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# Snap-Fit Latch Design

A snap-fit latch is the simplest, quickest, and most cost-effective method of assembling two or more parts. Assembly is accomplished without the use of any additional components or fasteners. A cantilever snap-fit latch basically consists of a protrusion (some type of bead or hook) at one end of the beam and a structural support at the other end. During assembly, pressure is applied to the two parts to be joined and the beam of the latch undergoes deflection due to interference between the hook and mating surface. Once inside the groove, the beam returns to its original shape and the interaction between the hook and the groove holds the assembly together.

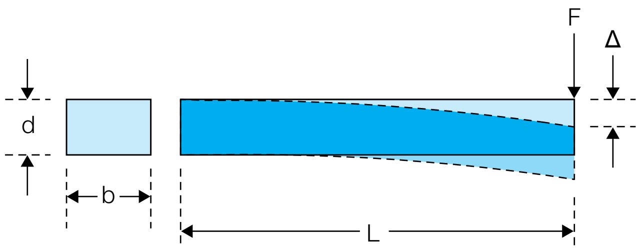
When designing snap-fit latches, one consideration is the expected service. Some latches will be used only once for assembly during the manufacturing process and not usually disassembled. These one-time or permanent latches are often used for non-serviceable items. Other latches will be snapped in and removed many times throughout the life of the item. The type of service will influence the design criteria.

The performance of a snap-fit latch greatly depends upon its engineering design. Snap-fit latches that are not designed properly can break in assembly or even during molding or shipping. One of the key design parameters is the amount of strain caused when the beam is deflected to achieve the snap-fit assembly.

## Calculating or Estimating Strain

A first approximation to the strain inherent in a latch design can be obtained by classical stress analysis. To illustrate this technique, the analysis of the simplest geometry for a latch design, a cantilever beam with a uniform rectangular cross-section, is presented.

**Figure 1:** Cantilever beam with rectangular cross section




## Cantilever Beam

According to the pure bending theory, the maximum stress (σ) in a cantilever beam with uniform rectangular cross-section (Figure 1) is given by the following:

$$(1)\ \sigma = \frac{FLd}{2I} \quad \text{where} \quad I = \frac{bd^3}{12}$$

and the deflection is given by

$$(2)\ \Delta = \frac{FL^3}{3EI}$$

Solving both equations for F gives

$$(3)\ F = \frac{2I\sigma}{Ld}$$

$$(4)\ F = \frac{3EI\Delta}{L^3}$$

Setting these expression equal to each other, simplifying, and solving for the stress gives

$$\sigma = \frac{3EI\Delta}{2L^3}$$

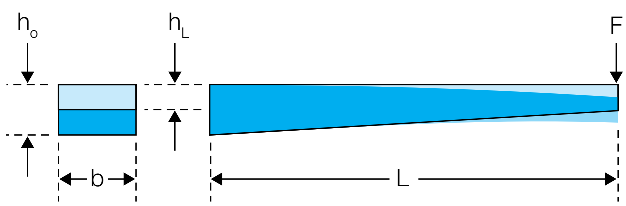
Since $E = \frac{\sigma}{\varepsilon}$, we can substitute and solve for the strain

$$\varepsilon = \frac{3\Delta d}{2L^2}$$

Where:
E = modulus of elasticity
F = force at the end of the latch
I = moment of inertia
L = effective length
σ = stress
b = width of the latch
ε = strain
Δ = deflection

From these equations, it is clear that the strain is highest at the root of the beam and is proportional to the deflection and inversely proportional to the modulus of elasticity E.

**Figure 2:** Tapered beam




## Beam with Uniform Stress (Tapered Beam)

A beam with a uniform rectangular cross-section concentrates the stress at its root, while a tapered beam, with the thickest portion at the base and its thickness decreasing toward the tip, can distribute the stress so that the stress is uniform throughout the entire length of the beam. The taper can be in the plane of the force, perpendicular to the plane of the force, or in both planes.

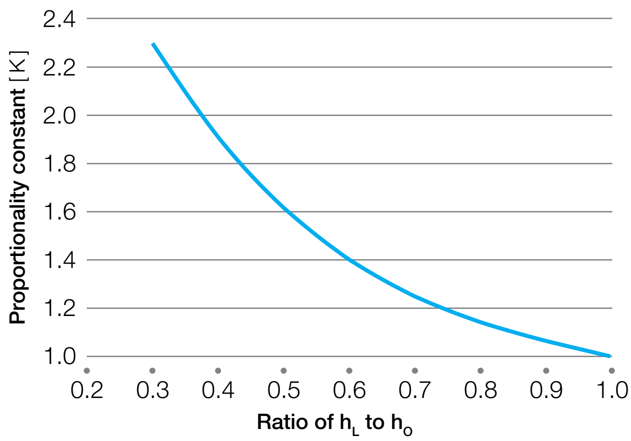
Figure 2 shows the latch tapered in the plane of the force. The taper of a beam can be described by the ratio of the beam height at the small end ($h_L$) to the beam height at the large end ($h_0$). For such a snap-fit latch, the maximum strain can be calculated by using the formula:

$$\varepsilon = \frac{3h_0\Delta}{2L^2K}$$

Where:
K = the proportionality constant for the tapered beam.

