Snap-fits

Design for HP MJF: Union joints design

Introduction

A snap-fit is an efficient assembly method used to attach plastic parts via a protruding feature on one part (e.g., a hook), which deflects during assembly to be inserted into a groove or a slot in the second part. After the assembly, the protruding feature returns to its initial position.

Snap-fits provide a simple and economical way to assemble plastic parts by drastically reducing assembly time. The way a snap-fit is designed determines whether it can be disassembled and reassembled several times and the force required to do so. This assembly method is suited to thermoplastic materials for their flexibility, high elongation, and ability to be printed into complex shapes.

HP Multi Jet Fusion technology allows for the designing and printing of parts with specific design features integrated, such as snap-fits, in order to connect them.

Types of snap-fits

The various types of snap-fits are listed below.

Cantilever snap-fit

The cantilever snap-fit is the most commonly used type of snap-fit. It consists of a cantilever beam with an overhang at the end. In this type of snap-fit there is a direct relationship between the robustness of the assembly and the strength of the snap-fit.

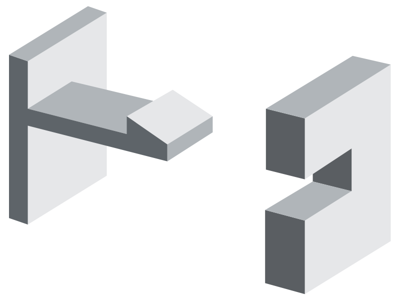


Figure 1. Cantilever snap-fit

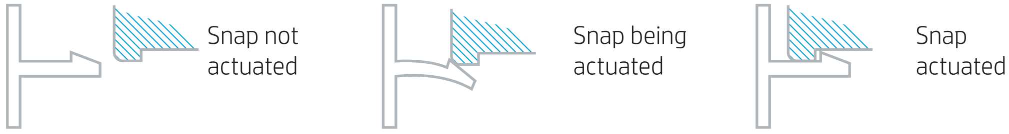


Figure 2. Cantilever snap-fit assembly operation

L-shaped snap-fit

When it is not possible to design a cantilever snap-fit without compromising the robustness of the assembly and the strength of the snap-fit due to material or geometrical constraints, an L-shaped snap-fit can be an alternative. Adding a groove to the base of the snap-fit increases its flexibility while reducing the strain on the beam, compared with a cantilever snap-fit.

Figure 3. L-shaped snap-fit

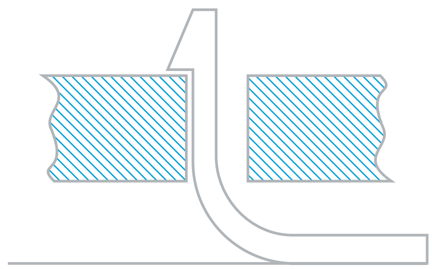


Figure 4. L-shaped snap-fit assembly operation

U-shaped snap-fit

The U-shaped snap-fit is another alternative to the cantilever snap-fit when it is necessary to increase the snap-fit flexibility within a reduced space. This U-shaped alternative is extremely flexible, and thus easier to remove. This type of snap-fit is usually used in cases where the parts need to be pulled apart repeatedly or when two parts don't require a lot of force to stay in position (e.g., in a battery compartment lid).

