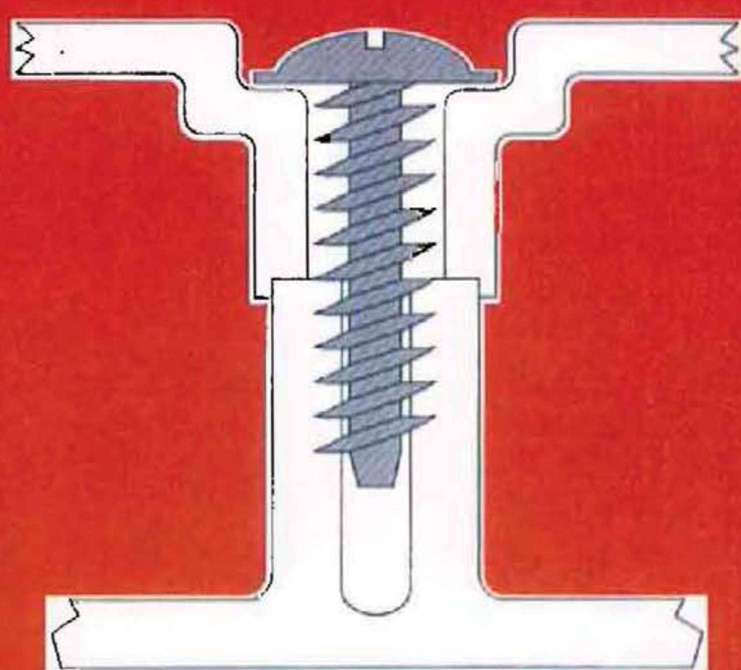
Robert A. Malloy

Plastic Part Design for Injection Molding



An Introduction

Ph.D. Polymer Chemistry/Plastics Engineering) is an sor in the Department of Plastics Engineering at the assachusetts Lowell. He teaches courses in areas of and Mold Design. His research interests include applied y, plastics recycling, and polymer process instrumenta- ing the University of Massachusetts Lowell, the author structor at the Algerian Petroleum Institute in Annaba, Past Chairman of the Society of Plastics Engineers g Division.

des an overview of the design process for injection arts. It describes an integrated approach to plastic part ic material selection and will assist the designer in the a plastic part design that is functional, manufacturable, pleasing. The book goes into great detail on subjects rent Engineering and Design for Manufacturability, he various phases of the injection molding process can esign. The book should serve as a reference and intro- uide for the plastic part design engineer, or as the text e or one semester course on the subject of plastic part

Consideration for Injection Molded Parts
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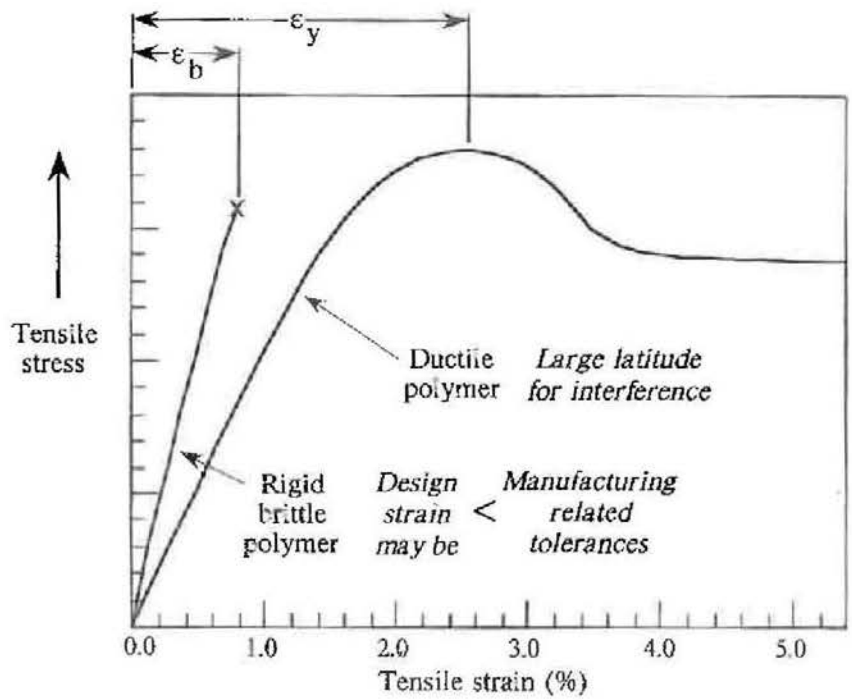
This book is dedicated to the memory of
S J, Eileen, and Ahn-Ahn Chen

ally for longer term applications where stress relaxation can occur, cannot exceed ps 10 - 40% of this breaking strain value. Permissible interference values are ly related to design strain values, and as a result, the difference in the inside ter of the hub and the outside diameter of the shaft may be less than the combined facturing tolerances of the hub and shaft. When more ductile polymers such as forced an polyethylene, nylon, or acetal are used, the design strain values are large h (relative to the combined manufacturing tolerances) that press fitting becomes a cal assembly option as shown in Figure 6.4.