The value of K for tapered beams can be found in Figure 3.

**Figure 3:** Proportionality constant for tapered beams




## Accuracy of Strain Calculations

When classical analysis is used, simplifying assumptions are necessary that may not be completely valid. For example, in the cantilever beam example, the end of the beam is assumed to be attached to a totally rigid body that will not deflect when the beam is deflected. In reality, deflecting the beam will cause deflection in the rest of the component. This deflection will reduce the strain on the beam, but calculating the magnitude of the reduction is difficult using classical methods. Therefore, strains calculated by the classical methods will usually be higher than those actually encountered in typical components.

In addition to the errors caused by using simplified equations that don't accurately describe the geometry of the components, plastic materials do not behave in the completely linear fashion assumed by the classical formulas. For most unfilled plastic materials, strain is not a linear function of stress. Allowable strain levels of 5 % to 20 % are common for some of these materials and at these high strains, the analytical approach does not give an accurate prediction. Even highly reinforced plastics only exhibit a linear stress to strain relationship for small strains.

In a majority of plastic latch designs one or both types of non-linearities come into play.

## Finite Element Analysis

Finite element analysis (FEA) is a technique for determining deflection and strains in a structure otherwise too complex for classical mathematical analysis. FEA can incorporate many of the factors commonly ignored in classical calculations, such as shear deflection, deflection of the material around the base, irregular geometry, and non-linear material properties. FEA can be done quickly for a simple latch with uniform longitudinal cross-section by using plane strain elements. FEA will usually give a more accurate estimate of the strain.

**Figure 4:** Typical stress/strain curve for an unfilled ductile resin

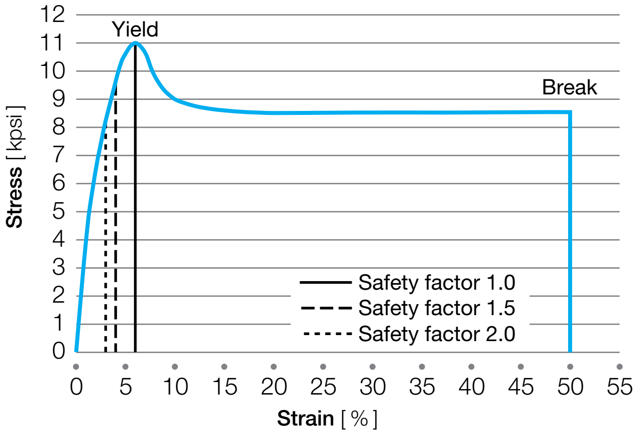
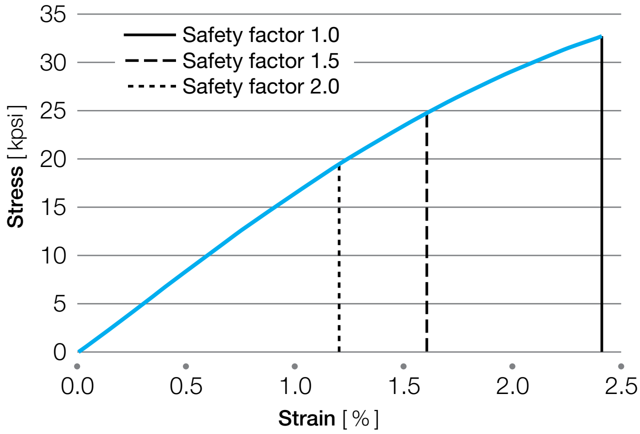



**Figure 5:** Typical stress/strain curve for a glass or mineral filled resin




## Relating Strain Estimates to Material Properties

Plastic materials encompass a wide range of stress/strain behavior. Some unfilled resins are quite ductile. Figure 4 shows typical stress/strain behavior for a ductile resin. This type of resin usually has a distinct yield point, but can endure significant additional strain before failure. Generally, the maximum strain should be held below the strain at yield or the latch will be permanently deformed.

Glass or mineral filled resins will exhibit a much lower strain to failure and will not have a distinct yield point. A typical stress/strain curve for this type of resin is shown in Figure 5. The stress/strain curve provides the designer with the information required to establish the design strain limits. Material data sheets often do not provide the stress/strain curve, but only the elongation at yield and rupture.