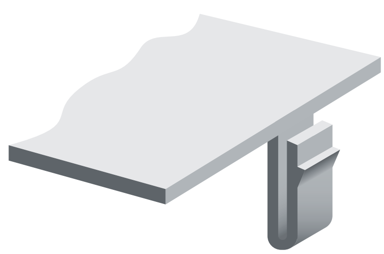


Figure 5. U-shaped snap-fit

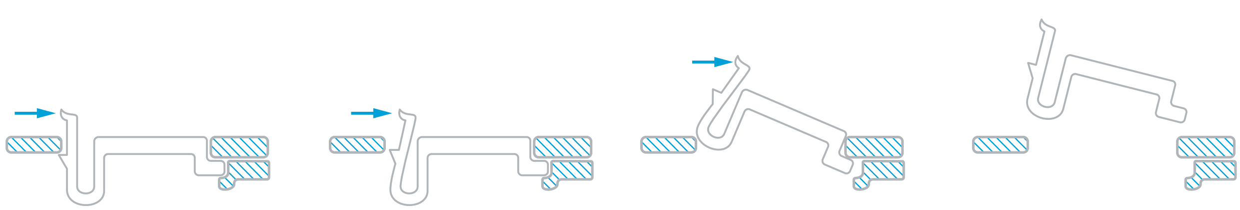


Figure 6. U-shaped snap-fit assembly operation

Annular snap-fit

The annular snap-fit is an assembly method usually used between two cylindrical or ring-shaped parts or between two rotationally symmetric parts, where the deformation required to assemble or disassemble the snap-fit is made in a 360º direction at the same time.

With this assembly method, one part is designed with an undercut and the other is designed with a mating lip. The joint occurs through the interference between both parts during the assembly operation.

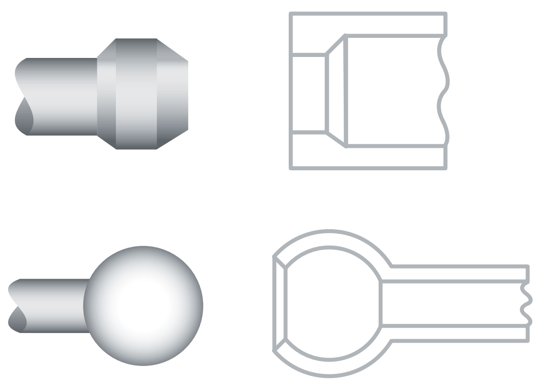


Figure 7. Annular snap-fit

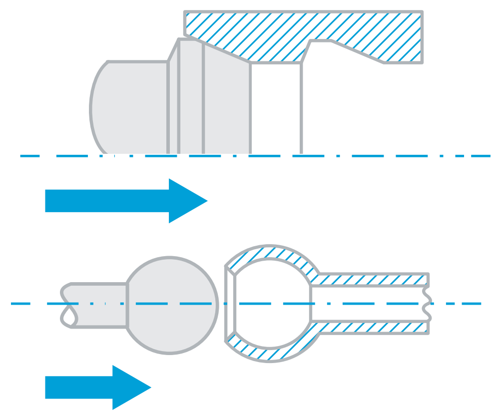


Figure 8. Annular snap-fit assembly operation

Torsional snap-fit

The torsional snap-fit is an assembly method where the flexible point is in a torsional bar instead of the self–snap-fit body. When the torsional bar is pushed down, it turns slightly and opens the joint.

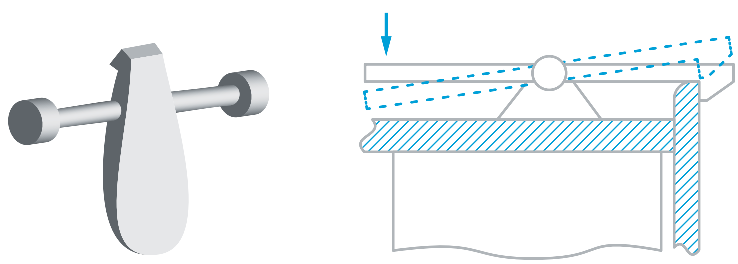


Figure 9. Torsional snap-fit

Design considerations

As mentioned previously, the most commonly used type of snap-fit is the cantilever snap-fit. When designing this type of snap-fit, it is important to design a balanced solution between the robustness of the assembly and the strength of the snap-fit cantilever beam.

This type of snap-fit can be approximated using a simplification of the general beam bending theory, which allows for the inspection of the snap-fit design feasibility. This approach models the cantilever snap-fit by a fixed-free beam with a point-applied end load:

Figure 10. Cantilever beam with a point-applied end load

Mating force and beam stress

The robustness of the assembly will be defined by the force (P) required to assemble and disassemble it. A weak force required to deflect the snap-fit beam will lead to a weak assembly that is unable to maintain the connection between both parts. Otherwise, a strong force will lead to an extremely robust assembly, which will be difficult to assemble and disassemble when required.

