

BASF Plastics

Snap-Fit
Design Manual

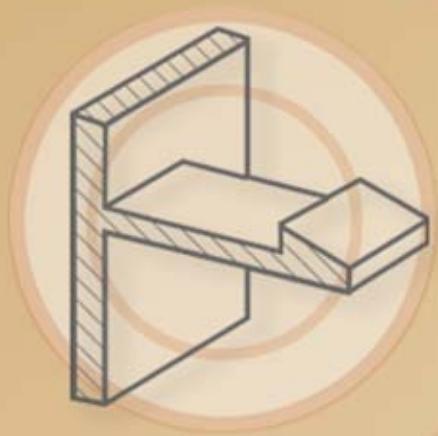


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Snap-Fit Design

This manual will guide you through the basics of snap-fit design, including: types of snap-fit designs and their applications; how to calculate the strength of the unit and amount of force needed for assembly; and the three common causes of failure in snap-fits and how to overcome them.

About BASF Performance Polymers

BASF Plastics is a fully integrated, global supplier of engineering resins from production of feedstocks to the compounding, manufacture and distribution of hundreds of resin grades.

BASF is committed to continuous product development to sustain rapid growth in the nylon resin market. In our Plastics Technology Laboratory, a highly experienced staff of research and development engineers continues to develop new resins to further extend the horizons of product performance.

BASF offers high-quality engineering resins, including:

Ultramid®/Capron® (nylon 6 and 6/6)

Nypel® (a post-industrial nylon 6)

Petra® (post-consumer recycled PET)

Ultradur® PBT Thermoplastic Polymer

Ultraform® Acetal (POM)

Ultrason® High Temp Polymers

These resins from BASF, coupled with the company's concept-through-commercialization expertise, can combine to help make possible the most efficient, cost-effective snap-fit for your product. Our technical support is ready to help you with all your needs. And for more information, you can always visit our web site at www.plasticsportal.com.

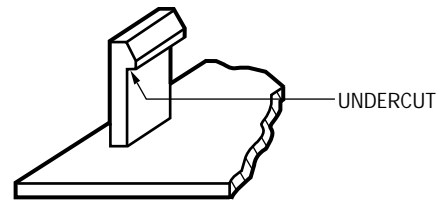
Snap-Fit Design Applications

Why use snap-fits? This chapter will give you a thumbnail sketch of the benefits of snap-fits and the materials used to make them.

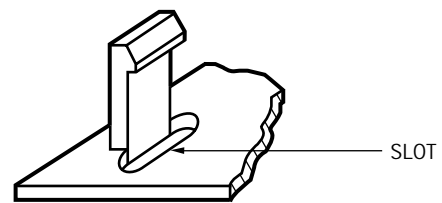
Snap-fits are the simplest, quickest and most cost-effective method of assembling two parts. When designed properly, parts with snap-fits can be assembled and disassembled numerous times without any adverse effect on the assembly. Snap-fits are also the most environmentally friendly form of assembly because of their ease of disassembly, making components of different materials easy to recycle.

Although snap-fits can be designed with many materials, the ideal material is thermoplastic because of its high flexibility and its ability to be easily and inexpensively molded into complex geometries. Other advantages include its relatively high elongation, low coefficient of friction, and sufficient strength and rigidity to meet the requirements of most applications.

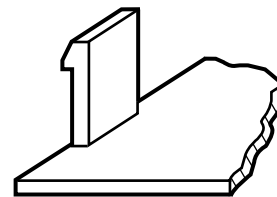
The designer should be aware that the assembly may have some play due to tolerance stack-up of the two mating parts. Some snap-fits can also increase the cost of an injection molding tool due to the need for slides in the mold. An experienced designer can often eliminate the need for slides by adding a slot in the wall directly below the undercut or by placing the snaps on the edge of the part, so they face outward (see Figure I-1).



REQUIRES SLIDE IN MOLD



NO SLIDE REQUIRED



NO SLIDE REQUIRED,
MOLD LESS COMPLEX

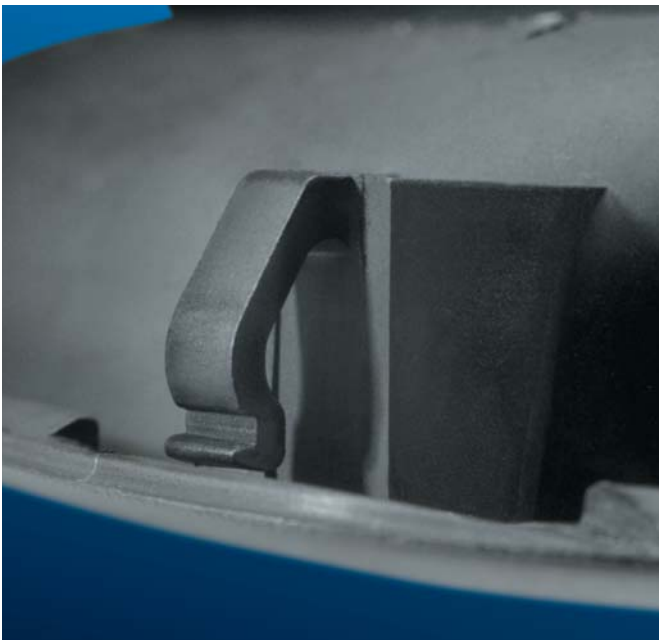
Figure I-1

SNAP-FIT DESIGN APPLICATIONS

Concluding points: Snap-fits solve the problem of creating an inexpensive component that can be quickly and easily joined with another piece. Thermoplastics are the ideal material for snap-fits because they have the flexibility and resilience necessary to allow for numerous assembly and disassembly operations.



Door handle bezel



Backside of bezel

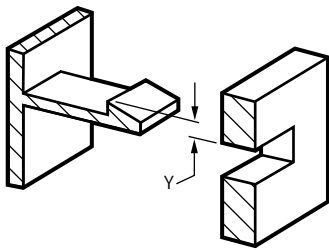


Detail of backside of bezel, cantilever design

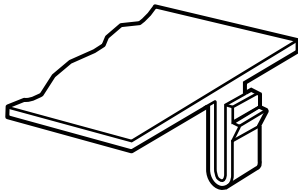
Types of Snap-Fits

This chapter provides an overview of the different types of cantilever snap-fits and gives an idea of when they are used.