.4. Typical stress-strain behavior for rigid and ductile polymers. Many rigid polymers are too r press fit applications. Tough, ductile polymers are more forgiving when considering the effects facturing tolerances.

b in a press fit assembly is subject to tensile loading, while the shaft is subject to ssive loading. The properties of both the hub and shaft materials must be taken count when designing press fit assemblies, especially when the shaft is produced plastic material. Tensile stresses in the hub are the primary concern in most Tensile stress levels for the hub should be kept as low as possible in order to a more reliable long term assembly. If the design stress values are low, the n torsional (or axial) strength due to stress relaxation will be reduced. In addition, ential for failure due to crazing, cracking, weld lines or chemicals is reduced at tress levels. Unfortunately, the axial and torsional strength of a press fit assembly directly related to contact pressure and therefore the tensile stress level. The , required to cause relative movement between the hub and the shaft can be given

$$= \mu \cdot P \cdot A \tag{6.1}$$

where μ is the coefficient of friction between the shaft and hub, P is the contact pressure (directly related to design stress), and A is the surface area of contact. The coefficient of friction for a given hub / shaft material pair is essentially a constant (it does in fact vary with stress level and surface quality). The optimum press fit design is achieved when the designer uses an interference value that results in low stress levels, but maximizes the surface area of contact as shown in Figure 6.2. Larger diameter shafts can also be used when possible. Additives that reduce the coefficient of friction of the polymer will reduce the torsional strength of the assembly. Additives such as mold releases or lubricants, should be avoided in press fit applications. The mating surfaces should be very clean and free from any type of chemical contaminate that could result in delayed failure [16]. Press fit assemblies subject to temperature variations should be produced with materials that have similar coefficients of thermal expansion whenever possible, to avoid changes in the effective interference due to the coefficient of thermal expansion mismatch.

6.2.3 Design of Press Fit Assemblies

The designer's objective in most press fit applications is to produce an assembly that has adequate resistance to both torsional and axial movement, while keeping component stress levels (especially tensile stress) within acceptable limits. The tensile stress level for a given shaft / hub material combination is determined by the overall geometry of the components and the interference value used. In most cases, the outer hub dimensions and often the shaft diameter are dictated by the end-use application. The designer must determine an acceptable interference value for each application. One way to determine the acceptable interference is to consult material manufacturers, who can provide graphical representations of interference limits (maximum recommended interference value), such as that shown in Figure 6.5.

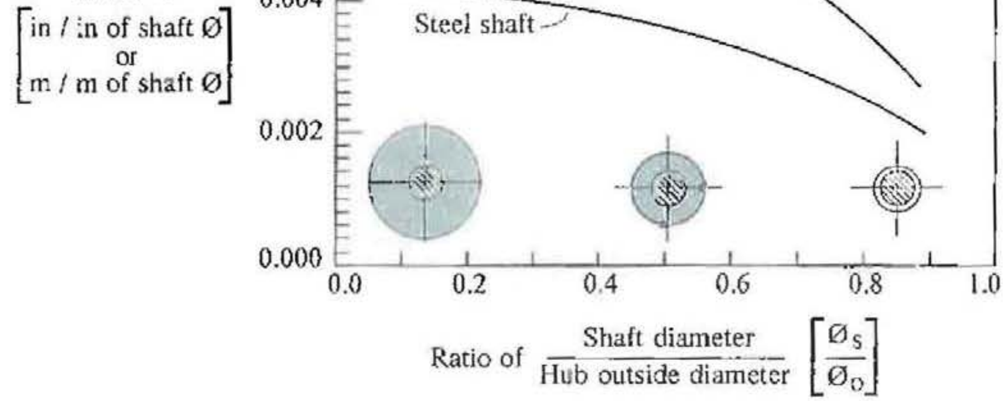


Figure 6.5. The amount of interference for a press fit can be determined using design equations, or graphs similar to that shown, indicating the maximum recommended interference for a particular material.

The information presented in figures of this type is material specific and is applicable only within a limited range of environmental conditions. Using the information given in Figure 6.5, the interference limit, I, for a nylon hub and steel shaft can be determined. Consider the following example.