Moreover, the design of the snap-fit must be strong enough to resist the stress (σ) suffered by the beam when it deflects due to the mating force (P) applied, without compromising the snap-fit integrity and performance.

For this reason, the mating force (P) and the beam stress (σ) must be the main considerations when designing a cantilever snap-fit, and according to the beam bending theory, they are dependent upon the snap-fit geometry and the material used to make it.

Material and geometry dependence

Because of their direct relationship with the assembly robustness and snap-fit strength, the snap-fit material and geometry are considered the most critical design parameters, and they are often dependent upon the available design space.

For this reason, geometry and material choice are usually the first steps when designing a snap-fit.

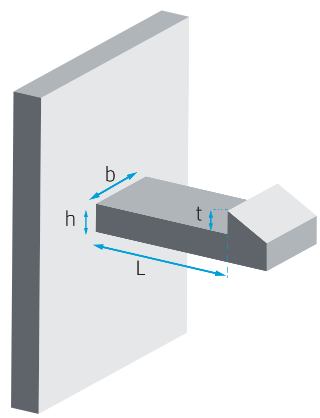


Figure 11. Snap-fit geometry

When choosing the snap-fit material and geometry (h, b, L, t), other dependent factors are clearly defined:

- Choosing the snap-fit cross-section geometry (h, b) allows the designer to calculate its moment of inertia (I), which, for a cantilever beam with a rectangular cross-section, is as follows:

$$I = \frac{b \cdot h^3}{12}$$

- Once the printing material is selected, the modulus of elasticity (E) is made clear since it is often provided in the material datasheet.

The product of the moment of inertia (I) and the modulus of elasticity (E) is known as the beam flexural rigidity (EI).

According to the beam bending theory, these dependent parameters, along with the snap-fit material and geometry, have a direct relationship with the required mating force (P) and the beam stress (σ), as shown below:

- Deflection (y) at the end of a cantilever beam with a point-applied end load:

$$y = \frac{P \cdot L^3}{3 \cdot E \cdot I} \quad (1)$$

- Maximum stress (σ) in a cantilever beam with a uniform rectangular cross-section:

$$\sigma = \frac{P \cdot L \cdot h}{2 \cdot I} \quad (2)$$

The minimum amount of deflection (y) at the end of the cantilever beam required to assemble and disassemble the snap-fit is usually a known parameter dependent upon the geometric constraints and the available design space. In fact, it is defined by the depth (t) of the snap-fit overhang:

- The minimum amount of deflection (y) must be at least equal to the depth (t) of the snap-fit overhang to allow a proper assembly and disassembly operation.

$$y \geq t$$

- A deeper overhang will lead to a strong assembly, but it will mean that the beam must deflect further and, as a consequence, it will require a greater matting force (P)—as shown in equation (1) —and the beam stress (σ) will also increase—as shown in equation (2)-.

Design calculations

The first step in checking the snap-fit design feasibility is to calculate the resultant mating force (P) and to check whether it is suitable. This calculation can be done by solving the equation (1) for P:

$$P = \frac{3 \cdot E \cdot I \cdot y}{L^3} \quad (3)$$

Based on the equation (3), the force (P) is dependent upon how much farther the snap-fit beam must deflect (y), but it also will depend on the material resistance against the bending deformation, which is known as beam bending stiffness (k), and its function of the beam flexural rigidity (EI), the length (L) of the beam, and beam boundary condition:

$$P = k \cdot y \quad (4)$$

The suitable mating force (P) value should not be greater than 50N to 100N, which is considered an ergonomic value for an estimated finger strength average.

Once the mating force (P) has been calculated and it results in a suitable value, the second step to check the snap-fit feasibility is to calculate the stress (σ) in the cantilever beam based on the equation (2).

