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A guide to plastic gearing



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INTRODUCTION

Plastic gears have positioned themselves as serious alternatives to traditional metal gears in a wide variety of applications. The use of plastic gears has expanded from low power, precision motion transmission into more demanding power transmission applications. As designers push the limits of acceptable plastic gear applications, more is learned about the behavior of plastics in gearing and how to take advantage of their unique characteristics.

Plastic gears provide a number of advantages over metal gears. They have less weight, lower inertia and run much quieter than their metal counterparts. Plastic gears often require no lubrication or can be compounded with internal lubricants such as PTFE or silicone. Plastic gears usually have a lower unit cost than metal gears, and can be designed to incorporate other features needed in the assembly (part consolidation). These gears are also resistant to many corrosive environments.

The first use of thermoplastic gears undoubtedly consisted of neat nylon and acetal gears carrying low loads at low speeds. As the advantages of using thermoplastic gears became clearer and new, higher performance materials became available, designers began using plastic gears in more demanding applications. The use of reinforcements and internal lubricants compounded into these materials has expanded their use even more.

The use of thermoplastic materials for gears is hampered by a lack of established load carrying and wear performance data, at least when compared to the reams of easily accessible gear/material performance data that has been put together for metals.

The data for metals has been collected and confirmed through numerous successful applications and is well understood by most gear designers.

Thermoplastics' late arrival as a gear material has not provided enough time for the compilation of extensive load rating data, and the unique mechanical and thermal behavior of thermoplastics has frustrated those who would attempt to interpolate these values from more readily available information.

Nonetheless, there are certain guidelines available for estimating the technical feasibility of using thermoplastic materials in gears. Most of these techniques have evolved from equations originally worked out for metals, and therefore do not take into account some of the unique behavior found in thermoplastic materials.

This brochure will attempt to reveal some of the important points that must be considered when using these equations and techniques to evaluate thermoplastic gears. The focus will be on spur gears; however the basic points covered can be extended to other types of gears.

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GEAR TYPES AND ARRANGEMENTS

There are many different types of gears, and they can be most easily categorized by the way the gear axis intersect. If the gears must operate on parallel axes, then spur or helical gears are required. If the axes are intersecting and at right angles, then bevel and worm gears are usually used. If the axes are both non-intersecting and nonparallel, then crossed-axis helical, worm gears, hypoid and spiroid gears are used. The most common plastic gears are spur, helical and worm gears, but the other types could be used if required.

A single gear can do no work, so gears are used in pairs. When the teeth of two gears are meshed, the rotation of one gear will cause the other gear to rotate also. If the two gears have different diameters, the smaller one (called the pinion) will turn faster and with less rotational force than the larger one (called the gear).

Spur gears are cylindrical in shape, with the flank of the gear tooth parallel to the gear axis. If the teeth of the gear point away from the axis, the gear is an external spur gear (figure 1). If the teeth point toward the axis, the gear is an internal spur gear (figure 2).

Spur gears are relatively simple in design and easy to manufacture. Spur gears impose only radial loads on their bearings and will operate on a variety of center distances, making them relatively simple to mount. Most designers use a 20° pressure angle, but 22 1/2° and 25° are also common. Pressure angles above 20° give higher load capacity but do not run as smoothly or quietly.

A helical gear is similar to a spur gear, but the tooth flank is now at an angle to the axis of the gear (figure 3). In fact, a helical gear with a helix angle of zero is a spur gear. Helical gears are used when both high speeds and high loads are involved. Single helical gears impose both radial and thrust loads, and are therefore not as simple to mount, but they do tend to run quieter and smoother than spur gears. Helical gears with opposite hands are often mounted on the same shaft to cancel out the thrust load. These are referred to as double helical gears (figure 4).

Bevel gears are conical in shape, and the teeth are tapered in both the tooth thickness and in the tooth height. At one end the tooth is large, and at the other it is small. While the tooth dimensions are listed based on the large end of the tooth, strength calculations are based on the central section of the gear tooth.

The simplest type of bevel gear is the straight bevel gear (figure 5). These gears are commonly used on intersecting 90° shafts, but they can operate at almost any angle. These gears impose both thrust and radial loads, and must be precisely mounted to operate correctly. While plastic bevel gears are not very common, designers are beginning to investigate their use. Other types of bevel gears are the spiral and zerol bevel gears.

Face gears are a special type of gear with teeth cut into the face of the gear (figure 6). In face gears the teeth point in the same direction as the gear's axis. The face gear will mate with a spur or helical gear. Like bevel gears, the axis of the two gears must intersect and the shaft angle is usually 90°.

There are three types of gears that are commonly referred to as worm gears. Worm gears can be mounted on non-intersecting, nonparallel axes; however, the most common arrangement is non-intersecting, 90° axis. Worm gears are characterized by one member having a screw thread. This member is referred to as the worm (figure 7). The gear it mates with is referred to as the worm gear.

It is very common in plastic gearing to mate a metal (or occasionally plastic) worm with a plastic helical gear. This arrangement is actually called non-enveloping worm gears or crossed-axis helical gears. Crossed-axis helical gears are mounted on axes which do not intersect and that are at an angle (usually 90°) to each other. Crossed-axis helical gears generate both radial and thrust loads on their bearings.

Crossed-axis helical gear sets are able to stand small changes in center distance and shaft angle without impairment in the accuracy of the gear. This makes it one of the easiest gears to mount. Unfortunately, crossed-axis helical gears have only point contact and, therefore, do not carry very high loads.

GEAR TYPES AND ARRANGEMENTS

However, if the gears can wear in for some time without failure, the point contact becomes line contact, which is more similar to a single enveloping worm gear, and the load carrying capacity increases. This is one of the reasons metal worms are mated with plastic helical gears. The helical gear wears first, and becomes, for all intents and purposes, a worm gear. The other reason metal worms are used with plastic worm gears or helical gears is that they help eliminate the large amounts of heat that can be generated with worm gear sets. It is not uncommon to see plastic worm gears fail due to heat related mechanisms.

True worm gear sets are described as single-enveloping or double-enveloping worm gears.

In single enveloping worm gearing the worm gear has a throated profile which wraps around the worm like a nut wraps around a thread (figure 8). This gives greater contact and increases the load carrying capability by a factor of 2-3 over a similar helical gear. In double-enveloping worm gears both the worm (figure 9) and the worm gear are throated and wrap around each other. It is very difficult to mold a throated worm or worm gear, and for that reason a worm and helical gear (crossed-helicals) combination is used most often.

FIGURE 1 SPUR GEARS

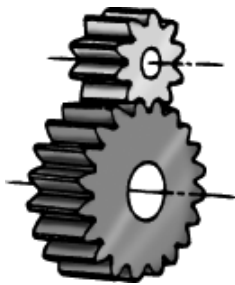


FIGURE 2 INTERNAL SPUR GEARS

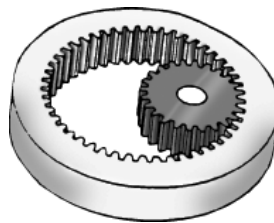


FIGURE 3 HELICAL GEARS

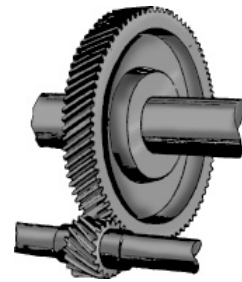


FIGURE 4 DOUBLE HELICAL (HERRINGBONE) GEARS

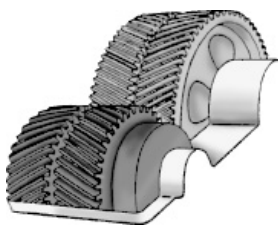


FIGURE 5 STRAIGHT BEVEL GEARS



FIGURE 6 FACE GEARS

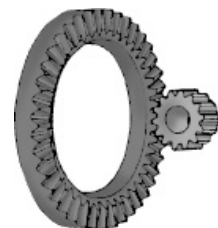


FIGURE 7 WORM GEAR



FIGURE 8 SINGLE ENVELOPING WORM GEAR



FIGURE 9 DOUBLE ENVELOPING WORM GEAR



GEAR ACTION

FIGURE 10A BEGINNING OF CONTACT

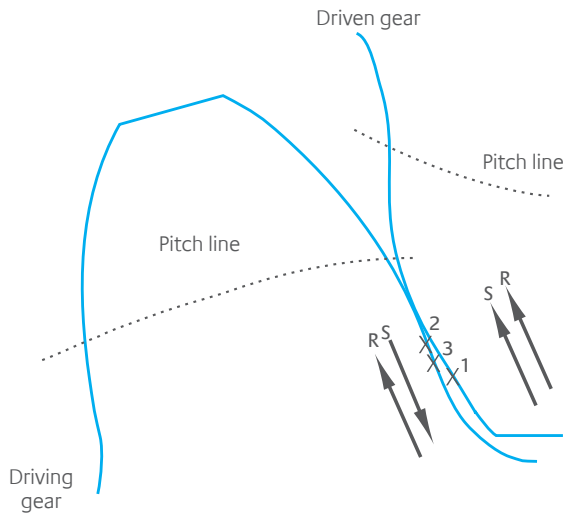
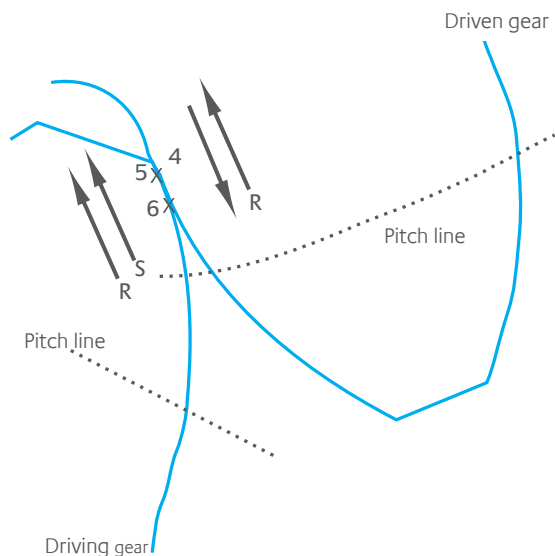


FIGURE 10B END OF CONTACT



Before we can begin to analyze the stresses in plastic gears, it is important to understand gear action. Each tooth is, in effect, a cantilever beam supported on one end. The contact tries to bend the beam and to shear the beam from the bulk of the material. Therefore, a gear material needs to have high flexural strength and stiffness.

The next effect is primarily a surface effect. Stress is generated on the surface of the tooth by frictional forces and by point or line contact (Hertzian contact stress). During gear movement the gear teeth roll on each other and slide past each other at the same time.

