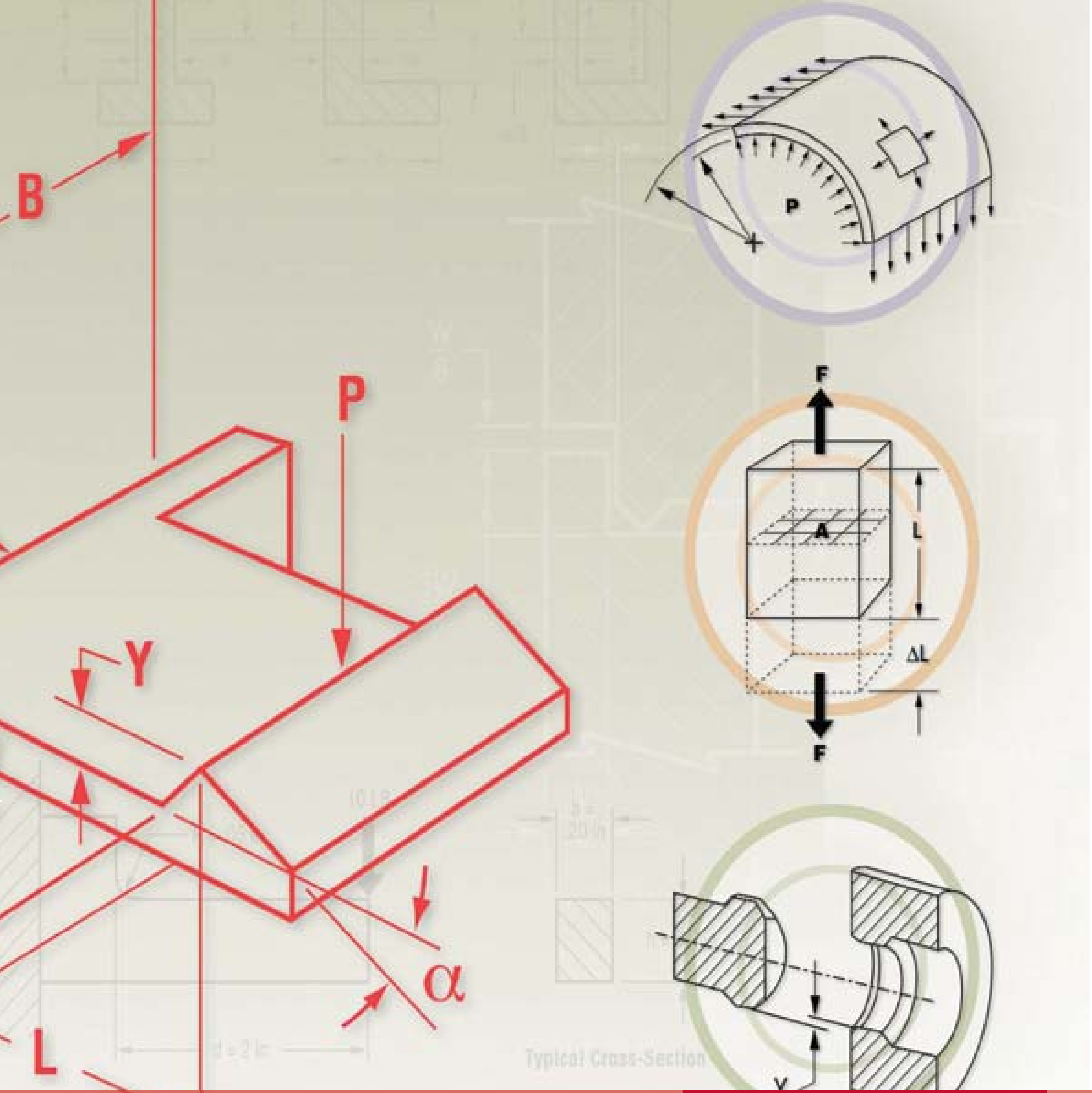


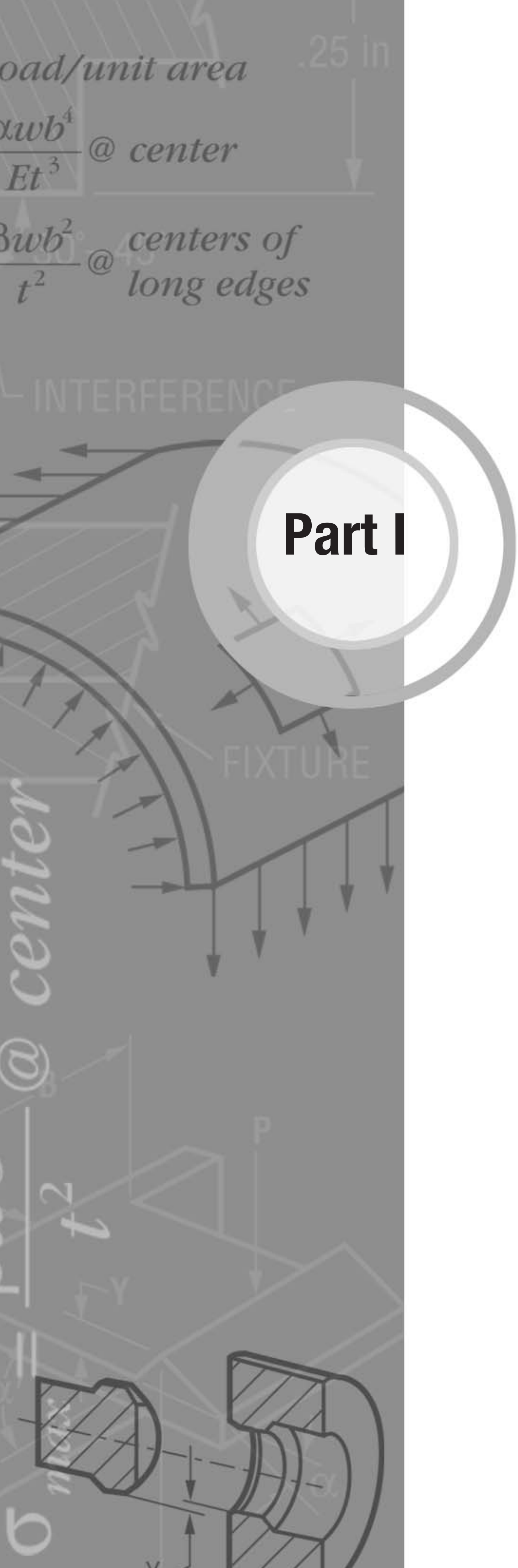
Design Solutions Guide

BASF
The Chemical Company


OF WELD
MIN.
LEAD-IN
.020 in
SLIP FIT

Table of Contents

Topic	Part/Page



Part I

Welcome!

Part I: Welcome!

As part of our customer-driven philosophy of doing business, we have prepared this guide to give you a general product design overview with a focus on plastic part design. It is our goal to provide all our customers with the optimum level of technical and design support during their product development process.

Overview

Proper design strategy includes:

a) a concern for safety and performance

b) appropriate material selection and preparation for processing to achieve the ultimate functional design goal

c) maximum functionality

d) minimum material usage

Our intent in developing this Design Solutions Guide is to supply general information for the customer on a variety of applications as a precursor to the more narrowly focused information which will appear in subsequent manuals. Manuals on specific applications will expand upon this general guide and address those precise topics. Your design success is our primary concern.

Recycling

Recycling is part of an all-important global drive toward reducing contamination, landfill volume and saving natural resources. Recycling is good business too, since in many cases, it results in reduced product lifestyle costs. Recycled plastic materials can often be specified into less-demanding applications.

There are some design implications which should be considered when using recycled products:

- One should use the same material in assembly applications where parts are permanently affixed to one another. Mixing material types is acceptable for mechanically assembled units which can be disassembled.
- Color availability is generally limited.
- Cadmium-free colors are available.

Safety

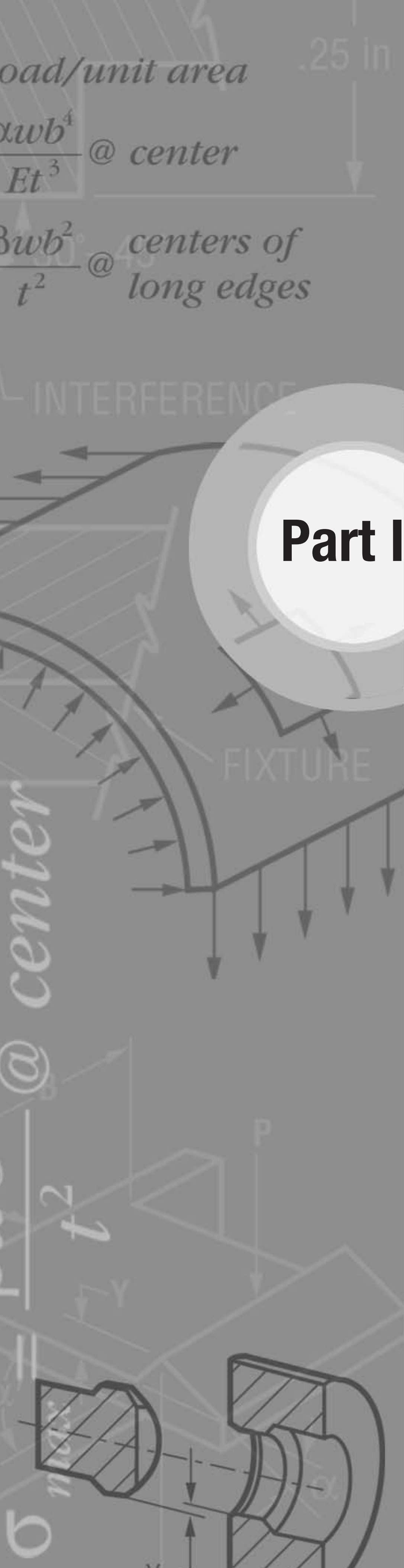
When designing parts, a factor of safety should be used to manage the risk of catastrophic, premature and short-term failures. The factor is contingent upon numerous conditions, including type of application, temperature, lack of material homogeneity, unforeseen overloads, unknowns, etc.

Having predetermined load conditions, the introduction of a factor will extend the service life of the product depending on the value used. The value used for the factor is based on the criticality of the function. Engineering handbooks cover this subject in more detail.

Nylon has some unique characteristics. For instance, in the presence of moisture, it changes its physical properties. Strength, stiffness, surface hardness and brittleness will decrease while elongation, ductility, impact resistance, dimensions and creep will increase. These characteristics need to be tempered with the safety factor during design. These are reasons for designing with information not found on data sheets which are readily issued by material suppliers.

Data sheet information is point data only.

We trust you will find this and our other manuals of great value. We are always available to assist when needed.



Part II

Design Considerations for Injection Molded Parts

Part II: Design Considerations for Injection Molded Parts

The injection molding process is the most common process for producing economical and automated thermoplastic parts. It commonly requires the use of steel molds, injection molding machinery and auxiliary equipment.

To injection mold a part, there are numerous design aspects which should be addressed. They are:

1. Parting Lines
2. Draft Angles
3. Wall Thickness
4. Fillets and Radii
5. Bosses
6. Ribs
7. Opening Formations
8. Shrinkage
9. Gating
10. Vents
11. Potential Knit Lines

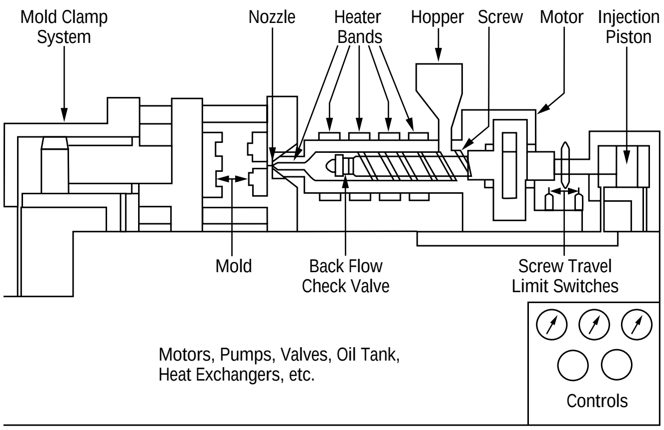


Figure II-1. Schematic of Reciprocating Screw Injection Molding Machine

Parting Lines

Parting line consideration depends upon shape and the function of the part. If a shaft diameter is used as a bearing surface and is going to be injection molded, it cannot tolerate a conventional parting line. In this situation, incorporating small flats on the shaft at the parting line will avoid mismatch and minimal flash conditions (see Figure II-2).

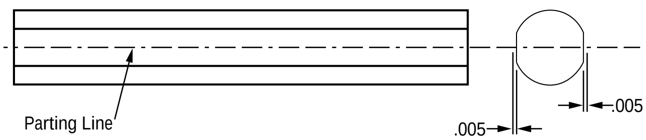


Figure II-2. Free Running Shaft

The parting line depends on the shape of the part. Figure II-3 illustrates an irregular parting line. When a parting line involves two mating halves with close tolerances, the mold mating steel parts should be interlocked for good positioning or take in an allowance for possible mismatches. The allowance should be in the 0.005 in to 0.010 in range relative to the finished dimension.

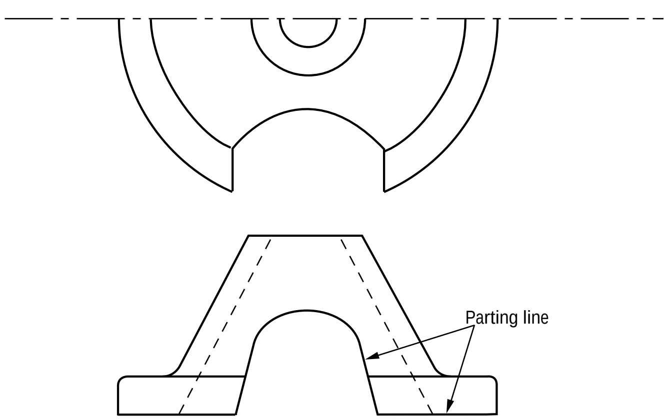


Figure II-3. Irregular Parting Line

Draft Angles

Draft is necessary for the ejection of the parts from the mold. Always design with draft angles. Recommended draft angle is normally 1° with 1/2° on ribs. Some draft angle is better than none and more draft is desirable if the design permits. Where minimum draft is desired, good polishing is recommended and feature depth should not exceed .5in.

Wall Thickness

The number one rule for designing plastic parts is *uniform wall thickness*. Uniform walls aid in material flow in the mold, reduce the risk of sink marks, molded-in stresses and differential shrinkage.

For non-uniform walls, the change in thickness should not exceed 15% of the nominal wall (see Figure II-4) and should transition gradually.

Corners should always be designed with a minimum fillet radius of 50% of the wall thickness and an outer radius of 150% of the wall thickness to maintain a uniform wall thickness (see Figure II-4).

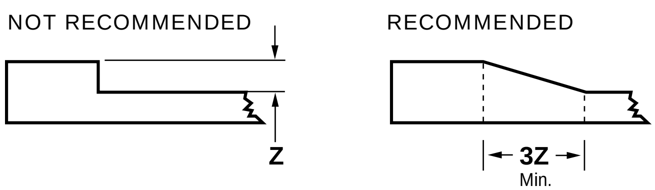


Figure II-4

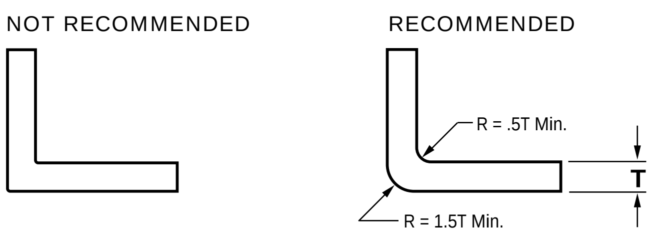


Figure II-4

Fillets and Radii

Sharp corners should be avoided. They are the number one cause of part failure, stress concentrations, poor flow patterns and increased tool wear (see Figure II-5).

Indicate radii at all inside and outside corners to the maximum which a design will allow.

Bosses

Bosses are usually designed to accept inserts, self-tapping screws, drive pins, etc., for use in assembling or mounting parts.

Avoid stand-alone bosses wherever possible. Bosses should be attached to walls or ribs by means of ribs or gussets for structural stability (see Figures II-5 & 6).

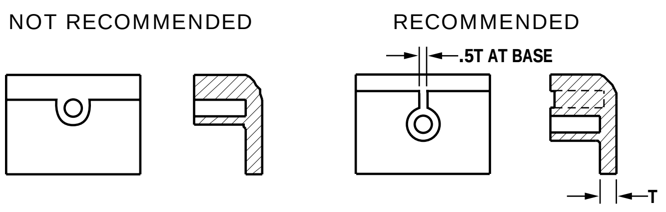


Figure II-5

The O.D. of the boss should ideally be 2.5 times the screw diameter for self-tapping screw applications. Thick-walled bosses with bases greater than 50% of the wall could form visible sink marks. To overcome this condition, a thinner-walled boss of 2.0 times screw diameter or less can have multiple ribs (see Figure II-6).

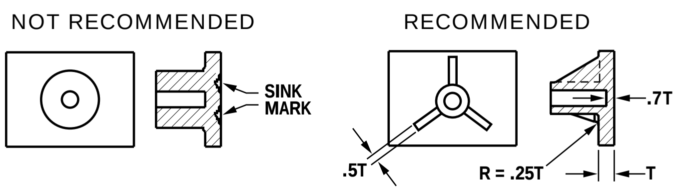


Figure II-6

The thickness at the base of the ribs and gussets used to stabilize bosses should not exceed 50% of the thickness of the adjoining wall.

Boss inside and outside diameters should have 1/2° draft per side. See Part V of this guide for additional information on bosses for press fits and self-tapping screws.

Ribs

Ribs should be used when needed for stiffness and strength or to assist in filling difficult areas.

In structural parts where sink marks are of no concern, rib base thickness (t) can be 75–85% of the adjoining wall thickness (T).

For appearance parts, where sink marks are objectionable, rib base thickness (t) should not exceed 50% of the adjoining wall thickness (T) if the outside surface is textured and 30% if not textured. Sink marks are also dependent on the material.

Rib height should be at least 2.5–3.0 times the wall thickness (T) for effective strength.

Draft should be 1/2° per side nominal.

Fillets at the base of the rib should be .020 in minimum.

Multiple ribs should be spaced at least 2 times the wall thickness apart to reduce molded in stress and problems in cooling of the mold (See Figure II-7).

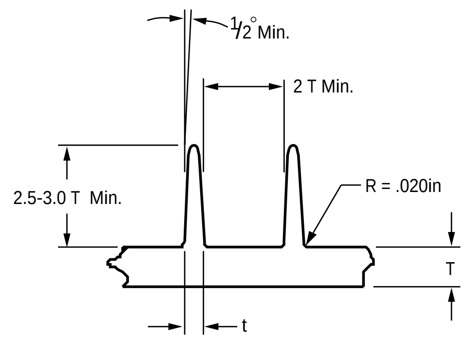


Figure II-7

Openings

When an opening is desired in a part (such as to accommodate a snap-fit) and is to be formed without core pulls, a 5° angle mating of the core and cavity is required (see Figure II-8).

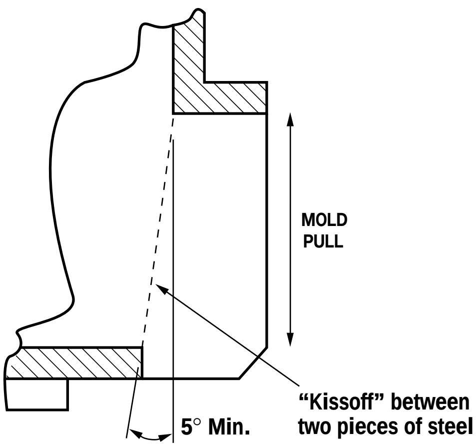


Figure II-8

Shrinkage

Shrinkage is a characteristic of resin which occurs during molding. Different resins have different mold shrinkages. Crystalline and semi-crystalline materials exhibit higher shrinkage than amorphous materials. Unreinforced plastics have higher shrinkage than reinforced grades. It is important that the grade of material be selected before the mold is constructed and that the proper mold shrinkage be specified. Basic shrinkage data is obtained from ASTM tests or ISO tests.

Material shrinkage can vary with part and tool design: thick walls will have higher shrinkage rates than thin, variation in section thickness can cause differential shrinkage and warpage; flow direction will effect shrinkage, particularly with glass fiber-reinforced grades (more when perpendicular to flow and less when parallel to flow; see Figure VII-14).

Shrinkage is also influenced by process conditions. As cavity pressure increases, shrinkage typically will decrease. The mold and melt temperature will also influence shrinkage. Cooler molds will reduce shrinkage while hotter melt temperatures will increase shrinkage especially with semi-crystalline materials.

Contact BASF Technical Services for shrinkage recommendations on any of our products.

Gating

The gate connects the part to the runner system. It is usually the thinnest cross-section in the entire system. The design of the gate is dependent on tool design, part geometry and the material selection.

Gate location, size, type and number must also be addressed.

- Gates should be located away from high stress or impact areas.
- Gate configuration and location should minimally affect part appearance.
- Gate design and location should eliminate secondary degating operations, if possible.
- The gate should be located to best fill the part; position flow for advantageous glass fiber orientation, if present, and locate knit lines in low-stress areas.

Refer to the BASF Injection Molding Processing Guide for more details.

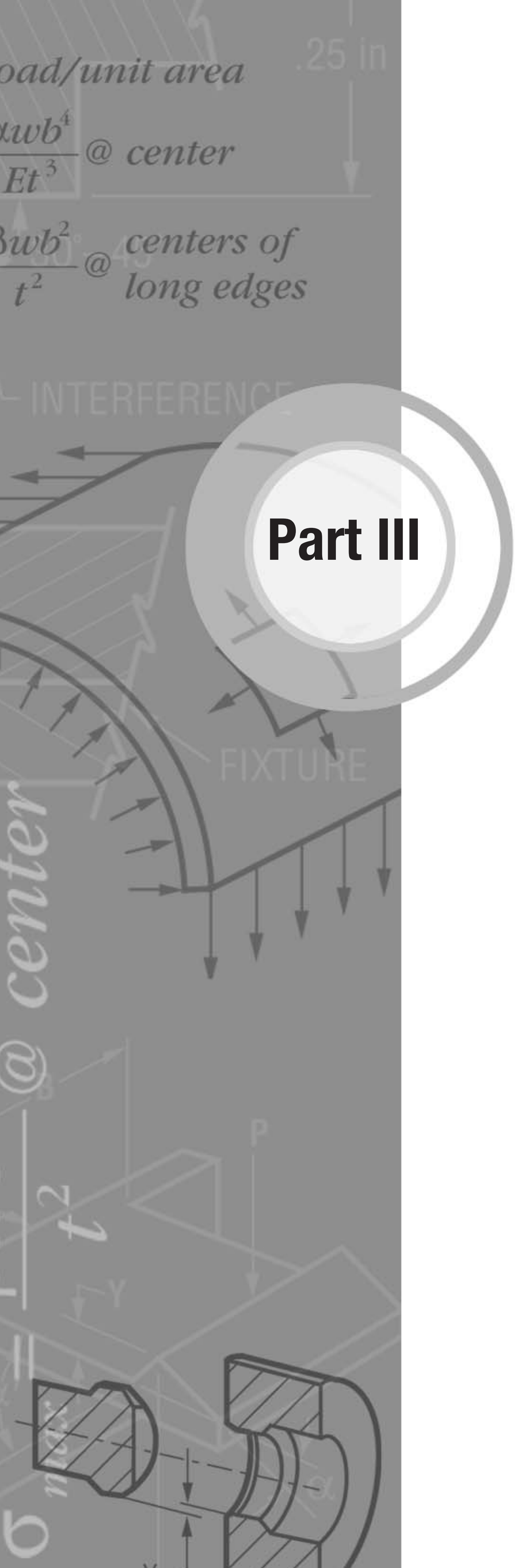
Vents

Vents are regions in the mold where clearance is used to permit trapped air and gases to escape. Lack of proper venting can cause excessive injection pressure, short shots, burn marks and splay. A cavity can be considered adequately vented when plastic can be injected at high rates without showing signs of burn marks.

There are many ways to vent a mold. Typically, this is done by machining numerous shallow channels at the parting line. The dimensions of the channels are dependent on the material injected. Contact BASF Technical Services for this information. Other ways to vent a mold are ejector pins, vent pins and runners. Flow analysis can identify areas needing specific venting for best results.

Potential Knit Lines

Knit lines are areas in the molded part where two or more flow fronts converge. This area generally has lower strength than the other areas of the part. One should anticipate knit lines, which show up well in flow analysis programs, and direct them away from anticipated high stress areas of the part where possible. Knit lines generally form on the opposite side of obstacles which are in the way of the normal flow path, such as pins that form holes in the part or bosses designed to accept inserts.



Part III

Structural Design

Part III: Structural Design

Stress

Stress-Strain

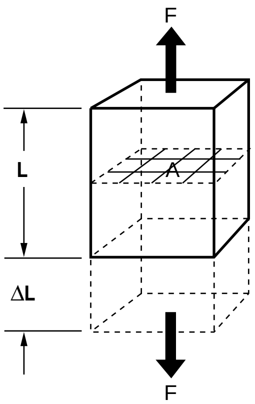
When a force is applied to a part, the result is a deformed part which is both stressed and strained. The stress (σ) in a part is determined by the load (F) applied per unit area.

$$\sigma = \frac{F}{A}$$

Strain (ε) is a change in the part's length over its original length (see Figure III-1).

$$\varepsilon = \frac{\Delta L}{L}$$

Figure III-1. Strain



Hooke's Law is the relationship between stress and strain, such that strain is proportional to stress and the modulus of elasticity (E) or Young's Modulus is the constant of proportionality:

$$E = \frac{\sigma}{\varepsilon}$$

.All plastic materials have a characteristic stress-strain curve (see Figure III-2).

