TICONA

SNAP-FITS FOR ASSEMBLY AND DISASSEMBLY

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GENERAL DISCUSSION

An economical and quick method of joining plastic parts is by a snap-fit joint. A snap-fit joint can be designed so it is easily separated or so that it is inseparable, without breaking one of its components. The strength of the snap-fit joint depends on the material used, its geometry and the forces acting on the joint.

Most all snap-fit joint designs share the common design features of a protruding ledge and a snap foot. Whether the snap joint is a cantilever or a cylindrical fit, they both function similarly.

When snap-fit joints are being designed, it is important to know the mechanical stresses to be applied to the snap beams after assembly, the required mechanical stresses or strains on the snap beams during assembly, the number of times the snap joint will be engaged and disengaged, and the mechanical limits of the material(s) to be used in the design.

Reasons to Use Snap-Fits:

- Reduces assembly costs.
- Are typically designed for ease of assembly and are often easily automated.
- Replaces screws, nuts, and washers.
- Are molded as an integral component of the plastic part.
- No welding or adhesives are required.
- They can be engaged and disengaged.

Things To Be Aware of When Using Snap-Fits:

- Some designs require higher tooling cost.
- They are susceptible to breakage due to mishandling and abuse prior to assembly.
- Snap-fits that are assembled under stress will creep.
- It is difficult to design snap-fits with hermetic seals. If the beam and/or ledge relaxes, it could decrease the effectiveness of the seal.

TYPES OF SNAP-FIT JOINTS

There are a wide range of snap-fit joint designs. In their basic form, the most often used are the cantilever beam (snap leg), Figure 1, and the cylindrical snap-fit joint, Figure 2. For this reason, these two designs and designs derived from these basics are covered in this text.

Figure 1
Cantilever Snap

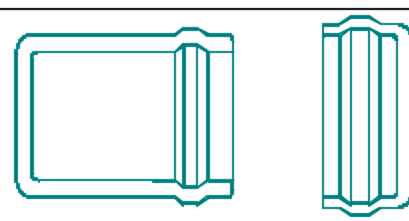


Figure 2
Cylindrial Snap

<u>Cantilever Snap Beams</u>

Using the standard beam equations, we can calculate the stress and strain during assembly of the snap beam. If we stay below the elastic limit of the material, we know the flexing beam will return to its original position. However, for such designs, there is usually not enough holding power with the low forces or small deflections involved.

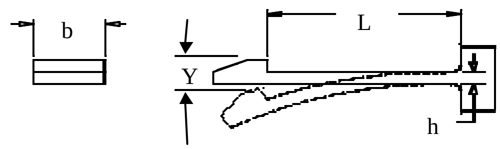


Figure 3

Therefore, much higher deformations are generally used. With most plastic materials, the bending stress calculated by using simple linear bending methods (Equations 1-3) can far exceed the recognized yield strength of the material. This is particularly true when large deflections are used and when the assembly occurs rapidly.

$$s = \frac{FL}{Z}$$ **Equation 1**

$$Y = \frac{FL^3}{3EI}$$ **Equation 2**

$$s = Ee$$ **Equation 3**

Therefore, it often appears as if the beam momentarily passes through the maximum deflection or strain, greatly exceeding its yield strength while showing no ill effects from the event.

What actually happens is described later.

For the present, simply note that the snap beams are usually designed to a stain rather than a stress.

<u>Where</u>:
σ = Maximum stress on beam
F = Force on the beam
L = Length of the beam
Z = $\frac{I}{c}$, Section Modulus
c = $\frac{d}{2}$ = Half the beam height
I = $\frac{bd^3}{12}$, Moment of inertia
h = Beam height
b = Beam width
ε = Maximum strain on beam
Y = Beam deflection

The strain should not exceed the allowable dynamic strain for the particular material being used. By combining Equations 1-3, the design equation (Equation 4) can be produced. Note that the strain is written in terms of the height, length, and deflection of the beam.

