9 – Assembly Techniques – Category I Screws, Press-Fit, Snap-fit

Introduction

Plastic parts can be joined using a variety of assembly techniques; some of them allowing disassembly (category I), others creating a permanent joint (welding, category II).

– *Mechanical Fasteners*
The self-tapping screw cuts or forms a thread as it is inserted, eliminating the need for moulding an internal thread or a separate tapping operation.

– *Plastic Threads*
When required, external- and internal threads can be automatically moulded onto the part, eliminating the need for mechanical thread-forming operations.

– *Press-Fittings*
This technique provides joints with high strength at low cost. In general, suggested interferences are larger between thermoplastic parts than metal parts because of the lower elastic modulus. The increased interference can produce production savings due to greater latitude in production tolerances. The effects of thermal cycling and stress relaxation on the strength of the joint must be carefully evaluated.

– *Snap-Fits*
Snap fitting provides a simple, inexpensive and rapid means of assembling plastic parts. Basically, a moulded undercut on one part engages a mating lip on the other. This method of assembly is uniquely suited to thermoplastic materials due to flexibility, high elongation and ability to be moulded into complex shapes.

– *Spin Welding (see Chapter 10)*
Spin welding produces welds that are strong, permanent and stress free. In spin welding, the part surfaces to be welded are pressed together as they are rotated relative to each other at high speed. Frictional heat is generated at the joint between the surfaces. After a film of melted thermoplastic has been formed, rotation is stopped and the weld is allowed to seal under pressure.

– *Ultrasonic Welding (see Chapter 10)*
Similar plastic parts can be fused together through the generation of frictional heat in ultrasonic welding. This rapid sealing technique, usually less than two seconds, can be fully automated for high speed and high production. Close attention to details such as part and joint design, welding variables, fixturing and moisture content is required.

– *Vibration Welding (see Chapter 10)*
Vibration welding is based on the principle of friction welding. In vibration welding, the heat necessary to melt the plastic is generated by pressing one part against the other and vibrating it through a small relative displacement at the joint. Heat generated by the friction melts the plastic at the interface. Vibration is stopped and the part is automatically aligned; pressure is maintained until the plastic solidifies to bond the parts together. The bond obtained approaches the strength of the parent material.

– *Hot Plate Welding (see Chapter 10)*
Hot plate welding is a technique used for joining thermoplastic parts. Non symmetric parts with fragile internal components are suitable for this technique.

– *Laser Welding (see Chapter 10)*
Two plastic parts, of which one must be made out of a transparent material, are welded together using laser light for melting both materials at the joint.

– *Cold or Hot Heading/Riveting (see Chapter 10)*
This useful, low-cost assembly technique forms strong, permanent mechanical joints. Heading is accomplished by compression loading the end of a rivet while holding and containing the body.

– *Adhesion Bonding (see Chapter 10)*
This technique is used to join plastics or plastics and dissimilar materials. It is useful when joining large or complicated shapes. Details on methods and techniques will be found in the individual product sections.

Design for Disassembly

In order to reduce the impact on the environment as much as possible, the design and the material should be selected to allow the most efficient use of the part over its service life. This may included re-use of the part or some of its components. For this reason, it is really important to "Design for Disassembly". In chapter 10 information and recommendations related to this subject are given, which should help designers in creating more optimal solutions.

Mechanical Fasteners

Self-Tapping Screws

Self-Tapping screws provide an economical means for joining plastics. Dissimilar materials can be joined together and the joint can be disassembled and reassembled.

The major types of self-tapping screws are thread forming and thread cutting. As the name implies, thread forming screws deform the material into which they are driven, forming threads in the plastic part. Thread cutting screws on the other hand, physically remove material, like a machine tap, to form the thread. To determine what kind of self-tapping screw is best for a job, the designer must know which plastic will be used, and its modulus of elasticity.

If the modulus is below 1500 MPa, thread forming screws are suitable, as the material can be deformed without entailing high hoop stress.

When the flexural modulus of a plastic is between 1500 and 3000 MPa, the proper type of self-tapping screw becomes somewhat indeterminate. Generally speaking, the stress generated by a thread forming screw will be too great for this group of resins, and thread cutting screws should be employed. However, plastics such as Zytel® nylon resin and Delrin® acetal resin work well with thread forming screws.

Thread cutting screws are still preferred unless repeated disassembly is necessary.

Thread forming screws *AB* and *B*, shown in Fig. 9.01 are fast driving, spaced-thread screws. The *BP* screw is much the same as the *B* screw except that it has a 45° included angle and unthreaded cone points. The cone point is useful in aligning mating holes during assembly. The *U* type, blunt point, is a multiple-thread drive screw intended for permanent fastening. The *U* type screw is not recommended where removal of the screw is anticipated. Special thread forming screws, like the Trilobular, which are designed to reduce radial pressure, frequently can be used for this range of modulus of elasticity, see Fig. 9.02. Screws with non-circular cross sections use to have slightly increased driving- and stripping torques.

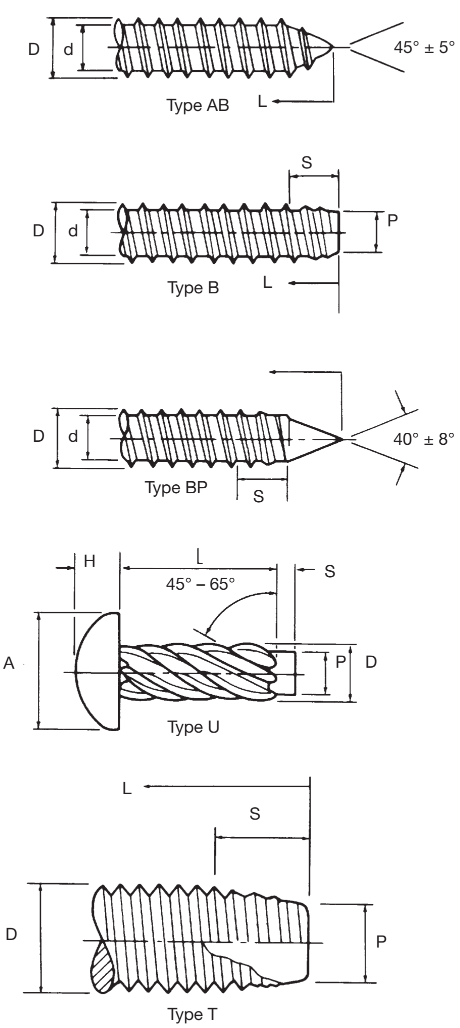
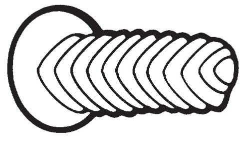
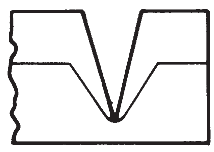


Fig. 9.01 **Types of self-tapping screws**

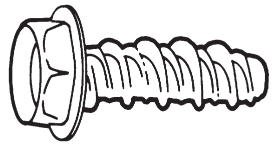
Trilobe
Triangular configuration designed by Continental Screw Co. (and licensed to other companies) is another technique for capturing large amounts of plastic. After insertion, the plastic cold-flows or relaxes back into the area between lobes. The Trilobe design also creates a vent along the length of the fastener during insertion, eliminating the "ram" effect in some ductile plastics, pressure builds up in the hole under the fastener as it is inserted, shattering or cracking the material.



Sharp Thread
Some specials have thread angles smaller than the 60° common on most standard screws. Included angles of 30° or 45° make sharper threads that can be forced into ductile plastics more readily, creating deeper mating threads and reducing stress. With smaller thread angles, boss size can sometimes be reduced.



HI-LO
Dual-height thread design from ELCO Industries boosts holding power by increasing the amount of plastic captured between threads.



EJOT Delta Special design

Fig. 9.02 **Special types of self-tapping screws**

Another unique thread form, the Hi-Lo fastener, has a double lead thread where one thread is high and the other low. A sharp 30° included thread angle allows for a deeper cut into the material and reduces the hoop stress that would be generated by a conventional 60° thread angle form. Another design feature is that the Hi-Lo screw has a smaller minor diameter than a conventional screw.

This increases the material in contact with the high flat thread, increasing the axial shear area. All of this contributes to a greater resistance to pull out and a stronger fastener. This style of screw can be either thread forming or thread cutting with the thread cutting variety used on even higher modulus materials.