Steel shaft diameter, $Ø_S$ = 0.400 inch
Nylon hub outside diameter, $Ø_O$ = 1.000 inch
then $Ø_S / Ø_O$ = 0.400 inch / 1.000 inch = 0.40

From the graph, the I limit value is 0.0036 inch / inch shaft diameter. The interference value for the 0.400 inch diameter shaft would be:

I = 0.400 inch x 0.0036 inch / inch shaft diameter = 0.0014 inch

The hub inside diameter, $Ø_i$, is then:

$$Ø_i = Ø_S - I = 0.400 \text{ inch} - 0.0015 \text{ inch} = 0.3985 \text{ inch}$$

The molded gear, after mold shrinkage; should have an inside diameter no less than 0.3985 inch to stay within the material manufacturers recommended strain limit.

Figures 6.5 shows that the permissible interference increases as the ratio of the radial wall thickness to shaft diameter increases. The figure also shows that the recommended interference increases when nylon, rather than steel, is used for the shaft material. This is due to the fact that the softer nylon shaft (relative to steel) actually deforms after the pressing operation. Even though the interference values are greater for the nylon hub / shaft combination compared to the nylon / steel combination, the effective (net) interference and tensile stress level in the plastic hub after pressing is approximately the same in both cases.

While the information presented in Figure 6.5 is extremely useful, it is limited, since it is both material and environment specific. As a more general approach, Equation 6.2 can be used to determine the interference value for any hub / shaft material combination:

$$I = [(\sigma_D \cdot Ø_S) / W] \cdot [\,[(W + \nu_h) / E_h] + [(1 - \nu_S) / E_S]\,] \qquad (6.2)$$

where: I = diametral interference (inch)
σ_D = design stress level (lbs / inch^2)
$Ø_O$ = hub outside diameter (inch)
$Ø_S$ = shaft diameter (inch)
E_h = hub modulus (lbs / inch^2)
E_S = shaft modulus (lbs / inch^2)
ν_h = Poisson's ratio for hub material
ν_S = Poisson's ratio for shaft material
$W = [1 + (Ø_S / Ø_O)^2] / [1 - (Ø_S / Ø_O)^2]$

The interference, or difference in the shaft diameter and the hub inside diameter, can be determined if the shaft and hub outside diameters, and material properties are known. When the shaft and hub are produced in the same polymer, the hub and shaft properties are equivalent, and the expression reduces to:

$$I = [(\sigma_D \cdot \varnothing_S)/W] \cdot [(W+1)/E] \tag{6.3}$$

where E is the modulus of the polymer (assumes tensile modulus is equal to compressive modulus). Equation 6.2 can also be reduced to a simpler form when a polymer hub is used with a metal shaft. For the metal shaft, $E_S \gg E_P$, and the I can be found using:

$$I = [(\sigma_D \cdot \varnothing_S)/W] \cdot [(W + \nu_h)/E_h] \tag{6.4}$$

This form of the equation is most commonly used since plastic hubs are typically used with steel shafts. The material property values used in the expression should be those that correlate with the end-use conditions.

The designer must determine what level of tensile stress (or strain) that is acceptable for the hub material. The design stress value is not a constant value but changes with variables such as temperature, relative humidity, chemical environment and time. As the service life of a plastic product increases, the recommended design stress levels decrease. Consider the press fit hub / shaft assembly in Figure 6.6. Press fit hubs are sometimes pushed over shafts that have been turned down to provide positive axial location.