$$e = \frac{3YH}{2L^2}$$ **Equation 4**

Strain Guidelines

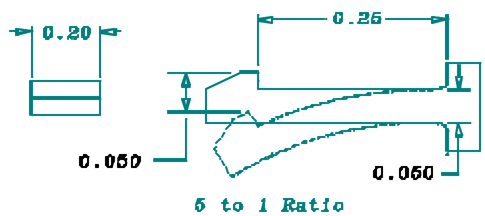


Figure 4

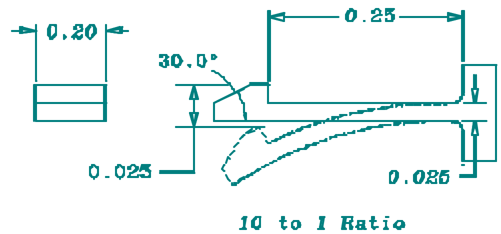


Figure 5

Generally speaking, an unfilled material can withstand a strain level of around 6% and a filled material of around 1.5%. As a reference, a 6% strain level could be a beam with a thickness that is equal to 20% of its length (a 5: 1 L/h_o) and a deflection that is also equal to 20% of its length (see Figure 5). A 1.5% strain level could be a beam with a thickness that is equal to 10% of its length (10: 1 L/h_o) and a deflection that is equal to 10% of its length (see Figure 4). If using a beam that is tapered so the thickness at the base of the snap foot is 50% that of the base of the beam, the length of the beam will approximately 78% (0.7819346 calculated) the length of the 6% and 1.5% beams with uniform thickness.

A more accurate guideline for the allowable dynamic strain curve of the material may be obtained from the material's stress strain curve. The allowable dynamic strain, for most thermoplastics materials with a definite yield point, may be as high as 70% of the yield point strain (see Figure 7). For other materials, that break at low elongations without yielding, a strain limit as high as 50% of the strain at break may be used (see Figure 6). If the snap joint is required to be engaged and disengaged more than once, the beam should be designed to 60% of the above recommended strain levels. However, the best source for allowable dynamic strain is the material supplier.

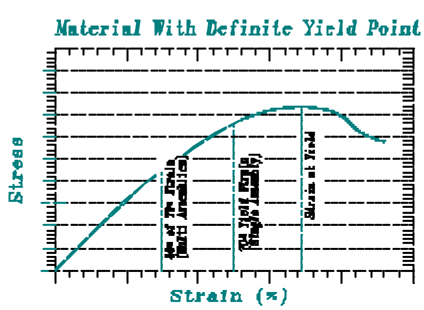


Figure 6

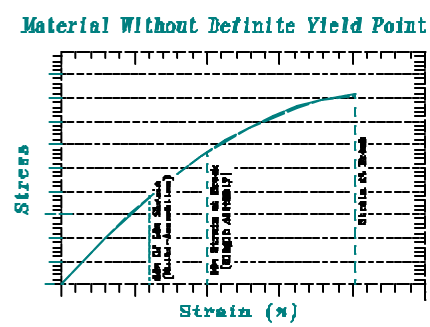


Figure 7

Modulus of Elasticity vs. Secant Modulus

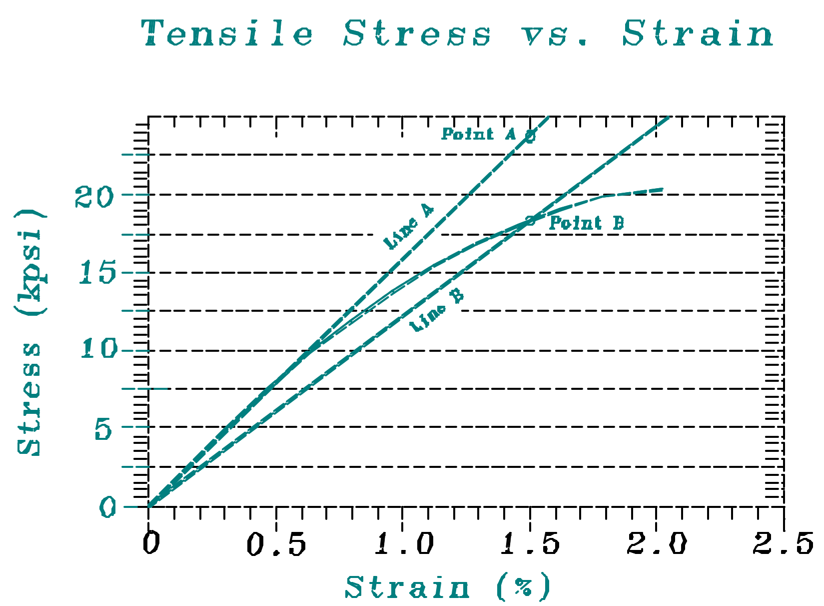


Figure 8

Before going further, we need to examine the actual stresses and forces developed in a snap finger. Figure 8 shows a stress strain curve for a brittle thermoplastic material. The straight line portion of the curve is the region where stress is proportional to strain. Line A is drawn tangent to this region

