.015 mm. The shaft must be reduced in size for assembly into the collar by cooling to a sufficiently low temperature that there is a clearance of 0.03 mm. Determine (a) the temperature to which the shaft must be cooled for assembly, (b) the radial pressure at room temperature after assembly, and (c) the maximum effective stress on the collar.

Solution: (a) From Table 4.1, the coefficient of thermal expansion for steel $\alpha = 12(10^{-6})\ °C^{-1}$. Rearranging Equation 31.10 to solve for the cooling temperature,

$$T_2 = \frac{(D2 - D_1)}{\alpha\, D_1} + T_1 = \frac{(30.00 - 0.03) - 30.015}{12(10^{-6})(30.015)} + 20 = -124.9 + 20 = \mathbf{-104.9°C}$$

(b) From Table 3.1, the modulus of elasticity for steel $E = 209(10^3)$ MPa. Using Equation (31.5) to solve for radial pressure, where $D_c = 50.00$ mm and $D_p = 30.025$ mm

$$P_f = \frac{Ei(D_c^2 - D_p^2)}{D_p D_c^2} = \frac{209(10^3)(0.015)(50^2 - 30.015^2)}{30.015(50^2)} = \mathbf{66.8\ MPa}$$

(c) The maximum effective stress is given by Equation 31.8:

$$\text{Max}\ \sigma_e = \frac{2p_f D_c^2}{D_c^2 - D_p^2} = \frac{2(66.8)(50^2)}{(50^2 - 30.015^2)} = \mathbf{209\ MPa}$$

Snap Fits and Retaining Rings Snap fits are a variation of interference fits. A ***snap fit*** involves joining two parts in which the mating elements possess a temporary interference while being pressed together, but once assembled they interlock to maintain the assembly. A typical example is shown in Figure 31.13: as the parts are pressed together, the mating elements elastically deform to accommodate the interference, subsequently allowing the parts to snap together; once in position, the elements become connected mechanically so that they cannot easily be disassembled. The parts are usually designed so that a slight interference exists after assembly.

Advantages of snap fit assembly include (1) the parts can be designed with self-aligning features, (2) no special tooling is required, and (3) assembly can be accomplished very quickly. Snap fitting was originally conceived as a method that would be ideally suited to industrial robotics applications; however, it is no surprise that assembly techniques that are easier for robots are also easier for human assembly workers.

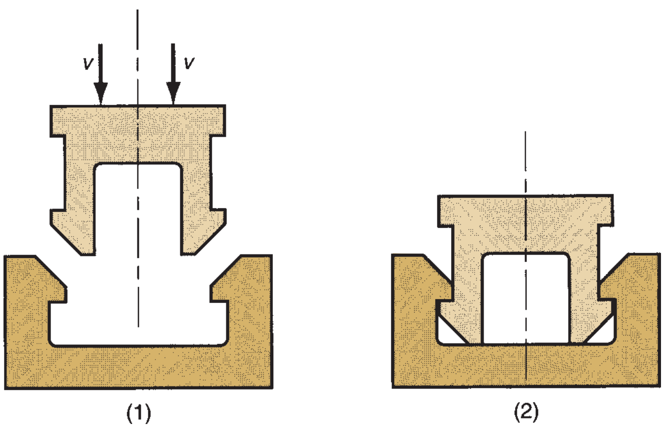

FIGURE 31.13 Snap fit assembly, showing cross sections of two mating parts: (1) before assembly, and (2) parts snapped together.

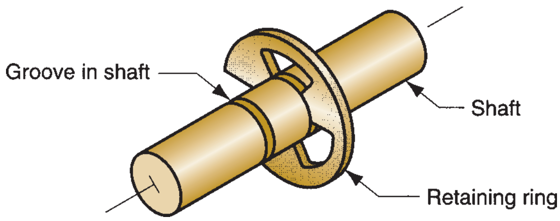

FIGURE 31.14 Retaining ring assembled into a groove on a shaft.