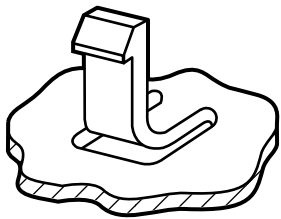
Most engineering material applications with snap-fits use the cantilever design (see Figure II-1) and, thus, this manual will focus on that design. The cylindrical design can be employed when an unfilled thermoplastic material with higher elongation will be used (a typical application is an aspirin bottle/cap assembly).



CANTILEVER



"U" SHAPED CANTILEVER



"L" SHAPED CANTILEVER

Figure II-1

When designing a cantilever snap, it is not unusual for the designer to go through several iterations (changing length, thickness, deflection dimensions, etc.) to design a snap-fit with a lower allowable strain for a given material.

Other types of snap-fits which can be used are the U or L shaped cantilever snaps (see Part V for more detail). These are used when the strain of the straight cantilever snap cannot be designed below the allowable strain for the given material.

Concluding points: Most applications can employ a cantilever type snap-fit in the design. In applications with tight packaging requirements, the U or L shaped snap may be required.



Automotive oil filter snaps



Cordless screw driver housing, cantilever snap-fit

Snap-Fit Design Using Classical Beam Theory

A design engineer's job is to find a balance between integrity of the assembly and strength of the cantilever beam. While a cantilever beam with a deep overhang can make the unit secure, it also puts more strain on the beam during assembly and disassembly. This chapter explains how this balance is achieved.

A typical snap-fit assembly consists of a cantilever beam with an overhang at the end of the beam (see Figure III-1). The depth of the overhang defines the amount of deflection during assembly.

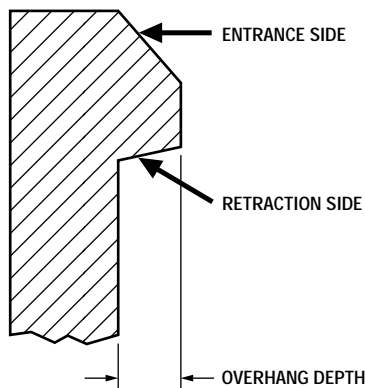
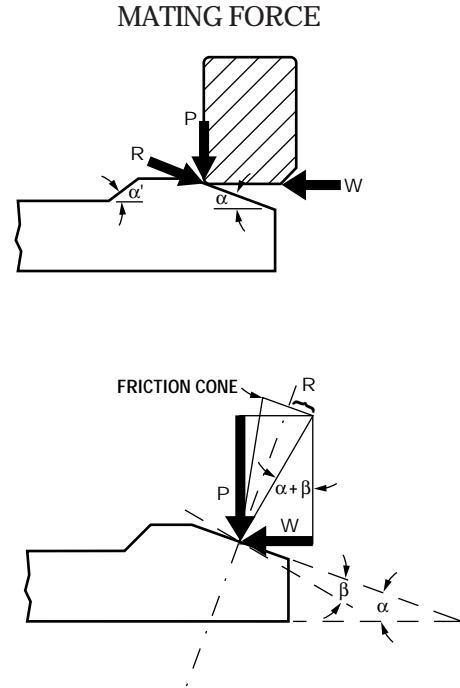


Figure III-1

The overhang typically has a gentle ramp on the entrance side and a sharper angle on the retraction side. The small angle at the entrance side (α) (see Figure III-2) helps to reduce the assembly effort, while the sharp angle at the retraction side (α') makes disassembly very difficult or impossible depending on the intended function. Both the assembly and disassembly force can be optimized by modifying the angles mentioned above.



$$\text{Friction Coefficient } \mu = \tan \beta$$

$$\text{Mating Force } = W$$

$$W = P \tan(\alpha + \beta)$$

$$W = P \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha}$$

Figure III-2

The main design consideration of a snap-fit is integrity of the assembly and strength of the beam. The integrity of the assembly is controlled by the stiffness (k) of the beam and the amount of deflection required for assembly or disassembly. Rigidity can be increased either by using a higher modulus material (E) or by increasing the cross sectional moment of inertia (I) of the beam. The product of these two parameters (EI) will determine the total rigidity of a given beam length.

The integrity of the assembly can also be improved by increasing the overhang depth. As a result, the beam has to deflect further and, therefore, requires a greater effort to clear the overhang from the interlocking hook. However, as the beam deflection increases, the beam stress also increases. This will result in a failure if the beam stress is above the yield strength of the material.

Thus, the deflection must be optimized with respect to the yield strength or strain of the material. This is achieved by optimizing the beam section geometry to ensure that the desired deflection can be reached without exceeding the strength or strain limit of the material.

The assembly and disassembly force will increase with both stiffness (k) and maximum deflection of the beam (Y). The force (P) required to deflect the beam is proportional to the product of the two factors:

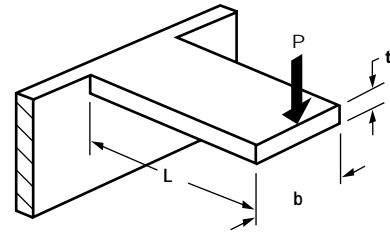
$$P = kY$$

The stiffness value (k) depends on beam geometry as shown in Figure III-3.

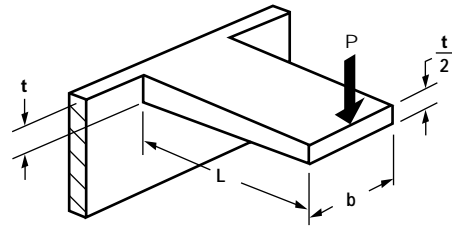
Stress or strain induced by the deflection (Y) is also shown in Figure III-3. The calculated stress or strain value should be less than the yield strength or the yield strain of the material in order to prevent failure.

When selecting the flexural modulus of elasticity (E) for hygroscopic materials, i.e., nylon, care should be taken. In the dry as molded state (DAM), the datasheet value may be used to calculate stiffness, deflection or retention force of snap design. Under normal 50% relative humidity conditions, however, the physical properties decrease and, therefore, the stiffness and retention force reduce while the deflection increases. Both scenarios should be checked.