The slope of Line A is generally reported as the modulus of elasticity (Young's modulus or initial modulus) of the material. Many plastics do not possess this straight-line region. For these materials, Line A is constructed tangent at the origin to obtain the modulus of elasticity. If we designed a snap beam at 1.5% strain for this material in Figure 8 using Equations 1-4 and a modulus of elasticity of 1.6 x 10^6 psi (given by the material) as determined from Line A, the resulting stress would be 24,000 psi, Point A on Line A. However, from the stress strain curve it can be seen that the true stress at 1.5% strain is about 18,000 psi, Point B on the curve. In addition, the deflection force predicted by Equations 1-4 will be high by the same proportions.

Now, to make our math easier, we need some method to force Equations 1-4 to give us the proper stress and force results. If we construct a secant line from the origin to Point B, Line B, the slope of Line B is the material modulus just as the slope of Line A is the modulus of elasticity. The slope of Line B is the secant modulus for the material at Point B and is approximately 18,000 psi divided by 1.5% strain or 1.2 x 10^6 psi. Obviously, the secant modulus can be calculated for any point on the stress-strain curve. Plots of the secant modulus vs. strain (or stress) can then be produced, if desired. Obviously, at the lower strains, the Scant Modulus should approach the modulus of elasticity of the material.

Radii and Stress

In designing the snap beam, it is very important to avoid any sharp corners or structural discontinuities, as stress will concentrate in such areas. To avoid such problems, inside corners should be designed with a minimum radius of 0.020 in. and where necessary to maintain a uniform wall, a radius equal to the inside radius plus the wall thickness should be placed on the outside of the corner. As indicated in Figure 9, an inside radius of 50% of the wall thickness is considered a good design standard.

Therefore, it is recommended that when possible a fillet radius equal to ½ the beam thickness be added to the base of a cantilever beam.

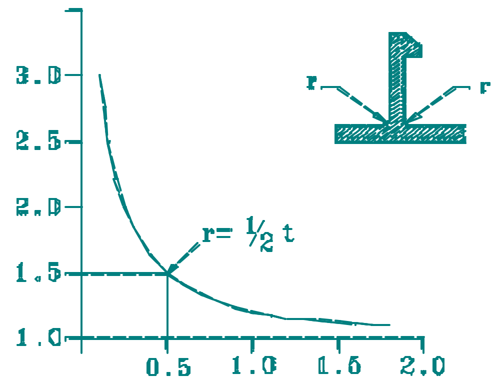


Figure 9

Tapered Beams (same length/stiffer snap)

An improved method of designing cantilever beam snap-fits is to use a tapered beam. The beam is tapered from the root to the base of the snap foot. Stresses in a straight beam concentrate at its base, as shown in Figure 10. Where as stresses in the tapered beam are distributed more uniformly through its length, therefore reducing stress, as shown in Figure 11.

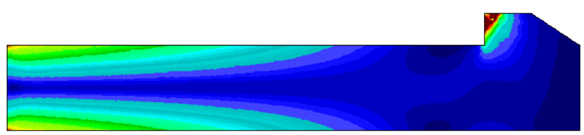


Figure 10

The taper effectively decreases the beam's strain and allows for a deflection greater than that of a straight beam with the same base thickness. Another use for a tapered snap beam is that when the straight beam is not stiff enough, the base of the beam can sometimes be increased to create a stiffer tapered beam.