If the beam stress (σ) is above the yield strength of the material, the snap-fit will deform, and some part of the deformation will be permanent and non-reversible, thus compromising the snap-fit performance and strength up to rupture.

$$\text{Beam stress } (\sigma) < \text{Material yield strength} \quad (5)$$

Considering that the yield strength is not a common property specified in technical datasheets when producing plastic parts, the best option to calculate the snap-fit strength is to use the material allowable strain (ε) and modulus of elasticity (E):

$$\text{Beam stress } (\sigma) < E \cdot \varepsilon \quad (6)$$

In order to obtain the allowable strain (ε) value, designers can refer to usual recommendations for other plastic manufacturing processes such as Injection Molding:

$$\text{Allowable strain } (\varepsilon) < \frac{1}{3} \cdot \text{Material elongation at yield} \quad (7)$$

All design considerations are shown in the following flowchart:

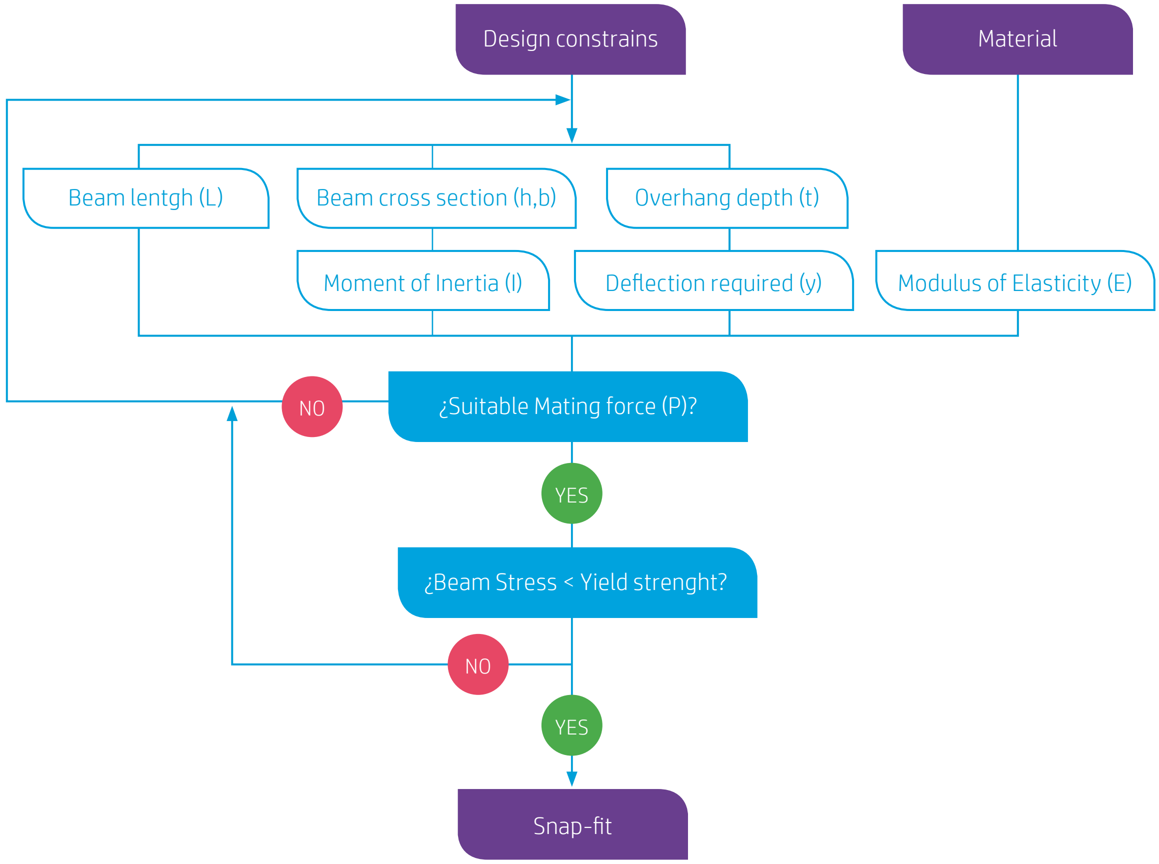


Figure 12. Design snap-fit flowchart

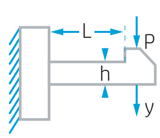
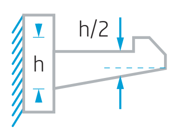
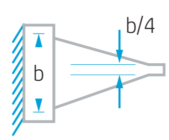
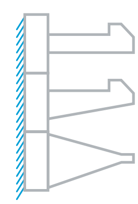
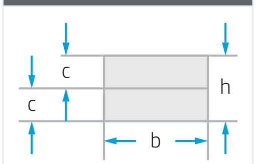
	(Permissible) deflection			Deflection force
Type of design				
Shape of the cross section  Rectangle	$y = 0.67 \cdot \frac{\varepsilon \cdot L^2}{h}$	$y = 1.09 \cdot \frac{\varepsilon \cdot L^2}{h}$	$y = 0.86 \cdot \frac{\varepsilon \cdot L^2}{h}$	$P = \frac{bh^2}{6} \cdot \frac{E\,\varepsilon}{L}$

*Figure 13. Snap-fit calculation equations**