A ***retaining ring***, also known as a ***snap ring***, is a fastener that snaps into a circumferential groove on a shaft or tube to form a shoulder, as in Figure 31.14. The assembly can be used to locate or restrict the movement of parts mounted on the shaft. Retaining rings are available for both external (shaft) and internal (bore) applications. They are made from either sheet metal or wire stock, heat treated for hardness and stiffness. To assemble a retaining ring, a special pliers tool is used to elastically deform the ring so that it fits over the shaft (or into the bore) and then is released into the groove.

31.4 Other Mechanical Fastening Methods

In addition to the mechanical assembly techniques discussed above, there are several additional methods that involve the use of fasteners. These include stitching, stapling, sewing, and cotter pins.

Stitching, Stapling, and Sewing Industrial stitching and stapling are similar operations involving the use of U-shaped metal fasteners. ***Stitching*** is a fastening operation in which a stitching machine is used to form the U-shaped stitches one at a time from steel wire and immediately drive them through the two parts to be joined. Figure 31.15 illustrates several types of wire stitches. The parts to be joined must be relatively thin, consistent with the stitch size, and the assembly can involve various combinations of metal and nonmetal materials. Applications of industrial stitching include light sheet-metal assembly, metal hinges, electrical connections, magazine binding, corrugated boxes, and final product packaging. Conditions that favor stitching in these applications are (1) high-speed operation, (2) elimination of the need for prefabricated holes in the parts, and (3) desirability of using fasteners that encircle the parts.

In ***stapling***, preformed U-shaped staples are punched through the two parts to be attached. The staples are supplied in convenient strips. The individual staples are lightly stuck together to form the strip, but they can be separated by the stapling tool for driving. The staples come with various point styles to facilitate their entry into the work. Staples are usually applied by means of portable pneumatic guns, into which strips containing several hundred staples can be loaded. Applications of industrial stapling include: furniture and upholstery, assembly of car seats, and various light-gage sheet-metal and plastic assembly jobs.

Sewing is a common joining method for soft, flexible parts such as cloth and leather. The method involves the use of a long thread or cord interwoven with the

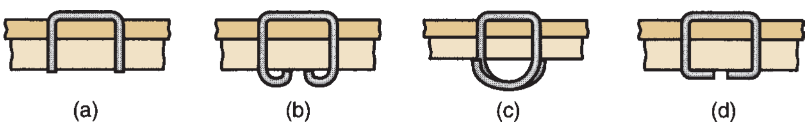

FIGURE 31.15 Common types of wire stitches: (a) unclinched, (b) standard loop, (c) bypass loop, and (d) flat clinch.

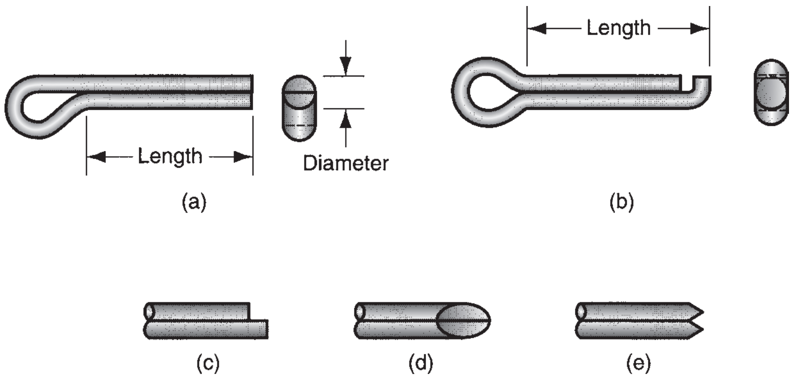

FIGURE 31.16
Cotter pins: (a) offset head, standard point; (b) symmetric head, hammerlock point; (c) square point; (d) mitered point; and (e) chisel point.

parts so as to produce a continuous seam between them. The process is widely used in the needle trades industry for assembling garments.

