Designing With Thermoplastics

Dow Plastics

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Contents

The general contents of this Design Manual are listed below. There is also a comprehensive index at the back of the Manual that will be helpful in locating specific information.

Introduction

Designing with Thermoplastics

Although basic design practices for plastics are the same as those for other materials and can be found in any good engineering handbook, successful plastics design presents some unique challenges to the engineer. Whereas the specific structural property values for metals remain relatively constant over a wide range of time and temperature, the same values for plastics are greatly dependent on temperature, stress levels, and the part's life expectancy.

Additives to thermoplastics, such as those for ignition resistance, also affect the structural properties of plastics, notably ultimate tensile strength, elongation at break, and tensile modulus values. As a result, the designer must understand and apply the design principles unique to ignition-resistant or other thermoplastics with special additives – to achieve optimal part design, manufacturing, and performance.

The purpose of this manual is to provide information and formulas that will help designers solve many of the problems typically encountered when designing structural plastic parts. However, the formulas and data expressed in this manual should be considered only **general guidelines**. Specific parts should be designed with their own special performance and end-use requirements in mind.

The nature of polymer materials demands special attention to appropriate safety factors during part design. Further, it is strongly recommended that after all the science, mathematics, and experience have been applied to solving the design requirements of a part, prototype parts be produced and thoroughly tested under expected end-use conditions before committing the design to full-scale production.

Quality

Quality can be summarized as "fitness of a product for use." It is inherent in good design, and many of the features and requirements are encompassed in the following chapters. Some aspects of quality include assessing the finish and appearance of components as well as checking dimensions. Other considerations should include the capability of the product techniques (materials, machines, and tools) to produce the required standards of the component.

Quality cannot be "inspected-in" for any part, but must be considered from the earliest stages of design and specification.

Dow Plastics† is firmly committed to a policy of supplying quality materials that promote downstream product quality and reliability.

Quality Control

Every manufacturing concern should have some type of quality control section or philosophy. Besides an inspection department, responsible for checking parts and assemblies, there should be engineers who work with design and production engineers to set quality standards. The engineers should also compile quality schedules, inspection charts, and (usually) arrange statistical sample inspection. Of course, in the smaller manufacturing concern, the quality functions may well be part of the designer's responsibility. In larger concerns, quality control may be aligned to national or aerospace standards.

Quality control should apply throughout production: from checking supply materials through finished parts and dispatch areas. Quality engineers will generally be responsible, with production engineers, for machine capability trials. These can include monitoring all features of injection molding machines and operating cycles in order to comply with dimensional standards, freedom from short shots, flashes, burns, and other appearance defects, etc.

There are numerous authoritative manuals and national standards relating to quality control that can provide you with further details.

Health and Safety

The labels and instructions provided with Dow products normally give general working and health and safety information. They specifically mention any health and safety concerns and provide precautionary information relating to the safe handling and use of the product. Material Safety Data sheets for Dow products, and any additional information you may need are available from the Customer Information Group (CIG).

†Dow Plastics, a business group of The Dow Chemical Company and its subsidiaries.

Reliability

Reliability is defined as the probability that an item will perform the required function, under the stated conditions, for a stated period of time. The required function can be as low as one or two operations (for example, packing material), up to thousands or millions of operations (as in electrical and electronic components). The stated conditions should include such functions as environmental conditions and loading, speed of operation, possible abuse, and overloading.

When discussing reliability, you must also specify what you will consider a "failure." For example, if manufacturing plastic milk bottles, leakage of the contents may well be considered a disastrous failure. But what caused the leakage may range from a molding fault such as a short shot or a hairline bad weld line, to an incorrectly fitting screw cap, due to mismatching of fits or tolerances.

For operational parts of assemblies, reliability can also cover functional testing. Functional tests can include static or cyclic pressure testing, operational life testing, and various electrical properties tests. Reliability engineers also may design tests for physical properties ranging from abrasion resistance to combustibility. Some of these tests may be to national standards, for example, testing electrical switchgear.

As with quality control, there are many good handbooks on reliability you can refer to for further guidance.

The Team Approach

Effective design and engineering practices are never functions of material choice and use alone. Invariably, they are also influenced by the constraints of fabrication equipment, die or mold design and construction, operator skills, material flow patterns, and a host of other factors. For example, it is impossible to design parts in thermoplastics without relating the design requirements (shape, size, function, etc.) to the intended method of fabrication. A capable designer in plastics will be familiar with the opportunities and limits associated with injection molding, extrusion, thermoforming, and blow molding.

We strongly recommend a team approach for problem-solving involving design engineering. The team should include materials suppliers, engineering, production, quality control, marketing, and business professionals. And they should be able to comment as to user requirements and environments, required functional life, shape aesthetics and design, fabrication equipment, practical "real" economics, and resin performance. That circumstance is probably when it all happens "best" – in a most timely and productive fashion.

Applications Development

Dow Applications Development Engineers (ADEs) are in-the-field specialists who act as your link with Dow design capabilities worldwide. ADEs serve major industries, including automotive, electronics and telecommunications, building and construction, packaging, recreation, health care and appliances.

Since Dow ADEs have extensive knowledge of materials, as well as industry expertise, they can help you determine the best resin for your product, keep development costs to a minimum, and streamline your development process. In addition, they can help you with design considerations and provide detailed information concerning every Dow resin that is used in your specific application.

Design Support Resources

Dow's Technical Service and Development (TS&D) staff offers the expertise, resources and design support you need to succeed in an increasingly competitive industry. With mold flow analysis, stress analysis, and other fabrication tests, we can help you trouble-shoot and fine-tune while your part is still in the design stage. And that helps you avoid costly alterations once the part is in full production.

If you have questions about how Dow's design support and technical services can help you with your application, call 1-800-441-4DOW.

Chapter 1

Thermoplastic Resins

Selection and Performance
Environmental Considerations

Selection and Performance

This chapter contains summary performance data that will be helpful in making or confirming your initial polymer selection.

Table 1 on pages 10 and 11 compares selected properties of a number of widely used thermoplastics. The data are useful in making preliminary evaluations of thermoplastic materials.

Designers can choose to specify thermoplastic materials for various components, and often this can result in commercially successful products, superior to similar components in other materials.

The correct and easy selection of suitable components for manufacture in thermoplastics, and the choice of the correct material for each component, obviously needs the designer's awareness of the performance properties and attributes of thermoplastic materials, and how these properties can be used in solving many design problems.

This can involve knowledge of one or more of the following:

- Fabrication techniques such as injection molding, structural foam molding, gas-assisted injection molding, co-injection molding, blow molding, sheet extrusion, and thermoforming – together with any constraints inherent in the method, due to shape, size and cost control.
- Assembly methods such as snap-fitting, solvent bonding, ultrasonic and thermal welding, riveting and screw fastening – and how these can affect component design.
- The dimensional stability of the materials together with any influence on probable service life in normal and severe environments.
- Food contact regulations and flammability standards. These can help determine the suitability of designs for selected packaging, construction and other uses.
- Hydrolytic stability and sterilizability (using steam, ethylene oxide or radiation techniques) in order to ensure suitability in many household, industrial and medical applications.
- How ultraviolet stability affects the design in both exterior and interior applications.
- How finishing techniques can affect the suitability of the materials for a number of uses.

Knowledge of these and other performance attributes affect not only shape and functional suitability, but often also the economic role of thermoplastics in a given design or component.

Some of these attributes are described in detail in this design manual. Others are briefly mentioned. However, if you require further details, call 1-800-441-4369, in Canada, call 1-800-363-6250.

The other chapters in this manual provide basic data, design principles and formulas. These will help designers and engineers to make well-informed judgments regarding the use of engineering thermoplastics.

The selection of a particular resin can be influenced by many things: strength, stiffness, electrical and physical characteristics among others. Here are a few examples:

- **Elasticity –** If your product requires a fair degree of flexibility, you have a good choice from polyethylene, vinyl, polypropylene, acetal and nylon. You can also use some of the more rigid plastics so long as the section is correctly designed. See Chapter 2 for viscoelastic properties.
- **Ignition resistance –** In many electronic applications such as enclosures or connectors, plastic components must demonstrate ignition resistance. Standards such as Underwriters Laboratory 94 spell out specific ignition-resistant test protocols. Certain grades of polycarbonate, PC/ABS, ABS, nylon, and polysulfone are suitable. See page 32 for other details to consider.
- **Gears and bearings –** Highly stressed gears can be produced in nylon and acetal especially when reinforced with glass fillings. Other useful reinforcements include graphite and molybdenum disulphide. Acetal resin is good for small, precisely dimensioned gears.
- **Impact resistance –** Polycarbonate, ABS and polyphenylene oxide (in its impact-modified form) have good impact characteristics. See page 44.
- **Odor and taste –** These will be of concern to you if you design for the food industries, either in packaging or in food processing machinery. Polystyrene, polyethylene, ABS, acrylic and polysulfone are among the satisfactory resins for such uses.
- **Surface wear –** Scratch resistance does not necessarily equate with hardness. Acrylic, ABS and SAN resins generally have good resistance against scratches due to handling.
- **Temperature –** Some materials will be eliminated from your choice because of thermal restrictions. For products operating above about 250°C (482°F), the silicones, polyimides, hydrocarbon resins or mica may be required. At the other extreme, polyphenylene oxide can be used at temperatures as low as -180°C (-292°F). Refer to Thermal Properties on page 32 for other guidance.

Table 1 – Property[1] Comparisons of Selected Engineering Thermoplastics

Thermoplastics	Notched Izod S.I. J/m	Notched Izod English ft-lb/in	Notched Izod Metric kg cm/cm	Yield Strength S.I. MPa	Yield Strength English psi	Yield Strength Metric kg/cm²	Tensile Modulus S.I. GPa	Tensile Modulus English psi	Tensile Modulus Metric kg/cm²
MAGNUM* 342EZ ABS	160.0	3.0	16.0	40	5,700	400	2.2	320,000	22,500
MAGNUM 4220 ABS[2]	171	3.0	17.0	35	5,400	380	2.2	325,000	23,000
MAGNUM 9020 ABS	320	6.0	32.5	40	5,700	400	2.2	320,000	22,500
Acetal	65 - 120	1.0 - 2.0	6.5 - 12.0	70	9,700	680	3.6	520,000	36,500
Acrylic	40 - 130	0.8 - 2.5	4.0 - 13.0	40	5,500	340	1.7	250,000	17,500
Amor Nylon	70	10	7	65	9,700	680	2.8	410,000	28,500
Nylon 6,6	110	2.0	1	45	6,500	460	1.3	190,000	13,500
Polybutylene Terephthalate	40 - 60	0.8-1.0	4.0 - 6.0	55	8,100	570	1.9	280,000	19,500
PULSE* 1725 PC/ABS[2]	530	9.9	55	55	8,400	590	2.4	350,000	24,500
PULSE 830 PC/ABS	640	12	65	50	7,700	540	2.1	310,000	22,000
CALIBRE* 300-15 Polycarbonate[3]	850	16	85	60	9,000	630	2.2	320,000	22,500
CALIBRE 800-10 Polycarbonate[2]	640	12	65	60	8,800	620	2.1	305,000	21,500
STYRON* 498 Polystyrene	70	1.0	7.0	25	3,800	270	2.2	320,000	22,500
STYRON 685 Polystyrene	10	0.25	1.4	45	6,400	450	3.2	460,000	32,500
STYRON XL-8023VC Polystyrene	110	2.0	12	30	4,800	340	2.3	330,000	23,500
STYRON 6075 Polystyrene[2]	110	2.0	12	25	3,600	250	2.1	305,000	21,000
Polysulfone	70	1.0	7	70	10,100	710	2.5	360,000	25,500
ISOPLAST* 101 Polyurethane	1,200	22.0	120	50	7,000	490	1.5	220,000	15,000
ISOPLAST 101-LGF40-NAT Polyurethane	430	8.0	40	190	27,000	1,900	11.7	1,700,000	119,000
Polyphenylene Oxide (impact modified)	320-330	6.0 - 6.2	32 - 33	50	7,500	530	2.4	350,000	25,000
TYRIL* 880B SAN	27	0.5	2.8	82	11,900	840	3.9	570,000	40,000

[1]Typical property values; not to be construed as specifications
[2]Ignition resistant resin
[3]General purpose resin, no incorporated additives, 15 Melt Flow Rate
*Trademark of The Dow Chemical Company

Flexural Strength			Flexural Modulus			HDT @ 1.8 MPa			Light Transmittance
S.I. MPa	English psi	Metric kg/cm²	S.I. GPa	English psi	Metric kg/cm²	S.I. °C	English °F	Metric °C	S.I./English/Metric %
60	9,900	690	2.2	320,000	22,400	80	175	80	opaque
60	9,200	650	2.2	315,000	22,100	80	180	80	opaque
65	9,600	670	2.3	330,000	23,500	100	220	100	opaque
95	14,000	980	2.8	407,000	28,600	125	260	125	opaque
60	9,000	630	1.7	250,000	17,300	80	180	80	90
90	13,200	940	2.6	378,000	26,500	125	255	125	opaque
40	6,100	430	1.3	189,000	13,300	75	170	75	opaque
100	15,000	1,050	2.5	360,000	25,500	50-80	120-175	50-80	opaque
90	13,300	940	2.8	410,000	28,600	90	190	90	opaque
80	12,000	850	2.3	330,000	23,500	120	250	120	opaque
100	14,100	990	2.4	350,000	24,500	125	260	125	90
100	14,000	990	2.5	360,000	25,500	125	260	125	opaque
50	7,500	530	2.1	305,000	21,400	90	190	90	opaque
85	12,300	870	3.3	485,000	34,000	100	220	100	90
65	9,400	660	2.4	380,000	24,500	80	170	80	opaque
50	6,800	480	2.3	330,000	23,500	90	200	90	opaque
100	15,400	1,100	2.7	390,000	27,500	175	350	175	75
70	9,800	690	1.8	260,000	18,400	80	170	80	opaque
300	45,000	3,200	10.3	1,500,000	105,100	90	200	90	opaque
70	10,000	700	2.4	350,00	24,500	90-135	190-275	90-135	opaque
100	16,000	1,100	4	580,000	40,800	100	220	100	87

Environmental Considerations

During the manufacture of our basic materials, Dow Plastics uses processes and techniques with due consideration and regard to the environment. Such considerations include factory planning and layout, machine capability studies, and employing processes that are economical in the total usage of power and in controlling emissions.

We are also highly conscious of our need to contribute to safe handling and transportation of all materials at all times, and the need for safety and training of operators together with the overall safety of other personnel and, of course, the general public.

These concerns make good sense and reflect our aims and policies to provide high quality products at the right price, backed by sound, helpful knowledge and advice.

We can assist you in designing plastic parts so that you can also take into account factors that permit the best and most economic use of your plant, labor and materials. Here are some guidelines for you to consider:

- Use as little material as possible. This does not mean that you should thin down all wall thicknesses to extremes, or design ridiculously tiny components; but the correct and intelligent use of thin wall sections, suitably stiffened with ribs and gussets, will be economically viable. This philosophy can result in more economic tooling and machinery, and less usage of power and shorter cycle times. It also implies that you should choose the best material for the job. And it implies that you should consider the best methods of assembly. For example, the use of snap fitting components will reduce or eliminate the need for bolts or screw fittings.
- Design with recycling in mind. This can affect your manufacturing plant, not only during normal production runs but also during the tool tryout, prototype and preproduction stages, reducing the need for disposal of scrap. And it can also affect your customers in their future purchases and planning.
- Consider how your customers will dispose of your products. Of course, there are long- and short-term factors. The intended life of your design or component may be a matter of weeks or may be measured in tens of years. While you should consider the future effects of disposal on the environment – it need not impose a constraint on your designs.

With these guidelines in mind, we at Dow Plastics encourage designs that:

- Minimize the number of different types of plastic in the component, aiding recycling.
- Reduce the combination of plastic and paint or decorative strips and finishes. This also aids recycling and disposal.
- Allow easy disassembly or replacement of the component.
- Minimize the number of separate pieces in any assembly or sub-assembly.
- Optimize wall thickness to reduce material usage, while still meeting the key specification or primary functions.
- Identify the various types of plastic in an assembly by labeling or molding-in identification.
- Minimize the number of non-plastic inserts.
- Allow a part to be able to carry out more than one function.

Keeping these guidelines in mind, your plastics designs can be both proactive and beneficial in addressing environmental issues.

Chapter 2

Viscoelastic Properties

Definitions
Modeling
Rheology

Definitions

A designer who has training in traditional engineering materials and who is now designing in plastics should have a grasp of the general concept of viscoelastic responses in order to understand the behavior of thermoplastics. When discussing viscoelasticity, the following definitions are relevant:

Viscoelastic Material

A material whose response to a deforming load combines both viscous and elastic qualities. The common name for such a material is "plastic."

Stress

The force per unit area, which is acting on a material and tending to change its dimensions. It is the ratio of the amount of force divided by the cross-sectional area of the body resisting that force. In engineering formulas, stress is represented by "σ." Among other types, stress can be tensile – as when the body is subject to a tension load; compressive – when the body is subject to compression loading; or shear – when the body is subject to a shearing load.

Strain

The percentage deformation of a body when subjected to a load. Tensile strain occurs when there is an increase in the original dimension, and is numerically expressed as the change in length per unit length of the specimen under load. It is represented in formulae by ϵ. There can also be compressive, shear, and volumetric strains.

Young's Modulus

This is the ratio between stress and strain, i.e., stress divided by strain and is denoted by "E," as shown in Figure 1.

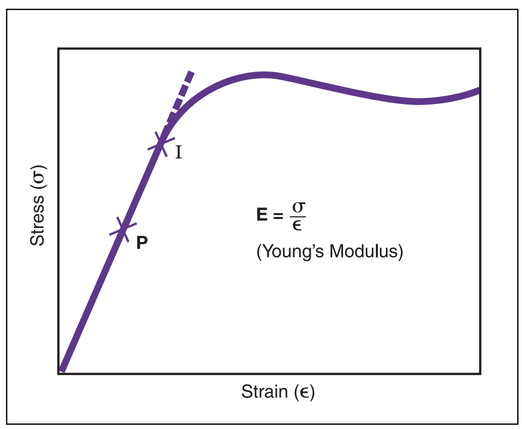
Elastic Material

A material that deforms under stress, but regains its original shape and size when the load is removed. A practical example of elastic material is any spring working within its limits. For completely elastic materials, stress is directly proportional to strain. However, when a material has viscous as well as elastic properties (as in plastics), deviation from this linear relationship occurs.

Proportional Limit

This is the point on the stress-strain curve (for a material involving both elastic and viscous components) where deviation from the linear relationship occurs. In Figure 1, this point is marked as "P." The point at which the proportional relationship deviates is often expressed in terms of stress (σ). Deformation is often expressed as strain (ϵ). Young's Modulus, or tensile modulus, represented by E, is defined as $\frac{\sigma}{\epsilon}$, as shown in Figure 1.

Figure 1 – Proportional Limit



Elastic Limit (Point I)

The point on the stress-strain curve that marks the maximum stress a material can absorb and still recover (with no permanent deformation) to its original dimensions. Recovery may not be immediate, and the elastic limit may occur at stress levels higher than the proportional limit. In Figure 1, the proportional limit is marked as “P.”

Viscous Material

A material which, after being subjected to a deforming load, does not recover its original shape and size when the load is removed. An example is a piston in a dashpot containing a viscous fluid. If a load is applied to move the piston in the dashpot, the piston will not return to its original position after the displacing load is removed, unless a returning load is applied – opposite to the original load.

Rheology

The science of the deformation and flow of materials under load.

Modeling

The Spring, Dashpot, and Voight-Kelvin Model

Most of the published data for plastic materials give only short-term, load-to-failure test results. So, for a viscoelastic material such as a thermoplastic, these data tend to reflect values that are predominantly affected by the elastic response of the plastic. However, it also is important to test and evaluate the viscous portion of the polymer's behavior (as in response to long-term loading) to determine whether any detrimental long-term effects will occur.

Long-term behavior is analyzed experimentally by two methods. In the first method the test specimen is subjected to a constant stress and the change in strain is monitored to determine creep. In the second, the strain on the specimen is held constant while the change in stress is monitored to determine stress relaxation.

One common way of representing how a plastic material responds to loading is by the use of a mechanical model, called a Voight-Kelvin model. This consists of an assembly of springs and dashpots – each dashpot being a cylinder containing a viscous fluid in which a piston is immersed.

The Spring A, in Figure 2, represents the elastic portion of a plastic material's response. When a load is applied to the spring, it instantly deforms by an amount proportional to the load. And when the load is removed, the spring instantly recovers to its original dimensions. As with all elastic responses, this response is independent of time.

The spring, of course, has a load limit. If that limit is reached, the spring fails or breaks. Each material represented has a load limit specific to itself.

The Dashpot A, shown in Figure 3, represents the viscous portion of a plastic material's response. The dashpot consists of a cylinder holding a piston that is immersed in a viscous, fluid-like material. When a load is applied to the dashpot, it does not immediately deform – the piston does not immediately move.

Figure 2 – The Spring (Elastic Response)

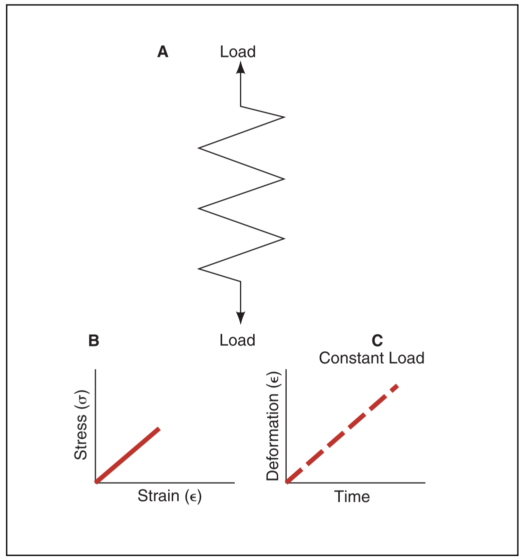
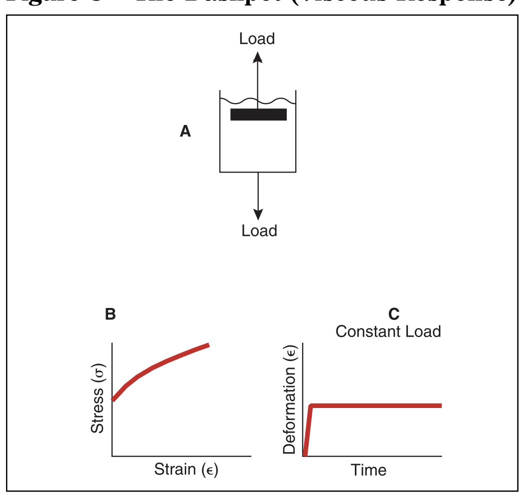


Figure 3 – The Dashpot (Viscous Response)



However, if the application of the load continues, the viscous material surrounding the piston is eventually displaced and the dashpot does deform. This occurs over some period of time, not instantly, and the length of time depends on both the load and the rate of loading.

Viscous response, therefore, has a rate of response that is time-dependent. Other important factors affecting the extent of deformation are environmental temperature and the length of time for which the load is applied.

Figure 4 – Typical Voight-Kelvin Mechanical Model

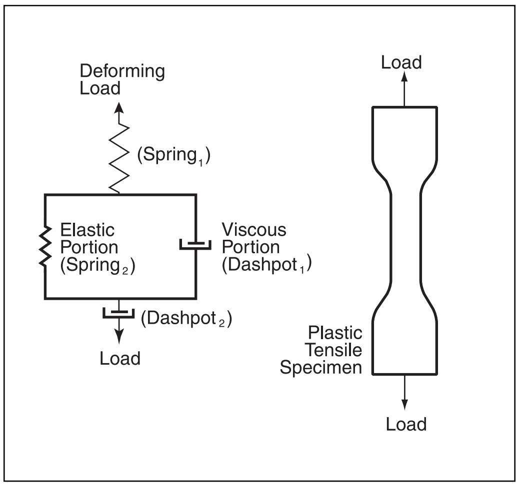
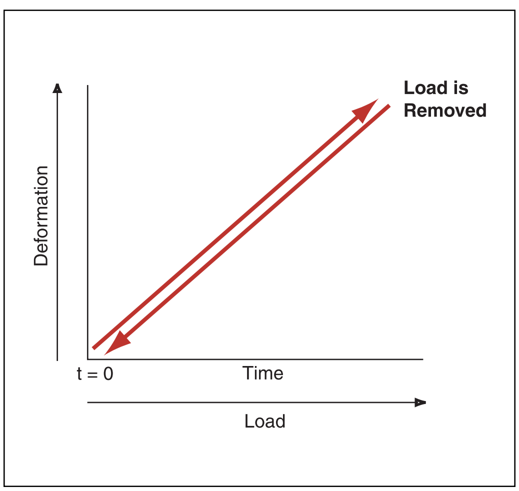


Figure 5 – Viscoelastic Response to Short-term Loading (Within Proportional Limit)



To represent the response behavior of a viscoelastic material, springs and dashpots are combined to form a Voight-Kelvin model, as shown in Figure 4. When a load is applied to this combination, Spring 1 immediately deforms to a given extent (proportional to the stress), but neither Dashpot 1 nor Dashpot 2 can move significantly in the same short period of time. Therefore, if the load is removed before the dashpots move significantly, the spring recovers from its deformation, and the model returns to its original position.

That sequence indicates what happens when, in testing a specimen of a plastic material, the load is applied and then removed before the proportional limit is reached. Figure 5 graphically represents how this response would be shown on a stress-strain curve.

If the same load is applied for a longer time, the model responds quite differently. Instantly, as before, Spring 1 deforms to an extent. Then, after a period of time – when the viscous material in the two dashpots can no longer resist the load, thus permitting the pistons to move – the dashpots also deform (flow). Once the dashpots flow, the stress-strain curve becomes non-linear (not proportional).

When the load is removed after this type of deformation, Spring 1 regains its original dimension, but the deformation of Dashpot 1 and Dashpot 2 is not instantly recovered. Dashpot 1 slowly returns to its original position – in time-dependent relaxation – under the compressing effect of Spring 2 returning to its original dimension. Dashpot 2, not having a spring in parallel with it to provide recompression, remains permanently deformed.

Figure 6 graphically represents the deformation of a Voight-Kelvin mechanical model.

A thermoplastic material can be represented by a mechanical model combining the appropriate number and values of springs and dashpots. Increasing the number of components in the model increases the accuracy of the theoretical response. When evaluating the response, it is important to remember that the theoretical viscosity of the fluid in a dashpot is dependent on temperature – similar to the viscous response of a plastic being affected by changes in temperature that alter the viscoelasticity of the plastic.

The response of a plastic is also affected by the duration of time for which the load is applied, and by the rate at which loading is applied. In the model, the longer time a load is applied, the greater is the amount of flow that occurs in the dashpot until the piston finally pulls free of the fluid, which is analogous to the failure of a part.

Additional information about the mechanical properties and behavior of thermoplastics is readily available. Excellent sources include *Introduction to Polymer Viscoelasticity*, by John J. Aklonsis, William J. MacKnight, and Mitchell Shen, published in 1972 by John Wiley and Sons, Inc.; and *Textbook of Polymer Science*, by Fred W. Billmeyer Jr., published in 1962 and 1971 by John Wiley and Sons, Inc.