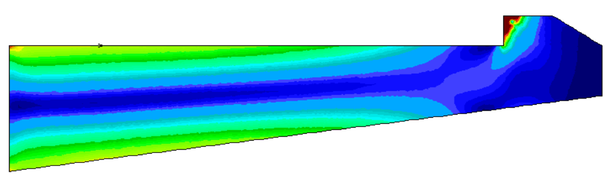


Figure 11

Formulas for Cantilever Beams

Straight Beams

Where
Y = Beam deflection
h_o= Beam thickness at base
L = Beam length
b = Beam width

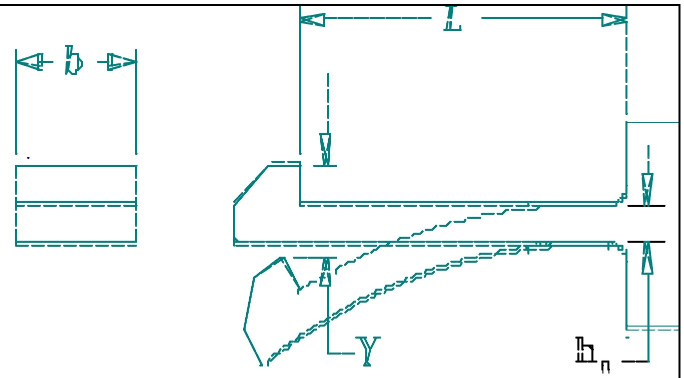


Figure 12

Equations 5-8 can be used to calculate the following properties:

Strain level in a straight beam	$e = \frac{3Yh_o}{2L^2}$	**Equation 5**
Deflection of a straight beam	$Y = \frac{2L^2e}{3h_o}$	**Equation 6**
Length of a straight snap beam	$L = \sqrt{\frac{3Yh_o}{2e}}$	**Equation 7**
Thickness of a straight snap beam	$h_o = \frac{2L^2e}{3Y}$	**Equation 8**

Tapered Beams

To calculate the geometry of a tapered beam, the ratio of the thickness of the beam at the snap foot vs. the base (h_L /h_O) must be known. On Table 1, find the h_L /h_O (Column 1) value and the corresponding K factor (Column 2). K is a geometry factor and is required in all of the formulas related to the tapered beam. Example: A h_L /h_O of 0.50, equates to a thickness at the snap foot of 50% he base. The K factor for an h_L /h_O of 0.50 is 1.636.

Table (K Values)

h_L/h_O	K	h_L/h_O	K	h_L/h_O	K	h_L/h_O	K
0.33	2.137	0.50	1.636	0.67	1.338	0.84	1.138
0.34	2.098	0.51	1.614	0.68	1.324	0.85	1.128
0.35	2.060	0.52	1.593	0.69	1.310	0.86	1.118
0.36	2.024	0.53	1.573	0.70	1.297	0.87	1.109
0.37	1.989	0.54	1.553	0.71	1.284	0.88	1.100
0.38	1.956	0.55	1.534	0.72	1.272	0.89	1.091
0.39	1.924	0.56	1.515	0.73	1.259	0.90	1.082
0.40	1.893	0.57	1.497	0.74	1.247	0.91	1.073
0.41	1.863	0.58	1.479	0.75	1.235	0.92	1.064
0.42	1.834	0.59	1.462	0.76	1.223	0.93	1.056
0.43	1.806	0.60	1.445	0.77	1.212	0.94	1.047
0.44	1.780	0.61	1.429	0.78	1.201	0.95	1.039
0.45	1.754	0.62	1.413	0.79	1.190	0.96	1.031
0.46	1.729	0.63	1.399	0.80	1.179	0.97	1.023
0.47	1.704	0.64	1.382	0.81	1.168	0.98	1.015
0.48	1.681	0.65	1.367	0.82	1.158	0.99	1.008
0.49	1.658	0.66	1.352	0.83	1.148	1.00	1.000

Equations 9-12 can be used to calculate the following properties:

Strain level in a straight beam

$$e = \frac{3Yh_o}{2L^2K}$$ **Equation 9**

Deflection of a straight beam

$$F_d = \frac{bh_o^2}{6} \times \frac{E_S e}{L}$$ **Equation 10**

Length of a straight snap beam

$$L = \sqrt{\frac{3Yh_o}{2Ke}}$$ **Equation 11**

Thickness of a straight snap beam

$$h_o = \frac{2L^2Ke}{3Y}$$ **Equation 12**

Snap Beam Assembly

Where:

Y = Beam Deflection
h_o = Beam thickness at its base
m = Coefficient of friction
e = beam fiber strain
F_a = Assembly force

b = Width of beam
E_s = Secant Modulus
a = Assembly angles
F_d = Deflection force