Cotter Pins Cotter pins are fasteners formed from half-round wire into a single two-stem pin, as in Figure 31.16. They vary in diameter, ranging between 0.8 mm (0.031 in) and 19 mm (3/4 in), and in point style, several of which are shown in the figure. Cotter pins are inserted into holes in the mating parts and their legs are split to lock the assembly. They are used to secure parts onto shafts and similar applications.

31.5 Molding Inserts and Integral Fasteners

These assembly methods form a permanent joint between parts by shaping or reshaping one of the components through a manufacturing process such as casting, molding, or sheet-metal forming.

Inserts in Moldings and Castings This method involves the placement of a component into a mold prior to plastic molding or metal casting, so that it becomes a permanent and integral part of the molding or casting. Inserting a separate component is preferable to molding or casting its shape if the superior properties (e.g., strength) of the insert material are required, or the geometry achieved through the use of the insert is too complex or intricate to incorporate into the mold. Examples of inserts in molded or cast parts include internally threaded bushings and nuts, externally threaded studs, bearings, and electrical contacts. Some of these are illustrated in Figure 31.17. Internally threaded inserts must be placed into the mold with threaded pins to prevent the molding material from flowing into the threaded hole.

Placing inserts into a mold has certain disadvantages in production: (1) design of the mold becomes more complicated; (2) handling and placing the insert into the cavity takes time that reduces production rate; and (3) inserts introduce a foreign material into the casting or molding, and in the event of a defect, the cast metal or plastic cannot be easily reclaimed and recycled. Despite these disadvantages, use of inserts is often the most functional design and least-cost production method.

Integral Fasteners Integral fasteners involve deformation of component parts so they interlock and create a mechanically fastened joint. This assembly method is most common for sheet-metal parts. The possibilities, Figure 31.18, include (a) ***lanced tabs*** to attach wires or shafts to sheet-metal parts; (b) ***embossed protrusions***,

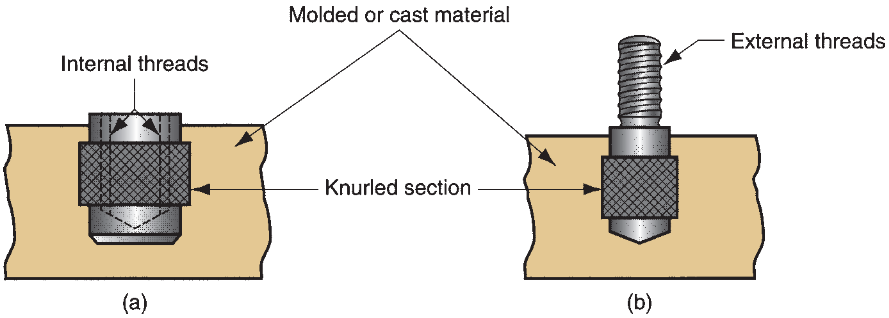

FIGURE 31.17 Examples of molded-in inserts: (a) threaded bushing, and (b) threaded stud.