Figure 6 – Viscoelastic Response to Long-Term Loading

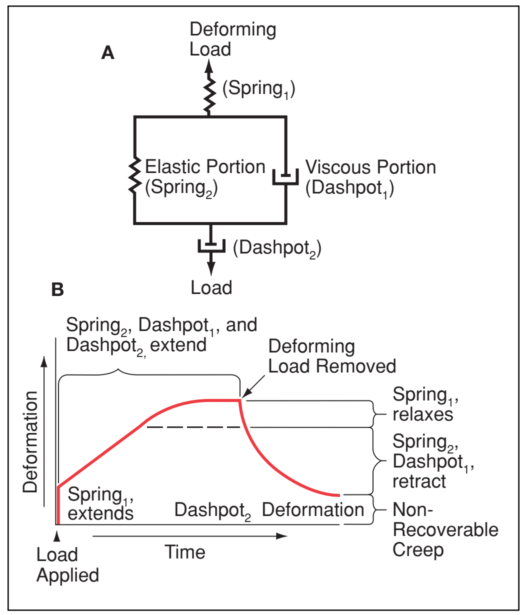


Figure 7 – Elastic Element

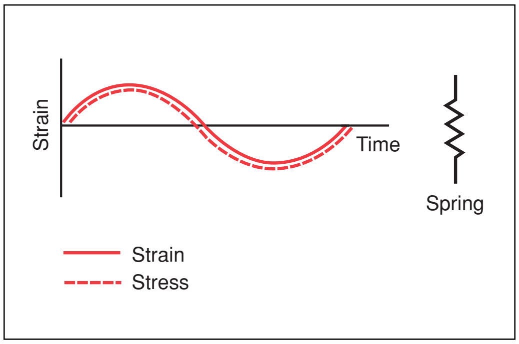


Figure 8 – Viscous Element

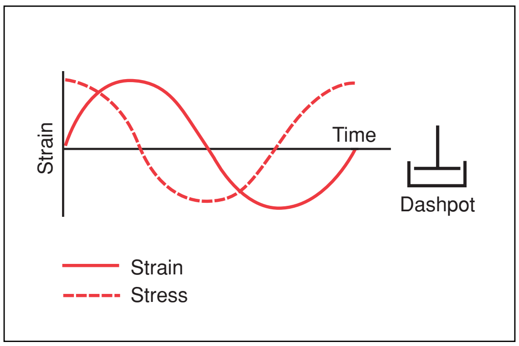
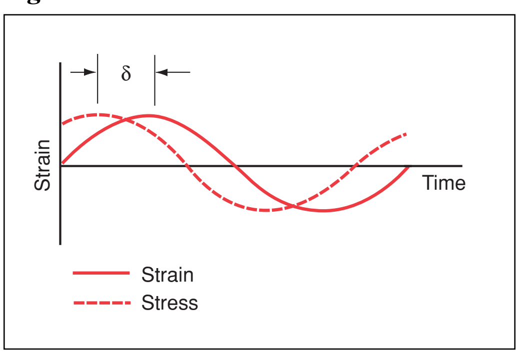


Figure 9 – Viscoelastic Element



A useful technique for measuring the viscoelastic properties of plastic materials employs a dynamic mechanical spectrometer (DMS). DMS evaluation measures the viscosity of a plastic in either of two conditions:

- where temperature is monitored and shear rate is varied, or
- where shear response is monitored and temperature is varied.

The resulting data indicate the difference between the viscous (dashpot) and elastic (spring) portions of the response of the plastic material to a shearing load.

For elastic materials that can be modeled by a spring (as in Figure 2, page 18), the shearing stress occurs in phase (proportional) with the strain or deformation, as shown in Figure 7, i.e., the strain occurs simultaneously with the stress.

On the other hand, for viscous materials that can be modeled by a dashpot (Figure 3, page 18), the shearing stress is 90° out of phase with the strain, as shown in Figure 8. In this case, the strain occurs some time after the stress is applied – an example of a time-dependent response.

In the case of a viscoelastic (plastic) material, the effects represented by Figures 7 and 8 are combined. The resulting rheological response thus reflects the same stress, but the strain is out of phase – occurring in a time-dependent response – by an amount δ, as shown in Figure 9.

Rheological data are valuable in indicating how much effect the viscous portion of a thermoplastic material's response will have on its final physical properties. Such data can also indicate whether a polymer has been inappropriately processed (by comparison between rheological spectra before and after the processing). Other practical uses of DMS data include the interpretation of processing conditions by determining the viscosity of the plastic under varied conditions. They also include the determination of molecular weight distribution, and the determination of dimensional stability of the fabricated plastic.

Chapter 3

Physical Properties

Density
Optical Properties
Physical Characteristics
Electrical Properties
Resistance to End-Use
Conditions and
Environments
Thermal Properties
Molding Properties

Density

Test Specimen Evaluation

Please note that the information and data presented in this manual, both general facts and property values, are for reference only.

Data based on test specimen evaluations have great practical value. But you can only be completely assured of design and product integrity by producing and testing prototype parts in the actual proposed conditions of fabrication and service.

Density and Specific Volume

Density is the measure of the weight per unit volume of material at 23°C (73°F) usually expressed as grams per cubic centimeter (gm/cm³) or as pounds per cubic inch (lb/in³).

In order to determine the relationship between the weight and the volume of material for any given part, it is necessary to know the density of the given material.

Specific Gravity

Specific gravity is the ratio of the weight of a given volume of material compared to an equal volume of water, both measured at 23°C (73°F).

Basically, you can think of:

$$\text{specific gravity} = \frac{\text{density of material @ } 23°C}{\text{density of water @ } 23°C}$$

At 23°C, water has a density slightly less than one.

The following conversion from specific gravity to density can be used.

Density, g/cm³ @ 23°C =
Specific gravity @ 23°C x 0.99756

Specific gravity is useful to the designer in calculating cost-weight and strength-weight ratios.

Table 2 – Density and Specific Volume of Various Thermoplastics[1]

Resin	Density			Specific Volume		
	S.I. g/cm³	English lb/in³	Metric g/cm³	S.I. cm³/g	English in³/lb	Metric cm³/g
ABS	1.05	0.038	1.05	0.95	26.3	0.95
Acetal	1.40	0.051	1.40	0.71	19.7	0.71
Acrylic	1.16	0.042	1.16	0.87	23.8	0.86
HDPE	0.96	0.035	0.96	1.04	28.8	1.04
LDPE	0.92	0.033	0.92	1.09	20.1	1.09
Nylon	1.15	0.042	1.15	0.87	24.0	0.87
PBT	1.30	0.047	1.30	0.77	21.3	0.77
PC	1.20	0.043	1.20	0.83	23.0	0.83
PP	0.90	0.033	0.90	1.10	30.7	1.10
PS	1.05	0.038	1.05	0.95	26.3	0.95
SAN	1.08	0.039	1.08	0.93	25.6	0.93

[1]Typical average values are shown. Consult product literature for exact values of specific resin grades.

Optical Properties

Luminous Transmittance

This is a principal indication of the transparency of a material, defined as the ratio of the amount of light transmitted through the material to the amount of incident light. Table 3 lists the luminous transmittance values of several transparent materials. Higher values indicate greater light transmittance or transparency.

Table 3 – Typical Luminous Transmittance of Transparent Materials

Material	Light Transmission[1] %
Glass	92
PMMA	92
CALIBRE polycarbonate	90
STYRON polystyrene	90
Cellulose acetate	89
TYRIL SAN	87

[1]Specimens 0.3 mm thick; tested by ASTM D 1003 procedures

Refractive Index

The refractive index of a material is another way of optically classifying clear materials. It is the ratio of the velocity of light in a vacuum, to its velocity in the material under study. Refractive index also can be defined as the ratio of the angle of incidence to the angle of refraction (i.e., sine incidence angle divided by sine refraction angle). Like luminous transmittance, refractive index is an important property to be considered in the design of optical systems.

Table 4 gives the refractive indices of a number of transparent materials. Lower values indicate that less refraction or distortion occurs as light passes through the material.

Table 4 – Typical Refractive Index of Transparent Materials

Material	Refractive Index[1] %
Polysulfone	1.633
CALIBRE polycarbonate	1.58
TYRIL SAN	1.57
Glass	1.52
Cellulose acetate	1.50
PMMA	1.49

[1]Tested by ASTM D 542

Haze

Haze is the percentage of transmitted light which, when passing through a specimen, deviates from the incident beam by forward scattering. Lower haze values imply greater transparency.

Haze is an important property when designing for a transparent "sight" application, in which observers must be able to see inside or through a part easily and clearly. If a material has a high haze value, it will have decreased transparency – making it more difficult to see inside or behind the "sight" part. A part with a high haze value will still transmit light, but images may appear foggy or blurred.

Typically, polycarbonate resins have a haze value of about 0.5 to 2.0%. Haze value ranges for other transparent materials are:

Polystyrene	0.1 to 3.0%
Styrene acrylonitrile	0.6 to 3.0%
Polymethylmethacrylate	1.0 to 3.0%
Cellulose acetate	0.5 to 5.0%
Glass	0 to 0.17%

Yellowness Index

Yellowness index (YI) is a numerical representation of how yellow a material is in comparison with a "clear" water-white standard. Lower YI values indicate greater clarity. The YI of polycarbonate resins is generally about 0.5 to 2.0. This is slightly less yellow than most other commercially available transparent polymers, which typically have YI values of 1.0 to 3.0.

Polymerization processes commonly induce a slight yellow or straw hue in the resins produced. Dow Plastics follows a practice normal in the manufacture of transparent polymers by making some materials that have a small amount of blue tint added to mask the yellowness. We can also supply resins without the added blue tint, in their "natural" form.

The presence of non-polymerized constituents or degraded material in a resin increases its YI. The YI value thus also indicates the statistical "cleanliness" of the final polymer. Excessive heat or shear stress during the fabrication processes tends to increase the YI of a natural resin. Thus, the normal injection molding conditions for a natural resin tend to increase the YI of the material.

To avoid raising the yellowness of a part significantly, fabricators should be careful to:

- Avoid excessive heat (caused by excessive melt temperatures and/or excessive length of exposure time at higher temperatures).
- Limit the amount of incorporated regrind material to the recommended maximum level of 25%, because regrind (with its heat history) tends to increase the YI.

Physical Characteristics

Abrasion Resistance

Abrasion resistance, tested by ASTM D 1044 procedures, is measured by applying a Taber Abrader with a 250 g weight and a CS 10-F textured abrader to the test specimen for a set number of cycles, and then measuring changes in specimen volume and transparency.

Table 5 shows typical abrasion resistance values for a number of selected thermoplastics. Lower values equate with greater abrasion resistance.

Hardness

To measure Rockwell hardness by ASTM D 785 procedures, an indenter is loaded with a minor load, a major load, and then again with the minor load. The increased depth of the impression made on the specimen is then measured. Thus, Rockwell hardness values indicate a material's resistance to surface deformation – thus greater hardness.

Table 6 shows the relative ranking of major thermoplastics from softest to hardest.

Table 5 – Abrasion Resistance of Selected Thermoplastics

Material	Abrasion Resistance[1]
PE	4.5×10^{-3}
PP	4.3×10^{-3}
PC	4.3×10^{-3}
Acetal	4.0×10^{-3}
Nylon 6/6	1.6×10^{-3}
PU	1.5×10^{-3}

[1]Grams removed after 100 cycles

Table 6 – Ranking of Selected Thermoplastics by Hardness

Ranking	Material
Softest	HDPE
↓	ABS
	Polysulfone
	PBT
	PC
	GPPS
	PET
	Acrylic
	Nylon 6
Hardest	Thermoplastic Polyimide

Coefficient of Friction

The coefficient of friction, determined by ASTM D 1894, numerically represents the resistance to movement when moving against another surface. Values are given for both static friction (the limiting friction between surfaces just before motion occurs) and kinetic friction (the friction after motion has occurred). The coefficient of friction is the ratio of the limiting friction to the normal reaction between the moving surfaces.

Table 7 lists the dynamic coefficient of friction of selected thermoplastics against steel surfaces. The lower the value, the "more slippery" the material.

Table 7 – Typical Coefficient of Dynamic Friction of Selected Plastics vs. Steel

Material	Coefficient
PC	0.55
ABS	0.5
SAN	0.5
Nylon	0.4
PMMA	0.4
PS	0.4
PPE	0.35
PP	0.33
HDPE	0.26

Electrical Properties

Figure 10 – Dielectric Strength – Typical Test Configuration

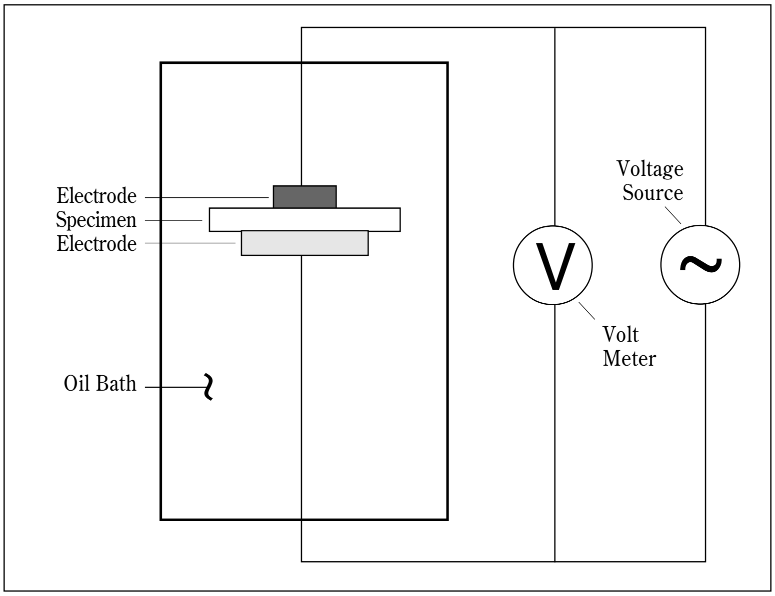


Table 8 – Dielectric Strength – Typical Values (ASTM D 149)

Material	MV/m	Volts/mil	kV/mm
ABS	13.8 - 19.7	350 - 500	13.8 - 19.7
ABS – 20% glass filled	17.7 - 18.1	450 - 460	17.7 - 18.1
Nylon 6/6	23.6[1]	600[1]	23.6[1]
Polycarbonate	15	380	15.0
Polycarbonate – 10% glass filled	20.9	530	20.9
Polypropylene	23.6	600	23.6
Polyphenylene ether – impact modified	20.9	530	20.9
Polystyrene – general purpose	19.7 - 22.7	500 - 575	19.7 - 22.7
Polystyrene – impact modified	21.7	550	21.7

[1]Dry as molded (approximately 0.2% moisture content).

Dielectric Strength

Dielectric strength is the maximum voltage a material can withstand without conducting electricity through the thickness of the material. Higher values indicate greater insulating efficiency. Test results will vary with the following: sample thickness, rate of voltage increase, duration of test and temperature. Figure 10 shows a typical test configuration. Table 8 represents typical values.

Volume Resistivity

Volume resistivity is a measurement of the resistance to the conduction of electricity provided by a unit cube of a material, at a given temperature and relative humidity. It is also described as the ratio of the voltage applied to one face of the specimen to the voltage exiting the opposite face of the cube. Higher values indicate greater insulation effectiveness. See Table 9 for typical values of volume resistivity.

Table 9 – Volume Resistivity – Typical Values (ASTM D 257)

Material	ohm-cm
ABS	1.0×10^{16}
ABS – 20% glass filled	1.0×10^{15}
ABS – ignition resistant	1.0×10^{14}
Nylon 6/6	1.0×10^{15}
Polycarbonate	1.0×10^{16}
Polycarbonate – 10% glass filled	6.0×10^{15}
Polypropylene	1.0×10^{16}
Polyphenylene ether – impact modified	1.0×10^{17}
Polystyrene – general purpose	1.0×10^{17}
Polystyrene – impact modified	1.0×10^{16}

Surface Resistivity

Surface resistivity is the resistance of a material to the conduction of electricity across its surface. As with volume resistivity, higher values indicate the material is less likely to allow a current to travel across its surface.

Dielectric Constant

Dielectric constant is the ratio of capacitance of a capacitor in which the specimen acts as the dielectric to the capacitance of a capacitor with dry air as the dielectric. It is also termed permittivity. Lower values indicate greater insulating ability of the material.

Dissipation Factor

The dissipation, or power factor, is the ratio of the power dissipated in an insulating material to the product of the effective voltage and current.

Dissipation factor = tan (90°– t)

where t is the phase angle between applied voltage and current within the material being tested.

The dissipation factor indicates an insulating material's usefulness in minimizing power loss caused by electrical heating. Lower values equate with greater system efficiency.

Arc Resistance

This property is stated as the length of time a specimen resists the formation of a continuous, conducting path across its surface. Higher values indicate increasing ability as an insulator.

Resistance to End-Use Conditions and Environments

Water Absorption

Most polymers produced by condensation polymerization, such as polycarbonate, are hygroscopic. They absorb water from direct exposure or from the water vapor present in the air. Parts fabricated from hygroscopic resins will also absorb water.

This behavior is important to understand because:

- To some extent, moisture absorbed by a finished part will affect part performance.
- Unless removed by drying before the processing begins, moisture absorbed by a resin before fabrication can cause serious degradation of properties during the molding process.

Weatherability

Ultraviolet stabilization can greatly extend the retention of key physical properties such as impact strength and appearance. The Dow Plastics product family includes several UV-stabilized resins.

Chemical Resistance

The practical chemical resistance of a material relates not only to exposure to the reagent, but also to the amount of stress on the part, the environmental temperature, and the duration of exposure. You should consider these factors in any application involving adverse environments. If your application involves unusual exposures or severe chemical environments, test samples or prototype parts should be fabricated, exposed to the actual materials and conditions of use, and then evaluated before full production.

Critical Stress

When a part made from a thermoplastic resin is designed for working in an environment involving chemical exposure, the amount of stress on the final part is critical, as mentioned previously. The more highly stressed the part, either from molded-in stress or from external loading in the final application, the more susceptible the part will be to chemical attack.

The critical stress – the maximum stress the material can withstand in air at 73°F (23°C) and 50% RH – is a reference point that can vary significantly with many factors. These factors may include temperature, the nature of the solvent, whether the strain is introduced by internal or external stress, and the resin's melt flow rate.

Solubility

Most chemical handbooks include solubility data for chemicals in general use. Such data can be used to estimate how a thermoplastic part will be affected by exposure to a particular chemical, by comparing the solubility of the thermoplastic with that of the chemical under consideration. If the solubility of the chemical is within 1.5 $(cal/cm^3)^{1/2}$ of the solubility of the thermoplastic, the chemical is probably aggressive, and parts made from that thermoplastic are not likely to be suitable for use when exposed to the chemical.

Gas Permeability

Gas permeability, or transmission, often must be taken into account when designing packaging and certain other associated applications. The value is determined by measuring the amount of a specific gas that passes through a volume of the material being tested, in a given time and under predetermined conditions.

The gases most frequently used in testing are nitrogen (N_2), oxygen (O_2), and carbon dioxide (CO_2).

Thermal Properties

Heat Distortion Temperature

Heat distortion temperature (HDT), measured by ASTM D 648 (ISO 75) test procedures, is the temperature at which an applied load causes a test bar of the plastic to deflect 0.25 mm. Like other thermal properties, HDT is sensitive to test variables.

In testing a resin for this value, there is particular sensitivity to the annealed/unannealed condition of test specimens. HDT values therefore should be used only to screen candidate materials, rather than as definitive guides for material selection.

Thermal Conductivity

The thermal conductivity of a material is a measure of the ability to transmit heat through the material. It is the same as the “K” factor for insulation, and is related inversely to the “R” value or thermal resistance.

The thermal conductivity of a polymeric material changes as the temperature changes. As the molecules of the polymer heat up, they vibrate at a higher frequency, enabling more energy (in the form of heat) to be transferred through the polymer. For that reason, thermal conductivity is generally reported at two temperatures: 0°F (-18°C) and 212°F (100°C).

Thermal conductivity values for a number of thermoplastics are shown in Table 10 and Figure 11.

Coefficient of Linear Thermal Expansion

The coefficient of linear thermal expansion (CLTE) of any material is the change in the material’s length per unit change in temperature. Typically, the CLTE of a material will increase with temperature. Values of CLTE for thermoplastics are typically 2 to 10 times larger than those for metals and glass. When designing parts where two different materials will be in fixed contact, allowances must be made for differences in CLTE to prevent warpage, breakage, or other damage or distortion of the finished article.

Vicat Softening Point

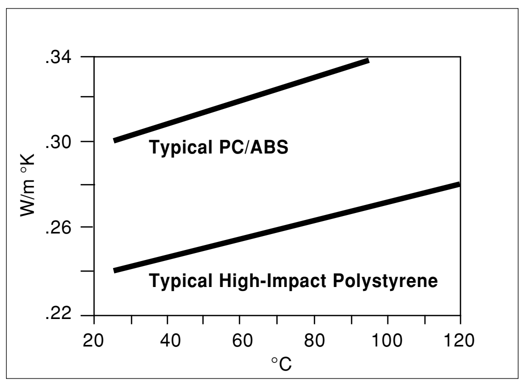
The Vicat softening point, measured by ASTM D 1525 (ISO 306) procedures, is the temperature at which a flattened needle of 1 mm^2 cross section, and under a specified constant load, penetrates a specimen of the plastic to a depth of 1 mm. It is useful as a rough comparative guide to a resin’s resistance to elevated temperatures.

Table 10 – Typical Thermal Conductivity, Selected Thermoplastics

	Thermal Conductivity[1]	
Material	W/m°K	BTU - in/hr ft^2 °F
PS	0.144	0.999
PBT	0.158	1.096
Nylon 6	0.173	1.249
PMMA	0.187	1.303
PC	0.202	1.401

[1]Data obtained from published literature

Figure 11 – Thermal Conductivity Change with Respect to Temperature



Maximum Use Temperature

The maximum use temperature of a plastic can be expressed as its relative thermal index in maximum °F or °C, as tested according to Underwriters Laboratory Test Method UL 764b.

Aging tests shows that when a material is used at or below its relative thermal index, the material's electrical and mechanical properties do not degrade significantly over the intended life of the final product.

Note that different thermal indices may be assigned, based on the thickness used and the properties evaluated.

Combustibility

The ignition resistance of a plastic is rated according to Underwriters Laboratory Standard 94 (UL94)[1].

The UL ratings represent the results of two separate tests. In the ratings from the first test (V-0, V-1, V-2, or HB), V-0 indicates the most ignition resistant material and HB indicates the least resistant. The rating from the second test (5VA), is added to the first rating if the plastic passes the second test. Thus, a combination of V-0 and 5VA is the highest UL94 rating possible, and HB is the lowest.

Ignition resistant (IR) resins are a large and important segment of the thermoplastic market.

Smoke Generation Limiting Oxygen Index

A limiting oxygen index (LOI) value represents the minimum concentration of oxygen (expressed as percent by volume) in a mixture of oxygen and nitrogen that will support flaming combustion of a material that is initially at room temperature. A higher value indicates a less flammable material.

[1]UL94 flammability ratings are based on small-scale tests and are not intended to reflect hazards presented by resins under actual fire conditions.

Table 11 – Typical Coefficient of Linear Thermal Expansion, Selected Thermoplastics

Material	mm/mm/°C	in/in/°F
SAN (styrene-acrylonitrile)	5.4×10^{-5}	3.0×10^{-5}
Polyethylene	5.9×10^{-5}	3.3×10^{-5}
Polymethylmethacrylate	6.3×10^{-5}	3.5×10^{-5}
Polystyrene	6.7×10^{-5}	3.7×10^{-5}
Polycarbonate (unfilled)	6.8×10^{-5}	3.8×10^{-5}
Polycarbonate (10% glass filled)	3.8×10^{-5}	2.1×10^{-5}
PBT (polybutylene terephthalate)	7.4×10^{-5}	4.1×10^{-5}
Nylon 6 (unfilled)	8.3×10^{-5}	4.6×10^{-5}
Nylon 6 (10% glass filled)	4.9×10^{-5}	2.2×10^{-5}
ABS (acrylonitrile-butadiene-styrene)	9.0×10^{-5}	5.0×10^{-5}

Molding Properties

Spiral Flow

Spiral flow data are obtained by using a standard spiral mold and measuring the distance the material has flowed under test conditions. The data allow a better understanding of a polymer's flow behavior. Machine, mold, and process conditions all influence the results of spiral flow tests.

A polymer's melt flow rate (MFR) is an indicator of its flow capabilities under given conditions. A higher MFR indicates an easier flowing resin.

Of course, processing at increased melt temperature, or with higher injection pressure, will increase the length of the polymer's flow, and varying either of these conditions is one method for achieving flexibility with any given thermoplastic resin.

Spiral flow and MFR data only provide a very rough guide to the processability of a resin. To accurately compare the flow characteristics of two resins, a complete curve of viscosity versus shear rate is required for each resin.

Mold Shrinkage Value

The mold shrinkage value reflects the amount of contraction from the actual mold dimensions that a finished part exhibits after removal from the mold and cooling to room temperature 23°C (73°F) for 48 hours.

It is possible to obtain less shrinkage by close control of all processing conditions. However, because of the slower cycles involved in such control, economics are usually adversely affected. The addition of fillers and/or reinforcements can also decrease mold shrinkage. In all practicality, however, part and mold design must take the stated amount of shrinkage into account.

Table 12 gives mold shrinkage values of several representative thermoplastics.

Table 12 – Mold Shrinkage Values of Various Thermoplastics[1]

Resin	mm/mm (in/in)
ABS	0.004 – 0.006
Acetal	0.015 – 0.025
Acrylic	0.003 – 0.006
HDPE	0.020 – 0.030
LDPE	0.015 – 0.030
Nylon	0.010 – 0.020
PC	0.005 – 0.007
PC/ABS	0.004 – 0.006
PP	0.012 – 0.020
PS	0.005 – 0.007
SAN	0.004 – 0.006

[1]Typical average values are shown. Consult product literature for exact values of specific resin grades.

Chapter 4

Mechanical Properties

Tensile Properties
Flexural Properties
Poisson's Ratio
Fatigue
Compressive Properties
Shear Strength
Impact Strength
Tensile Stress-Strain Behavior
Creep
Relaxation

Tensile Properties

Among the many mechanical properties of plastic materials, tensile properties are probably the most frequently considered, evaluated, and used throughout the industry. These properties are an important indicator of the material's behavior under loading in tension. Tensile testing provides these useful data: tensile yield strength, tensile strength at break (ultimate tensile strength), tensile modulus (Young's modulus), and elongation at yield and break. Figure 12 shows a typical stress-strain curve for a ductile thermoplastic, with the following important points indicated.

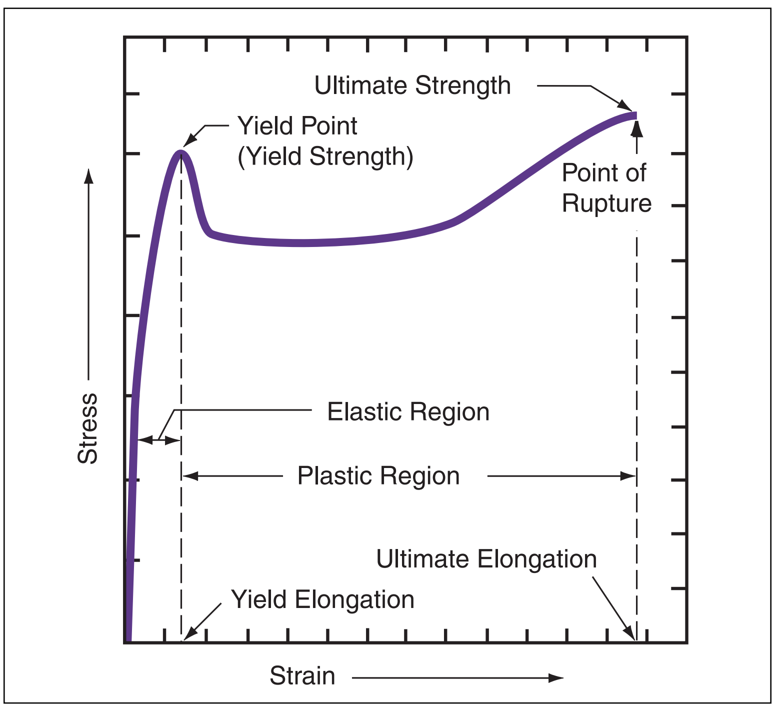
Tensile Yield Strength

Tensile yield strength is the maximum engineering stress, in MPa, at which permanent, non-elastic deformation begins. (ISO 527)

Yield Point

Yield point is the first point (load) at which the specimen yields, where the specimen's cross-sectional area begins to decrease (neckdown) significantly, or an increase in strain occurs without an increase in stress.

Figure 12 – Typical Stress-Strain Curve



Ultimate Tensile Strength

Ultimate tensile strength is the maximum stress a material can withstand before failing, whichever occurs at the higher stress level.

Elongation

Elongation at yield is the strain that the material undergoes at the yield point, or the percent change in length that occurs while the material is stressed to its yield point.

Elongation at break is the strain at failure, or the percent change in length at failure. (ISO 527)

Tensile Modulus or Young's Modulus

Tensile, or Young's modulus, is the ratio of stress to strain within the elastic region of the stress-strain curve (prior to the yield point). Note: The tensile modulus is usually measured at very low strains where the proportionality of stress to strain is at its maximum.

Sometimes, secant modulus is reported in place of tensile modulus. Secant modulus is the ratio of stress to corresponding strain at a specified strain level. It is usually employed when the stress-strain curve for a material does not exhibit linearity of stress to strain.

The shape of the stress-strain curve gives a clue to the material's behavior. A hard, brittle material shows a large initial slope and fails with little strain. A soft and tough material, on the other hand, exhibits a very small initial slope, but strain hardens and withstands larger strains before failure.

A material's stress-strain curve also indicates the overall toughness of the material. The area under the curve, in units of MPa, is a measure of a material's toughness. The greater that area is, the tougher the material is, and the greater the amount of energy required to break it. Figure 14 shows several typical engineering stress-strain curves for selected thermoplastics.