**Note 1: Beam deflection (y) expressed in terms of allowable strain (ε), based on equations (1), (2)*
**Note 2: Mating or deflection force (P) expressed in terms of allowable strain (ε), based on equation (2)*

Design guidelines

There are several design recommendations when designing snap-fits with HP Multi Jet Fusion:

Minimum thickness (h)

The minimum recommended thickness at the base of the cantilever is 1 mm.

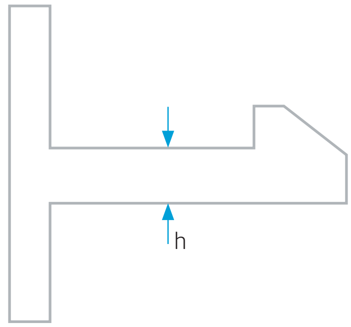


Figure 14. Minimum thickness at the base of the cantilever

Minimum overhang depth (t)

The minimum overhang depth (t) should be at least 1 mm.

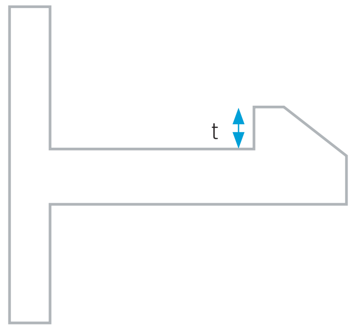


Figure 15. Minimum overhang depth (t)

Recommended common radius

It is recommended to add a common radius at the base of the cantilever to avoid sharp corners and reduce the stress concentration. This common radius should be at least half of the thickness (h) of the base of the cantilever.

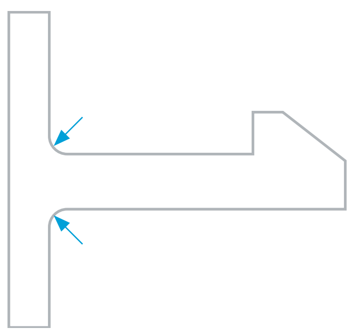


Figure 16. Common radius

Snap-fit overhang

It is recommended to avoid sharp edges at the end of the snap-fit overhang, adding a small chamfer to prevent breaking during the assembly operation.

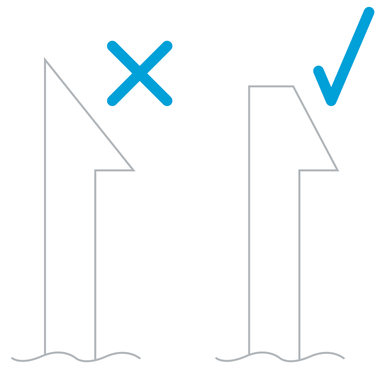


Figure 17. Snap-fit overhang

Assembly angle (α)

As mentioned previously, the snap-fit overhang usually has a gentle chamfer to facilitate the assembly operation. The inclination of this chamfer angle (α) directly affects the mating force (P). If the angle (α) is reduced, the mating force (P) will also reduce. The recommended assembly angle value should be between 35º and 40º.