The third group of resins with elastic moduli in the 3000 and 7000 MPa range gain their strength from reinforcing glass fibres. Typical of resins in this category are the 13% glass-reinforced Zytel® nylon resin materials and Minlon® mineral-reinforced materials. These resins are best fastened with thread cutting screws. In these more rigid materials, thread cutting screws will provide high thread engagement, high clamp loads, and will not induce high residual stress that could cause product failure after insertion.

The last group of plastics, those with flexural moduli above 7000 MPa are relatively brittle and at times tend to granulate between the threads causing fastener pull out at lower than predicted force values. Resins in this higher modulus category are the 33% and 43% glass-reinforced Zytel® nylon resins, Rynite® PET-reinforced polyester terephthalate resin, and Crastin® PBT-reinforced polybutyl terephthalate resin and DuPont high performance polyamide resin Zytel® HTN.

For these materials, the finer threads of the type T screw are recommended. Even with the fine pitch screws, backing out the screw will cause most of the threads in the plastic to shear, making reuse of the same size screw impossible. If fastener removal and replacement is required in this group of materials, it is recommended that metal inserts be used, or that the boss diameter be made sufficiently larger to accommodate the next larger diameter screw (see also Fig. 3.25).

The larger screws can be used for repairs and provide greater clamp loads than the original installation. If metal inserts are chosen, there are five types available: ultrasonic, heated inserts, moulded-in, expansion, or solid bushings (Fig. 9.03).

The inserts are held in place by knurls, grooves and slots, and are designed to resist both axial and angular movement.

– *Ultrasonic Insert*

  This insert is pressed into the plastic melted by high-frequency ultrasonic vibrations and is secured by melt solidification. This is a preferred choice where applicable because of low residual stress.

– *Heated Insert*

  The insert is heated 30 to 50°C above processing melt temperature and pressed into the slightly too small hole.

– *Moulded-In Insert*

  The Insert is placed in the mould, and has an external configuration designed to reduce stress after cooling.

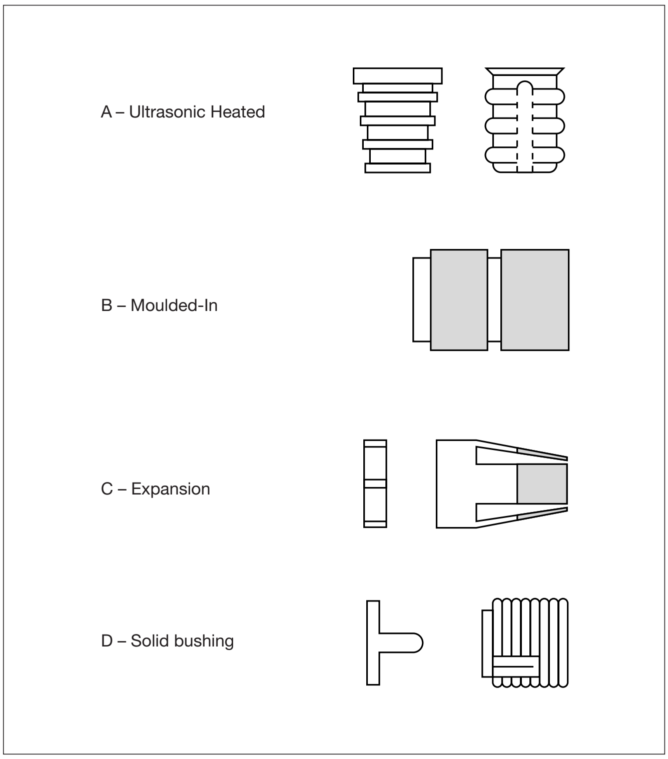


Fig. 9.03 **Types of inserts**

– *Expansion Insert*

  The expansion insert is slipped into the hole and does not lock in place until the screw is inserted to expand the insert wall.

– *Solid Bushings*

  The bushings are generally a two-piece insert. The body is screwed into a prepared hole and a ring locks the insert in place.

Recommended Design Practice

When designing for self-tapping screws in plastics, a number of factors are important: (see also Fig. 3.09 for design).

– *Boss Hole Dimension*

  For the highest ratio of stripping to driving torque, use a hole diameter equal to the pitch diameter of the screw, ($d_h \cong 0{,}8\ D_s$, see Tables 9.01-9.02).

– *Boss Outside Dimension*

  The most practical boss diameter is 2,5 times the external screw diameter. Too thin a boss may crack, and no acceptable increase in stripping torque is achieved with thicker bosses.

– *Effect of Screw Length*

  Stripping torque increases with increasing length of engagement and levels off when the engaged length is about 2,5 times the pitch diameter of the screw.

A practical tool for evaluating the manufacturing feasibility of a fastener joint is the strip-to-drive ratio, which is the ratio of stripping torque to driving torque.
For high volume production with power tools, this ratio should be about 5 to 1. With well trained operators working with consistent parts and hand tools, a 2 to 1 ratio may be acceptable. In any case, lubricants must be avoided because they drastically reduce this ratio.

Stripping Torque

Stripping torque may be calculated from:

$$T = F\,r\left(f_1 + f_2 + \frac{p}{2\pi r}\right)$$

where:

T = Torque to develop pull-out force

r = Pitch radius of screw = $D_p/2$

p = thread pitch

F = Pull-out force

f_1 = Coefficient of friction screw-plastic, Table 7.01

f_2 = Coefficient of friction screwhead – material underneath

* Hub fracture under the screw

Pull-Out Force

The ultimate test of a self-tapping screw is the pull-out force. It can be calculated by equation:

$$F = \tau \pi D_p L / S$$

where:

F = Pull-out force

τ = Shear stress = $\left(\frac{\sigma_t}{\sqrt{3}}\right)$

σ_t = Tensile yield stress or design stress

D_p = Pitch diameter

L = Axial length of full thread engagement

S = safety factor = $1{,}2\ c_1\ c_2$

c_1 = 1,0 for special screws

c_1 = 1,5 for ordinary screws

$c_2 = 10/\varepsilon_{br}$ ($\geqslant$ 1.0)

ε_{br} = elongation at break, (%)

The above information can be verified by running prototype test on boss plaques or flat plaques moulded in the plastic selected.

Tables 9.01 and 9.02 give numerical values of the pull-out strengths, stripping torque and dimensions for Type AB screws of various sizes. The nomenclature for self-threading screws is described. Engaged length *L*, is 2,5 times the screw diameter. Applications of self-tapping screws are shown in Fig. 9.41–9.43.

Pull-Out Force Metal Inserts

For the calculation of the pull-out force of metal inserts the formula for self-tapping screws can be used, but with an effective length of 0,3–0,5 L, see also Figures 9.03 a/b.

Plastic Threads

Introduction

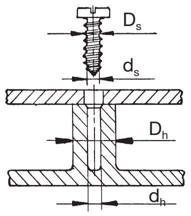
This conventional method of holding parts together can be applied to DELRIN® and ZYTEL® or other thermoplastic materials.

It can be used for assembling parts made out of different materials and the thread can be moulded into the parts.

Basic Principles

To design a screwed joint all sharp interior corners must be eliminated. The beginning as well as the end of the thread should be rounded off in order to avoid notch effects. See Fig. 9.04 A.