As the teeth come into mesh, there is an initial contact loading. The rolling action of the gears pushes the contact stress (which is a special compressive stress) just ahead of the point of contact.

At the same time, sliding is occurring because the contact length on the parts of the gear tooth in mesh are not the same. This causes frictional forces that develop a region of tensile stress just behind the point of contact. The arrows in figure 10 marked R show the direction of rolling and the arrows marked S show the direction of sliding.

In the areas where these two motions are in opposite directions the resulting forces cause the most problems.

In figure 10a, the gears have just come into contact. At point 1 on the driving gear the material is under compression from the rolling action toward the pitch point, and under tension due to frictional resistance to the sliding motion away from the pitch point. This combination of forces can cause surface cracking, surface fatigue and heat build up. All of these factors can lead to considerable wear.

At the pitch point the sliding force changes direction and a null point for sliding is created (pure rolling). One might assume this section of the gear shows the least surface failure; however, the pitch point is one of the first areas where serious failures occur. Although the pitch point does not see compound stresses, it does see high unit loading.

During initial gear contact or at the end of contact, the previous tooth pair or the next tooth pair should bear some of the load, and the unit loading is reduced. The highest point loading occurs when the gears are contacting at or slightly above the pitch line. At that point, one tooth pair will usually be carrying all or most of the load. This can lead to fatigue failure, severe heat build-up and surface deterioration.

GEAR DESIGN

STRESS ANALYSIS

The most important part of a gear is the gear teeth. Without the teeth, the gear is simply a wheel and is of little use in transmitting motion or power. The basic measurement of a gear's ability to carry a given load is to estimate the gear's tooth strength. Although prototyping of a gear is always recommended, it can be expensive and time consuming, so some method of determining a gear's feasibility is required.

BENDING STRESS

The bending stress on a gear tooth of a standard tooth form loaded at the pitch line can be calculated using the Lewis Equation

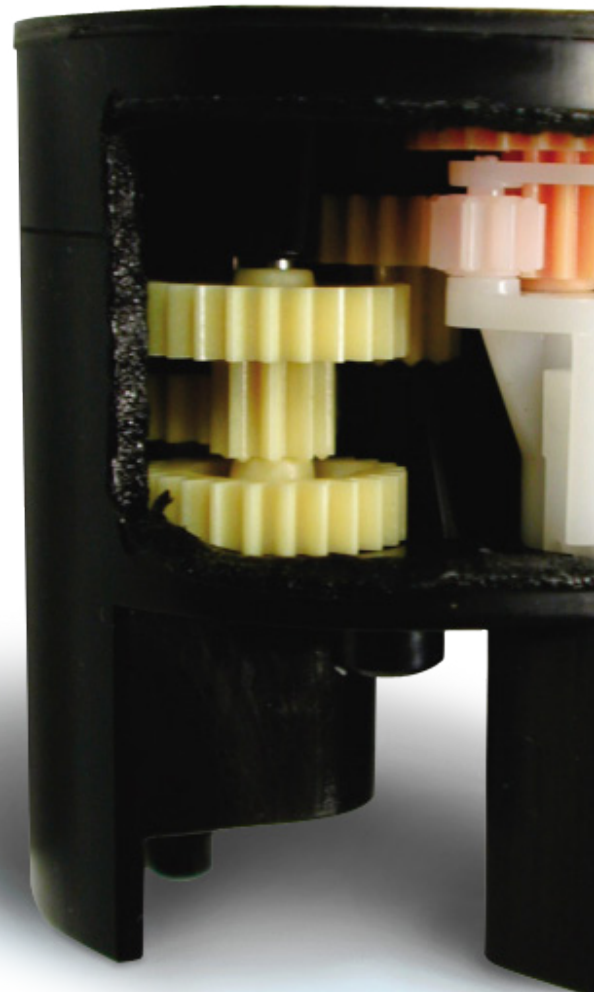
$$S_b = \frac{F P_d}{f Y}$$

where S_b = bending stress
 F = tangential tooth loading at the pitch line
 P_d = diametral pitch
 f = face width
 Y = Lewis form factor for plastic gears, loaded at the pitch point

Tests have shown that the most severe tooth loading occurs when the gear tooth is loaded tangentially at the pitch line and the number of pairs of teeth in contact approaches one. Another useful approach, if the horsepower required by the system is known, is to use the equation

$$S_b = \frac{HP \ 126050 P_d}{f Y D w}$$

where HP = horsepower
 D = pitch diameter
 w = speed, rpm





Typical service factors, which describe the quality of the input torque and the duty cycle of the gear are:

TABLE 1 SERVICE FACTORS

Type of load	24 hrs/day	8-10 3 hrs/day	Intermittent 1/2 hrs/day	Occasional
Steady	1.25	1.00	0.80	0.50
Light shock	1.50	1.25	1.00	0.80
Med. shock	1.75	1.50	1.25	1.00
Heavy shock	2.00	1.75	1.50	1.25

Another variation of the Lewis equation incorporates the pitch line velocity and a service factor

$$S_b = \frac{HP \ 55(600+V) \ P_d \ C_s}{f_y V}$$

where y = Lewis form factor at the tooth tip
 V = pitch line velocity (fpm)
 C_s = service factor

With any stress equation, allowable stress, S_{all} , can be input for S_b in order to solve for other variables. The safe or allowable stress is not a typical data sheet stress level, but an allowable stress determined from actual testing of a material as a gear with standard tooth form.

An allowable stress already has a material safety factor incorporated into the value. For any given material the allowable stress level is very dependent on a large number of factors.

These include:

- Lifetime cycles
- Operating environment
- Pitch line velocity
- Counterface
- Lubrication

TOTAL PLASTIC GEARS DESIGN

Since allowable stress is equal to a strength value divided by a material safety factor ($S_{all} = S/n$), this is a good point to talk about factors of safety for gears. Safety refers to the ability of a part to perform its proper function for its service life without failure. Function, service life and part failure must be defined for an application before a factor of safety can be chosen.

Factors-of-safety may be defined in many ways; however, they basically relate what is permissible or allowable to what will cause failure. A factor-of-safety can be applied in three basic ways. The entire factor can be applied to a material property such as strength; or the entire factor can be applied to the loading; or separate factors can be used for each load and a material property.

The latter case is often the most useful because each load can be investigated and then a factor-of-safety applied to determine the absolute maximum load. Each maximum load is then used in the stress analysis such that the geometry and boundary conditions produce an allowable stress. The allowable stress limit is determined by applying a strength factor-of-safety to the material strength at end-use conditions.

The load factor-of-safety can be determined in the traditional manner. The strength factors-of-safety for plastic materials, however, are often difficult to determine. This is because the strength of a plastic is not a constant, but a statistical distribution of strength under end-use conditions.

Consequently, design engineers need to understand the end use conditions, for example, temperature, strain rate and load duration. Fabrication knowledge is needed to understand weldline situations, anisotropic effects, residual stresses and process variants.

Material knowledge is most important because the better a material behavior is understood under end-use conditions, the more accurately a factor-of-safety can be established, resulting in an optimum part geometry.

The poorer the definition and the greater the number of unknowns, the larger the factor-of-safety required. A minimum factor-of-safety of two is recommended, even when an application has been carefully analyzed.

If pre-calculated allowable stress data is not available, and for plastics, it usually isn't, then the gear designer must be extremely careful to consider all of the factors outlined earlier, so that a proper factor of safety can be determined and S_{all} can be calculated.

Whether similar experience exists or not, it is essential that a prototype mold be built and that the gear be tested at the expected application conditions.

CONTACT STRESS

The equations we have looked at so far have been examining the forces trying to bend the gear teeth and shear them from the bulk of the material. These forces lead to failures by tooth breakage due to static loading or fatigue action. The other forces we saw in our examination of gear action generated surface stresses by contact of the gear teeth and their relative motion to each other. These stresses lead to failure in the surface of the gear teeth, or wear. To assure a satisfactory life, the gears must be designed so that the dynamic surface stresses are within the surface endurance limit of the material.

The following equation was derived from the Hertz theory of contact stress between two cylinders, and modified to employ notation used in gearing

$$S_H = \sqrt{\frac{W_t}{f D_p} \frac{1}{\pi \left(\frac{1-\mu_p^2}{E_p} + \frac{1-\mu_g^2}{E_g} \right)} \frac{1}{\frac{\cos \phi \sin \phi}{2} \frac{m_g}{m_g + 1}}}$$

where

- S_H = surface contact stress (Hertzian stress)
- W_t = transmitted load
- D_p = pitch diameter, pinion
- μ = Poisson's ratio
- E = modulus of elasticity
- ϕ = pressure angle
- m = speed ratio, N_g/N_p
- N = number of teeth

The subscripts p and g refer to the pinion and gear, respectively. The surface contact stress is calculated for a gear and then compared to the surface endurance limit of the material. However, for plastics, this data is seldom available. Once again, the best way to determine this type of data is through actual testing sets of gears running under service conditions. This calculation, however, can give the designer some idea of how stressed the surface of the gear will be relative to the pure compressive strength of the material, which is readily available.

GEAR TOOTH DESIGN

Hobs used to cut teeth in metal gears are available "off-the-shelf", and for economic reasons, designers of commercially cut gears seldom use any other tooth forms. Injection molded gears are not constrained to use these standard hobs since special tooling must be used when cutting the mold to compensate for shrinkage.

If a hob with a standard pressure angle is used to cut a mold, a serious tooth profile error will result due to the mold shrinkage of the material. The gear designer is therefore free to use a variety of techniques to maximize the performance of his gear. While a variety of plastic gear tooth profiles are available, they all use basic plastic design techniques to optimize the design of the gear tooth.

The gear modifications most commonly used on plastic gears were developed from modifications used in heavily loaded metal gears in critical applications. The basic modifications most commonly made are full fillet radius modification, tip relief modification, elimination of undercut condition and balanced circular tooth thickness.

TOTAL PLASTIC GEARS DESIGN

FULL FILLET TOOTH RADIUS MODIFICATION

Sharp corners in plastic molded parts are undesirable since they act as stress risers. Using a full fillet radius between two teeth in a gear eliminates these sharp corners and can reduce stress by up to 20% or more. Full fillet radii should be used in all plastic gears.

TIP RELIEF MODIFICATION

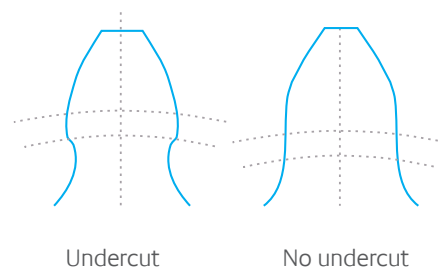
When a tooth deflects under load it can get in the way of the next oncoming tooth. This happens in heavily loaded metal gears and to varying extent in most plastic gears. This type of interference can cause noise, excessive wear and a loss of smooth uniform motion.