The stress-strain behavior of polymers is strongly dependent on temperature and strain rate. As temperature increases, tensile modulus and the tensile yield strength as well as ultimate tensile strength generally decrease. However, the elongation at yield and break tend to increase. As the strain rate is increased, in general, the modulus, yield strength and tensile strength all increase.

Additives to thermoplastics, such as those for ignition resistance or mold release, may decrease the ultimate tensile strength, elongation at break, and tensile modulus values.

Temperature also affects tensile properties – as environmental temperature increases, tensile values decrease. (ISO 527)

Proportional Limit

This is the point on the stress-strain curve (for a material involving both elastic and viscous components) where deviation from the linear relationship occurs. In Figure 13, this point is marked as "P." The point at which the proportional relationship deviates is often expressed in terms of stress (σ). Deformation is often expressed as strain (ϵ). Young's Modulus, or tensile modulus, represented by E, is defined as $\frac{\sigma}{\epsilon}$, as shown in Figure 13.

Elastic Limit (Point I)

The point on the stress-strain curve that marks the maximum stress a material can absorb and still recover (with no permanent deformation) to its original dimensions. Recovery may not be immediate, and the elastic limit may occur at stress levels higher than the proportional limit. In Figure 13, the proportional limit is marked as "P."

Figure 13 – Proportional Limit

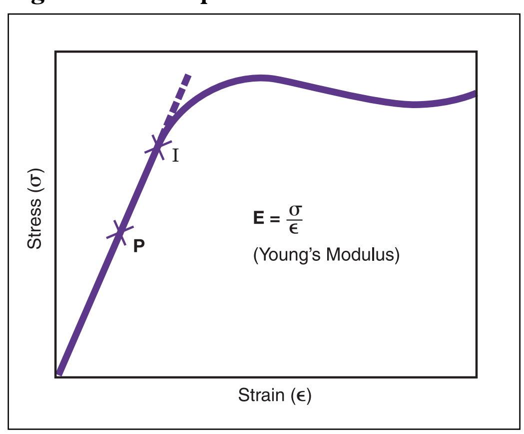
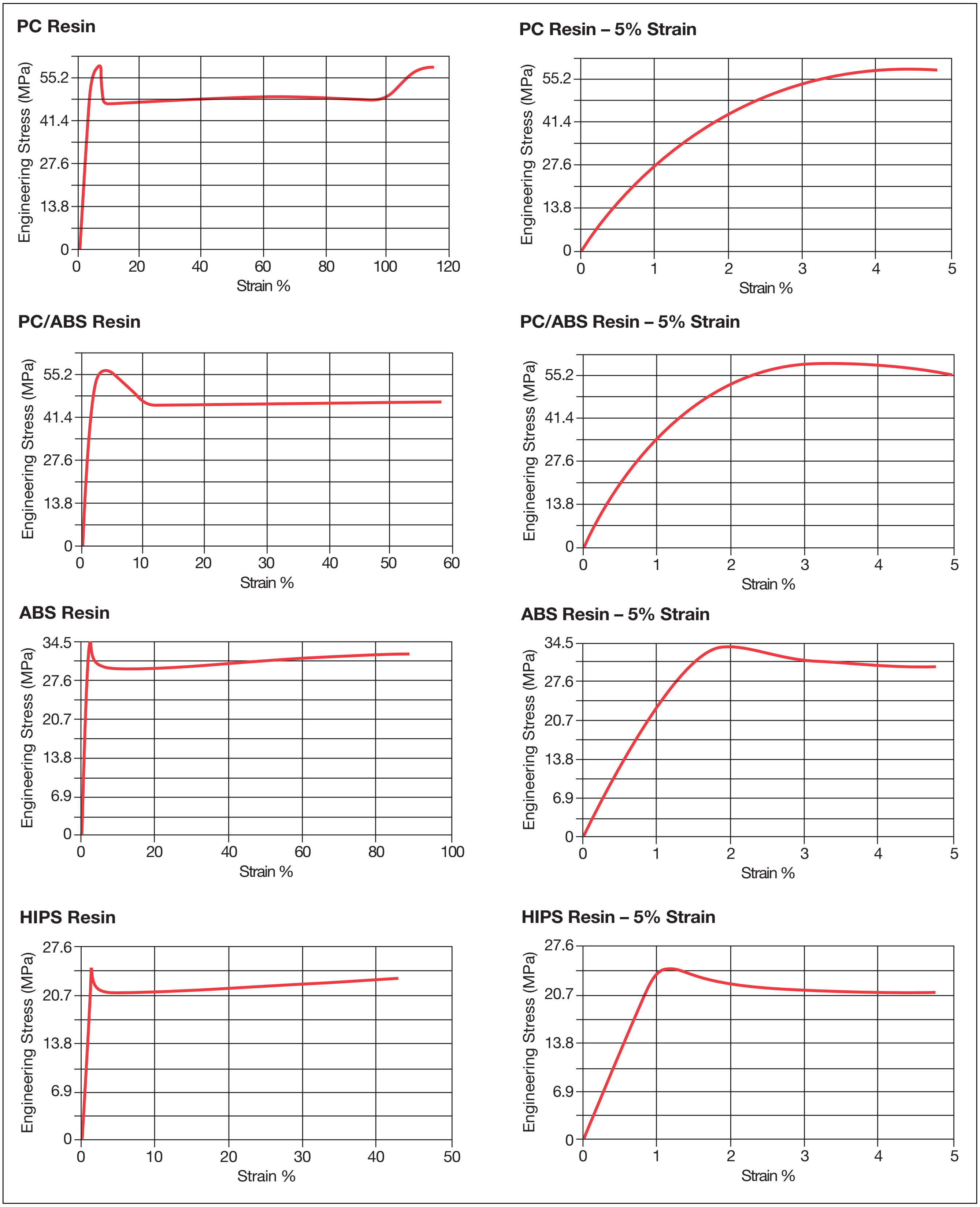


Figure 14 – Typical Engineering Stress-Strain Curves for Selected Thermoplastics[1]



[1]Engineering stress-strain curves are measured using a specimen's original cross-section (prior to neckdown). For uses such as non-linear Finite Element Analysis (FEA), true stress-strain data may give a more accurate indication of a material's behavior.

Flexural Properties

Flexural Strength

Flexural strength is the maximum stress in the outer fibers of a specimen at the moment of crack or break. This property is a measure of a material's ability to resist bending. (ISO 178)

Flexural Modulus

Flexural modulus is the ratio of stress to strain within the elastic limit (when measured in the flexural mode) and is similar to the tensile modulus. This property is used to indicate the bending stiffness of a material. The values of the flexural properties for a number of thermoplastics are provided in Table 13. (ISO 178)

Table 13 – Flexural Properties of Selected Thermoplastics

Material	Flexural Strength			Flexural Modulus		
	S.I. MPa	English psi	Metric kg/cm^3	S.I. GPa	English psi	Metric kg/cm^3
ABS	69	10,000	690	2.2	320,000	22,500
Nylon 6,6	91	13,000	930	2.6	380,000	26,500
PC/ABS	92	13,500	940	2.8	405,000	28,500
Polycarbonate	83	12,000	990	2.4	350,000	24,500
High Impact Polystyrene	96.5	14,000	985	2.5	365,000	25,500
General Purpose Polystyrene	52	7,500	530	2.1	305,000	21,500
Polyphenylene Oxide	69	10,000	700	2.4	350,000	24,500
Styrene Acrylonitrile	110	16,000	1,120	4	580,000	41,000

Poisson's Ratio

When a part molded from a plastic is subjected to tensile or compressive stress, it deforms in two directions: along the axis of the load (longitudinally) and across the axis of the load (transversely). See Figure 15.

Poisson's ratio, under tensile stress, is defined as the ratio of the lateral contraction per unit width to the longitudinal extension per unit length.

The minus sign indicates that the part decreases in cross-sectional area under tension, or increases in cross-sectional area under compression, at right angles to the load.

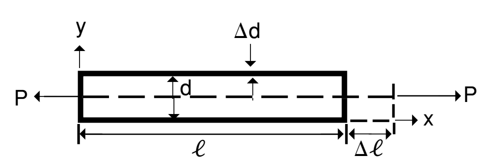
$$v = \left(\frac{\epsilon_y}{\epsilon_x}\right) = \left(\frac{\text{Lateral strain}}{\text{Longitudinal strain}}\right) = \left(\frac{\frac{\Delta \ell}{\ell}}{\frac{\Delta d}{d}}\right)$$

ϵ = strain

Table 14 – Poisson's Ratio for Selected Thermoplastics @ 23°C (73°F)

Material	Ratio
Polycarbonate	.39
Polycarbonate/Polyester	.38
Polycarbonate/ABS	.36
ABS	.35
High Impact Polystyrene	.34

Figure 15 – Poisson's Ratio



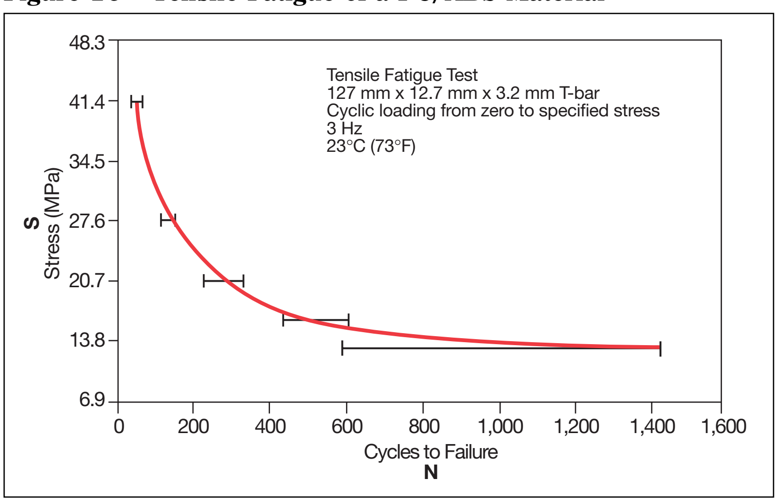
ℓ = original length
Δℓ = change in length
d = original width
Δd = change in width
x = longitudinal axis
y = transverse axis
P = load

Fatigue

In fatigue testing, a specimen of the material being tested is subjected to repeated cycles of short-term stress or deformation. Eventually, micro-cracks or defects form in the specimen's structure, causing decreased toughness, impact strength, and tensile elongation – and the likelihood of failure at stress levels considerably lower than the material's original ultimate tensile strength. The number of cycles-to-failure at any given stress level (called fatigue strength) depends on the inherent strength of the resin, the size and number of defects induced at that stress level, and the environment of the test or specimen.

Continual cyclic load on a part is fatigue. Fatigue failures are among the most common service failures. It is the principal stress involved in rotating shafts, reciprocating connecting rods, and running gear teeth. Fatigue strength is therefore an important property to consider when designing any part that will be exposed to vibration or any type of frequent, intermittent loadings.

Figure 16 – Tensile Fatigue of a PC/ABS Material



Testing specimens at different stress levels (S) and measuring the number of cycles-to-failure (N) produces an S-N curve. That S-N curve allows designers to make a direct estimate of the expected life of the part in terms of stress – a basic design parameter. By locating on the graph the number of cycles similar to that expected during the service life of the part, designers can identify the appropriate design stress.

Several factors must be recognized when working with S-N curves.

1. Your actual end-use conditions will never be identical to the test conditions used to generate published S-N curves. The validity of the S-N values for your particular application depends on the similarity of the test conditions to those of your actual end-use condition.
2. A safety factor must be included in all design calculations to compensate for:
   a. Possible flaws in the part or basic design such as voids, contaminants in the material, sharp corners or abrupt wall thickness changes in the design.
   b. Averaging of data to create an S-N curve leaves worst-case data points below the indicated curve.

Figure 16 represents an S-N curve typical of a PC/ABS material.

It should be noted that a frequency of 3 Hz (180 rpm) is used rather than the typical 30 Hz (1,800 rpm) used in a majority of fatigue data reported in published literature.

A major factor in the failure mechanism of a specimen tested at 30 Hz is due to softening because of hysteretic heating. At 3 Hz, the effect due to hysteresis is reduced to a point that is insignificant in most real world design problems, and more closely approximates their typical cyclic loading conditions.

Best Practices in Designing with Fatigue Data

1. Estimate the worst-case number of cycles your part will have to endure in its life. Let this be the "design life."
2. Refer to an S-N curve, for the design material, that was produced using loading, environmental and geometry conditions similar to those of your worst-case conditions. Locate on the S–N curve the point corresponding to the number of cycles used in your "design life."
3. Draw a straight line from this point over to the stress axis of the curve. The intersect point is your "target stress."
4. Add a factor of safety onto your "target stress." Start with a minimum of 10%. Add greater factors of safety for each occurrence of the following:[1]
   1. Sharp corners
   2. Changes in wall section
   3. Surface flaws
   4. Voids or contaminants.

   This value will be your "working stress."
5. Compare the calculated "working stress" to the actual stress the part is subjected to. If the working stress is greater than the actual stress, the design is satisfactory.
6. If the working stress is less than the actual stress, two options are available:
   a. Select a new material with a more favorable S-N curve at the current loading, environmental and geometry conditions.
   b. Modify the design to lower the actual stress by increasing the section modulus. This may be accomplished through the addition of ribs or gussets.
7. **Warning**: Under no circumstances should the "working stress" be used as the sole basis for a fatigue analysis. It is a starting point around which prototype testing of injection-molded parts must be performed under actual loading and environmental conditions to determine the "true fatigue stress" of any design.

   Very large differences in "working stress" vs. "true fatigue stress" are common in plastics. These differences may be attributable to inaccurately estimating one or more of the factors in step 4 or differences in test loading, environmental or geometry conditions.

[1] Values assigned for these factors are not well documented because they vary so widely with respect to differing designs, applications and materials.

Compressive Properties

Compression strength and modulus are tested, using ASTM D 695 procedures, by placing a specimen between two parallel platens and compressing it to rupture.

Compression strength is the amount of stress necessary to cause rupture or deformation by a predetermined percentage. Compression modulus, similar to other moduli, is the ratio of stress to strain below the proportional limit of the material.

Most engineering thermoplastics have fairly high compression properties, which will only rarely place limitations on designs. In most cases tensile or flexural strength impose many more constraints.

Table 15 – Compressive Strength Typical Values

Material	MPa	psi	kg/cm²
ABS	48	7,000	490
ABS – 20% glass filled	62	9,000	635
Nylon 6,6	34	5,000	350
Polycarbonate	86	12,500	880
Polycarbonate – 10% glass filled	96	14,000	985
Polypropylene	21	3,000	210
Polyphenylene ether – impact modified	110	16,000	1,125
Polystyrene – general purpose	83	12,000	845
Polystyrene – impact modified	44	6,400	450

ASTM D 695

Shear Strength

The shear strength of a material is determined by the maximum shear stress necessary to penetrate the surface of a flat specimen of the material completely. The shear strength test standardized by ASTM D 732 is performed with a punch tool. Since the test takes into account neither stress concentrators nor shear rate – to which plastics tend to be sensitive – the resultant test data can vary significantly. These factors make this test most useful as a way to compare the shear strength of one material to another. Data from different tests should not be compared directly.

Torsional Strength

By analogy with the tensile stress-strain relation, we can write for shear that:

$$\frac{\text{Shear stress}}{\text{Shear strain}} = \text{constant } (G)$$

where the constant G is known as the modulus of rigidity of the material.

Impact Strength

Impact Testing

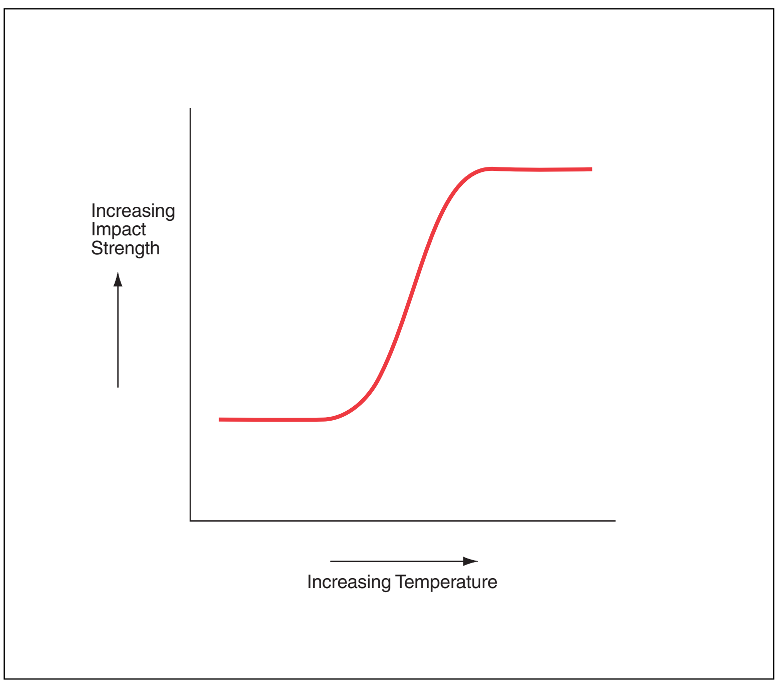
Impact testing measures the energy required to break a specimen by dynamically applying a load. Impact strength is one of the most commonly tested and reported properties of plastics. As the plastics industry grows, so do the number of different methods for measuring impact strength – with each method having its own inherent advantages and disadvantages.

Izod Impact

The Izod impact strength of a material, measured by ASTM D 256 procedures, is the amount of energy necessary for a swinging pendulum to break a notched specimen that is secured at one end. The notch in the specimen acts as a stress concentrator or crack growth site. While the test gives a good indication of a material's notch sensitivity, the values may have little validity for the behavior of unnotched parts in actual service. Table 1, pages 10 and 11, shows the Izod impact strength of selected thermoplastic materials.

If the Izod impact strength of a particular resin increases as the notch radius increases, the resin is said to be notch-sensitive. For such resins, sharp corners should be eliminated from product designs whenever possible. For curved corners, the more generous the radius, the greater the impact strength of the corner. (See page 56 for a discussion of radius corners.)

Figure 17 – Ductile-Brittle Transition Curve



Temperature Effects

The impact strength of thermoplastics is affected by temperature. This effect is seen most clearly in the way the specimen fails. At temperatures above a certain level, a test bar molded from the resin breaks via ductile fracture; that is, it stretches until failure. At lower temperatures, the break occurs by brittle fracture, with little deformation occurring during failure.

The temperature at which the behavior of the material changes (as measured by a particular test) is called the ductile-brittle transition temperature (DBTT). See Figure 17.

Thickness Effects

The thickness of the test specimen also affects the impact strength of plastics. As thickness of a part increases, the impact strength also increases – until a critical thickness is reached. At that critical thickness, impact strength decreases drastically. Brittle failure occurs. This is known in engineering terms as the transition from "plane stress to plane strain." Temperature alters the critical thickness: as the temperature increases, so does critical thickness.

Charpy Impact

Values determined by Charpy impact testing are sometimes used in place of Izod impact test results as material selection criteria. In the Charpy test, to DIN 53453, the specimen is simply supported at both ends, and the test measures the amount of energy needed to break a specimen fixed across a 100 mm span.

Because the tests are similar, results are also similarly affected by variations in resin MFR and temperature. However, the values of Charpy tests tend to be slightly higher than Izod test values, particularly when low values are obtained. This is because of the orientation of the notch on the sample.

Instrumented Dart Impact

An instrumented dart impact test, as standardized by ASTM D 3763, measures the amount of energy necessary for a high-speed, round-tipped dart to puncture a 3.2 mm thick specimen. The dart delivers a uniaxial impact on an unnotched disk. This test, by eliminating the effect of notch sensitivity, is an important indicator of impact strength for designs that do not include sharp corners.

Tensile Impact

The tensile impact test, using ASTM D 1822 procedures, measures the energy required to rapidly stretch a specimen to failure. As is the case with other impact tests, tensile impact does vary with temperature, the rate at which the test force is applied, and the polymer's MFR.

Deformation Under Load

Deformation under load, determined by ASTM D 621 procedures, is a test used in some countries. It is the percentage change that has occurred in a dimension of a 12.7 mm specimen cube when a constant force has been applied for 24 hours. The test is a short-term, low-load, compression creep test. (See page 49 for a discussion of creep.)

Data on deformation under load are particularly useful when designing assemblies to be held together by bolts, rivets, or similar fastening devices. Once the assembly is fastened, compressive creep occurs – and the fastener may loosen if the creep deformation is sufficiently great. (See "Assembly Design" on page 95 for a discussion of fasteners.)

Deformation under load also varies with temperature.

Note: This test method differs from Heat Distortion Temperature. See page 32.

Tensile Stress-Strain Behavior

Stress-Strain Curve

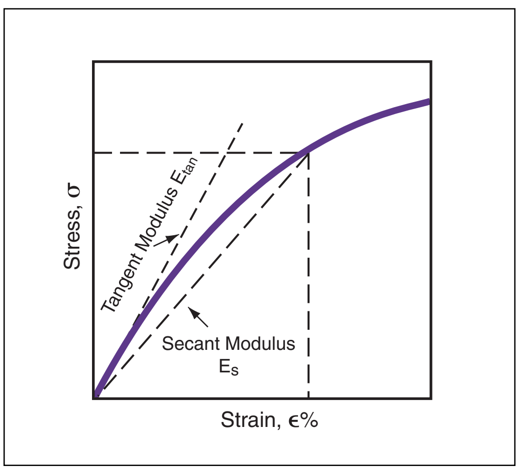
Short-term stress-strain (σ - ϵ) data are useful in these ways:

- They can be used in quality control applications to ensure consistent properties during production.
- They give a general picture of the strength and stiffness of the resin and permit comparisons for material selection.
- They provide general information about the ductility and toughness of the resin.

Toughness

The toughness of a thermoplastic is quantified by determining the area under the material's stress-strain curve. (See "Tensile Properties," page 36 for additional information on yield strength, ultimate tensile strength, elongation at break, and toughness.)

Figure 18 – Determination of Tangent and Secant Moduli



Stiffness

The stiffness of a thermoplastic is indicated by its tensile modulus. The value of the tensile modulus is determined by the steepness of a line drawn tangential to the low-strain portion of the ϵ curve, as shown in Figure 18. (The tensile moduli of some thermoplastics are given in Chapter 1, Table 1, pages 10 and 11). This modulus is often called the tangent modulus, E_{tan}, or Young's modulus.

Using Tangent and Secant Moduli

Stress-strain data can also be used in designing parts that will be subjected to infrequent, short-term, intermittent stresses, such as the stresses on a snap-fit cantilever beam that is briefly deflected during assembly. Because the stresses used to generate the ϵ data are brief in duration, the stress-strain curve can be used instead of creep data in calculations for snap-fit designs.

Most unfilled thermoplastics have no true proportional limit. The stress-strain curve of a typical unfilled thermoplastic, at normal loading rates and temperatures, is curvilinear.

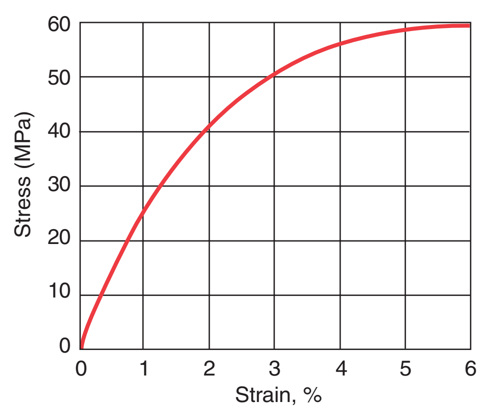
As the level of strain increases, the deviation from linearity also increases – and calculations using the tangent modulus become increasingly inaccurate. In such cases, the secant modulus should be used instead.

The secant modulus E_s, is defined – using Hooke's Law – as the ratio of stress to strain, or the slope of a line drawn from the origin to a point on the stress-strain curve corresponding to the particular strain, typically between 0.2 and 7.0% for plastics.

As seen in Figure 18, a point of higher strain on the stress-strain curve will decrease the slope of the line and decrease the value of the secant modulus. Also, the greater the strain chosen to determine the secant modulus, the greater will be the difference in values between the secant and tangent moduli. Because of that difference, a part in service may deflect more than is indicated by calculations based on the tangent modulus. Therefore, calculations using standard design equations and the secant modulus on the stress-strain curve will give more accurate predictions of deflection and stress in momentary, high-strain applications.

A typical stress-strain curve for thermoplastic is shown in Figure 19.

Figure 19 – Typical Polycarbonate Stress-Strain Curve @ 6% Strain – 23°C (73°F)



The following calculations of stress and deflection illustrate the difference between using the secant modulus and the tangent modulus to find the maximum stress (Max σ), and deflection (y), or applied force (P) on the beam shown in Figures 20 and 21.

Calculation 1

A. If the applied force (P) at the free end of the beam is 35 N, find the maximum stress and deflection.

1. Calculate the maximum stress (Max σ)

$$\text{Max } \sigma = \frac{6P\ell}{bh^2} = \frac{6\,(35)\,(25)}{12\,(4)^2}$$
$$= 27.34 \text{ MPa}$$

2. Find the deflection (y) at the free end of the beam.

a. Using the tangent modulus (E_{tan})

1. E_{tan} = 2411.5 MPa

2. Calculate the deflection (y)

$$y = \frac{4P\ell^3}{E_{tan}\, b\, h^3} = \frac{4\,(35)\,(25)^3}{2411.5\,(12)\,(4)^3}$$
$$= 1.18 \text{ mm}$$

B. Using the secant modulus (E_{sec})

1. Find the value of E_{sec}

a. From stress-strain curve, Figure 19 at σ = 27.34 MPa, ϵ = 1.37%

b. Calculate E_{sec}

$$E_{sec} = \frac{\sigma}{\epsilon} = \frac{27.34}{0.0137}$$

2. Calculate the deflection (y)

$$y = \frac{4\,P\ell^3}{E_{sec}\, b\, h^3} = \frac{4\,(35)\,(25)^3}{1995\,(12)\,(4)^3}$$
$$= 1.43 \text{ mm}$$

Note that the deflection calculated with the tangent modulus is 1.18 mm, while that calculated with the secant modulus is 1.43 mm, 0.25 mm (21%) greater.

Calculation 2

A. If the same beam is deflected 1.43 mm, find the deflection force (P) and the maximum stress (Max s) where ℓ = 25 mm.

1. Calculate the strain, ϵ

   $$\epsilon = \frac{3\,h\,y}{2\ell^2} = \frac{3\ (4)\ (1.43)}{2\ (25)^2}$$

   $= 0.0137\ (1.37\%)$

2. Find the stress,

   a. Using the tangent modulus (E_{tan})

      1. E_{tan} = 2411.5 MPa
      2. Calculate the maximum stress (Max σ)

         $\sigma = E_{tan} \times \epsilon$

         $= 2411.5 \times 0.0137$

         $= 33.04$ MPa

   b. Using the secant modulus (E_{sec})

      1. From the stress-strain curve, *Figure 19*:

         at ϵ = 1.37%, σ = 27.34 MPa

B. Find the deflection force (P)

1. Using the tangent modulus (E_{tan})

   a. E_{tan} = 2411.5 MPa

   b. Calculate the deflection force

      $$P = \frac{E_{tan}\,y\,b\,h^3}{4\ell^3} = \frac{2411.5\ (1.43)\,(12)\,(4)^3}{4\ (25)^3}$$

      $= 42.37$ N

2. Using the secant modulus (E_{sec})

   a. Find E_{sec}

      1. From the stress-strain curve, *Figure 19:*

         at ϵ = 1.37%, σ = 27.34 MPa

      2. Calculate E_{sec}

         $$E_{sec} = \frac{\sigma}{\epsilon} = \frac{27.34}{0.0137}$$

         $= 1995$ MPa

   b. Calculate the deflection force

      $$P = \frac{E_{sec}\,y\,b\,h^3}{4\ell^3} = \frac{1995\ (1.43)\ (12)\ (4)^3}{4\ (25)^3}$$

      $= 35$ N

Note the different values produced by these calculations. The maximum stress based on the tangent modulus is 33.04 MPa, while that based on the secant modulus is 27.34 MPa. Similarly, the deflection force based on the tangent modulus is 42.37 N, and that based on the secant modulus is 35 N.