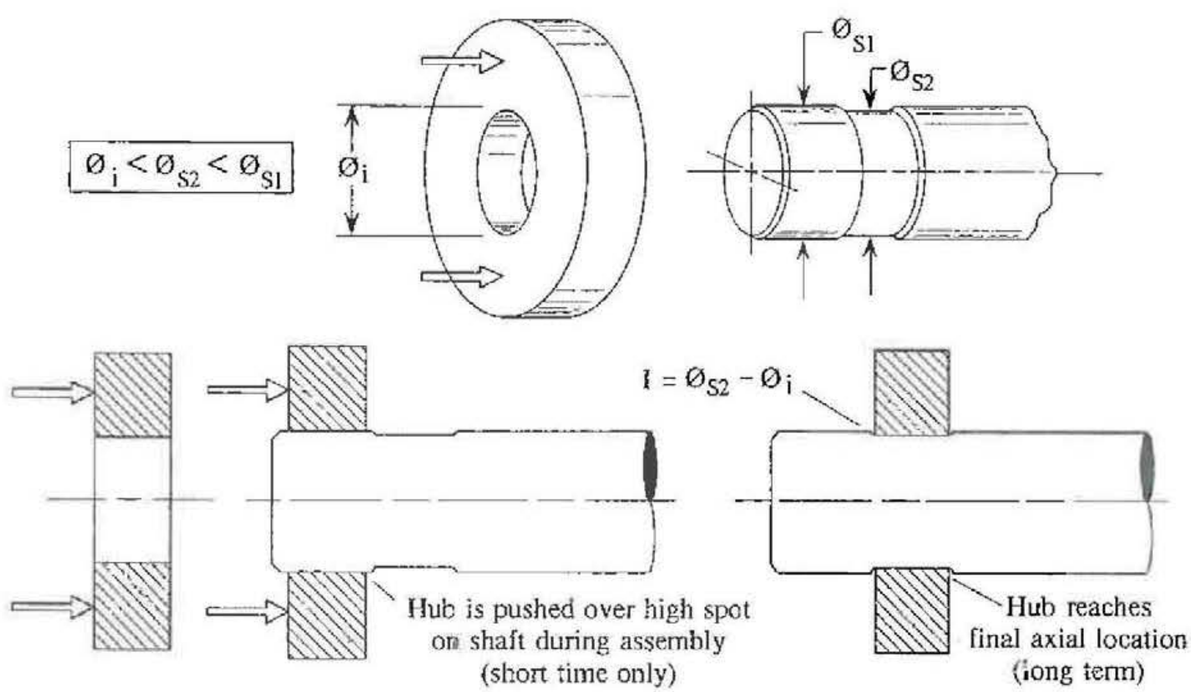


Figure 6.6. Use of an undercut shaft for press fit hub / shaft assemblies will locate the hub axially. The stresses associated with both the long term and short term interference (as the hub is pushed onto the shaft) must be considered by the designer.

The hub inside diameter and shaft diameter at the point where the shaft is undercut, should be designed using an interference value that is suitable for a long term application. The interference value (and strain level) are significantly higher when the hub is pushed over the shoulder onto the shaft during assembly, though for only a very short period of time. Acceptable short term stress (or strain) values could be used for this very short term assembly interference calculation. The tensile stress values that are suitable for long term press fit applications are determined by the stress relaxation characteristics of the polymer, and are often less than 20-25% of the short tensile yield stress for the material. In general, the design stress values for long term applications should be kept as low as possible, in order to minimize stress relaxation and the potential for premature failure due to slippage or cracking. However, when the hub is pushed over the shoulder on the shaft (the portion of the shaft that has not been turned down) during installation, the stress levels reach a higher value, perhaps 40 - 60% of the tensile yield stress, for a very short period of time.

The force required to push a press fit (undersized) hub over a shaft is important for a number of reasons. The push-on force, F, determines the assembly characteristics, torsional strength, and axial strength of the assembly. The push-on force can be determined using:

$$F = \mu \cdot P \cdot A = \mu \cdot P \cdot \pi \cdot \text{Ø}_S \cdot L \tag{6.5}$$

Where L is the axial length or width of the hub. The contact pressure, P, can be determined using:

$$P = \sigma_D / W \tag{6.6}$$

The push-on force value, P, is the force required for assembly, and is related to the torsional strength (slippage torque), T, of the assembly.

$$T = F \cdot \text{Ø}_S / 2 \tag{6.7}$$

Push-on or assembly force values for the hub / shaft assembly can be reduced using lubricants that reduce the coefficient of friction between the polymer hub and metal shaft, however, lubricants will also reduce the torsional strength (i.e. slippage torque) of the press fit assembly. The combined effect of the lubricant and the surface tensile stress in the hub can also lead to problems such as stress cracking over the long term. The preferred method of reducing assembly stresses is to cause a temporary change in the interference value by cooling the metal shaft, heating the polymer hub, or both. The most common approach is to simply cool the shaft using a freezer, as this eliminates the potential for problems such as oxidation, warpage, or softening of the polymer hub that could occur if the hub was heated. The temporary change in the shaft diameter, $\Delta\text{Ø}$, is given by:

$$\Delta\text{Ø}_S = \alpha_S \cdot \text{Ø}_S \cdot \Delta T \tag{6.8}$$

where α is the linear coefficient of thermal expansion for the shaft, and ΔT is the temperature change.