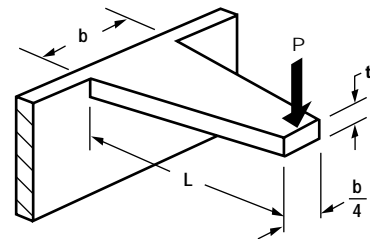
Cantilever Beam: Deflection-Strain Formulas



- I) Uniform Cross Section, Fixed End to Free End
 Stiffness: $k = \frac{P}{Y} = \frac{Eb}{4} \left(\frac{t}{L}\right)^3$
 Strain: $\epsilon = 1.50 \left(\frac{t}{L^2}\right) Y$



- II) Uniform Width, Height Tapers to t/2 at Free End
 Stiffness: $k = \frac{P}{Y} = \frac{Eb}{6.528} \left(\frac{t}{L}\right)^3$
 Strain: $\epsilon = 0.92 \left(\frac{t}{L^2}\right) Y$



- III) Uniform Height, Width Tapers to b/4 at Free End
 Stiffness: $k = \frac{P}{Y} = \frac{Eb}{5.136} \left(\frac{t}{L}\right)^3$
 Strain: $\epsilon = 1.17 \left(\frac{t}{L^2}\right) Y$

Where:

- E = Flexural Modulus
- P = Force
- Y = Deflection
- b = Width of Beam

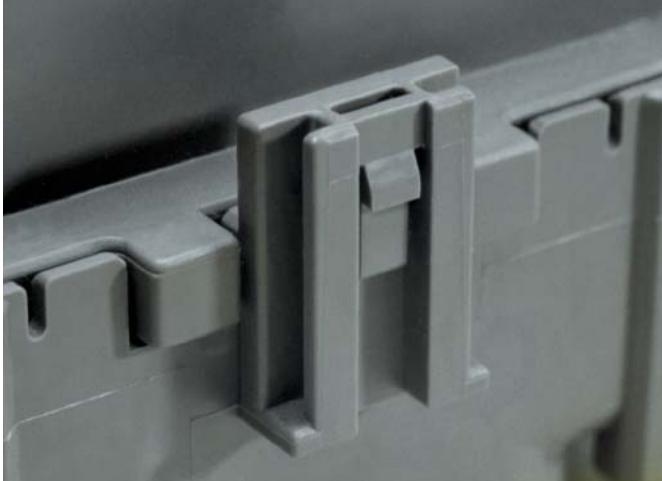
Figure III-3

SNAP-FIT DESIGN USING CLASSICAL BEAM THEORY

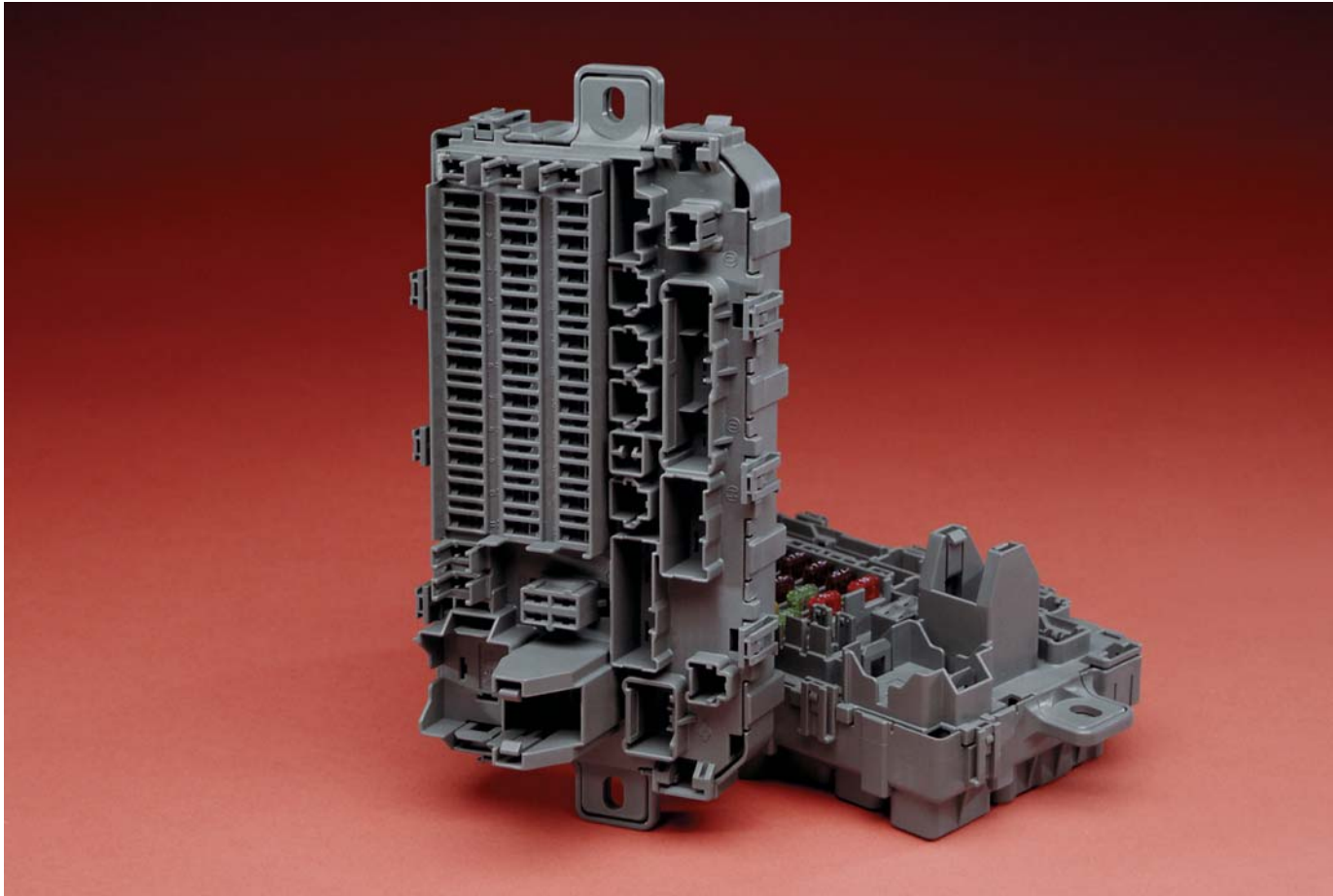
Concluding points: In a typical snap-fit, the strength of a beam is dependent on its geometry and maximum deflection during assembly. The force to assemble and disassemble snap-fit assemblies is highly dependent on the overhang entrance and retraction angles.