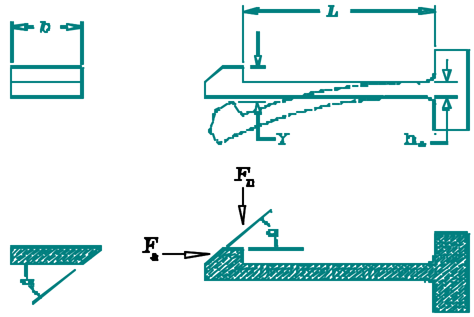


Figure 13

To calculate the **deflection** force of a straight or tapered cantilever beam, use Equation 13.

$$F_d = \frac{bh_o^2}{6} \times \frac{E_S e}{L}$$ **Equation 13**

The assembly angle along with deflection force and coefficient of friction between the mating parts determines the assembly force. The greater the angle and/or coefficient of friction, the higher the assembly force. It may not be possible to assemble parts with assembly angles 45º and a high coefficient of friction. It is recommended that assembly angles between 15º and 30º be used.

To calculate the **assembly force** of a straight or tapered cantilever beam, use Equation 14.

$$F_a = F_d \frac{m + \tan a}{1 - m \tan a}$$ **Equation 14**

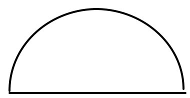
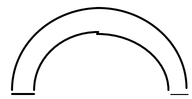
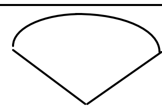
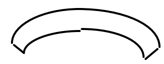
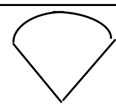
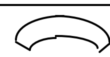
The **retaining force** is determined by the angle of the mating surfaces of the snap foot and ledge. To a point, the greater the angle, the greater the holding strength of the snap. This is true only up to the shear strength of the snap and the effects of bending moments applied to the beam. It is a common belief that a retaining angle of 90º will prevent the snap joint beam from failing. However, the forces on the snap foot can create a bending moment that is high enough to rotate the snap foot back and disengage the snap joint without a shear failure (beam retention is discussed later). *For detachable joints, it is recommended that a retaining angle between 30º and 45º be used.*

To calculate the retaining force, use Equations 13 and 14 and substitute the retaining angle for the assembly angle.

When designing snap-fits that require the ability to be engaged and disengaged repeatedly, a safety factor is needed to predict the beam performance. Therefore, when designing such snap-fits, replace e in Equations 5 through 13 with 0.6e.

Designing Cantilever Snap-Fit Joints From Beams of Circular Sections

The following examples can be used in cases where snap-fits are used on a circular part, such as a boss.

Half circle cross section	 or  *Figure 14*	$Y_{max} = 0.578e\dfrac{L^2}{r}$	**Equation 15**
One third circle cross section:	 or  *Figure 15*	$Y_{max} = 0.580e\dfrac{L^2}{r}$	**Equation 16**
One quarter circle cross section:	 or  *Figure 16*	$Y_{max} = 0.555e\dfrac{L^2}{r}$	**Equation 17**

Snap-fit Retention

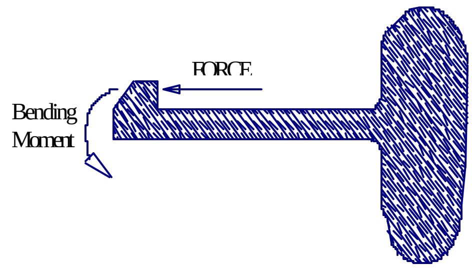


Figure 17

It is often thought that if the retaining angles (the interfacing surface between the snap foot and ledge) are 90º from their base and parallel, the snap foot or ledge must shear to fail. However, the forces that are applied to the snap foot after it is engaged will create a bending moment (Figure 17) on the beam. In some cases, this bending moment will roll the foot back off of the ledge.

To reduce the risk of this occurring, the following are two recommendations.

<u>*The Snap-Fit Loop*</u>

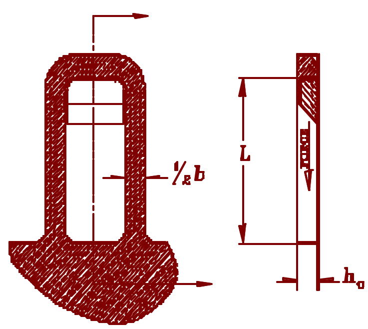


Figure 18

The snap loop in Figure 18 is one alternative. Since the force is directly in-line with the snap loop, the failure modes, after assembly, are shearing of the support ledge or a tensile failure of the snap loop. The roll off failure, as with the standard cantilever beam, is eliminated.