Figure 20 – Typical Cantilever Beam at Rest

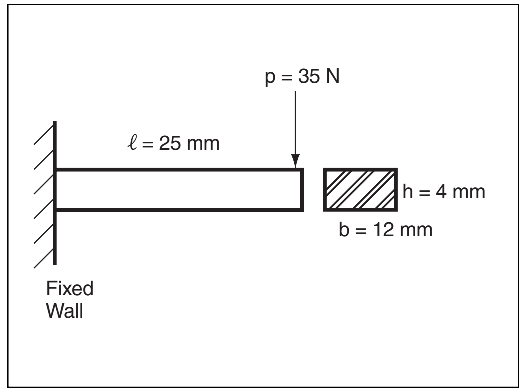
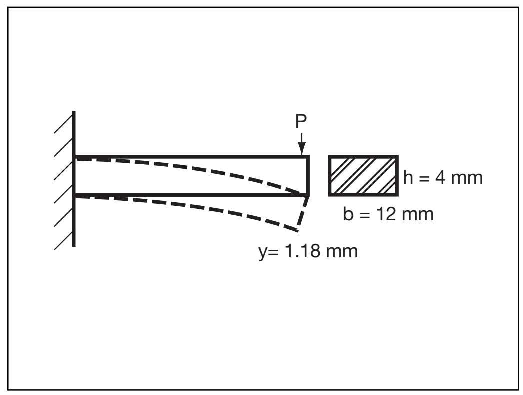


Figure 21 – Typical Cantilever Beam at Deflection



Creep

Creep and Creep Modulus

When a continuous load within the elastic range is suddenly applied to a plastic part, the part quickly deforms by an amount roughly predictable by the flexural modulus of the plastic. (The flexural modulus is a material's low-strain modulus and is explained in "Flexural Properties," page 39.) The part then continues to deform at a slower rate – indefinitely, or, if the load is great enough, until rupture.

Should the load be removed, the part will partially recover its original dimensions, but depending on the material, some portion of the deformation will remain permanently. This non-recoverable deformation is called creep. It is dependent on temperature, the duration, and amount of the load. (For more information about creep, see "Viscoelasticity," page 15.)

Most data for mechanical and physical properties reflect values for short-term loading. Such values, while indicative of a material's overall characteristics, do not accurately reflect how the material will perform when subjected to long-term loading. When loading is more than momentary, creep data, which provide realistic values for strength and rigidity properties, must be considered for purposes both of material selection and of basic design. Because both strength and stiffness are time-dependent, the design life of the part becomes an important design requirement.

Creep Strength

At a specific stress level known as the creep limit, creep becomes negligible and can be ignored in long-term loading applications (below that stress level at that temperature). A designer accustomed to working with metals should give close attention to creep strength and modulus when designing for thermoplastics.

Crazing Strength

Before a part molded from a thermoplastic resin finally breaks, it will often develop a network of fine cracks at or below its surface – a phenomenon called crazing. While these small cracks do not in themselves cause immediate fracture, they are stress concentrators that act as notches and thus significantly reduce impact strength. Crazing strength is the time-dependent and temperature-dependent level of stress at which crazing begins. Below that stress level, crazing will not occur within the selected design life of the part.

Creep Modulus

The creep modulus, E_t, represents the modulus of a material at a given stress level and temperature over a specified period of time (t). Creep modulus is expressed as:

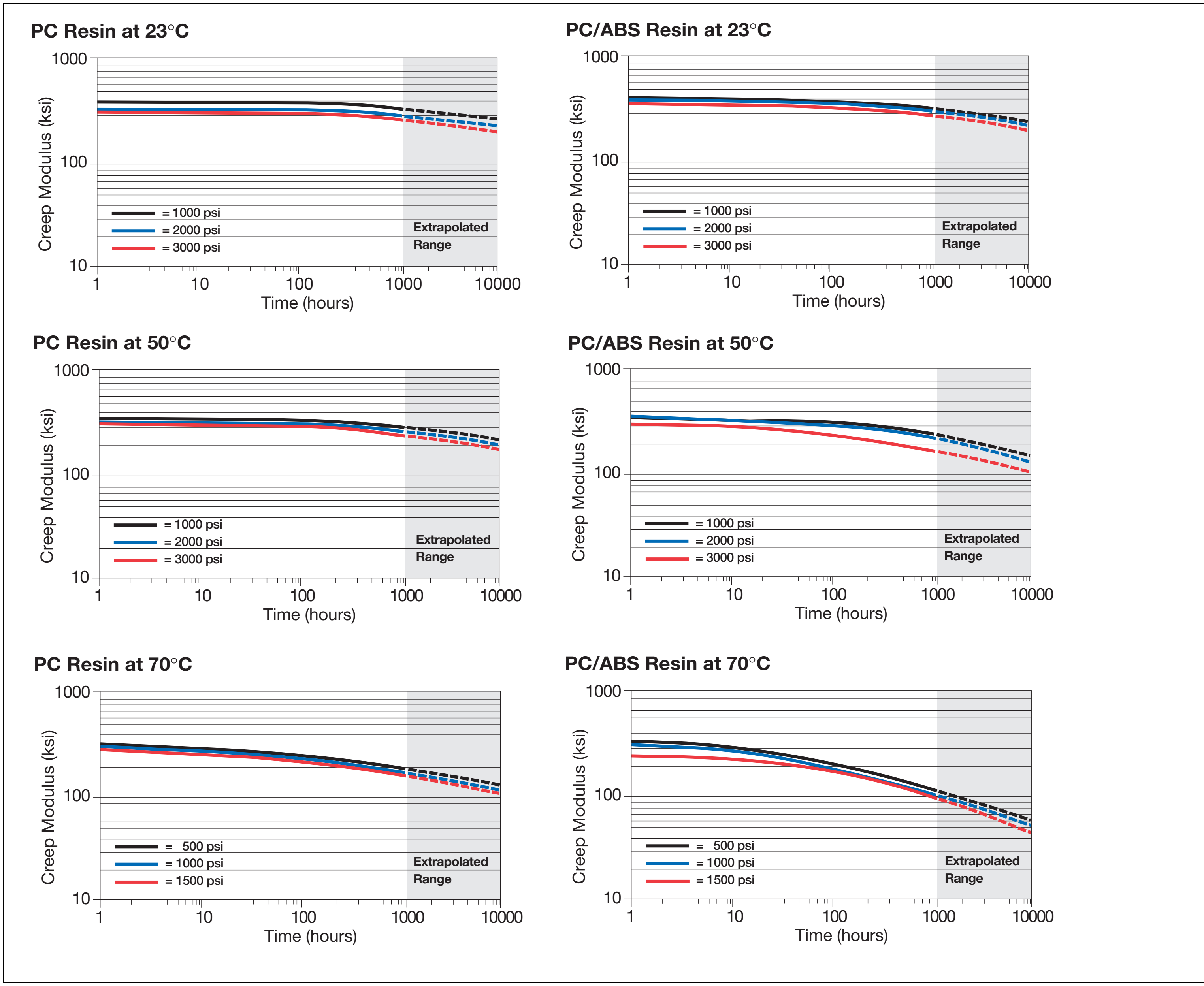
$$E_t = \frac{\text{Stress}}{\text{Total strain at time (t)}} = \frac{\sigma}{\epsilon_t}$$

The appropriate way to use creep data in continuous-load designs is to substitute the time- and temperature-dependent creep modulus – also called the apparent modulus – into the standard design equation.

Figure 22 shows typical creep modulus curves for selected thermoplastics.

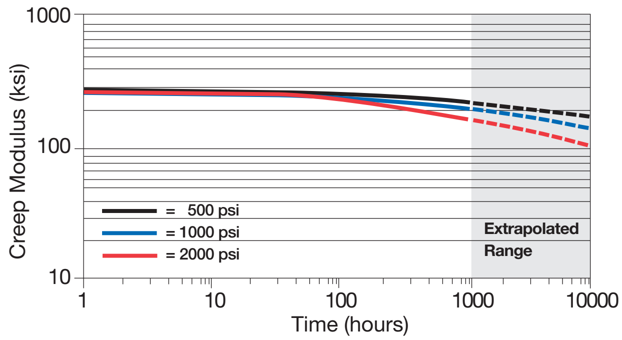
A simple illustration of creep is represented in Figure 23, page 53, where a load is introduced to a simple beam and the beam deforms (creeps) in response to the load.

Figure 22 – Typical Creep Modulus Curves for Selected Thermoplastics

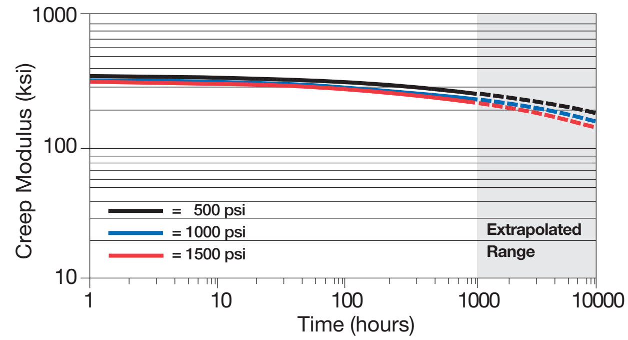


ASTM D 2990 – Injection molded
(5" x 5" x .125") specimen.
Flexural load – simple beam bending, load at center, 2" span.
Strain measurement = Deflection at center of beam.

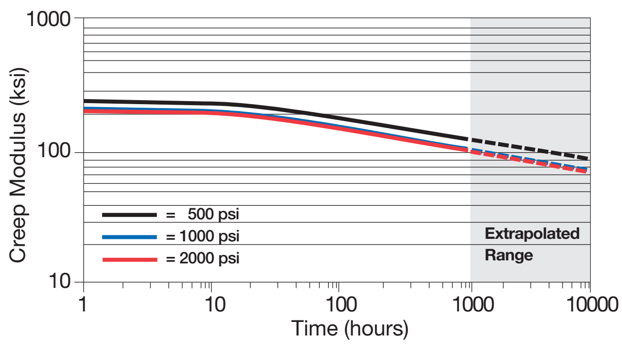
ABS Resin at 23°C



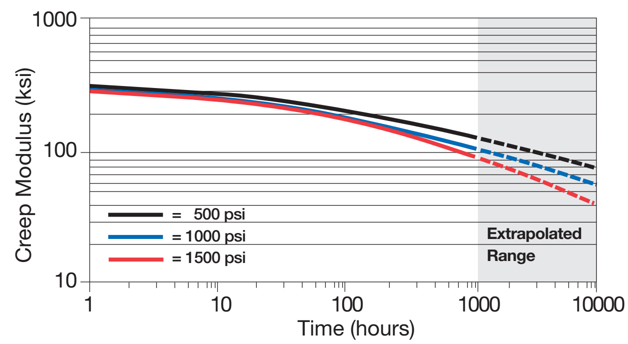
HIPS Resin at 23°C



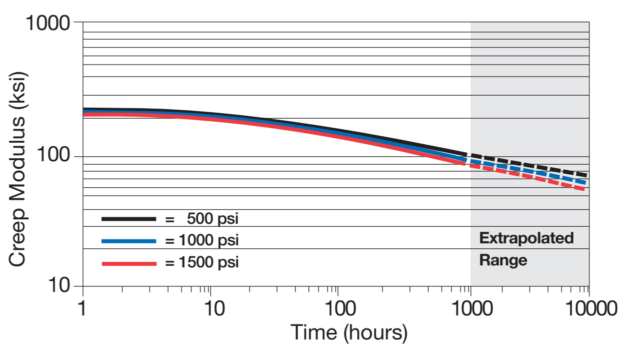
ABS Resin at 40°C



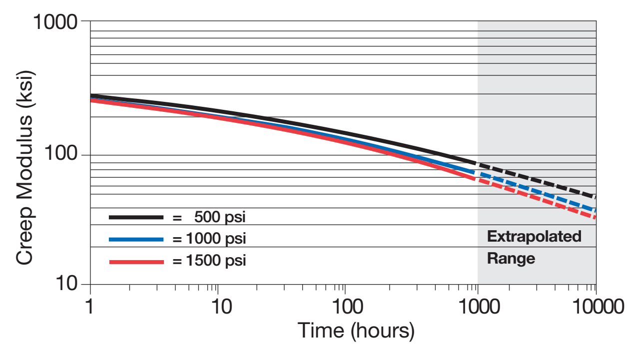
HIPS Resin at 40°C



ABS Resin at 60°C



HIPS Resin at 50°C



How to Design with Creep

With plastic articles, maintaining the part stiffness during its service life is an important consideration if the part is subjected to any amount of stress over extended periods of time. In such instances, the material stiffness will be substantially less than that predicted by elastic stress-strain behavior. The design engineer needs to consider the more realistic apparent creep modulus, rather than the standard modulus of elasticity as the design criterion. In using creep modulus for design calculations, the standard strength of material's design formulas are generally used. The application of creep data in design procedure is as follows:

1. Select the design life of the part.
2. Consult the creep modulus curves for the material of interest at the temperature the part will be used, and the stress level to which the part may be subjected. If the design stress level has not been defined, select a creep modulus curve at a conservative stress level. Later, after accurately calculating the design stress level, this choice should be rechecked.
3. Read off the apparent creep modulus value that corresponds to the design life selected. This is the *design modulus*. Since the creep data plotted on log-log scale shows generally a less pronounced curvature, extrapolation is possible. However, it should be done with caution. Although there are no specific rules, good judgment is necessary. Most experts suggest an extrapolation limit of one decade beyond the available range.
4. Apply a safety factor to the design modulus to calculate a *working modulus*, to make up for any uncertainties arising from extrapolations or other compromises that may have been made. Safety factors of 50% to 75% are typical.
5. Substitute the calculated working modulus in the part design equation to determine the part deflection.
6. If the maximum allowable deflection is lower than the calculated value, then offsetting it by increasing the part thickness is an option.

To illustrate this procedure, take the example of a cantilever beam with a rectangular cross section with a concentrated load of 1 kg on the free end. Assume the beam is 150 mm long, 12.5 mm high and 6 mm deep.

The moment of inertia of the beam cross section is given by:

$$l = (b.h^3)/12 = 976.6 \text{ mm}^4$$

The maximum bending moment and maximum bending stress are given by:

$$M_{max} = P.L. = 150 \text{ kg-mm}$$

$$S_{max} = (M_{max}\ h)/(2.l) = 9.4 \text{ MPa}$$

The initial deflection is calculated using the short term modulus of elasticity, E_o. Suppose for the material of interest, E_o = 1675 MPa.

To calculate the deflection after 1000 hours, working creep modulus corresponding to a stress level of 9.4 MPa should be used. Referring to the apparent creep modulus curves, suppose the design modulus (E_d) is found to be 1600 MPa. Then, using a safety factor of 0.75, the working modulus (E_c) is estimated to be 1200 MPa. Substituting this value,

Long term deflection Y_{max} =
$-(P.L^3)\ /(3.E_c.l) = -9.4$ mm

If the maximum allowable deflection is 7.5 mm, the part thickness will have to be increased to 13.5 mm instead of 12.5 mm.

It should be noted that there is no substitute to confirming the design and long-term performance by thoroughly testing prototype parts at end-use conditions.

Relaxation

If a plastic part is subjected to a constant deformation (strain), the force (stress) necessary to maintain that deformation decreases with time, as shown in Figure 23. This behavior is known as relaxation or stress relaxation and, like creep, is both time- and temperature-dependent.

The relaxation modulus is defined as:

$$E_{t\,relax} = \frac{\text{Stress at time (t)}}{\text{Strain}} = \frac{\frac{P\ell_t}{A_o}}{\frac{\Delta\ell}{\ell_o}}$$

Where:

A_o = original cross-sectional area

ℓ_o = original length

$\Delta\ell$ = change in length

P = load at time

The relaxation modulus and the creep modulus are essentially equal in the same service conditions. Therefore only one time- and temperature-dependent modulus is necessary for calculations. In standard practice, the creep modulus is preferred. When designing parts such as springs or press-fits that will be affected by relaxation, the creep modulus is commonly substituted for the tangent modulus in the standard equations.

Figure 23 – Calculation of Creep Modulus in a Beam

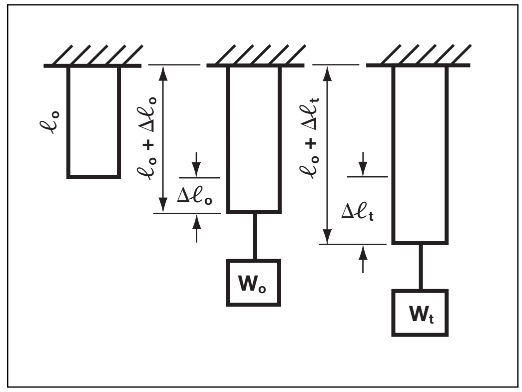
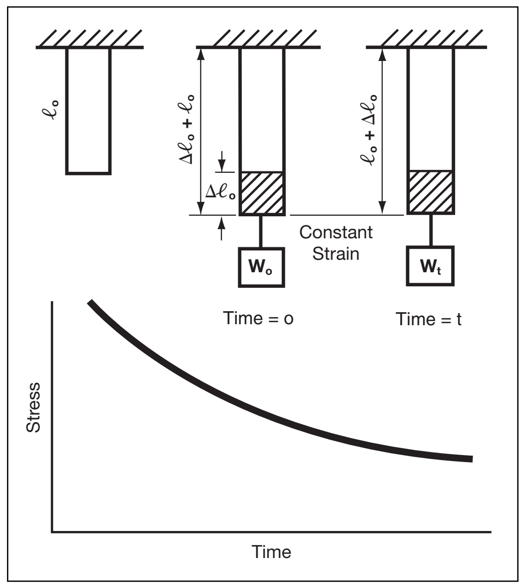


Figure 24 – Graphic Representation of Relaxation



Chapter 5

Product and Mold Design

Product Design

Although component design in thermoplastics is complex, following a few fundamental principles will help you minimize problems during molding and in part performance. Of course, the guidelines given here are general. Depending on the particular requirements of the part, it may not always be possible to follow all of our suggestions. But these guidelines, in furthering your understanding of the behavior of thermoplastics, can help you effectively resolve some of the more common design problems.

Nominal Wall Thickness

For parts made from most thermoplastics, nominal wall thickness should not exceed 4.0 mm. Walls thicker than 4.0 mm will result in increased cycle times (due to the longer time required for cooling), will increase the likelihood of voids and significantly decrease the physical properties of the part. If a design requires wall thicknesses greater than the suggested limit of 4.0 mm, structural foam resins should be considered, even though additional processing technology would be required.

In general, a uniform wall thickness should be maintained throughout the part. If variations are necessary, avoid abrupt changes in thickness by the use of transition zones, as shown in Figure 25. Transition zones will eliminate stress concentrations that can significantly reduce the impact strength of the part. Also, transition zones reduce the occurrence of sinks, voids, and warping in the molded parts.

A wall thickness variation of ±25% is acceptable in a part made with a thermoplastic having a shrinkage rate of less than 0.01 mm/mm. If the shrinkage rate exceeds 0.01 mm/mm, then a thickness variation of ±15% is permissible.

Radii

It is best not to design parts with sharp corners. Sharp corners act as notches, which concentrate stress and reduce the part's impact strength. A corner radius, as shown in Figure 26, will increase the strength of the corner and improve mold filling. The radius should be in the range of 25% to 75% of wall thickness; 50% is suggested. Figure 27 shows stress concentration as a function of the ratio of corner radius to wall thickness, R/T.

Draft Angle

So that parts can be easily ejected from the mold, walls should be designed with a slight draft angle, as shown in Figure 28. A draft angle of $1/2°$ draft per side is the extreme minimum to provide satisfactory results. 1° draft per side is considered standard practice. The smaller draft angles cause problems in removing completed parts from the mold. However, any draft is better than no draft at all.

Parts with a molded-in deep texture, such as leather-graining, as part of their design require additional draft. Generally, an additional 1° of draft should be provided for every 0.025 mm depth of texture.

Figure 25 – Suggested Design for Wall Thickness Transition Zone

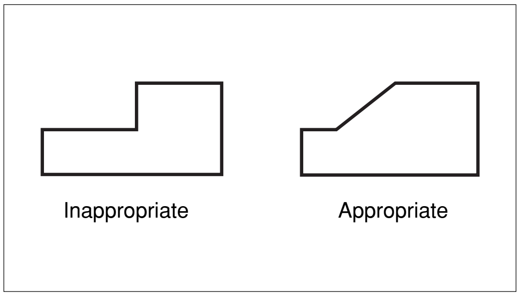


Figure 26 – Suggested Design for Corner Radius

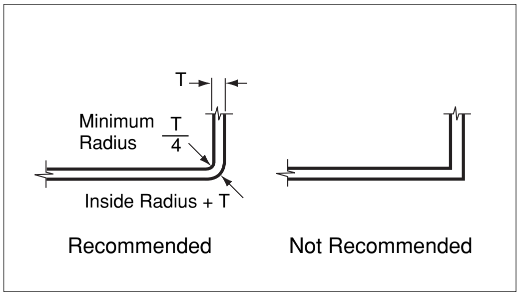


Figure 27 – Stress Concentration as a Function of Wall Thickness and Corner Radius

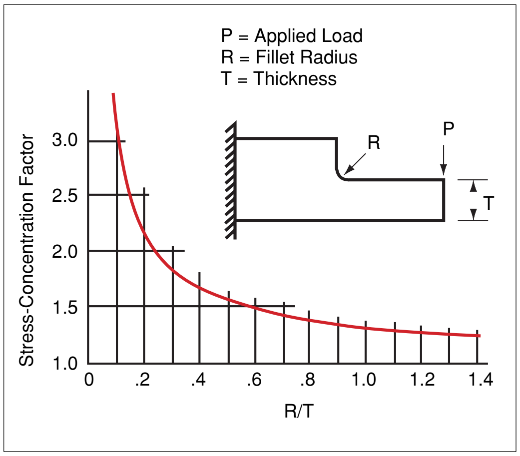
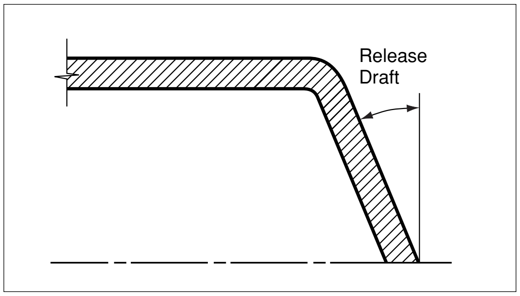


Figure 28 – Exaggerated Draft Angle



Ribs and Gussets

When designing ribs and gussets, it is important to follow the proportional thickness guidelines shown in Figures 29 and 30. If the rib or gusset is too thick in relationship to the part wall, sinks, voids, warpage, weld lines (all resulting in high amounts of molded-in stress), longer cycle times can be expected.

The location of ribs and gussets also can affect mold design for the part. Keep gate location in mind when designing ribs or gussets. For more information on gate location, see page 66. Ribs well-positioned in the line of flow, as well as gussets, can improve part filling by acting as internal runners. Poorly placed or ill-designed ribs and gussets can cause poor filling of the mold and can result in burn marks on the finished part. These problems generally occur in isolated ribs or gussets where entrapment of air becomes a venting problem.

Note: It is further recommended that the rib thickness at the intersection of the nominal wall not exceed one-half of the nominal wall in HIGHLY COSMETIC areas. For example, in Figure 29, the dimension of the rib at the intersection of the nominal wall should not exceed one-half of the nominal wall.

Experience shows that violation of this rule significantly increases the risk of rib read-through (localized gloss gradient difference).

Figure 29 – Example of Rib Design

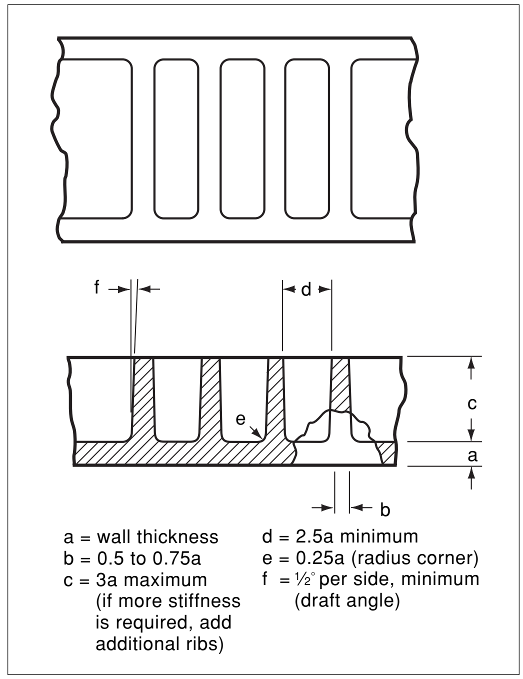
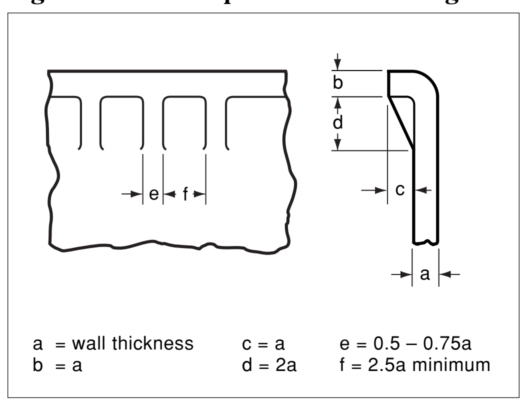


Figure 30 – Example of Gusset Design



Bosses

Bosses are used in parts that will be assembled with inserts, self-tapping screws, drive pins, expansion inserts, cut threads, and plug or force-fits. Avoid stand-alone bosses whenever possible. Instead, connect the boss to a wall or rib, with a connecting rib as shown in Figure 31. If the boss is so far away from a wall that a connecting rib is impractical, design the boss with gussets as shown in Figure 32.

Figures 33 and 34 give the recommended dimensional proportions for designing bosses at or away from a wall. Note that these bosses are cored all the way to the bottom of the boss.

Figure 31 – Recommended Design of a Boss Near a Wall (with Ribs and Gussets)

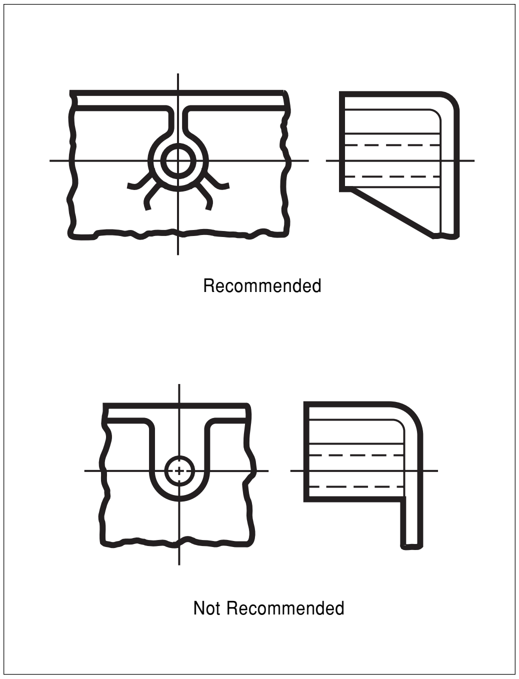


Figure 32 – Recommended Design of a Boss Away From a Wall (with Gussets)

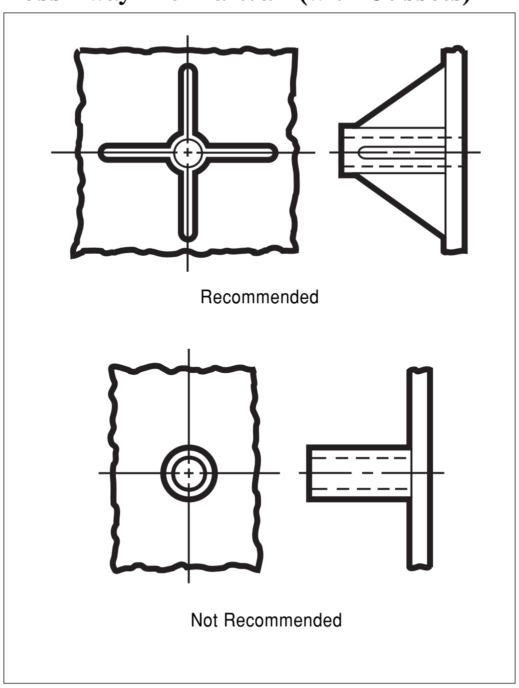
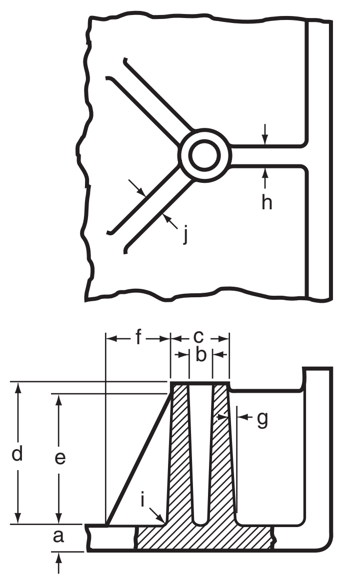
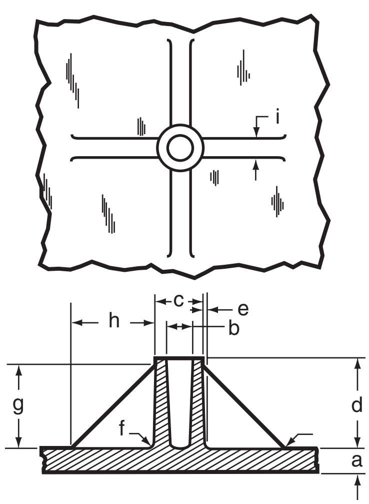


Figure 33 – Recommended Dimensions for a Boss Near a Wall (with Rib and Gussets)



a = wall thickness
b = diameter of core (at top of boss)
c = 2.5b
d = 3a
e = 0.9d
min f = 0.3e
max f = e
g = ½° per side (draft angle)
h = 0.6a (at base)
i = 0.25a (radius corner)
j = 0.6a (at base)

Figure 34 – Recommended Dimensions for a Boss Away From a Wall (with Gusset)



a = wall thickness
b = diameter of core (at top of boss)
c = 2.5b
d = 3a
e = ½° per side (draft angle)
f = 0.25a (radius corner)
max g = 0.95d
min h = 0.3g
max h = g
i = 0.6a (at top of gusset)

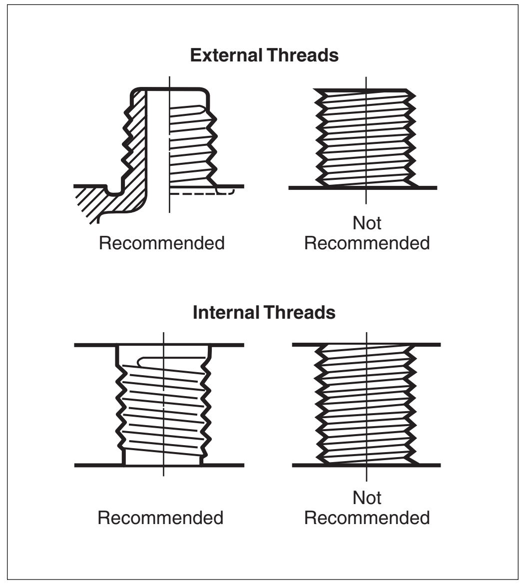
Threads

Molded-in threads can be designed into parts made of engineering thermoplastic resins. Threads always should have radiused roots and should not have feather edges – to avoid stress concentrations. Figure 35 shows examples of good design for molded-in external and internal threads. For additional information on molded-in threads, see page 105. Threads also form undercuts and should be treated as such when the part is being removed from the mold i.e., by provision of unscrewing mechanisms, collapsible cores, etc. Every effort should be made to locate external threads on the parting line of the mold where economics and mold reliability are most favorable.