To compensate for this deflection, the tip of the tooth is gradually thinned from half way up the addendum (the top half of the gear tooth). This modification is most useful in gears that are highly loaded (for that particular material) and is not always required in plastic gears.

ELIMINATION OF UNDERCUT

The teeth of gears having a small number of teeth will often be undercut at the root of the gear. This will weaken the gear tremendously and should be avoided in plastic gears (figure 13).

FIGURE 13 UNDERCUTTING

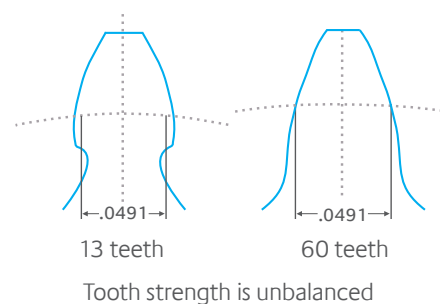


- weakens tooth
- inhibits continuity of motion
- increases wear

BALANCED CIRCULAR TOOTH THICKNESS

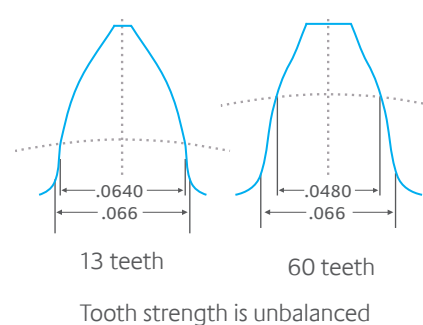
If two gear teeth in mesh are designed as standard, then the gear with the smaller number of teeth (pinion) will have teeth that are thinner at the root than the teeth of the gear (figure 14).

FIGURE 14 STANDARD 32 PITCH DIAMETER PINION AND GEAR



The pinion will not be able to transmit as much power as the gear could carry and will be the weak link in the design. In order to optimize the load carrying capability of the gear set, the circular tooth thickness of the pinion should be increased and the circular tooth thickness of the gear should be decreased (figure 15).

FIGURE 15



Two tooth forms for plastic gears that incorporate these modifications are the AGMA PT tooth forms (figure 16) and the ISO R53 modified (figure 17). These tooth forms are essentially the same, only differing in nomenclature. The ISO form uses the metric module, m while the AGMA PT tooth forms use diametral pitch, P_d . While these forms are useful, they are by no means the only forms available. Other designs may be used to optimize a gear set for its particular application

FIGURE 16 AGMA PT TOOTH FORM

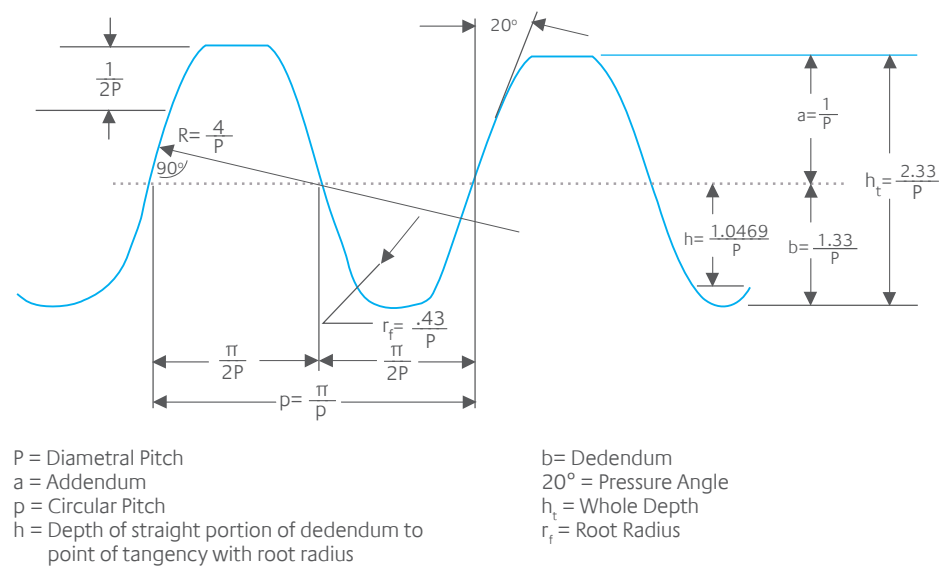
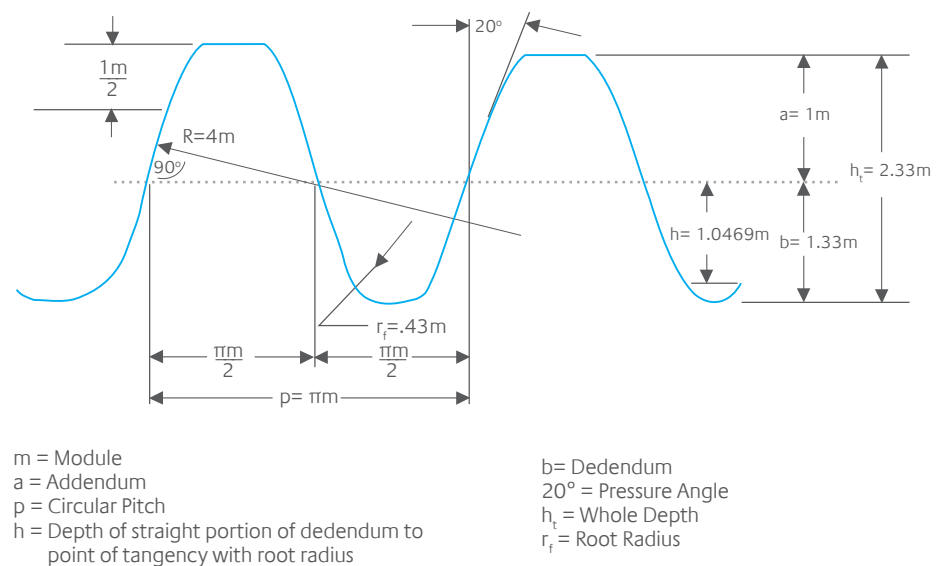


FIGURE 17 MODIFIED ISO BASIC RACK



TOTAL PLASTIC GEARS DESIGN

When using these types of modifications, some adjustment must be made to the equations for tooth bending stress and allowable stress. If tooth thickness has been modified, then the Lewis form factors for standard tooth thickness should be multiplied by the ratio of the thickness of the modified tooth to the thickness of the standard tooth.

OVERALL PART DESIGN

The tooth modifications outlined above apply to the design of the gear tooth itself, but are adaptations of basic plastic part design guidelines. When designing any plastic part, these rules must be taken into consideration. Since your plastic gear teeth will be attached to some plastic part, you must also apply the rules to your overall design.

NOMINAL WALL

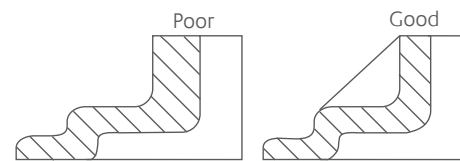
One of the most important features in a good plastic design is the nominal wall. The nominal wall is the basic feature of a part which gives the part its shape. The thickness of the nominal wall will influence the strength, cost, weight and precision of the part. Typical injection molding techniques work best when the nominal wall of the part is in the range of 0.030 to 0.200 inches thick.

Although there is no such thing as an average wall thickness for an injection molded plastic part, 0.125 inches is a very common dimension. It is also very important that changes in the nominal wall should be held to less than 25% for low shrink materials and 15% for high shrink materials. If a more radical change in wall thickness is needed, it should be made in several steps (figure 18).

The biggest problem associated with large changes in wall thickness is that the thicker sections will not cool as quickly as the thin sections and will therefore, shrink more. This can result in part warpage and out of tolerance parts. One way to keep a uniform wall thickness is to core the part equally from both sides (figure 19).

FIGURE 18 WALL TRANSITION DESIGN

Variable Wall Section Design



Wall Transition Design

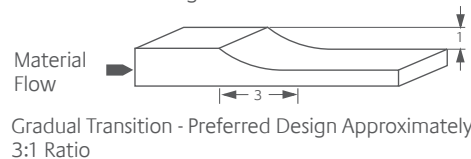
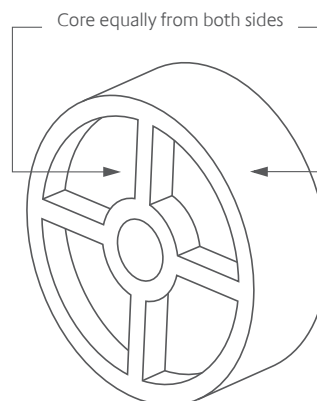


FIGURE 19 CORING



Goal – uniform wall with even pressure distribution and even cooling

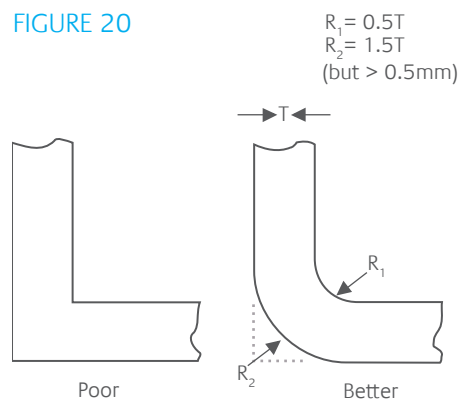
RADIUS

When two walls meet in a plastic part and form a corner, there is a potential for stress concentrations and a reduction in flow. By radiusing the inside corner, stresses are spread out over a larger area. By radiusing an outside corner you improve the material flow path and maintain a nominal wall thickness.

The general recommendation is for inside corners to be radiused a minimum of 25% of the nominal wall with a maximum being 75%. Larger radii reduce stress concentrations, but the design trade off is the resulting thick section of material.

When an inside corner has a corresponding outside corner the outside radius should be sized to maintain a uniform wall. If the inside radius is 50% of the nominal wall thickness, then the outside radius should be 150% (figure 20).