The length, deflection, thickness, and beam taper, (if required) are all calculated in the same manner as the cantilever beam snap-fit. To reduce stress concentrations, the inside corners should have fillet radii. To accommodate the radii on the snap loop, the ledge may have mating radii or clearance for the radii.

Snap Through Support

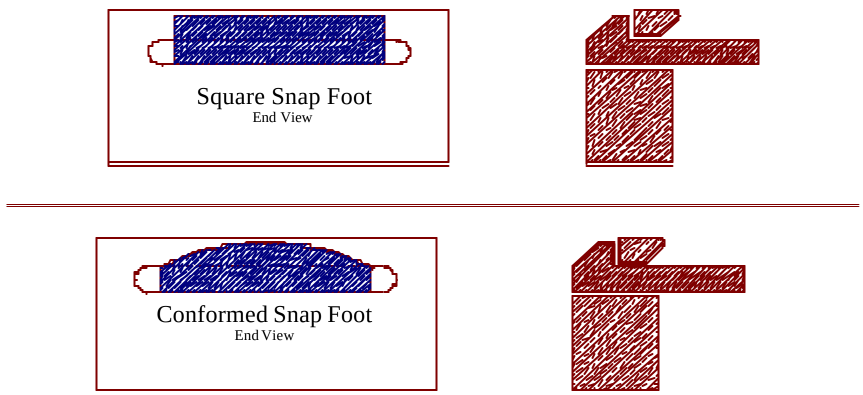


Figure 19

The snap through support is a beam fixed on both ends (Figure 19) creating a hole for a snap foot to pass through. The inside corners of this hole should have full radii in its corners to reduce stress, just as with a cantilever beam. When using a snap through support, the cantilever beam does not have to deflect; the support does all the bending. In areas that are too short for the design of a cantilever beam, this approach is ideal. The deflection of the support beam is dependent on the width of the cantilever beam. The closer the snap foot and snap through hole are in width, the less deflection is allowable. That is, if the cantilever beam and the support beam are equal in width, the support beam has very little deflection before its strain limit is reached.

Alternately, the snap foot of the cantilever beam can be designed to conform to the flexure of the support beam; i. e., use a radius profile instead of a square one (Figure 19). If a profiled snap foot is used, the supported beam will deflect the same as if the forces are a point contact in the center.

The only caution in using this method of beam retention is that it usually has a weld line in the support beam. By increasing the wall section in the support beam, the strength of the weld line can be improved, but the allowable deflection would decrease. After assembly, the possiblefailure modes for a fixed support beam system are the weld line, shear at one end, or a failure of the cantilever beam due to tensile stress which disengages the snap-fit.

Calculating the deflection of the support beam:

To calculate the deflection of the fixed support beam with a square snap foot, use Equation 18.

Where:
b = Cantilver beam width
L = Length of fixed beam
T_{beam} = Thicknes of fixed beam
e = Strain in Fixed Beam
Ymax = Maximum deflection of fixed beam

$$Y_{max} = \frac{1}{12} \times \frac{L^2}{T_{Beam}} \times \frac{\left(1-\frac{b}{L}\right)^2 \times \left(1+3\left(\frac{b}{L}\right)\right)}{1+\left(\frac{b}{L}\right)} \times e$$ **Equation 18**

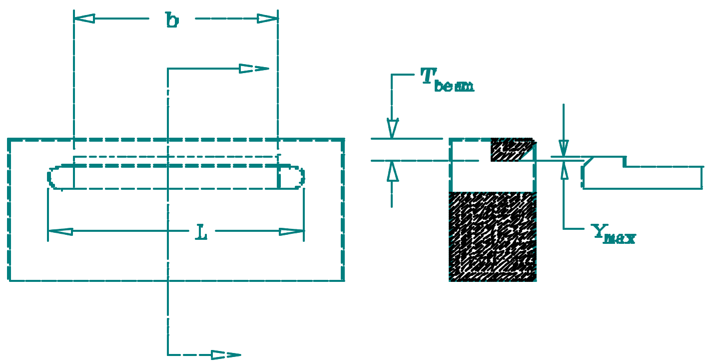


Figure 20

To calculate the deflection of the fixed support beam with a snap foot that conforms to the deflection of the support beam, use Equation 19.