Undercuts

Because of the rigidity of most engineering thermoplastic resins, undercuts in a part are not recommended. However, should a design require an undercut, make certain the undercut will be relieved by a cam, core puller, or some other device when the mold is opened.

Figure 35 – Recommended Design for Molded-in Threads



Mold Design

Proper design of the injection mold is crucial to producing a functional plastic component. Mold design has great impact on productivity and part quality, directly affecting the profitability of the molding operation. This section provides general guidelines for the design of a good, efficient mold for making thermoplastic parts.

Figure 36 – Three Common Sprue Pullers

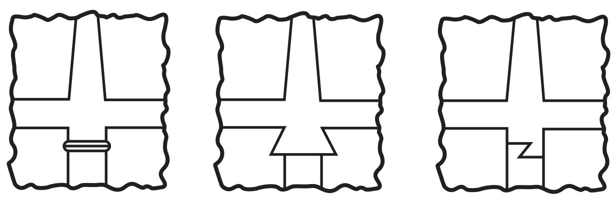
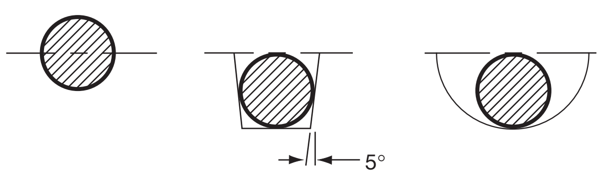


Figure 37 – Three Conventional Runner Profiles



Sprue Bushings

Sprue bushings connect the nozzle of the injection molding machine to the runner system of the mold. Ideally, the sprue should be as short as possible to minimize material usage and cycle time. To ensure clean separation of the sprue and the bushing, the bushing should have a smooth, tapered internal finish that has been polished in the direction of draw (draw polished.) Also, the use of a positive sprue puller is recommended. Figure 36 shows three common sprue puller designs.

Runner Geometry of Conventional Mold

Runner systems convey the molten material from the sprue to the gate. The section of the runner should have maximal cross-sectional area and minimal perimeter. Runners should have a high volume-to-surface area ratio. Such a section will minimize heat loss, premature solidification of the molten resin in the runner system, and pressure drop.

The ideal cross-sectional profile for a runner is circular. This is known as a full-round runner, as shown in Figure 37. While the full-round runner is the most efficient type, it also is more expensive to provide, because the runner must be cut into both halves of the mold.

A less expensive yet adequately efficient section is the trapezoid. The trapezoidal runner should be designed with a taper of 2 to 5° per side, with the depth of the trapezoid equal to its base width, as shown in Figure 37. This configuration ensures a good volume-to-surface area ratio.

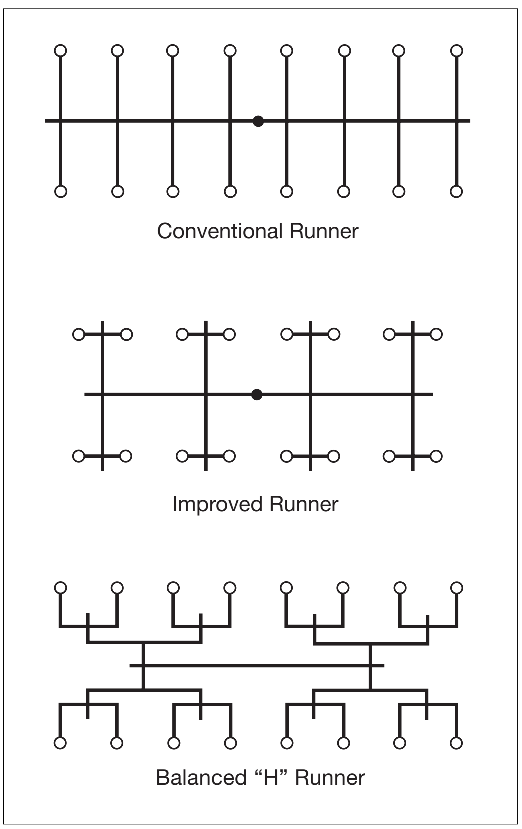
Half-round runners are not recommended because of their low volume-to-surface area ratio. Figure 37 illustrates the problem. If the inscribed circles are imagined to be the flow channels of the polymer through the runners, the poor perimeter-to-area ratio of the half-round runner design is apparent in comparison to the trapezoidal design.

Runner Diameter Size

Ideally, the size of the runner diameter will take many factors into account – part volume, part flow length, runner length, machine capacity gate size, and cycle time. Generally, runners should have diameters equal to the maximum part thickness, but within the 4 mm to 10 mm diameter range to avoid early freeze-off or excessive cycle time. The runner should be large enough to minimize pressure loss, yet small enough to maintain satisfactory cycle time. Smaller runner diameters have been successfully used as a result of computer flow analysis where the smaller runner diameter increases material shear heat, thereby assisting in maintaining melt temperature and enhancing the polymer flow.

Large runners are not economical because of the amount of energy that goes into forming, and then regrinding the material that solidifies within them.

Figure 38 – Runner System Layouts



Runner Layout

Similar multicavity part molds should use a balanced “H” runner system, as shown in Figure 38. Balancing the runner system ensures that all mold cavities fill at the same rate and pressure. Of course, not all molds are multicavity, nor do they all have similar part geometry. As a service to customers, Dow Plastics offers computer-aided mold filling analysis to ensure better-balanced filling of whatever mold your part design requires. Utilizing mold filling simulation programs enables you to design molds with:

- Minimum size runners that deliver melt at the proper temperature, reduce regrind, reduce barrel temperature and pressure, and save energy while minimizing the possibility of material degradation.
- Artificially balanced runner systems that fill family tool cavities at the same time and pressure, eliminating overpacking of more easily filled cavities.

Cold Slug Wells

At all runner intersections, the primary runner should overrun the secondary runner by a minimum distance equal to one diameter, as shown in Figure 39. This produces a feature known as a melt trap or cold slug well. Cold slug wells improve the flow of the polymer by catching the colder, higher-viscosity polymer moving at the forefront of the molten mass and allowing the following, hot, lower-viscosity polymer to flow more readily into the mold-cavity. The cold slug well thus prevents a mass of cold material from entering the cavity and adversely affecting the final properties of the finished part.

Figure 39 – Recommended Design of a Cold Slug Well

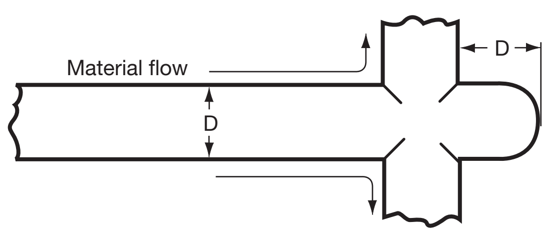
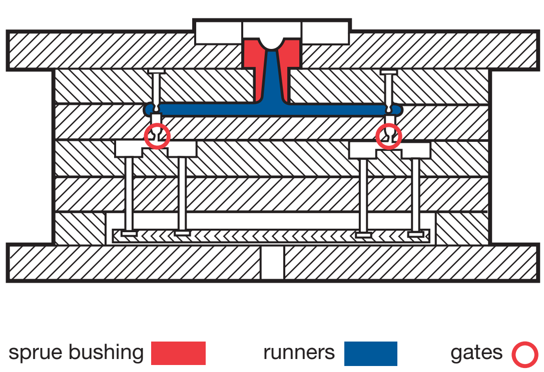


Figure 40 – Conventional Cold Runner Mold



Runnerless Molds

Runnerless molds differ from the conventional cold runner mold (Figure 40) by extending the molding machine's melt chamber and acting as an extension of the machine nozzle. A runnerless system maintains all, or a portion, of the polymer melt at approximately the same temperature and viscosity as the polymer in the plasticating barrel. There are two general types of runnerless molds: the insulated system, and the hot (heated) runner system.

Insulated Runners

The insulated runner system (Figure 41) allows the molten polymer to flow into the runner, and then cool to form an insulating layer of solid plastic along the walls of the runner. The insulating layer reduces the diameter of the runner and helps maintain the temperature of the molten portion of the melt as it awaits the next shot.

The insulated runner system should be designed so that, while the runner volume does not exceed the cavity volume, all of the molten polymer in the runners is injected into the mold during each shot. This full consumption is necessary to prevent excess build-up of the insulating skin and to minimize any drop in melt temperature.

The many advantages of insulated runner systems, compared with conventional runner systems, include:

- Less sensitivity to the requirements for balanced runners.
- Reduction in material shear.
- More consistent volume of polymer per part.
- Faster molding cycles.
- Elimination of runner scrap – less regrind.
- Improved part finish.
- Decreased tool wear.

However, the insulated runner system also has disadvantages. The increased level of technology required to manufacture and operate the mold results in:

- Generally more complicated mold design.
- Generally higher mold costs.
- More difficult start-up procedures until running correctly.
- Possible thermal degradation of the polymer melt.
- More difficult color changes.
- Higher maintenance costs.

Hot Runners

The more commonly used runnerless mold design is the hot runner system, shown in Figure 42. This system allows greater control over melt temperatures and other processing conditions, as well as a greater freedom in mold design – especially for large, multicavity molds.

Hot runner molds retain the advantages of the insulated runner over the conventional cold runner, and eliminate some of the disadvantages. For example, start-up procedures are not as difficult. The major disadvantages of a hot runner mold, compared with a cold runner mold, are:

- More complex mold design, manufacture, and operation.
- Substantially higher costs.

These disadvantages stem from the need to install a heated manifold, balance the heat provided by the manifold, and minimize polymer hang-ups.

The heated manifold acts as an extension of the machine nozzle by maintaining a totally molten polymer from the nozzle to the mold gate. To accomplish this, the manifold is equipped with heating elements and controls for keeping the melt at the desired temperature. Installing and controlling the heating elements is difficult. It is also difficult to insulate the rest of the mold from the heat of the manifold so the required cyclic cooling of the cavity is not affected.

Another concern is the thermal expansion of the mold components. This is a significant detail of mold design, requiring attention to ensure the maintenance of proper alignment between the manifold and the cavity gates. (For more information on thermal expansion, see the section on thermal stress analysis, page 32.)

Currently there are many suppliers and many available types of runnerless mold systems. In most cases, selection of such a system is based primarily on cost and design limitations – be careful in evaluating and selecting a system for a particular application.

Figure 41 – Insulated Runner Mold

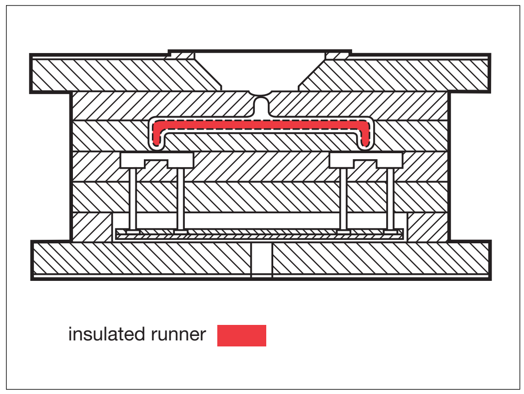
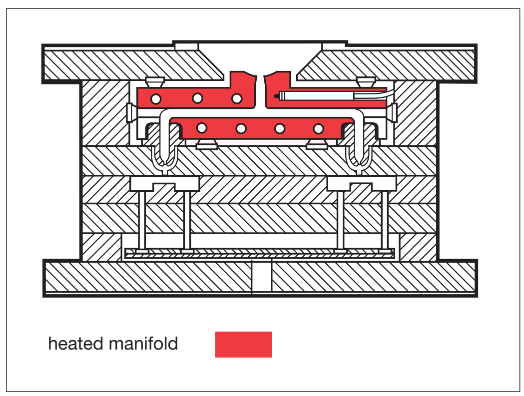


Figure 42 – Hot Runner Mold



Gates

The gate serves as a transition zone between the runner and the part, and should be designed to permit easy filling of the mold.

Gate Size

Gates should be small enough to ensure easy separation of the runner and the part. However, they should be large enough to prevent premature freezing-off of the polymer flow, which can affect the consistency of part dimensions. When specifying gate size, it is best to be “steel safe.” Start with a gate size smaller than you think will do the job, and increase the size until proper filling of the mold is achieved consistently. The minimum size we suggest for gate diameter is 0.75 mm, and, as a rule, it should not exceed the runner or sprue diameter. Gates are often designed to be half the nominal wall thickness of the part.

Gate Location

Correct location of gates has a critical effect on finished part performance. You should consider the following guidelines when determining gate location.

Appearance

Residual vestiges of a gate are normally unacceptable on a visible surface. Therefore, position gates on a non-visible surface whenever possible.

Stress

Do not place gates near highly stressed areas. The gate itself, and degating of the part that may be required, result in high residual stresses near the gate area. Also, the rough surface left by the gate creates stress concentrators.

Pressure

Place gates in the thickest section of a part to ensure ample pressure for packing-out the thick section, and prevention of sinks and voids.

Orientation

Gate location affects the molecular orientation of the polymer. Molecular orientation becomes more pronounced as the depth of the flow channel decreases in thin part sections. Because of flow stress orientation, most of the molecules align in the same direction.

High degrees of orientation result in parts having uniaxial strength. And such parts are primarily resistant only to forces acting in one direction. To minimize molecular orientation, position gates so that as soon as the molten polymer enters the cavity, the flow is diverted by an obstruction.

Weld Lines

In general, place gates to equalize flow length throughout the cavity. Also, place gates to minimize the number and length of weld lines.

Figure 43 shows how weld lines are formed and how they can be prevented. When weld lines are unavoidable, place the gate close to the obstruction forming the weld line – to maintain a high melt temperature and ensure a strong weld.

Filling

Select gate locations so that the polymer impinges against walls (or other projections, such as pins) as shown in Figure 44. This will eliminate jetting and also will help to prevent flow marks and gate-blush on the surfaces of the part.

Figure 43 – Positioning Gates to Eliminate Weld Lines

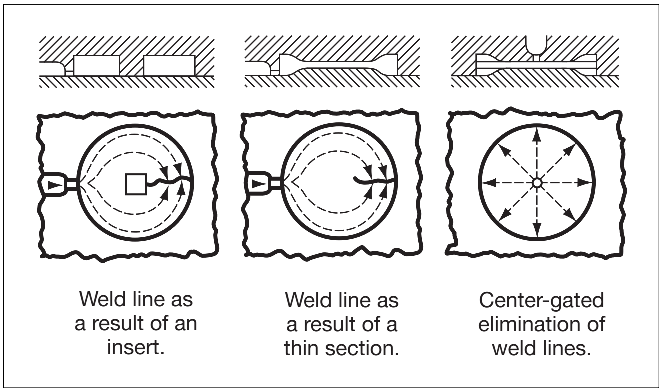
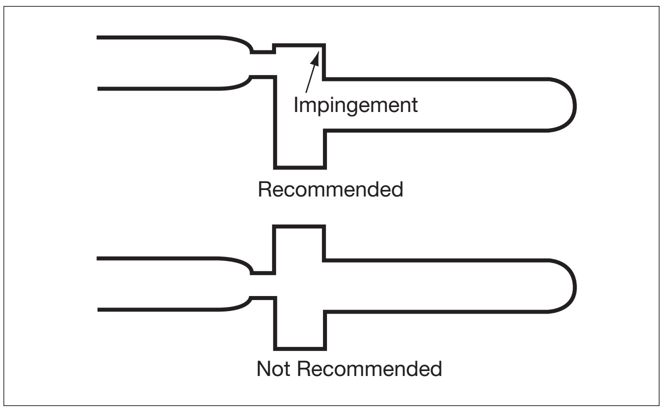


Figure 44 – Positioning Gates to Improve Polymer Flow



Types of Gates

Selecting the best type of gate for a given mold design is as important as the location and size of the gate. Many gate designs are readily available. The most commonly used gates are described here to help you to select the type best-suited for specific kinds of applications. See Figures 45 to 54.

Figure 45 – The sprue gate is recommended for single-cavity molds or for molds for circular parts requiring symmetrical filling. This gate is suitable for thick sections.

Figure 46 – The side, or edge gate is used for multicavity two-plate molds and is suitable for medium and thick sections.

Figure 47 – The pin gate (a three-plate tool) is often substituted for an edge gate to minimize finishing and provide a centrally located gate. It is good for applications that require automatic degating, but is suitable only for thin sections.

Figure 48 – The restricted, or edge pin gate allows simple finishing and degating. Like the pin gate (Figure 47), it is used only for thin sections.

Figure 49 – The tab gate is a restricted gate that prevents “jetting” and minimizes molding strain.

Figure 50 – The diaphragm gate is used for single-cavity molds for single-shaped parts that have a small or medium internal diameter.

Figure 51 – The internal ring gate is similar to the diaphragm gate, and is used for single-cavity molds to make ring-shaped parts having large internal diameters.

Figure 52 – The external ring gate is used for multicavity molds for ring-shaped parts when the diaphragm gate is not practical.

Figure 53 – The flash gate is a development of the edge pin gate for larger volume cavities.

Figure 54 – The geometry of the submarine gate.

Figure 45 – Sprue Gate

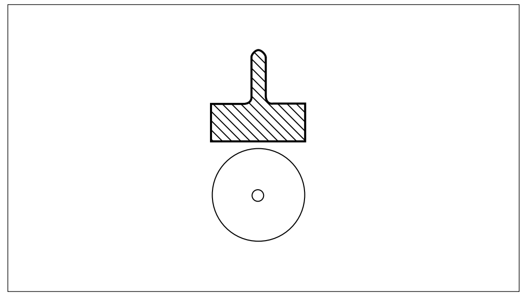


Figure 46 – Edge Gate

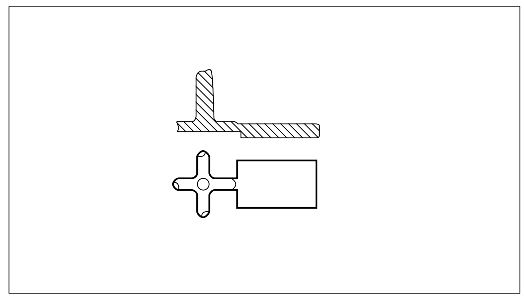


Figure 47 – Pin or Drop Gate (3-Plate Mold)

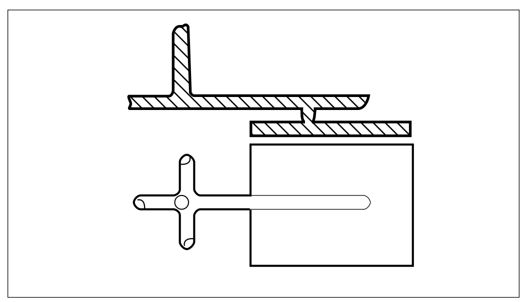


Figure 48 – Restricted Edge Pin Gate

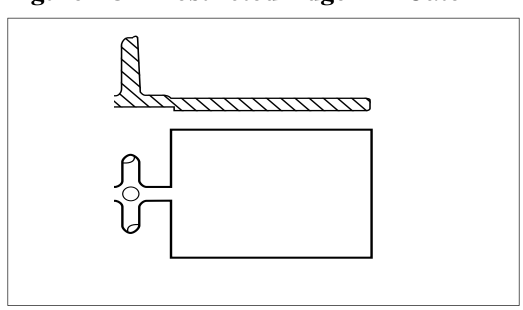


Figure 49 – Tab Gate

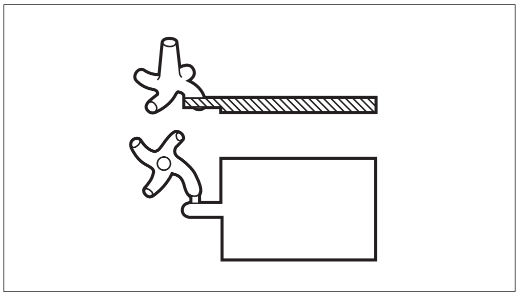


Figure 50 – Diaphragm Gate

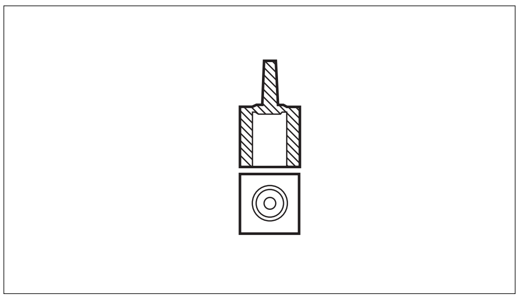


Figure 51 – Internal Ring Gate

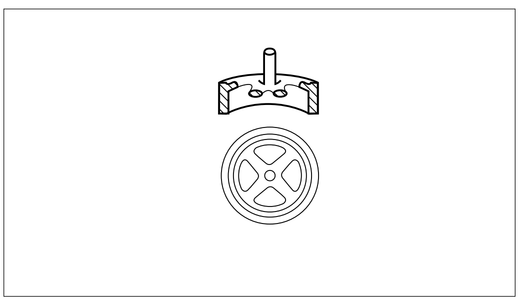


Figure 52 – External Ring Gate

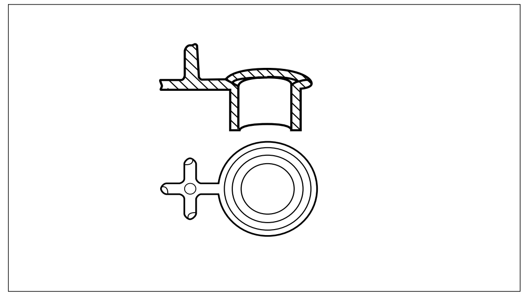


Figure 53 – Flash Gate

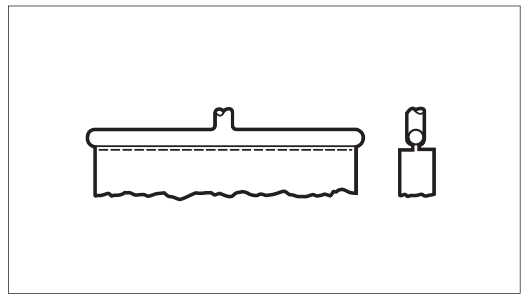
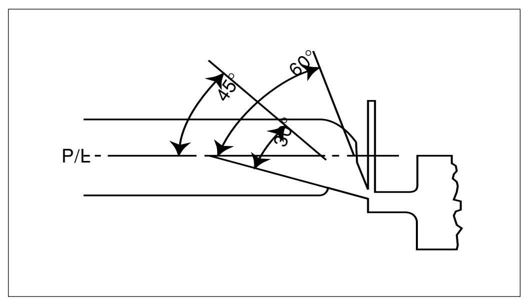


Figure 54 – Geometry of Submarine Gate



Vents

All mold cavities must be vented in order to release the air that is displaced when the polymer flows into them. Poor venting can result in short shots, weak weld lines, burn marks, and high molded-in stresses resulting from high packing pressures.

The number of vents in a mold is often limited by the economics of mold construction.

Good part design practices include specifying vent location on part prints.

Note: In general, higher melt flow materials must use smaller vents than a low melt flow version of the same material.

Example: Polycarbonate with a 3 melt flow rate may prove to mold sufficiently with a 0.08 mm (0.003") vent, showing no vent vestige. However, when a polycarbonate with a melt flow rate of 22 is run in the same mold, small vestiges may appear on the part at the vent entrance.

Vent Size

The vent depth should be as indicated in Table 16, from 0.02 mm to 0.05 mm for at least the first 0.25 mm distance from the edge of the mold cavity. The vent depth then should increase to a minimum of 0.75 mm to the outer edge of the mold and the vent width should be a minimum of 3 mm.

As in the sizing of gates, vents should be cut "steel safe." Begin with shallow vents and cut them larger, if needed, until molding is satisfactory. Vents that are too small tend to become clogged, reducing or eliminating their ability to release air from the cavity of the mold. Large vents can lead to flash on the part at the vent location.

Table 16 – Venting Techniques

Polymer	Depth of Vent mm (inch)
Polystyrene	.02-.05 (0.001-0.002)
ABS	.04-.06 (0.0015-0.0025)
PC/ABS	.02-.05 (0.001-0.002)
Polycarbonate	.02-.05 (0.001-0.002)
Polyethylene	.01-.02 (0.0005-0.001)

Figure 55 – Venting Geometry

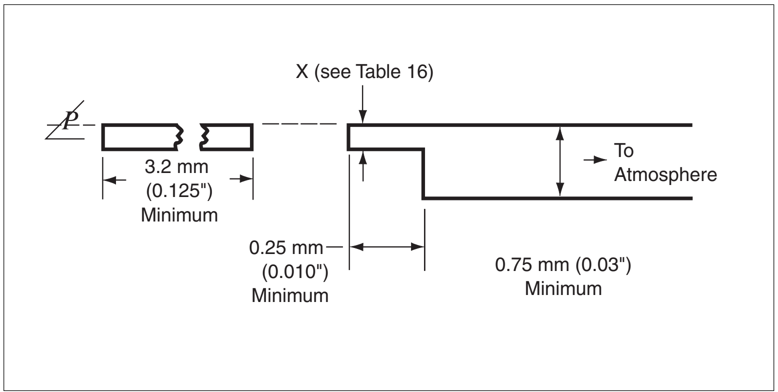
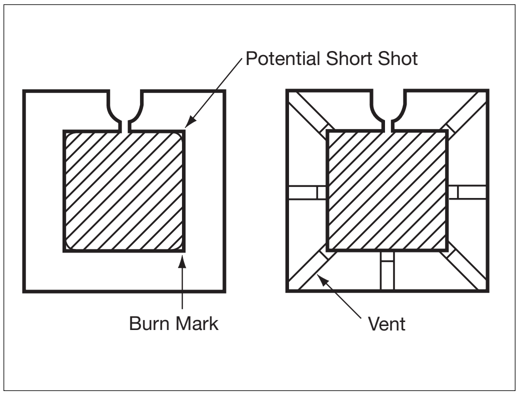


Figure 56 – Venting Techniques



Vent Location

Vents can be positioned anywhere along the parting line of the mold, particularly at last-to-fill locations as shown in Figure 56. A reasonable guide is to have vents spaced at 25 mm pitch. For blind ribs and bosses, vents may be incorporated into the mold by grinding flat spots along the major axis of an ejector pin or cavity.

Another option for venting is the use of sintered metal inserts. These inserts enable gas to pass into them but do not allow the polymer to clog them. Sintered metal inserts should be used only on non-visual surfaces and only as a last resort.