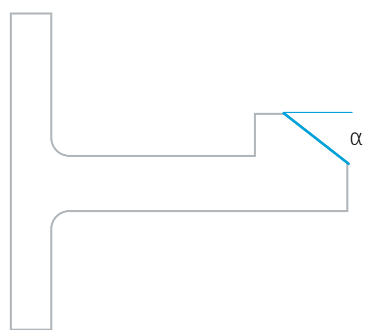


Figure 18. Assembly angle

Disassembly angle (β)

The way the overhang is designed determines whether the snap-fit can be disassembled and reassembled several times. The disassembly angle (β) affects the ease of joint disassembly. For example, a 90º angle (β) can never be disassembled. However, a snap-fit with a disassembly angle (β) equal to the assembly angle (α) will need the same mating force (P) for both operations.

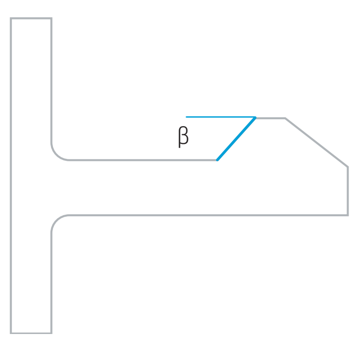


Figure 19. Disassembly angle

Tolerances between parts

When designing a snap-fit, there must be a gap between the protruding feature and the groove to ensure a proper performance, even including the worst tolerance case as shown in the following figure:

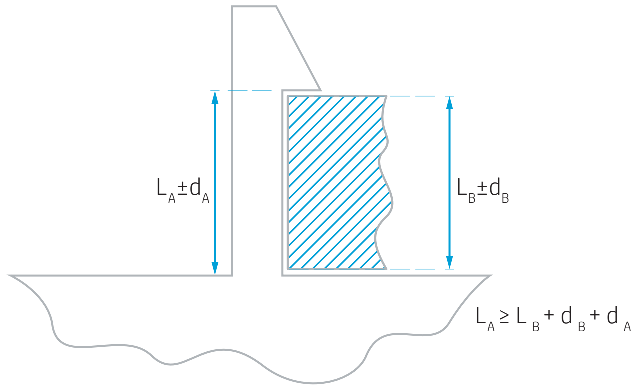


Figure 20. Tolerances between parts

Modifying the mating force (P)

Sometimes, after choosing the snap-fit material and geometry, the resulting mating force (P) is a non-desirable value. Based on the equation (3) and bearing in mind that when designing a snap-fit, the most common restrictive tolerances are the length (L) of the beam and the depth (t) of the overhang, the most common solution when modifying the mating force (P) is needed is to change the cantilever cross-section (h, b).

Reducing the mating force (P) will also reduce the beam stress (σ).

Tapered beam

One of the most recommended changes in the snap-fit cross-section is to design a tapered beam. While a snap-fit beam with a uniform cross-section has an uneven distribution of strain and concentrates the stress at its base, a tapered beam uses less material and results in a more even distribution of strain throughout the cantilever, thus reducing stress (σ) concentration and the assembly and disassembly force (P).

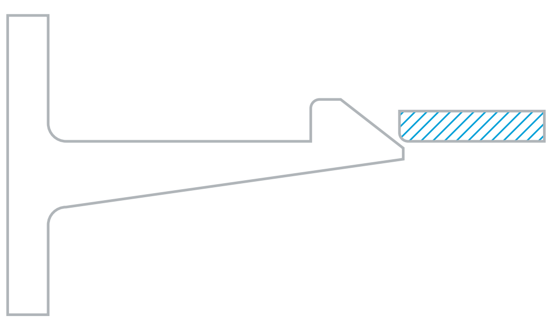


Figure 21. Tapered beam

Printing orientation

There are some recommended orientations when printing a snap-fit regarding its accuracy and proper performance.

For tight snap-fits

When printing tight snap-fits where the length of the beam (L) is critical, the XY plane orientation is recommended to achieve the best accuracy and, thus, a better performance.