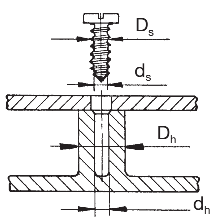
Table 9.01 **Pull-out load performances for various screw dimensions and materials**



		Screw No.	6	7	8	10	12	14
		D_s mm	3,6	4	4,3	4,9	5,6	6,5
		d_s mm	2,6	2,9	3,1	3,4	4,1	4,7
		D_h mm	8,9	10	10,8	12,2	14	16,2
		d_h mm	2,9	3,3	3,5	4,1	4,7	5,5
Pull-out Force	DELRIN® 500 NC010	N	3100	3800	4500	5250	6500	9 000
	DELRIN® 570	N	3050	3600	4250	4950	6000	8 300
	ZYTEL® 101L NC010	N	2250	3250	3850	4300	5100	6 400
	ZYTEL® 79G13L	N	2200	3100	3400	3700	4400	5 900
	ZYTEL® 70G30HSL	N	2300	3200	3500	3900	4850	6 200
	MINLON® 10B140	N	3200	3330	5370	5690	8710	10 220
	MINLON® 11C140	N	2880	3200	3540	4510	5070	6 480
	RYNITE® 530	N	3300	4100	4400	4900	*	*
	RYNITE® 545	N	4300	4470	4500	5660	6020	*
	RYNITE® 555	N	2480	2940	2740	3780	4120	*

* Hub fracture under the screw

Table 9.02 **Stripping torque performances for various screws dimensions and materials**



		Screw No.	6	7	8	10	12	14
		D_s mm	3,6	4	4,3	4,9	5,6	6,5
		d_s mm	2,6	2,9	3,1	3,4	4,1	4,7
		D_h mm	8,9	10	10,8	12,2	14	16,2
		d_h mm	2,9	3,3	3,5	4,1	4,7	5,5
Stripping torque	DELRIN® 500 NC010	N.m	2,5	3,5	4,6	5,8	7,5	11,2
	DELRIN® 570	N.m	2,5	3,5	4,7	6,2	8,2	12,0
	ZYTEL® 101L NC010	N.m	1,6	2,5	3,6	5,0	7,0	10,0
	ZYTEL® 70G13L	N.m	2,0	3,0	4,0	5,3	6,9	8,5
	ZYTEL® 70G30HSL	N.m	2,5	3,5	4,8	6,3	8,0	10,0
	MINLON® 10B140	N.m	2,4	3,5	4,8	6,4	10,2	13,8
	MINLON® 11C140	N.m	2,5	3,0	4,3	6,0	7,3	11,3
	RYNITE® 530	N.m	3,3	4,3	4,6	7,2	-	-
	RYNITE® 545	N.m	4,7	5,1	5,3	8,6	10,4	11,8
	RYNITE® 555	N.m	4,3	4,7	4,2	6,0 [a)]	6,6 [a)]	9,8 [a)]

[a] For these sizes, ‘‘d_h’’ was increased by 10% to avoid hub fracture at the weld line.

If both parts are made in plastic, the shape of the thread should be changed to one of the two types shown in Fig. 9.04 B-9.04 C.

Engineering plastics usually have better resistance to compressive stresses than to tensile stresses and therefore the threads should be made on the outside of the plastic part when it is to be screwed to a metallic tube, Fig. 9.05.

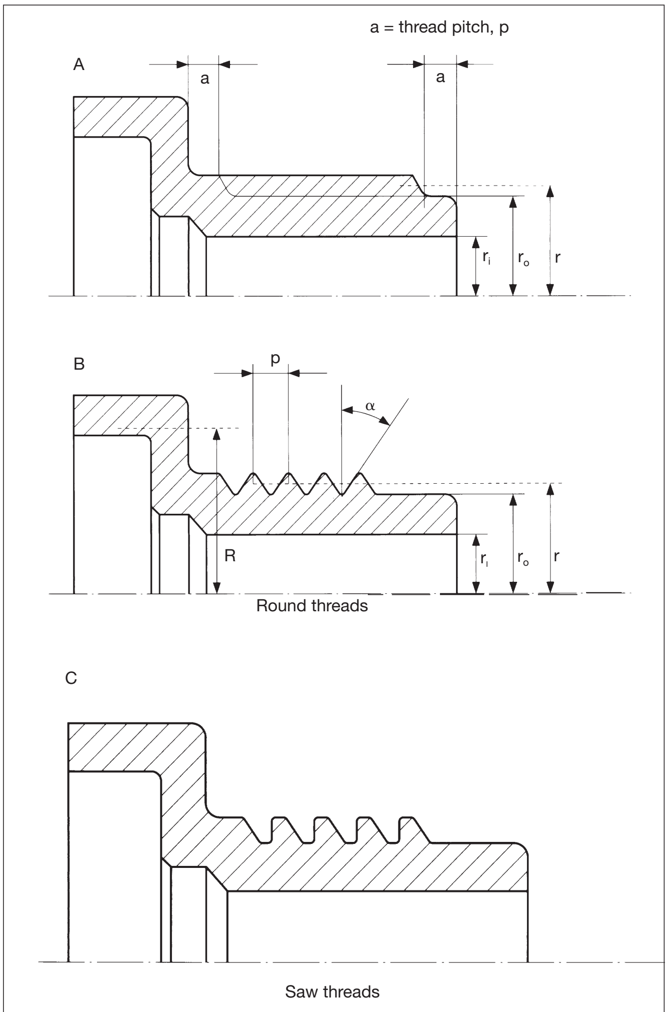


Fig. 9.04 **Plastic screws**

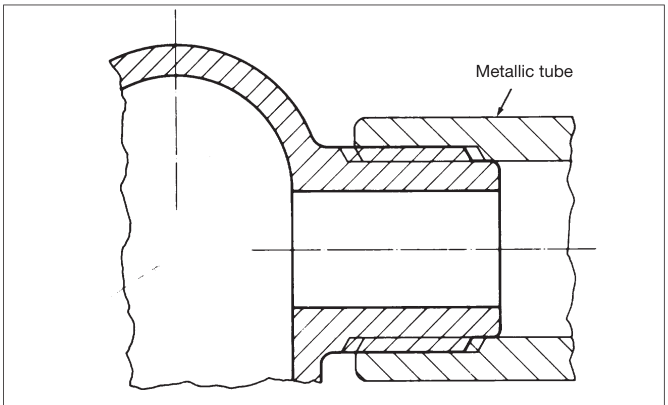


Fig. 9.05 **Preferred plastic-metal joint**

Practical Examples of Screwed Joints

For examples of plastic screws, see Figs. 9.06-9.08.

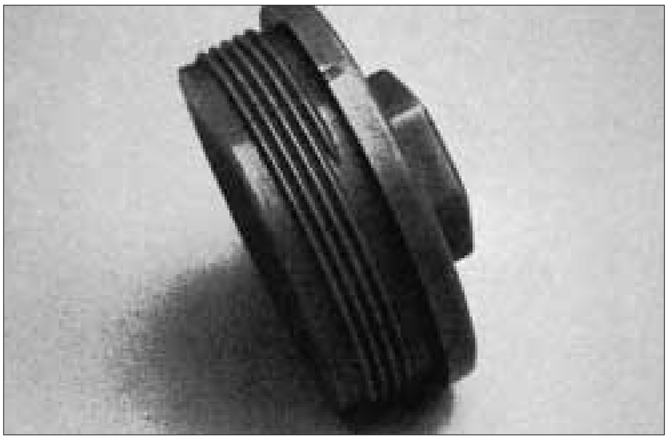


Fig. 9.06 **Plug**

Fig. 9.07 **Hose-coupling**

Fig. 9.08 **Coupling**

Design of Plastic Screws

Theoretical equations to calculate the strength of plastic screwed joints.

Torque on screw head:

$$M_h = F\,r\left(\frac{f_1 R}{r} + \frac{f_2}{cos(\alpha)} + \frac{p}{2\pi r}\right) = F\,r\,f_a$$

where:

F = axial force in screw [N]

R = radius of screw head contact surface

r = pitch radius of thread, Fig. 9.04 B

f_1 = friction between screw head and part

f_2 = friction between threads

p = thread pitch, Fig. 9.04B

α = angle of thread in radial direction, Fig. 9.04 B

Torque in thread:

$$M_T = F\,r\left(\frac{f_2}{\cos(\alpha)} + \frac{p}{2\pi r}\right) = F\,r\,f_b$$

Stresses in screw shaft:

axial: $\sigma_{ax} = F / A$

shear: $\tau = r\,M_T / I_p$

equivalent: $\sigma = \sqrt{\sigma^2_{ax} + 3\,\tau^2} \leq \sigma_y$

where:

$$A = \pi\,(r_o^{\,2} - r_i^{\,2})$$

$$I_p = \frac{\pi}{2}\,(r_o^{\,4} - r_i^{\,4})$$

r_o = outside radius of screw core, Fig. 9.04 A

r_i = inside radius of hollow screw; (solid: $r_i = 0$)

σ_y = tensile yield strength at design conditions

Maximum torque on screw head:

$$M_{h,\,max} = \sigma_y \Big/ \sqrt{\left(\frac{1}{r\,f_a A}\right)^2 + 3\left(\frac{r f_b}{I_p f_a}\right)^2}$$

Shear stress in thread

Due to differences in axial stiffness of screw and “nut”, the loads on the windings of the thread are not uniformly distributed over the length of the screw. Finite element studies have shown that in the case of a plastic screw with a steel nut, the first winding will take up to 50% of the total axial load. To avoid failure of the thread, the axial load in this case should be limited to,

$$F_{ax} \leq 2\,\pi\,r\,p\,\frac{\sigma_y}{\sqrt{3}}$$

Press Fittings

Press fitting provides a simple, fast and economical means for parts assembly. Press fits can be used with similar or dissimilar materials and can eliminate screws, metal inserts, adhesives, etc. When used with dissimilar materials, differences in coefficient of linear thermal expansion can result in reduced interference due either to one material shrinkage or expanding away from the other, or the creation of thermal stresses as the temperature changes. Since plastic materials will creep or stress relieve under continued loading, loosening of the press fit, at least to some extent can be expected. Testing under expected temperature cycles is obviously indicated.