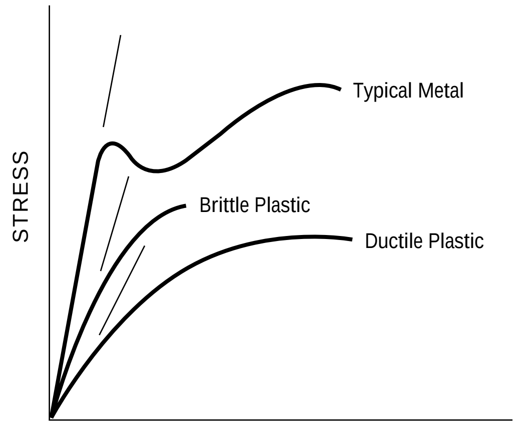


Figure III-2. Stress vs. Strain

In order to obtain a stress-strain curve for a resin, a tensile test is performed at room temperature. The part is axially loaded with the force directed away from the part. The stress-strain curve describes the resin's response to a force applied at a predetermined rate (.2–.5in/min). The yield point (deviation from the straight line) is dependent upon the temperature at which it is measured. Plastic materials do not have a distinct linear response like that of metals. Temperature and humidity can change these curves. Higher temperatures and humidity generally reduce stress carrying ability and increase strain (deflection).

When a plastic part is subjected to a high enough external force, it will exceed its elastic limit (the straight line portion of the curve in Figure III-2). Its original size and shape will no longer remain constant. The material behaves linearly as long as the stress is kept well below the yield point. Once the yield point is reached, the material at that point is in its plastic (non-linear) range. Exceeding the linear range results in some permanent deformation of the material. It is only when the part has not been stressed beyond its elastic limit that Hooke's Law applies. There are many types of stresses: Normal, Shear, Torsional, and Bending. Each will be discussed in detail.

Normal Stress

Normal stress (σ) is the ratio of the force applied over a given cross-sectional area (A):

$$\sigma = \frac{F}{A}$$

When a load is applied perpendicular (normal) to the plane of a surface, it results in a stress normal to the cross-section. A normal stress is either tensile or compressive, depending on the direction of the force applied. When the force is directed away from the part, the stress is tensile (see Figure III-3), and when the force is directed toward the part, the stress is compressive (see Figure III-4).

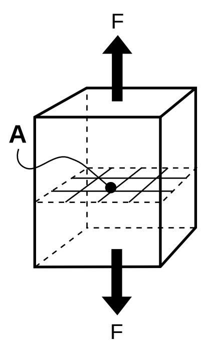


Figure III-3. Tensile Stress

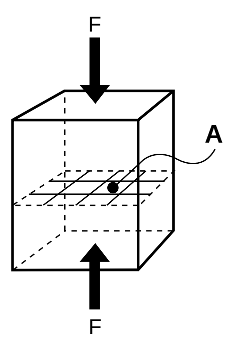


Figure III-4. Compressive Stress

Shear Stress

Shear Stress (τ), like tensile and compressive stress, is also expressed as the force applied over a cross-sectional area (A).

$$\tau = \frac{F}{A}$$

The difference is that the result of the force being applied is a stress which is parallel to the cross-section (see Figure III-5).

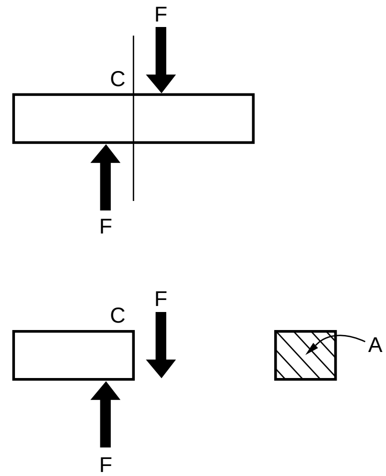


Figure III-5. Shear Stress

Illustration credit: Beer & Johnson, Mechanical Materials.

Torsional Stress

When a part is in torsion (T), twisted along its longitudinal axis, there is, at any point on the plane of the section, a shear stress (τ) (see Figure III-6). The maximum shear stress of a shaft in torsion is calculated by:

$$\tau = \frac{Tc}{K}$$

Where variable c is the distance from the center of the shaft to the outer surface where the maximum stress occurs, and K is the torsional constant.*

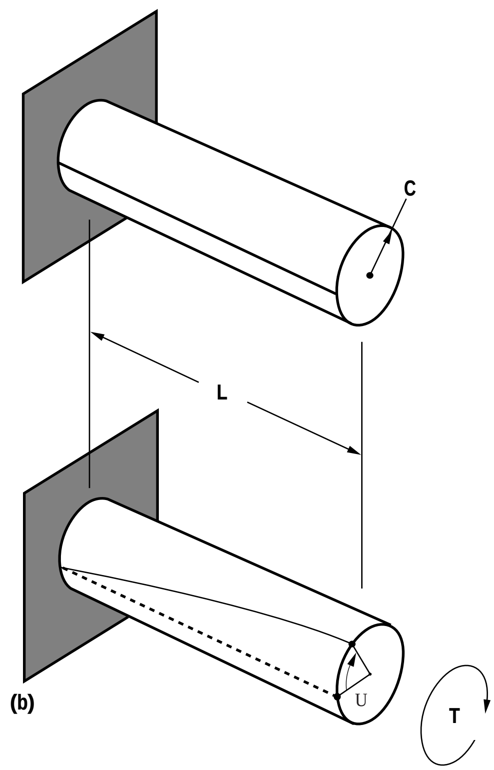


Figure III-6. Torsional Stress

Illustration credit: Beer & Johnson, Mechanical Materials.

Torque is calculated by:

$$T = \frac{\theta K G}{L}$$

Where variable θ is the angle of twist due to the torque, G is the modulus of rigidity, and L is the length of the member. The equation can be manipulated to calculate the angle of twist (θ):

$$\theta = \frac{TL}{KG}$$

Example for solid circular shaft:

A 5in long solid circular shaft of .5in diameter, is subjected to a torque of 8 in–lb. Calculate the shear stress and angle of twist.

Using Ultramid 8267 resin (40% mineral/glass), at room temperature and dry as molded (DAM):

E = *Modulus of elasticity* = 1,110,000 psi

υ = *Poisson's Ratio* = 0.35

$$G = \frac{E}{2(1+\upsilon)} = \frac{1{,}110{,}000psi}{2(1+0.35)} = 411{,}111\ psi$$

$$K = \frac{1}{2}\pi R^4 = \pi\frac{(0.25)^4}{2} = 0.006136in^4$$

$$\tau = \frac{(8in\text{-}lb)(0.25in)}{.006136in^4} = 326\ psi$$

$$\theta = \frac{(8in\text{–}lb)(5in)}{(411{,}111psi)(.006136in^4)} = .0158\ rad$$

The shear stress is extremely low in comparison to the resin's tensile strength, therefore, the shaft can withstand the 8 in–lb torque applied.

* Please reference the formula for the torsional constant of various cross-sections in the Torsional Formula Section (see Figure III-8).

Bending Stress

When a simply supported structural member is in flexure, the top section will be in compression and the bottom surface will be in tension.

The center of the member is the neutral axis (N.A.), and is a region of zero stress. The maximum stresses will occur at the extreme fibers (a and b). Bending stress is expressed as:

$$\sigma = \frac{Mc}{I}$$

where M is the bending moment, c is the distance from the neutral axis to the extreme outer fiber and I is the moment of inertia (see Figure III-7).

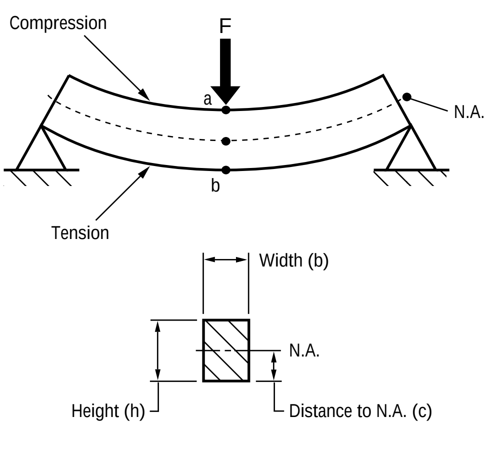


Figure III-7. Neutral Axis in Bending

Example for cantilever beam:

The I-beam shown has applied force of 20lb (see Figure III-8). Once the moment is calculated, then the bending stress can be calculated. The chosen material is Ultramid® 8267 (40% mineral/glass), tensile strength at room temperature and dry as molded (DAM) is 20,000 psi.

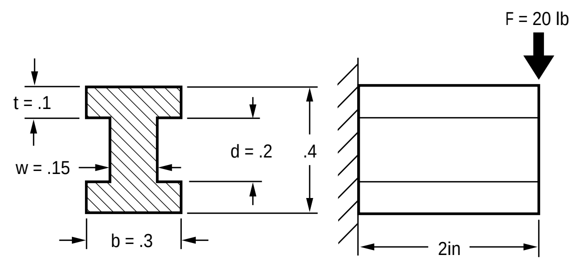


Figure III-8.
Cantilever I-Beam Cross-Section

M = Force x distance = (20lb)(2in) = 40in-lb

$$I^* = \frac{b(d+2t)^3}{12} - \frac{(b-w)d^3}{12} = \frac{.3(.2+2(1))^3}{12} - \frac{(.3-.15)(.2)^3}{12} = .0015in^4$$

$$\sigma_b = \frac{(40in\text{-}lb)(.2in)}{.0015in^4} = 5333\ psi$$

In comparing the bending stress with the material's tensile strength, the I-beam will be able to withstand the 20 lb force applied.

* Please reference the Beam Sections that follow for the moment of inertia of various cross-sections.

Section Properties of Various Cross-Sections (Straight Beams)

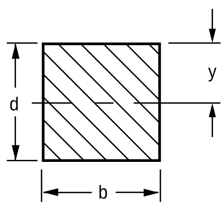
Explanation of Variables

The table below provides the following useful section properties for Figures III-9 through III-14:

A = Area
Y = Distance from centroid to extreme fiber
I = Moment of Inertia about principal axis
r = Radius of gyration about principal axis

Beam Sections

Rectangle

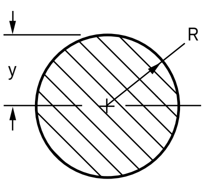


$A = bd \qquad I = \frac{1}{12} bd^3$

$y = \frac{d}{2} \qquad r = 0.2887d$

Figure III-9

Solid Circle

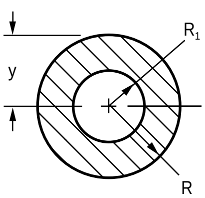


$A = \pi R^2 \qquad I = \frac{\pi}{4} R^4$

$y = R \qquad r = \frac{R}{2}$

Figure III-10

Hollow Circle

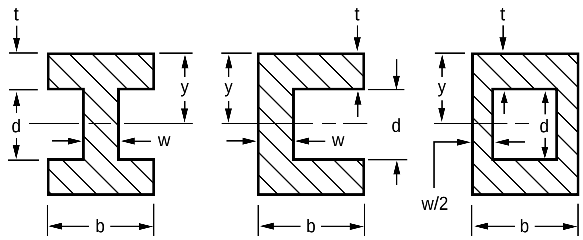


$A = \pi(R^2 - R_i^2) \qquad I = \frac{\pi}{4}(R^4 - R_i^4)$

$y = R \qquad r = \frac{1}{2}\sqrt{R^2 + R_i^2}$

Figure III-11

I-Beam, C channel and Hollow Rectangle

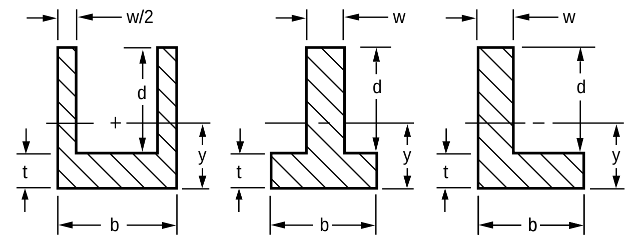


$A = 2bt + wd \qquad I = \frac{b(d+2t)^3}{12} - \frac{(b-w)d^3}{12}$

$y = \frac{d}{2} + t \qquad r = \left(\frac{I}{A}\right)^{1/2}$

Figure III-12

U channel, T section and L section

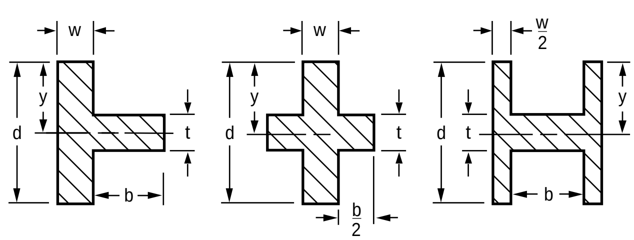


$A = tb + wd \qquad I = \frac{b}{3}(d+t)^3 - \frac{d^3}{3}(b-w) - 9Ad + t - y)^2$

$y = \frac{bt^2 + wd(2t+d)}{2(tb+wd)} \qquad r = \left(\frac{I}{A}\right)^{1/2}$

Figure III-13

Side T section, cross-section and H section



$A = wd \qquad I = \frac{wd^3 + bt^3}{12}$

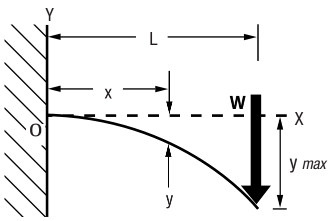
$y = \frac{d}{2} \qquad r = \left(\frac{I}{A}\right)^{1/2}$

Figure III-14

Formulas for Common Beams in Bending

The following equations can be utilized to determine the maximum moment, M_{max}; displacement at a point, y; maximum displacement, y_{max}; and maximum stress, σ_{max}, of many commonly used beam structures; c = distance from centroid of cross-section (Figures III-15 through III-22).

Cantilever Beam End Load



$$M_{max} = WL$$

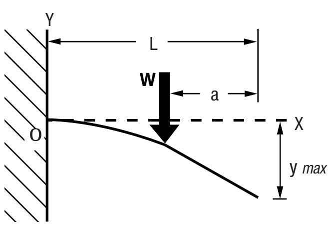
$$y = \frac{-Wx^2(3L - x)}{6EI}$$

$$y_{max} = \frac{-WL^3}{3EI}$$

$$\sigma_{max} = \frac{WLc}{I} \ @ \ x = 0$$

Figure III-15

Cantilever Beam, Intermediate Load



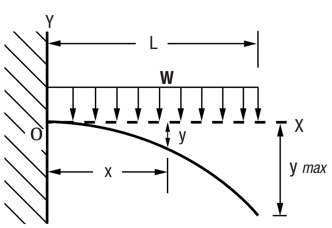
$$M_{max} = W(L-a)$$

$$y_{max} = \frac{-W(2L^3-3L^2a+a^3)}{6EI}$$

$$\sigma_{max} = \frac{W(L-a)c}{I} \ @ \ x = 0$$

Figure III-16

Cantilever Beam, Fully Distributed, Uniform Load



w = load/unit length

$$M_{max} = \frac{wL^2}{2} \ @ \ x = 0$$

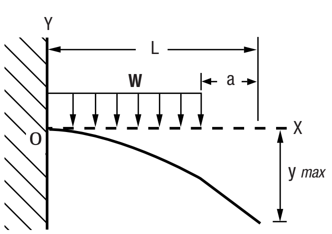
$$y = \frac{-wx^2}{24EI}(6L^2-4Lx+x^2)$$

$$y_{max} = \frac{-wL^4}{8EI} \ @ \ x = 0$$

$$\sigma_{max} = \frac{wL^2c}{2I} \ @ \ x = 0$$

Figure III-17

Cantilever Beam, Partially Distributed, Uniform Load



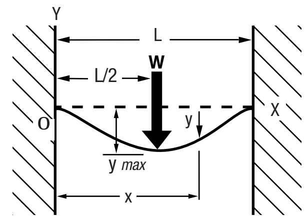
$$M_{max} = \frac{w(L-a)^2}{2} \ @ \ x = 0$$

$$y_{max} = \frac{-w}{24EI}(L-a)^3\,(3L+a)$$

$$\sigma_{max} = \frac{w(L-a)^2c}{2I} \ @ \ x = 0$$

Figure III-18

Fixed Beam, Center Load



$$M_{max} = \frac{WL}{8} \ @ \ x = \frac{L}{2}$$

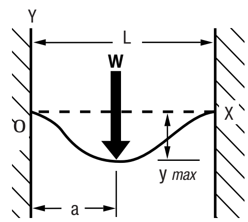
$$y = \frac{-W}{48EI}(3Lx^2-4x^3)$$

$$y_{max} = \frac{-WL^3}{192EI} \ @ \ x = \frac{L}{2}$$

$$\sigma_{max} = \frac{WLc}{8I} \ @ \ x = 0, L$$

Figure III-19

Fixed Beam, Intermediate Load



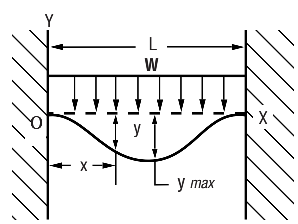
$$M_{max} = \frac{Wa^2(L-a)}{L^2}$$

$$y_{max} = \frac{-2W(L-a)^2a^3}{3EI(L+2a)^2} \ @ \ x = \frac{2aL}{L+2a} \ \text{if } a > L/2$$

$$\sigma_{max} = \underline{-Wa^2(L-a)c}$$

Figure III-20

Fixed Beam, Fully Distributed, Uniform Load



w = load/unit length

$$M_{max} = \frac{wL^2}{12} \ @ \ x = 0,L$$

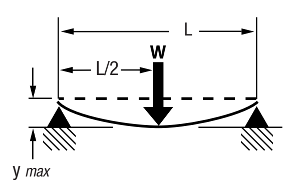
$$y = \frac{wx^2}{24EI}(2Lx-L^2-x^2)$$

$$y_{max} = \frac{-wL^4}{384} \ @ \ x = \frac{L}{2}$$

$$\sigma_{max} = \frac{wL^2c}{12I} \ @ \ x = 0,L$$

Figure III-21

Simply Supported Beam, Center Load



$$M_{max} = \frac{WL}{4} \ @ \ x = \frac{L}{2}$$

$$y = \frac{-W(3L^2x-4x^3)}{48EI} \ \text{for } O \geq x \leq \frac{L}{2}$$

$$y_{max} = \frac{-WL^3}{48EI}$$

$$\sigma_{max} = \frac{WLc}{4I} \ @ \ x = 0, L$$

Figure III-22

Formulas for Torsional Deformation and Stress

By using these formulas on beams of common cross-sections, angle of twist, U, and maximum shear stress, T_{max}, can be calculated. (See Figures III-23 through III-29.)

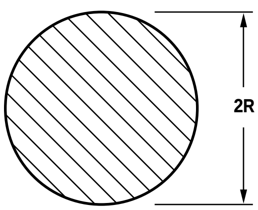
$$\theta = \frac{TL}{KG}$$

Where: T = Twisting moment (force-length)
L = Length of beam
G = Modulus of rigidity (force per unit area)
K = Cross-section dependent function (length^4)

Reference: Roark, Raymond & Young, Warren, *Formulas for Stress and Strain,* McGraw Hill.

Beam in Torsion

Solid Circular Section

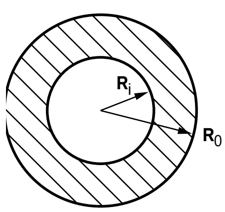


$$K = \frac{1}{2}\pi R^4$$

$$\tau_{max} = \frac{2T}{\pi R^3} \text{ @ boundary}$$

Figure III-23

Hollow Circular Section

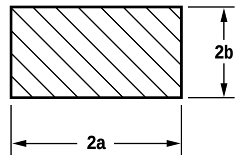


$$K = \frac{1}{2}\pi (R_o^4 - R_i^4)$$

$$\tau_{max} = \frac{2TR_o}{\pi (R_o^4 - R_i^4)} \text{ @ outer boundary}$$

Figure III-24

Solid Rectangular Section

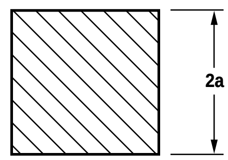


$$K = ab^3\left[\frac{16}{3} - 3.36\frac{b}{a}\left(1 - \frac{b^4}{12a^4}\right)\right] \text{ for } a \geq b$$

$$\tau_{max} = \frac{T(3a + 1.8b)}{8a^2b^2} \text{ @ midpoint of each longer side}$$

Figure III-25

Solid Square Section



$$K = 2.25a^4$$

$$\tau_{max} = \frac{0.601T}{a^3} \text{ @ midpoint of each side}$$

Figure III-26

I, T and L Sections

For sections I, T and L, the maximum shear stress occurs where the largest inscribed circle, D, touches the boundary. A=cross-sectional area.

$$K_1 = ab^3\left[\frac{1}{3} - 0.21\frac{b}{a}\left(1 - \frac{b^4}{12a^4}\right)\right]$$

$$\tau_{max} = \frac{Tc}{K}$$

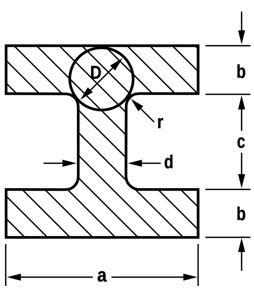
$$c = \frac{D}{1 + \frac{\pi^2 D^4}{16A^2}}\left\{1 + 0.762\left[0.118\log_e\left(1 + \frac{D}{2r}\right) + 0.238\frac{D}{2r}\right]\right\} \text{where D touches radius r}$$

$$c = \frac{D}{1 + \frac{\pi^2 D^4}{16A^2}}\left[1 + 0.15\left(\frac{\pi^2 D^4}{16A^2}\right)\right] \text{ where D touches flat surface}$$

*If b < d then t = b and t_1 = d

If b > d then t = d and t_1 = b

I Section



$$K = 2K_1 + K_2 + 2\alpha D^4$$

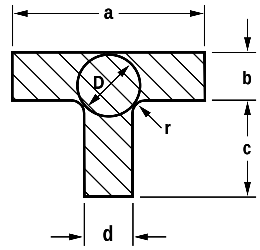
$$K_2 = \frac{1}{3}cd^3$$

$$\alpha = \frac{t}{t_1}\left(0.15 + \frac{0.1r}{b}\right)$$

$$D = \frac{(b+r)^2 + rd + \frac{d^2}{4}}{2r + b}$$

Figure III-27

T Section



$$K = K_1 + K_2 + \alpha D^4$$

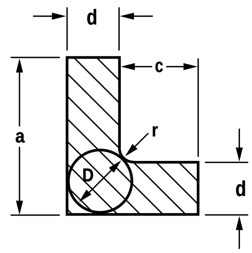
$$K_2 = cd^3\left[\frac{1}{3} - \frac{0.105d}{c}\left(1 - \frac{d^4}{192c^4}\right)\right]$$

$$\alpha = \frac{t}{t_1}\left(0.15 + \frac{0.10r}{b}\right)$$

$$D = \frac{(b+r)^2 + rd + \frac{d^2}{4}}{(2r+b)} \text{ for } d < 2(b+r)$$

Figure III-28

L Section



$$K = K_1 + K_2 + \alpha D^4$$

$$K_2 = cd^3\left[\frac{1}{3} - \frac{0.105d}{c}\left(1 - \frac{d^4}{192c^4}\right)\right]$$

$$\alpha = \frac{d}{b}\left(0.07 + \frac{0.076r}{b}\right)$$

$$D = 2\left(d + b + 3r - \sqrt{2(2r+b)(2r+d)}\right) \text{ for } b < 2(d+r)$$

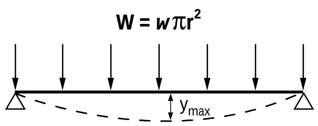
Figure III-29

Formulas for Flat Plates

Both circular and rectangular plates with constant thickness may use these formulas to determine maximum displacement, y_{max} and maximum bending stress, σ_{max}, of a plate under uniform loading. ν is Poisson's Ratio (See Figures III-30 through III-33).

Flat Plate Equations

Circular Disk, Edge Supported, Uniform Load



w = load/unit area

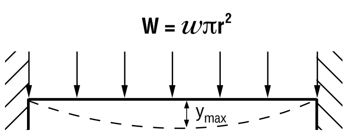
$$m = \frac{1}{\nu}$$

$$y_{max} = \frac{-3W(m-1)(5m+1)r^2}{16\pi E m^2 t^3} \text{ @ center}$$

$$\sigma_{max} = \frac{-3W(3m+1)}{8\pi m t^2} \text{ @ center}$$

Figure III-30

Circular Disk, Fully Fixed, Uniform Load



w = load/unit area

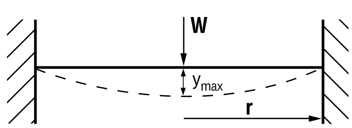
$$y_{max} = \frac{-3W(m^2-1)r^2}{16\pi E m^2 t^3} \text{ @ center}$$

$$\sigma_{max} = \frac{3W}{4\pi t^2} \text{ @ edge}$$

$$m = \frac{1}{\nu}$$

Figure III-31a

Circular Disk, Fully Fixed, Concentrated Load at Center



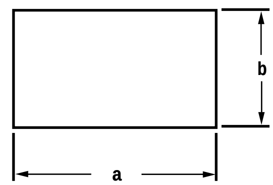
$$y_{max} = \frac{-3W(m^2-1)r^2}{4\pi E m^2 t^3} \text{ @ center}$$

radial stress

$$\sigma = \frac{0.01261W(m-1)}{mr^2} - \frac{3W}{2\pi t^2} \text{ @ edge}$$

Figure III-31b

Rectangular Plate, Uniform Load, Simply Supported



$$y_{max} = \frac{\alpha w b^4}{E t^3} \text{ @ center}$$

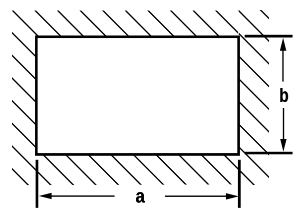
$$\sigma_{max} = \frac{\beta w b^2}{t^2} \text{ @ centers of long edges}$$

a/b	1	1.2	1.4	1.6	1.8
β	0.2874	0.3762	0.4530	0.5172	0.5688
α	0.0444	0.0616	0.0770	0.0906	0.1017

a/b	2	3	4	5	∞
β	0.6102	0.7134	0.7410	0.7476	0.750
α	0.1110	0.1335	0.1400	0.1417	0.1421

Figure III-32

Rectangular Plate, Uniform Load, Fully Fixed



w = load/unit area

$$y_{max} = \frac{\alpha w b^4}{E t^3} \text{ @ center}$$

$$\sigma_{max} = \frac{\beta w b^2}{t^2} \text{ @ center}$$

a/b	1	1.2	1.4	1.6	1.8
β	0.2874	0.3762	0.4530	0.5172	0.5688
α	0.0444	0.0616	0.0770	0.0906	0.1017

Figure III-33

Pressure Vessels

Pressure vessels, containers, or tanks can be analyzed by the use of shell theory because of their shell-like shape and symmetrical loading. To distinguish between thick and thin wall shell or cylinders, the relationship of the wall thickness (t) to the radius (r) must be considered:

- If 10t is < r, the thin wall theory applies.
- If 10t is > r, the thick wall theory applies.