FIGURE 20



RIBS

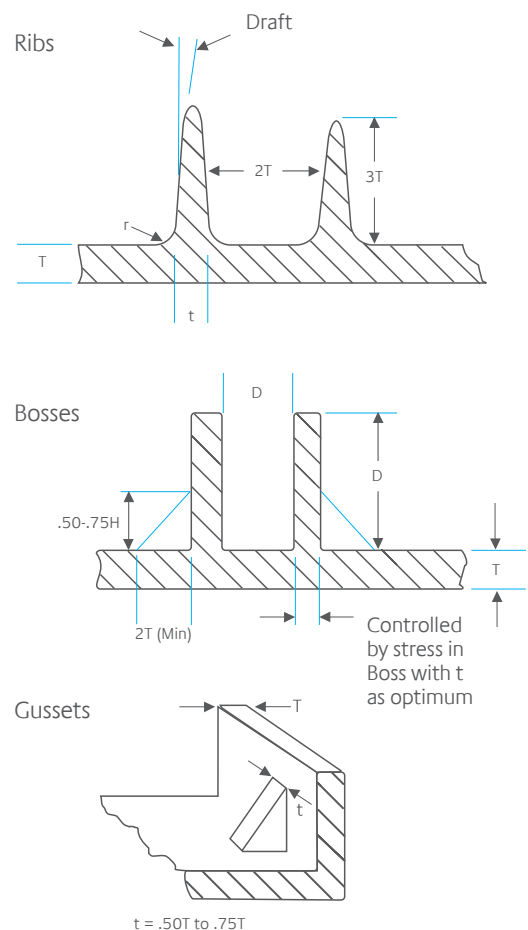
All but the simplest plastic parts have projections of some type off the nominal wall. These projections can come in the form of reinforcing ribs, gussets and bosses. The most common projection is the reinforcing rib.

Reinforcing ribs are generally added to a part to increase its stiffness or to control the flow of the melt across the cavity. In general, the height of the rib should be no more than two and a half to three times the thickness of the nominal wall. Although a taller rib will increase the stiffness of the part, it will be difficult to mold properly. Tall ribs are difficult to fill, vent and eject. For this reason it is usually preferable to add two shorter ribs in the place of one tall one.

The thickness of a rib should be approximately half that of the nominal wall for high shrink materials and 75% of the wall in low shrink materials. This will help control the shrinkage at the junction of the rib and wall. The junction should be radiused to a minimum of 25% of the nominal wall thickness.

Larger radii will increase the thickness of the junction and create sink marks in the surface opposite the reinforcing rib. When using multiple ribbing, the ribs should be no closer to each other than two times the thickness of the nominal wall. Ribs placed closer together will be very difficult to cool and may result in a large amount of molded-in stress (figure 21).

FIGURE 21 RIBS/BOSSES/GUSSETS/ PROJECTIONS

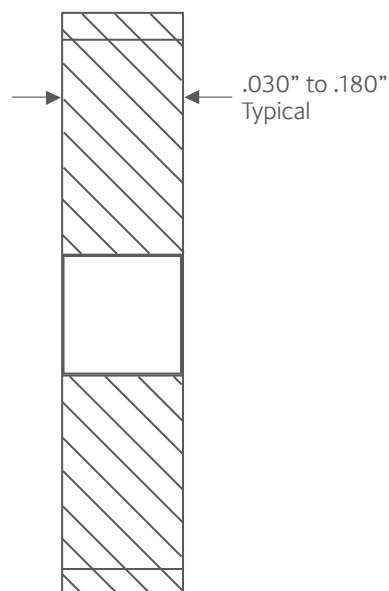


TOTAL PLASTIC GEARS DESIGN

GEAR DESIGN

When designing a gear to be molded out of thermoplastic material, it is important to remember the basic plastic design guidelines outlined above. The simplest gear is the flat gear with no rim or hub (figure 22) gated in the center. This gear will have minimal differential shrinkage since it has a single nominal wall with no changes in thickness. These gears should not be more than 0.250" thick, and web and hub designs may become more practical if the gear is over 0.180" thick.

FIGURE 22 HUBLESS AND RIMLESS GEARS

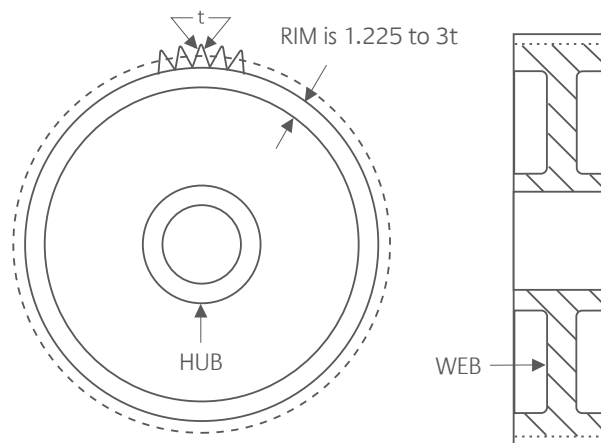


When designing a plastic gear that has a hub and a rim, careful consideration must be made to the thickness of the various parts. Tooth thickness and height has already been determined by the requirements of tooth strength. The difficulty lies in deciding which part of the gear is the nominal wall and what is the relationship between that feature and the other parts.

Each part of the gear should be designed to perform the desired function without forgetting the basic plastic design guidelines. As with any design guidelines, compromise will undoubtedly have to be made.

If the gear teeth are treated as a projection off of a wall (the rim), then the thickness of the rim should be 1.25 to 3 times the thickness of the gear tooth (figure 23). The web and hub should be at least as thick as the rim. Since most gears are gated on the web, the web could be made thicker than the hub and rim for better filling.

FIGURE 23



RIM, HUB and WEB should be uniform thickness

Once again, the web should not be more than 1.25 to 3 times thicker than rim and hub. If the hub must be thick (i.e., press fits) then the gear should be gated on the hub or diaphragm gated in the center. In all cases, center diaphragm gates will provide the most even fill and are recommended. Remember to radius all inside corners 50 to 75% of the wall thickness.

Holes in the web should be avoided, as they only serve to weaken the gear by adding knit lines and inducing variable areas of high and low shrinkage in the rim, which leads to difficulty controlling tolerances (figure 24). Ribs can also affect tolerances for the same reason and should be avoided unless absolutely necessary. If ribs must be added, they should be added to both sides of the gear, and they should not be directly opposite each other (figure 25).

FIGURE 24

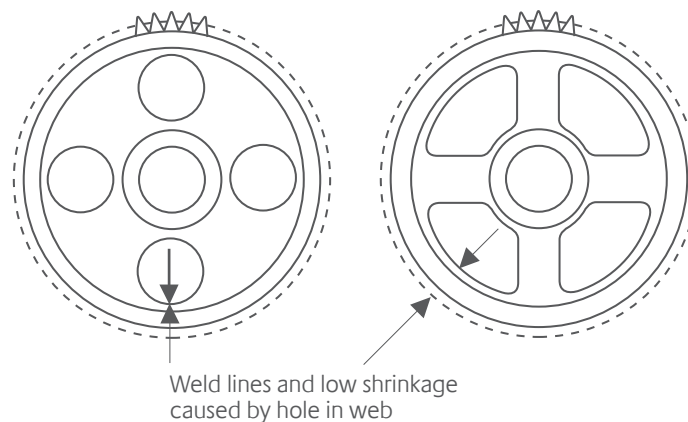
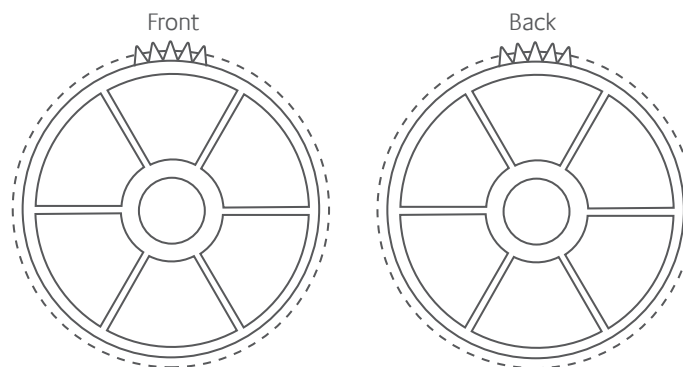


FIGURE 25



If needed, ribs should be used on both sides and offset front to back to prevent thick sections

ASSEMBLIES

The four "modifications" to standard gear designs and basic thermoplastic part design guidelines outlined will provide stronger injection molded plastic gears. However, the gears must remain in the mesh at the proper points. When two gears are brought into close mesh the distance between their centers is half the sum of their standard pitch diameters and is referred to as the standard center distance.

TOTAL PLASTIC GEAR DESIGN

Spur and helical gears will operate at a wide range of center distances, and it is rare that the best operating center distance is the standard center distance. Also, the gear designer must compensate for any environmental conditions that might affect the center distance. If the center distance between the gears is too small, thermal and environmental effects may cause the center distance to close in and bind the gears.

Factors which can affect the operating center distance of the gears include thermal expansion of the gears, shafts and housing, dimensional changes due to moisture absorption, runout in the bearings used to locate the gears and the overall accuracy of the gears themselves. In order to prevent the gears from binding because of these changes, it may be necessary to increase the center distance. This increase in center distance can be calculated with the equation.

$$\Delta c = \frac{T_{ct1} + T_{ct2}}{2} + C[(T - 70) \frac{a_1 N_1 + a_2 N_2}{N_1 + N_2} - a_H] + \frac{M_1 N_1 + M_2 N_2}{N_1 + N_2} - M_H + \frac{TIR_1 + TIR_2}{2}$$

where

- Δc = required increase in center distance
- T_{ct} = maximum total composite tolerance in gear
- C = close mesh center distance
- T = maximum operating temperature gears will see in °F
- a = coefficient of linear thermal expansion of the material (in./in./°F)
- M = expansion due to moisture absorption of hub material (in./in.)
- TIR = maximum allowable runout of bearing

The subscripts 1, 2, and H refer to gear 1, gear 2 and the housing. The coefficient of linear thermal expansion can usually be found on material data sheets provided by the material supplier. Expansion due to moisture absorption is not readily available and is not the same as the rate of water absorption normally reported on data sheets. If the gears in question will not immediately be exposed to high humidity, then the expansion of most plastics is minimal and may be offset by the slight shrinkage of the plastic that occurs as molded in stresses relax gradually over time .

