Design of Snap-Fits

Cantilever Snap-Fit Joints

The Cantilever snap-fit joint between the Splash Guard (Part 13) and Outer Cover (Part 1) is illustrated below in Fig. 16. Our Cantilever snap-fit beam design has a constant rectangular cross-section. This is in reference to Table 1 on page 112 of the ME452 Coursepack[1] (Appendix E1). With this design, the Splash Guard can be removed easily from the Outer Cover since the snap fitting arms can be released by pressing inward on the end of the two beams. This redesign accomplishes the goals stated in our mission statement in the following ways: Reduces number of parts by four (4), increases efficiency of manufacturing process, and increases ease of use. In addition, this redesign decreases time of assembly for the manufacturer and has a greater sense of aesthetic appeal for the customer.

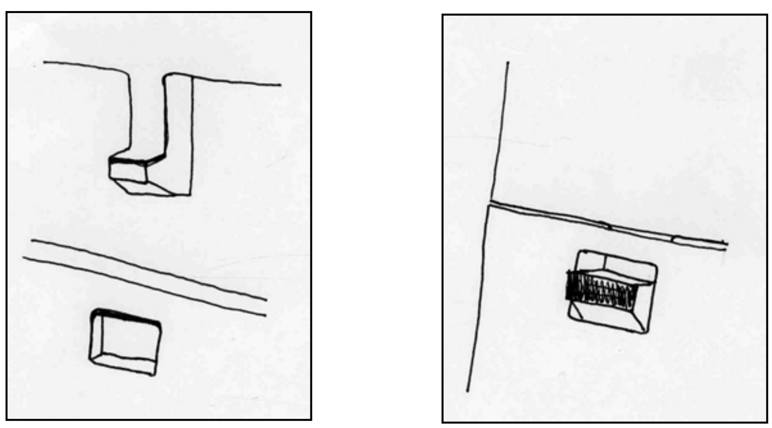

Fig. 16: Cantilever snap-fit Assembly

Equations

Beam deflections and forces can be derived from Castigliano's second theorem that states: $y = \delta U/\delta P$, where y is the deflection, U is the flexural energy of the beam and P is the deflection force. All of the following equations are based on the assumption that the beam base is rigidly fixed and deflection is due to bending stresses.

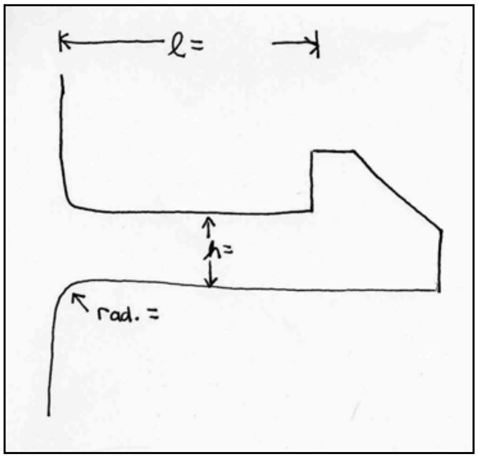

Fig. 17: Dimensions

Permissible deflection, (y)

The amount of beam deflection is dependent on the length *l* and the thickness *h* of the beam and the maximum strain of the material, *ε*. The following equation solves for the maximum (or *permissible*) deflection of a rectangular cantilever beam of constant thickness:

$$y = \frac{2\varepsilon l^2}{3h} = 23.47\ mm$$

where $l = 65$ mm

$h = 3$ mm
$\varepsilon = 0.025$ for ABS/Polycarbonate

Generally permissible deflection of a constant thickness beam should be less than or equal to 0.5 times the beam length. The following calculation shows that our redesign holds true under these conditions:

$y \leq 0.5\ l$
0.5 x 65 mm = 32.5 mm
$y \leq 32.5$ mm

Deflection force, (P)

The deflection force (P) is the force required to bend the cantilever by the amount of the undercut (y). Using the secant modulus of the material (E_s) this value can be calculated from the following equation:

$$P = \frac{bh^2}{6} x \frac{E_s \varepsilon}{l} = 19.04\ N$$

where l = 65 mm
b = 20 mm
h = 3 mm
ε = 0.025 for ABS/Polycarbonate
E_s = 1650 N/mm^2

The secant modulus of the material (E_s) is the ratio of stress to strain at a point on th stress-strain curve corresponding to a particular strain. This value is used instead of the elastic modulus (E_o) because it results in a more accurate estimation of deflection and stress in situations of high levels of short-term application. We used the graph in Appendix E2 to determine this value by using a strain value of 2.5% which was obtained from our known material of the product (ABS).

Engagement force, (W):

The following equation is for determining the engagement force (also known as assembly force) which is the force required to engage the snap:

$$F_t = P\left[\frac{\mu + \tan\alpha}{1 - \mu\tan\alpha}\right] = 36.18\ N$$

where $\alpha = 30^o$
$\mu = 0.6$
P = 19.04 N

The engagement force is dependent on the deflection force, the coefficient of friction, μ, and the lead-in angle of the latch, α. The coefficient of friction was determined by using the table in Appendix E3. The value for $\left[\dfrac{\mu + \tan\alpha}{1 - \mu\tan\alpha}\right]$ was determined by using the α value with the graph in Appendix E3.