Ejection Mechanisms

When designing plastic parts, the method of part ejection from the mold must be considered in the concept phase. Designing with ejection in mind largely eliminates use of costly and complex ejection systems pressed into service later, when a part is difficult to eject.

Four factors should be considered in designing the ejection mechanism:

- Shape and geometry of the part.
- Type of material and wall thickness.
- Projected production volume.
- Component position relative to the parting line.

These factors will usually indicate to the designer which mechanism is most suited for the designed part. The following guidelines will help you decide on particular mechanisms.

Ejector or Knockout Pins

These are very common and inexpensive ejection methods. The pins are preferably located where changes in shape occur (at corners, ribs, bosses, etc.), because these features increase the difficulty of ejection. Among the various pin geometries are stepped pins, blade pins, valve pins, and standard flat pins.

Other Methods

Ejector sleeves are often used around part bosses. Stripper rings/plates are used with thin-wall containers. Air ejection is used to eject parts having an "enclosed" geometry (a flat part would not contain the air long enough to blow the part off the mold).

Regardless of the ejection method selected, the designer must calculate the area of part surface required if the part is to be ejected effectively. If the surface area of ejection is inadequate, the part surface can be damaged by the ejection mechanism. You can use the following equation to calculate the ejection force required to remove the part from the mold.

$$P = \frac{S_t \times E \times A \times \mu}{d\,[d/2t - (d/4t \times v)]}$$

where:

P = Ejection force (N)
S_t = Thermal contraction of the plastic across diameter d = Coefficient of thermal expansion x ΔT
ΔT = Temperature difference (°C)
d = Diameter of circle whose circumference is equal to the perimeter length of the molded part surrounding the male core (mm)
E = Elastic modulus (MPa)
A = Area of contact that shrinks onto core in the direction of ejection (mm^2)
μ = Coefficient of friction, plastic/steel
t = Thickness of molded part (mm)
v = Poisson's ratio of the plastic

Cooling

Molds must be cooled to remove heat from the just-molded plastic part so the part can be ejected from the mold as quickly as possible. Cooling is accomplished by drilling or machining passages in the mold and circulating a heat-transfer fluid through those passages. Other than passages for cooling in the molding block or plates, the molding surfaces of the core and each cavity should also have direct cooling passages. To remove heat from the just-molded article and thus permit ejection, cooling must occur efficiently and effectively. Inefficient cooling can be very costly because cooling accounts for, on average, 70 to 80% of the cycle time.

Bore diameter for cooling channels should be drilled to accept pipes in the range of 6 to 10mm. Do not use smaller pipes unless there is a size constraint. The hoses used to interconnect passages in the mold should have the same inside diameter as the passages.

To maximize the cooling rate, the cooling fluid – water or ethylene glycol/water mixture – should flow turbulently. Turbulent flow achieves three to five times as much heat transfer as does non-turbulent flow.

The cooling rate is also affected by the material used for making the mold. A beryllium copper mold transfers twice as much heat as does a carbon steel mold, and four times as much as a stainless steel mold. This does not mean that using a beryllium copper mold will permit molding cycles four times as fast as a stainless steel mold. However, a beryllium copper mold will run some thin-wall parts significantly faster.

Beryllium copper molds are not recommended for molding thermoplastics that require elevated mold temperatures. The high thermal conductivity of the beryllium copper allows so much heat to transfer to the surroundings that it is difficult to maintain adequate heat economically. Dow Plastics offers computer-aided analysis of mold-cooling networks to help you ensure adequate and uniform cooling of your molded part.

Chapter 6

Design Formulas

Stress Formulas
Strength of Materials
Beam Formulas, Bending Moments
Properties of Sections, Moments of Inertia
Flat Plate Formulas
Designing for Equal Stiffness
Designing for Impact Resistance
Designing for Thermal Stress

Whether you are designing in metals or plastics, it is necessary to choose the specific structural property values for use in standard design equations. With metals, such property values are relatively constant over a wide range of temperatures and time. But for plastics, the appropriate values are dependent on temperature, stress level, and life expectancy of the part.

As far as design practices are involved, the principles defined in many good engineering handbooks are applicable. However, the nature of high polymer materials requires even more attention to appropriate safety factors.

The information and formulas provided in this chapter can help you solve many of the design problems commonly met in the structural design of plastic parts.

However, it is important that designers and design engineers understand that the formulas and the data expressed in this brochure are given only as guides. They may not be pertinent to the design of a particular part, with its own special requirements and end-use environments.

Generally, the symbols used in this manual's various figures, formulas, and text have the definitions shown in the boxed column on this page.

Our customers can expect efficient design assistance and aid from the technical support services at Dow Plastics. We invite you to discuss your needs with us.

Above all, there is an aspect of professional and competent design engineering that holds true throughout. That is the fact that, after all the science, mathematics, and experience have been properly used in "solving" the design needs of a part, it is strongly recommended that prototype parts be produced and thoroughly tested in the expected end-use conditions and environments before committing the design to full-scale production.

Partial list of engineering symbols and letters used, and meanings.

θ	=	Angle
A	=	Area . . . cross-sectional
α	=	Coefficient of linear thermal expansion
y	=	Deflection of cantilever; height of undercut
ρ	=	Density
D	=	Diameter
MD	=	Diameter, major
PD	=	Diameter, pitch
c	=	Distance from neutral axis to outer fiber, centroid
z	=	Distance from q to neutral axis
P	=	Force . . . P = deflection force
μ	=	Friction, coefficient
a,b,h,t	=	Height or thickness
I	=	Inertia, moment of (neutral axis)
ℓ	=	Length
$\Delta\ell$	=	Length, change
E	=	Modulus (Young's)
q	=	Point within a beam or internal pressure
R	=	Radius
r	=	Radius
E_s	=	Secant modulus
M	=	Sectional bending moment
τ	=	Shear stress
ϵ	=	Strain
σ	=	Stress
TCF	=	Thickness conversion factor
T	=	Temperature
ΔT	=	Temperature, change
v	=	Poisson's ratio
ω	=	Velocity, constant angular, radius/second
b_o	=	Width at base
a	=	Width . . . wall thickness

Stress Formulas

Within the elastic limits of the materials, design formulas developed for metals can also be applied to plastics. Stress levels are determined only by load and part geometry, so standard equations can be used. Deflection is determined by two other material property values: the elastic, or Young's modulus (E); and Poisson's ratio (v). Since the modulus of a plastic material varies with temperature and duration of the stress, this modulus may need replacement in deflection equations by the appropriate creep modulus. It may be helpful to review various sections of Chapter 4 for assistance in choosing modulus values appropriate to the specific stress level, temperature, and design life of the part.

Poisson's ratio varies with temperature, strain level, and strain rate. These differences are too small to significantly affect a calculation. For example, Poisson's ratio at room temperature for CALIBRE polycarbonate resin is 0.37, and it ranges from 0.35 to 0.40 over the operational temperature range. By selecting the correct modulus and assuming the value of Poisson's ratio to be constant, standard equations can be used to design a part for fabrication in thermoplastics.

Tensile or Compressive Stress

Tensile or compressive stress σ is the force carried per unit of area and is expressed by the equation:

$$\sigma = \frac{P}{A} = \frac{P}{ab}$$

Where:

σ	=	stress
P	=	force
A	=	cross-sectional area
a	=	width
b	=	height

The force (P) produces stresses normal (i.e., perpendicular) to the cross section of the part. If the stress tends to lengthen the part, it is called tensile stress. If the stress tends to shorten the part, it is called compressive stress. (For compression loading, the part should be relatively short, or it must be constrained against lateral bucking.)

Strain

Strain is the ratio of the change in the part's length, over the original length. It is expressed as the percentage of change in length, or percent elongation.

In direct tension and compression loading, the force is assumed to act along a line through the center of gravity of members having uniform cross-sections, called centroids.

Stress Acting at an Angle

The standard stress equation is valid when the cross-section being considered is perpendicular to the force. However, when the cross-section is at an angle other than 90° to the force, as shown in Figure 57, the equation must be adapted. These stresses are always less than the standard case, i.e., maximum normal stress occurs when $\theta = 0$.

Shear Stress

In addition to the normal stress calculated in the previous section, a plane at an angle to the force has a shear stress component. Here, unlike tensile and compressive stress, the force produces stress in the plane of the cross-section, i.e., the shear stresses are perpendicular to tensile or compressive stresses. The equations for calculating planar shear stress, based on Figure 58 are:

$$\tau\theta = \frac{P}{A} \sin\theta \cos\theta$$

$$\text{Max } \theta = \frac{P}{2A} \text{ (when } \theta = 45° \text{ or } 135°)$$

Torsional Stress

When a stress acts to twist a component, it produces torsional stress. If a solid circular shaft, or shaft-like component, is subject to a twisting moment, or torsion, the resulting shear stress (q) is calculated by:

$$q = \frac{G\,\theta\,r}{\ell}$$

where:

- q = shear stress
- G = modulus of rigidity (see Chapter 4, page 35)
- θ = angle of twist, in radians
- r = radius of shaft
- ℓ = length of shaft

The torque (T) carried by the shaft is given by

$$T = \frac{G\,\theta}{\ell} I_P$$

where I_P is the polar second moment of

$$\text{area} = \frac{\pi d^4}{32}$$

A useful rearrangement of the formula is

$$\theta = \frac{T\ell}{GI_P}$$

Figure 57 – Diagram of Stress Acting at an Angle θ

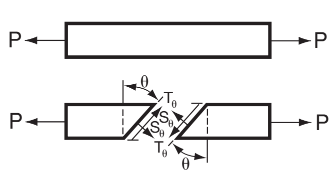
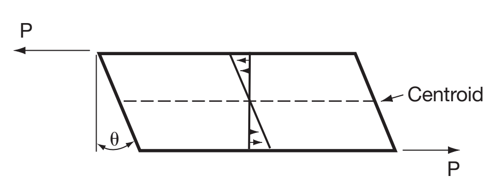


Figure 58 – Representation of Shear Stress

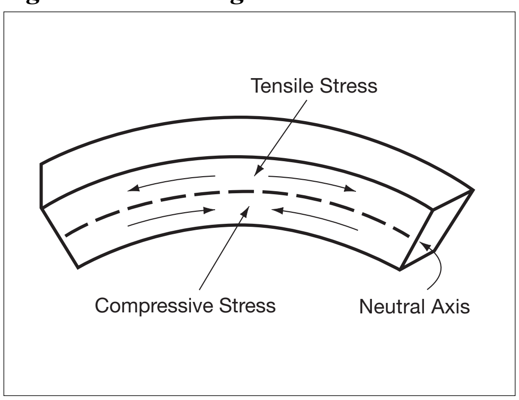


Beams

When a straight beam of uniform cross-sectional area is subjected to a perpendicular load, the beam bends. If shear is negligible, the vertical deflection is largely due to bending. Fibers on the convex side of the beam lengthen, and fibers on the concave side compress.

There is a neutral surface within any beam that contains the centroids of all sections and is perpendicular to the plane of the load for such deflections. In a uniform, symmetrical beam, the neutral axis of the beam is the horizontal, central axis. Tensile or compressive stress and strain on the neutral axis are essentially zero. At all other points within the beam, the stress is a tensile stress if the point lies between the neutral axis and convex surfaces of the beam, and is a compressive stress if the point lies between the neutral axis and concave surfaces of the beam, see Figure 59.

Figure 59 – Bending of a Beam



The fiber stress σ for any point (q) within the beam is calculated using the equation:

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

where:

M = bending moment of the section containing q (values can be taken from the appropriate beam formula, Figures 60 to 68).
z = the distance from q to the neutral axis
I = the moment of inertia with respect to the neutral axis (values can be taken from the appropriate cross-sectional area formula, Figures 69 to 91).

The maximum fiber stress in any section occurs at the points farthest from the neutral surface and at the section of greatest bending moment, i.e., when z = Max z, and M = Max M. Maximum fiber stress is given by the equation:

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

where:

c = the distance from the neutral axis to the extreme outermost fiber.

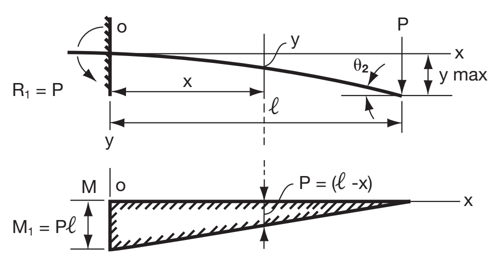
Such equations are valid if:

- The beam is of homogeneous material, so that it has the same modulus of elasticity in tension and compression.
- Plane sections remain planar.

If several loads are applied at the same time, the total stress and deflection at any point are found by superimposition. Compute the stress and deflection for each load acting on the point, and add them together.

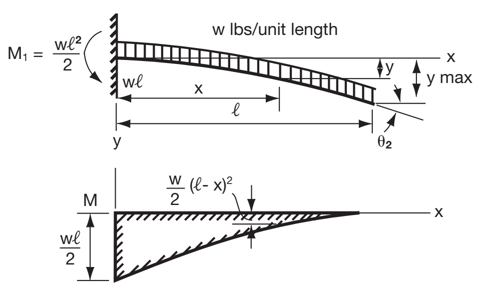
Beam Formulas, Bending Moments

Figure 60 – Cantilever Beam, concentrated load at free end



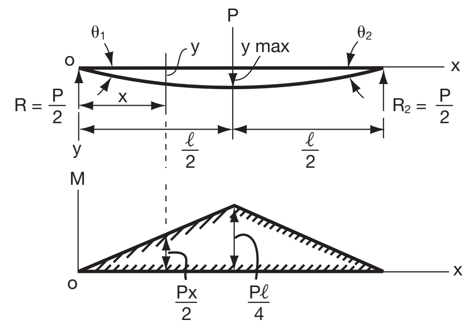
$\theta_2 = \frac{P\ell^2}{2EI}$ $\quad$ $y\ max = \frac{P\ell^3}{3EI}$ $\quad$ $y = \frac{P}{6EI}(3\ell x^2 - x^3)$

Figure 61 – Cantilever Beam, uniform load, w per unit length, total load W



$\theta_2 = \frac{w\ell^2}{6EI}$ $\quad$ $y\ max = \frac{w\ell^3}{8EI}$ $\quad$ $y = \frac{wx^2}{24EI\ell}(x^2 + 4\ell x - 6\ell^2)$

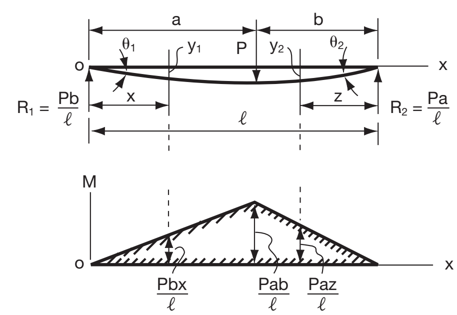
Figure 62 – Simple Beam, concentrated load at center



For $o < x \leq \frac{\ell}{2}$; $y = \frac{P}{48EI}(3\ell^2 x - 4x^3)$

$\theta_1 = \theta_2 = \frac{P\ell^2}{16EI}$ $\quad$ $y\ max = \frac{P\ell^3}{48EI}$

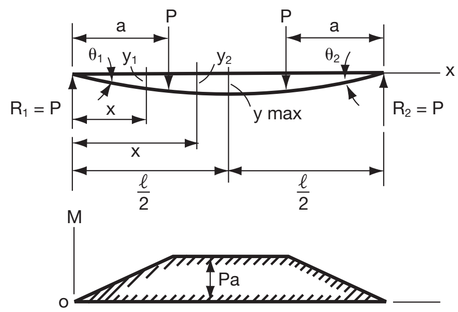
Figure 63 – Simple Beam, concentrated load off center



$\theta_1 = \frac{Pab\,(\ell + b)}{6\ell EI}$ $\quad$ $y_1 = \frac{Pbx}{6\ell EI}(\ell^2 - b^2 - x^2)$

$\theta_2 = \frac{Pab\,(\ell + a)}{6\ell EI}$ $\quad$ $y_2 = \frac{Paz}{6\ell EI}(\ell^2 - a^2 - z^2)$

Figure 64 – Simple Beam, two equal, concentrated loads, symmetrically placed

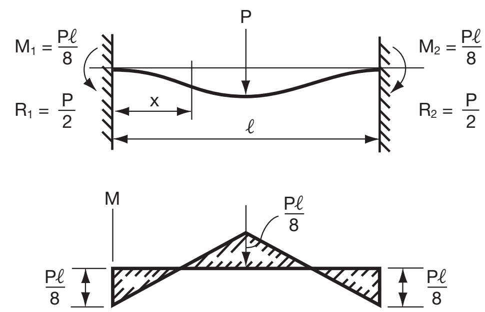


For $o < x \le a$; $y_1 = \frac{Px}{6EI}[3a(\ell - a) - x^2]$

For $a \le x \le (\ell - a)$; $y_2 = \frac{Px}{6EI}[3x(\ell - x) - a^2]$

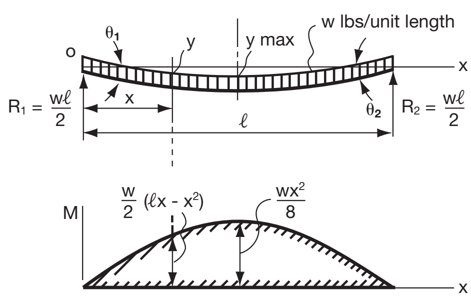
$\theta_1 = \theta_2 = \frac{Pa(\ell - a)}{2EI}$ $\quad$ $y\ max = \frac{Pa}{24EI}(3\ell^2 - 4a^2)$

Figure 66 – Beam fixed at both ends, concentrated load at center



$\theta_1 = \theta_2 = O$ $\quad$ $y = \frac{Px^2}{48EI}(3\ell - 4x)$ $\quad$ $y\ max = \frac{P\ell^3}{192EI}$

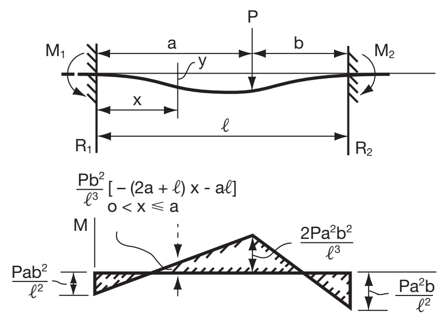
Figure 65 – Simple Beam, uniform load, w per unit length, total load W



$\theta_1 = \theta_2 = \frac{w\ell^3}{24EI}$ $\quad$ $y\ max = \frac{5w\ell^3}{384EI}$

$y = \frac{wx(\ell - x)}{24EI\ell}(\ell^2 + x[\ell - x])$

Figure 67 – Beam fixed at both ends, concentrated load at any point



$a > b$

$R_1 = \frac{Pb^2}{\ell^3}(3a + b);$ $\quad M_1 = \frac{Pab^2}{\ell^2}$

$R_2 = \frac{Pa^2}{\ell^3}(a + 3b)$ $\quad M_2 = \frac{Pa^2b}{\ell^2}$

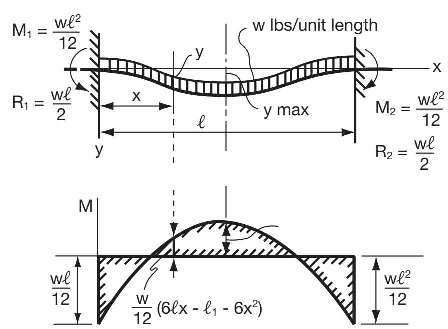
For $o < x \le a$; $y = \frac{Pb^2}{6\ell^3 EI}[3\,a\ell x^2 - (3a + 6)x^3]$

$\theta_1 = \theta_2 = O$ $\quad y\ max = \frac{2Pa^3b^2}{3EI\,(3a + b)^2}$

$ex = \frac{2a\ell}{\ell + 2a}$

At load $y = \frac{Pa^3b^3}{3EI\ell^3}$

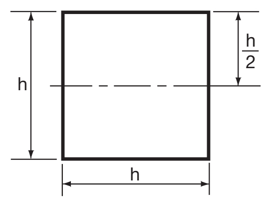
Figure 68 – Beam fixed at both ends, uniform load w per unit, total load W



$\theta_1 = \theta_2 = 0$ $\quad y = \frac{wx^2}{24EI}(\ell - x)^2$ $\quad y\ max = \frac{w\ell^4}{384EI}$

Properties of Sections, Moments of Inertia

Figure 69



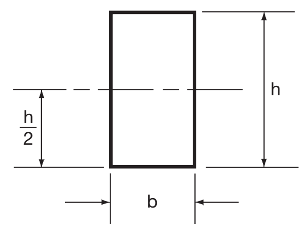
$A = h^2$

$I = \frac{h^4}{12}$

$r = \frac{h}{\sqrt{12}} = 0.289\ h$

$Z = \frac{h^3}{6}$

Figure 72



$A = bh$

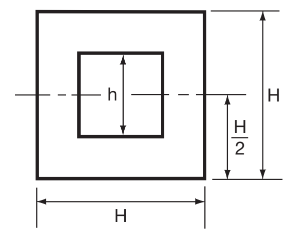
$C = \frac{h}{2}$

$i = \frac{bh^3}{12}$

$Z = \frac{bh^2}{6}$

$r = \frac{h}{\sqrt{12}}$

Figure 75



$A = H^2 - h^2$

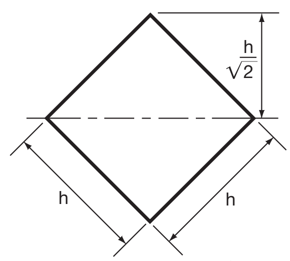
$C = \frac{H}{2}$

$I = \frac{H^4 - h^4}{12}$

$Z = \frac{1}{6}\left(\frac{H^4 - h^4}{H}\right)$

$r = \sqrt{\frac{H^2 + h^2}{12}}$

Figure 70



$A = h^2$

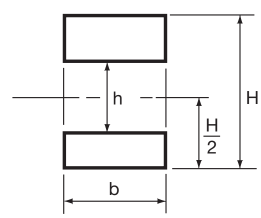
$C = \frac{h}{\sqrt{2}}$

$I = \frac{h^4}{12}$

$Z = \frac{h^3}{6\sqrt{2}}$

$r = \frac{h}{\sqrt{12}}$

Figure 73



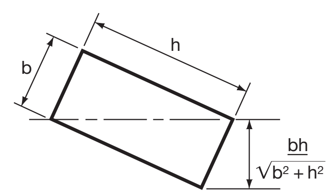
$A = b\,(H - h)$

$I = \frac{b}{12}\,(H^3 - h^3)$

$Z = \frac{b}{6}\left(\frac{H^3 - h^3}{H}\right)$

$r = \sqrt{\frac{H^3 - h^3}{12\,(H - h)}}$

Figure 76



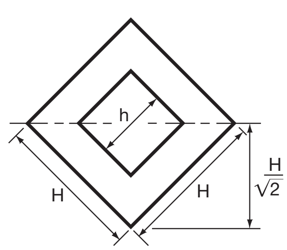
$A = bh$

$I = \frac{b^3h^3}{6(b^2 + h^2)}$

$Z = \frac{b^2h^2}{6\sqrt{b^2 + h^2}}$

$r = \frac{bh}{6\sqrt{(B^2 + H^2)}}$

Figure 71



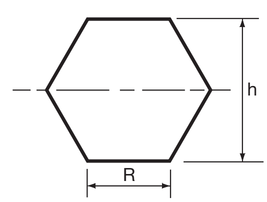
$A = H^2 - h^2$

$I = \frac{H^4 - h^4}{12}$

$Z = \frac{\sqrt{2}\,(H^4 - h^4)}{12H}$

$r = \sqrt{\frac{H^2 + h^2}{12}}$

Figure 74



$A = \frac{3R^2}{2\tan 30^\circ}$

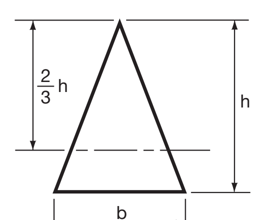
$C = \frac{h}{2}$

$I = \frac{5\sqrt{3}}{16}\,R^4$

$r = 0.456\sqrt{R}$

$Z = 0.625R^3$

Figure 77



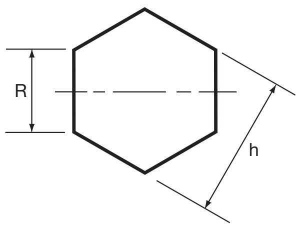
$A = \frac{1}{2}\,bh$

$I = \frac{bh^3}{36}$

$Z = \frac{bh^2}{24}$

$R = \frac{h}{\sqrt{18}}$

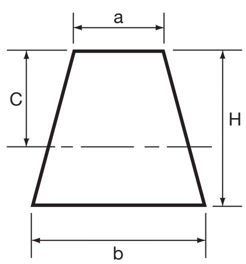
Figure 78



$C = 0.577\ h$ $\quad r = 0.456\sqrt{R}$

$I = 0.541R^4$ $\quad Z = 0.541R^3$

Figure 79



$$A = \frac{h\,(a + b)}{2}$$

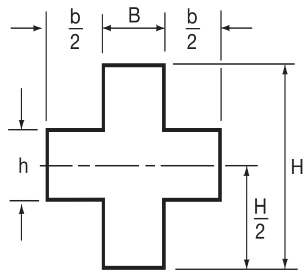
$$C = \frac{h}{3}\left(\frac{a + 2b}{a + b}\right)$$

$$I = \frac{h^3}{36}\left(\frac{a^2 + 4ab + b^2}{a + b}\right)$$

$$Z = \frac{h^2}{12}\left(\frac{a^2 + 4ab + b^2}{a + 2b}\right)$$

$$r = \frac{h}{\sqrt{18}}\left(a + b\right)\sqrt{a^2 + 4ab + b^2}$$

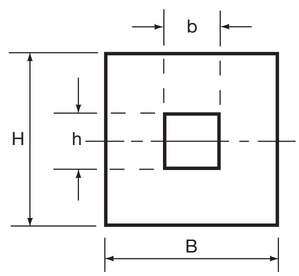
Figure 80



$A = HB + hb$ $\quad Z = \dfrac{BH^3 + bh^3}{6H}$

$I = \dfrac{BH^3 + bh^3}{12}$ $\quad r = \sqrt{\dfrac{BH^3 + bh^3}{12\,(BH + bh)}}$

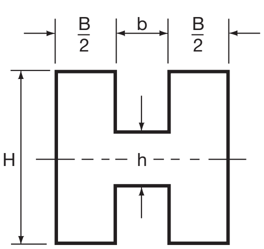
Figure 81



$A = BH - bh$ $\quad Z = \dfrac{BH^3 - bh^3}{6H}$

$I = \dfrac{BH^3 - bh^3}{12}$ $\quad r = \sqrt{\dfrac{BH^3 - bh^3}{12\,(BH - bh)}}$

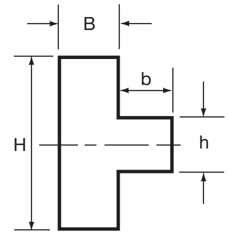
Figure 82



$A = HB + hb$ $\quad Z = \dfrac{BH^3 + bh^3}{6H}$

$I = \dfrac{BH^3 + bh^3}{12}$ $\quad r = \sqrt{\dfrac{BH^3 + bh^3}{12\,(BH + bh)}}$

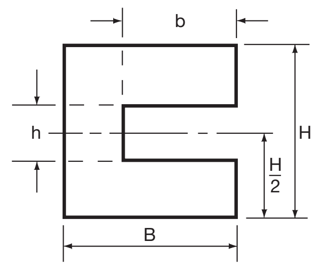
Figure 83



$A = HB + hb$ $\quad Z = \dfrac{BH^3 + bh^3}{6H}$

$I = \dfrac{BH^3 + bh^3}{12}$ $\quad r = \sqrt{\dfrac{BH^3 + bh^3}{12\,(BH + bh)}}$

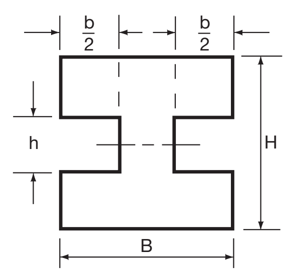
Figure 84



$A = BH - bh$ $\quad Z = \dfrac{BH^3 - bh^3}{6H}$

$I = \dfrac{BH^3 - bh^3}{12}$ $\quad r = \sqrt{\dfrac{BH^3 - bh^3}{12\,(BH - bh)}}$

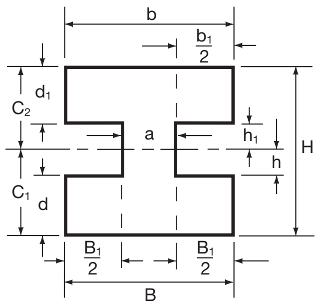
Figure 85



$A = BH - bh$ $\quad Z = \dfrac{BH^3 - bh^3}{6H}$

$I = \dfrac{BH^3 - bh^3}{12}$ $\quad r = \sqrt{\dfrac{BH^3 - bh^3}{12\,(BH - bh)}}$

Figure 86



$$A = (Bd + bd_1) + a(h + h_1)$$

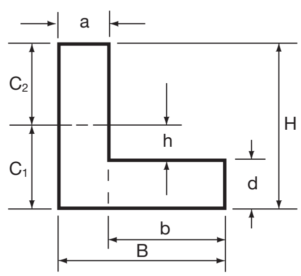
$$I = \frac{1}{3}(BC_1^3 - B_1h^3 + bc_2^3 - b_1h_1^3)$$

$$C_1 = \frac{1}{2}\,\frac{aH^2 + b_1d^2 + b_1d_1\,(2H - d_1)}{aH + B_1d + b_1d_1}$$

$$C_2 = H - C_1$$

$$r = \sqrt{\frac{I}{(Bd + bd_1) + a(h + h_1)}}$$

Figure 87



$$A = Bd + a(H - d)$$

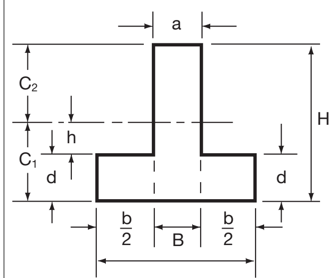
$$I = \frac{1}{3}(BC_1^3 - bh^3 + aC_2^3)$$

$$C_1 = \frac{1}{2}\left[\frac{aH^2 + bd^2}{aH + bd}\right]$$

$$C_2 = H - C_1$$

$$r = \sqrt{\frac{I}{Bd + a(H - d)}}$$

Figure 88



$$A = Bd + a(H - d)$$

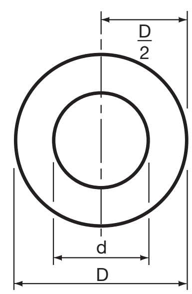
$$I = \frac{1}{3}(BC_1^3 - bh^3 + aC_2^3)$$

$$C_1 = \frac{1}{2}\left[\frac{aH^2 + bd^2}{aH + bd}\right]$$

$$C_2 = H - C_1$$

$$r = \sqrt{\frac{I}{Bd + a(H - d)}}$$

Figure 89



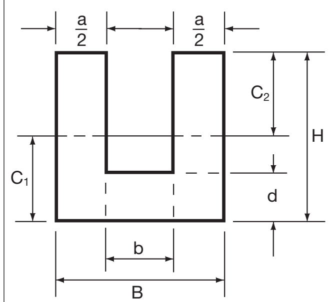
$$A = \frac{\pi}{4}(D^2 - d^2)$$

$$I = \frac{\pi(D^4 - d^4)}{64} = 0.049\,(D^2 - d^2)$$

$$Z = \frac{\pi(D^4 - d^4)}{32D} = 0.098\,\frac{(D^4 - d^4)}{D}$$

$$r = \frac{\sqrt{D^2 + d^2}}{4}$$

Figure 90



$$A = Bd + a(H - d)$$

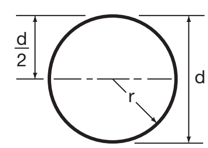
$$I = \frac{1}{3}(BC_1^3 - bh^3 + aC_2^3)$$

$$C_1 = \frac{1}{2}\left[\frac{aH^2 + bd^2}{aH + bd}\right]$$

$$C_2 = H - C_1$$

$$r = \sqrt{\frac{I}{Bd + a(H - d)}}$$

Figure 91



$$A = \frac{\pi d^2}{4} \qquad Z = \frac{\pi d^3}{32}$$

$$I = \frac{\pi d^4}{64} \qquad r = \frac{d}{4}$$

Flat Plate Formulas

Flat Plates

A flat plate of uniform thickness is used in many designs to support a load perpendicular to the plate. Figures 92 to 95 give stress and deflection equations for several common plate configurations. Again, these equations are valid when working with a homogeneous, isotropic material, and when deflection is less than about one-half of the plate thickness.