In dealing with pressure vessels, only those vessels having internal pressure resulting in a tensile failure will be addressed in this manual. External pressure resulting in buckling failure is not covered here because it seldom occurs in practice. The equations are as follows:

Internal pressure: (See Figure III-34)

Thin wall
Maximum stress (hoop or circumferential) is:

$$\sigma = \frac{Pr_i}{t}$$

where P is the internal pressure, r_i is the inner radius, and t is the wall thickness.

Thick wall
Maximum stress is:

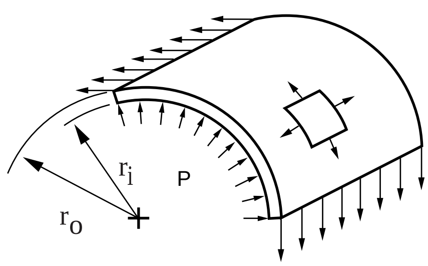
$$\sigma = \frac{P(r_o^{\,2} + r_i^{\,2})}{(r_o^{\,2} - r_i^{\,2})}$$

where r_o is the outside radius, r_i is the inside radius, and P is the internal pressure.

Shells or Curved Surfaces

•Thick Wall: Radial and Hoop (Tensile) Stresses

•Thin Wall: Hoop Stress only



`

The critical or the highest-stressed area of a pressure vessel is the knuckle or transition section, located at the juncture between the end cover and the shell or body of the vessel (see Figure III-35).

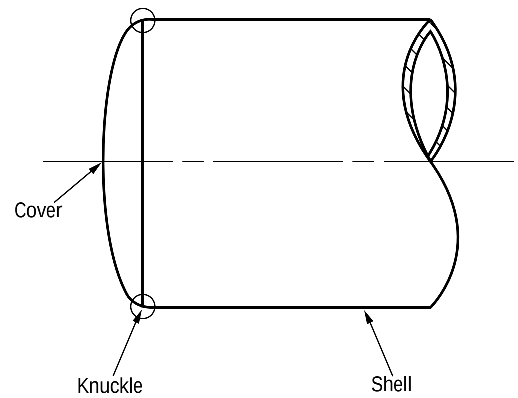


Figure III-35

High localized stresses at the knuckle section are caused by stress risers. (These can be linked to the effects of stress concentration factors. See section III-12.) The more abrupt the change, the higher the stress will be. As a result, a spherical shaped end cover is best.

Pressure Vessels *(cont.)*

Example:

Design a cylinder container to withstand an internal pressure of 50 psi, if the diameter is to be 12in. Material has been selected to be Ultramid® 8233 (33% glass reinforced nylon 6). The wall thickness, for molding convenience, has been set at 0.250 in maximum.

Test for wall condition: Thin or Thick?

$$10t$$
$$(10)(.250in) = 2.5in$$
$$r = 12/2 = 6in$$
$$2.5\ in < 6in$$

Therefore, thin wall approach is acceptable.

$$\sigma = \frac{Pr}{t} = \frac{(50psi)(6in)}{(.250in)} = 1200psi$$

Since we are dealing with long-term effects, we need to compare the 1,200 psi against the allowable tensile strength of the material to determine if this design is satisfactory. Fortunately, this 1,200 psi is well below the allowable tensile strength of the material (18,400 psi @ 50% RH), therefore, the design is satisfactory for short term applications. One must check acceptability for long term conditions, where applicable.

NOTE: for a spherical shape the stress is:

$$\sigma = \frac{Pr}{2t} = \frac{(50psi)(6in)}{2(.250in)} = 600psi$$

This demonstrates that spherical is a good shape for the end cap.

Thermal Expansion and Stress

Thermal stresses are typically not of significant concern except in the case of dissimilar materials in an assembly which sees temperature variations. This occurs when a thermoplastic part is fixed to metal.

If the plastic is clamped to the metal, then the material with the greater expansion would tend to buckle from the resultant compressive loading. To eliminate this potential, the allowable stress of the larger expanding material must be less then the compressive stress developed due to expansion. Euler's critical buckling load (P_e) equation allows one to calculate the critical compressive stress.

$$P_e = \frac{4p^2\ EI}{L^2}$$

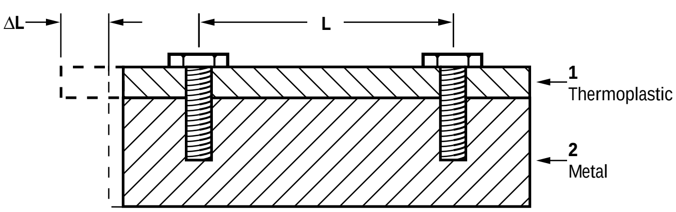


Figure III-36

$\sigma_{cc} = P_e / A_1$

σ_{cc} = critical compressive stress
A_1 = cross-section area of material 1
I = moment of inertia of material 1
E = modulus of elasticity

The following equation calculates the thermal expansion difference between two dissimilar materials (1 & 2).

$\Delta L = (\alpha_1 - \alpha_2)\Delta TL$
where α_1 = coefficient of thermal expansion of material 1
α_2 = coefficient of thermal expansion of material 2
ΔL = change of length
ΔT = change in temperature
L = length between fixed points

To calculate thermal stress, use the following equation:

$$\sigma_c = (\alpha_1 - \alpha_2)\, E\Delta T$$

or increase the section modulus for the larger expanding material.

If the $\sigma_c > \sigma_{cc}$, buckling will occur. To avoid this potential: 1) add additional bolts, 2) increase the section modulus (I/c) of material 1, or 3) provide clearance between bolts and holes for expected movement.

Impact Stresses

An impact situation results when the loading of the part occurs over a very short time frame. When designing for impact, certain concerns should be considered. One important factor is to minimize stress concentrations. Various places in a part, such as holes, notches, grooves, depressions, sharp corners, ribs and bosses, can create high stress concentrations and induce impact failures.

Modify the part, where possible, to reduce and spread the stress over a larger area.

One method to predict impact stresses, deflections or strains is to calculate the static deflection of the part. This information can then be used to calculate an amplification factor. By multiplying the static deflection, stress or strain by the amplification factor, an approximation of the dynamic, deflection, stress or strain can be determined. The amplification factor is as follows:

Amplification factor:

$$K_D = 1 + \sqrt{1 + \frac{2h}{y_{static}}}$$

where h = height of drop
y = static deflection

Example:
The following simulates the impact of a 1.2 lb. load from a height of 4 feet on the center of a fully fixed 1/4 in. thick, 3in circular thermoplastic disk.

ν = Poisson's Ratio

$\nu = .35 \quad E = 1{,}300{,}000\nu$
$m = 1/\nu$

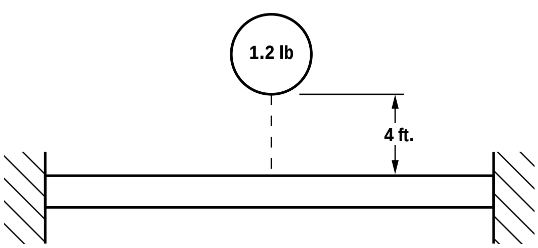


Figure III-37

First determine the static deflection and stress by using the formulas for flat plates, shown in Figure III-31b.

$$y_{static} = \frac{-3W(m^2-1)r^2}{4\pi E m^2 t^3}$$
$$= \frac{-3(1.2)\left(\left(\frac{1}{0.35}\right)^2 - 1\right)1.5^2}{4\pi 1{,}300{,}000\left(\frac{1}{0.35}\right)^2(0.25)^3}$$
$$= -2.7846 \text{ x } 10^{-5}\ in.$$

radial stress @ edge:

$$\sigma_{static} = \frac{0.01261W(m-1)}{mr^2} - \frac{3W}{2\pi t^2}$$
$$= \frac{0.01261(1.2)\left(\frac{1}{0.35} - 1\right)}{\frac{1}{0.35}(1.5)^2} - \frac{3(1.2)}{2\pi(0.25)^2}$$
$$= -9.163\ psi$$

Next, calculate the amplification factor:

$$K_D = 1 + \sqrt{1 + \frac{2h}{y_{static}}}$$
$$= 1 + \sqrt{1 + \frac{2(-48)}{-2.7846x10^{-5}}}$$
$$= 1858$$

Then, the predicted dynamic deflection and stress are as follows:

$$y_{dynamic} = K_D\, y_{static}$$
$$= (1858)(-2.7846 \text{ x } 10^{-5})$$
$$= -0.052in\ @\ center$$
$$\sigma_{dynamic} = K_D \sigma_{static}$$
$$= 1858\ (-9.163)$$
$$= -17025\ psi\ @\ edge$$

Please note that this method is an approximation and generally will be conservative. All calculations should be verified by experimental testing.

Stress Concentrations

Irregularities in a structure subjected to loading may produce high localized stress, or stress concentration (see Figure III-38). These irregularities or stress risers include holes, sharp corners, notches, abrupt changes in wall thickness, or numerous other geometric discontinuities.

Stress Concentration = Stress Riser

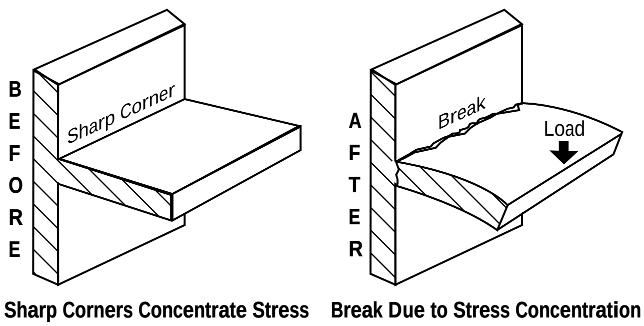


Figure III-38

In many instances it is difficult to accurately compute the actual stress, but, good information does exist which provides for a reasonable estimate. Figure III-39 shows a graph for a given configuration. When the corner radius is small compared to the wall thickness, a high stress concentration factor results.

Stress Concentration Factor

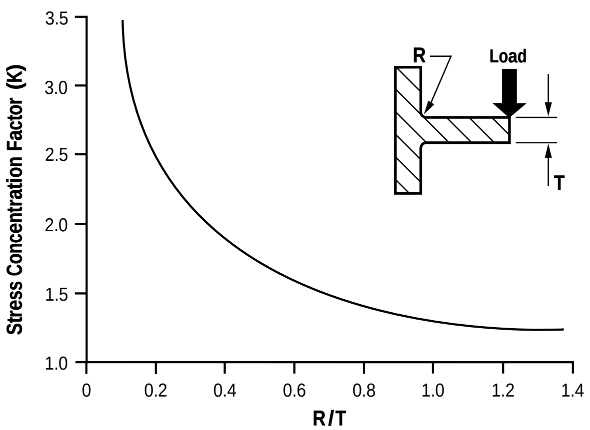


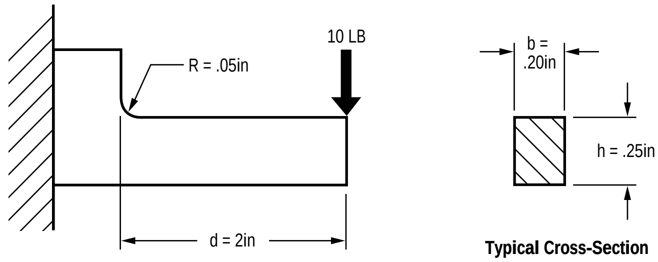
Figure III-39

Illustration credit: Peterson, R.E., Stress Contribution Factors.

The actual stress is now predicted by simply multiplying the calculated bending stress by the K factor.

Example:

Cantilever Beam



$$M = Fd = (10lb)(2in) = 20in\text{–}lb$$

$$C = \frac{h}{2} = 0.125in$$

$$I = \frac{bh^3}{12} = \frac{(.20in)(.25in)^3}{12} = 2.6 \times 10^{-4} in^4$$

Wrong Way

$$\sigma = \frac{Mc}{I} = \frac{(20in\text{–}lb)(.125in)}{2.6 \times 10^{-4} in^4}$$

$$\sigma = 9600 \text{ psi}$$

Right Way

$$\sigma = K\frac{Mc}{I}$$

$$\frac{R}{t} = \frac{.05in}{.25in} = .2$$

$$K = 2.5$$

$$\sigma = 2.5(9600 \text{ psi})$$

$$\sigma = 24{,}000 \text{ psi}$$

The design and load shown should be expected to experience a stress of 24,000 psi. Note that changing the radius to .25 in. changes the expected stress concentration to 12,500 psi.

Rib Design

When designing a part, it is often necessary to determine the number of ribs needed to produce an equivalent displacement or stress based on an un-ribbed part of different thickness. An example is the conversion of an aluminum/steel part into thermoplastic. It is typically required that the thermoplastic provide equal or better stiffness and strength as the metal part.

The following method can be used to determine the number of required ribs. It also results in minimizing the mass of the part without compromising performance or manufacturability.

Draft angle (in degrees) = 1/2°

Base rib thickness to wall thickness (T/W) = .75

All values are per unit of plate wall thickness, W

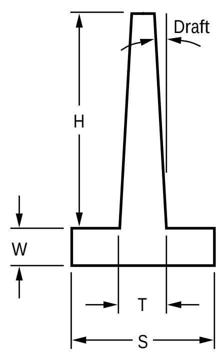
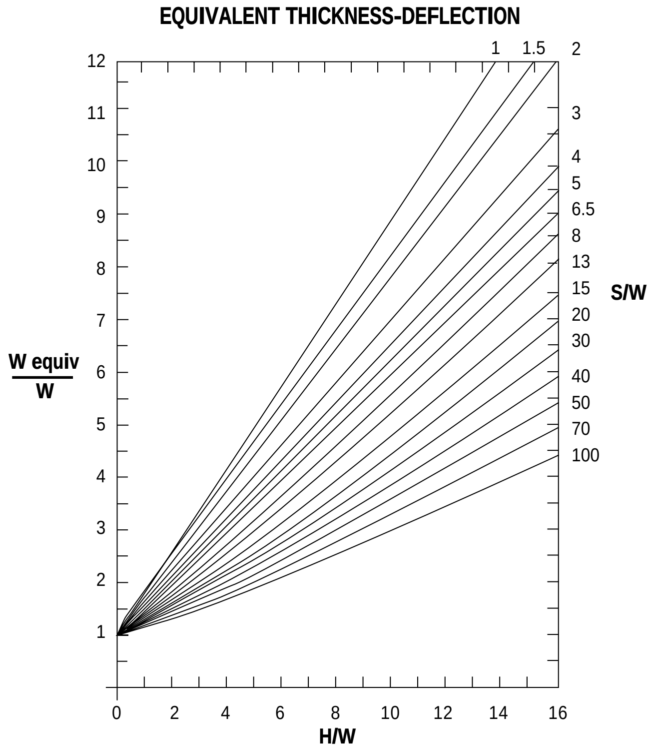


Figure III-40. Ribbed Plate With Draft

Where W = Wall thickness
T = Base rib thickness
S = Distance between ribs
H = Height of rib

Figure III-40 shows the geometric parameters used in this method. The following curves (Figure III-41) have been generated for a plate with ribs having 1/2° draft per side and a T/W = .75. There are numerous other similar curves for other variations.



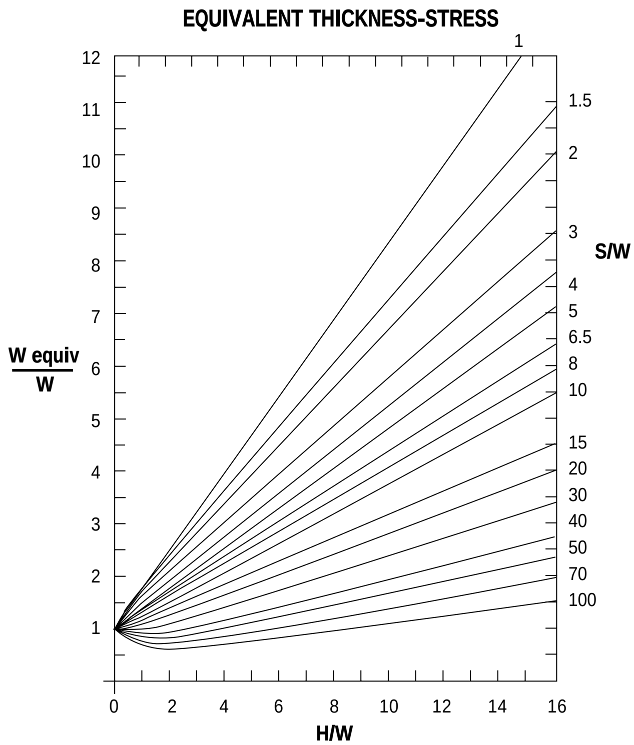


Figure III-41. Equivalent Thickness Charts

To use these charts a few ratios need to be calculated. To convert an aluminum part to thermoplastic of equal rigidity, it is necessary to calculate the equivalent thickness of the plastic part without ribs. Next, a nominal thermoplastic wall thickness must be selected along with either the rib height or number of ribs.
Using the curves, the appropriate rib pattern can be determined. This process can be reversed as well if there is a desire to determine displacement or stress for a ribbed member. It basically reduces the ribbed member to a simple flat plate.

Example:
The following shows the conversion of a flat aluminum plate of 6 in. x 10 in. with a thickness of 0.125 in. to a typical glass reinforced nylon plate with ribs and having equivalent stiffness. For flat plates of equal rigidity, the following ratio is valid: (see page III-15.)

$$E_A t_A^3 = E_P t_P^3$$

(see page III-18)

knowing:

$$E_{aluminum} = 1.0x10^7 psi$$

$$E_{plastic} = 5.0x10^5 psi$$

then

$$t_P = \left(\frac{E_A t_A^3}{E_P}\right)^{\frac{1}{3}}$$

$$= \left(\frac{1.0 \ x \ 10^7 (0.125)^3}{5.0x10^5}\right)^{\frac{1}{3}}$$

$$t_P = W_{equiv} = 0.339in$$

This is the thickness that the thermoplastic part would need to be if no ribs were present. Since this wall thickness is thicker than desirable for injection molding, the addition of ribs is an alternative.

We must now choose a value for two of the following: nominal wall thickness (W), rib height (H), or distance between the ribs (S). For this exercise, let us make W = 0.125in and H = 0.725in.

Therefore:

$$\frac{W_{equiv}}{W} = \frac{0.339}{0.125} = 2.712$$

$$\frac{H}{W} = \frac{0.725}{0.125} = 5.8$$

$$\frac{T}{W} = 0.75 \therefore T = (0.125)0.75$$

$$= 0.094in$$

Since we are interested in equivalent deflection, we can find the curve that corresponds to these 2 ratios and find that S/W = 20, therefore S = (0.125)20 = 2.5in. This means that for a plate of 6 in. x 10 in.we will need 3 and 4 ribs respectively. The equivalent plate would look like Figure III-42.

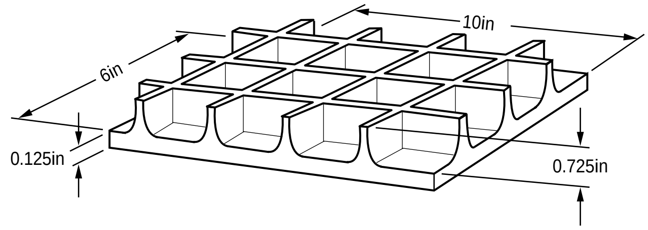


Figure III-42. Equivalent Plate Using Cross-Ribbing

NOTE: To lower stresses in the plate, it is recommended that generous radii be placed at the base of the ribs.

Table III-1 further illustrates the weight-to-stiffness advantage of various rib heights.

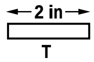
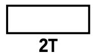
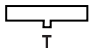
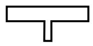
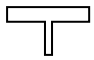
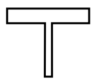
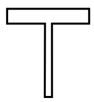
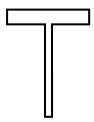
Effect of 1/8in Thick Rib of Various Heights on the Strength of a 2in x 1/4in Beam					
Case Number	Shape	Rib Size	Rib Height/ Wall Thickness	% Increase in Weight	% Increase in Stiffness
0		N/A	N/A	N/A	N/A
1		N/A	N/A	100	700
2		1/8in W x 1/8in H	1:2	3.12	23
3		1/8in W x 1/4in H	1:1	6.25	77
4		1/8in W x 1/2in H	2:1	12.5	349
5		1/8in W x 3/4in H	3:1	19.0	925
6		1/8in W x 1in H	4:1	25.0	1901
7		1/8in W x 1 1/4in H	5:1	31.0	3352
T = Thickness = 1/4in					

Table III-1

Design for Equivalent Stiffness

In order to replace metal parts with plastic, the equivalent stiffness of a plastic part can be determined. When the two parts are of equivalent stiffness, deflection is the same. Deflection is inversely proportional to the rigidity modulus (R):

$$R = EI$$

where E is the modulus of elasticity and I is the moment of inertia. (The moment of inertia will vary for each geometry. See Figure III-43.) Therefore, by equating the modulus of rigidity of the metal and plastic parts the condition of equivalent stiffness will be satisfied.

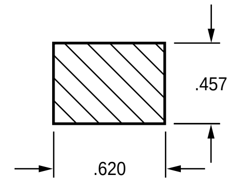
$$E_{aluminum} \quad I_{aluminum} \quad = \quad E_{plastic} \quad I_{plastic}$$

For solid shape of equal width;

$$E_{aluminum} \quad h^3_{aluminum} \quad = \quad E_{plastic} \quad h^3_{plastic}$$

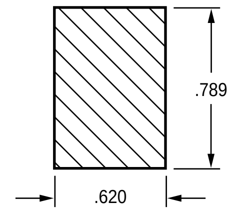
Sections of Equivalent Stiffness in Bending:

Aluminum



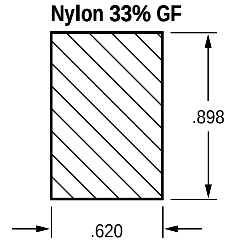
E = 10.3 x 10^6 psi
I = 0.0049in^4
EI = 5.08 x 10^4 lb–in^2
A = 0.283in^2

Zinc



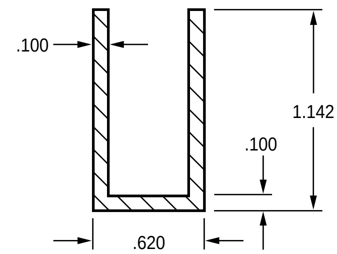
E = 2.0 x 10^6 psi
I = 0.0254in^4
EI = 5.08 x 10^4 lb–in^2
A = 0.489in^2

Nylon 33% GF



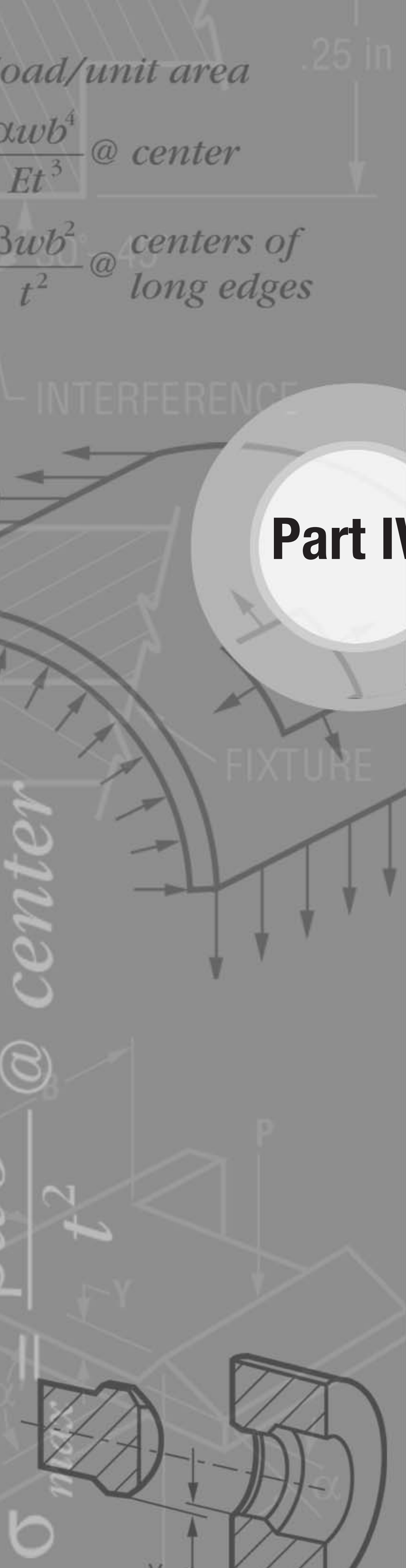
E = 1.36 x 10^6 psi DAM
I = 0.0374in^4
EI = 5.08 x 10^4 lb–in^2
A = 0.557in^2

Nylon 33% GF



E = 1.36 x 10^6 psi DAM
I = 0.0374in^4
EI = 5.08 x 10^4 lb–in^2
A = 0.270in^2

Figure III-43



Part IV

Design Examples

Part IV: Design Examples

Design Example #1

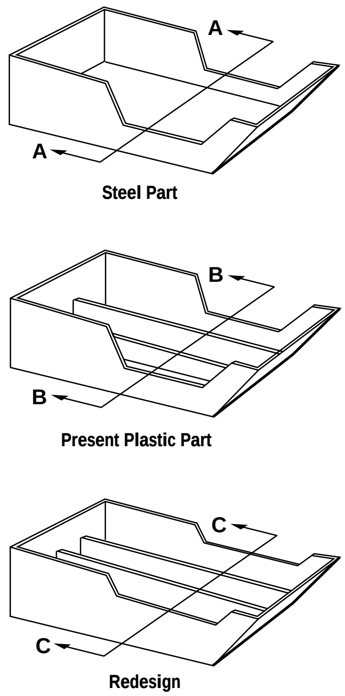


Figure IV-1. Cruise Control Bracket

Application: CRUISE CONTROL BRACKET

Problem: Plastic Bracket Bending Under Load After Conversion From Steel

Potential Reasons for Part Failure:
a) Material
b) Processing
c) Design

Analysis: Simple Cantilever Beam (Closely similar to the end-use condition). (Note actual outlined part.)