For hygroscopic materials such as nylons, the expansion may be more important. Some common allowances for moisture are:

TABLE 2		If the material you are working with is not on this table, you can use the polycarbonate numbers for low moisture materials and the Nylon 6/6 numbers for hygroscopic materials.
Material	M (%)	
Acetal	0.05	
Nylon 6/6	0.25	
Nylon 6/6 + 30% glass fiber	0.15	
Polycarbonate	0.05	

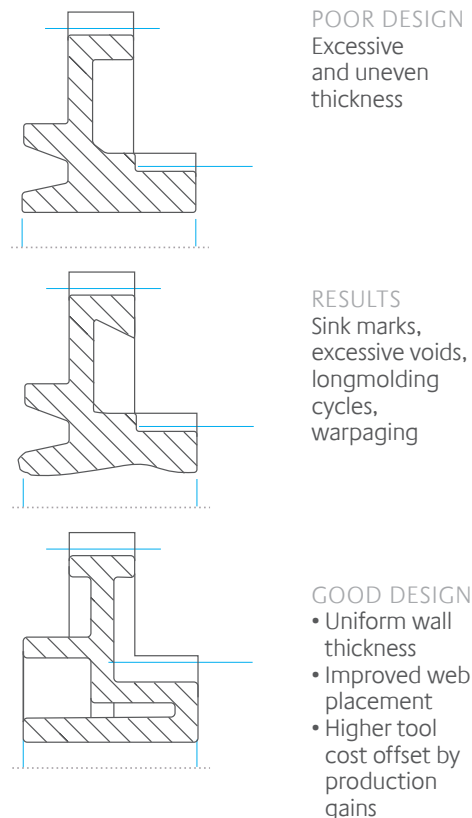
TESTING

PART CONSOLIDATION

One of the most useful features of thermoplastic injection molded gears is the ability to consolidate a number of different parts into one multi-functional design (part consolidation). The simplest form of this is molding the gear shaft and gear as a single unit.

It is also very common to mold two or more spur or helical gears as a one piece unit. This is referred to as a compound gear. When molding compound gears it is important to remember the rules concerning nominal wall thickness and radii. Simply stacking one gear on another will lead to thick sections, unequal cooling and low tolerance. Figure 26 shows the difference between a good design and one that is too thick.

FIGURE 26



TESTING

Plastic and metal gears fail by the same type of mechanism if the gear materials design limits are exceeded. All new applications should be prototype tested near or at operational conditions. Testing gears machined from rod stock may seem the simplest way to test a gear, but the results may differ from a molded gear due to differences in surface finish, molded in stresses, accuracy and many other issues. The only way to really know how a gear will perform is to test a prototype molded gear.

Accelerated tests at speeds higher than required of a given application are often of no value. Increasing temperature above normal working temperature may cause rapid failure, whereas under normal operating conditions the gear may work well. Test conditions should always be chosen to come as close to actual conditions as possible.

For instance, if a gear operates at high loads but only intermittently, it should not be tested in a continuous run, since under operating conditions the gear will have an opportunity to cool down between cycles. If a gear is slow running and infrequently operated, it can be tested continuously if the rise in gear tooth temperature will be minimal.

If a gear will reach its maximum use temperature very quickly during use, then continuous testing may also be allowed. Static tooth load tests may be useful if the gear does not run close to its endurance limit. Static test loads should be 8 to 10 times expected working loads if the gear is to have a long service life.

GEAR FAILURE MECHANISMS

GEAR FAILURE MECHANISMS

ADHESIVE OR “NORMAL” WEAR

This type of wear results from the intermittent welding and tearing of small areas of the opposing wear surfaces. If the welding is at a microscopic level then the result will be a normal uniform wear rate. External lubrication of the gears works to keep the surfaces separated and inhibit wear.

PTFE compounded into the thermoplastic acts as a lubricant by forming a thin film of PTFE on both the gear and its mate. This PTFE transfer film has low friction and wear rates. In plastic on plastic gear pairs, at least one of the gears should contain PTFE. Using an external lubricant with PTFE lubricated gears may not give as good a result, since the grease may act as a release agent and prevent the formation of the transfer film.

However, since there is a period of break-in on PTFE lubricated gears that will have higher wear rate while the transfer layer forms, a light external lubrication may slow the wear of the gear on start-up if it does not inhibit the formation of the layer.

In unlubricated plastic gears failure at the pitch line usually occurs due to nonuniform or excessive wear. This kind of wear increases frictional heat (softens material) and increases the pitch line loading on a tooth with a reduced cross-section (figure 27).

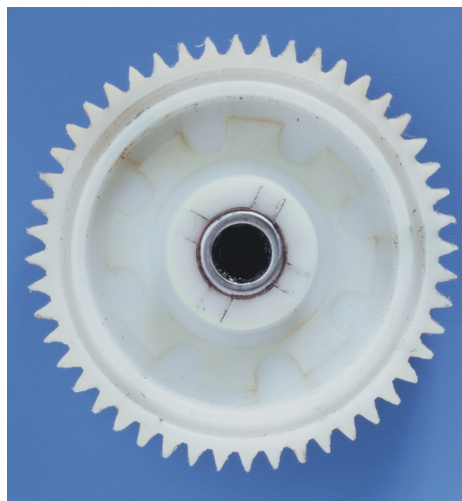
This usually bends the tooth over at the pitch line, resulting in tooth smearing or complete breakage. This may look like a fatigue failure, but it is really a wear failure. If a gear is well lubricated, then frictional forces are reduced, which will lower the heat build-up and wear.

In general, dissimilar materials wear better than similar materials. However, this is not always the case and some sort of wear testing should be performed followed by prototype testing of the gear pair in question if the wear test results look acceptable. If a plastic gear is to be run against a metal gear, the metal gear face should have a finish of 16 $\mu\text{in.}$ for good wear resistance.

ABRASIVE

Abrasive wear takes place whenever a hard particle is present between the contact surfaces. This material may be wear debris from one of the gears or dirt from the environment. This type of wear may also be present if one of the gears (usually metal) has a rougher surface than the other. The particles first penetrate the material and then “plow” off pieces of material from the surface. Abrasive wear conditions should be avoided

FIGURE 27 TOOTH THINNING DUE TO HIGH WEAR



PITTING

Pitting is defined as a surface fatigue failure that occurs when the endurance limit of the material is exceeded. Gears under load are subject to surface and subsurface stresses. If the loads are high enough and the stress cycles repeated often enough, areas will fatigue and fall from the surface. The area of the pitch line receives the highest stress and is most prone to pitting. Pitting is fatigue related and is generally independent of lubrication. Pitting is rare in plastics but can occur, especially if the system is well lubricated (low wear).

PLASTIC FLOW

Plastic flow is caused by high contact stresses and the rolling and sliding action of the mesh. It is a surface deformation resulting from the yielding of the surface and subsurface material. Since plastics are insulators and have low melting temperatures (compared to metals) they tend to melt and flow in situations where metal gears would score. In plastic gears the initial plastic flow is in the radial direction. It may not be detrimental as it may relieve itself. However, in more severe cases the flow will be in the axial direction and tooth breakage will soon follow.

Plastic flow indicates that the operating conditions are too severe and that failure is not far away. Lubrication (internal and external) can help prevent this condition by lowering the amount of heat generated by friction (figure 28).

FRACTURE

Fracture is failure by tooth breakage of a whole tooth or at least a good part of it. This can be the result of overloading (stall, impact) or from cycle stressing (fatigue) of the tooth beyond the endurance limit of the material. These types of fractures generally occur at the root fillet and propagate along the base of the tooth. Fractures in unlubricated systems are usually due to overload. Fractures higher on the tooth are usually wear related (figure 29).

THERMAL CYCLIC FATIGUE

Unlubricated and lubricated gears may fail due to thermal cyclic fatigue. Tooth bending stresses always result in some hysteresis heating and since plastics are such good thermal insulators, this results in a material operating temperature rise. This temperature rise can lower the strength of the material and cause pitch line deformation failure (tooth fold over).

FIGURE 28 TOOTH DEFORMATION DUE TO EXCESSIVE HEAT



FIGURE 29 TOOTH FRACTURE AT ROOT DUE TO OVER LOADING



MATERIALS

Gear materials have some basic requirements. The materials must be strong enough to handle the gear tooth loading and also have good wear and friction characteristics against the material of the mating gear. Impact and corrosion resistance are also important to some applications. The gear designer must carefully evaluate the requirements, both environmentally and mechanically, that the gear demands, and compare these to the properties associated with the intended materials.

As stated earlier, the values used to evaluate a gear are seldom found on a data sheet. Standard mechanical and physical properties are evaluated at conditions that are seldom seen in gears. If engineering properties, such as isochronous stress strain curves, tensile creep or flexural fatigue data are available at a variety of temperatures and strain rates, then a better picture of how a material will behave can be seen. Even if the required data is available, prototype testing is still strongly recommended.

While the majority of wear data does not directly apply to gear applications, thrust washer, block-on-ring or pin-on-disc wear data can give a comparative ranking of possible candidates. Extensive thrust washer test data has been generated on thermoplastic composites against steel and other metals (aluminum, brass, etc.), and thermoplastic composites at room and at elevated temperature. This type of data can be useful in screening potential candidates for gear prototyping. Unfilled nylon 6/6's wear factor of 200 is the benchmark for determining whether a composite has an acceptable wear rate. Wear factors over 200 indicate a material has an unacceptably high wear rate and is unsuitable for most gearing applications. Factors lower than 200 indicate a potentially viable gearing material.

Another number that should be used when selecting a gear material is the limiting PV (pressure-velocity) value. This indicates a composite's load or speed

limitations. In the PV test, the load on a rotating bearing increases incrementally until failure. Composite selection is generally based on a maximum of 50% of the limiting PV to allow for a factor of safety.

For a more complete look at thrust washer and limiting PV testing, see "A Guide to LNP's Internally Lubricated Thermoplastics." Unfilled acetal and unfilled nylon 6/6 were among the first thermoplastics used commonly in gearing applications. These crystalline resins possess good inherent wear resistance, low coefficients of friction and good chemical resistance. However, their high mold shrinkage and reduced speed/load capability limit the number of potential applications. Today, many thermoplastic resins are compounded with internal lubricants for increased wear and lower friction, and reinforcements for added strength.

LUBRICANT ADDITIVES

Among the most widely used lubricants are PTFE powder (polytetrafluoroethylene) and silicone fluid. PTFE particles smear from the shear between the wear surfaces resulting in a PTFE film which transfers to the mating wear surface. This PTFE versus PTFE transfer film results in significantly reduced coefficients of friction and wear rates.

For example, when PEI (polyetherimide) is 15% lubricated with PTFE, the dynamic coefficient of friction decreases from 0.51 to 0.30 and the wear factor drops from 3940 to 106.

With a wear factor below 200, this amorphous resin can now be considered as a potential gear candidate (figure 30). Amorphous resins are important because they have a lower shrinkage rate in the mold than crystalline resins and can be molded to produce more accurate gear.