Where:

a = radius of circular plate

$D = \frac{Eh^3}{12\,(\ell - \mu^2)}$ flexural rigidity of plate

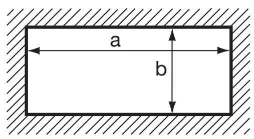
E = apparent modulus of elasticity

h = plate thickness

v = Poisson's ratio

q = uniform load per unit area

Figure 92 – Rectangular plate, all edges fixed, uniform load



Rectangular plate, all edges fixed, uniform load

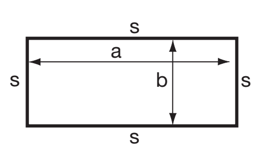
Deflection at center:

$$y = \frac{0.0284\ qa^4}{Eh^3\,[1 + 1.05\,(\frac{a}{b})^5]}$$

Maximum stress (at center of long edge):

$$\max S = \frac{qa^2}{2h^2\,[1 + .0623\,(\frac{a}{b})^6]}$$

Figure 93 – Rectangular plate, all edges simply supported, uniform load



Rectangular plate, all edges simply supported, uniform load

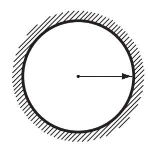
Deflection at center:

$$y = \frac{0.142\ qa^4}{Eh^3\,[1 + 2.21\,(\frac{a}{b})^3]}$$

Maximum stress (at center):

$$\max S = \frac{0.75qa^2}{h^2\,[1 + 1.61\,(\frac{a}{b})^3]}$$

Figure 94 – Circular plate, fixed edges, uniformly distributed load



Circular plate, fixed edges, uniformly distributed load

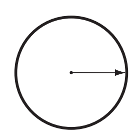
Deflection at center: $y = \frac{qa^4}{64D}$

Moment at center: $M = \frac{qa^2\,(1+\mu)}{16}$

Maximum stress (at center):

$\max S = \frac{3qa^2}{4h^2}$

Figure 95 – Circular plate, simply supported edges, uniformly distributed load



Circular plate, simply supported edges, uniformly distributed load

Deflection at center: $y = \frac{qa^4\,(5+\mu)}{64D\,(1+\mu)}$

Moment at center: $M = \frac{qa^2\,(3+\mu)}{16}$

Maximum stress (at center):

$\max S = \frac{3qa^2\,(3+\mu)}{8h^2}$

Thin-Walled Tubing

Figure 96 and the equations provided can be used to calculate the stress and deformation of thin-walled tubing under internal pressure when neither end of the tubing is closed. This also applies to fairly long tubes, or in situations remote from the tube ends. As long as the wall thickness is less than about one-tenth of the radius, the circumferential or hoop stress (σ_2) is practically uniform throughout the thickness of the wall, and the radial stress (σ_3) is negligible. As usual, the appropriate time- and temperature-dependent modulus must be calculated for specific applications. Significant error can result if the thin-wall equations are used in calculations that involve thick walls.

$$\sigma_1 = \frac{qr}{2t}$$

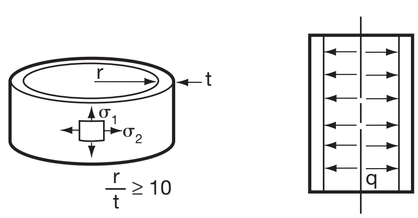
$\sigma_1 = 0$, if longitudinal pressure is zero or is externally balanced

$$\sigma_2 = \frac{qr}{t}$$

$$\Delta r = \frac{qr}{Et}$$

See Figure 96 for definitions.

Figure 96 – Thin Walled Tubing



r = radius
t = thickness
q = internal pressure
σ_1 = see calculation
σ_2 = hoop stress
σ_3 = radial stress
E = modulus
Δr = change in radius

Thick-Walled Pressure Vessels

Equations for design of thin-walled pressure vessels can be used to design thick-walled pressure vessels to be fabricated from thermoplastics. However, several guidelines need to be considered. First, include generous safety factors in the design to allow for the geometrical differences at the joint of the end-plate and the cylinder. These differences can cause maximum stresses, many times the nominal hoop stress, depending on the plate-to-wall joint design. Also, the ratio of wall thickness to mean radius should not exceed approximately 1:10 to avoid a triaxial stress state – with stresses acting in three directions – which can reduce the ductility of plastics and most other materials. And, of course, the modulus must be selected carefully.

Remember always that design analysis and calculations cannot take into consideration such factors as weld lines, the effect of gate location, orientation of the polymer, or variations in polymer density. Therefore, the design should always be verified by fabricating and testing prototypes. For example, a typical pressure vessel evaluation would include fatigue testing (cyclic pressurization) and hydrostatic burst testing. (For more information on the effect of weld lines and gate location, see page 66.) For the equations appropriate to a specific situation, consult your general engineering handbook.

Rotating Disks

Because of their high strength-to-weight ratio, dimensional stability, resistance to creep and relaxation, and their impact strength, engineering thermoplastics are excellent materials for rotating disks, such as impellers.

The total stress on an impeller is calculated by adding:

- Bending stresses due to the pressure differential.
- Localized bending stresses due to the attachment of a blade.
- Inertial stresses due to high-speed rotation.

Make sure that the total stress is within the design limits based on service conditions.

Bending stresses are calculated using standard stress and deflection equations. The inertial stresses developed by high-speed rotation can be estimated by using the following flat-disk equations. In all of the equations, v is Poisson's ratio, which is defined on page 40.

Rotating Disk Equations

A. For a solid, homogeneous, circular disk of uniform thickness, having radius R (mm) and density r (g/cm³), rotating about its centroidal axis with a constant angular velocity, v (rad/sec):

1. Radial tensile inertia stress (s_r) at a point which is distance r from the center, is given as:

$$\sigma_r = \frac{1}{8} \times \frac{\rho\,\omega^2}{386.4}(3+v)(R^2 - r^2)$$

2. Tangential tensile inertia stress σ_t) is given as

$$\sigma_t = \frac{1}{8} \times \frac{\rho\,\omega^2}{386.4}\quad (3+v)R^2 - (1+3v)r^2$$

3. Maximum radial and maximum tangential stresses are equal and occur at the center (r = 0).

$$\text{Max } \sigma_r = \text{Max } \sigma_t = \frac{1}{8} \times \frac{\rho\,\omega^2}{386.4}(3+v)R^2$$

B. For a homogeneous, annular disk of uniform thickness with an outer radius R (mm), a central hole of radius R_0 (mm), and density r (g/cm³), rotating about its centroidal axis with a constant angular velocity v (rad/sec):

1. At any point a distance r from the center radial tensile stress (σ_r) is given as

$$\sigma_r = \frac{3+v}{8} \times \frac{\rho\,\omega^2}{386.4}\left(\frac{(R^2 + R_0^{\,2} - R^2R_0^{\,2} - r^2)}{r^2}\right)$$

2. Tangential tensile inertia stress (σ_t) is given as

$$\sigma_t = \frac{1}{8} \times \frac{\rho\,\omega^2}{386.4}\left[(3+v)\left(R^2 + R_0^{\,2} + \frac{R^2R_0^{\,2}}{r^2}\right) - (1+3v)r^2\right]$$

3. Maximum radial stress (Max σ_r) occurs at $r = \sqrt{RR_0}$ and is given as

$$\text{Max } \sigma_r = \frac{3+v}{8} \times \frac{\rho\,\omega^2}{386.4}\quad (R - R_0^{\,2})^2$$

4. Maximum tangential stress (Max σ_t) occurs at the perimeter of the hole and is given as

$$\text{Max } \sigma_t = \frac{1}{4} \times \frac{\rho\,\omega^2}{386.4}\left[(3+v)R^2 + (1-v)R_0^{\,2}\right]$$

Designing for Equal Stiffness

Equivalent Thickness

When a thermoplastic is specified as replacement for another material (a metal, for example) the new part often needs to have the same stiffness as the old one. Essentially, that means making sure that the new part, when subjected to the same load, will have the same deflection as the old part.

Deflection in bending is proportional 1/EI (E = modulus and I = moment of inertia), and I is proportional to t^3 (t = thickness). Thus, the equivalent thickness of a plain, flat part to be made from a thermoplastic can be calculated by the following equation:

$$t_2 = t_1 \sqrt[3]{\frac{E_1}{E_2}}$$

where:

E_1 = flexural modulus of material being replaced
E_2 = flexural modulus or creep modulus of replacement thermoplastic
t_1 = thickness of old material
t_2 = required thickness of thermoplastic

A thickness conversion factor (TCF) can be calculated on the basis of the cube root of the ratio of the moduli of the two materials. Table 17 lists the thickness conversion factors for several common structural materials relative to steel. These factors are based on the short-term, room temperature modulus values. Conversion factors based on the long-term and/or high temperature modulus (that is, the creep modulus) will be different from those shown here.

For example, to find what thickness of a thermoplastic component is required for equal stiffness relative to steel, multiply the thickness of the steel component by the conversion factor, TCF, in Table 17:

$$t_2 = t_1 \times TCF$$

where:

$$TCF = \sqrt[3]{\frac{E_1}{E_{ST}}}$$

and E_{ST} = flexural modulus or creep modulus of steel.

To determine the thickness of material (Y) required for a thermoplastic part that will give the same stiffness as when the part is made with a material (Z) other than steel, multiply the thickness of the part in material (Z) by the TCF (from Table 17) for the thermoplastic relative to steel, and then divide by the TCF for the material (Y) relative to steel.

Table 17 – Thickness Conversion Factors for Common Structural Materials Relative To Steel

Replacement Material	Flexural Modulus S.I. GPa	Flexural Modulus English ksi	Flexural Modulus Metric kg/cm²	Thickness Conversion Factor
ABS	2.6	3.8×10^5	2.7×10^4	4.29
Acrylic	3.0	4.4×10^5	3.1×10^4	4.12
Aluminum, cast	71.0	1.0×10^7	7.2×10^5	1.43
Brass	96.5	1.4×10^7	9.9×10^5	1.29
Ceramics ($A\ell_2 0_3$)	344.8	5.0×10^7	3.5×10^6	0.84
Glass	69.0	1.0×10^7	7.0×10^5	1.44
PC	2.4	3.5×10^5	2.5×10^4	4.41
PP	1.2	1.7×10^5	1.2×10^4	5.63
PS	3.3	4.8×10^5	3.4×10^4	3.97
Polysulfone	2.5	3.6×10^5	2.6×10^4	4.37
Steel	206.9	3.0×10^7	2.1×10^6	1.00
Timber (average of a variety of structural timbers)	11.7	1.7×10^6	1.2×10^5	2.60
SAN	3.6	5.2×10^5	3.7×10^4	3.88
Zinc, die cast	44.8	6.5×10^6	4.6×10^5	1.66

The following calculations illustrate both methods of finding equivalent thickness when redesigning in polycarbonate.

To calculate the thickness of a part that, when made in polycarbonate, will have the same deflection as a 0.75 mm thick aluminum part at 73°F (23°C).

A. Using the moduli of the two materials:

E_1 = modulus of aluminum at 73°F (23°C)
= 7.2×10^4 MPa

E_2 = modulus of polycarbonate at 73°F (23°C)
= 2.41×10^3 MPa

$t_1 = 0.75$

$t_2 = ?$

$$t_2 = t_1 \sqrt[3]{\frac{E_1}{E_2}}$$

$$= 0.75 \sqrt[3]{\frac{71,000}{2,410}}$$

$$= 2.3 \text{ mm}$$

B. Using the thickness conversion factors from Table 17:

$TCF_{AL/ST}$ = TCF for aluminum relative to steel
= 1.43

$TCF_{PC/ST}$ = TCF for polycarbonate relative to steel
= 4.41

$TCV_{PC/AL}$ = TCF for polycarbonate relative to aluminum

$$= \frac{TCV_{PC/ST}}{TCF_{AL/ST}}$$

$$= \frac{4.41}{1.43}$$

$$= 3.08$$

Therefore: $t_2 = 0.75 \times 3.08$
= 2.3 mm

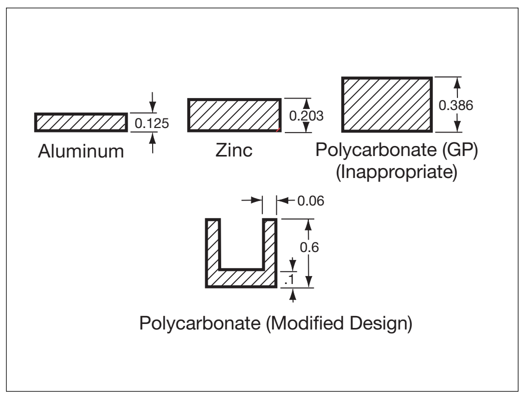
Remember that stiffness is proportional to thickness cubed (t^3). This means an increase in thickness of only 26% will double part stiffness.

Ribs

Occasionally, the calculations for an equivalent thickness of a thermoplastic to a plain, flat plate can give results that would be too thick to be economical or practical. As the moment of inertia is proportional to thickness cubed, the addition of ribs to a relatively thin plate is an effective way to increase the stiffness.

Figure 97 shows four cross-sections of equal stiffness. The straight conversion factor for polycarbonate is bulky, uneconomical and inappropriate. The use of ribs in the part made with polycarbonate will allow a thinner overall wall thickness. By allowing thinner walls, ribbing also reduces molding cycle time and cross-sectional area, and reduces material usage and product weight without sacrificing physical properties. You may wish to consider other methods of stiffening such as corrugating and doming.

Figure 97 – Calculations for Equal Stiffness, Ribbing with Polycarbonate Resins



Designing for Impact Resistance

The impact resistance exhibited by an actual part depends on the design of the part, the material used, and the conditions of fabrication.

Designing for impact is complex. The shape and stiffness of the striking body, the shape of the part, the inertia of both, and end-use conditions can all affect impact strength. The following section gives you general design guidelines for improving impact strength. These guidelines comprise a sound approach to the design challenge, but are not a substitute for production of and testing for prototype parts in the actual conditions of use.

Part Design for Impact Resistance

Because the part must be able to absorb the energy of impact, part design is probably the greatest single factor – other than proper material selection – in determining impact strength. Part design will improve the impact resistance when you take care to:

- Provide walls that flex rather than rigidly resist impact loading.
- Use rounded corners so that they can give with the impact and provide a smoother transfer of energy. (See the discussion on corner radius in “Product Design” page 56.)
- Avoid any abrupt changes in stiffness (due to changes in wall thickness or structural reinforcement), which tend to concentrate impact loading. This includes such features as ribs, holes, and machined areas. (See Chapter 5, page 56 for design guidelines on wall thickness, transition zones and ribs.)

Mold Design

Impact strength can also be improved by good mold design. In this:

- Position gates away from high impact areas. (See page 66 for more information on gate location.)
- Place weld lines, whenever possible, away from high impact areas. (See page 66 for more information on weld lines.)
- Core-out thick sections to reduce packing stresses and improve flexibility.

Assembly

The method of assembly can also affect a part’s impact strength. Rigid joints can cause abrupt transitions in energy flow, which can break the joint. Joints, like walls and corners, should be flexible. Assembly techniques are discussed in Chapter 7.

Designing for Thermal Stress

Thermal expansion and contraction are important considerations in plastics design, and are often overlooked. Expansion-contraction problems often arise when two or more parts made of materials having different coefficients of thermal expansion are assembled at a temperature other than that of the end-use environment. When the assembled parts go into service in the end-use environment, the two materials react differently, and the resultant thermal stresses can cause unexpected part failure.

So, you must consider the effects of thermal expansion and/or contraction early in the design of parts that involve close fits, molded-in inserts, and mechanical fastenings. Coefficients of thermal expansion for some common materials are given in Table 18.

Thermal stress can be calculated by using the following equation:

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

$$\text{or } \epsilon = (\alpha_1-\alpha_2)\Delta T$$

Where:

α_1 = coefficient of thermal expansion of one material
α_2 = coefficient of thermal expansion of second material
E = modulus
ΔT = change in temperature, °F (°C)
ϵ = strain, mm/mm
κ = constant (roughly 1.0 for most conditions)

The following calculations illustrate the use of thermal stress equations:

Calculate the strain (ϵ) on a part made of polycarbonate and close fitting onto a steel bracket. The parts are assembled at a room temperature of 73°F (23°C) and operated at an environmental temperature of 180°F (82°C).

A. Select values of coefficients from Table 18:

a_1 = coefficient of polycarbonate = 6.8×10^{-5}

a_2 = coefficient for steel = 1.2×10^{-5}

B. Calculate the change in temperature:

DT = 180°F (82°C) - 73°F (23°C) = 138°F (59°C)

C. Choose the appropriate thermal stress equation and insert values:

$$e = (a_1-a_2)\,DT$$
$$= (6.8 \times 10^{-5} - 1.2 \times 10^{-5}\ \text{mm/mm/°C}) \times 59°C$$
$$= 0.0033\ \text{mm/mm}\ (0.33\%)$$

Because the steel bracket restrains the expansion of the polycarbonate part, a strain of 0.33% is induced in the part.

Table 18 Coefficients of Thermal Expansion of Various Structural Materials

Material	Coefficient of Thermal Expansion S.I. mm/mm/°C	English in/in/°F	Metric mm/mm/°C
ABS	9.5×10^{-5}	5.3×10^{-5}	9.5×10^{-5}
Aluminum	2.2×10^{-5}	1.2×10^{-5}	2.2×10^{-5}
Brass	1.8×10^{-5}	1.0×10^{-5}	1.8×10^{-5}
Nylon	8.1×10^{-5}	4.5×10^{-5}	8.1×10^{-5}
PBT	7.4×10^{-5}	4.1×10^{-5}	7.4×10^{-5}
PC	6.8×10^{-5}	3.8×10^{-5}	6.8×10^{-5}
PE	12.0×10^{-5}	6.7×10^{-5}	12.0×10^{-5}
PP	5.8×10^{-5}	3.2×10^{-5}	5.8×10^{-5}
PS	8.1×10^{-5}	4.5×10^{-5}	8.1×10^{-5}
SAN	6.7×10^{-5}	3.7×10^{-5}	6.7×10^{-5}
Steel	1.1×10^{-5}	0.6×10^{-5}	1.1×10^{-5}

Chapter 7

Designing for Machining and Assembly

Machining
Assembly Design
Mechanical Assembly
Bonding
Welding

Machining

Tools designed for cutting steel work well when cutting most thermoplastics, and generally provide a long service life. Either high-speed steel or carbide tooling can be used, but carbide types offer higher feed and speed capabilities. If extensive machining is necessary, use tools with optimum geometry (as defined by The Society of the Plastics Industry in the United States) to ensure maximum productivity and good surface finish.

No matter what tool-tip geometry is used, the tool must be sharp, honed, and polished. A dull tool causes poor finish, gumming, and dimensional problems due to heat build-up.

Finished products may require polishing to prevent machine marks acting as stress concentration points.

Assembly Design

Engineering thermoplastics are often used in demanding, complex, and diverse applications that require post-mold assembly of finished parts. Typically, two or more components – of similar or dissimilar materials – must be mated into an assembly.

Assembly methods for parts made of thermoplastics are numerous: ranging from relatively simple mechanical fits to complex welding operations.

Table 19 indicates the effectiveness of various assembly methods when used for parts made from several types of plastic materials.

Each assembly method has advantages and disadvantages when used for thermoplastic parts. The decision as to which method is best suited for a particular application should be based on several factors: product requirements, technical expertise, production requirements, equipment availability, and costs. It is important to consider all these factors during the product design stage, so the parts and tooling can be designed to meet assembly needs.

The following section gives detailed descriptions of the different techniques used to assemble parts made from thermoplastics.

Table 19 – Effectiveness of Various Assembly Methods for Common Thermoplastics

Material	Mechanical	Solvents	Adhesives	Welding
PC	G	G	G	G
ABS	G	G	G	E
Acetal	E	P	P	F – G
Acrylic	G	G	G	G
Nylon	G	P	P	F – G
PP	P	P	P	F – G
PS	F – G	G	G	E
Polysulfone	G	G	G	F – E
SAN	G	G	G	E

E = Excellent G = Good F = Fair P = Poor

Mechanical Assembly

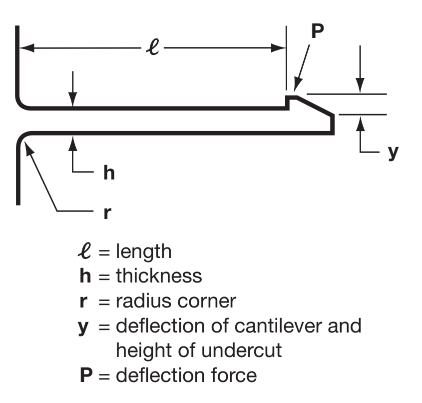
Mechanical methods of assembly are the most basic means of fastening plastic parts, partly because these methods have been used traditionally in the metals industry. There are two major types of mechanical assemblies: those using fits and those using fasteners. Fits include snap-fits, press-fits, and staking. Fasteners include screws, thread-forming screws, or machine screws with nuts or clips, and rivets.

Snap-Fits

Correctly designed snap-fits are simple, economical, fast, and dependable. Snap-fits can be applied to any combination of materials. Their strength comes from mechanical interlocking and, once assembled, a properly designed snap-fit is not under load, so its strength does not decrease with time and will not loosen under vibration.

The most common type of snap-fit is the cantilever, shown in Figure 98. This design is most suitable for thermoplastics having low mold shrinkage, high resistance to creep, and good overall dimensional stability.

Figure 98 – Cantilever Snap-Fit Design



Permissible Deflection

The deflection (y) that occurs during assembly of a cantilever snap-fit is equal to the undercut, as shown in Figure 98. The permissible deflection for a cantilever beam of constant rectangular cross-section is calculated as:

$$y = \frac{2}{3} \frac{\epsilon \ell^2}{h}$$

Where:

y = maximum deflection
ϵ = maximum fiber strain
ℓ = length of beam
h = thickness

This deflection should not be exceeded during ejection from the mold or during assembly.

You can increase the permissible deflection by increasing the beam's length or decreasing its thickness. Increasing length is more effective since length appears in the equation to the second power. (Another method of increasing deflection is discussed in "Tapered Cantilever Beams," page 100.)

Permissible Strain

Permissible deflection depends on the permissible strain (ϵ) as well as the shape of the beam. Amorphous materials, such as polycarbonate, polystyrene, PC/ABS, and ABS, can be strained up to approximately 70% of the yield strain during a single, brief snap-fit. However, if frequent assembly and disassembly are anticipated, the strain level should be reduced to about 60% of that value. For example, the permissible strain for polycarbonate is 4% for a single assembly and about 2.4% for frequent assembly/disassembly.

Material	**Permissible Strain[1] (single snap or assembly)**
ABS	1.4
Polycarbonate	4.0
High-Impact Polystyrene	0.7
PC/ABS	2.4

[1]These are general guidelines only. Accurate part design requires the use of data produced with loading, environmental and geometry conditions similar to the part's end-use conditions.

Deflection Force

The transverse deflection force (P) required to bend the cantilever by the amount of the undercut (y) is calculated as:

$$P = \frac{bh^2}{6} \times \frac{E_S \epsilon}{\ell}$$

where:

E = secant modulus
ϵ = strain at which cantilever is operated
b = width (or base)
h = thickness at the base
ℓ = length

Assembly Force

To assemble the snap, the deflection force (P) and frictional force have to be overcome. The assembly force (W) is applied at the end of the beam via a lead-in angle and is calculated as:

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

where:

W = assembly force
P = deflection force
μ = coefficient of friction
α = lead-in angle

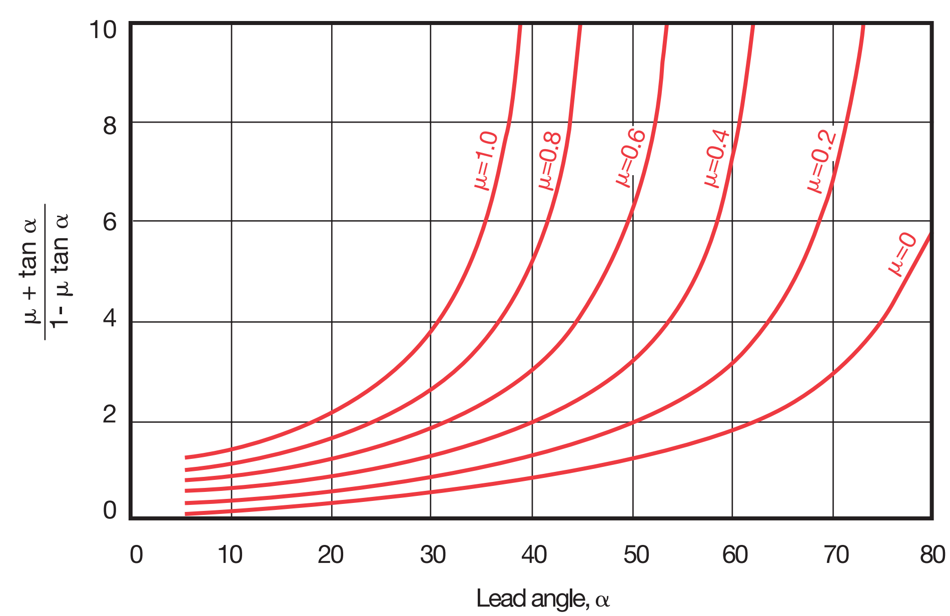
Values for $\left[\frac{\mu + \tan\alpha}{1 - \mu\tan\alpha}\right]$

can be taken directly from Figure 99. The assembly force can be made greater than, equal to, or less than the deflection force by choice of lead-in angle.