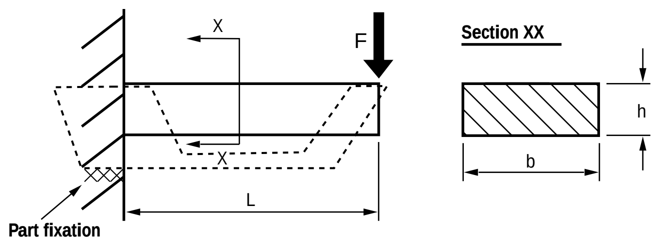


Figure IV-2. Rigidity Modulus, R = EI

E = Modulus of Elasticity I = Moment of Inertia
R can be increased by increasing E or I

NOTE: The small section was analyzed because the left portion of the part, which is basically clamped, has a large section modulus and is therefore much stiffer than the middle of the part.

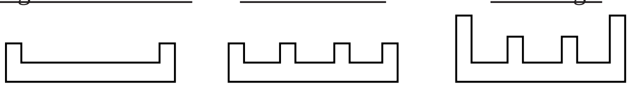
Since $I = bh^3/12$, a small change in h will result in a cubed effect or a large increase in R, a very effective change.

Example: If h is doubled, it will increase the R by a factor of 8!!!

To make the plastic part more rigid than the steel part, $EI_{plastic} \geqq EI_{steel}$ (lb. in^2)

If the E for plastic is 740,000 (Ultramid·8233) @ 50% RH, and for steel 30,000,000 psi, then the results would be as noted:

Section AA	Section BB	Section CC
	Plastic	
Original Steel Part	Present Part	Redesign



I = .0002	I = .0008	I = .041
EI = 6,000	EI = 592	EI = 30,340

Figure IV-3

Conclusion:

a) A material change would not be effective enough; it could increase the cost and require new approvals.

b) Processing was not determined to be the problem by lab analysis.

c) Redesign was implemented with successful results.

Design Example #2

Application: COVER CAP

Problem: Oversize Parts Out of Specification

Potential Reason for Part Failure:

a) Material, or
b) Processing

Customer Input:

1) The parts were initially inspected, approved by Q. C. and placed into stock.

2) Five months passed, when a reorder was issued.

3) Parts were .0045in oversize and rejected by the customer.

Analysis:
Parts were in Ultramid® 8233. The critical dimension, and the one in question, is 2.002 in. in diameter. It was assumed that the problem could well be the growth of the part due to moisture absorption.

If the parts were measured soon after molding, the parts were in the DAM (dry as molded) condition. Using the plot shown for part growth versus moisture absorption or percent relative humidity, it can be seen that the growth of the part at 50% RH is .0025in/in. This is a worldwide average which should be used in all calculations for critical dimension determinations. The formula for the circumference of a round part is:

$$C = 2 \times \pi \times r$$
or
$$C = \pi \times D$$

Where:

r = radius
D = diameter

If we multiply the 2.002in diameter by .0025in/in, a .005in part growth results. This was the amount the parts in stock were oversize and therefore the problem.

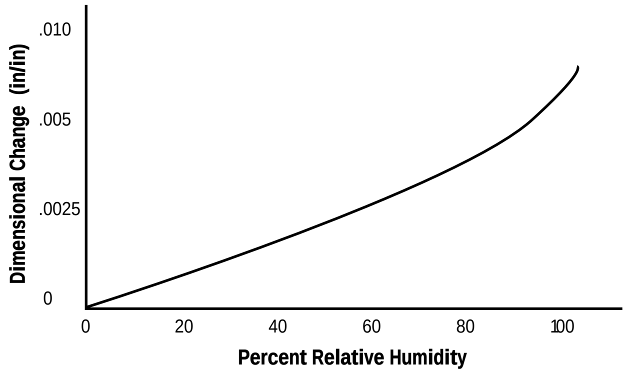


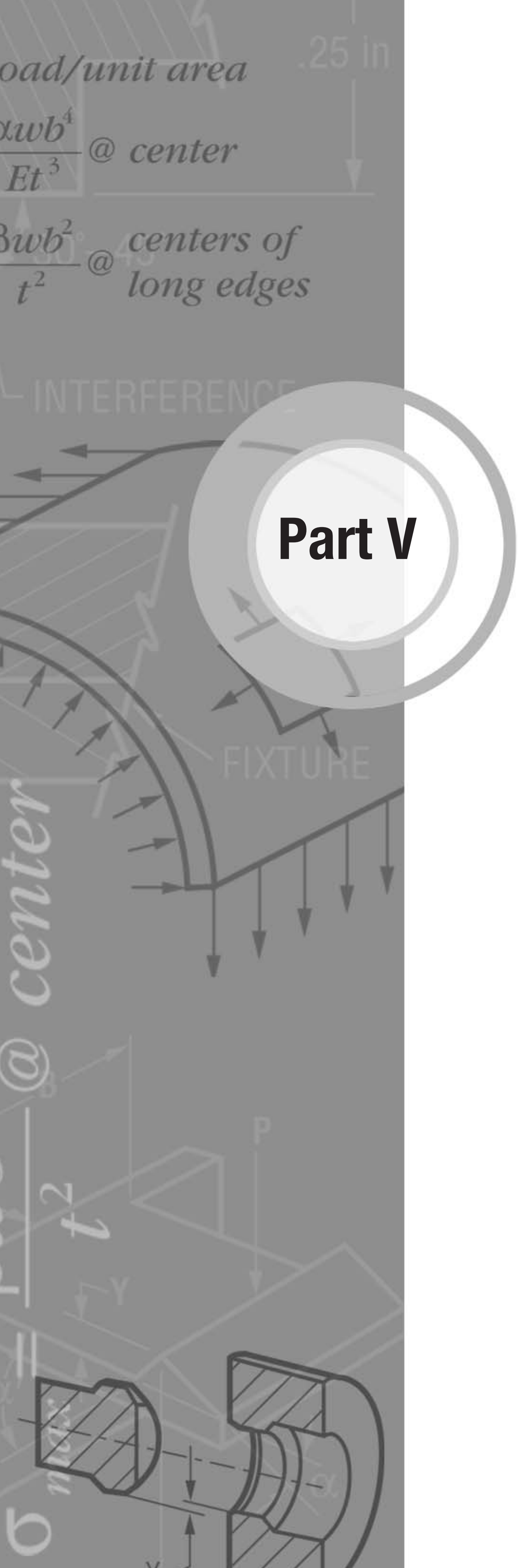
Figure IV-4

Conclusion:

a) A concern only when very tight tolerances are important, as this was.

b) Can be corrected by preparing the mold for the anticipated long term size and 50% RH conditions.

c) Parts can be conditioned initially to stabilized dimensions.



Part V

Assembly

Part V: Assembly

Snap-Fit Assembly

Snap-Fit Design

When assembling two parts, snap-fits are simple, cost-effective, and a quick method of assembly. When designed correctly, snap-fits can be assembled and disassembled many times without any adverse affect on the assembly. Snap-fits are also one of the more environmentally-friendly form of assembly because of their easy disassembly.

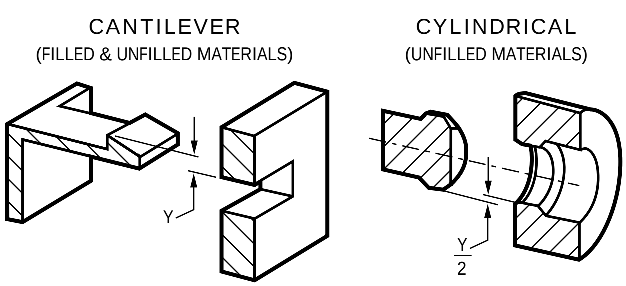
The designer should be aware that snap-fits do have some limitations. These include a possible clearance condition due to the tolerance stack-up of the two mating parts, and low pullaway forces. Snap-fits can also increase the cost of an injection molding tool, if slides are needed in the mold. The designer can eliminate the need for slides by adding a slot directly underneath the snap ledge or by placing the snap at the outside edge of the part.

Cantilever vs. Cylindrical

Most applications use the cantilever snap-fit design (see Figure V-1). The cylindrical design can be employed when an unfilled thermoplastic material is selected (a typical application is an Aspirin bottle/cap assembly).

When designing a cantilever snap, the designer may have to go through several iterations (changing length, thickness, deflection dimensions, etc.) to design a snap-fit which results in a strain lower than the allowable strain of the material (Figure V-6).

Figure V-1



Tapered Cantilever

For most applications, the uniform section cantilever (see Figure V-2) is sufficient in designing a snap-fit. A tapered section beam is desirable, if additional deflection is desired.

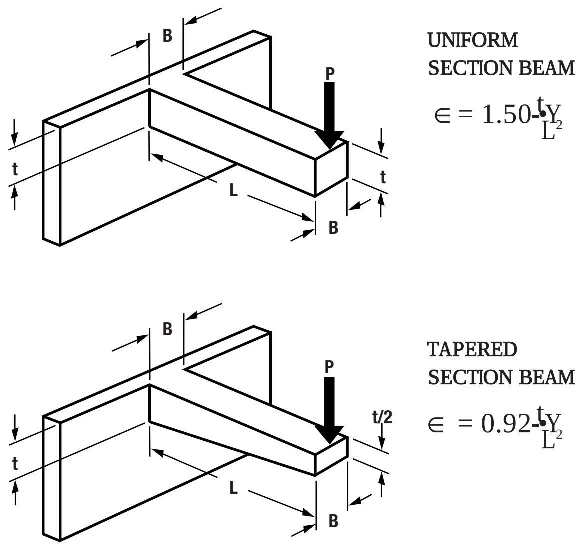


Figure V-2.
Conventional Cantilever Formulas

Short Cantilever Design

The conventional cantilever formulas used in analyzing snap-fit deflections predict a much smaller deflection than observed in the field for short cantilevers. The wall from which the snap protrudes is assumed to be rigid in the conventional formulas. This is a valid assumption for long cantilevers, but not short cantilevers. The intersecting wall actually deforms under load for short beams.

BASF has proven[1] this both experimentally and by Finite Element Analysis. The results of this study were compiled and are shown in Figure V-4 for various configurations (see Figure V-3). An example illustrates the procedure for designing a snap-fit. Formulas for calculating maximum strain, deflection and amount of force required to assemble the parts are also given. A special snap-fit manual with more detail is available.

[1] 1987 SPE ANTEC, Chul S. Lee, Alan Dublin and Elmer D. Jones, ‘‘SHORT CANTILEVER BEAM DEFLECTION ANALYSIS APPLIED TO THERMOPLASTIC SNAP-FIT DESIGN,’’ Held in Los Angeles, California, USA.

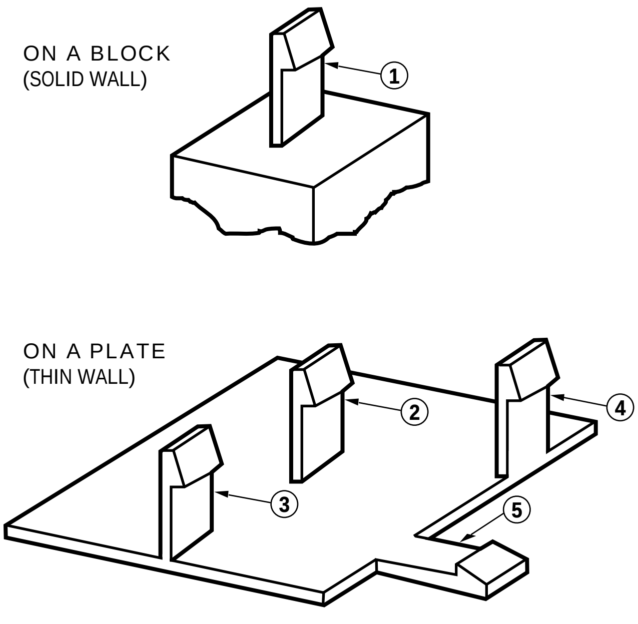


Figure V-3. Beam Configurations

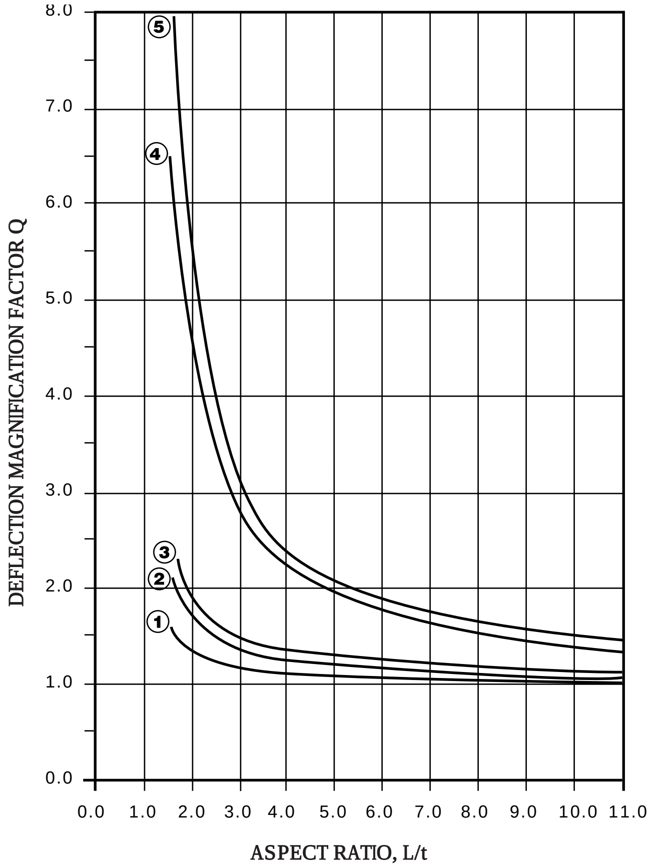
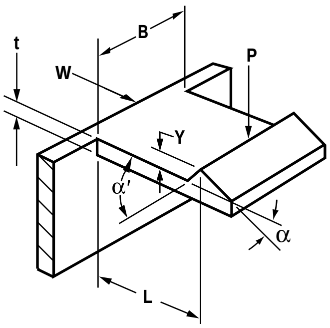


Figure V-4. Q Factor

NOTE: Numbers inside of the circles correspond to the beam configuration cases shown in the preceding figure.

New Formulas



MAXIMUM STRAIN

$$\in_o = 1.5\frac{tY}{L^2Q}$$

Figure V-5

Where:

$\in_o$ = Maximum strain at the base
t = Beam thickness
Y = Deflection
L = Beam length
Q = Deflection magnification factor
(refer to graph for proper Q values)

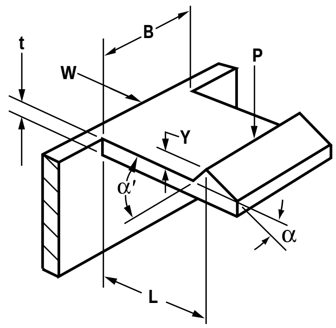
MATING FORCE

$$W = P\frac{\mu + \tan\alpha}{1 - \mu\tan\alpha} \qquad P = \frac{Bt^2 \in E}{6LQ}$$

Where:

W = Push-on force
W' = Pull-off force
P = Perpendicular force
μ = Coefficient of friction
α = Lead angle
α' = Return angle
B = Beam width
t = Beam thickness
E = Flexural modulus
∈ = Strain
L = Beam length
Q = Deflection magnification factor
(refer to graph for proper Q values)

Example:



GIVEN:
Material: PETRA 130
(PET)

t = 0.10in
L = 0.50in
B = 0.25in
E = 1,300,000 psi
μ = 0.2
α = 30.0°
$\in_o$ = 1.5%

Figure V-6

DETERMINE:

a) THE MAXIMUM DEFLECTION OF SNAP
b) THE MATING FORCE

SOLUTION:

a) THE MAXIMUM ALLOWABLE DEFLECTION OF SNAP

$$\in_o = 1.5\frac{tY_{max}}{L^2Q} \Rightarrow Y_{max} = \frac{\in_o L^2 Q}{1.5\,t}$$

$$\frac{L}{t} = 5.0 \Rightarrow Q = 2.0 \text{ (from Q Factor graph)}$$

$$Y_{max} = \frac{(0.015)(0.5)^2\,(2.0)}{(1.5)(0.1)} = 0.05\text{in}$$

Therefore, in an actual design, a smaller value for deflection (Y) would be chosen for an added factor of safety.

b) THE MATING FORCE

$$P = \frac{Bt^2 E\in_o}{6L}$$

$$P = \frac{(0.25)(0.1)^2\,(1{,}300{,}000)\,(0.015)}{6(0.5)} = 16.2\text{lb}$$

$$W = P\frac{\mu + \tan\alpha}{1-\mu\tan\alpha}$$

$$W = 8.1\frac{0.2 + \tan 30}{1 - 0.2\,(\tan 30)} = 14.2\text{lb}$$

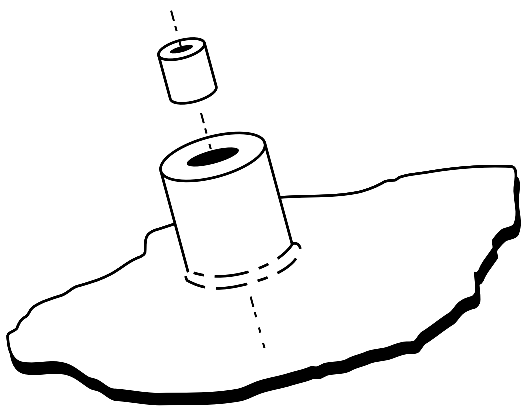
Therefore, it will take 14.2 pounds to force the snap-fit into position.

(More detailed information can be found in the BASF Snap-Fit Design Manual.)

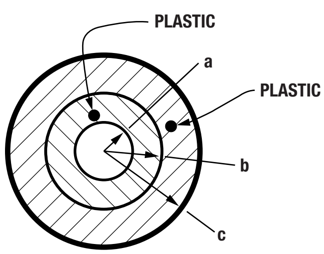
Press-Fit Assembly

Two parts can be assembled by press-fitting them together (see Figure V-7). Although this is a common assembly method in metals, a good design is more critical for thermoplastics. Since plastics creep (or stress relax), the designer must allow for a large reduction of the initial press-fit clamp force. A good design will minimize the strain on the plastic (see Figures V-8 & V-9), take tolerance stack-up into account and estimate the final residual clamp force due to plastic creep relaxation.

Figure V-7



RADIAL DEFORMATION



$$\delta = \frac{2 \in b^3 (c^2 - a^2)}{(c^2 + b^2)(b^2 - a^2)}$$

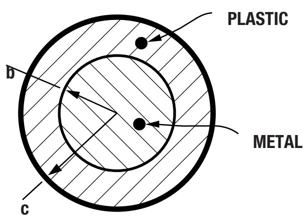
WHERE:
δ = Radial deformation
$\in$ = Strain

WHEN a = 0

$$\delta = \frac{2\in bc^2}{c^2 + b^2}$$

Figure V-8. Press-Fit With Two Identical Materials

NOTE:
1. Radial deformation, δ, must be doubled for the total interference fit on a diameter.
2. a, b and c are radii, not diameters.
3. Formula is only valid when the shaft and hub are the same material.
4. Creep must be fully analyzed.



RADIAL DEFORMATION

$$\delta = b \in \left(\frac{c^2 - b^2}{c^2 + b^2}\right) \left(\frac{c^2 + b^2}{c^2 - b^2} + v_{PL}\right)$$

WHERE:
δ = Radial deformation
$\in$ = Strain
v_{PL} = Poisson's Ratio.

Figure V-9. Press-Fit a Metal Shaft into a Plastic Hub

NOTE:
1. Radial deformation, δ, must be doubled for the total interference fit on a diameter.
2. b and c are radii, not diameters.
3. Formula assumes zero deformation of the metal shaft.
4. Creep must be fully analyzed.

EXAMPLE:

GIVEN: A metal insert (O.D. = 0.50in) is to be press-fit into a Ultramid 8233 boss (O.D. = 0.75in). Determine the maximum interference of the assembly using a 2% allowable strain for Ultramid 8233.

SOLUTION:

$$d = b \in \left(\frac{c^2 - b^2}{c^2 + b^2}\right) \left(\frac{c^2 + b^2}{c^2 - b^2} + v_{PL}\right)$$

$$= (.250)(.02)\left(\frac{.375^2 - .250^2}{.375^2 + .250^2}\right) \left(\frac{.375^2 + .250^2}{.375^2 - .250^2} + .35\right)$$

$$d = .0057\text{in}$$

Therefore, the Capron® 8233 boss I.D. should be designed with an .011in (2δ) maximum interference (.489in minimum diameter).

Adhesive Bonding

Another method for assembling parts is by applying an adhesive. Two similar or dissimilar materials can be assembled together in a strong leak-tight bond. Various joint designs are shown in Figure V-10.

The choice of the adhesive depends on the application and its end-use environment. Details of some adhesives, which can be used with BASF Ultramid® and Petra® products, are highlighted in Table V-1.

Polyurethanes – High strength, good impact resistance, good low temperature flexibility, two parts (usually), limited moisture resistance, long cure times, and usually needs to be fixtured.

Epoxies – High strength, high temperature resistance, two parts (usually), poor impact resistance, long cure times, and usually needs to be fixtured.

Cyanoacrylates (example: Krazy Glue) – High strength, very fast cure time, one part, limited service temperature (about 200°F), poor impact resistance, and limited moisture resistance.

Silicones – Low strength, very high heat resistance, two parts (usually), good low temperature flexibility, good impact resistance, good sealing capability, very long cure times, usually needs to be fixtured, and very high material cost.

*3M, Structural Adhesive Guide for Industrial Product Design and Assembly.

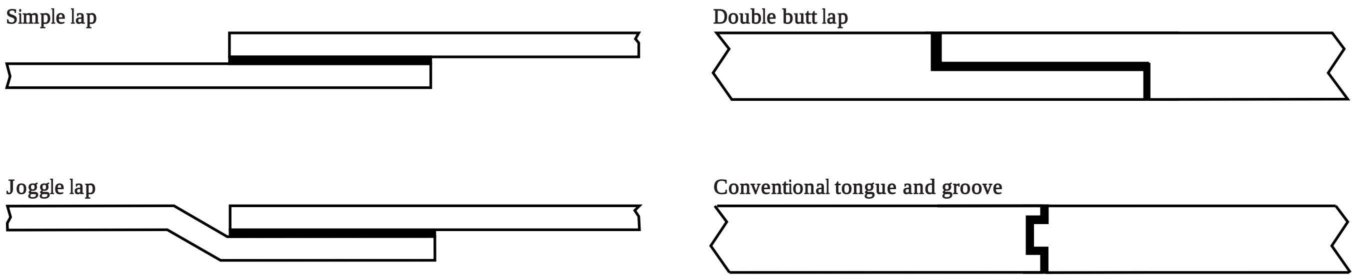


Figure V-10. Joint Design*

ADHESIVE	TYPE	CURE	MANUFACTURER
UR 1100	1 part urethane	30 min @ 250° F (121° C)	HB Fuller 1-800-328-7307
FE 6046 (Flexible Epoxy)	2 part epoxy	60 min @ 200° F (93° C)	HB Fuller 1-800-328-7307
Hysol 934	2 part epoxy	60 min @ 200° F (93° C)	Hysol Aerospace Products (510) 458-8000
Scotch-Weld 2214 Regular	1 part epoxy	40 min @ 250° F (121° C)	3M Company 1-800-362-3550
Scotch-Weld 2214 Hi-Temp	1 part epoxy	40 min @ 250° F (121° C)	3M Company 1-800-362-3550
Scotch-Weld 2216	2 part epoxy	5 min @ 250° F (121° C)	3M Company 1-800-362-3550
Tyrite 5700 A/C	2 part epoxy	15 min @ 200° F (93° C)	Lord Industrial Adhesives (814) 868-3611
Superbonder 498	Cyanoacrylate	30 sec @ 73° F (23° C)	Loctite Corporation 1-800-562-8483
Cylok P	Cyanoacrylate	10 - 30 sec @ 73° F (23° C)	Lord Industrial Adhesives (814) 868-3611
Permabond 268	Cyanoacrylate	10 sec @ 73° F (23° C)	Permabond Int'l 1-800-526-4741
3-0100	Silicone	24 hrs @ 73° F (23° C)	Dow Corning (517) 496-6000
Plexus MA310	2 Part Methacrylate	15-18 min @ 73° F (23° C)	ITW Plexus 1-800-851-6692

Table V-1. Recommended Adhesives for BASF Resins

Bolts, Nuts, and Machine Screws

Standard metal fasteners are also used to assemble thermoplastic components, although self-tapping screws are more common. Bolts and screws are used to join plastic to metal or plastic to plastic. Care must be taken to prevent excessive compressive stress on the plastic.

- Assembly must be limited to a prescribed torque level and controlled. Rapid application of torque should be avoided since most thermoplastics are rate sensitive.
- High torques generally produce high compressive stress. A rapid initial stress reduction takes place before leveling off over time. The higher the stress, the greater the stress relaxation. Elevated temperatures will further increase relaxation.
- A larger head screw or addition of a large diameter metal washer under the bolt head and/or nut will increase the contact area and reduce stress. Figure V-11 illustrates this concept.
- Flat head screws and rivets should be avoided in plastic applications. These conical shaped fasteners cause a wedging action which results in high hoop stress and possible failure of the part.
- As stress relaxation occurs, the clamping force and torque retention drop and the fastener will loosen. A spring washer can be used to maintain acceptable force and torque levels. Figure V-12 shows various options to help counteract stress relaxation. Options 1 & 2 use a shoulder washer or bolt in combination with the spring washer. The main clamping is metal to metal while a smaller force holds the plastic.