Another popular lubricant, silicone fluid, migrates to the wear interface and is present upon start-up. Silicone fluid is

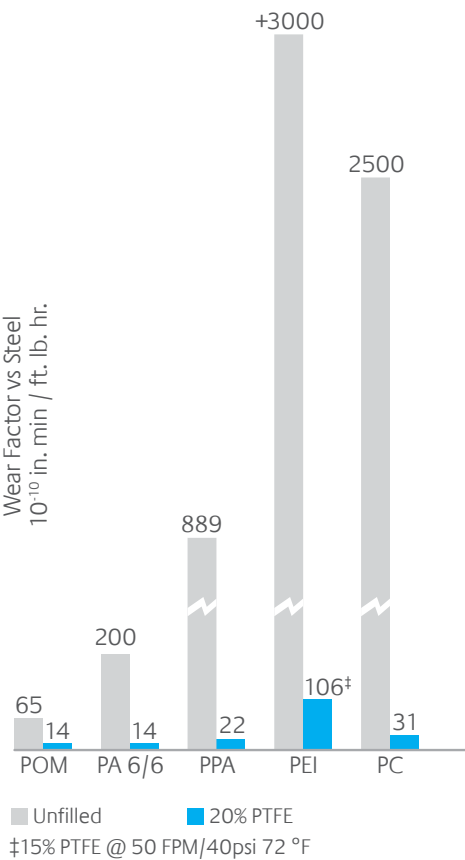
used alone or in conjunction with PTFE for even lower wear factors. The wear factor for a 2% silicone lubricated polycarbonate drops from 2500 to 386.

A combination of 13% PTFE and 2% silicone further reduces the wear factor to 42. The PTFE/silicone combination offers improvements at high speeds and is often selected for applications where oscillating speed or motion occurs.

GEAR PAIRS

The wear of a plastic gear depends significantly on the opposing gear, or counterface. For plastic on metal gear pairs, composites may wear differently against a relatively hard metal such as 1141 steel than against soft metals such as aluminum and brass, and certain formulations will wear better than others.

FIGURE 30 EFFECT OF PTFE LUBRICATION ON VARIOUS RESINS



The surface finish of the metal gear will also have an effect on the wear of the plastic gear. A good surface finish range for metal gears wearing against plastic gears is 12-16 µin.

For example, consider two nylon 6/6 formulations one is a 30% carbon fiber reinforced 15% PTFE lubricated nylon 6/6, and the other is 10% aramid fiber reinforced, 10% PTFE lubricated nylon 6/6. Both have wear factors of 13 against steel.

However, against aluminum, the carbon fiber formulation has a wear factor of 175, while the aramid fiber formulation has a wear factor of 45. Also of importance is the fact that the aluminum shaft will wear more against the carbon fiber reinforced composite (95) than it will against the aramid composite (4). Aramid fiber reinforcements are good against soft metal and powdered metal because they help minimize the generation of abrasive metal particles.

PLASTIC-ON-PLASTIC WEAR

For plastic-on-plastic gear pairs, composite selection becomes increasingly complex. These wear combinations are extremely difficult to predict and can only be confirmed by testing. In general it is easier to find acceptable wear combinations using dissimilar materials, although there are certain thermoplastic composites that will wear well against themselves.

In many cases materials with good natural lubricity have high wear rates. When unfilled acetal (generally considered to have good natural lubricity) is paired against itself, the wear factor is over 10,000. The addition of 20% PTFE, however, yields acceptable wear factors against itself of around 40.

ELEVATED TEMPERATURE GEARS

Thermoplastic composites are limited in elevated temperature gear applications because mechanical properties degrade as temperature increases to the melt point/glass transition temperature.

MATERIALS

When designing plastic gears for high temperature applications, it is important to understand the mechanical performance of the candidate materials at the application temperature. This includes wear data, since wear rates also tend to increase as temperature increases.

Most high temperature gear applications utilize high melt temperature/high glass transition temperature resin such as PES (polyethersulfone), PEI (polyetherimide), PPS (polyphenylene sulfide), PPA (polyphthalamide) and PEEK (polyetheretherketone). High temperature gearing applications almost always have fiber reinforcement and/or internal lubrication.

REINFORCEMENTS

Reinforcing fibers such as glass, carbon or aramid compounded into a resin improve mechanical performance. Carbon fiber reinforcement provides the greatest enhancement in mechanical strength and stiffness followed by glass fibers and then aramid fibers.

The addition of reinforcing fibers alone significantly lowers wear factor of most resin systems (table 3). Combining PTFE and fiber reinforcements results in further reduction in wear factor. For 15% PTFE lubricated nylon 6/6 with typical fiber loadings (30% for glass and carbon, 15% for aramid) wear factors decrease to less than 20.

The main disadvantage of glass and carbon fibers is that they induce anisotropic shrinkage in the mold, which could yield less accurate gears. Aramid fibers behave more isotropically, with lower differential between flow and transverse shrinkage values. Composite formulations with minimal reinforcing fiber may benefit from particulate fillers such as milled glass or glass beads, which do not increase differential shrinkage. However, these materials will generally have lower mechanical properties and higher wear rates.

One of the advances in reinforced composites which offers the potential for increased growth in replacing metal gears is the use of long fiber technology. A comparison of the properties of long and short glass nylon 6/6 with glass fibers and PTFE reveals significant improvements in flexural strength and impact strength with long fiber reinforcement. Wear rate is not dramatically increased since the number of fiber ends is reduced.

The higher strength and impact properties translate to improved tooth strength and fatigue endurance under the high torque achieved in some gear applications (table 4).

TABLE 3 COMPARISON OF REINFORCING FIBERS IN A NYLON 6/6 MATRIX

Property	ASTM method	Units	Unfilled	30% glass fiber	30% carbon fiber	15% aramid fiber
Shrinkage	D955	% (flow/trans)	1.5/1.8	0.40/1.5	0.08/0.56	0.02/0.03
Tensile strength	D630	psi/MPa	12000/83	24000/165	38600/266	1400/97
Flexural modulus	D790	ksi/GPa	410/2.8	1370/9.4	2720/18.8	560/3.9
Coefficient of friction	LNP	Static/dynamic	0.55/0.65	0.57/0.11	0.30/0.32	0.75/0.73
Wear factor	LNP	10 ⁻¹⁰ in ⁵ -min ft.-lb.-hr..	200	75	36	19

TABLE 4 COMPARISON OF LONG AND SHORT GLASS FIBER REINFORCED PTFE LUBRICATED NYLON 6/6

Property	ASTM	Units	40% long glass fiber 10% PTFE	30% short glass fiber 10% PTFE
Tensile strength	D630	psi/MPa	30000/207	26600/183
Tensile elongation	D630	%	2.5	3.2
Flexural modulus	D790	Ksi/GPa	1730/11.9	1350/9.3
Notched Izod impact	D256	ft.-lb./in. / J/m	5.1/257	2.1/106

TABLE 5 APPROXIMATE RELATION TO 1982 TRADE STANDARDS

Designation	Description of level	AGMA 2000-A88	DIN 3963/1978	
AA Ultra-high accuracy	Highest possible accuracy. Requires special tool room methods. Used for master gears, unusually critical high speed gears, or when both highest load and highest reliability are needed.	14 or higher	2 or 3	
A High accuracy	High accuracy, achieved by grinding or shaving with first rate machine tools and skilled operators. Used extensively for turbine gearing and aerospace gearing	12 or 13	4 or 5	
B Medium-high accuracy	A relatively high accuracy, achieved by grinding or shaving with emphasis on production rate rather than high quality. May be achieved by hobbing or shaping with best equipment and favorable conditions. Used in medium speed industrial gears and the more critical vehicle gears.	10 or 11	6 or 7	
Typical plastics range	C Medium accuracy	A good accuracy, achieved by hobbing, shaping or injection molding with first-rate machinery and operators. Typical use is for vehicle gears and electric motor industrial gears running at lower speeds.	8 or 9	8 or 9
	D Low accuracy	A nominal accuracy for hobbing, shaping and molding can be achieved with older equipment and less skilled operators. Typical use is for low-speed gears that wear into a reasonable fit (soft metals) and most plastic gears.	6 or 7	10 or 1
E Very low accuracy	An accuracy used for gears at slow speeds and light loads. Can be cast or molded. Typical use is for toys and gadgets. May be used in low-hardness power gears with limited life and reliability needs.	5 or 4	12	

FABRICATION

The overall accuracy of injection molded gears is a function of material composition, part design and fabrication (tool design and processing). No matter how good the material or part design, an accurate gear cannot be made in a poorly designed tool or with improper processing. Before we talk about how accurate gears can be molded, we need to discuss how accuracy in gears is measured.

If two mating gears with standard tooth thicknesses are brought into close mesh, the distance between their centers will be half the sum of their standard pitch diameters. This distance is called their standard center distance. These gears could only rotate at their standard center distance if both gears were perfect, and any error in the gears would cause them to bind at some point during rotation.

The types of errors present can be classified as:

- Runout
- Lateral runout or wobble
- Pitch error
- Profile error

The pitch error plus the profile error add up to what is called tooth-to-tooth composite error, or TTE. This describes the variance in shape and position of one tooth to another on the gear being inspected. The overall runout of the gear, or the amount the gear is out of round, is added to the tooth-to-tooth composite error to get the total composite error, or TCE.

The tooth-to-tooth and total composite error of a gear can be measured by rotating the gear in close mesh with a master gear of known accuracy in a variable center distance fixture. As the gears rotate, the center distance will vary with the accuracy of the gear being tested.

This radial displacement can be measured and charted. An example of the chart is shown in figure 31. If the gear was perfect, the chart would be a straight line. For plastic gears, the greater error is usually in overall runout of the gear, and not in tooth-to-tooth error.

Two accepted standards for rating the quality of an injection molded gear is AGMA 2000-A88 and DIN 3963/1978. In the AGMA system gears are classified by number in accordance with their accuracy in terms of the maximum tooth-to-tooth and total composite tolerances allowed them.

This number is called the AGMA quality number, and the higher the number, the tighter the tolerances on the gear. These standards are the same standards used for metal gears, so an AGMA Q8 molded plastic gear is the same quality as an AGMA Q8 hobbed steel gear.

For example, for a 48 diametral pitch, 1.00" pitch diameter spur gear to be an AGMA Q7, it would allow a maximum tooth-to-tooth tolerance (TTE) of 0.00138" and a maximum composite tolerance (TCE) of 0.00275". For the same gear to be an AGMA Q10, the allowable maximum TTE would be 0.00036" and the maximum TCE would be 0.00010".

The AGMA quality numbers and the corresponding maximum tolerances, by diametral pitch and pitch diameter, are listed in the American Gear Manufacturers Association "Gear Handbook, 390.03".

For reference, these quality levels can be broken into six major levels of accuracy. Most molded gears fall into the AGMA Q4 to Q8 range, but AGMA Q10 gears have been molded.

When designing a gear the designer must choose an accuracy level that is within the capability of the gear shops available. In addition, the gears need to be made at a reasonable cost. In the competitive world, it is not simply the best gear that is needed. It is the lowest-cost gear that will adequately meet load, life, reliability and quietness requirements.