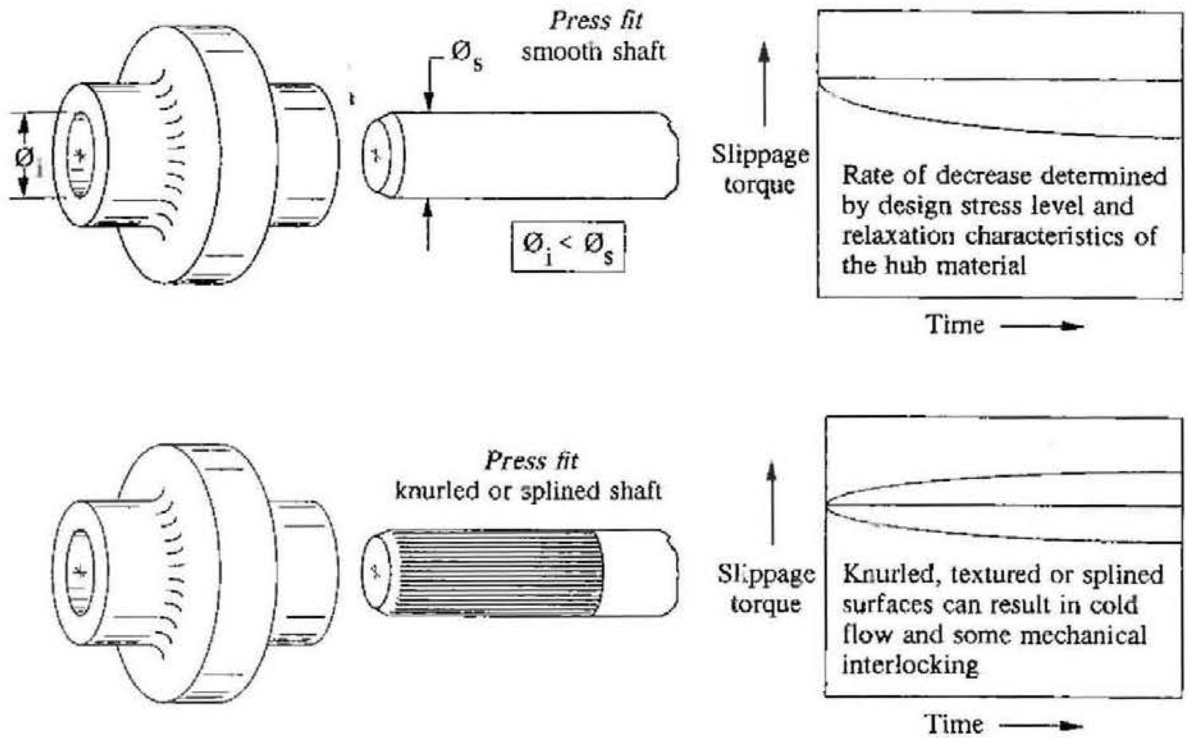


Figure 6.7. The shafts used with press fit assemblies can be smooth, textured or even knurled. With smooth shafts, torsional strength can decrease with time due to stress-relaxation effects. The torsional strength for textured or knurled shafts involves some degree of mechanical interlock.

Torque retention is a significant problem for press fit assemblies, due to stress relaxation. Once a hub is pressed into place over a smooth metal shaft, the contact pressure and torsional strength decrease over the long term as shown in Figure 6.7. However, if the same hub is pressed over a knurled, splined, textured or bead blasted surface, the creep or cold flow of material into the various topographical features can result in a more constant or even an increase in torsional strength over time. Smooth shaft surfaces are recommended for rigid, amorphous polymers, while rougher surfaces can be used with more ductile, semi-crystalline polymers that are less sensitive to stress concentration effects. Mechanical design modifications, such as keyways or other shaft geometries, can also be used to increase the torsional strength of a hub / shaft assembly, however, their use negates the need for and simplicity of the press fit assembly concept.

6.3 Snap Joint Assemblies

6.3.1 Introduction

Snap or interference fit assembly methods for molded plastic products provide an attractive alternative to more conventional assembly approaches. The use of snap joint assemblies is increasing at a rapid rate, since parts assembled using the snap technique satisfy the requirements of both *Design for Assembly* and *Design for Disassembly*. From an assembly viewpoint, snap joints are economical because they are molded directly into the product as an integral feature. This eliminates the need for additional parts / materials such as mechanical fasteners or adhesives. The assembly operations associated with snap fit assemblies are also relatively simple, usually requiring only straight insertion. There is no need for rotational motion or part fixturing. The snap joints can also be designed to be reversible, permitting access for repair and improved product recyclability. Snap assemblies are simple, yet they are perhaps the most versatile means of plastic product assembly [6-14].