Disassembly force, (F_i)

The disassembly force is calculated using the same equation as above, except with the using β, the pull-out angle instead.

$$F_t = P\left[\frac{\mu + \tan\beta}{1 - \mu\tan\beta}\right] = 57.12\ N$$

where $\beta = 40^o$
$\mu = 0.6$
$P = 19.04$ N

Stress concentration

Can be determined by finite element analysis.

Assumptions

- We are assuming that our design does not require complicated or expensive molds for manufacturing. This consideration of ease of molding is pertinent to our mission statement of "*reducing cost of manufacturing*" and this is why we are assuming that our design can be produce with less complex molds with lower tolling costs.
- All equations used are based on the assumption that the beam base is rigidly fixed and deflection is due to bending stresses.
- We are assuming that the material of our Juiceman Jr.®, before redesign (ABS), is suitable for snap-fit design and can withstand high levels of short-term strain as well as can be deflected without damage (i.e. no residual deflection)

Annular Snap-Fit Joints

The Annular Snap-Fit joint between the Base (Part 7) and the Outer Cover (Part 1) is illustrated below in Fig. 18. For our redesign, the shaft (Base) is rigid and the hub (Outer Cover) is flexible and the wall thickness of the annular snap-fit is uniform. The annular snap-fit joint is redesigned so that it is a permanent joint without the aid of screws or additional parts. This redesign accomplishes the goals stated in our mission statement in the following ways: Reduces number of parts by four (4), and increases efficiency of manufacturing process. Due to complicated assembly forces in annular snap-fit designs, the following calculations are based on a model of a beam of infinite length resting on a resilient foundation.

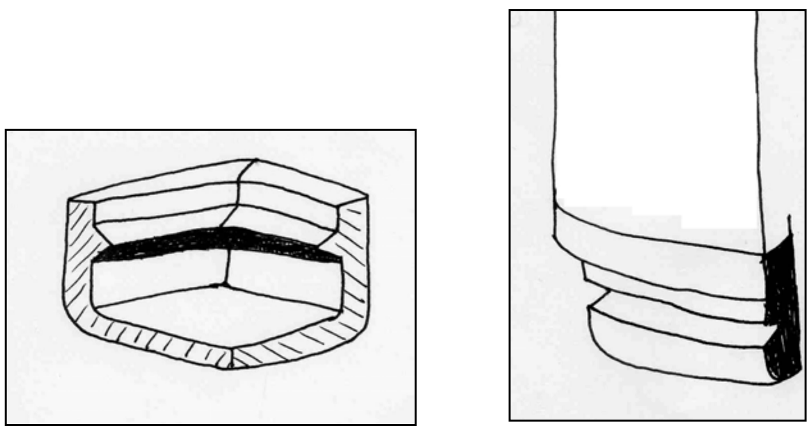

Fig. 18: Annular snap-it Assembly

Equations

As discussed above, beam deflections and forces can be derived from Castigliano’s second theorem that states: $y = \delta U/\delta P$, where y is the deflection, U is the flexural energy of the beam and P is the deflection force.

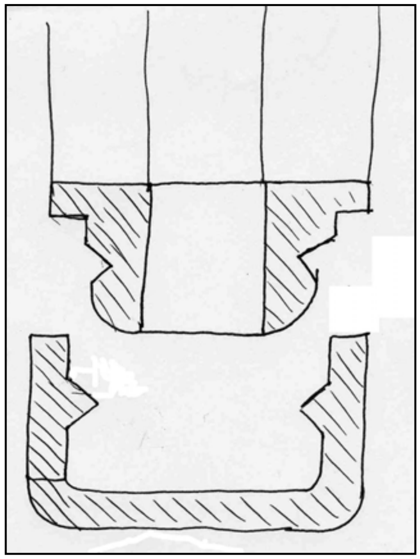

Fig. 19: Dimensions

Permissible deflection, (y)

The equation for maximum deflection in annular snap-fits depends on the maximum strain of the material, ε, and the joint diameter of the hub (Outer Cover), d. The permissible deflection is divided by two to account for the cylindrical shape.

$$y = \varepsilon\, d$$

where $d = 200$ mm
$\varepsilon = .025$

This annular snap-fit joint is redesigned so that it is a permanent joint without the aid of screws or additional parts. This means that our redesign requires split cavity molds for part ejection and 90° return angles.

Deflection Force, (P)

The equation for the deflection force of joints near the end of the hub depends on the undercut, y, the joint diameter, d, the secant modulus of the material, E_s, and the geometric factor, X, which accounts for the rigidity of the components material:

$$P = y\, d\, E_s\, X_s = 782\ N$$

where y = 1 mm (2 parts)

d = 200 mm

E_s = 2300 N/mm^2 (from graph in Appendix E1)

X_s = 1.7 x 10^{-3}

$$\frac{d_0}{d} = \frac{200mm + 2x2.5mm}{200mm} = 1.025$$

Engagement Force, (W)

The engagement force is similar to the above calculation for the cantilever beams:

$$W = P\left[\frac{\mu + \tan\alpha}{1 - \mu\tan\alpha}\right] = 3128\ N$$

where $\alpha = 45^{o}$

$\mu = 0.6$

P = 782 N

Assumptions

- All calculations are based on a model of a beam of infinite length resting on a resilient foundation.
- All calculations are based on the ideal circular shape for a annular snap-fit. The shape of the annular snap-fit in the product slightly deviates from that circular geometry and would be slightly more difficult to eject from the mold and assemble.