FIGURE 31

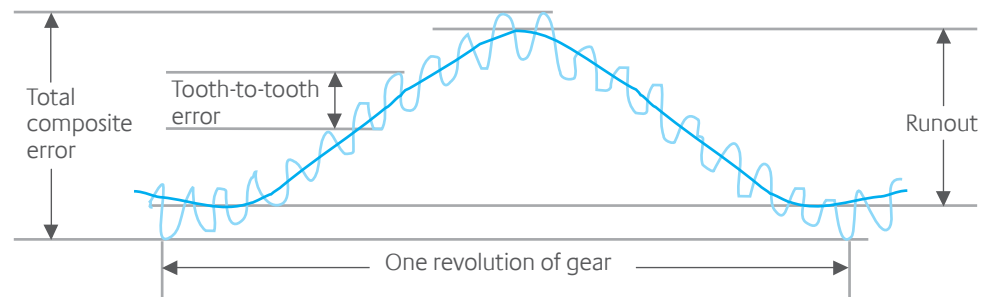
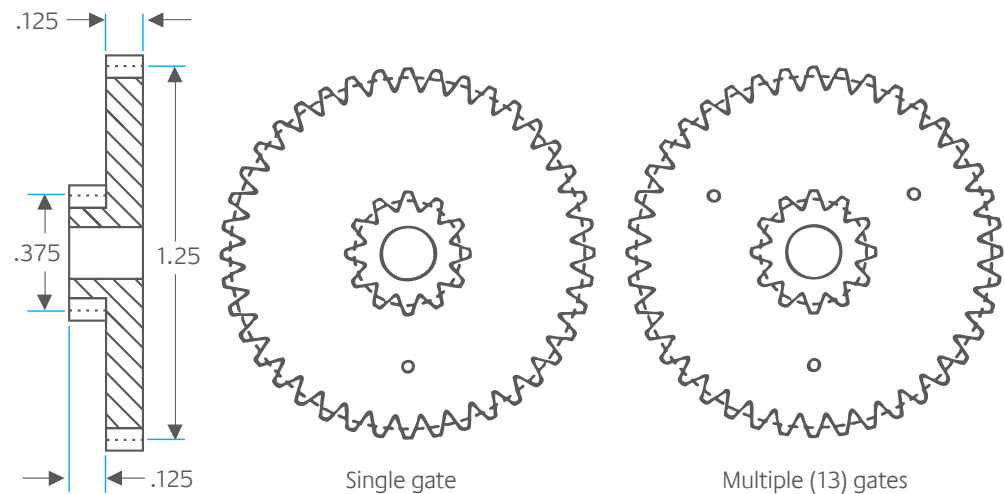


FIGURE 32

Test gears 32 diametral pitch compound gear



FABRICATION

MATERIAL EFFECT ON GEAR ACCURACY

Material choice can have an effect on gear accuracy. Amorphous resins have shrinkage characteristics which tend to be more isotropic than crystalline resins, and particulate fillers behave more isotropically than fibrous reinforcements. If the shrinkage behavior of a material is well understood, then a mold cavity can be cut to mold an accurate gear from that material. However, isotropic shrinkage is easier to compensate for when cutting a gear cavity.

To evaluate the effects of different resins and filler systems on the accuracy of molded gears, various thermoplastic compounds were injection molded as a 32 pitch, 20° pressure angle, 1.25" pitch diameter, 0.125" wide spur gear, integrally molded with a smaller pinion gear (figure 32). The gear had a single, off-center gate on the web of the gear. The base resins chosen were nylon 6/6 and polycarbonate.

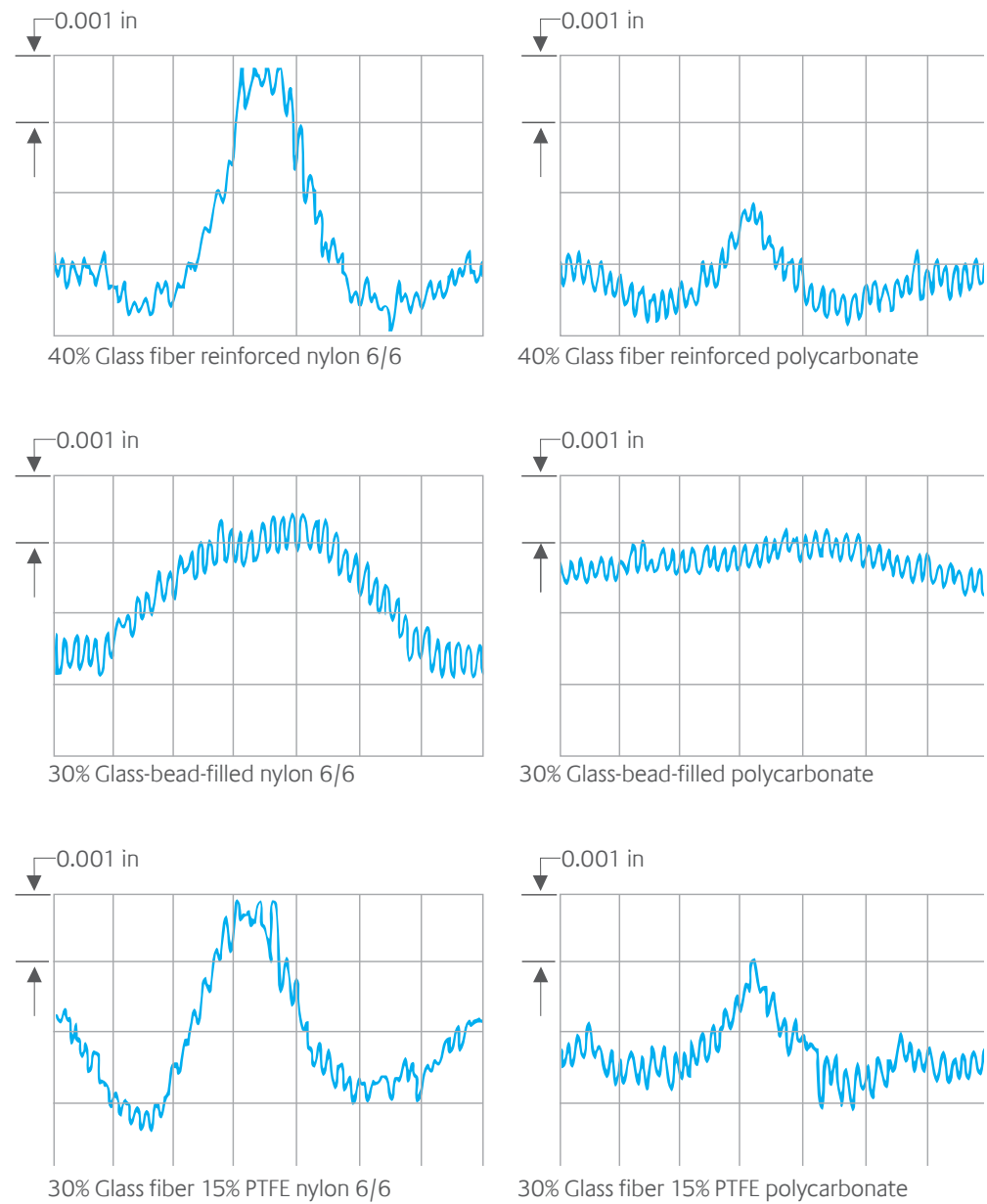
These commonly used gear materials represent the two major types of thermoplastic resins high shrinkage crystalline materials (nylons, acetals, and olefins) and low shrinkage amorphous compounds (polycarbonates, polysulfones, ABS and SAN). Typical molding conditions were used for each resin, and molding conditions were held constant for each base resin regardless of the filler type or content.

For each resin, a 40% glass fiber reinforced, a 30% glass bead filled and a 30% glass fiber, 15% PTFE lubricated formulation was molded. TCE charts for each formulation are shown in figure 33. For both base resins, the glass fiber reinforced formulations showed a single large peak. This peak is a high spot in the gear, and is the result of high fiber orientation on the side of the gear opposite the gate.

The glass bead filled nylon 6/6 compounds also showed a single peak, but the magnitude of the peak is reduced compared to the glass fiber reinforced nylon. This is because particulate fillers shrink isotropically and their alignment on the far side of the gear is irrelevant. The peak in this compound is due to the anisotropic nature of crystalline materials. The glass bead filled polycarbonate shrank essentially isotropic, giving a flat curve. This compound produced the most accurate gear.

The TCE chart for 30% glass fiber, 15% PTFE lubricated compounds showed a single peak, similar to the 40% glass fiber reinforced compounds. The addition of the PTFE to a glass reinforced compound has little effect on runout.

FIGURE 33



TOOL DESIGN AND GEAR ACCURACY

To get accurately molded thermoplastic gears, you must have an accurate mold. The alignment of the tool halves and cavity bores will be critical in gear molding. The use of interlocks between the mold halves to remove running fits in the guide systems is recommended.

Air hardened steel is preferred over oil hardened steel due to its improved dimensional stability during the heat treating process. Also, a steel with a higher carbon content for overall hardness and higher chromium content for better wear resistance (to filled materials) is suggested for tight tolerance designs.

For good control of tolerances the use of H-13 or A-2 steel with D-2 steel used for gate inserts, core pins and other high wear areas of the part is suggested.

Cooling is crucial for tolerance control in molding gears. An even temperature must be maintained across the mold in order to allow the material to shrink at an even, controlled rate. Uneven shrinkage will lead to dimensional tolerance variations. Special attention should be given to core pins and deep cores, as they tend to run hot.

Three plate molds with naturally balanced runner systems are preferred for tight tolerance gear molding. While multi-cavity molds are common, family molds are not recommended.

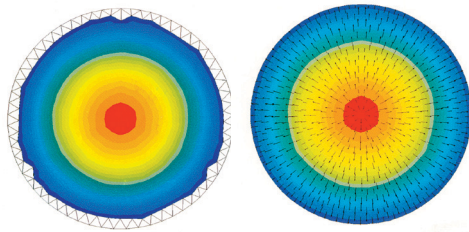
The use of runnerless (hot runner) systems is possible, but may reduce the tolerance capacity of the tool. The heat required to keep a runner hot will also heat a portion of the tool, so additional cooling will be required. If a hot-runner system is chosen, an adequate cooler plate must be used to properly control the mold temperature.

Venting is important, since too little venting traps air in the mold and can lead to variations in the melt temperature and the cavity pressure as the part is filled. Either of these conditions can also affect tolerance capability. Provide for as much venting as possible in the tool, especially at the areas that are last to fill. Ejection systems must be designed to insure minimum distortion of the part as it is ejected from the tool.