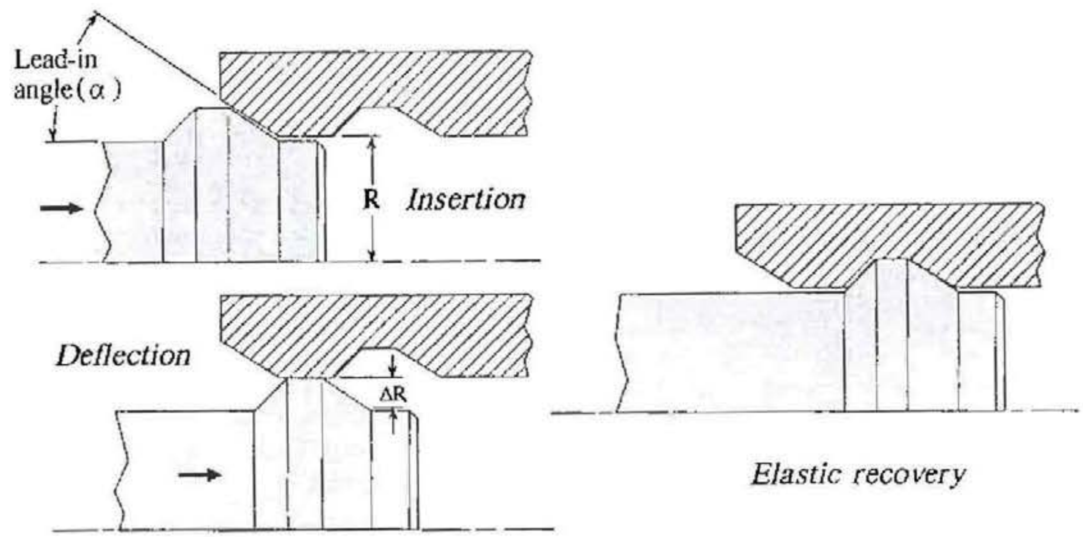


Figure 6.8. Insertion, deflection and recovery. While there are many different snap fit geometries, the snap fitting process always involves a momentary deflection during assembly / disassembly, followed by elastic recovery.

Snap joints can have a variety of geometries, however, the principles of operation remain the same in each case. When two parts are joined using the snap assembly process, a protruding feature on one component, such as a hook or beam, is deflected briefly during the product assembly operation due to an interference, after which the protruding part recovers elastically, and catches in an undercut or indentation on the mating component. The deflection during the assembly operation can be relatively large resulting in high stress or strain levels, however, once the assembly is snapped in place, the components are generally designed to be in a relatively stress free state (unlike press fits) [6].

Perhaps the most significant disadvantage associated with the use of snap assemblies is the consequence of joint failure. Snap assemblies that undergo repeated assembly operations can fail due to fatigue: Even one time assembly applications can fail due to improper handling. This can be a particular problem for snap assemblies produced from brittle, filled, or fiber reinforced polymers. Since the snap member is an integral feature of the molding itself, snap failure can mean component failure. Snaps are difficult or impossible to repair. As a result, it may be desirable to "overdesign" the number of snap joints required for a particular product to account for the possibility of individual snap damage. The redundancy may have some impact on the tool and ultimate product cost, however, the useful service life of the part may be extended. Strain limiting features, adjacent to, the deflecting component, can also be added to limit the permissible deflection and minimize the potential for snap damage.

Another disadvantage associated with the use of snap joint assemblies is the need for tighter control over part tolerances. The tightness of a snap assembly is controlled by both the snap geometry and the stress state after assembly. Excessive interference or stress can lead to the potential for joint failure, while a lack of interference may result in poor location or a loosening of the parts. Preload control can also be difficult to accomplish with snap assemblies, however, creative joint design and careful control over part tolerances improve the ability to control the preload.

6.3.2 Types of Snap Joints

Snap joints are commonly categorized as (i) snap hooks or beams, (ii) annular or ring snaps, (iii) ball and socket snaps, or (iv) torsional snap joints. Snap joints are further categorized as being either separable or non-separable [6,12,14]. Annular snap joints can be used to assemble rotationally symmetric parts. The annular snap joints shown in Figures 6.9 to 6.11 represent common configurations.