Close-up of automotive fuse box, snap on sides of box



Close-up of automotive fuse box snap



Close-up of automotive fuse box, full view

Improved Cantilever Snap-Fit Design

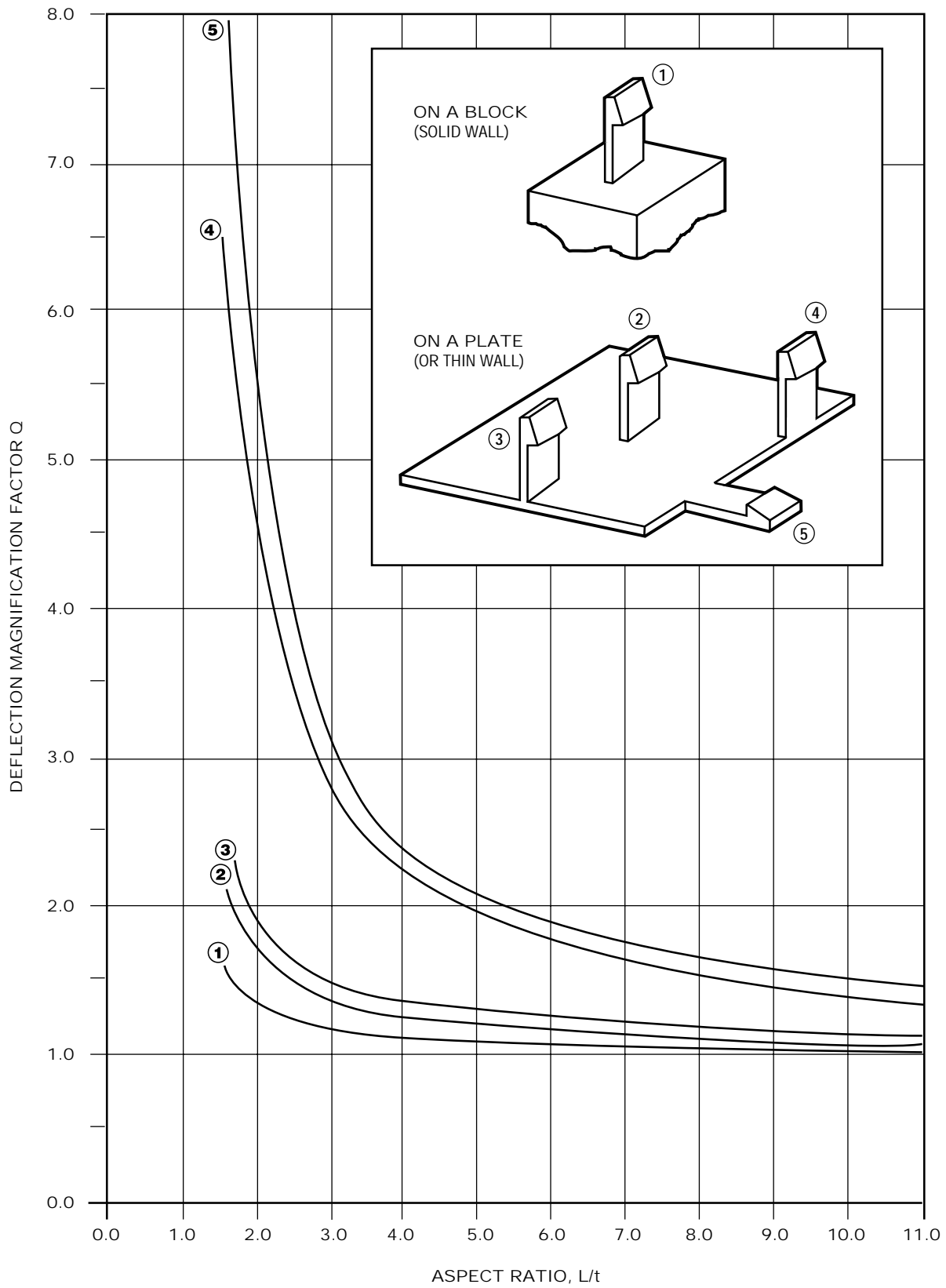
The cantilever beam formulas used in conventional snap-fit design underestimate the amount of strain at the beam/wall interface because they do not include the deformation in the wall itself. Instead, they assume the wall to be completely rigid with the deflection occurring only in the beam. This assumption may be valid when the ratio of beam length to thickness is greater than about 10:1. However, to obtain a more accurate prediction of total allowable deflection and strain for short beams, a magnification factor should be applied to the conventional formula. This will enable greater flexibility in the design while taking full advantage of the strain-carrying capability of the material.

BASF Plastics has developed a method for estimating these deflection magnification factors for various snap-fit beam/wall configurations as shown in Figure IV-1. The results of this technique, which have been verified both by finite element analysis and actual part testing¹, are shown graphically in Figure IV-1. Figure IV-2 shows similar results for beams of tapered cross section (beam thickness decreasing by 1/2 at the tip).

Snap-Fit Design Examples 1 & 2 illustrate this procedure for designing snap-fits, including calculating the maximum strain developed during assembly and predicting the snap-in force required.