The ratio of the strain at yield or rupture to the design strain is known as a safety factor. For example, if the yield elongation or strain is 6 %, and the design maximum strain is 3 %, the design has a safety factor of 2.

As a general rule of thumb, when designing latches for materials that have a distinct yield point, the safety factor of 1 may be used for one-time assembly latches and a factor of 2 should be used for latches that will undergo multiple assemblies and disassemblies. When designing for the materials that do not exhibit a distinct yield point, such as glass or mineral filled polymers, a safety factor of 1.5 should be used for one-time assembly latches and a safety factor of 2 for latches expected to endure multiple assemblies and disassemblies.

### Part Geometry Constraints

The geometrical non-linearity is mainly due to the part configuration. The analytical results are appropriate and of sufficient accuracy for slender beams. From experimental studies, it has been found that the deflection of a latch with length to thickness (L/T) ratio higher than 20 behaves non linearly. Also, in many cases, the base of the latch is not as rigid as it is assumed in the derivation. This is mainly due to the geometry as well as the lower stiffness of the plastic material. For these types of designs, it is highly recommended to use finite element analysis (FEA) for the accurate strain level prediction.

## Design Considerations

There are a number of issues to be considered while designing a snap-fit latch for a particular part. The actual strain level that will be acceptable in any given design depends upon a number of factors, e.g., fiber orientation, distance from gate, and weld-line location. For these reasons, testing of prototype parts is strongly recommended to verify the acceptable performance.

The following design options should be considered where the calculated strain level is higher than the design level of the material before finalizing the design.

**Figure 6:** L-shaped cantilever

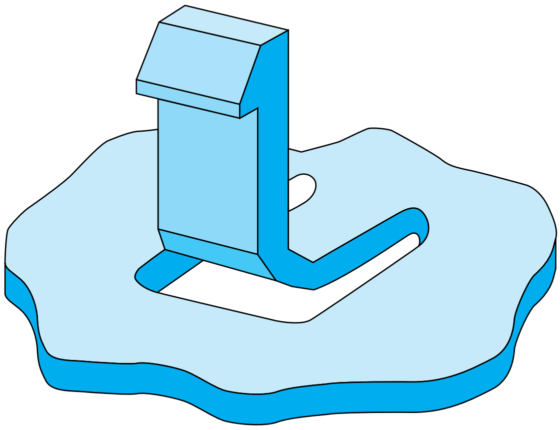
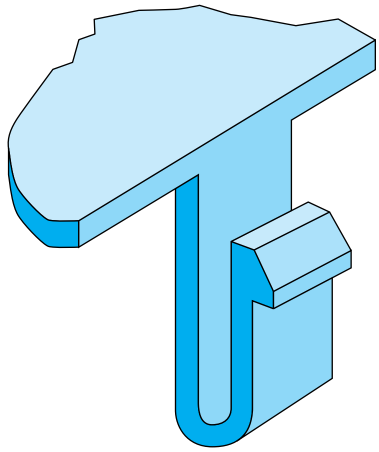


**Figure 7:** U-shaped cantilever



### Length/Thickness Ratio

An important design objective is to prevent the latch from taking permanent set, or retaining residual deflection. The amount of residual deflection depends upon the ratio of length (L) to its thickness (T). The larger the L/T ratio, the greater the transverse deflection that can be accommodated without taking a permanent strain. From the experimental study, it has been found that the deflection of a latch with L/T ratio higher than 20 behaves non-linearly.

### L-Shaped and U-Shaped Latches

These alternative designs can accommodate very large deflections without inducing high strain at the base. They are viable design options for materials with lower strain limits, such as glass or mineral filled thermoplastics.

## Tooling Considerations

Snap-fit latch fasteners often have features such as the hook or the groove that may become undercuts in the tool and require special slides and/or cam-actuated mechanisms. Such devices increase the tooling cost because of their complex design and tight tolerances. In addition, they require more frequent tool maintenance. It is always beneficial to strive for a tool design that eliminates the need for undercuts, which require special side pulls or lifter pins.

### General Guidelines

The following guidelines should be considered in snap-fit latch designs:

- The latch should be designed so that once it is assembled it is essentially at zero stress.
- The snap-fit joints using high strain levels should not be subjected to multiple assemblies and disassemblies.
- Excessive strain can be reduced by reducing beam thickness, increasing the beam length, or reducing the latch deflection.
- Provide a positive stop for latches requiring frequent assembly and disassembly. The positive stop prevents unintentional excessive strain.
- Use tapered latches for optimum weight and performance.
- Use small radii in corners to reduce stress concentration. Increasing the corner radii beyond that required for a smooth corner will make the latch stiffer and increase the stress level at the base. In this case, bigger is not always better

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