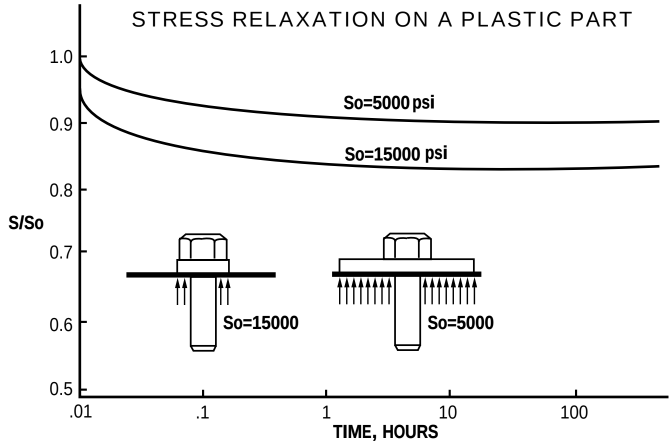


Figure V-11. Application of Stress Relaxation to Plastic Part Design

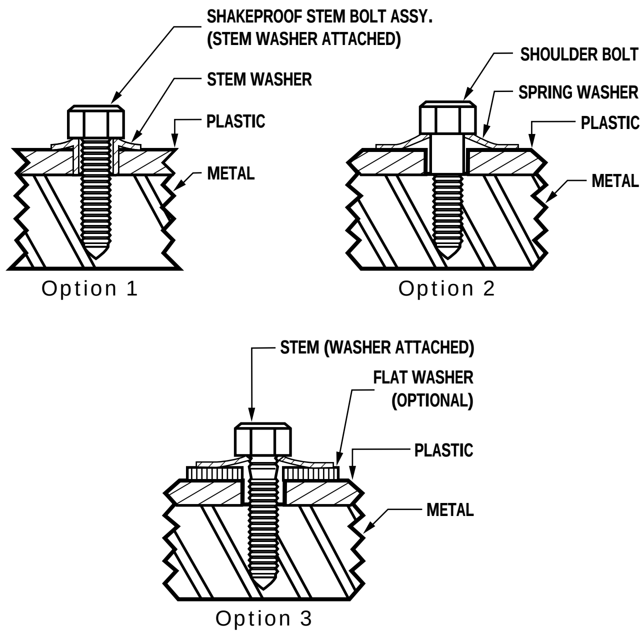


Figure V-12. Minimizing Stress Relaxation

Molded-In Threads

One of the advantages of injection molded parts is the ability to mold in many kinds of functional features such as threads. These can be traditional forms or specially modified versions tailored to specific applications. Coarse threads are generally preferred due to their higher strength and torque limits. For applications requiring high pullout or subject to high pressure loads, Acme• or Buttress• thread forms can be used.

The two main types of threads are external and internal. Both types should be designed with lead-in thread relief. Generally 1/32 in. is sufficient to prevent high stresses at the end of the threads. Also it is good design practice to include radii of .005 in.–.010 in. at the thread roots to minimize stress concentration. Examples of external and internal threads are shown below.

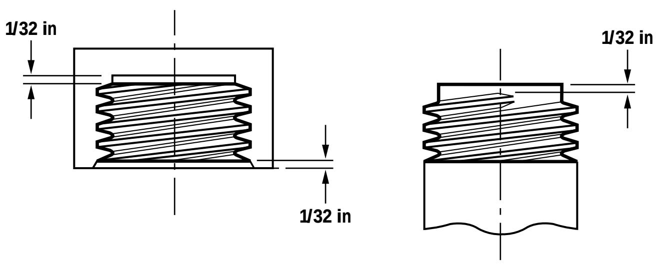


Figure V-13. Molded-In Thread Relief

External threads should be located on the tool parting line, if at all possible, to avoid undercuts and the need for an unscrewing mechanism. This will lower tool cost and lower mold cycle time. Internal threads are usually formed by an unscrewing or collapsible core. If a single thread that is slightly less than 360° around is adequate, then it can be formed using a straight core pull. (See Figure V-14.)

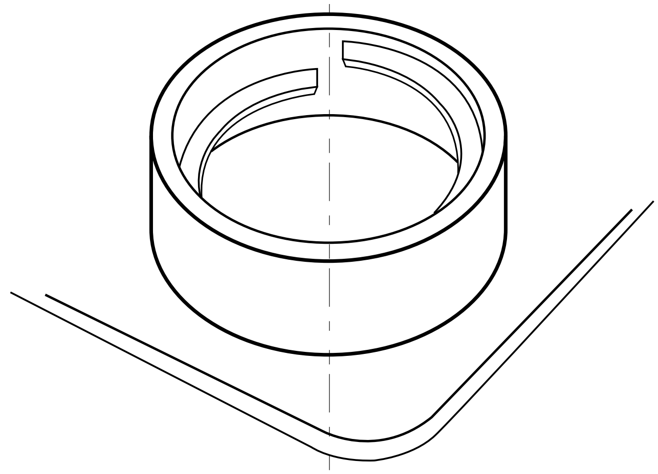


Figure V-14. Single Screw Thread

Tapered Pipe Threads

Special care should be taken when designing internal tapered pipe threads that will be mated with a metal pipe. These threads act as a wedge, causing high hoop stress that may crack the plastic member, if over-tightened. Some means of providing a positive stop should be incorporated such as a shoulder at the bottom of the internal plastic thread.

Self-Tapping Screws

There are two main types of self-tapping fasteners used in plastic parts: thread cutting and thread forming.

Thread cutting screws are generally used only on brittle plastics, such as thermosets and highly filled (+50%) thermoplastics. They cut threads by means of a slotted shank. Because they actually remove material when inserted, thread cutting screws should not be reinstalled and a chip reservoir should be added.

Thread forming screws are generally preferred for most thermoplastic applications. These types of screws can be reinstalled a limited number of times (3-7). For repeated assembly and disassembly, some form of metal inserts should be used. There are several styles of thread forming screws designed specifically for plastics. Three of the more widely used are:

Plastite™
These screws have a trilobular cross-section which roll threads by moving material out of the way as they are installed. After installation, the material fills back around the shank lowering the residual stress in the screw boss. This feature also gives the Plastite• screws excellent resistance to loosening due to vibration. Higher hoop stress is produced with these screws.

Hi-Lo™
These screws feature a dual lead with a high thread having a 30° included angle and the low thread having a 60° angle. These screws have a high strip torque to drive torque ratio which is important for small sizes. Lower hoop stress is produced but higher stress concentrations result due to acute angle threads.

PT™
This thread design has a single 30° included angle which reduces hoop stress in the boss and also provides a high strip torque to drive torque ratio.

Guidelines for self-tapping fasteners:

1. Thread engagement length should be 2.5 times the screw diameter.
2. Boss diameter should be at least 2 times the pilot hole diameter.
3. Pilot hole diameter should be based on 50%-70% thread engagement. This can vary with the material and the type of fastener (check with your Basf design representative).
4. Cored holes should have 1/4° to 1/2° draft/side.
5. Holes should be counterbored or chamfered to a depth of .020 in. to aid alignment and reduce the chance of boss cracking.
6. Strip to drive torque ratio should be at least 3:1, but the difference in strip to drive torque is more important than the ratio.
7. Seating torque should be no more than 2/3 strip torque.

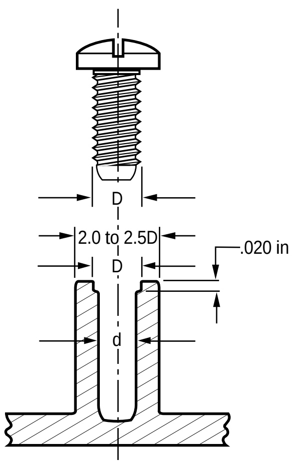


Figure V-15. Bosses for Self-Tapping Screws

Inserts

Inserts of various types are used with plastics. The most common are threaded metal inserts; either internally threaded nuts or externally threaded studs.

Threaded metal inserts are used when the assembly application requires repeated assembly and disassembly or the assembly needs to resist creep and compressive relaxation. There are several methods of installing inserts:

Ultrasonic

This method uses the same equipment as ultrasonic welding. The high frequency horn vibrations cause frictional heat between the insert and plastic, thereby melting it into the boss. This process takes under 5 seconds and features low residual stress and excellent pullout strengths.

Thermal

This is similar to ultrasonics in that the insert is melted into the boss but the insert is heated by a device like a soldering iron. This method is relatively slow and also yields a low stress assembly with good pull out strengths.

Self-tapping

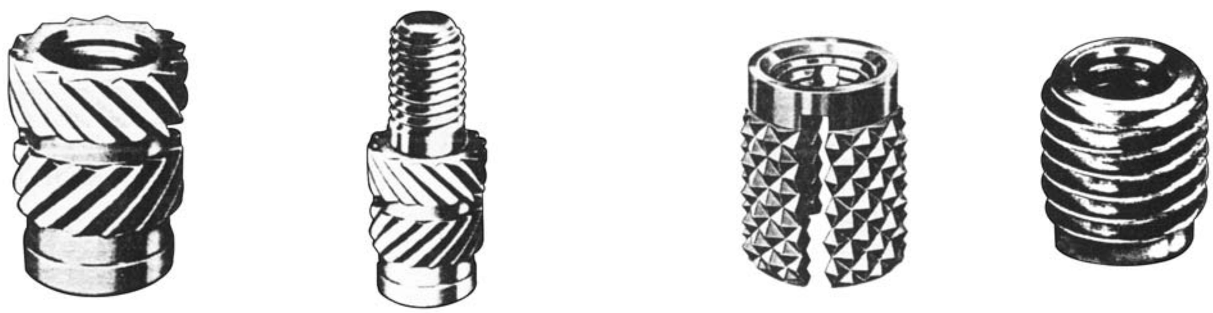
These inserts have an external self-tapping screw thread and are driven into the hole using low cost equipment. (See Figure V-7.)

Press-fit and Expansion

This type of insertion is not normally desirable. The insert is pressed in with an interference fit. The expansion insert is designed to expand into the side walls of the boss with a tool. Both methods impart a high stress to the boss, and they have lower mechanical performance.

Molded-in

This method is often used for large or special inserts. As the name implies, these inserts are placed in the mold cavity and the plastic is injected, thereby encapsulating them. The need to place the insert in the mold increases cycle time and mold damage can occur.



Thermal Insert | Ultrasonic Insert | Expansion Insert | Self-Tapping Insert

Figure V-16. Typical Threaded Metal Inserts

Illustration Credit: Spirol Inserts CEM Corporation Inc. and In-X Fasteners Corp.

Ultrasonic Welding

Ultrasonic welding is a quick and reliable way to assemble the same or very similar thermoplastic parts. Electrical energy is converted into mechanical vibrations causing frictional heat between mating parts, thereby melting the plastic. The parts are held in a fixture under pressure while the ultrasonic energy is applied. The energy is then shut off and the pressure maintained until the weld surface has solidified. Total weld time is generally around 0.5-1.0 second. Standard welder frequencies are 20 kHz, although 40 kHz units are available for small delicate parts.

The major factor determining the quality of an ultrasonic weld is the joint design. The two major types of joint designs are shear joints and energy director joints. The choice depends on the type of material to be welded and the end use requirements.

Shear Joint

A shear joint is more commonly used on semi-crystalline materials such as nylon* and polyester. Due to their sharp melting points, semi-crystalline resins often do not achieve strong welds with energy director joints. The molten material flowing from the weld area quickly resolidifies before welding to the opposite interface.

In a shear joint, a small contact area is initially melted and then continues down the weld surface as one part is forced into the other. Due to the good material mixing between the welded parts, strong structural and hermetic seals can be obtained.

Typical joint designs and interference guidelines are shown in Figure V-17 (a–c). As with energy director joints, flash traps can be included in the joint design.

* NOTE: Hygroscopic materials such as nylon should be welded in their "dry as molded" condition to prevent brittle welds. This can be achieved by welding parts soon after molding, or by placing parts in moisture proof bags to assure dry as molded condition.

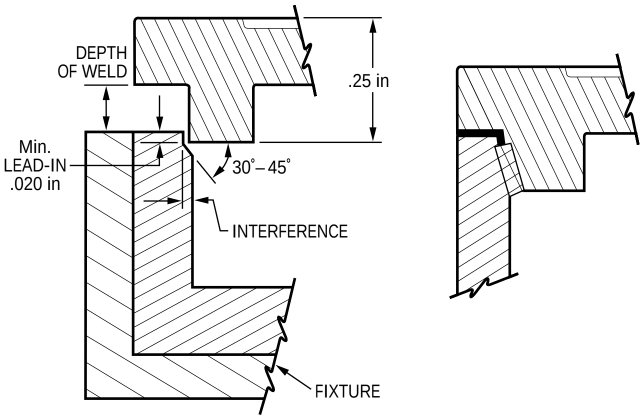


Figure V-17 (a). Shear Joint

Illustration credit: "Vibration Welding," Branson Ultrasonics Corp., 1980.

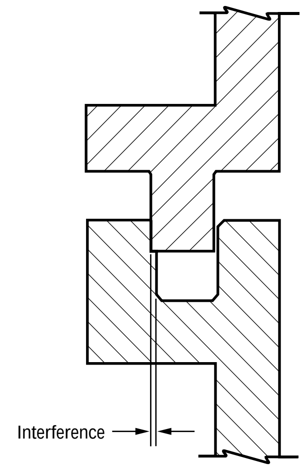


Figure V-17 (b). Shear Joint (Flash Trap)

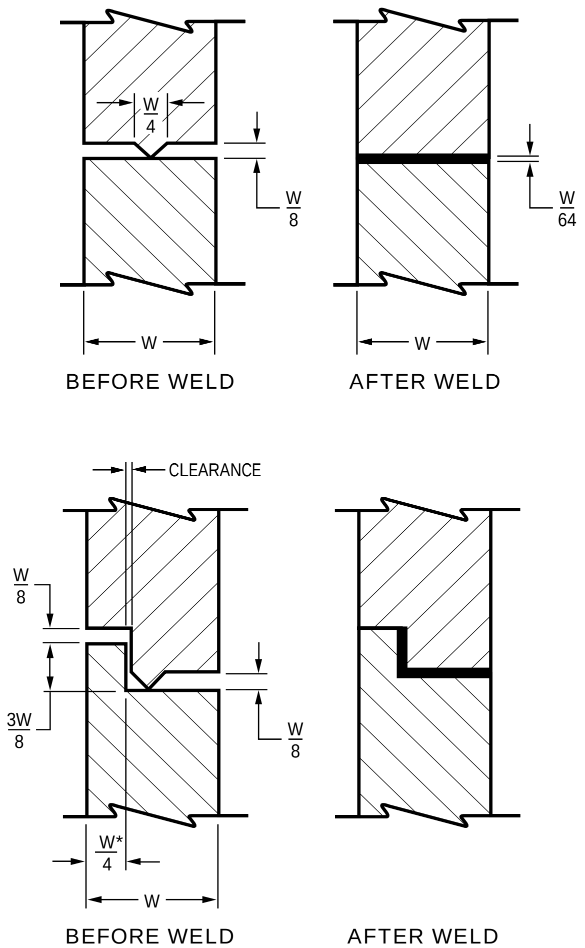
Maximum Part Dimension (In)	Interference for Size (Range)	Part Dimension Tolerance (In)
<0.75	0.008 to 0.012	±0.001
0.75 to 1.50	0.012 to 0.016	±0.002
>1.50	0.016 to 0.020	±0.003

Figure V-17 (c). Shear Joint Interference Guidelines

Energy Director

An energy director is a raised triangular bead molded on one of the joint surfaces. It concentrates ultrasonic energy causing a rapid initiation of the melt and welding of the material.

Energy director joints are normally used for amorphous materials. Typical joint designs are shown below in Figure V-18. For appearance parts, flash traps can be designed into the joint. (Figure V-19.)



*Minimum of .024 in

Figure V-18 (a) & (b). Energy Director

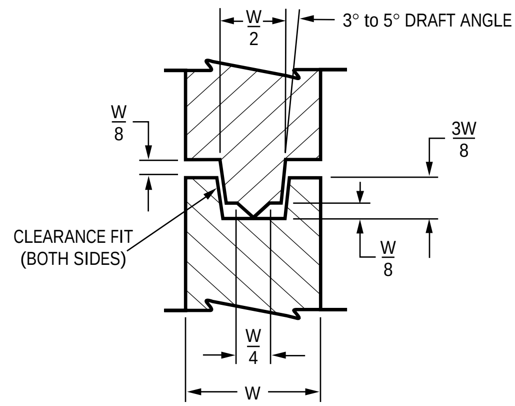


Figure V-19. No Flash

Illustration Credit: Holtz, Richard, "Vibration Welding: Fast, Quiet, Efficient," Assembly Engineering, Hitchcock Publishing.

Vibration Welding

Vibration welding is a preferred method for assembling large structural parts of the same or very similar thermoplastic parts. In this process, frictional heat is developed by moving the two parts relative to each other under pressure. Strong hermetic bonds can be achieved using this process.

Depending on the equipment (Figure V-20), welding frequency is either 120 Hz or 240 Hz with weld peak to peak amplitudes being .060 in.–.140 in. and .030 in.–.065 in., respectively. Allowance for this amplitude must be built into the joint. Weld time is generally 2–3 seconds. The welding cycle is described in more detail below.

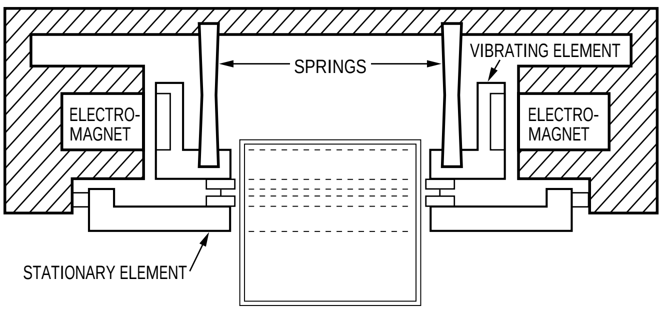


Figure V-20. Components of a Vibration Welder

Illustration credit: “Designing Parts for Ultrasonic Welding,” Branson Ultrasonics Corp., 1980.

Standard machines can accept parts up to 16 in. x 24 in. The back and forth motion of vibration welding also helps to remove surface contaminants such as mold release from the weld area.

One of the main limitations on this welding process is that the weld joint must be designed so that the reciprocation motion takes place on a single plane. However, the plane need not be flat, it may be bowed or bent as in automotive intake manifolds. Separate welded areas can be incorporated on parallel planes to the direction of motion on the same part. This method is used to create separate sealed integral gas and oil compartments on chain saw chassis.

A proper holding fixture is critical in achieving a good weld. Care must be taken when designing fixturing to prevent unsupported part walls from flexing during welding. A new method called orbital vibration welding makes welding unsupported walls easier by producing a constant circular motion so no wall is ever perpendicular to the weld direction.

Some common joint designs are shown in Figure V-21, including flash traps to provide for a cosmetic appearance. The flash trap must accommodate the melt area (shaded areas are basically equal in volume).

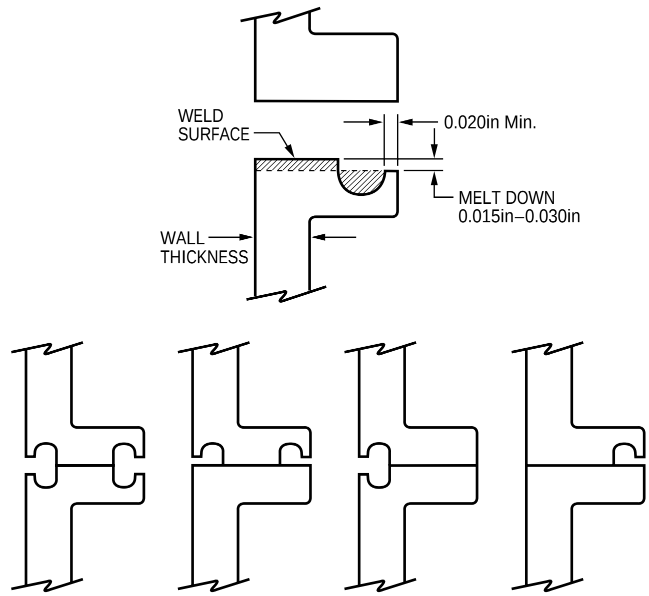


Figure V-21. Vibration Weld Joints

Illustration Credit: Holtz, Richard, “Vibration Welding: Fast, Quiet, Efficient, Assembly Engineering,” Hitchcock Publishing.

Other Assembly Techniques

Thermoplastic Staking

Staking (including ultrasonic, heat and hot gas) is a common assembly technique to join two dissimilar materials. A stud configuration or boss molded into one of the plastic parts protrudes through a hole or matching configuration in the second part. A specially contoured horn contacts and melts the top of the stud, forming a head and locking the second part in place. Staking is simple, fast and permanent. It produces a tight assembly with a variety of head contours to choose from (Figure V-22 shows a dome configuration).

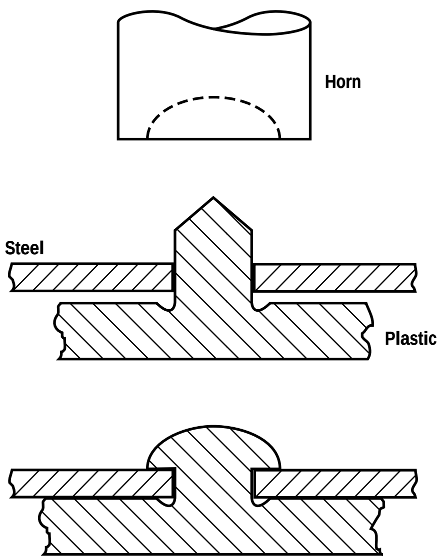
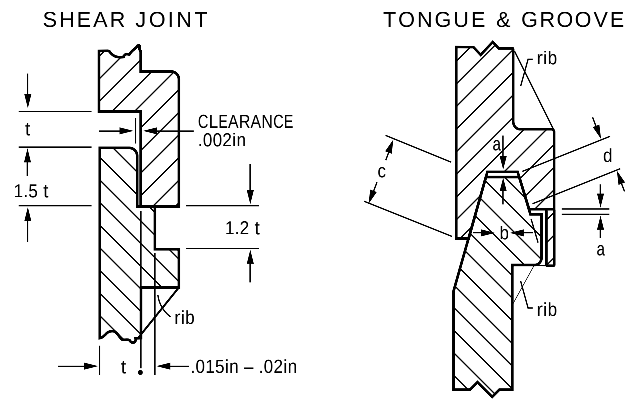


Figure V-22. Dome Stake

Spin Welding

Spin welding is a fast and practical assembly technique for joining circular parts or surfaces. Most thermoplastics can be spin welded, particularly rigid resins. Welds are made by rotating one part against the other fixed part at high speed and under pressure. Frictional heat melts both surfaces. Rotation is stopped and pressure is maintained until the weld solidifies. Strong, permanent and hermetic welds can be obtained, but accurate orientation between the parts is difficult. Cycle times are generally 1 to 2 seconds and ordinary machinery equipment can be used. Part configuration or a keying feature is needed for rotating a part.

The welding joint can be either flat, angled or V-shaped, usually with a flange for increased surface area and rigidity. Flash from the weld can also be hidden with special joint designs (see Figure V-23).



Where:

t = wall thickness of the part

a = depth of weld. Should be .5 to .8 times the wall thickness.

b = angle of stationary joint interface. Must be 30 degrees or greater to avoid jamming.

c+d = weld surface. Should be up to 2.5 times the wall thickness.

Figure V-23

Credit: Forward Technology Industries.

Electromagnetic Welding

Electromagnetic welding provides a simple, rapid and reliable assembly technique to produce a strong and hermetic joint. A specially designed strand is placed between the two parts to be welded. This assembly is then exposed to an induction heat field which melts the strand and plastic to form a strong bond at the interface. Our studies show the shear strength of the weld to be about 5000 psi. A variety of joints can be used as in other welding techniques. A tongue and groove joint is shown in Figure V-24.

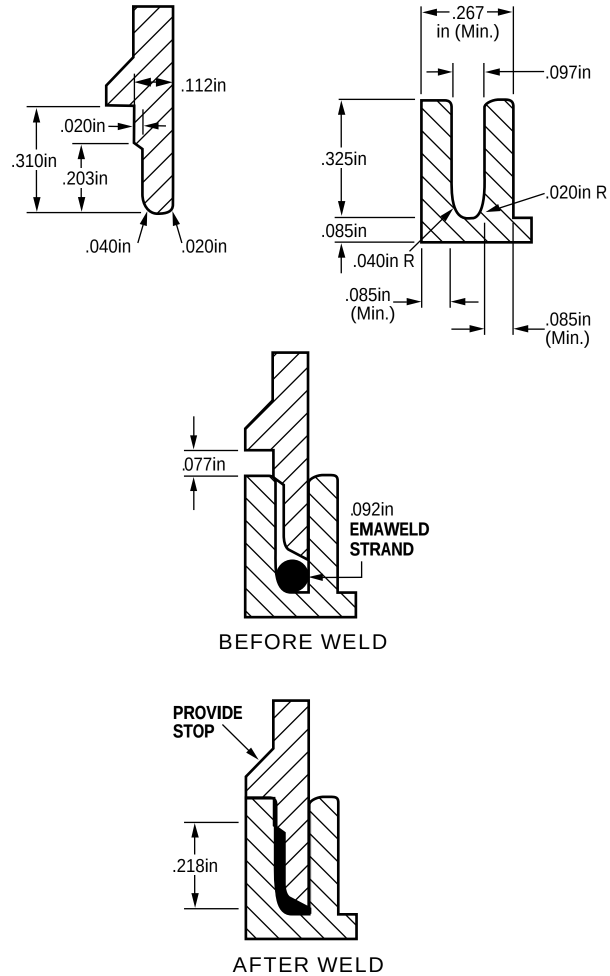


Figure V-24

Credit: Emabond Systems.