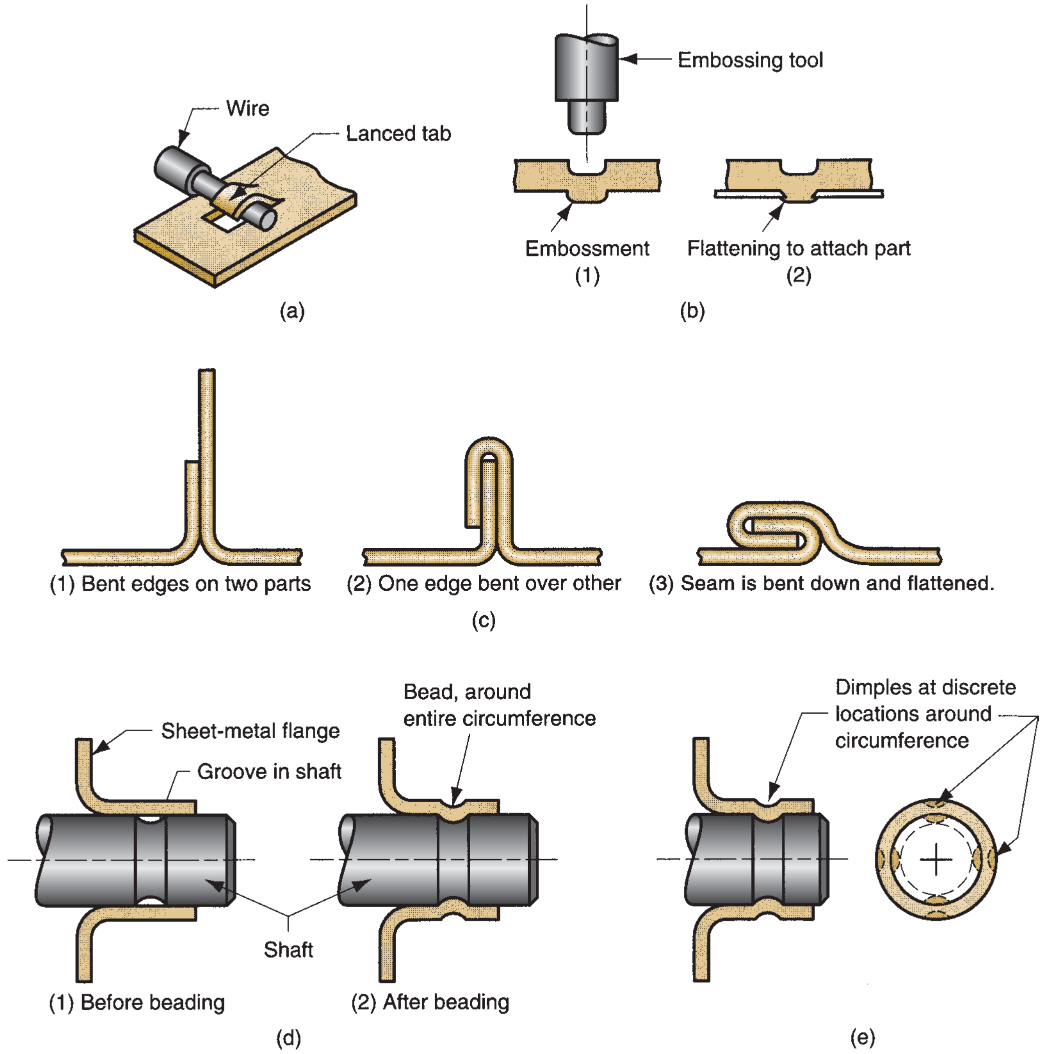

FIGURE 31.18 Integral fasteners: (a) lanced tabs to attach wires or shafts to sheetmetal, (b) embossed protrusions, similar to riveting, (c) single-lock seaming, (d) beading, and (e) dimpling. Numbers in parentheses indicate sequence in (b), (c), and (d).

in which bosses are formed in one part and flattened over the mating assembled part; (c) ***seaming***, where the edges of two separate sheet-metal parts or the opposite edges of the same part are bent over to form the fastening seam—the metal must be ductile in order for the bending to be feasible; (d) ***beading***, in which a tube-shaped part is attached to a smaller shaft (or other round part) by deforming the outer diameter inward to cause an interference around the entire circumference; and (e) ***dimpling***—forming of simple round indentations in an outer part to retain an inner part.

Crimping, in which the edges of one part are deformed over a mating component, is another example of integral assembly. A common example involves squeezing the barrel of an electrical terminal onto a wire (Section 34.5.1).

31.6 Design for Assembly

Design for assembly (DFA) has received much attention in recent years because assembly operations constitute a high labor cost for many manufacturing companies. The key to successful design for assembly can be simply stated [3]: (1) design the product with as few parts as possible, and (2) design the remaining parts so they are easy to assemble. The cost of assembly is determined largely during product design, because that is when the number of separate components in the product is determined, and decisions are made about how these components will be assembled. Once these decisions have been made, there is little that can be done in manufacturing to influence assembly costs (except, of course, to manage the operations well).