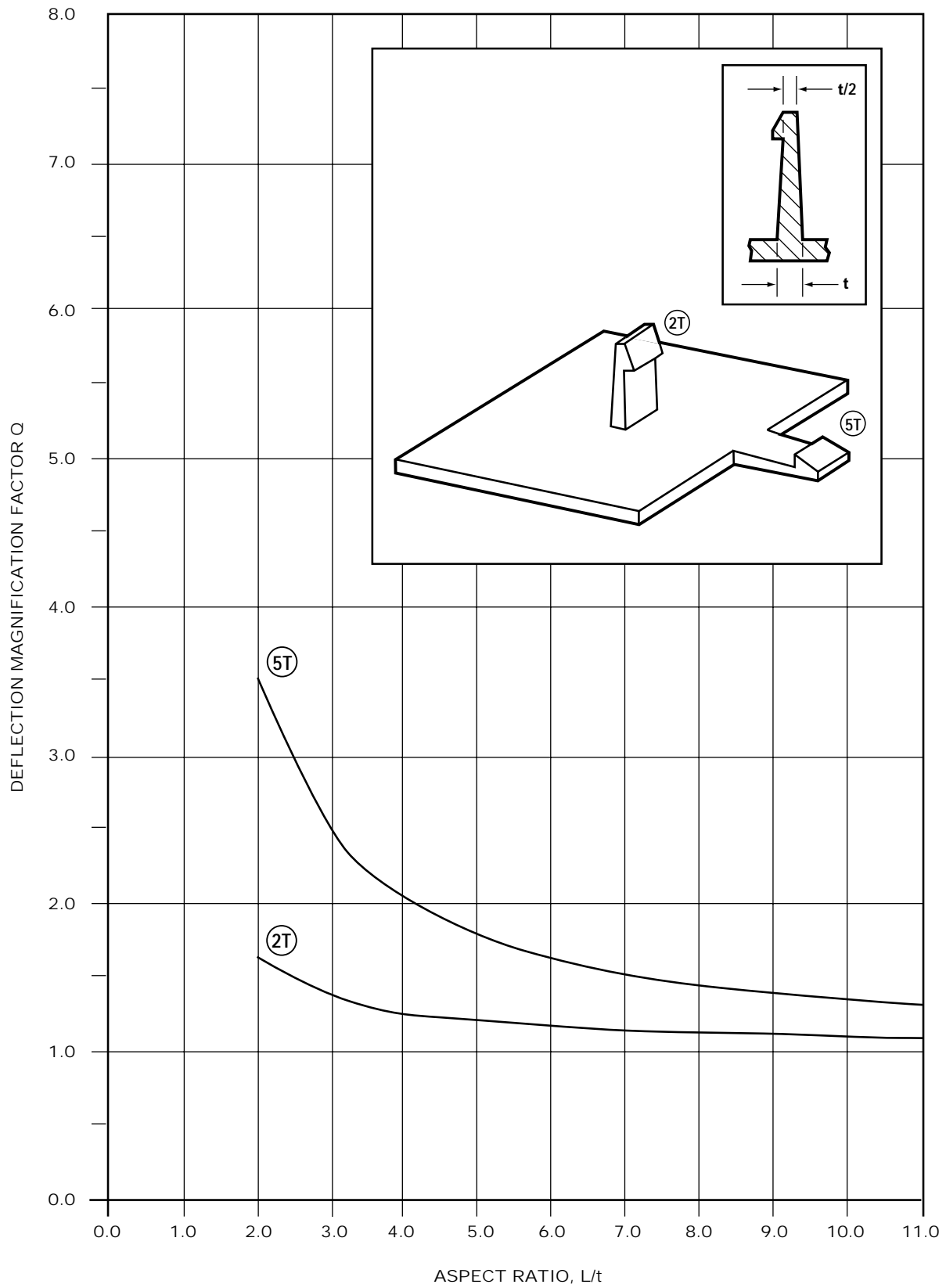
¹ Chul S. Lee, Alan Dubin and Elmer D. Jones, Short Cantilever Beam Deflection Analysis Applied to Thermoplastic Snap-Fit Design, 1987 SPE ANTEC, held in Los Angeles, California, U.S.A.

IMPROVED CANTILEVER SNAP-FIT DESIGN



Uniform Beam, Q Factor
Figure IV-1

IMPROVED CANTILEVER SNAP-FIT DESIGN



Tapered Beam, Q Factor
Figure IV-2

Improved Formulas

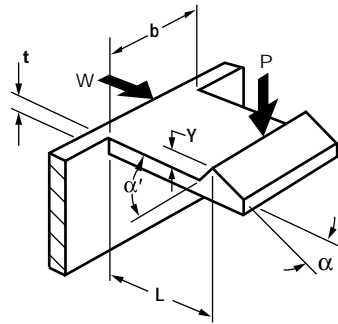


Figure IV-3

MAXIMUM STRAIN (@ BASE)

$$\epsilon = 1.5 \frac{tY}{L^2 Q}$$

MATING FORCE

$$W = p \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha}$$

$$p = \frac{bt^2 E \epsilon}{6L}$$

Where:

- W = Push-on Force
- W' = Pull-off Force
- P = Perpendicular Force
- μ = Coefficient of Friction
- α = Lead Angle
- α' = Return Angle
- b = Beam Width
- t = Beam Thickness
- L = Beam Length
- E = Flexural Modulus
- ε = Strain at Base
- ε_o = Allowable Material Strain
- Q = Deflection Magnification Factor
(refer to Figure IV-2 for proper Q values)
- Y = Deflection

Allowable Strain Value, ε_o

MATERIAL	UNFILLED	30% GLASS
PEI	9.8% ⁽²⁾	
PC	4% ⁽¹⁾ - 9.2% ⁽²⁾	
Acetal	1.5% ⁽¹⁾	
Nylon 6 ⁽⁴⁾	8% ⁽⁵⁾	2.1% ⁽¹⁾
PBT	8.8% ⁽²⁾	
PC/PET	5.8% ⁽²⁾	
ABS	6% - 7% ⁽³⁾	
PET		1.5% ⁽¹⁾

Table IV-I

NOTES:

- ⁽¹⁾ 70% of tensile yield strain value
- ⁽²⁾ G.G. Trantina. *Plastics Engineering*. August 1989.
- ⁽³⁾ V.H. Trumbull. 1984 ASME Winter Annual Conference
- ⁽⁴⁾ DAM - Dry As Molded condition
- ⁽⁵⁾ BASF test lab; Note 4% should be used in Mating Force Formula

Coefficient of Friction⁽¹⁾

MATERIAL	μ
PEI	0.20 - 0.25
PC	0.25 - 0.30
Acetal	0.20 - 0.35
Nylon 6	0.17 - 0.26
PBT	0.35 - 0.40
PC/PET	0.40 - 0.50
ABS	0.50 - 0.60
PET	0.18 - 0.25

Table IV-II

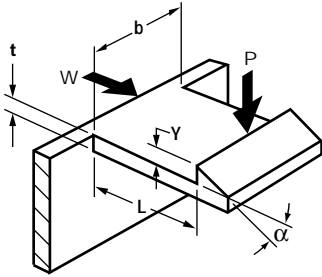
NOTES:

- ⁽¹⁾ Material tested against itself



Wheel cover with cantilever snaps

Snap-Fit Design Example #1



GIVEN:
 Material ⇒ Petra 130 (PET)
 $t = 0.10$ in
 $L = 0.50$ in
 $b = 0.25$ in
 $E = 1.3 (10^6)$ psi
 $\mu = 0.2$ (From Table IV-II, Coefficient of Friction)
 $\alpha = 30.0^\circ$
 $\epsilon_o = 1.5\%$ (From Table IV-I, Allowable Strain Value)

Figure IV-4

DETERMINE:

- A) THE MAXIMUM DEFLECTION OF SNAP
- B) THE MATING FORCE

SOLUTION:

- A) THE MAXIMUM ALLOWABLE DEFLECTION OF SNAP

$$\epsilon_o = 1.5 \frac{tY_{max}}{L^2 Q} \Rightarrow Y_{max} = \frac{\epsilon_o L^2 Q}{1.5 t}$$

$$\frac{L}{t} = 5.0 \Rightarrow Q = 2.0 \text{ (from Q Factor Graph)}$$

$$Y_{max} = \frac{(0.015)(0.5)^2 (2.0)}{(1.5)(0.1)} = 0.050 \text{ in}$$

Therefore, in an actual design, a smaller value for deflection (Y) would be chosen for an added factor of safety.