Part VI

Plastic Materials

Part VI: Plastic Materials

Plastics are man-made materials. They are made up of long chains of large molecules. Each molecule consists of many units of organic chemicals, thus called a polymer (many units), or macromolecules. At room temperature, the material is solid and rigid, and it can withstand significant structural load. Some of the materials retain rigidity at relatively high temperatures and can replace metallic components in such high temperature environments as automobile underhood applications.

The material can be formed into a finished shape by molding, extruding or shaping under high temperature conditions.

Classification of Plastic Materials

General classification of plastic materials is shown in Table VI-1.

PLASTIC MATERIALS		
THERMOPLASTIC		THERMOSET
CRYSTALLINE	AMORPHOUS	
		EPOXY
		MELAMINE
PP	PC	PHENOLICS
PE	PS	UNSATURA
	POLYESTER	
PET	PPS	BUTYL RUBBER
PBT	PPO	
NYLON (PA)	PVC	
ACETAL (POM)	ABS	
	SAN	
	POLYSULFONE	
	POLYARYLATE	
	POLYETHERIMIDE	
	PMMA (ACRYLICS)	

Table VI-1. Family of Plastic Materials

Thermoplastic material is processed in a molten state at elevated temperature, while thermoset plastics are processed in an uncured state and then cured in the mold. Thermoplastics can be reprocessed by melting the finished product while a thermoset material cannot because it has no melting point. It will degrade or char if exposed to high temperature.

Thermoplastic material can be classified into two categories: Crystalline and Amorphous. Figure VI-1 shows molecular structure of the material in various forms.

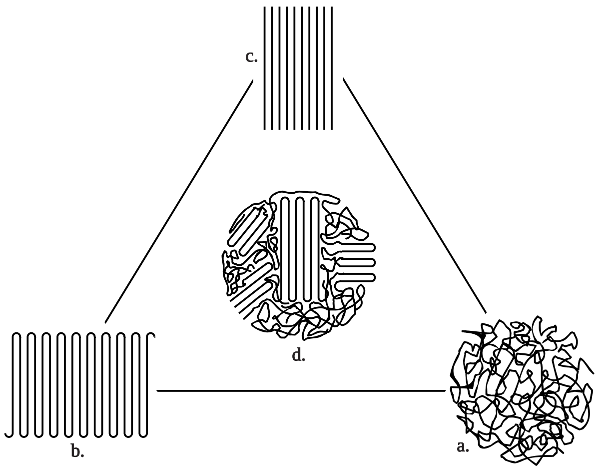


Figure VI-1.
Molecular Configuration of Crystalline Polymers and Amorphous Polymers

a) Amorphous Polymer Solid
b) Folded Chain Polymer Crystal
c) Extended Chain •Drawn Polymer Fiber
d) Semi-Crystalline Polymer

The molecular structure of crystalline polymers shows periodic folding of molecules, whereas amorphous polymers consist of randomly entangled polymer chains.

When heated, the crystalline polymers exhibit a distinct melting point (T_m) and change from rigid plastic to easy-flowing liquid. Amorphous polymers change their rigidity gradually as the temperature rises. They start in a rigid state at room temperature and get softer above the glass transition temperature (T_g) until they become liquid in the higher temperature regions.

Figure VI-2 shows the temperature dependence of the modulus for various classes of plastics.

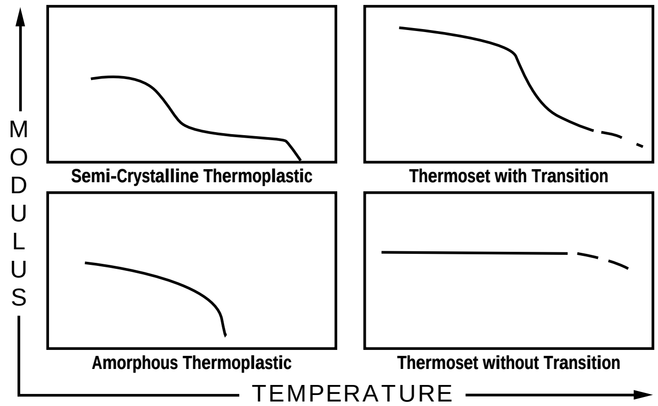


Figure VI-2.
Mechanical Characteristics of Various Plastics

In real molding conditions, most of the crystalline polymers cannot achieve full crystallinity. Instead, they will form some crystalline regions and some amorphous regions. Therefore, they are sometimes called semi-crystalline polymers. The proportion of the two phases depends on the cooling rate during molding. Faster cooling rates will result in higher amorphous content, as shown in Figure VI-3 for a Nylon 6 material.

Since a semi-crystalline polymer contains both crystalline and amorphous phases, it exhibits both a glass transition behavior and a sharp melting point.

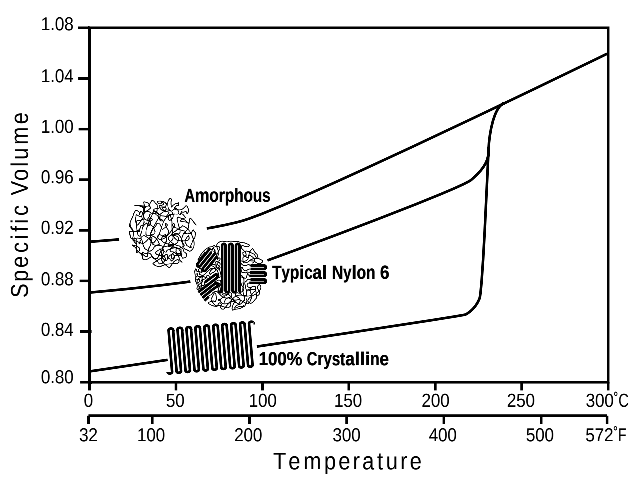


Figure VI-3.
Volume vs. Temperature Behavior of Nylon 6

Molecular Weight Distribution

Molecular chain length in plastic materials varies from very short to very long, and the distribution of such chain lengths, or molecular weight, creates a form of bell curve, as shown in Figure VI-4.

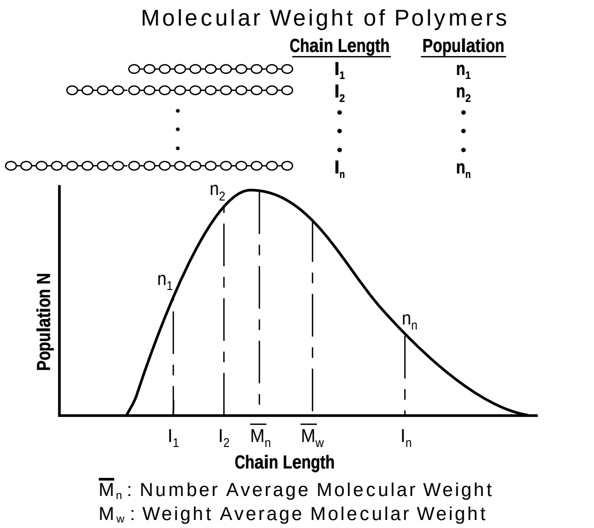


Figure VI-4.
Molecular Weight Distribution of Polymers

The weight averaged value (M_w) of the molecular weight distributions affects important physical properties, such as melt viscosity and part strengths.

The full spectrum of molecular weight distributions can be obtained through an analytic lab technique called Gel Permeation Chromatography (GPC). However, this method is rather time consuming and costly. A simple method to evaluate relative value of a given polymer is to use the Melt Index (MI) method. Molten plastic is placed in a heated capillary chamber, and it is pushed through a nozzle by placing a specified weight on a plunger. The amount of plastic collected at the bottom of the capillary in a given time is called MI, and this number is related to the molecular weight of the sample. The higher the number, the lower the molecular weight. There are similar but a little more complicated methods of estimating the molecular weight distribution. They are Formic Acid Viscosity (FAV) and Intrinsic Viscosity (IV). FAV is normally used for Nylon materials, and IV is used for thermoplastic PET materials.

Physical Properties

Plastic materials have significantly different physical properties compared to metallic materials. Some key properties are compared against metallic materials in the following topics:

Density

Plastic material is significantly lighter than most metallic materials, as shown in Figure VI-5. Plastic materials replace metals in many applications where product weight reduction is desired.

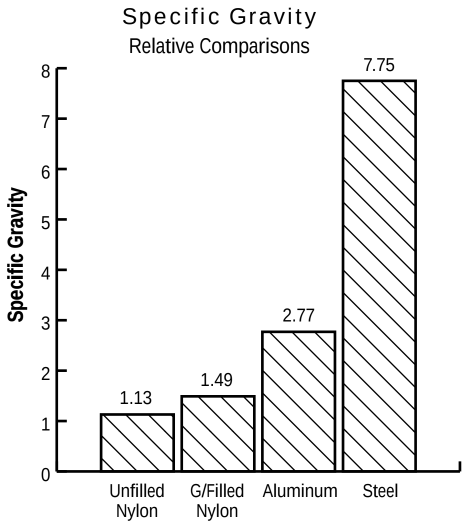


Figure VI-5.
Specific Gravity

Thermal Expansion

Plastic materials change dimensions significantly with temperature. Therefore, the product design engineer should calculate dimensional changes over the service temperature range, to verify that critical dimensions will remain within acceptable limits. Figure VI-6 compares the thermal expansion coefficient of various materials.

Care must be taken when joining materials having different coefficient of thermal expansion for buckling, tensile and shear stress, etc. Large parts are also more of a concern than smaller parts.

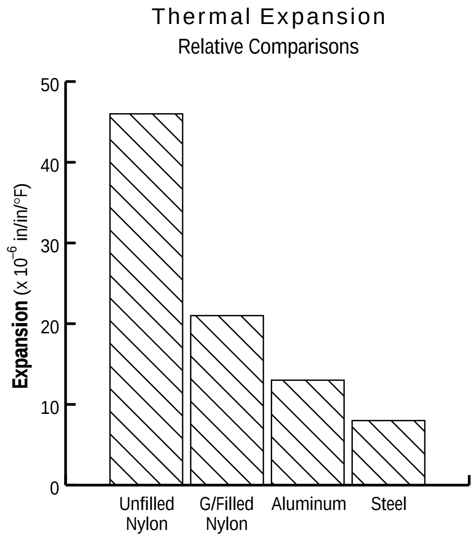


Figure VI-6.
Thermal Expansion Coefficient

Thermal Conductivity

Plastic materials do not conduct heat well and are about two orders of magnitude less conductive than the metals. Plastic is then a good thermal insulator. This thermal characteristic can be a positive or negative factor depending on the application. Figure VI-7 shows thermal conductivities of various plastic materials as compared to other materials.

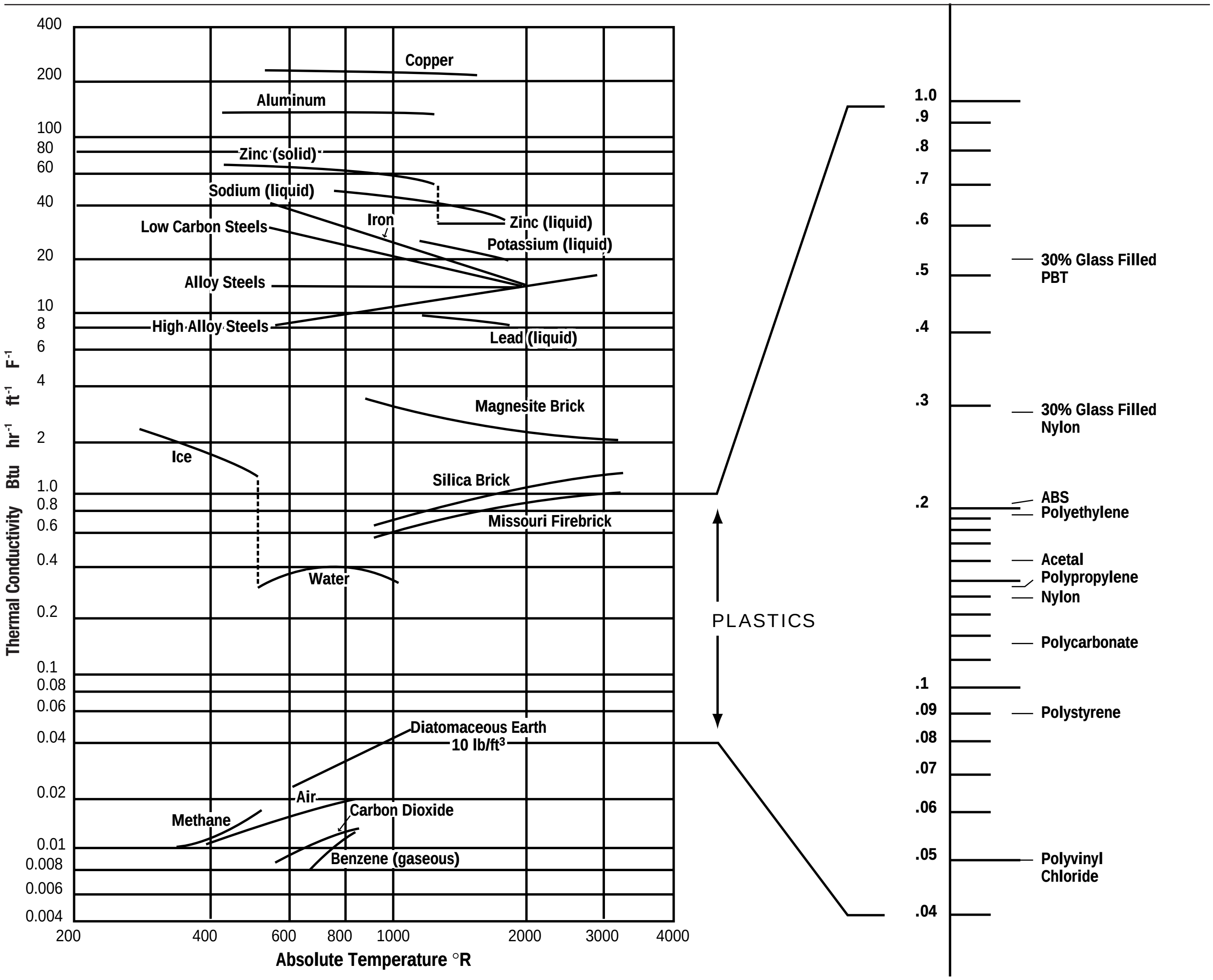
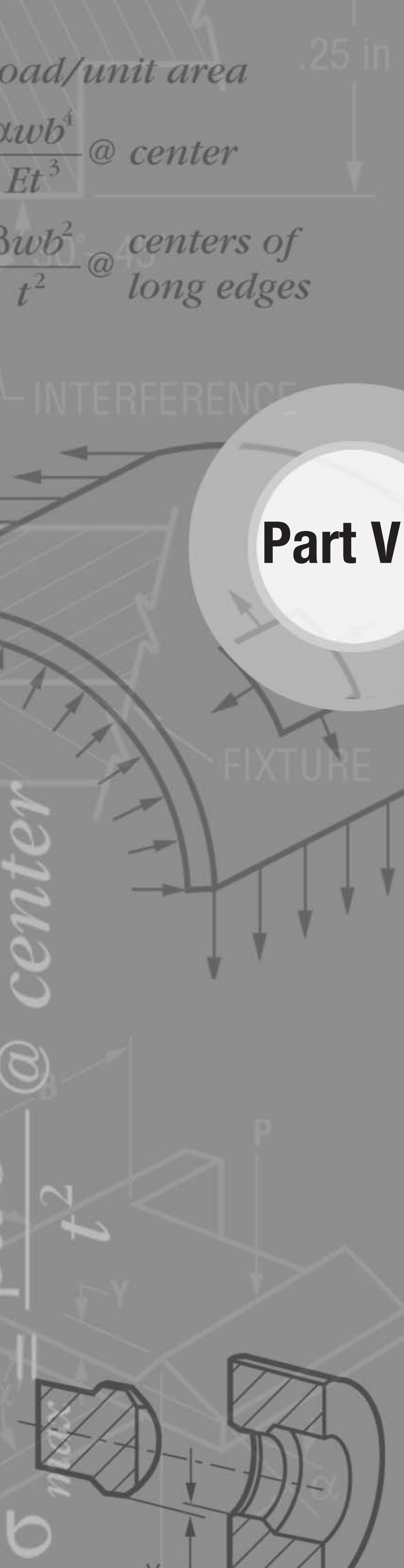


Figure VI-7. Thermal Conductivities of Solids, Liquids and Gases with Temperature

Illustration credit: Arpaci, Vedal S., "Convection Heat Transfer," Addison-Wesley.



Part VII

Physical Properties

Part VII: Physical Properties

The Mechanical Properties of Plastics

The mechanical properties of plastic materials can vary depending on the service environments, the duration of the service loading, types of loading, part configuration, etc. So the short-term properties cannot be applied to long-term applications. Therefore, design engineers should obtain property data that is applicable to the service conditions and life expectancy of the product.

Definitions of various physical properties and terminologies are listed in Appendix I, at the end of this guide.

Short-Term Properties

Standard mechanical properties are normally obtained by test methods as specified by the American Standard Testing Materials (ASTM) or by the International Standard Organization (ISO) methods. These two standards are cross referenced in Appendix II. You can find these properties for individual resins listed in BASF's product data sheets. (Product data on all our individual resins is also available via the Internet. Call BASF for details.)

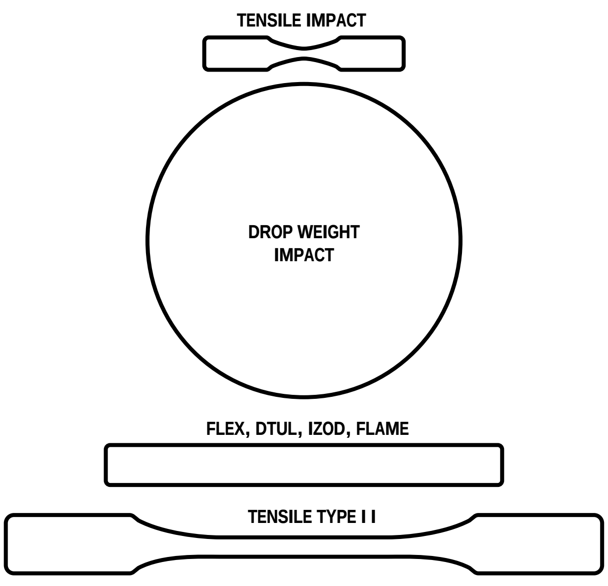


Figure VII -1. Test Specimens

Test specimens used for some standard test methods are shown in Figure VII-1. The material properties, however, can be affected significantly by the following factors:

Notches

Notches or sharp corners introduce stress concentrations and can induce premature failures, especially during impact. Sudden change of cross-sections are to be avoided when designing with plastics.

Resistance of material to the combined effects of notches and impact is measured by the Notched Izod Impact Test. The notched Izod specimen geometry is shown in Figure VII-2. The influence of the notch radius on the impact resistance is shown in Figure VII-3 and it indicates the notch sensitivity of materials. Impact resistance of materials is better measured by drop weight impact loading (see Figure VII-4 for testing method). It produces significantly different ranking of materials compared with those obtained by the notched Izod test method, as shown in Figure VII-5.

Rate of Loading

Higher rates of loading tend to reduce elongation of a plastic material, and it can result in brittle failures. Some materials are more resilient to such rapid loading effects than others.

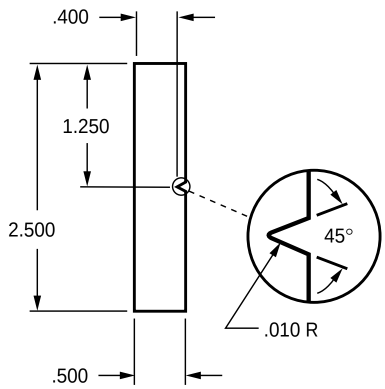


Figure VII-2. Impact Specimen

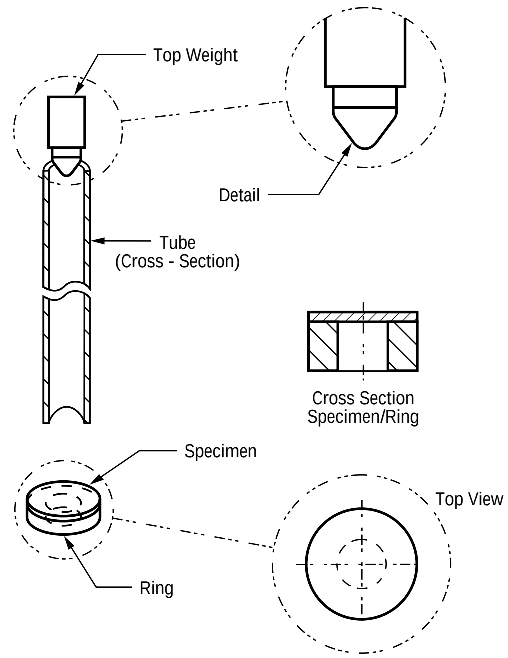


Figure VII-4. Drop Weight Impact Test

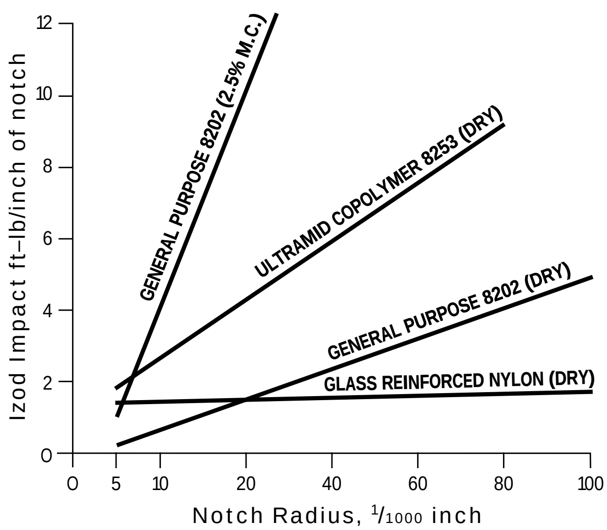


Figure VII-3. Izod Impact Strength vs. Notch Radius

Drop Weight (ft-lbs)		IZOD (ft-lbs/in notch)	
Polycarbonate	>180	Polycarbonate	12.0
Nylon 6	135	Modified Nylon 6/6	3.0
Modified Nylon 6/6	125	Nylon 6 Copolymer	2.5
Nylon 6 Copolymer	125	Modified PPO	1.9
PBT	95	20% GR Polycarb	1.8
Nylon 6/6	75	30% GR PBT	1.3
Nucleated Nylon 6	75	30% GR Nylon 6	1.3
30% GR Nylon 6	3.0	PBT	0.8
20% GR Polycarb	2.0	Nylon 6/6	0.6
30% GR Nylon 6/6	1.5	Nylon 6	0.5
30% GR PBT	1.0	Nucleated Nylon 6	0.4

Figure VII-5. Drop Weight Impact Values vs. Izod Impact Values of Various Engineering Thermoplastics

Temperature

Properties of plastics change significantly with temperature. Typical property changes with temperature are shown in Figure VII-6. Property values corresponding to the service temperature should be used for designing a plastic part. Deflection Temperature Under Load (DTUL) is sometimes used for screening high temperature grade materials. The test apparatus is illustrated in Figure VII-7. However, this test method is based on a specific deflection during test rather than a total time history. Results should be used in combination with other inputs.

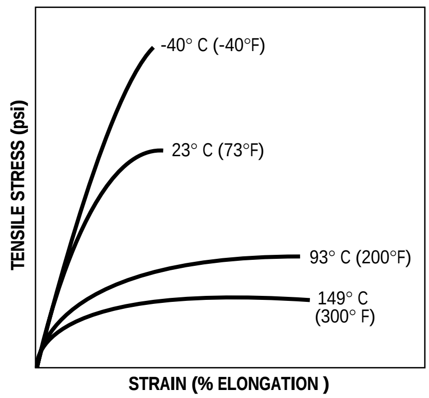


Figure VII-6. Tensile Stress-Strain Curve of Petra 130 at Various Temperatures

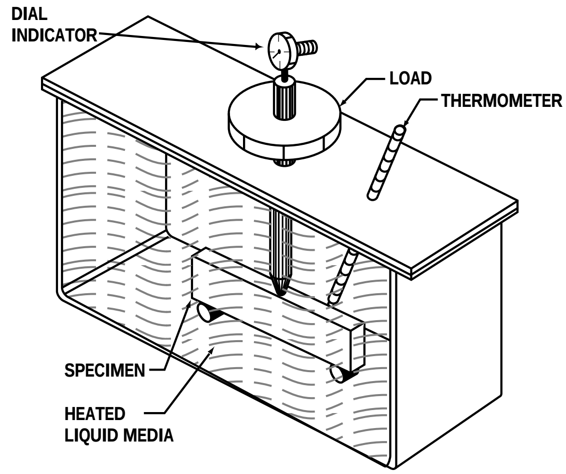


Figure VII-7. DTUL Apparatus

Thermal Aging

Plastic degrades under extended exposure to high temperature environment. Figure VII-8 shows the effect of heat aging on an Ultramid® nylon product. And Figure VII-9 shows how extensive heat aging affects thermoplastics.