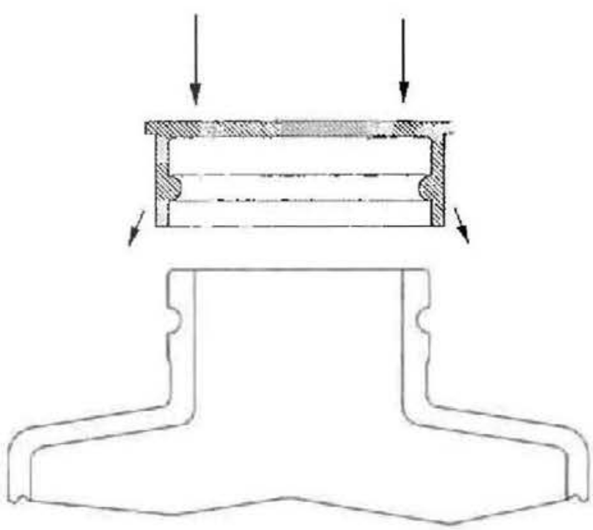


Figure 6.9. Annular snap fits are commonly used with more flexible polymers. A common application for an annular snap assembly is a push on bottle cap.

The bottle cap shown in Figure 6.9 has a circumferential bead that deflects briefly during assembly as it is pushed over the neck of the bottle. The design permits both assembly and disassembly, and is therefore described as reversible. Unlike press fits, snap assemblies are usually designed to be in a stress free (or very low) stress state after assembly.

The cylindrical components shown in Figure 6.10 differ from one another in that the component on the right is reversible, while the one on the left is self locking. The separable joint incorporates both a lead-in and a return angle or "ramp" that permits both insertion and separation, while the inseparable joint is self locking because it incorporates a 90° return angle. These lead-in and return angles can be used as one means to control the relative "push on" and "pull off" forces for a given snap geometry.

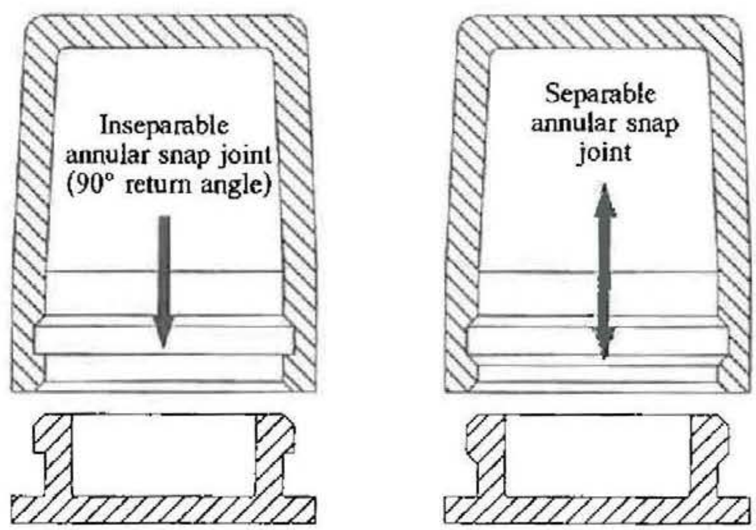


Figure 6.10. Snap assemblies can be designed to be either separable or inseparable.

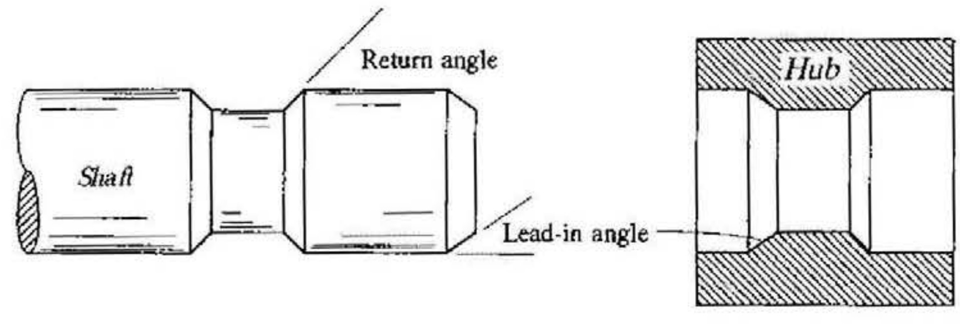


Figure 6.11. The snap fit lead-in and return angles influence the push-on and the push-off forces.

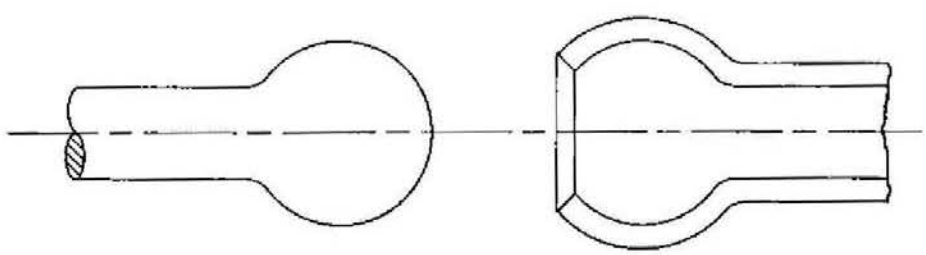


Figure 6.12. Example of a ball and socket snap fit assembly.