- B) THE MATING FORCE

$$P = \frac{bt^2 E \epsilon_o}{6L}$$

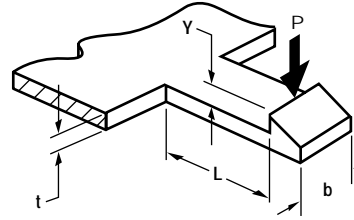
$$P = \frac{(0.25)(0.1)^2 (1.3)(10^6) (0.015)}{6(0.5)} = 16.2 \text{ lb}$$

$$W = P \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha}$$

$$W = 16.2 \frac{0.2 + \tan 30^\circ}{1 - 0.2 (\tan 30^\circ)} = 14.2 \text{ lb}$$

Therefore, it will take 14.2 lb mating force to assemble parts, if the part deflected to the material's allowable strain.

Snap-Fit Design Example #2



GIVEN:
 Material ⇒ Unfilled Nylon 6
 $t = 0.063$ in
 $Y = 0.090$ in
 $L = 0.225$ in
 $b = 0.242$ in

Figure IV-5

DETERMINE:

IS THIS TYPE OF SNAP-FIT ACCEPTABLE FOR USE IN NYLON 6 (CAPRON 8200 NYLON)

SOLUTION:

$$\epsilon = 1.5 \frac{tY}{L^2 Q} \text{ (From Q Factor Graph, Figure IV-1)}$$

$$\frac{L}{t} = 3.57 \Rightarrow Q = 2.7$$

$$\epsilon = 1.5 \frac{(0.063)(0.090)}{(0.225)^2 (2.7)} = 6.2\%$$

Therefore, it is acceptable for unfilled Nylon 6 (See Allowable Strain Value, Table IV-1).

Concluding points: Unlike conventional formulas, BASF includes the deflection magnification factor in all calculations. The examples show how to calculate the maximum strain during assembly and how to predict the force needed for assembly.



Close-up of automotive wheel cover snaps

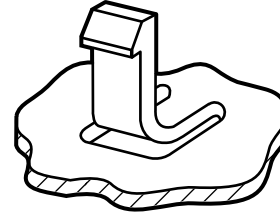
U & L Shaped Snaps

The cantilever beam snap-fit design isn't appropriate for all applications. This chapter defines L and U shaped snaps and tells when they are used.

Occasionally, a designer will not be able to design a cantilever snap-fit configuration with a strain below the allowable limit of the intended material. This is usually due to limited packaging space which can restrict the length of the snap. This is the ideal time to consider using either an L shaped snap or a U shaped snap.

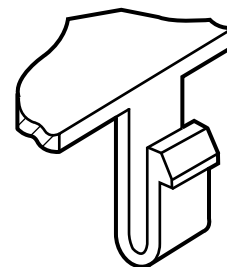
The L shaped snap (see Figure V-1) is formed by designing in slots in the base wall which effectively increases the beam length and flexibility compared to a standard cantilever beam. This allows the designer to reduce the strain during assembly below the allowable limit of the selected material. It should be noted that adding a slot to the base wall may not be acceptable in some designs for cosmetic or air flow concerns.

The U shaped snap (see Figure V-2) is another way to increase the effective beam length within a limited space envelope. With this design, even materials with low allowable strain limits (such as highly glass-filled materials) can be designed to meet assembly requirements. The U shaped design usually incorporates the undercut on the outer edge of the part to eliminate the need for slide in the mold, unless a slot is acceptable in the wall from which the snap projects.



"L" SHAPED CANTILEVER

Figure V-1



"U" SHAPED CANTILEVER

Figure V-2

U & L SHAPED SNAPS (CONSTANT CROSS SECTION)

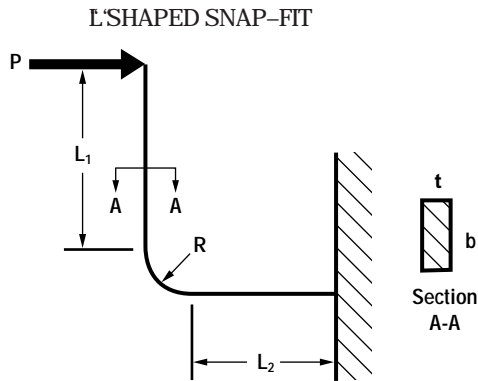


Figure V-3

$$L_2 = \frac{(6/\epsilon_o)Yt(L_1 + R) - 4L_1^3 - 3R(2\pi L_1^2 + \pi R^2 + 8L_1R)}{12(L_1 + R)^2}$$

or,

$$Y = \frac{P}{12EI} [4L_1^3 + 3R(2\pi L_1^2 + \pi R^2 + 8L_1R) + 12L_2(L_1 + R)^2]$$

Where:

- L_2 = Length of slot as shown in sketch
- ϵ_o = Allowable strain of material
- Y = Maximum deflection required in direction of force
- t = Thickness
- L_1 = Length as shown in sketch
- R = Radius as shown in sketch (at neutral axis)
- P = Force
- b = Beam Width
- E = Flexural Modulus
- I = Moment of Inertia

L Shaped Snap-Fit Example

A) Calculate the minimum length (L_2) of the slot (see sketch, Figure V-3) in the main wall for Capron 8233 nylon in the configuration below. The required deflection is .38 inches.

B) Calculate the required force (P) to deflect the snap .38 inches.