Interference Limits

The general equation for thick-walled cylinders is used to determine allowable interference between a solid shaft and a hub:

$$I = \frac{\sigma_d D_s}{W} \left[\frac{W+\nu_h}{E_h} + \frac{1-\nu_s}{E_s} \right] \qquad (15)$$

and

$$W = \frac{(D_h^2 + D_s^2)}{(D_h^2 - D_s^2)} \qquad (16)$$

Where:

I = Diametral interference, mm
σ_d = Design yield stress, MPa
D_h = Outside diameter of hub, mm
D_s = Diameter of shaft, mm
E_h = Modulus of elasticity of hub, MPa
E_s = Elasticity of shaft, MPa
ν_h = Poisson's ratio of hub material
ν_s = Poisson's ratio of shaft material
W = Geometry factor

Case 1. Shaft and Hub of same plastic.

$E_h = E_s$; $\nu_h = \nu_s$. Thus equation 15 simplifies to:

$$I = \frac{\sigma_d D_s}{W} \times \frac{W+1}{E_h}$$

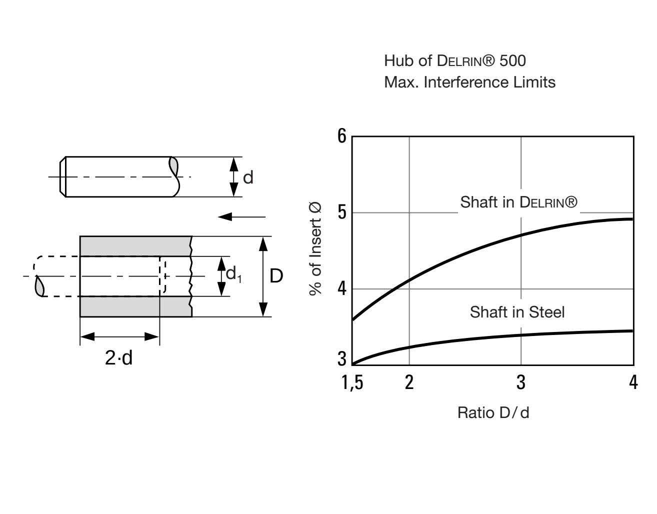


Fig. 9.09 **Maximum interference limits**

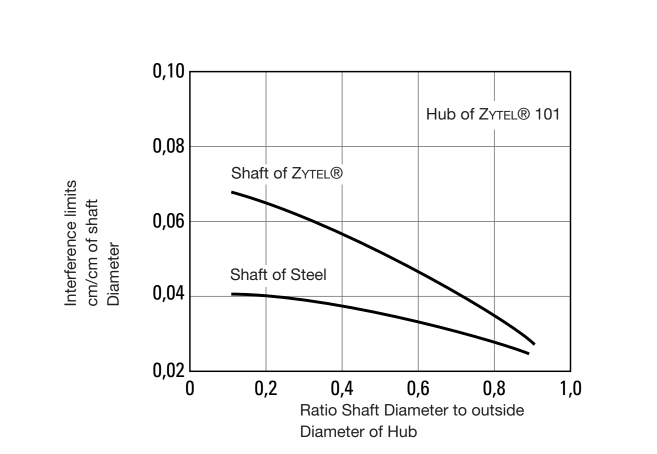


Fig. 9.10 **Theoretical interference limits for press fitting**
Based on yield point and Elastic Modulus at Room Temperature and Average Moisture Conditions

Case 2. Metal Shaft; Hub of plastic. When a shaft is of a high modulus metal or any other high modulus material, E greater than 50×10^3 MPa, the last term in equation 15 becomes negligible and the equation simplifies to:

$$I = \frac{\sigma_d D_s}{W} \times \frac{W+\nu_h}{E_h}$$

Theoretical Interference Limits for DELRIN® acetal resin and ZYTEL® nylon resin are shown in Fig. 9.09 and 9.10.

Press fitting can be facilitated by cooling the internal part or heating the external part to reduce interference just before assembly.

The change in diameter due to temperature can be determined using the coefficient of thermal expansion of the materials.

Thus:

$$D - D_o = \alpha\,(T - T_o)\,D_o$$

Where:

- D = Diameter at temperature T, mm
- D_o = Diameter at initial temperature T_o, mm
- α = Coefficient of linear thermal expansion, (1/K).

Effects of Time on Joint Strength

As previously stated, a press-fit joint will creep and/or stress relax with time. This will reduce the joint pressure and holding power of the assembly. To counteract this, the designer should knurl or groove the parts. The plastic will then tend to flow into the groves and retain the holding power of the joint.

The results of tests with a steel shaft pressed into a sleeve of DELRIN® acetal resin are shown in Figs. 9.11-9.13. Tests were run at room temperature. Higher temperature would accelerate stress relaxation. Pull out force will vary with shaft surface finish.

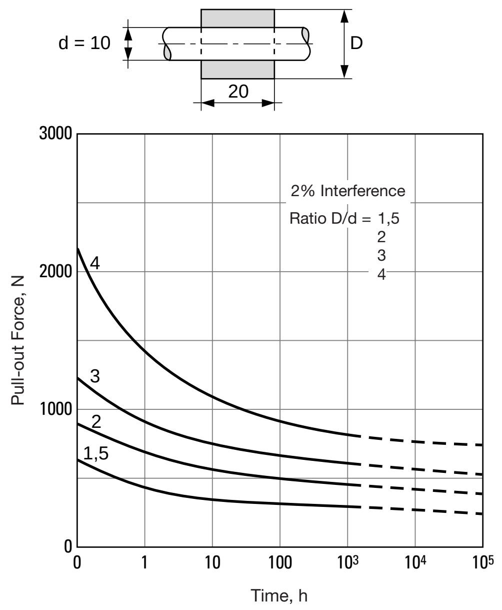


Fig. 9.11 **Time vs joint strength – 2% interference**

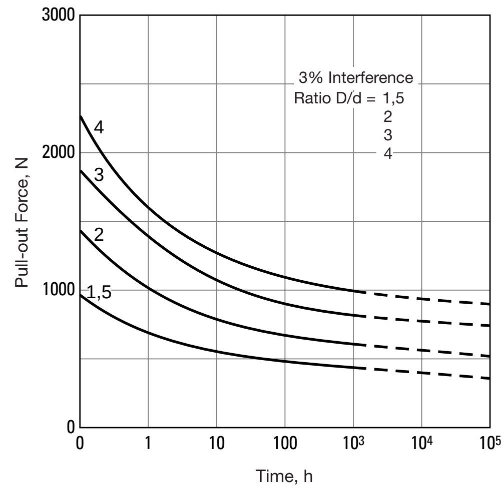


Fig. 9.12 **Time versus joint strength – 3% interference**

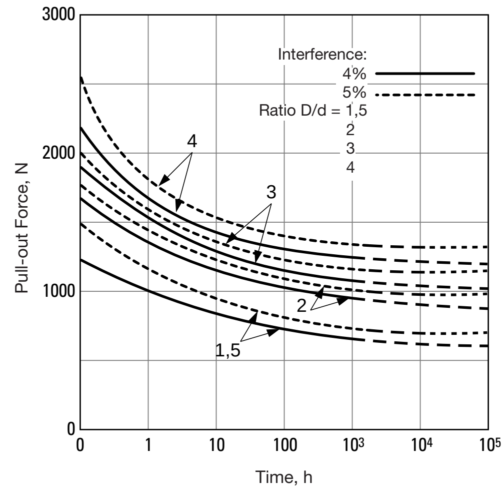


Fig. 9.13 **Time versus joint strength – 4 and 5% interference**

Assembly of Press-Fit Joints

The force required to press two parts together may be approximated by the equation:

$$F = \pi f P D_s L$$

and

$$P = \frac{\sigma_d}{W}$$

where:

- F = Assembly force
- f = Coefficient of friction
- P = Joint pressure
- D_s = Diameter of shaft
- L = Length of press-fit surfaces
- σ_d = Design stress
- W = Geometry factor (equation 16)

Coefficient of friction is dependent on many factors and varies from application to application. Coefficients from Table 7.01 may be used as approximations for rough strength calculations. When greater accuracy is required, tests on prototype parts are recommended.