The ball and socket snap assembly shown in Figure 6.12 is a modification of the annular snap. Annular snap assemblies are used most commonly with ductile or flexible materials. Push-on / pull-off forces for parts produced in more rigid materials can be extremely high. For these more rigid materials, slotted annular snaps such as that shown in Figure 6.13, are commonly used.

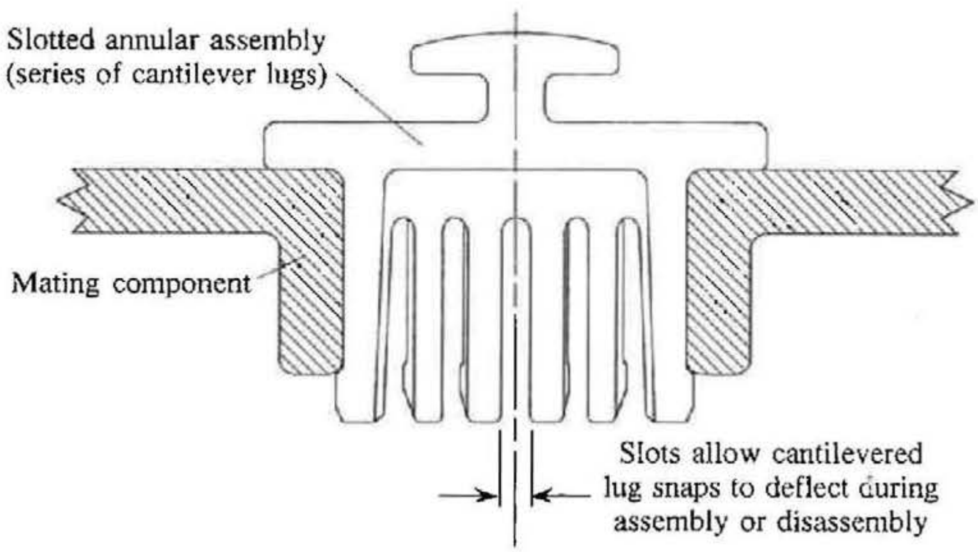


Figure 6.13. The slotted annular snap assembly is actually a series of cantilever snap beams. This approach is more suitable for rigid polymers.

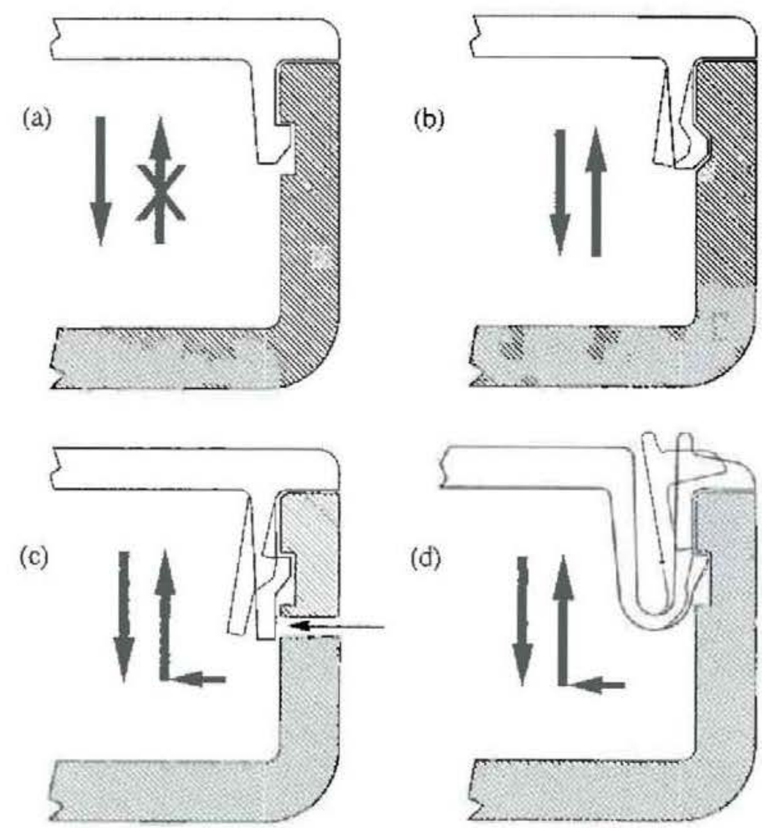


Figure 6.14. Cantilever snap beams are commonly used for the assembly of plastic parts. A variety of both separable and inseparable configurations are possible.