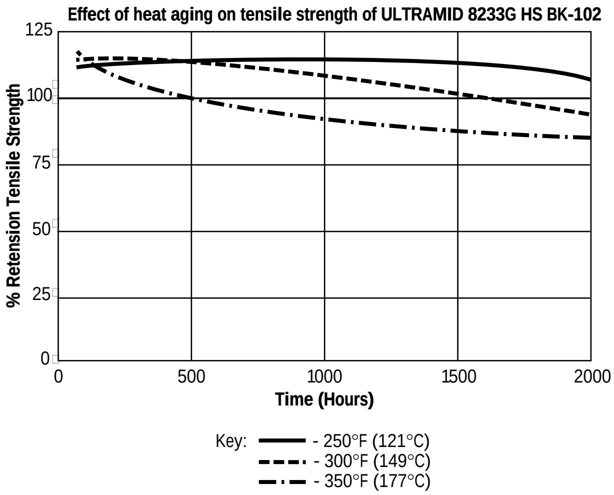


Figure VII-8. Heat Aging Effect

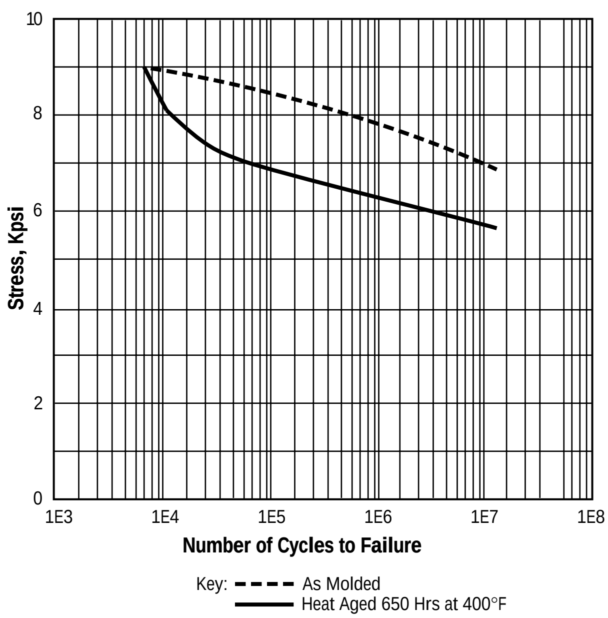


Figure VII-9. Fatigue Life of Petra 130 Before and After Heat Aging

Moisture

Nylon 6 or 6/6 absorb moisture from the air and environment. Mechanical properties and dimensions will change depending on the amount of the absorbed moisture. Figure VII-10 shows the flexural modulus change due to temperature and moisture. (See •Dimensional Considerations •Moisture Absorption• in Part VII of this guide.)

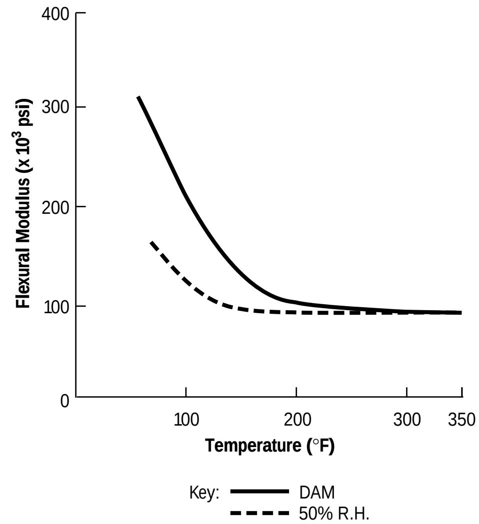


Figure VII-10. Flexural Modulus vs. Temperature and Moisture of Ultramid®8202

Dimensional Considerations (Moisture Absorption)

Effects of Moisture

All nylons are hygroscopic. The amount and rate of moisture absorbed from the atmosphere depends upon the ambient humidity and temperature. A study performed by BASF showed that the annual average relative humidity throughout the USA varies between 40% to 60%.

Design engineers should account for moisture effects when designing parts with nylon. The conventional practice is to choose 50% relative humidity at 73°F and base the design on property values at this condition.

Nylon will increase in impact resistance, toughness and size, while its strength and stiffness properties will decrease as it absorbs moisture. The time it takes for nylon to come to equilibrium depends on the thickness. The moisture absorption rate is shown in Figure VII-11 for Ultramid 8233 (33% GR Nylon).

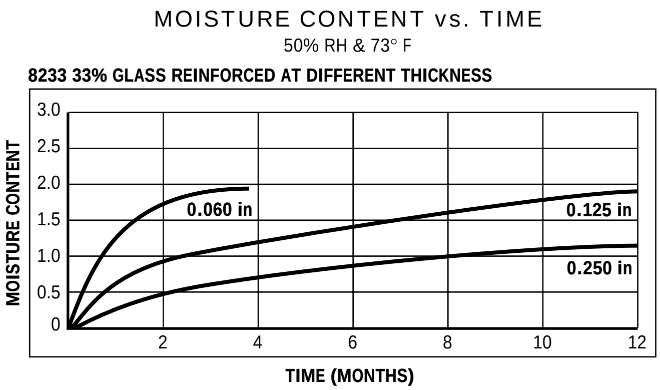


Figure VII-11

Dimensional Changes

Nylon parts will expand with exposure to moisture, as can be seen in Figure VII-12. These changes are small and need only be considered for applications with very large dimensions or very tight tolerances.

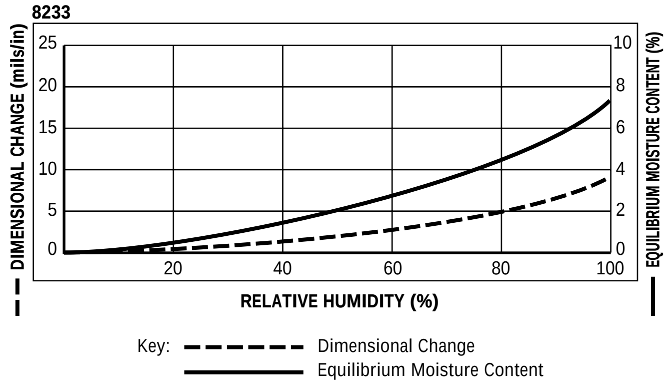


Figure VII-12

Accelerated Moisture Conditioning

Some applications having critical dimensional, property, or impact requirements may need moisture conditioning prior to use or testing. Submerging the parts in room temperature water or high temperature water can accelerate the time to achieve the design moisture content (Compare Figure VII-11 with Figure VII-13.)

Once at the new average condition, further dimensional changes will be minimal.

See Dimensional Considerations manual for other materials.

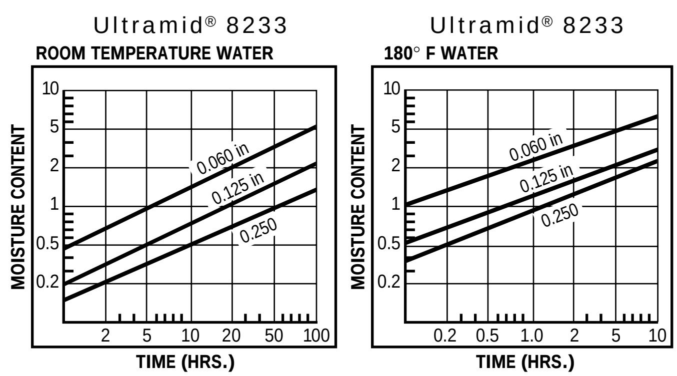


Figure VII-13

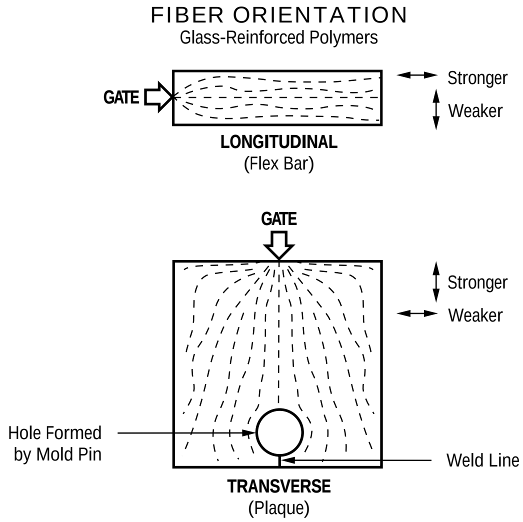


Figure VII-14

Process Induced Property Variations

Fiber orientation, introduced during molding, creates different directional property characteristics: stronger in the flow direction and weaker in transverse direction. (See Figure VII-14.) A typical property profile of an injection molded Petra® 130 shell structure is shown in Figure VII-15.

The molded part will also be very weak at a weld line, especially for glass reinforced material (see Figure VII-14). Excessive amounts of regrind will reduce strength as well. Improper preparation, such as poor drying of the resin, and improper processing, will degrade the resin, thereby reducing physical properties.

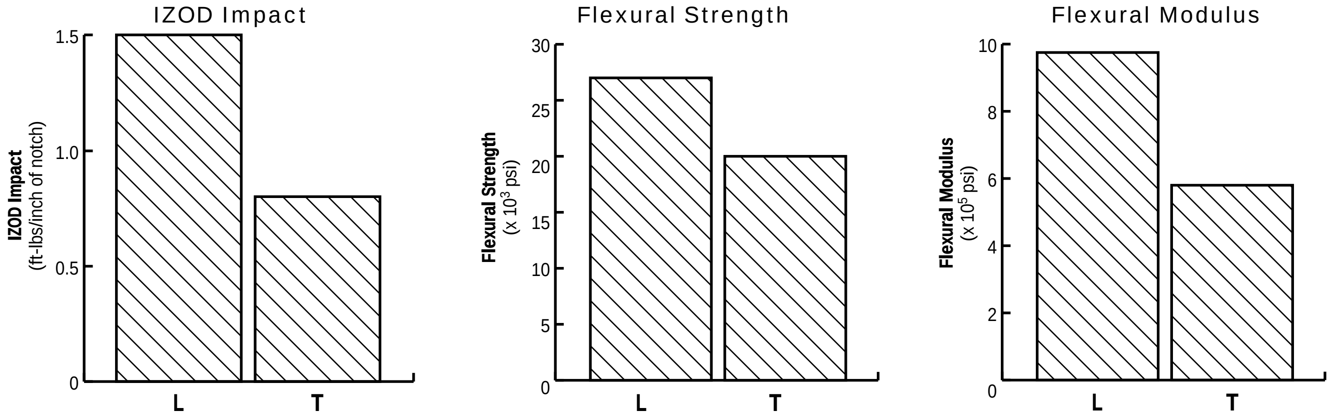


Figure VII-15. Property Variations With Fiber Orientation in a Molded Petra 130 Shell Structure

Where:

L-Longitudinal Direction

T-Transverse Direction

Additives (Color)
Carbon black or colorants do affect strength. The influence of colorants varies, depending on the ingredients and quantity used to achieve a specific color.

Ultraviolet (UV) Light
Ultraviolet light affects the polymer structure chain and reduces the physical properties of plastics. Most of the degradation is localized to the outermost layer so the overall strength decay is minimal. However, the surface appearance can be significantly affected. Special UV resistant grades are available from BASF. These grades are formulated to extend the surface appearance, and therefore, service life of the material.

Chemicals
Certain chemicals attack plastics and reduce their physical properties. Each polymer behaves differently when exposed to various chemicals. A design engineer should refer to the chemical resistance table for each material (See the Chemical Resistance Guide) to make sure the service environment is not harmful to the plastic material being used, and testing is recommended.

Long-Term Properties
Creep, Stress Relaxation and Service Life

When a load is applied on a plastic part, the part will elongate or collapse with time. The amount of the elongation depends on the magnitude of the load, duration, and if the load is constant or diminishing with time. Such time dependent change is called *creep*. The creep phenomenon is illustrated in Figure VII-16(a).

When a sustained load is applied Figure VII-16(a), creep will continue and lead to eventual failure. On the other hand, if the load is applied and then fixed, the stress will decrease with time Figure VII-16(b). This last case is called stress relaxation. One can observe this phenomenon in plastic assemblies such as mechanically fastened plastic components.

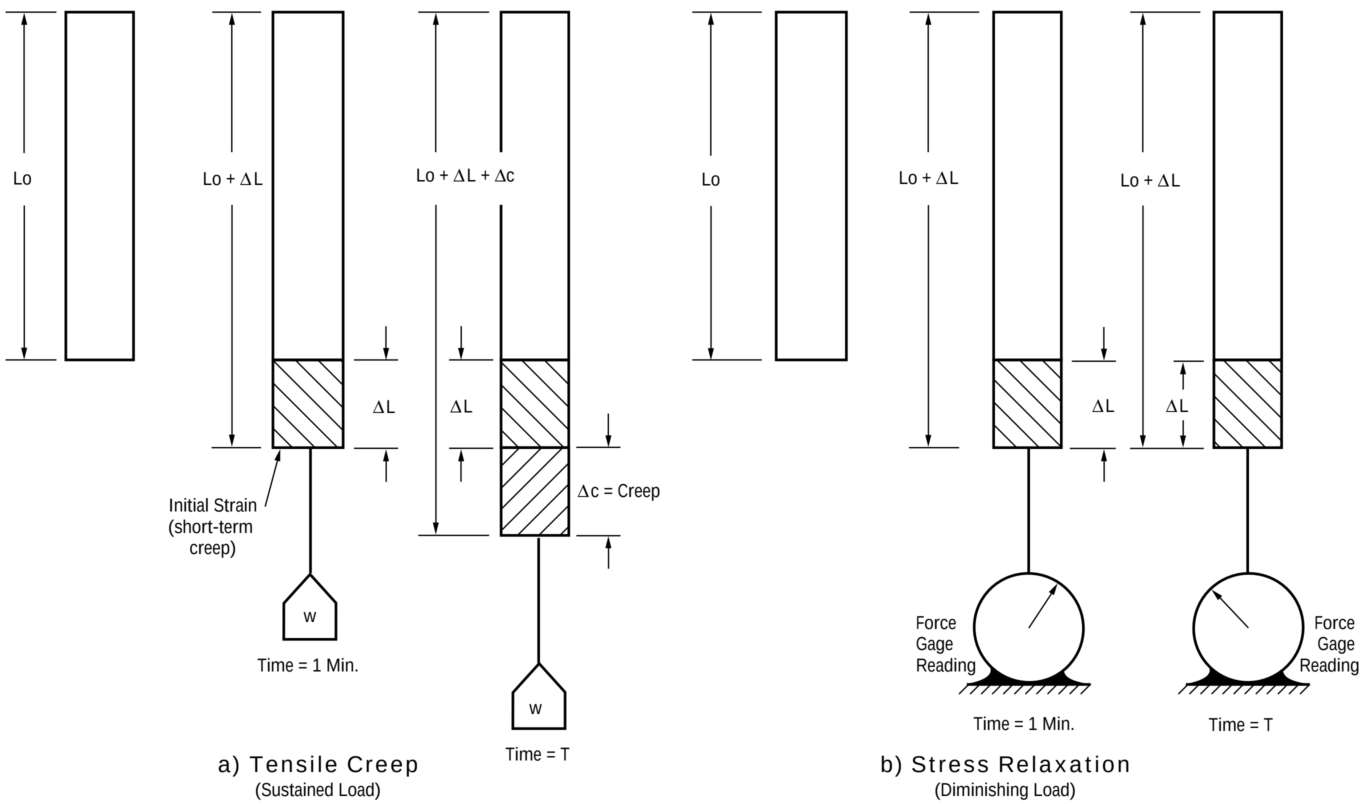


Figure VII-16. (a) & (b)
Creep and Stress Relaxation Phenomena

The creep strain is added to the initial elastic strain to arrive at a total strain. Elastic strain can be recovered immediately upon release of the load. Creep strain does not recover immediately but takes time to recover after the load is removed. Generally, a significant portion of the creep strain is unrecoverable. The amount of the creep strain and the rate of elongation depends on the applied load. The higher the load or stress, the higher the strain and the faster the rate. See Figure VII-17.

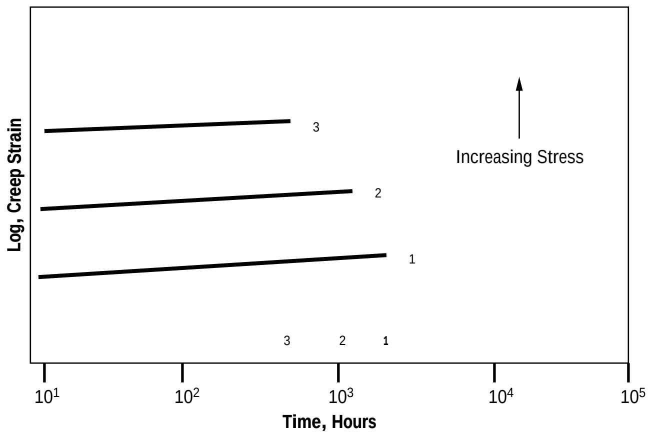


Figure VII-17. Creep Strain Showing Stress and Time Dependency

The rate of the stress decay takes place faster with a higher initial stress level, as shown in Figure VII-18. Based on this principle, reducing stress in a mechanical fastening application reduces the clamping force decay.

The amount of deformation and failure time depends on the stress level. If the failure time is plotted against the applied stress level on a log-graph, an approximate linear relationship can be found. One can now predict the service life of a part under sustained loading. Extrapolation of a curve for more than one decade in the time scale is not recommended. Examples of the service life prediction scheme are shown in Figures VII-19 and 20.

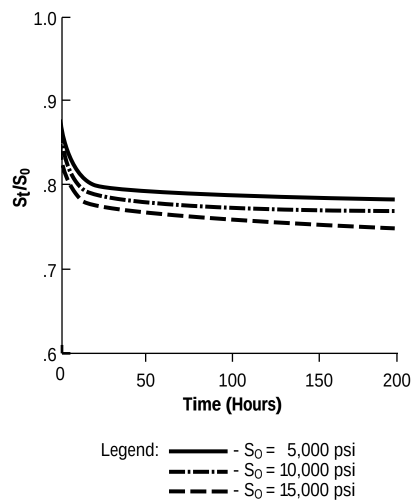


Figure VII-18. Compressive Relaxation of 33% Glass Reinforced Nylon

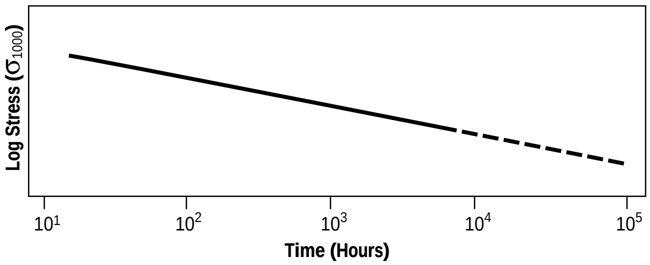


Figure VII-19. Extrapolation of a Creep-Rupture Curve for Service Life Prediction

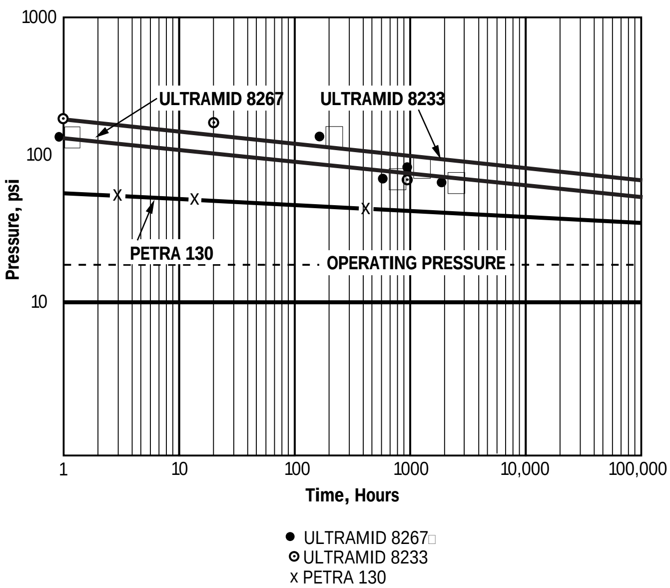


Figure VII-20. Pressure vs. Time to Rupture Curve of Wheel Assemblies (Air Filled) @ 73° F

Coefficient of Friction

The coefficient of friction COF (μ) is defined in ASTM D-1894 as the ratio of the frictional force (F) to the force, usually gravitational, acting perpendicular to the two surfaces in contact (N).• Therefore the COF, $\mu = F/N$, and is dimensionless. The COF is a measure of the relative difficulty of one surface moving over another. Static COF (μ_s) relates to the force required to initiate the movement. Kinetic COF (μ_k) relates to the force required to sustain the movement. μ_s is generally greater than μ_k. The lower the COF value, the easier it is to move one part relative to another.

The following COF data was generated by ASTM D-1894, in which a sled of one material was pulled over a plane of a second material (See Figure VII-21). This test is very sensitive to surface irregularities, imperfections and specimen warpage. Although the data presented in Table VII-1 are believed to be representative, a conservative design approach is recommended.

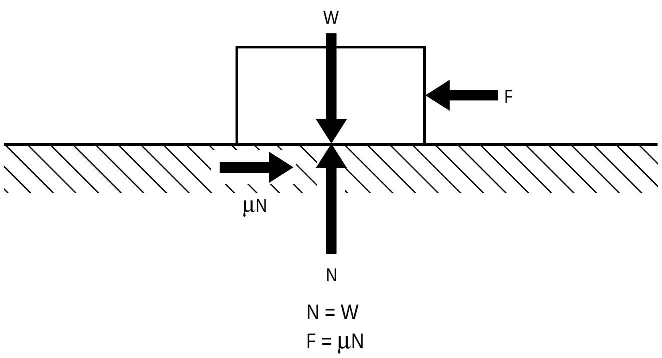
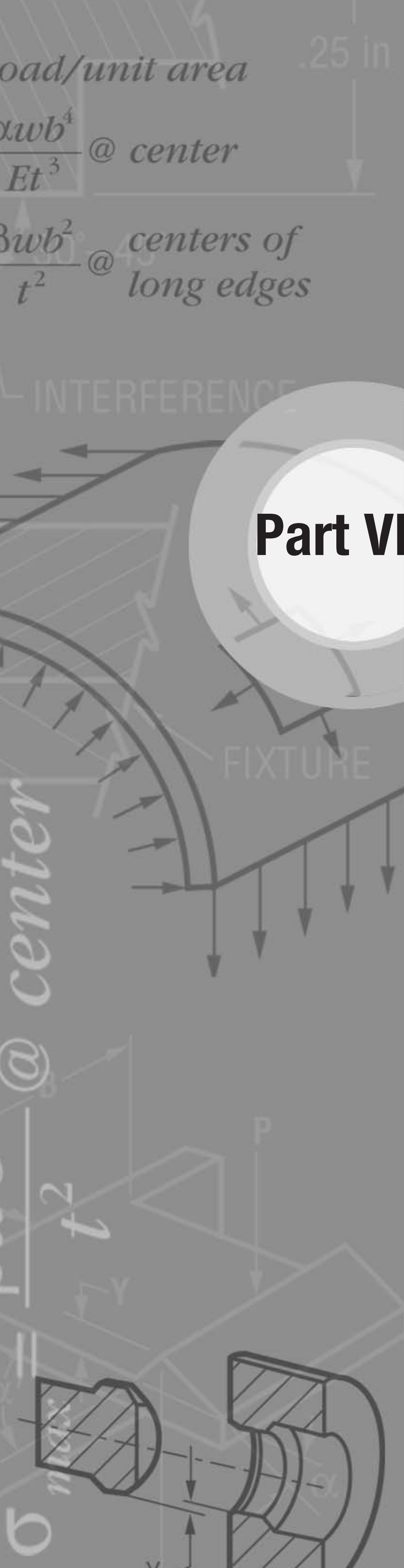


Figure VII-21

PRODUCT	POLYMER-TO-POLYMER		POLYMER-TO-STEEL	
	STATIC	KINETIC	STATIC	KINETIC
ULTRAMID®				
8200 HS	0.26	0.21	0.17	0.16
8202C HS	0.53	0.33	0.24	0.16
8224 HS	0.48	0.47	0.23	0.17
8253 HS	0.33	0.32	0.25	0.16
8254 HS BK-102	0.48	0.47	0.32	0.22
8350 HS	0.35	0.32	0.25	0.17
8351 HS BK-102	0.4	0.35	0.25	0.18
D-8358 HS BK-102	0.39	0.38	0.25	0.19
8233G HS	0.34	0.32	0.25	0.16
8267G HS	0.3	0.22	0.23	0.17
D-8333G HS GY5723	0.34	0.32	0.26	0.17
8360 HS	0.41	0.22	0.23	0.15
PETRA·				
130	0.22	0.21	0.15	0.13
230	0.37	0.27	0.2	0.15
D-242 BK-112	0.33	0.32	0.21	0.16
132	0.33	0.32	0.18	0.15

ULTRAMID and PETRA are registered trademarks of Basf Inc.

Table VII-1. Coefficient of Friction



Part VIII

Gas Assist Molding

Part VIII: Gas Assist Molding

Gas assist molding is used to partially core out thick sections. It produces large, dimensionally stable parts with good surface and mechanical properties. The process can also lower costs because it reduces cycle times and uses less material.

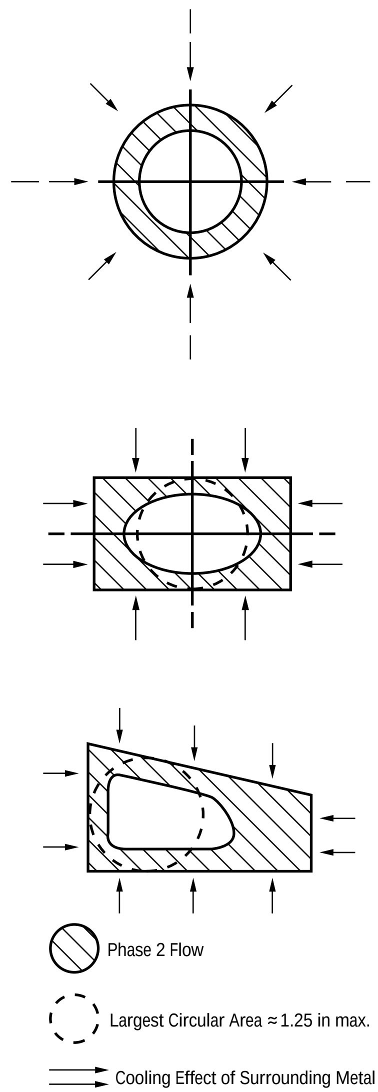
Gas assist molding is a form of injection molding in which the mold cavity is partially filled with molten plastic followed by injecting an inert gas, usually nitrogen, into the melt. Depending on the process, the gas can be introduced through the machine nozzle, into the runner, or into the part itself. The gas pressure is maintained until it is vented just prior to part ejection. In this way, the gas takes up the volume shrinkage of the plastic as it cools, packing out sink marks and greatly reducing molded-in stress that can cause the part to warp.

There are generally three ways gas assist molding is utilized:

1. Hollow Molding
2. Short Shot
3. Full Shot

Hollow Molding

This method is normally used to core out parts like chair arms and various types of handles, including those found on chain saws, cars and large appliances. The final cross-section is determined by part geometry, gas and resin flow, material type, and filler content. Some common cross-sections are shown in Figure VIII-1. The maximum circular area is generally limited to 1.25in.