Torsional Strength

The torsional strength of an interference joint is given by the equation:

$$T = F \frac{D_s}{2} \ (N \cdot mm)$$

Examples

Examples of press fittings are shown in Fig. 9.14-9.15.

This handle for a drill-crank is assembled with the three studs going into the three hubs with an interference fit of 4%.

Ball bearings are press-fitted into the grooved pulley.

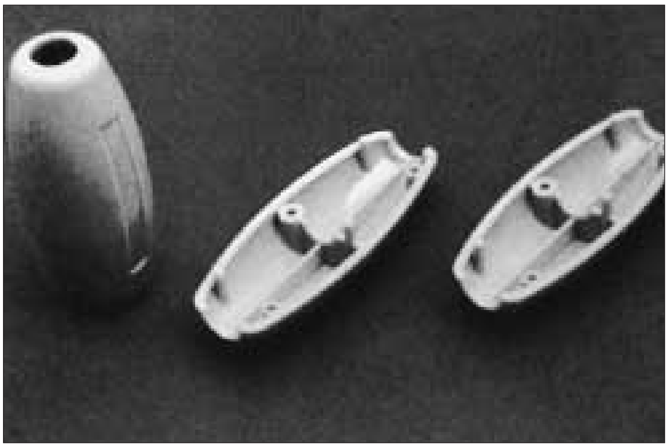


Fig. 9.14 **Drill-crank handle**

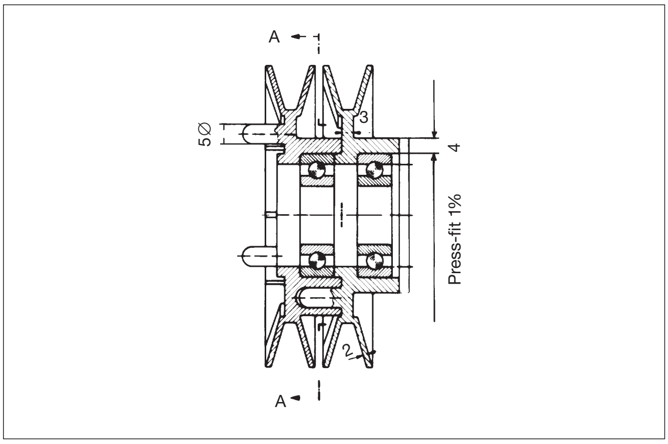


Fig. 9.15 **Ball bearing**

Snap-Fits

Introduction

The most common types of snap -fits are:

1) those with a full cylindrical undercut and mating lip (Fig. 9.16, Table 9.03),
2) those with flexible cantilevered lugs (Fig. 9.17),
3) those with spherical undercut (Fig. 9.18).

Spherical snap-fits can be seen as a special cylindrical snap-fit.

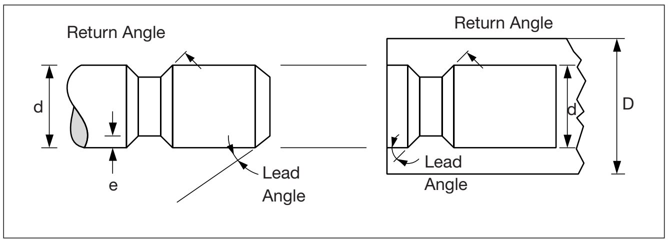


Fig. 9.16 **Cylindrical snap-fit joint**

Table 9.03 **Dimensions cylindrical snap-fit**

d	D (max., mm)		e (mm)	
mm	DELRIN®	ZYTEL® 101	DELRIN®	ZYTEL® 101
2	5		0,05	
3	8		0,07	
4	10	12	0,10–0,15	0,12
5	11	13	0,12–0,18	0,16
10	17	20	0,25–0,35	0,30
15	22	26	0,35–0,50	0,45
20	28	32	0,50–0,70	0,60
25	33	38	0,65–0,90	0,75
30	39	44	0,80–1,05	0,90
35	46	50	0,90–1,20	1,05

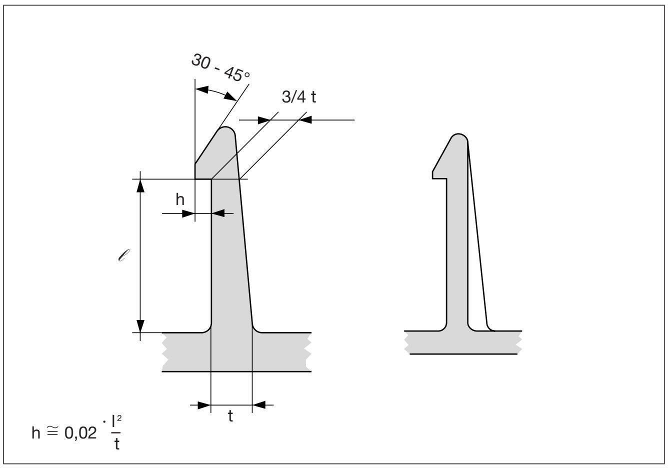


Fig. 9.17 **Snap-fit cantilevered lug**

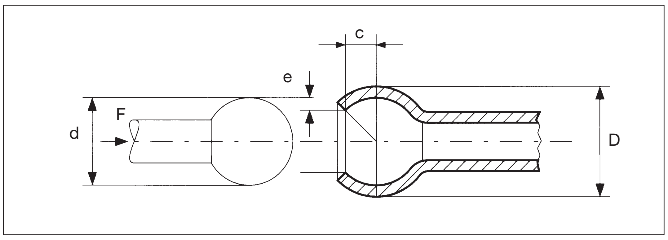


Fig. 9.18 **Spherical snap-fit**

Cylindrical snap-fits are generally stronger, but require greater assembly force than cantilevered lugs. In cylindrical snap-fits, the undercut part is ejected by snapping off a core. This requires deformation for removal from the mould. Materials with good recovery characteristics are required. For moulding complex parts, cantilevered lugs may simplify the moulding operation.

Design of Undercut Snap-Fits

In order to obtain satisfactory results, the undercut type of snap-fit design must fulfill certain requirements:

- *Uniform Wall Thickness*
  It is essential to keep the wall thickness constant throughout. There should be no stress risers.
- *Free to Move or Deflect*
  A snap-fit must be placed in an area where the undercut section can expand freely.
- *Shape*
  For this type of snap-fit, the ideal geometric shape is a circular one. The more the shape deviates from a circle, the more difficult it is to eject and assemble the part. Rectangular shaped snap-fits do not work satisfactorily.
- *Gates – Weld Lines*
  Ejection of an undercut from the mould is assisted by the fact that the resin is still at a very high temperature, thus its modulus of elasticity is lower and elongation higher. This is not the case, later, when the parts are being assembled. Often an undercut part will crack during assembly due to weak spots produced by weld lines, gate turbulence, or voids. If a weld line is a problem and cannot be avoided by changing the overall design or by moving the gate to some other location, the section at the weld line can be strengthened by means of a bead or rib.

Force to Assemble
During assembly, cylindrical snap-fit parts pass through a stressed condition due to the designed interference. The stress level can be calculated following the same procedure outlined in the previous section on press fits. With snap-fits, higher stress level and lower design safety factor is permissible due to the momentary application of stress.

The force required to assemble and disassemble snap-fit parts depends upon part geometry and coefficient of friction. This force may be divided arbitrarily into two elements: the force initially required to expand the hub, and the force needed to overcome friction.