$$Y_{max} = \frac{1}{12} \times \frac{L^2}{T_{Beam}} \times e$$ **Equation 19**

CYLINDRICAL SNAP JOINTS

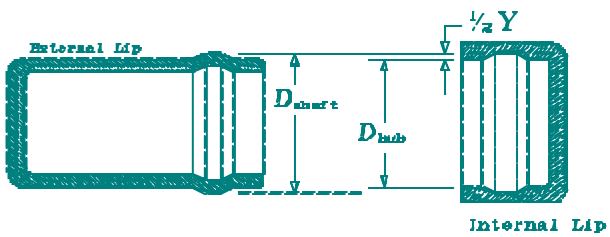


Figure 21

A cylindrical snap joint consists of a cylindrical part with an external lip (snap foot) which engages a cylindrical part with a corresponding internal lip (ledge) as shown in Figure 21. Generally, the shaft is considered rigid and the hub elastic. Variable *Y* in cylindrical snap joints is the total allowable diametral interference. Strain applies to cylindrical snap joints in the same manner as for cantilever beams.

Dynamic Strain in a Cylindrical Snap Design

$e = \dfrac{D_{hub} - D_{shaft}}{D_{shaft}} = \dfrac{Y}{D_{shaft}}$	*Where* D_{hub} = *Inside diameter of the hub* Y = *Total diametral interference* e = *Material strain* D_{shaft} = *Outside diameter of the shaft*

The difference in the largest diameter of the shaft and the smallest diameter of the hub is the deflection *Y*.

Unlike the cantilever beam, the assembly force of the cylindrical snap fit can be only roughly estimated. This is because the length *A* of the hub, Figure 24, deformed during assembly, is difficult to predict. Length *A* depends on both the wall thickness of the hub and the depth of the undercut (½*Y*). As it is difficult to predict a reference, a dimension of twice the width *b* of the hub should be used.

Cylindrical Snap-Fit Assembly

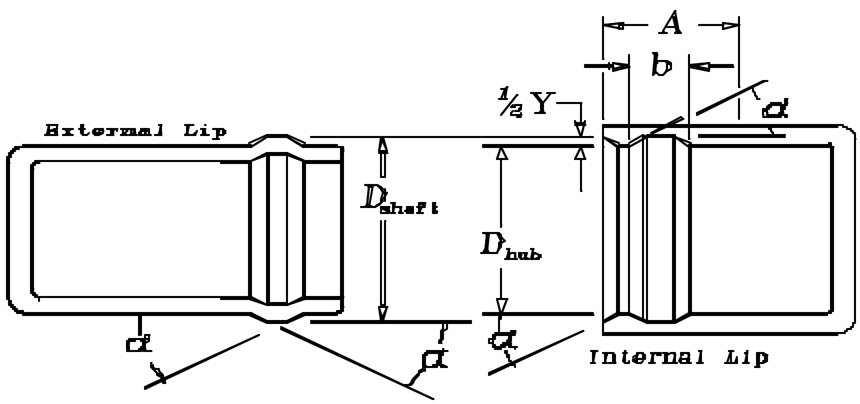


Figure 22

To roughly estimate the assembly forces for a cylindrical snap, we must first calculate the geometry factor K, using Equation 30. It will be assumed that the shaft is rigid and that the total part interference is accommodated by the hub. Equation 30 shows the geometry factor K as a function of the diameter ratio of D_{oh} / D_{shaft}

$$K = \frac{\left(\dfrac{D_{oh}}{D_{shaft}}\right)^2 + 1}{\left(\dfrac{D_{oh}}{D_{shaft}}\right) - 1} + 1$$

Equation 30

The joint pressure (p) can be calculated using Equation 31.

$$p = \frac{0.5Y}{D_{hub}} \times E_s \times \frac{1}{K}$$

Equation 31

Then using factor p, the assembly and pull-out forces can be calculated, using Equation 32.

$$F = p \times p \times D_{oh} \times 2b\left(\frac{m + \tan a}{1 - m \times \tan a}\right)$$