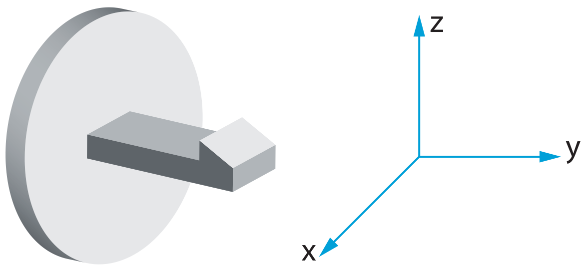


Figure 22. XY plane orientation

When the width of the snap-fit (b) is critical, the XZ or YZ plane orientation is recommended to achieve the best accuracy and to avoid excessive clearances on the XY plane, which can lead to noise and vibrations.

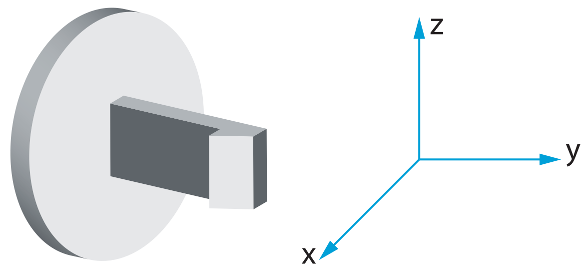


Figure 23. XZ or YZ plane orientation

To reduce printing issues

Printing the snap-fit inclined slightly in the X, Y, and Z axes can reduce the likelihood of typical printing issues.

Post-processing recommendations

HP MJF technology allows for different post-processing methods that can affect the finishing of the printed part. Although most of the post-processing methods should not affect a 3D printed snap-fit, there can be some automatic post-processes that affect it, such as the tumbler post-process.

The tumbler post-process involves hitting the 3D printed part with small abrasive pellets in order to reduce its roughness. In return, some dimensions and/or small features can be affected by the process.

In the case of the snap-fits, a tumbler process can reduce the mating force (P) of the assembly and even break it depending on the snap-fit geometry.

For this reason, if automatic post-processes are required, it is recommended to protect the part with a sinter box to prevent damage.

Calculation example

The following figure illustrates the calculation needed when designing a cantilever snap-fit.

In this particular case, a clipping system for an optical sensor must be designed as follows:

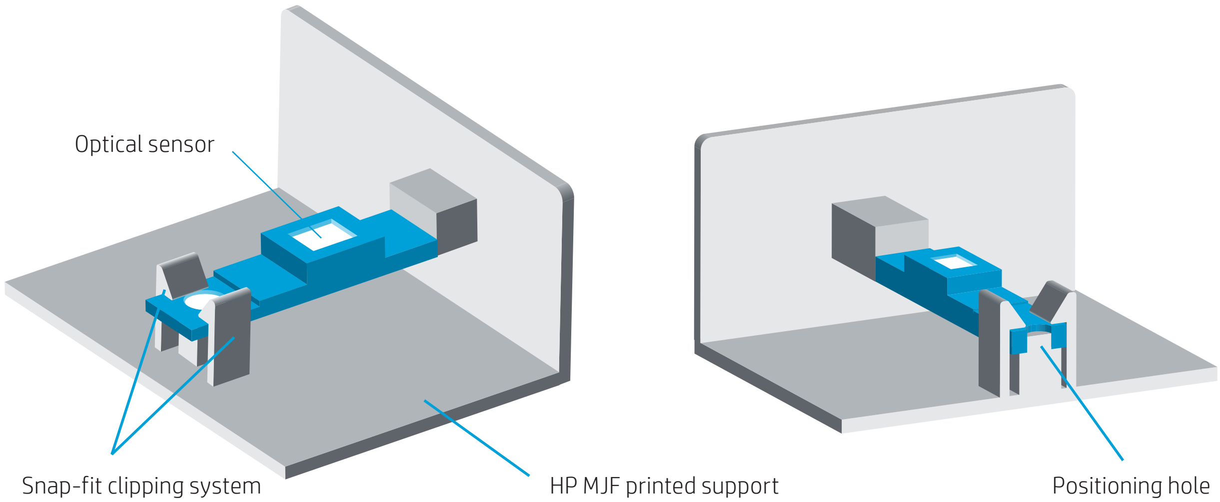


Figure 24. Optical sensor clipping system

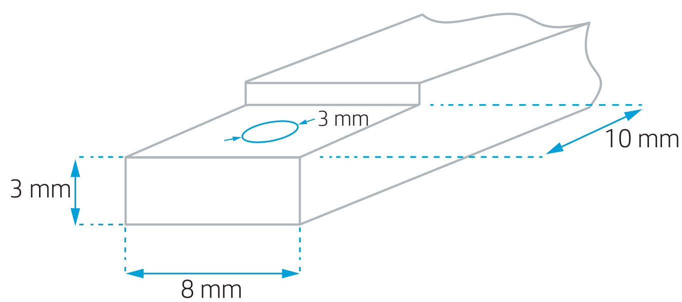


Figure 25. Optical sensor dimensions

The design requirements are listed below:

- The material used to print the part is HP 3D HR PA 12, with an elastic modulus of elasticity (E) of 1800 MPa.
- Due to optical requirements, the sensor must lay 5 mm above the base. Thus, the snap-fit total length must consider the worst-case tolerances and the optical requirements:

$$L = 3\ mm + 0.1\ mm + 5\ mm + 0.2\ mm + 0.2\ mm = 8.5\ mm$$

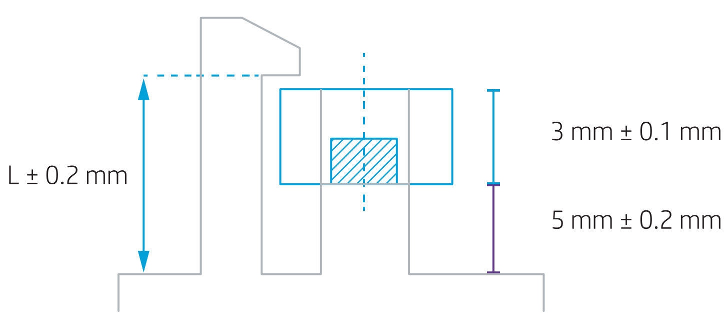


Figure 26. Snap-fit length calculation

- Due to constructive constraints, the snap-fit cannot overlap the positioning hole, which means that the overhang depth (t) must be between 1 mm—the minimum recommended value—and 2.5 mm—the maximum allowable distance to avoid contact between the snap-fit overhang and the sensor positioning hole:

$$\text{Overhang depth } (t) = y = 1 \text{ mm}$$

- The width must be smaller than 10 mm due to geometrical constraints:

$$b = 9.5 \text{ mm}$$

Once the snap-fit material and geometry (h, b, L, t) are clearly defined, the resulting mating force (P) must be calculated to check whether it is suitable. This calculation can be done using the equation (3):

$$P = \frac{3 \cdot E \cdot I \cdot y}{L^3} = \frac{3 \cdot E \cdot (b \cdot h^3) \cdot y}{12 \cdot L^3} = \frac{3 \cdot 1800\text{MPa} \cdot 9.5 \text{ mm} \cdot (1.5 \text{ mm})^3 \cdot 1 \text{ mm}}{12 \cdot (8.5 \text{ mm})^3} = 23.49\text{N}$$

The calculated mating force (P) value is inside the ergonomic range. Therefore, based on the equation (2) and (6), the next step is to check the strength of the snap-fit calculating the allowable strain (ε):

$$\sigma = \frac{P \cdot L \cdot h}{2 \cdot I} = E \cdot \varepsilon$$

$$\varepsilon = \frac{P \cdot L \cdot h}{2 \cdot I \cdot E} = \frac{12 \cdot P \cdot L \cdot h}{2 \cdot (b \cdot h^3) \cdot E} = \frac{12 \cdot P \cdot L}{2 \cdot (b \cdot h^2) \cdot E} = \frac{12 \cdot 23.49\text{N} \cdot 8.5 \text{ mm}}{2 \cdot (9.5 \text{ mm} \cdot (1.5 \text{ mm})^2\,) \cdot 1800 \text{ MPa}} = 0.03 = 3\%$$

The calculated allowable strain (ε) shows that the snap-fit does not deform when it deflects due to the mating force (P) applied, without compromising its integrity and performance.

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