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Fundamentals of Annular Snap-Fit Joints

Annular snap fits aren't well understood but can be a handy means of making assemblies easily disassembled or permanently joined together.

Stephen Mraz | Jan 06, 2005

Snap-fit joints are the most widely used way of joining and assembling plastics. They are classified according to their spring element and by separability of the joint: Will it be detachable, difficult to disassemble, or permanent? The most common snap fit is the cantilever. It is based on a flexural beam principle where the retaining force is a function of the material's bending stiffness. The second most widely used is the torsional snap joint. Deflection comes from torsional deformation of the joint's fulcrum and shear stresses carry the load after assembly.

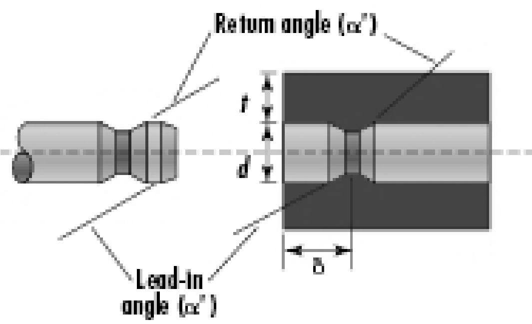
But perhaps the least understood type is the annular snap joint (ASJ). Classic examples of ASJs include ballpoint pens with snap-on caps, and the child-resistant cap on Tylenol bottles. This type of snap fit is best for assembling axis-symmetrical (cylindrical) profiles. But ASJs are often good choices for compact, stiff joints even if the part is not annular.

ASJs are generally stronger, but need greater assembly force than their cantilevered counterparts.

ASJs are basically interference rings. The smaller-diam has a bump or ridge feature around its circumference. The ridge diameter is slightly larger than the inside diameter of the mating tube-shaped female hub. Key to ASJ operation is to make the plug from a more rigid material than its mating female hub. Then the ridge will deflect the hub outward. Deflection imposes a relatively high onceonly or repeated short-term load (hoop stress) distributed along the axis of the hub as the plug slides into it. The ridge feature engages into an undercut grove molded into the inside diameter of the hub, at which point the assembly returns to a stress-free condition.

The maximum permissible deformation of the hub, *P*, is limited by the maximum permissible strain or proportional limit of the material. This limit is typically 50% of the strain at break for most reinforced plastics (safety factor of two). It can be upwards of 60 to 70% of strain at break for more elastic polymers assuming the appropriate safety factor.

The geometry of the ridge determines the assembly force, *F*, needed to engage the snap joint and whether or not the annular snap joint will be detachable or permanent. The lead angle, , is generally < 30°. Its corresponding return angle,



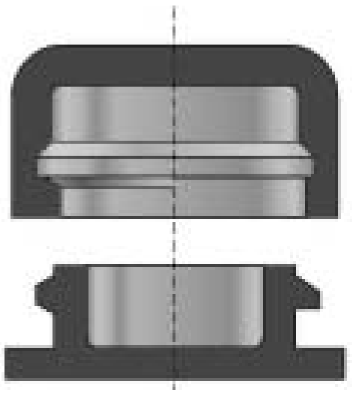
α

, determines assembly separability. A shallow return angle (30°) easily separates, a 90° angle is permanent, and a 45° angle is typical of most applications that disassemble.

Returning to the example of the Tylenol bottle, one can see that the ridge profile has a 90° return angle. This in principal should make the cap impossible to remove from the bottle. The packaging designer employed a clever trick, explains design engineer

Mark Wichmann of **DuPont** to transform a permanent snap joint into one that easily disengages. "The trick," he says, "is notches molded into the ridge on the bottle. This intentional breach in the snap fit, under the right orientation (arrows lined up), lets the cap pop easily off."

Despite the prolific use of snap-fit joints in every conceivable configuration for plastic assemblies, says Wichmann, designers often shy away from annular snap fits for two reasons. The first is ASJs are generally harder to mold and require more complex tooling than cantilevered versions.