Coefficient of Friction

In the context of assembling parts with snap-fits, the values of the coefficient of friction in Table 20 depend on the relative speed of the assembly, the pressure applied during assembly, and the surface finish quality of the mating parts. The values represent friction between two different plastic materials. With two components of the same plastic material, the friction coefficient is generally higher, as noted in Table 20. (For more information on the coefficient of friction, see page 28.)

Figure 99 – Determination of Lead-in Angle by Magnitude of Assembly Force



Disassembly Force

When designing separable joints, the disassembly force is calculated with the same equation as the assembly force, substituting the return angle (α_1) for the lead-in angle (α). In either case, the angle can be varied so the assembly (or disassembly) force is greater than, less than, or equal to the deflection force. The smaller the angle is, the easier it is to snap in or snap out. If the return angle (α_1) exceeds ($90° - \tan^{-1}\mu$), the joint is inseparable or self-locking.

Figure 100 shows the details of a cantilever undercut, with all forces and angles labeled.

Figure 100 – Design of the Cantilever Undercut for Snap-Fit Assemblies, Disassemblies

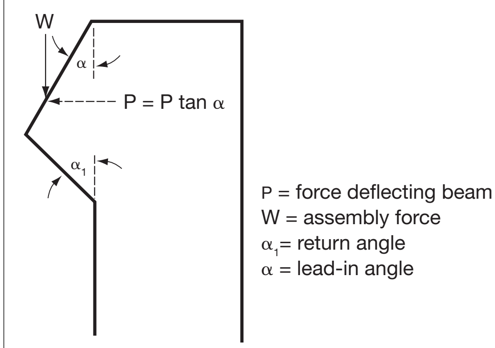


Table 20 – Coefficient of Friction – Between Two Different Plastic Materials

Material	Friction Coefficient	Conversion Factor for Components of Same Material
ABS	0.50 – 0.60	x 1.2
PC	0.35 – 0.40	x 1.2
Nylon	0.30 – 0.40	x 1.5
PBT	0.35 – 0.40	x 1.2
PE flexible	0.55 – 0.60	x 1.2
PE rigid	0.20 – 0.25	x 2.0
Acrylic	0.50 – 0.60	x 1.2
PP	0.25 – 0.30	x 1.5
PS	0.40 – 0.50	x 1.2
PVC	0.55 – 0.60	x 1.0
SAN	0.45 – 0.55	x 1.2

Tapered Cantilever Beams

One way to increase the permissible deflection for a cantilever beam is to taper either the thickness (h) or width (b) of the beam from the base to the hook. Tapering the beam provides a more uniform distribution of stress and reduces material usage for the same deflection force and assembly force. For example, tapering the thickness of a beam to half its base dimension (while holding the other variables constant) increases the permissible deflection by more than 60% over that of the same beam with a constant thickness.

Tapering either the thickness or width will change the value of K, the proportionality constant, in the tapered beam deflection equation:

$$Y = \frac{2}{3} K \frac{\epsilon \ell^2}{h}$$

Values for K can be found in Figure 101 for tapered thickness beams and in Figure 102 for tapered width beams.

Figure 101 – Proportionality Constant for Tapered Thickness Beams

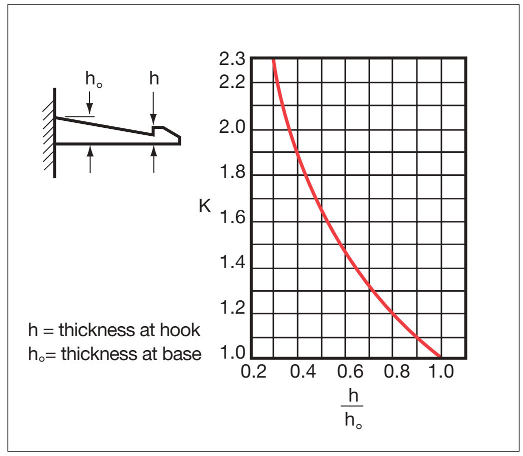


Figure 102 – Proportionality Constant for Tapered Width Beams

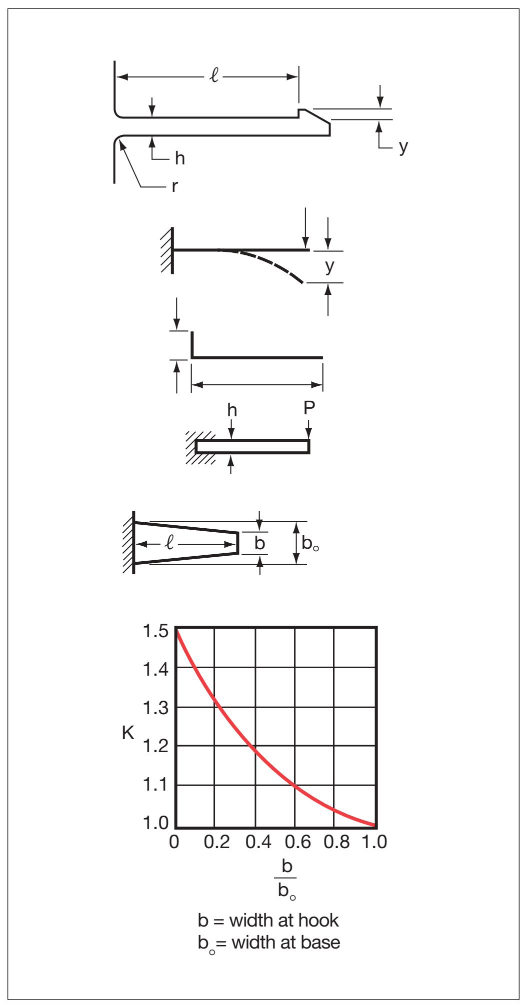
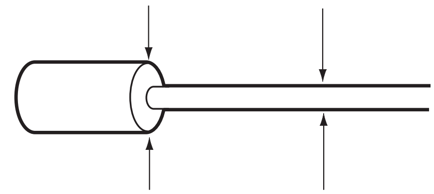


Figure 103 – Example of Interference Fit: Plastic Hub Pressed on Steel Shaft



$$\frac{\text{Shaft Diameter}}{\text{Hub Outer Diameter}} = \frac{.250}{.500} = .5$$

From Figure 104, the maximum Diametral

$$\text{Interference} = 8 \frac{\text{mils}}{\text{inch of shaft diameter}}$$

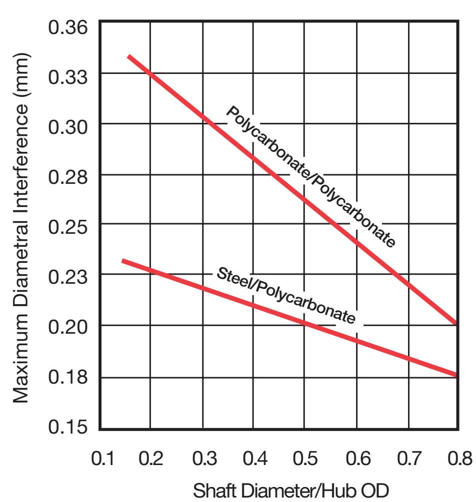
$\therefore$ Diametral Interference = .008 x .250
= .002 inches

Thus, for: Shaft Diameter = .250
Hub Outer Diameter = .500

Then: Hub Inner Diameter = .248
Hub Wall = .126

Also note that tolerances of the two mating parts may cause the interference to vary considerably.

Figure 104 – Recommended Diametral Interference @ 23°C (73°F), Polycarbonate Resins



Press-Fits

The press-fit is perhaps the most basic of all assembly techniques. It is fast, relatively simple, and economical. But it can also be the most troublesome if incorrectly designed or badly manufactured.

When designing components for a press-fit, make sure the design provides holding strength adequate to meet the assembly requirements without over-stressing the assembly. This potential problem is complicated by three factors:

- Press-fit designs require close manufacturing tolerances.
- Most thermoplastics will fail under long-term loading if the stress exceeds a critical value. For example, the critical stress value of polycarbonate for long-term loading is approximately 14 MPa.
- Part dimensions will change with time due to creep and relaxation. (See page 49 for a discussion of creep, and page 53 for a discussion of relaxation.)

Interference

In designing a press-fit between two parts made of rigid materials, you should minimize the interference between the two parts to keep the assembly stresses at acceptable levels. An example of a press-fit is a steel shaft pressed into a polycarbonate hub, as shown in Figure 103. In this design, the maximum obtainable hoop stress has to be evaluated for the case of maximum shaft diameter, and minimum obtainable hoop stress has to be evaluated for the case of maximum shaft diameter and minimum inside diameter hub to determine that the hoop stress does not exceed the allowable stress. Diametral interferences for polycarbonate resins (at 23°C, 73°F) are given in Figure 104.

Creep

Another consideration in the design of press-fits is the effect of stress relaxation (creep) over time. Creep is the change in dimensions of a molded part resulting from cold flow incurred by continual loading. The amount of creep and the time necessary to produce creep deformation depend on several factors: the type of material, stress levels, and environmental conditions such as temperature, humidity, etc.

Creep can cause a press-fit that was considered satisfactory at the time of assembly to loosen to an unacceptable condition or even failure.

For the press-fit design under consideration, a common method of overcoming this problem is to incorporate grooves in the shaft. This reduces the assembly stresses, and thereby the degree of creep. After assembly and over time, the plastic will cold flow into the grooves and maintain the desired holding strength of the fit.

Staking

Staking is another basic assembly method transferred from the metals industry for use with thermoplastics. In this process, a metal insert (such as a threaded insert, electrical connector post, stud, hypodermic needle, etc.) is placed into a plastic boss. Then the plastic is forced to cold flow onto or around the metal insert. Normally such metal inserts are undercut or knurled, and the plastic flows into the undercut, improving retention.

However, cold staking does produce a high level of residual stress in the plastic part and is not suitable for use with materials that may crack under the residual stress. For example, cold staking is not recommended with polycarbonate.

Screw Fastening

Assembling with screws allows components to be repeatedly assembled and disassembled. This is important in many applications where the unit incorporating the plastic part can be expected to undergo modifications, repairs, or where it may provide access into an assembly. Like metals, most thermoplastics can accommodate many types of screw assemblies. The four major methods are:

- To screw directly into the thermoplastic part, using self-tapping screws.
- To screw into a threaded insert that is incorporated within the part.
- To pass the screw through the part and secure it with an external nut or clip.
- To mold threads into or onto the thermoplastic.

Self-tapping Screws

There are two types of self-tapping screws: thread forming and thread cutting. Thread-forming screws are not recommended for use with materials that crack under sustained loads because they induce high stresses into the plastic as the thread is formed. Thread-cutting screws (such as Type 23 or 25) are recommended for use in these materials because they form threads by actually cutting away the plastic material, inducing minimal deformation and reducing hoop stress. Countersunk-head screws should be avoided because the wedging action causes high hoop stress.

If the unit is to be repeatedly assembled and disassembled, we advise you to specify Type 23 screws. If a self-tapping screw is removed from an assembly, always replace it with a standard-pitch machine screw on reassembly. Otherwise, the self-tapping screw may cut a new thread over the original thread, resulting in a stripped thread.

This assembly method allows for only a minimal number of disassemblies and reassemblies; repeated removal and insertion of the screw decreases the strength of the material. For applications requiring frequent reassembly, ultrasonically applied metal inserts are suggested (see page 104).

Screw Manufacturers

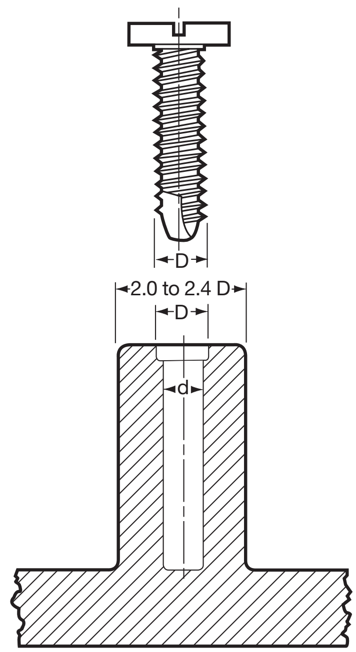
Suppliers can provide you with specifications for screw design. Generally, several types of screws are available in most standard thread sizes.

Design criteria for the mating plastic boss to be molded in thermoplastics are summarized in the recommendations in Figure 105. Additional information on boss design is provided on page 59.

The following recommendations are applicable to Figure 105:

- The entry counterbore diameter should be equal to the major diameter (D) of the screw thread and approximately equal to a depth of one pitch.
- The inside diameter of the boss (d) should be equal to the pitch diameter of the screw thread.
- The outside diameter of the boss should be 2 to 2.5 times the major diameter (D) of the screw thread.
- The minimum thread engagement should be 2.5 times the pitch diameter.
- Either a through hole or a blind hole in the boss will provide adequate melt flow. The base thickness of a blind hole should be equal to nominal wall thickness.

Figure 105 – Example of Boss Design for Self-Threading Screws, Polycarbonate Resins



Thread Size	Dia (mm) d	Dia (mm) D
6-20	2.92	3.50
8-18	3.56	4.17
10-16	4.01	4.83
12-14	4.62	5.49

Threaded Inserts

Threaded inserts, usually made of nonferrous metal, are used in many designs. They can be built in the plastic part as molded-in inserts, heat or pressure inserts, ultrasonic inserts, or expansion inserts. The preferred method for embedding the insert into a thermoplastic part is ultrasonic insertion, because it imparts low residual stresses and is inexpensive. The least preferred method, because of the high residual stresses that result, is expansion insertion.

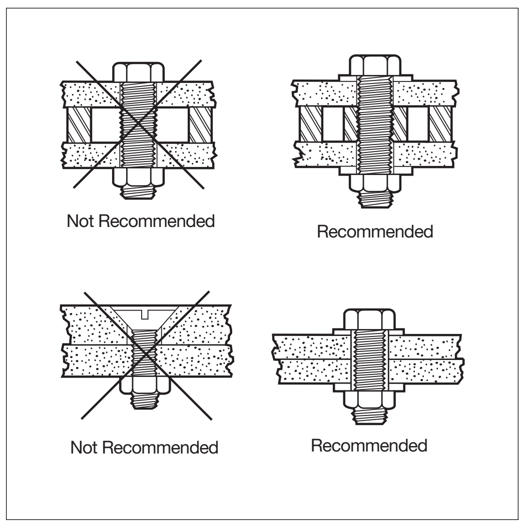
For good design, inserts should not have sharp corners or edges that could act as notches or stress concentrators. An undercut with a flat or smooth knurl minimizes notch sensitivity, yet still provides acceptable pull-out and torque levels. If you are considering using threaded metal inserts, consult the insert supplier and/or your Dow Information Center for more information.

Bolts with Nuts

Bolts or screws that pass through the plastic part and are retained by an external nut or clip provide a simple, convenient assembly method. This method can be used for multiple reassemblies and, if correctly designed, is unaffected by the amount of torque applied to the plastic. Figure 106 illustrates good and bad practices. Good design for this assembly method requires attention to the following:

- Design the joint area to eliminate any space between the two plastic surfaces being assembled. This puts the assembly in compressive loading instead of bending or tensile loading, reducing tensile stresses that can cause failure. A spacer or boss may be needed to accomplish this, as shown in Figure 106.
- Use a washer to distribute the high torque loading over a greater surface area.

Figure 106 – Assembly Design for Bolts with Nuts



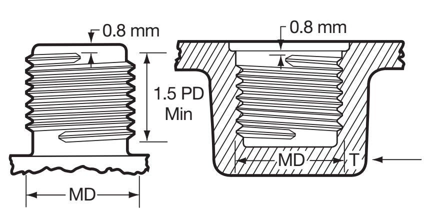
Molded-in Threads

Most standard thread designs (including those with multi-start threads) can be molded into thermoplastics. There are only three limitations. First, avoid extra-fine threads. These are difficult to fill, and usually are not strong enough to withstand torque requirements. Second, threads should not have sharp corners. These can form notches and decrease the screw-retention values. Third, threads should always have a radius at the root to avoid stress concentrators.

The following dimensional guidelines should be helpful in designing molded-in threads. For more information, see “Product Design,” page 56.

- Avoid running threads out to the edge of the screw base. Leave a gap of approximately 0.8 mm, as shown in Figure 107.
- Minimum active thread length should be 1.5 times the pitch diameter of the thread.
- Minimum wall thickness around the internal thread should be 0.5 times the major diameter of the thread.
- Avoid the use of tapered pipe threads. As the threaded part is increasingly tightened, the hoop stress increases.

Figure 107 – Example of Design for Molded-in Threads, Polycarbonate Resins



Rivets

Assembling with rivets (either solid or “pop”) can be useful because of the low cost, ease of use, and high degree of precision of this method. Rivets can be used to attach a thermoplastic part to itself, to metals, or to other plastics.

Use rivets made of aluminum, because of its ability to deform under load (which limits the compressive forces imparted during the riveting process), and because the coefficients of thermal expansion of aluminum and many thermoplastics are similar. (See the sections on deformation under load on page 45, and on thermal expansion, page 91, for further information.) As with most mechanical assemblies, incorporating a washer to distribute the loading over a greater area is advisable.

Bonding

Solvent Bonding

Amorphous thermoplastics are more suitable for solvent bonding than crystalline materials. A solvent commonly used is methylene chloride at varying concentrations. The main limitation of this technique is in the handling of the solvent.

In solvent bonding, the solvent is applied to the joint area of one or both components, and the components are then held together in a fixture. While the parts are held together, and being subjected to pressure, the bond cures to form a joint. The pressure and time required depends on the thermoplastic material, the solvent, and the joint design.

Adverse environmental conditions, such as elevated temperatures, can cause stress crazing. Therefore, the bond should be dried for 24 to 48 hours at a temperature just below the maximum anticipated operating temperature. This often eliminates crazing, which can be caused by entrapped solvent.

Adhesive Bonding

Adhesive bonding allows great freedom in design because it can be used effectively to bond a thermoplastic to a wide variety of materials: to itself, other plastics, metals, wood, glass, and ceramics, among others. However, adhesive bonding is not without some severe constraints. The primary concerns are slow processing rates, limited use in certain environments, and difficulties in applying the adhesive during the assembly operation.

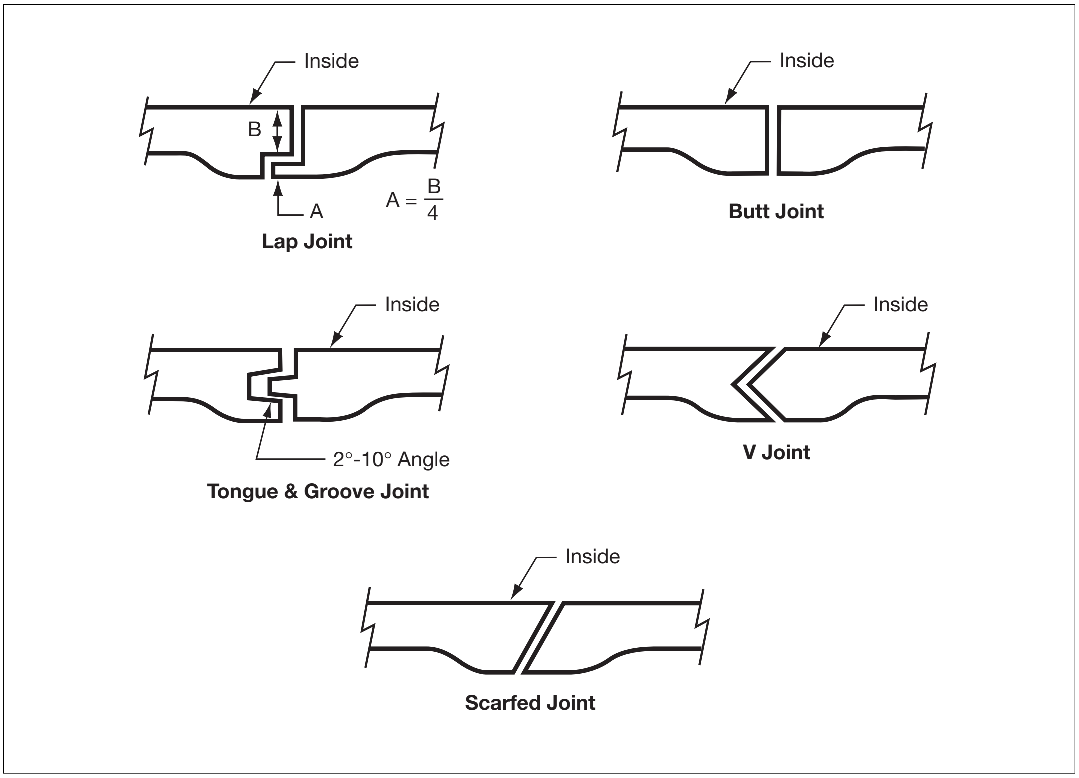
Certain adhesives can be hazardous. Follow proper national and industry guidelines. Before working with any such materials, request Material Safety Data Sheets for safe-handling recommendations from your supplier.

Adhesive bonding with thermoplastics will be satisfactory if:

- The proper adhesive (epoxy, polyurethane, or silicone) is selected for the application and environment.
- The joint design is adequate for the size and shape of the product. (Figure 108 shows the most commonly used joint designs.)
- The joint surface is not smooth. The surface should have molded-in texture or matte finish, or it should be treated in a secondary, roughing operation.
- The joint surface is clean and free of foreign materials, such as dirt, mold release spray, water, oil, etc.
- The adhesive is properly applied and cured.

Avoid using adhesives that have constituents incompatible with the thermoplastic being bonded. For more information on chemical resistance, see page 31.

Figure 108 – Joint Design for Adhesive Bonding



Welding

Welding Techniques

The technology for welding of thermoplastics comprises several techniques: spin, ultrasonic, induction, heated tool, and vibrational welding. These varied techniques give the designer and the manufacturer great flexibility in choosing a welding method that best suits the assembly requirements of a particular application.

All of the listed welding techniques involve subjecting the plastic components to the controlled generation of heat. First, heat is induced in the joint area of each component to be bonded. The plastic undergoes a softening or melting phase at the bond joint, and these "melt" layers on the joining surfaces will form the bond. When the mating surfaces are held together for a period of time, without heat and (sometimes) under pressure, the melted plastic resolidifies, and the weld bond is formed.

Spin Welding

Spin welding is a simple process that is quick and cost-effective. It requires minimal, basic equipment – usually nothing more than a modified drill press and a fixture. If the production requirements warrant, spin welding equipment can be completely automated.

Spin welding is frictional and is limited to cylindrical joint applications. Frictional heat is generated by holding one component still, while rotating the other at high speed and with controlled pressure. After the melt layer is formed, the rotation is halted and the plastic resolidifies.

Three variables affect the spin weld process: speed of rotation, duration of rotation, and pressure applied to the joint. Each of the variables depends on the material and the diameter of the joint. In most cases, the actual spin time should be approximately 0.5 seconds, with an overall weld time of 2 seconds. When assembly by spin welding is proposed, a series of prototype evaluations should be done to determine the rotation speed and time, pressure, and holding time that suits the application.

Figure 109 – Joint Design for Ultrasonic Welding, Butt Joint with Energy Director

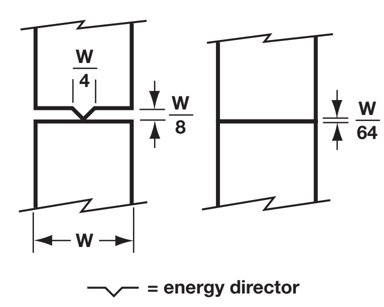


Figure 110 – Shear Joint Design for Ultrasonic Welding

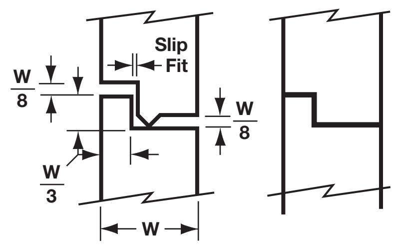
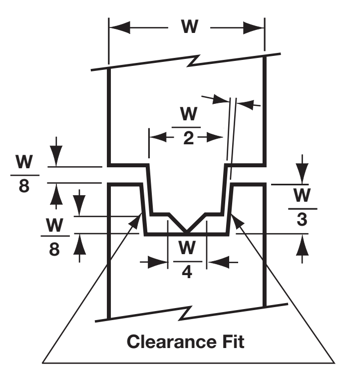


Figure 111 – Tongue and Groove Joint Design for Ultrasonic Welding



Ultrasonic Welding

Ultrasonic welding applies frictional heat to a plastic assembly by creating a high frequency (20 kHz to 40 kHz) mechanical vibration. The frictional heat that melts the polymer occurs at the molecular level, which results in a fusion or molecular bond. The bond is uniform and strong, and in some cases is stronger than the parent material. The welding cycle usually takes less than two seconds and can be completely automated.

An important factor in the ultrasonic welding process is the design of the joint. Joint design must cater to the material, the part geometry, and the requirements of the product. Several standard joint designs are available to meet those needs. Figures 109, 110, and 111 show three of the most common joint designs for ultrasonic welding.

Induction Welding

In induction welding of thermoplastics, heat is supplied to the plastic components by a metal member positioned in an electromagnetic field. The metal component can be an insert that conforms to the shape of the bond area, a foil-like tape, or small metal particles dispersed throughout a compatible resin mixture. An electromagnetic coil creates the magnetic field, and the current passing through the metal component – insert, foil, or dispersed particles – generates heat. The increasing temperature of the metal brings the plastic to its melt temperature.

Induction welding is relatively fast and clean, lends itself to use with most thermoplastics, and allows welding of parts having irregularly contoured surfaces. Of the three methods of induction welding, the metal insert is the most commonly used. The main drawback of this method is the need for a metal "preform" to act as the conductor. The preform can be relatively expensive and difficult to handle, especially in automated assembly processes.

If assembly by induction welding is part of your design plan, contact the manufacturer of the welding equipment for additional information.

Hot Tool Welding

Hot tool welding is basic, and is exactly what its name suggests. A tool is heated and brought into contact with the two plastic components to induce melting on the surfaces to be welded. Once melting occurs, the tool is removed, and the parts are held together under slight pressure. As the plastic cools, a weld bond is formed. There are several types of suitable tools: hot plate, soldering iron, strip heater, hot knife, or hot wire. In the case of the hot wire, the wire remains in the weld bond. Most hot tools are covered with a non-stick coating to prevent plastic from building up on the tool.

Vibrational Welding

Vibrational welding is a relatively new technology. There are two kinds of vibrational welding: linear and angular, with linear being the more common. In linear welding, one component is rapidly vibrated a certain (small) distance in relationship to the other. Friction between the two plastic parts develops the heat required to melt and form the joint. The advantage of vibrational welding is that parts with non-planar surfaces can be bonded by this method. The main disadvantages are the critical requirements of the welding equipment.

Polymer Glossary Index

Glossary

Common Designations for Thermoplastics

ABS	Acrylonitrile butadiene styrene
Apec	Aromatic polyester carbonate
ASA	Acrylonitrile styrene acrylate
GPPS	General-purpose polystyrene
HDPE	High density polyethylene
HIPS	High impact polystyrene
LCP	Liquid crystal polymer
LDPE	Low density polyethylene
PA	Polyamide
PB	Polybutylene
PBT	Polybutylene terephthalate
PC	Polycarbonate
PE	Polyethylene
PEEK	Polyether ether ketone
PEI	Polyetherimide
PES	Polyethersulfone
PET	Polyethylene terephthalate
PMMA	Polymethyl methacrylate
POM	Polyacetal
PP	Polypropylene
PPE	Polyphenylene ether
PPO	Polyphenylene oxide
PPS	Polyphenylene sulphide
PS	Polystyrene
PSU	Polysulfone
PVC	Polyvinyl chloride
SAN	Styrene acrylonitrile
SMA	Styrene maleic anhydride
TPU	Thermoplastic polyurethane

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PULSE* Engineering Resins
SABRE* Engineering Resins
STYRON* Polystyrene Resins
TYRIL* SAN Resins

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