As the beveled edges slide past each other, the maximum force for expansion occurs at the point of maximum hub expansion and is approximated by:

$$F_e = \frac{[\tan(\gamma) + f]\, \sigma_d\, \pi\, D_s L_h}{W}$$

Where:

F_e = Expansion force, N
f = Coefficient of friction (Table 7.01)
γ = Angle of beveled surface, lead angle
σ_d = Stress due to interference, MPa
D_s = Shaft diameter, mm
W = Geometry factor (Press-Fitting equation 16)
L_h = Length of hub expanded, mm

For the formulae for maximum diametral interference, I, see eq. (15), (pressfittings). For blind hubs, the length of hub expanded L_h may be approximately by twice the shaft diameter. Poisson's ratio can be found in the product data.

The force required to overcome friction can be approximated by:

$$F_f = \frac{\pi f\, \sigma_d D_s L_s}{W}$$

Where:

L_s = Length of interference sliding surface

Generally, the friction is less than the force for hub expansion for most assemblies. The value of [γ + *atan (f)*] should be less than 90° to be able to assemble the parts.

Examples

Suggested dimensions and interferences for snap-fitting a steel shaft into a blind hub of ZYTEL® nylon resin are given in Table 9.03. Terminology is illustrated in Fig. 9.16. A return bevel angle of 45° is satisfactory for most applications. A permanent joint can be achieved with a return angle of 90° in which case the hole in the hub must be open at the other end. It is a good practice to provide a 30° lead-in bevel on the shaft end to facilitate entry into the hub.

The toothed pulley in Fig. 9.19 is not subjected to significant axial load. A snap-fit provided with slots is, therefore, quite adequate. It allows a deeper groove and, therefore, a higher thrust bearing shoulder, which is advantageous since it is subject to wear.

Another example of press fitting is shown in the brake handle of Fig. 9.20.

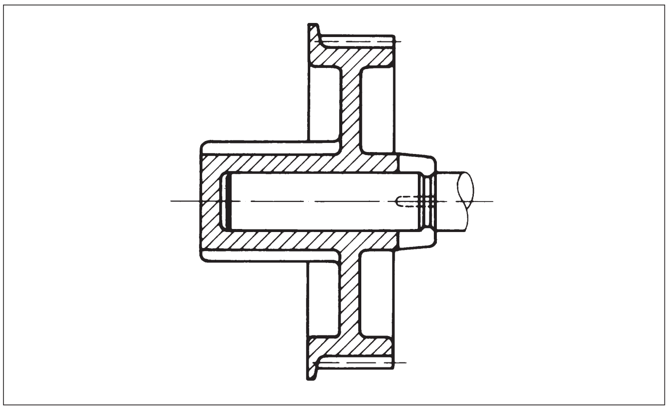


Fig. 9.19 **Tooth belt pulley**

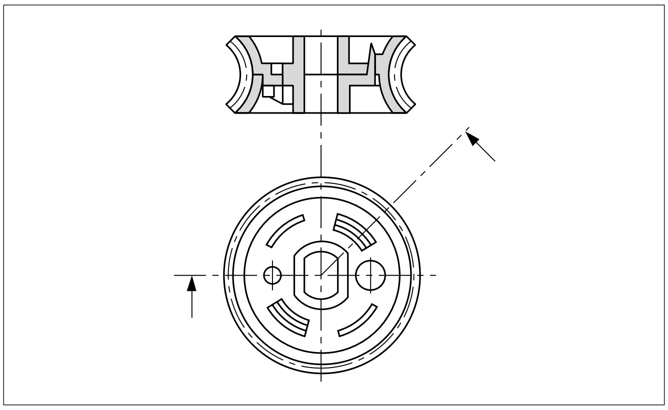


Fig. 9.21 **Snap-fit worm gear**

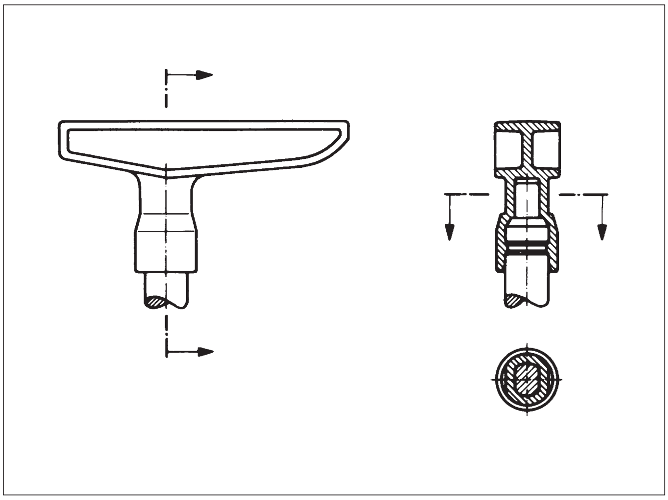


Fig. 9.20 **Brake handle**

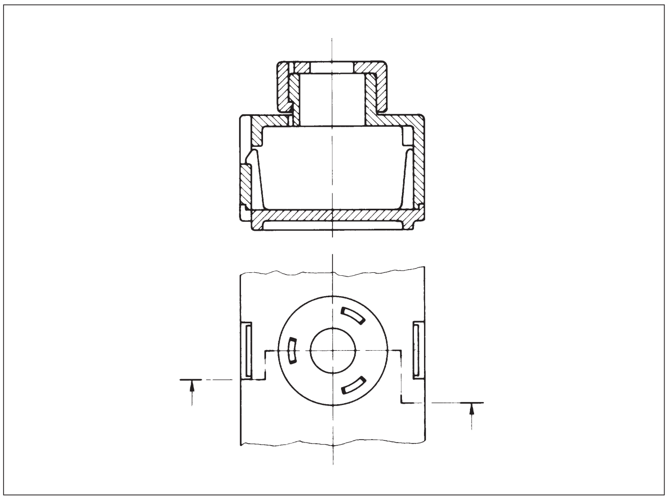


Fig. 9.22 **Micro-switch housing**

Cantilever Lug Snap-Fits

The second category into which snap-fits can be classified is based on cantilevered lugs, the retaining force of which is essentially a function of bending stiffness. They are actually special spring applications which are subjected to high bending stress during assembly.

Under working conditions, the lugs are either completely unloaded for moving parts or partially loaded in order to achieve a tight assembly. The typical characteristic of these lugs is an undercut of 90° which is always moulded by means of side cores or corresponding slots in the parts.

The split worm gear of Fig. 9.21 shows an example in which two identical parts (moulded in the same cavity) are snapped together with cantilevered lugs that also lock the part together for increased stiffness. In addition, the two halves are positioned by two studs fitting into mating holes.

The same principle is especially suitable for noncircular housings and vessels of all kinds. For example, there is the micro-switch housing in Fig. 9.22 where an undercut in the rectangular housing may not be functionally appropriate.

A similar principle is applied to the ball bearing snap-fit in Fig. 9.23. The center core is divided into six segments. On each side three undercuts are moulded and easily ejected, providing strong shoulders for the heavy thrust load.

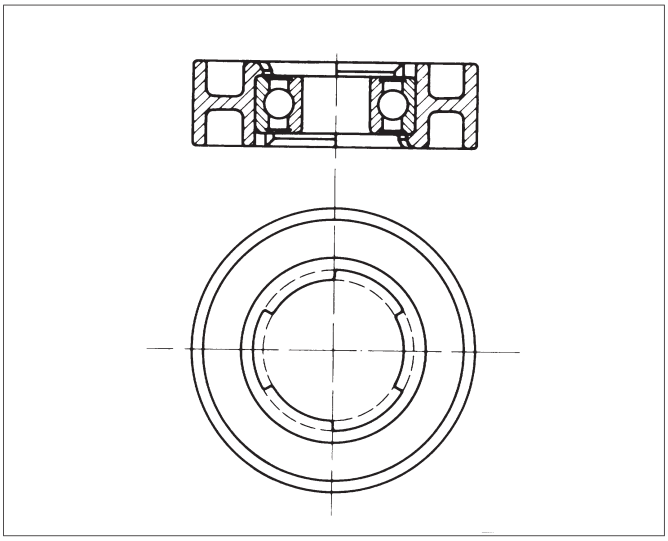


Fig. 9.23 **Snap-fit ball bearing**

Design of Cantilevered Lug Snap-Fits

Cantilevered lugs should be designed in a way so as not to exceed allowable stresses during assembly operation. This requirement is often neglected when parts in DELRIN® acetal resin are snapped into sheet metal. Too short a bending length may cause breakage (Fig. 9.24). This has been avoided in the switch in Fig. 9.25, where the flexible lugs are considerably longer and stresses lower.