GIVEN:

$$\epsilon_{8233} = .021$$

$$t = .1 \text{ in}$$

$$L_1 = .5 \text{ in}$$

$$R = .12 \text{ in}$$

I = Moment of Inertia (rectangle)

$$I = \frac{bt^3}{12} = \frac{1(.1)^3}{12} = 8.333(10^{-5})$$

$$E = 1.31 (10^6)$$

$$b = 1.0 \text{ in}$$

$$Y = .38$$

$$A) L_2 = \frac{(6/\epsilon) Yt(L_1 + R) - 4L_1^3 - 3R(2\pi L_1^2 + \pi R^2 + 8L_1R)}{12(L_1 + R)^2}$$

$$= \frac{(6/.021)(.38)(.1)(.62) - 4(.5)^3 - .36[.5\pi + .12^2\pi + 4(.12)]}{12(.62)^2}$$

$$L_2 = \underline{1.187 \text{ in}}$$

$$B) Y = \frac{P}{12EI} [4L_1^3 + 3R(2\pi L_1^2 + \pi R^2 + 8L_1R) + 12L_2(L_1 + R)^2]$$

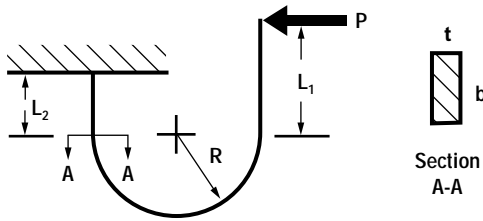
$$.38 = \frac{P}{(12)(1.31)(10^6)(8.333)(10^{-5})} [4(.5)^3 + (.36)[.5\pi + .12^2\pi + 8(.5).12] + 12(1.187)(.62)^2]$$

$$.38 = \frac{P}{1.31(10^8)} (6.718)$$

$$P = \underline{74.1 \text{ lb}}$$

U & L SHAPED SNAPS

U Shaped Snap-Fit



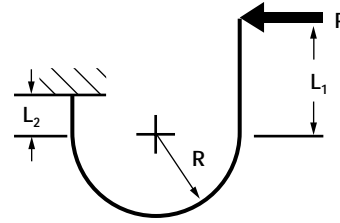
Case 1

$$Y = \frac{\epsilon}{9(L_1 + R)t} [6L_1^3 + 9R \{L_1(2\pi L_1 + 8R) + \pi R^2\} + 6L_2(3L_1^2 - 3L_1L_2 + L_2^2)]$$

or,

$$Y = \frac{P}{18EI} [6L_1^3 + 9R \{L_1(2\pi L_1 + 8R) + \pi R^2\} + 6L_2(3L_1^2 - 3L_1L_2 + L_2^2)]$$

U Shaped Snap Example #1



Case 1

A) Calculate the amount of deflection at the tip of the beam for a 1.0 pound load

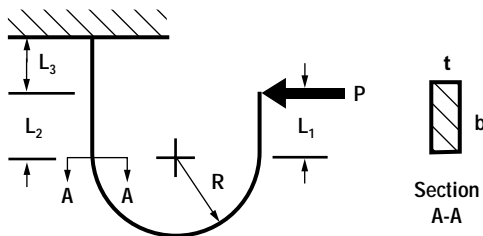
GIVEN:

- P = 1.0 lb
- I = 0.833 x 10⁻⁴ in⁴ = bt³/12 (rectangular cross section)
- E = 534,000 psi
- R = 0.15 in
- L₁ = 1.4 in
- L₂ = 0.973 in
- t = 0.1 in
- b = 1.0 in

$$A) Y = \frac{P}{18EI} [6L_1^3 + 9R \{L_1(2\pi L_1 + 8R) + \pi R^2\} + 6L_2(3L_1^2 - 3L_1L_2 + L_2^2)]$$

$$Y = \frac{1}{18(534,000)(0.833 \times 10^{-4})} [6(1.4)^3 + 9(0.15) \{ (1.4)(2\pi \cdot 1.4 + 8 \cdot 0.15) + \pi(0.15)^2 \} + 6(0.973) \{ 3(1.4)^2 - 3(1.4)(0.973) + (0.973)^2 \}]$$

$$= 0.064 \text{ in}$$



Case 2

$$Y = \frac{\epsilon}{3(L_1 + R)t} [4L_1^3 + 2L_3^3 + 3R \{L_1(2\pi L_1 + 8R) + \pi R^2\}]$$

or,

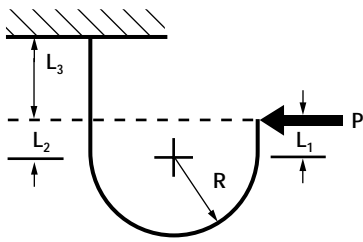
$$Y = \frac{P}{6EI} [4L_1^3 + 2L_3^3 + 3R \{L_1(2\pi L_1 + 8R) + \pi R^2\}]$$

Where:

Variables defined on previous page.

U" & L" SHAPED SNAPS

U" Shaped Snap Example #2



Case 2

A) Calculate the amount of deflection at the tip of the beam for a 1.0 pound load

GIVEN:

$$\begin{aligned} I &= 0.833 \times 10^{-4} \text{ in}^4 \\ E &= 534,000 \text{ psi} \\ R &= 0.15 \text{ in} \\ L_1 &= 0.7 \text{ in} \\ L_1 &= L_2 \\ L_3 &= 0.273 \text{ in} \\ t &= 0.1 \text{ in} \end{aligned}$$

$$\begin{aligned} Y &= \frac{P}{6EI} [4L_1^3 + 2L_3^3 + 3R \{L_1(2\pi L_1 + 8R) + \pi R^2\}] \\ &= \frac{1}{6(534,000)(0.833 \times 10^{-4})} [4(0.7)^3 + 2(0.273)^3 + \\ &\quad 3(0.15) \{0.7(2\pi \cdot 0.7 + 8(0.15)) + \pi (0.15)^2\}] \\ &= \underline{0.012 \text{ in}} \end{aligned}$$



Inset shot of a U" shaped snap-fit design

Concluding points: Snap-fits can use either the U" or L" shaped design to overcome space limitations. Both the L" and U" shaped snaps effectively reduce strain during assembly, thus making it ideal for materials with lower allowable strain limits.



Automotive wheel cover



Close-up of above cover backside featuring the L" shaped snap-fit design (from a top angle)

Part VI

General Design Guidelines

Three basic issues should be reviewed before finalizing a snap-fit design: stress concentration, creep/relaxation, and fatigue. Below are descriptions of these problems and suggestions to prevent them. All should be considered as part of good design practice for any thermoplastic design.