This section considers some of the principles that can be applied during product design to facilitate assembly. Most of the principles have been developed in the context of mechanical assembly, although some of them apply to the other assembly and joining processes. Much of the research in design for assembly has been motivated by the increasing use of automated assembly systems in industry. Accordingly, our discussion is divided into two sections, the first dealing with general principles of DFA, and the second concerned specifically with design for automated assembly.

31.6.1 GENERAL PRINCIPLES OF DFA

Most of the general principles apply to both manual and automated assembly. Their goal is to achieve the required design function by the simplest and lowest cost means. The following recommendations have been compiled from [1], [3], [4], and [6]:

- ***Use the fewest number of parts possible to reduce the amount of assembly required***. This principle is implemented by combining functions within the same part that might otherwise be accomplished by separate components (e.g., using a plastic molded part instead of an assembly of sheet metal parts).
- ***Reduce the number of threaded fasteners required***. Instead of using separate threaded fasteners, design the component to utilize snap fits, retaining rings, integral fasteners, and similar fastening mechanisms that can be accomplished more rapidly. Use threaded fasteners only where justified (e.g., where disassembly or adjustment is required).
- ***Standardize fasteners***. This is intended to reduce the number of sizes and styles of fasteners required in the product. Ordering and inventory problems are reduced, the assembly worker does not have to distinguish between so many

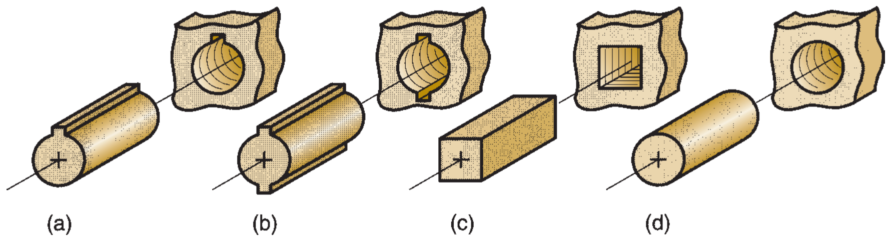

FIGURE 31.19 Symmetrical parts are generally easier to insert and assemble: (a) only one rotational orientation possible for insertion; (b) two possible orientations; (c) four possible orientations; and (d) infinite rotational orientations.

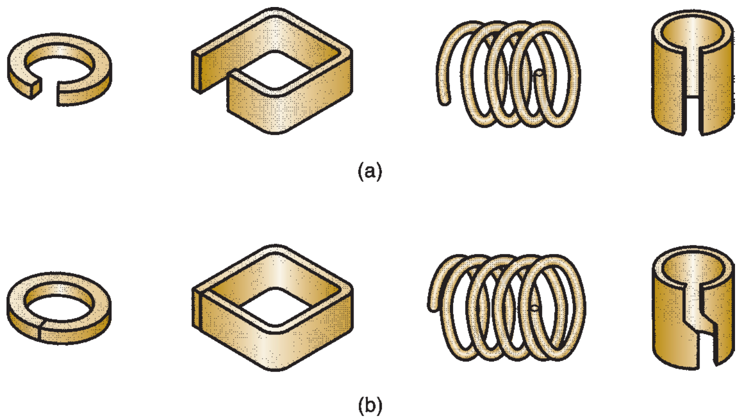

FIGURE 31.20 (a) Parts that tend to tangle, and (b) parts designed to avoid tangling.

separate fasteners, the workstation is simplified, and the variety of separate fastening tools is reduced.

- ➢ ***Reduce parts orientation difficulties***. Orientation problems are generally reduced by designing a part to be symmetrical and minimizing the number of asymmetric features. This allows easier handling and insertion during assembly. This principle is illustrated in Figure 31.19.
- ➢ ***Avoid parts that tangle***. Certain part configurations are more likely to become entangled in parts bins, frustrating assembly workers or jamming automatic feeders. Parts with hooks, holes, slots, and curls exhibit more of this tendency than parts without these features (see Figure 31.20).