The second is material. Stress (σ) is a material's response to distortion. For any given uniaxial distortion (strain, ε) the stress response is given by the appropriate stress-strain curve. Stiffer, reinforced plastics don't recover as easily from deformation and will need engagement features (undercuts and ridges) that are quite small so the plastic doesn't stretch beyond its elongation limit in the hoop direction, says Wichmann.

In an annular snap fit, the hoop-direction distortion (strain) required of a hub when inserting a press-fit plug is a matter of the geometric interference. This required strain may be compared to the allowable strain of a material. Or it may be converted to how the material responds to the strain, its hoop stress, and then compared to the material's allowable stress. Hoop stress is a tensile stress for the female hub and a compressive stress for the male plug. If the joint stresses the material beyond its proportional limit, the plastic may stress-craze or crack causing the joint to fail.

WHEN ANNULAR SNAP FITS MAKE SENSE

It doesn't take an FEA Pro or an engineering Ph.D. to find out whether or not annular snap fits can handle a particular application, says Wichmann. Instead, he suggests making a first order approximation using tried and true handbook calculations and a few basic assumptions.

"Although it's a powerful design tool, a thorough, nonlinear FEA simulation (with contact elements) is time consuming and certainly not trivial for snap fits," says Wichmann. "Using simple handbook equations, it's possible, however, to calculate reasonably close approximations, often within 90 to 95% of an FEA analysis. The resulting ballpark, first-order approximation, is often enough for a working prototype. The prototype in turn gives feedback on fit, function, and resulting assembly/disassembly forces needed.

For example, designers of a recent motor application wanted to employ an annular snap-fit connection between the encapsulated stator and motor housing. The first order of business, says Wichmann, was getting the motor designers to not worry about every conceivable material property that may affect assembly forces and other loads. Often, it's best to first make a few critical assumptions to see if the part geometry and material will handle an annular snap fit before going through lengthy FEA calculations.

The first step was determining the necessary safety factor for the assembly. The engineers determined a safety factor of two was reasonable for the motor. Their thinking was the components would be assembled in a carefully controlled and fixtured assembly process. There would be no repeated disassembly, and no one's life would be at stake if the snap fit failed.

Next, came an evaluation of material and geometry compatibility. Two approaches are possible, says Wichmann. "One is to first establish a particular geometry and then find a material which survives the deflection force (i.e., strain of assembly, *P*). In this case, the encapsulated stator slipped inside the motor mount, deflecting the annular snap joint in the process." The permissible deflection depends on the permissible strain for the material. Amorphous materials such as polycarbonate, polystyrene, PC/ABS, and ABS can be strained up to 70% of the yield strain during a single brief snap fit, he says. "Reinforced materials such as those the motor designers were evaluating are less forgiving. So you would use the rule of thumb — 50% strain at break (safety factor 2) — to make a reasonable material evaluation."

The second approach starts with a material rather than a geometry. "This will give a known permissible strain (i.e., 50% strain at break). The geometry can then be tweaked to meet the permissible strain constraints," explains Wichmann. In the case of the motor, the geometry was the inside diameter of the female hub and the outside diameter of the male (stator) part plus ridge height.

For the motor, the designers opted for the first method and initially defined the annular snap geometry. Next they assumed that all the deflection would come entirely from the motor mount housing. Then the deflection force or strain of assembly of the annular snap fit can be estimated from the geometry of each component. Minimum inside diameter of motor mount housing was 64.5 mm. Maximum outside diameter of stator was 66 mm with a 3-mm ridge or hook.

The percent strain the motor-mount housing will see can be estimated by taking the ratio:

$$\frac{(D_s - D_H)}{D_s} 100\%$$

where *DS* = the outside diameter of the stator (66 mm) and *DH* = the inside diameter of the motor mount housing (64.5 mm). The result, 2.3% strain, is a first-order approximation that is best for membrane stresses when the wall thickness is < 0.1 the nominal diameter. For the motor, wall thickness was 2.5 mm, which is less than 0.1 66 = 6.6 mm so the approximation is reasonable.

The next step in the approximation is to see whether the candidate polymer for the motormount housing, DuPont's Hytrel 8238 (generalpurpose thermoplastic polyester), can survive the expected deflection stress at the predicted 2.3% strain.

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