The single most common cause of failure in snap-fits is stress concentration due to a sharp corner between the snap-fit beam and the wall to which it is attached. Since this location normally coincides with the point of maximum stress, a sharp corner can increase the stress beyond the strength of the material, causing point yielding or breakage. This is more critical for rigid plastics like glass-reinforced nylon, which have relatively low ultimate elongation. More ductile materials, like unreinforced nylon, tend to yield and deform before they break, redistributing the peak stress over a broader region. One solution is to incorporate a fillet radius at the juncture between the beam and the wall (see Figure VI-1), so that the ratio of radius to wall thickness (R/t) is at least 50%. Going beyond 50% results in a marginal increase in strength and may cause other problems like internal voids and sink marks. If sink marks are an issue, a smaller radius can be used, but it may increase the stress in this area. Another option is to add the radius only on the tensile side of the beam.

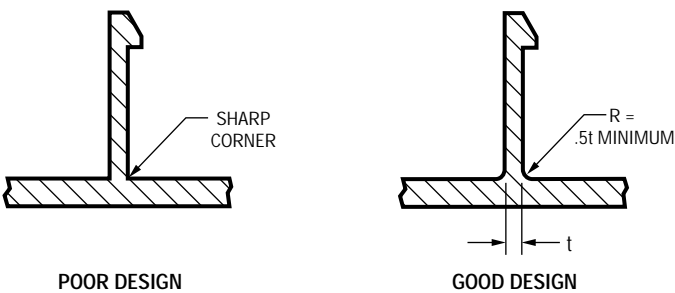


Figure VI-1

Creep, or more accurately stress relaxation, can result in a reduction of the holding force between the two components connected by the snap-fit. Stress relaxation will occur gradually over time. If there is a gasket or seal

between the parts, relaxation at the joint can result in loss of seal pressure, resulting in leakage of the contained fluid. Another problem often seen is excessive play between the parts due to tolerance variations, sometimes resulting in noise and vibration. Several ways to minimize these phenomena include: designing a low stress snap beam, designing the snap-fit to incorporate a 90° return angle so that it relaxes in tension versus bending (see Figure VI-2). This will prevent the mating part from slipping past or becoming loose. Another way is to use a large return angle and increase the land length in the return angle area (see Figure VI-3). Increasing the overhang depth and evaluating the worst case scenario in a tolerance study will allow the design to retain given pull-off force even after relaxation occurs.

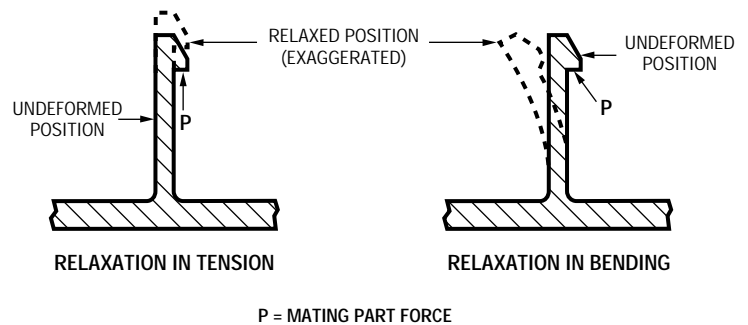


Figure VI-2

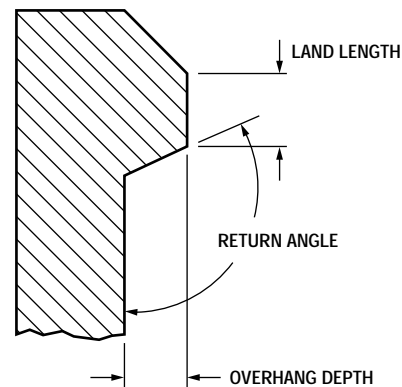


Figure VI-3

GENERAL DESIGN GUIDELINES

Fatigue, or repetitive loading, is the third major cause of failure. Fatigue concerns primarily apply if hundreds or thousands of cycles are anticipated. While the design stress level might be well within the strength of the material, the repeated application of this stress can result in fatigue failure at some point in the future. Some polymers perform better than others in this regard, making them ideal candidates for snap-fits or living hinges that must flex repeatedly. The first way to avoid a fatigue failure is to choose a material known to perform well in fatigue. This can be done by comparing the so-called S-N curves of the materials, which show the expected number of cycles to failure at various stress levels and at different temperatures of exposure. The second way, still using the S-N curves, is to choose a design stress level, at the correct temperature, that results in the required number of load applications prior to failure. This method will usually be conservative since S-N curves are typically generated at much higher frequencies than would be anticipated for repeated application of a snap-fit assembly.

For hygroscopic materials like nylon, the effects of moisture on final part dimensions and mechanical properties also must be considered. For further information, please consult the BASF Plastics Design Solutions Guide.

Concluding points: There are a number of ways to overcome the issues of stress concentration, stress relaxation and fatigue. A well thought-out design and using the right polymer for a given application will minimize these issues. This allows the application to benefit from all the advantages of a snap-fit design.



Aerator



Close-up of automotive fuel rail cover, snap-fit design



Close-up of truck mirror patch cover



Circular saw handle inset shot featuring snap-fit closure and mating

Notes

English/Metric Conversion Chart

To Convert English System	To Metric System	Multiply English Value by...
DISTANCE		
inches	millimeters	25.38
feet	meters	0.30478
MASS		
ounce (avdp)	gram	28.3495
pound	gram	453.5925
pound	kilogram	0.4536
U.S. ton	metric ton	0.9072
VOLUME		
inch ³	centimeter ³	16.3871
inch ³	liter	0.016387
fluid ounce	centimeter ³	29.5735
quart (liquid)	decimeter ³ (liter)	0.9464
gallon (U.S.)	decimeter ³ (liter)	3.7854
TEMPERATURE		
degree F	degree C	$(^{\circ}\text{F} - 32) / 1.8 = ^{\circ}\text{C}$
PRESSURE		
psi	bar	0.0689
psi	kPa	6.8948
ksi	MN/m ²	6.8948
psi	MPa	0.00689
ENERGY AND POWER		
in lb _f	Joules	0.113
ft lb _f	Joules	1.3558
kW	metric horsepower	1.3596
U.S. horsepower	Kw	0.7457
Btu	Joules	1055.1
BTU in / (hr ft ² F)	W/m ² K	0.1442
VISCOSITY		
poise	Pa s	0.1
BENDING MOMENT OR TORQUE		
ft lb	N m	1.356
DENSITY		
lb/in ³	g/cm ³	27.68
lb/ft ³	kg/m ³	16.0185
NOTCHED IZOD		
ft lb/in	J/m	53.4

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