31.6.2 DESIGN FOR AUTOMATED ASSEMBLY

Methods suitable for manual assembly are not necessarily the best methods for automated assembly. Some assembly operations readily performed by a human worker are quite difficult to automate (e.g., assembly using bolts and nuts). To automate the assembly process, parts fastening methods must be specified during product design that lend themselves to machine insertion and joining techniques and do not require the senses, dexterity, and intelligence of human assembly workers. Following are some recommendations and principles that can be applied in product design to facilitate automated assembly [6], [10]:

- ➢ ***Use modularity in product design***. Increasing the number of separate tasks that are accomplished by an automated assembly system will reduce the reliability of the system. To alleviate the reliability problem, Riley [10] suggests that the design of the product be modular in which each module or subassembly has a maximum of 12 or 13 parts to be produced on a single assembly system. Also, the subassembly should be designed around a base part to which other components are added.
- ➢ ***Reduce the need for multiple components to be handled at once***. The preferred practice for automated assembly is to separate the operations at different stations rather than to simultaneously handle and fasten multiple components at the same workstation.
- ➢ ***Limit the required directions of access***. This means that the number of directions in which new components are added to the existing subassembly should be minimized. Ideally, all components should be added vertically from above, if possible.
- ➢ ***High-quality components***. High performance of an automated assembly system requires that consistently good-quality components are added at each workstation. Poor quality components cause jams in feeding and assembly mechanisms that result in downtime.
- ➢ ***Use of snap fit assembly***. This eliminates the need for threaded fasteners; assembly is by simple insertion, usually from above. It requires that the parts be designed with special positive and negative features to facilitate insertion and fastening.

References

[1] Andreasen, M., Kahler, S., and Lund, T. ***Design for Assembly***. Springer-Verlag, New York, 1988.

[2] Blake, A. ***What Every Engineer Should Know About Threaded Fasteners***. Marcel Dekker, New York, 1986.

[3] Boothroyd, G., Dewhurst, P., and Knight, W. ***Product Design for Manufacture and Assembly***. 3rd ed. CRC Taylor & Francis, Boca Raton, Florida, 2010.

[4] Bralla, J. G. (ed.). ***Design for Manufacturability Handbook***, 2nd ed. McGraw-Hill, New York, 1998.

[5] Dewhurst, P., and Boothroyd, G. “Design for Assembly in Action,” ***Assembly Engineering***, January 1987, pp. 64–68.

[6] Groover, M. P. ***Automation, Production Systems, and Computer Integrated Manufacturing***, 3rd ed. Pearson Prentice-Hall, Upper Saddle River, New Jersey, 2008.

[7] Groover, M. P., Weiss, M., Nagel, R. N., and Odrey, N. G. ***Industrial Robotics: Technology, Programming, and Applications***. McGraw-Hill, New York, 1986.

[8] Nof, S. Y., Wilhelm, W. E., and Warnecke, H-J. ***Industrial Assembly***. Chapman & Hall, New York, 1997.

[9] Parmley, R. O. (ed.). ***Standard Handbook of Fastening and Joining***, 3rd ed. McGraw-Hill, New York, 1997.

[10] Riley, F. J. ***Assembly Automation, A Management Handbook***, 2nd ed. Industrial Press, New York, 1999.

[11] Speck, J. A. ***Mechanical Fastening, Joining, and Assembly***. Marcel Dekker, New York, 1997.

[12] Whitney, D. E. ***Mechanical Assemblies***. Oxford University Press, New York, 2004.

[13] Wick, C., and Veilleux, R. F. (eds.). ***Tool and Manufacturing Engineers Handbook***, 4th ed., Vol. IV, ***Quality Control and Assembly***. Society of Manufacturing Engineers, Dearborn, Michigan, 1987.