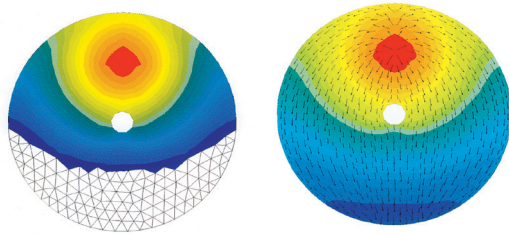
Core pins, slides and side actions can be found in many gear molds. Whenever possible, those features should go through the part and lock into a nest in the other mold half. This prevents deflection of the feature over time, which can be caused simply by the repeated impact of the plastic flow front during processing.

FIGURE 34

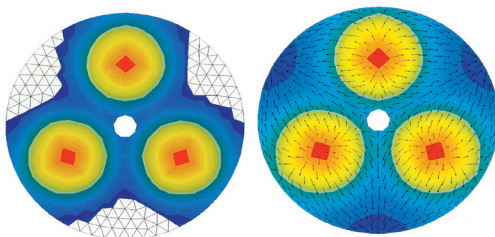
Filling pattern and resultant fiber orientation from a single center gate. Fiber orientation is uniform



A single off-center gate results in non-uniform fiber orientation, causing differences in mold shrinkage



Using multiple gates results in a more uniform fiber orientation and filling pattern. Effect of weld lines is minimized



Gate position in molded gears can have a dramatic effect on the accuracy of the gear, particularly the runout. The best type of gate for an injection molded gear is a disc or diaphragm gate. Figure 34 shows a mold filling analysis for an extremely simple gear with a disc, single and multiple gates. The disc gate provides completely uniform flow in the radial direction and no weld lines. This results in a gear that shrinks the same in all directions. Since this is not usually practical for production gears, the gates are usually placed on the web of the gear.

With gates on the web it is preferable to use multiple gates evenly spaced around the gear. When a single gate is used, the plastic must flow around the central core pin. This forms a small meld line near the core pin, and then the flow front moves out from the center. This flow pattern results in a high degree of fiber orientation in the radial direction on the side of the gear opposite the gate.

TOOL DESIGN AND GEAR ACCURACY

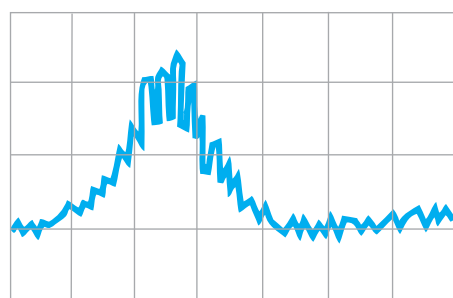
With multiple gates the flow pattern remains more random. The flow moves radially out from the gates, and where the flow fronts meet, three weld lines are formed. At a weld line, the fibers tend to orient themselves parallel to the flow front. In a gear, this will result in the fibers being oriented radially at the weld lines and more randomly in the rest of the gear. This results in an area of low shrinkage along the weld lines. The difference in orientation between the weld lines and the rest of the gear is less than that in the single gated gear, so the gear is more accurate.

To illustrate this even further the same 32 pitch, 20° pressure angle, 0.125" thick spur gear used in the material study (figure 32 on page 31) was molded using a single gate and three equidistant gates in the web area of the gear. Figure 35 compares the TCE charts for a 40% glass fiber reinforced nylon 6/6 with one and three gates.

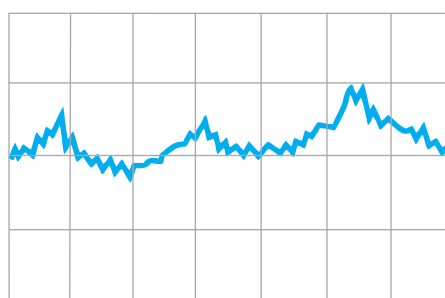
The single gated gear had a single large peak, indicating an egg shaped gear. This peak is a high spot in the gear, and correlates to the high degree of fiber orientation on the side opposite the gate. The radially oriented fibers reduced the shrinkage on one side of the gear, resulting in a high spot on the gear.

In the triple gated gear, there are three high spots, due to the three weld lines. However, the high spots are reduced in magnitude, since the plastic had less distance to flow, and less fiber orientation takes place. The multiple gate system is closer to providing the condition of concentric, uniform flow which would result from a disc-gated gear.

FIGURE 35 GEAR TOLERANCES AS A FUNCTION OF GATING



Single gate
40% glass-reinforced nylon 6/6



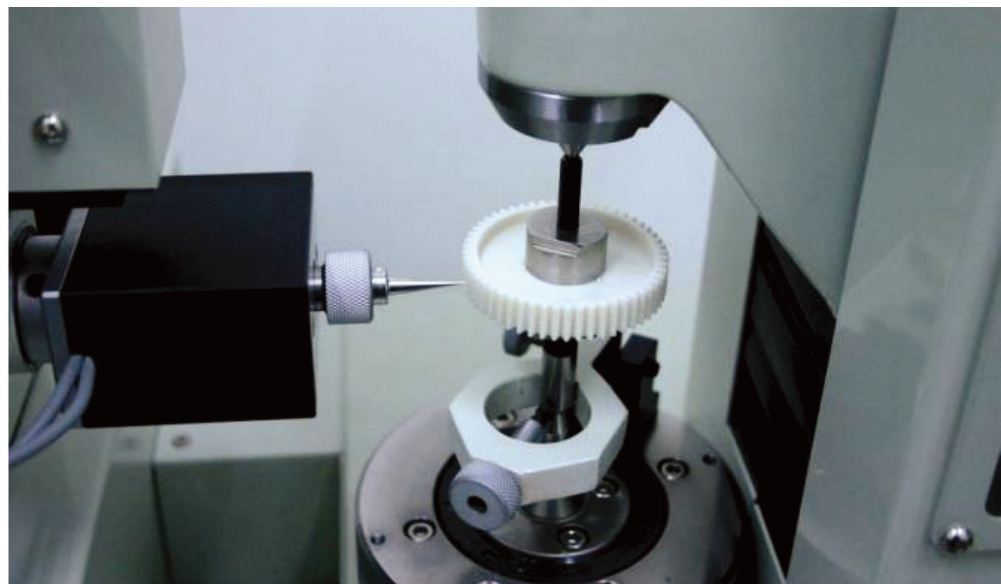
Triple gate
40% glass-reinforced nylon 6/6

EFFECTS OF MOLDING PARAMETERS

Process variables do appear to have some effect on overall gear accuracy but this appears to differ between crystalline and amorphous resins. Using the single gated gear as discussed above, standard process conditions were established for 40% glass fiber reinforced nylon 6/6 and 30% glass fiber reinforced polycarbonate. Process variables were adjusted for higher and lower injection pressure, injection rate, holding pressure, cylinder temperature and mold temperature (table 6 on page 38).

Crystalline nylon had the lowest TCE's (more accurate) when parameters which control material solidification were optimized (lower mold temperature, maintain holding time until cool). Amorphous polycarbonate produced the most accurate gears when molding conditions placed minimal shear on the melt (i.e. higher melt temperature).

What is probably of more importance in molding plastic gears is the stability and repeatability of the injection molding process itself. Use of closed loop process controls is highly recommended. As noted above, changes in packing pressure, melt temperature, and material mixing can have a substantial impact on the shrinkage of the materials, and closed loop process controls allow the molder to make the adjustments needed to maintain constant molding parameters.



FABRICATION

TABLE 6

Molding variables	30% glass fiber reinforced nylon 6/6		30% glass fiber reinforced polycarbonate	
	TCE (in x 10 ⁻⁴)	TTE (in x 10 ⁻⁴)	TCE (in x 10 ⁻⁴)	TTE (in x 10 ⁻⁴)
Typical conditions	33	7	18	7
Lower injection pressure	26	7	18	7
Higher injection pressure	40	8	20	6
Slower injection speed	30	7	18	6
Shorter hold pressure	70	13	18	6
Longer hold pressure	43	5	18	5
Lower barrel temp.	40	9	–	–
Higher barrel temp.	46	6	13	5
No cushion	50	6	19	6
Hotter mold	48	8	–	–
Colder mold	–	–	20	7

LEWIS FORM FACTORY

USED WHEN LOADING AT TOOTH TIP

Number of teeth	20° full depth	20° stub
12	0.245	0.311
13	0.261	0.324
14	0.276	0.339
15	0.289	0.348
16	0.295	0.361
17	0.302	0.367
18	0.308	0.377
19	0.314	0.386
20	0.320	0.393
21	0.327	0.399
22	0.330	0.405
24	0.336	0.415
26	0.346	0.424
28	0.352	0.430
30	0.358	0.437
34	0.371	0.446
38	0.383	0.456
43	0.396	0.462
50	0.408	0.474
60	0.421	0.484
75	0.434	0.496
100	0.446	0.506
150	0.459	0.518
300	0.471	0.534
Rack	0.484	0.550

LEWIS FORM FACTORY

USED WHEN TANGENTIAL TOOTH LOADING IS
CALCULATED AT THE PITCH POINT

Number of teeth	20° full depth	20° stub
12	0.415	0.502
13	0.442	0.524
14	0.468	0.540
15	0.490	0.565
16	0.500	0.577
17	0.512	0.588
18	0.520	0.605
19	0.533	0.617
20	0.544	0.626
21	0.551	0.640
22	0.557	0.646
24	0.571	0.665
26	0.587	0.677
28	0.595	0.687
30	0.605	0.697
34	0.629	0.712
38	0.650	0.730
43	0.671	0.738
50	0.696	0.756
60	0.712	0.775
75	0.734	0.791
100	0.758	0.807
150	0.780	0.832
300	0.802	0.854
Rack	0.824	0.882

LIST OF SYMBOLS

C	=	close mesh center distance
C_s	=	service factor
D	=	pitch diameter
D_p	=	pitch diameter, pinion
E	=	modulus of elasticity
f	=	face width
F	=	tangential tooth loading at the pitch line
HP	=	horsepower
m_g	=	speed ratio, N_g/N_p
M	=	expansion due to moisture absorption of hub material (in./in.)
n	=	factor of safety
N	=	number of teeth
P_d	=	diametral pitch
T	=	maximum operating temperature gears will see in °F
T_{ct}	=	maximum total composite tolerance in gear
TIR	=	maximum allowable runout of bearing
V	=	pitch line velocity (fpm)
w	=	speed, rpm
W_t	=	transmitted load
y	=	Lewis form factor at the tooth tip
Y	=	Lewis form factor for plastic gears, loaded at the pitch point
a	=	coefficient of linear thermal expansion of the material (in./in./°F)
Δ_c	=	required increase in center distance
μ	=	Poisson's ratio
S_b	=	bending stress
S_h	=	surface compressive stress (Hertzian stress)
ϕ	=	pressure angle

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