Equation 32

Where
D_{hub} = Inside diameter of the hub
Y = Total diametral interference
e = Material strain
D_{shaft} = Outside diameter of theshaft
❍ = *Coefficient of friction*
♋ = *Assembly an d retaining angle*

APPENDIX

Example of a Cantilever Snap-Fit Calculation

Straight Beam

The material specified has an allowable fiber strain of 1.5% (0.015), and at a 1.5% strain the secant modulus is 1,200,000 psi. The length of the beam has to be 0.50 inches, the width of the beam is 0.200 inches, the required deflection is 0.030 inches, and the assembly angles are both 30. (Figure 23)

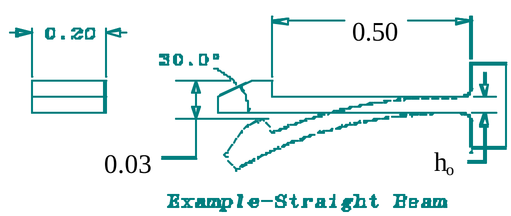


Figure 23

Using Equation 8, the beam's thickness can be calculated:		
$h_o = \dfrac{2L^2e}{3Y}$	$.085 = \dfrac{2(0.50^2)(0.015)}{3(0.030)}$	The thickness of the straight beam is 0.085 inches.
Using Equation 13, the deflection force of the beam can be calculated:		
$F_d = \dfrac{bh_o^2}{6} \times \dfrac{E_S e}{L}$	$8.7 = \dfrac{(0.20)(0.085^2)}{6} \cdot \dfrac{(1200000)(.015)}{0.50}$	The calculated force deflection for the straight cantilever snap beam is 8.7 lbs.
Using Equation 14, the assembly forces can be calculated:		
$F_a = F_d \dfrac{m + \tan a}{1 - m \tan a}$	$7.376 = 8.9 \times \dfrac{(.17 + \tan 30)}{1 - (0.17 \bullet \tan 30)}.$	The calculated assembly force for the straight cantilever beam is 7.4 lbs.

Tapered Beam

To increase the strength of this beam and maintain the same deflection and strain level, the beam can be tapered (Figure 24).

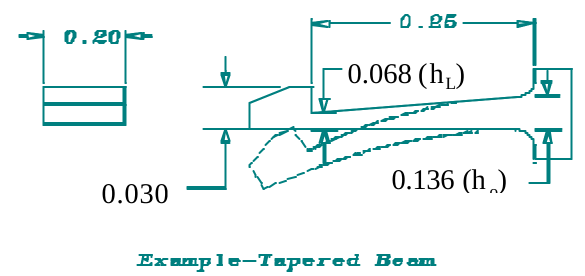


Figure 24

There are now two unknowns:

1. the thickness at the base
2. the thickness at the snap foot.

These two unknowns are related by the *K* factors in Table 1. A decision on the base thickness and thickness ratio (h_L/h_o) is needed. A good starting point is with a base twice as thick as the straight beam with a thickness ratio of 50%. For an h_L/h_o ratio of 0.5, the *K* factor is 1.636.

Using Equation 12, the base thickness can be calculated:		
$h_o = \frac{2L^2Ke}{3Y}$	$22.2 = \frac{(0.2)(136^2)}{6} \times \frac{(1200000)(0.015)}{0.5}$	The calculated deflection force for the tapered beam is 22.20 lbs.
Using Equation 13, the deflection force of the beam can be calculated:		
$F_d = \frac{bh_o^2}{6} \times \frac{E_s e}{L}$	$18.4 = 22.2 \times \frac{0.17 + \tan 30}{1 - (0.17)(\tan 30)}$	The calculated deflection force for the tapered beam is 22.20 lbs.
Using Equation 14, the assembly forces can be calculated:		
$F_a = F_d \frac{m + \tan a}{1 - m \tan a}$	$18.4 = 22.2 \times \frac{0.17 + \tan 30}{1 - (0.17)(\tan 30)}$	The calculated assembly force for the tapered beam is 18.4 lbs.

The strength of the beam's deflection may be increased by decreasing the h_L / h_o value or decreased by increasing the h_L / h_o value. An h_L / h_o equal to 1 would be the same as calculating the straight cantilever beam.

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