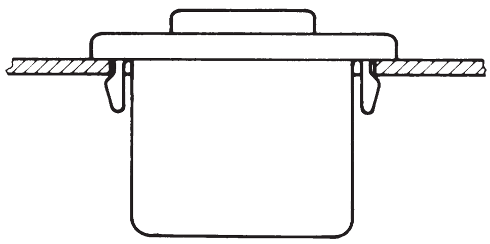


Fig. 9.24 **Undersized snap-fit lugs**

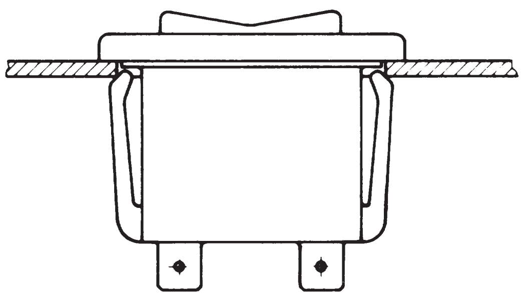


Fig. 9.25 **Properly sized snap-fit lugs**

Cantilevered snap-fit lugs should be dimensioned to develop constant stress distribution over their length. This can be achieved by providing a slightly tapered section or by adding a rib (Fig. 9.17). Special care must be taken to avoid sharp corners and other possible stress concentrations, which can cause failure during assembling.

To check the stress levels in a cantilevered lug, the beam equations can be used:

Deflection: $h = \dfrac{F\ell^3}{3\,EI}$ [min.]

Force: $F = \dfrac{3\,EI\,h}{\ell^3}$ [N]

Assembly force (per lug): $F_a = F\,(f + \tan\gamma)$ [N]

Stress: $\sigma = C\,\dfrac{F\ell}{I}\,y$ [MPa] (elastic)

Strain: $\varepsilon = \dfrac{100\,\sigma}{E}$ [%]

where:

F = force to deform snap-fit with interference h [N]

ℓ = effective length of snap-fit [mm]

E = modulus of elasticity [MPa]

I = moment of inertia of mean cross section[1], Table 4.01 [mm^4]

f = coefficient of friction

γ = top angle, [γ + *atan* (*f*)]<90

y = distance to neutral axis on tension side, Table 4.01 [mm]; cantilever: y = 0,5 × thickness

C = stress concentration factor

C = 1,0 for filleted snap-fits

C = 2,0 for poorly rounded off corners at the critical cross section

The allowable amount of strain depends on material and on the fact if the parts must be frequently assembled and disassembled.

Table 9.04 shows suggested values for allowable strains.

Table 9.04 **Suggested allowable strains (%) for lug type snap-fits**

	Allowable strain	
Material	Used once (new material)	Used frequently
DELRIN® 100	8	2–4
DELRIN® 500	6	2–3
ZYTEL® 101, dry	4	2
ZYTEL® 101, 50% RH	6	3
ZYTEL® GR, dry	0,8–1,2	0,5–0,7
ZYTEL® GR, 50% RH	1,5–2,0	1,0
RYNITE® PET GR	1	0,5
CRASTIN® PBT GR	1,2	0,6
HYTREL®	20	10

[1] If the snap-fit lug is tapered, the accuracy of the beam formulae drops. In that case a more complicated model (e.g. Finite Elements) is recommended.

Examples

For certain applications the snap-fit area can be provided with slots as shown in Fig. 9.26. This principle allows much deeper undercuts, usually at the sacrifice of retaining force. For parts which must be frequently assembled and disassembled this solution is quite convenient. For example, it is used successfully for assembling a thermostat body onto a radiator valve (Fig. 9.27). Here a metal ring is used to insure retention.

Pressure operated pneumatic and hydraulic diaphragm valves or similar pressure vessels sometimes require higher retaining forces for snap-fits.

This can be achieved by means of a positive locking undercut as shown in Fig. 9.28. A certain number of segments (usually 6 or 8) are provided with a 90° undercut, ejection of which is made possible through corresponding slots.

In the portions between the segments there are no undercuts. This design provides very strong snap-fit assemblies, the only limitation being elongation and force required during assembly. It is also conceivable to pre-heat the outer part to facilitate the assembly operation.

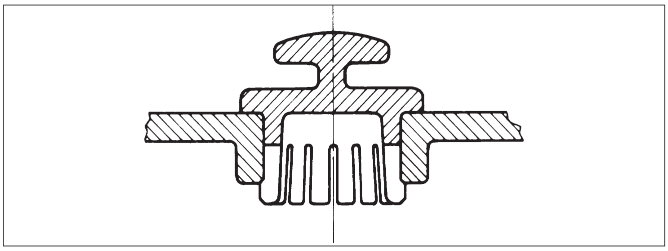


Fig. 9.26 **Slotted snap-fit**

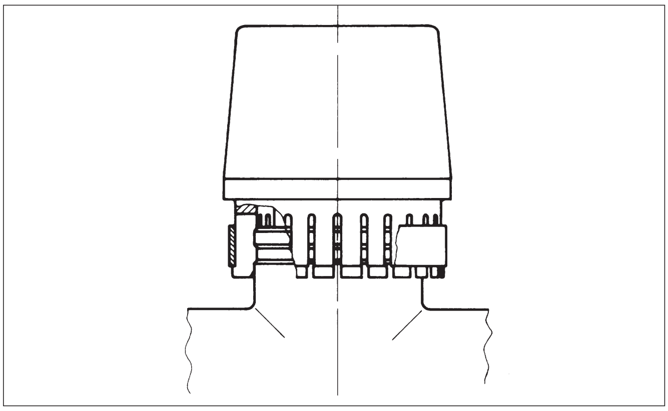


Fig. 9.27 **Snap-fit thermostat body**

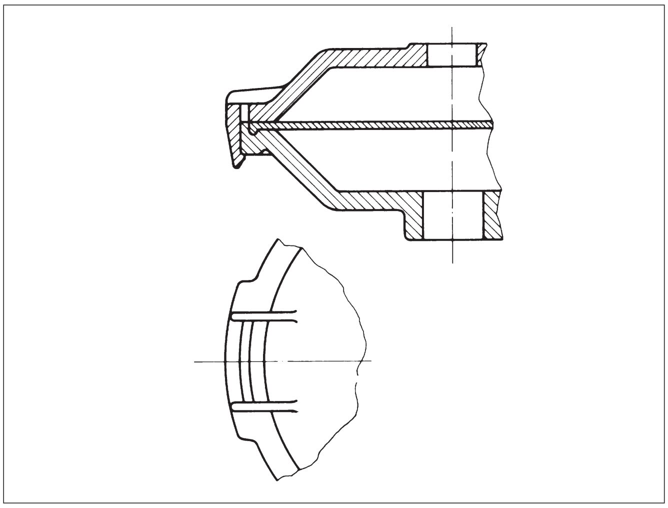


Fig. 9.28 **Snap-fitted diaphragm valve**

Hub Joints

This assembly method is generally used for parts transmitting a torque from one shaft to another by means of gears, or transmitting a mechanical movement with a cam, pump impellor or fan etc.

The connection is made generally by a key, by screws or a special shaft cross section.

In plastic, the design of such hub connections should be carefully done, fillets are very important. A lot of mistakes have been made. In order to avoid the repetition of such errors and also to avoid long explanations, a look at some practical examples of good design is suggested.

Practical Examples, see Fig. 9.29–9.35.