(A) A circular core is produced during Phase 2 Flow that retains its form during Phase 3 Packing.

(B) A rectangular form produces thicker sections on shorter sides.

(C) Offset Phase 2 Flow produces solid sections on the opposite side.

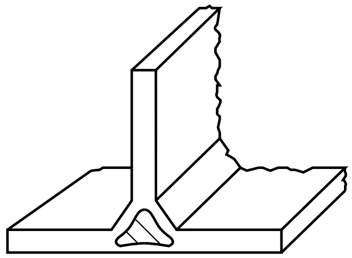
Phase 1. Plastic Injection
Phase 2. Gas Injection
Phase 3. Packing final channel shape

Figure VIII-1

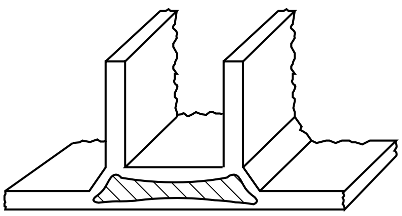
Reference: "Application of Gas Injection Technology" by Matthew Sayer, Cinpress, Ltd.

Short Shot Molding

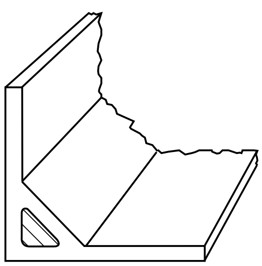
This version is normally used for structural parts where heavy ribs are desired for stiffness. The gas channels are generally positioned so the gas will flow along the base of ribs and under bosses, thus packing out sink marks. They can also be run along the base of the side walls to help stiffen the part and prevent warpage. Some design examples of these channels are given in Figure VIII-2.



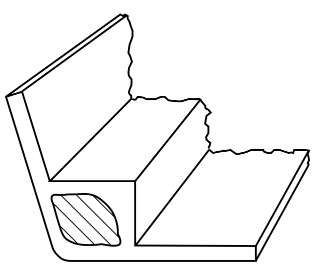
A. Vertical Rib



B. Bridge Rib



C. Sidewall



D. Sidewall

E. Channels Created Using Form

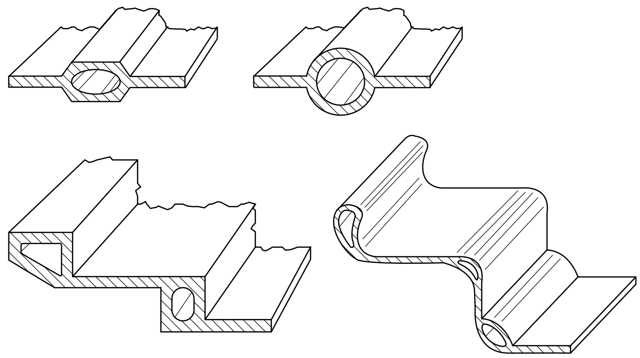
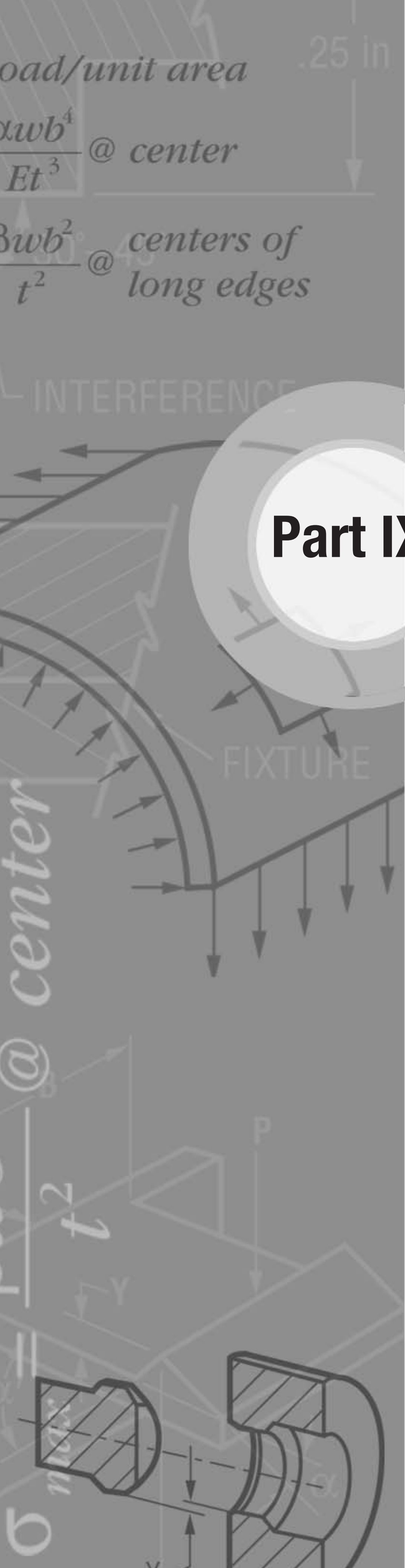


Figure VIII-2

Reference: "Design Tips for Gas-Assisted Injection Molding," diagram of rib designs by Indra Baxi of Sajar Plastics, Plastics Design Forum, (July/August, 1990)

Full Shot Molding

Here the mold cavity is filled completely with plastic before the gas is applied. The coring is limited to the volume shrinkage of the plastic. This is primarily used for highly aesthetic parts where elimination of sink marks is critical, especially those that will be painted or chrome-plated, such as mirror housings or door handles.



Part IX

Finishing

Part IX: Finishing

Electroplating

The mineral reinforced nylon 6 material (Ultramid® 8260), which is designed to accept plating, results in excellent plating adhesion and appearance. Plating can be used to enhance conductivity, shielding and/or aesthetics. An electrolytic pre-plating process of copper or nickel prepares the surface for final electroplating of copper, nickel and chrome in that sequence. Other plating materials can also be used where desired. Good design practice for an effective end product includes:

a) avoiding deep cavities or sharp corners

b) application of abundant radii to avoid plating build-up.

Deep pockets can be expected to be void of plating. Special surface preparation may be needed in some cases. The total thickness of plating is in the range of .001in to .005in. Refer to BASF's Finishing Manual for more detailed information.

Painting

Most plastics accept paint systems well, especially the amorphous resins. With special preparation of the surface for better adhesion (cleaning is essential), even the more difficult plastics, i.e., PE, PP and Acetal, which have more slippery surfaces and chemical resistance, can be painted.

BASF materials will accept paint systems well. Nylon and PET are excellent resins for paint applications, especially where high-temperature curing is required. Their ability to tolerate high temperatures for long periods of time without softening is a key advantage.

Printing/Hot Stamping

All known printing methods are effective when using BASF materials. Occasional surface preparation may be needed for improved adhesion quality. A well-cleaned surface is the most important preparation for quality adhesions.

Machining

Nylon and PET are readily machinable using conventional metal-cutting equipment. Cutting techniques for plastics are different than those for metals, and special preparations should be taken. When cutting, you should remember to:

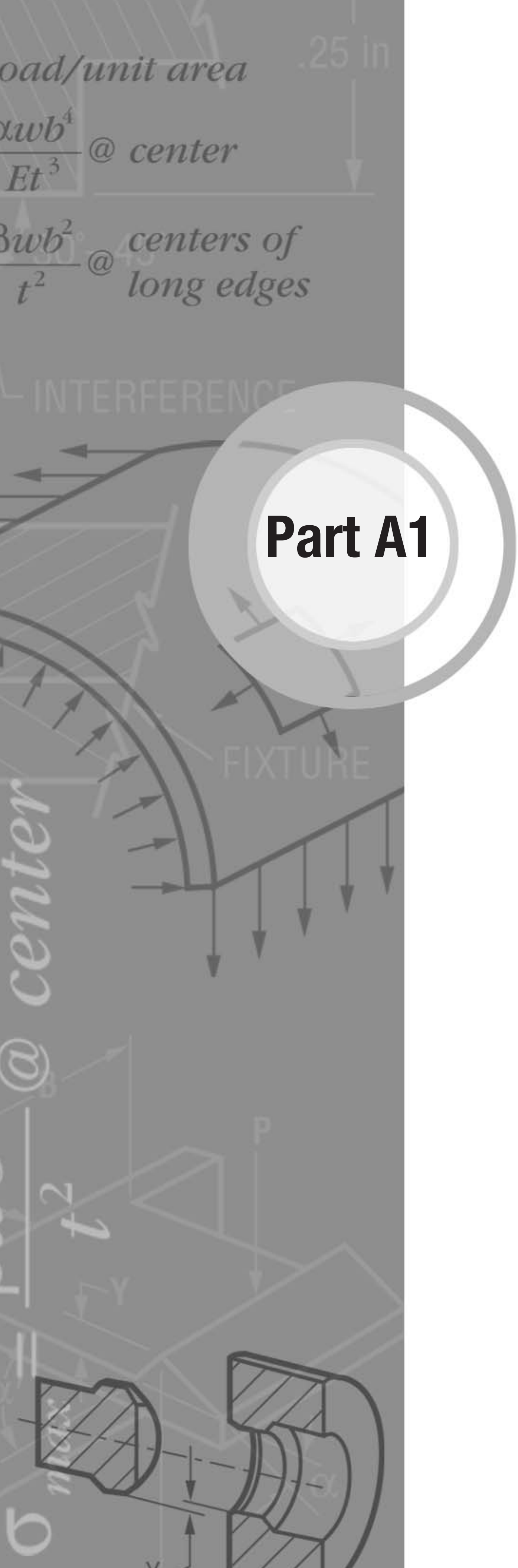
- Provide for cooling during the cutting process.
- Maintain a sharp tool with relief after the cut, especially for reinforced resins.
- Provide good support at the cutting area.
- Apply low cutter forces.
- Use carbide where possible.
- Use cutting points with a radius.

Surface Treatment

A designer can chose from a variety of surface treatments and plastic colors. Both can be molded into the plastic part and require no further finishing operations. The injection molding process will accurately duplicate the mold surface. Part function and/or aesthetics usually dictate the surface requirements. A smooth, uniform surface is often preferred for plating and painting, and high gloss is popular on many consumer applications. However, matte or textured surfaces are also attractive, are less slippery, provide contrast, hide sink marks, and disguise wear and abuse.

Specifying mold finish has often been arbitrary or neglected completely. A practical guide to surface finish selection is the SPI Mold Finish Guide which is available from the Society of the Plastics Industry. The finish should be specified by SPE/SPI number where possible.

Uniform matte and textured mold surfaces are usually less costly than a high polish and can be obtained by vapor blast or glass blast, while machining or chemical etching can produce a variety of patterns and textures. A wide selection of textures are available from companies that specialize in mold engraving.



Part A1

Appendix I: Physical Properties and Terminology

Appendix I: Physical Properties and Terminology

Density, specific gravity, mold shrinkage, water absorption, elasticity, plasticity, ductility, toughness, brittleness, notch sensitivity, tribological (lubricity and abrasive resistance) properties, anisotropy and isotropy are crucial physical properties determining the usefulness and durability of a plastic. These terms and others are defined below.

Anisotropy

Anisotropic material properties depend on the direction in which they are measured. Glass and mineral reinforced thermoplastics have a high degree of property dependence on orientation of fiber reinforcement.

Brittleness

Brittleness is the opposite of toughness. As a rule, reinforced thermoplastics show higher stiffness and lower impact properties, or more brittleness than unfilled plastics.

Density

The density of plastics is the mass in air per unit volume of material at 23° C (73° F), expressed in pounds per cubic inch or grams per cubic centimeter.

Ductility

The ability of plastic to be stretched, pulled or rolled into shape without destroying its integrity. Typical ductile failure of plastics occurs when molecules slide along or past each other.

Elasticity

Elasticity is the ability of material to return to its original size and shape after being deformed. Elastic limit is the greatest stress which a material is capable of sustaining without any permanent strain remaining upon complete release of stress. It is expressed in force per unit area. The elastic region is very important for linear analysis (FEA) of thermoplastic components. Most plastic materials have limited elasticity.

Friction and Wear

Friction is resistance against change in the relative positions of two bodies touching one another. If the plastic in the area of contact is loaded beyond its strength, wear or abrasion will take place. Although plastics may not be as hard as metal, their resistance to abrasion and wear may still be excellent.

Hardness

Hardness is closely related to wear resistance, scratch resistance, strength, stiffness, and brittleness. The various hardness tests provide different behavior characteristics for plastics:

—the resistance of a material to indentation by an indentor;

—the resistance of a material to scratching by another material;

—the measurement rebound efficiency or resilience.

Isotropy

An isotropic material is a material that retains the same physical properties when measured in any direction.

Lubricity

Lubricity refers to the load-bearing characteristics of a plastic under conditions of relative motion. Those with good lubricity tend to have a low coefficient of friction either with themselves or other materials and have no tendency to gall.

Mold Shrinkage

Mold shrinkage is the amount of contraction from mold cavity dimensions that a molded part exhibits after removal from the mold and cooling to room temperature. Mold shrinkage starts the very moment plastic is injected into the cavity of a closed mold, so optimum mold design will choose the best gate(s) position, runner diameter, cycle times, and the smoothest flow path to prevent excessive shrinkage or differential shrinkage across the whole part.

For a given material, mold shrinkage can vary with design and molding variables, such as part walls thickness, flow direction, and injection molding conditions. Mold shrinkage is very important for concurrent design, for material substitution (plastic for metal; plastic for plastic) and for specific applications.

Notch Sensitivity

Notch sensitivity is the ability of crack propagation through a plastic from existing stress concentration areas (sharp corners, grooves, holes, abrupt changes in the cross-sectional area). In evaluating plastics for a particular impact or cyclic loading condition, design variables (stress concentration areas) are quite important.

Plasticity

Plasticity is the ability of material to preserve the shape and size to which it is formed. Plasticity occurs when the stress goes beyond yield strength on the stress-strain curve for plastic. Increases in temperature affect plasticity of plastic materials.

Specific Gravity (Relative Density)

The specific gravity (relative density) is the ratio of the mass in air of a unit volume of the impermeable portion of the material at 23°C (73°F) to the mass in air of equal density of an equal volume of gas-free water at same temperature.

Toughness

Toughness refers to the ability of plastic to absorb mechanical energy without fracturing. This process is done with both elastic and plastic deformation. Material toughness is often measured as the area under the stress-strain curve. As a rule, unfilled resins have excellent toughness. Sometimes the toughness is measured by the amount of energy consumed to generate unit area of a fracture surface.

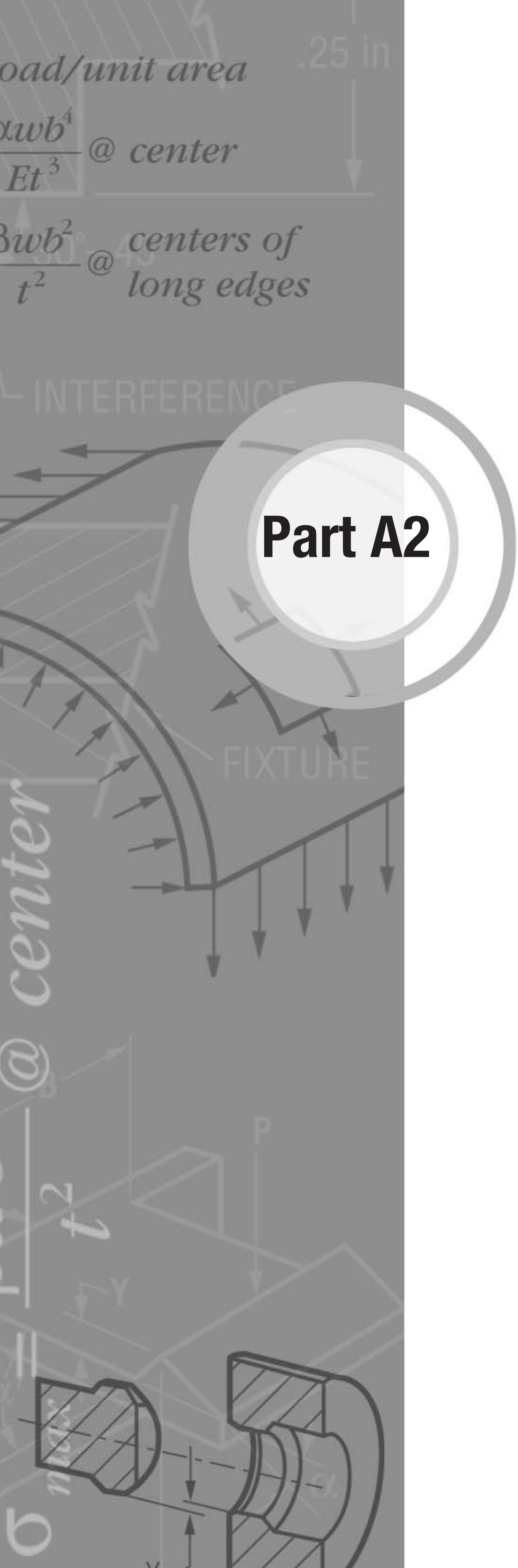
Warpage

Complex shapes of the finished part promote varying contraction rates relative to molded part dimensions and can cause internal stress to build up during the cool-down process while molding. This can cause warpage.

The warpage and plastic part-distortion can be caused by internal stresses generated by non-uniform shrinkage. Part warpage can be significantly greater than the in-plane mold shrinkage value.

Water Absorption

Water absorption is the percentage of increase in the weight of a plastic part during its immersion in water, or exposure to a humid air environment. Plastic parts (materials) can be either absorbent (hygroscopic) or non-absorbent (non-hygroscopic). Most engineering plastics show absorbent tendencies in their dry conditions (DAM–dry as molded). Plastic parts absorb water by direct exposure or from airborne water vapors at a rate specific to each material. Standard test specimens of a material whose physical property values would be appreciably affected by exposure to high temperatures in the neighborhood of 110°C (230°F) shall be dried in an oven for 24 hours at 50°C (122°F), cooled in a desiccator, and immediately weighed to the nearest 0.0001g. Generally, the rate of water absorption is measured when a material is exposed to 50% relative humidity air (50% RH). Saturation is given by percentage of part dry weight. The presence of water (water absorption) in the plastic part influences its physical, mechanical and electrical properties, as well as dimensional stability. Moisture in the resins before molding, unless removed by drying prior to processing, can cause serious degradation of physical properties.



Part A2

Appendix II: ISO and ASTM Test Methods

Appendix II: ISO and ASTM Test Methods

Property Number in ISO 10350:1998	Property	ISO Standard	ASTM* Standard	SI Units for ISO Test	SI Units for ASTM Test	U.S. Units for ASTM Test
1	Rheological Properties					
1.1	Melt mass-flow rate	1133	D 1238	g/10 min	g/10 min	g/10 min
1.5	Shrinkage, flow direction	294-4	-	%	%	%
1.6	Shrinkage, transverse direction	294-4	-	%	%	%
2	Mechanical Properties					
2.1	Tensile modulus	527-1 & 2	D 638	MPa	MPa	psi
2.2	Yield Stress	527-1 & 2	D 638	MPa	MPa	psi
2.3	Yield Strain	527-1 & 2	D 638	%	%	%
2.4	Nominal strain at break	527-1 & 2	-	%	-	-
2.5	Stress at 50% strain	527-1 & 2	-	MPa	-	-
2.6	Stress at break	527-1 & 2	D 638	MPa	MPa	psi
2.7	Strain at break	527-1 & 2	D 638	%	%	%
2.10	Flexural modulus	178	D 790	MPa	MPa	psi
2.11	Flexural strength	178	D 790	MPa	MPa	psi
2.12	Charpy impact strength	179	-	kJ/m^2	-	-
2.13	Charpy notched impact strength	179	-	kJ/m^2	-	-
—	Izod impact strength at 23°C	180	D 4812	kJ/m^2	J/m	ft–lb/in
—	Izod notched impact strength at 23°C	180	D 256	kJ/m^2	J/m	ft–lb/in

* For each applicable ISO method, corresponding ASTM methods are provided for reference. These methods may not be technically equivalent.

Property Number in ISO 10350:1993	Property	ISO Standard	ASTM Standard	SI Units for ISO Test	SI Units for ASTM Test	U.S. Units for ASTM Test
3	Thermal Properties					
3.1	Melting temperature	11357-3	D3418	C	C	F
3.3	Temperature of deflection at 1.8 MPa	75-1& 2	D 648	C	C	F
3.4	Temperature of deflection at 0.45 MPa	75-1& 2	D 648	C	C	F
3.7	CLTE*, flow direction, 23-55°C	11359-2	E228 or 831	E-4 1/K	1/C	1/F
3.8	CLTE*, transverse direction, 23-55°C	11359-2	E228 or 831	E-4 1/K	1/C	1/F
3.9	Flammability at 1.6mm Thickness	1210	(UL94) (UL94)	class (as HB, V-2, V-1, V-0) mm	 mm	 mm
3.11	Flammability•5V Thickness	10351	(UL94) (UL94)	class mm	class mm	class mm
3.13	Limiting Oxygen Index	4589	D 2863	%	%	%
4	Electrical Properties					
4.1	Relative permittivity, 100Hz	IEC 250	D 150			
4.2	Relative permittivity, 1MHz	IEC 250	D 150			
4.3	Dissipation factor, 100 Hz	IEC 250	D 150	E-4		
4.4	Dissipation factor, 1MHz	IEC 250	D 150	E-4		
4.5	Volume resistivity	IEC 93	D 257	ohm cm	ohm cm	ohm cm
4.6	Surface resistivity	IEC 93	D 257	ohm	ohm	ohm
4.7	Electric strength	IEC 243-1	D 149	kV/mm	kV/mm	V/mil
4.9	Comparative tracking index	IEC 112	-			
5	Other Properties					
5.1	Water absorption 24 hr immersion/23°C	2	D570	%	%	%
5.2	Water absorption saturation at 23°C	62	D570	%	%	%
•	Water absorption at 23°C/50% RH	62	-	%	%	%
5.3	Density	1183	D792	g/cm^3	g/cm^3	lb/ft^3
•	Specific gravity	1183	D792			

* CLTE = Coefficient of Linear Thermal Expansion
** TMA = Thermo-Mechanical Analysis

Chart reference: Wigotsky, Victor, "The Road to Standardization," Plastics Engineering.

Part B1

Bibliography

Selective Bibliography

Design Considerations for Injection Molded Parts

Dym, Joseph B., Injection Molds and Molding, Second Edition, Van Nostrand Reinhold Company.

Structural Control

Beer & Johnson, Mechanical Material.

Peterson, R.E., Stress Contribution Factors.

Flat plate, torsional deformation, and stress formulas thanks to:
Roark, Raymond, and Young, Warren, Formulas for Stress and Strain, McGraw Hill.

Assembly

"Designing Parts for Ultrasonic Welding," ©Branson Utrasonics Corporation, Danbury, CT.

"Ultrasonic Insertion," Branson Utrasonics Corporation, Danbury, CT.

"Vibration Welding," Branson Utrasonics Corporation, Danbury, CT.

Holtz, Richard, "Vibration Welding: Fast, Quiet, Efficient," Assembly Engineering, Hitchcock Publishing.

1987 SPE ANTEC, Chul S. Lee, Alan Dubin and Elmer D. Jones, "Short Cantilever Beam Deflection Analysis Applied to Thermoplastic Snap-Fit Design," Held in Los Angeles, California, USA.

3M, "Structural Adhesive Guide for Industrial Product Design and Assembly."

Plastic Materials

Arpaci, Vedal S., "Convection Heat Transfer," Addison-Wesley.

Forward Technology Industries, "Joint Designs."

Emabond Systems, "Emaweld, Electromagnetic Welding System for Assembling Thermoplastic Parts."

Gas Assist Molding

Sayer, Matthew, "Application of Gas Injection Technology," Cinpress, Ltd.

"Design Tips for Gas-Assisted Injection Molding," diagram of rib designs by Indra Baxi of Sajar Plastics, Plastics Design Forum.

Appendix II: ISO and ASTM Test Methods

Wigotsky, Victor, "The Road to Standardization," Plastics Engineering.

English/Metric Conversion Chart

To Convert English System	To Metric System	Multiply English Value by...
DISTANCE		
inches	millimeters	25.38
feet	meters	0.30478
MASS		
ounce (avdp)	gram	28.3495
pound	gram	453.5925
pound	kilogram	0.4536
U.S. ton	metric ton	0.9072
VOLUME		
$inch^3$	$centimeter^3$	16.3871
$inch^3$	liter	0.016387
fluid ounce	$centimeter^3$	29.5735
quart (liquid)	$decimeter^3$ (liter)	0.9464
gallon (U.S.)	$decimeter^3$ (liter)	3.7854
TEMPERATURE		
degree F	degree C	[(F)-32] / 1.8 = (C)
PRESSURE		
psi	bar	0.0689
psi	kPa	6.8948
ksi	MN/m^2	6.8948
psi	MPa	0.00689
ENERGY AND POWER		
in lbf	Joules	0.113
ft lbf	Joules	1.3558
kW	metric horsepower	1.3596
U.S. horsepower	Kw	0.7457
Btu	Joules	1055.1
Btu in (h ft^2 F)	W(m K)	0.1442
VISCOSITY		
poise	Pa s	0.1
BENDING MOMENT OR TORQUE		
ft lb	Nm	1.356
DENSITY		
lb/in^3	g/cm^3	27.68
lb/ft^3	kg/m^3	16.0185
NOTCHED IZOD		
ft–lb/in	J/m	53.4

BASF Corporation
Engineering Plastics
1 (800) 527-TECH

BASF: The Chemical Company. We don't make a lot of the products you buy. We make a lot of the products you buy better.®

BASF Corporation, headquartered in New Jersey, is the North American affiliate of BASF AG, Ludwigshafen, Germany. BASF is the world's leading chemical company. Our portfolio ranges from chemicals, plastics, performance products, agricultural products and fine chemicals to crude oil and natural gas. As a reliable partner to virtually all industries, BASF's intelligent solutions and high-value products help customers to be more successful. BASF develops new technologies and uses them to open up additional market opportunities. We combine economic success with environmental protection and social responsibility, thus contributing to a better future. Further information on BASF is available on the Internet at www.basf.com.

BASF
The Chemical Company