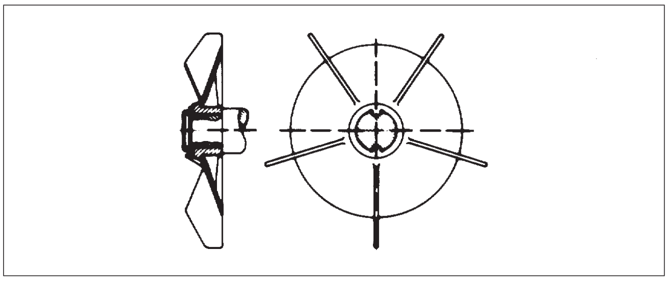


Fig. 9.29 **Fan-blower**

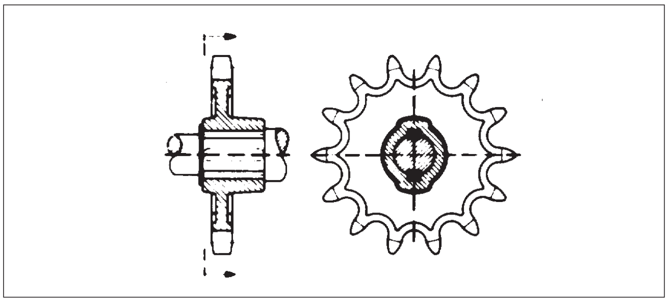


Fig. 9.30 **Chain-wheel**

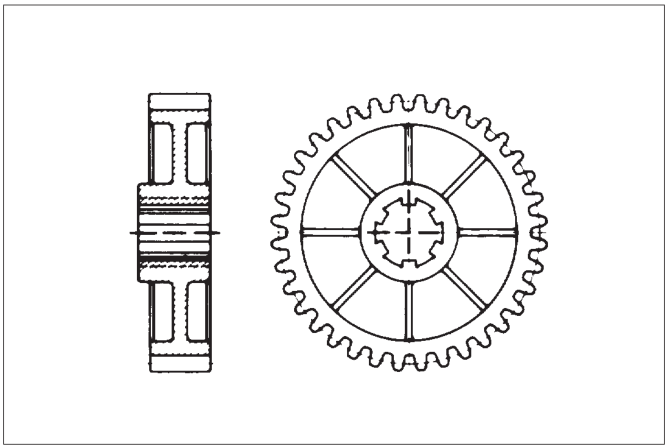


Fig. 9.31 **Gear**

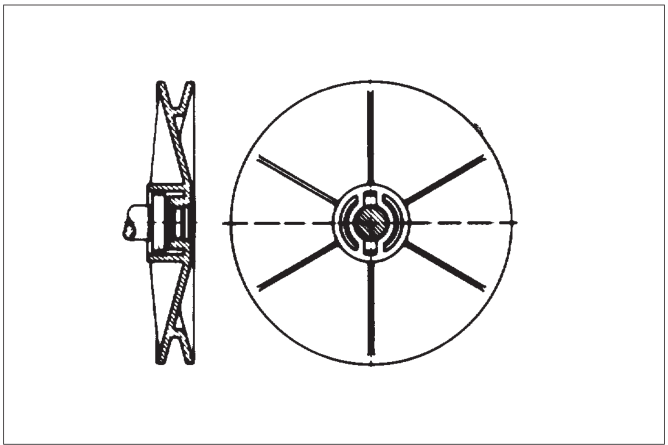


Fig. 9.32 **V belt-wheel**

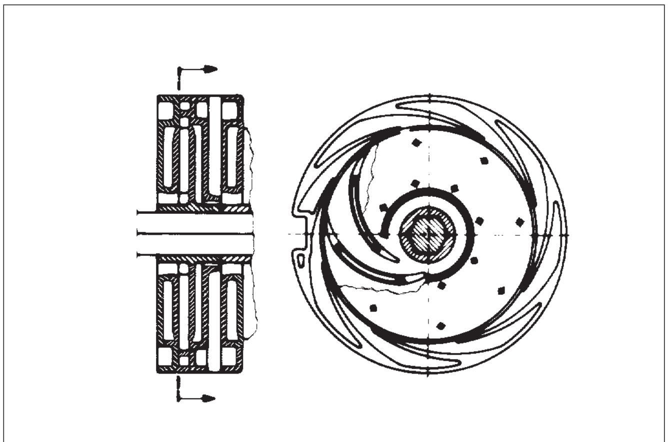


Fig. 9.33 **Centrifugal pump**

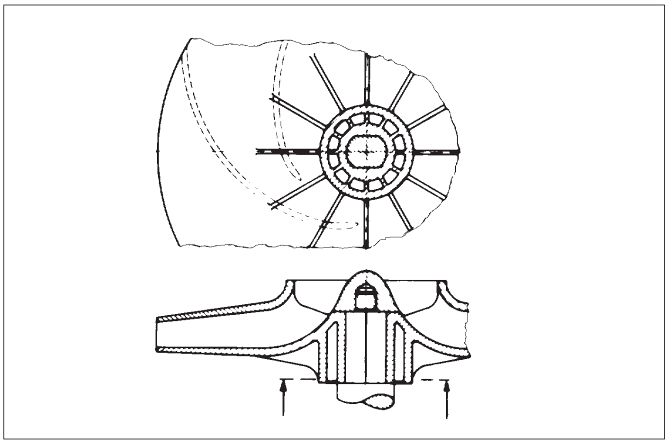


Fig. 9.34 **Outward-flow turbine**

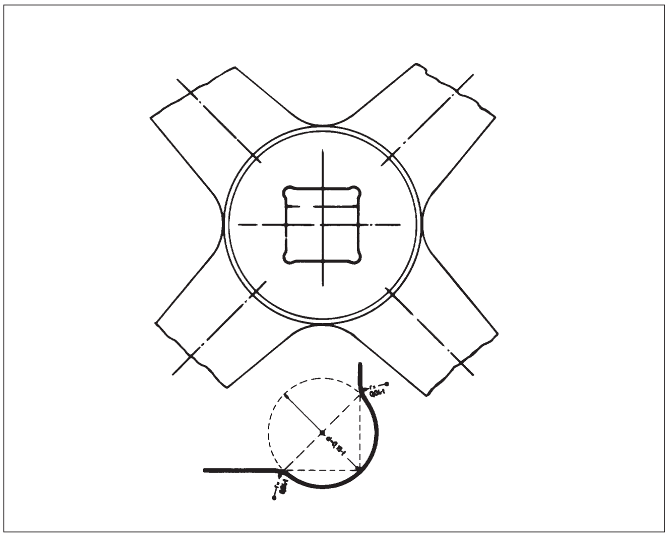


Fig. 9.35 **Suggestion for a hub connection**

Fig. 9.36 **Washing machine pump**

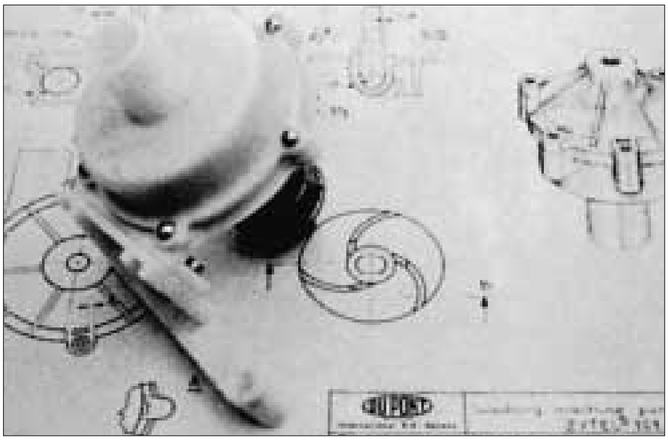


Fig. 9.37 **Washing machine pump**

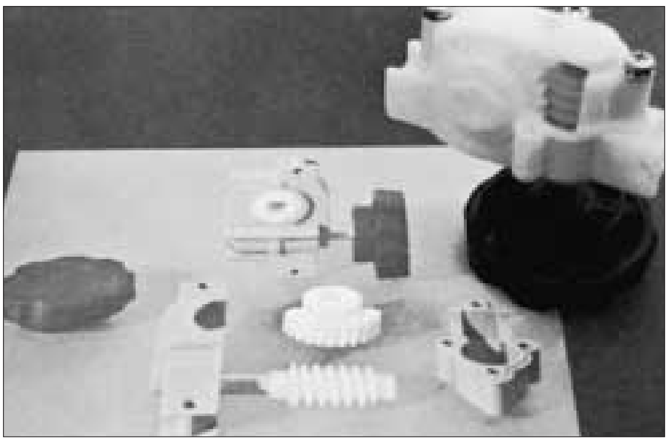


Fig. 9.38 **Window gearbox**