Plastic Gear Design Basics

LNP Engineering Plastics, Inc.

Plastic gears are 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 are quieter than metal. 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 with part consolidation in mind to incorporate other features needed in an assembly. These gears are also resistant to many corrosive environments.

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 that available for metal gears; nonetheless, there are certain guidelines available for estimating the technical feasibility of their use. However, these guidelines have evolved from equations originally worked out for metals and do not take into account some of the unique behavior found in thermoplastic materials.

This article 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 as well.

Plastic Gear Tooth Design

Hobs used to cut teeth in metal gears are available off the shelf, and for cost reasons, designers of commercially cut gears seldom use any other tooth forms. Injection-molded gears are not affected by the constraints of 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.

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 a varying extent in most plastic gears. This type of interference can cause noise, excessive wear and a loss of smooth uniform motion. To compensate, the tip of the tooth is gradually thinned from half way up the addendum. This modification is most useful in gears that are highly loaded (for their 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.

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

Two tooth forms for plastic gears that incorporate these modifications are the PGT tooth forms (Fig. 1) and the ISO R53 Modified (Fig. 2). These tooth forms are essentially the same, differing only in nomenclature. The ISO form uses the metric module, m, while the PGT

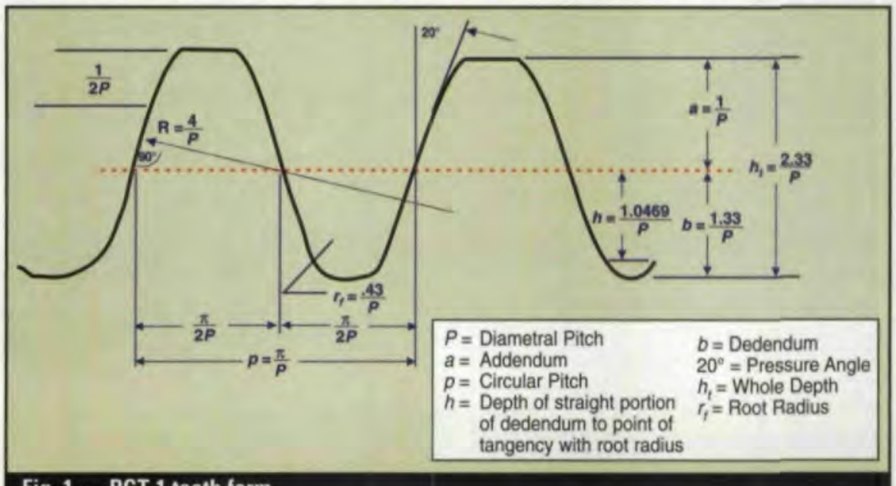

Fig. 1 — PGT-1 tooth form.

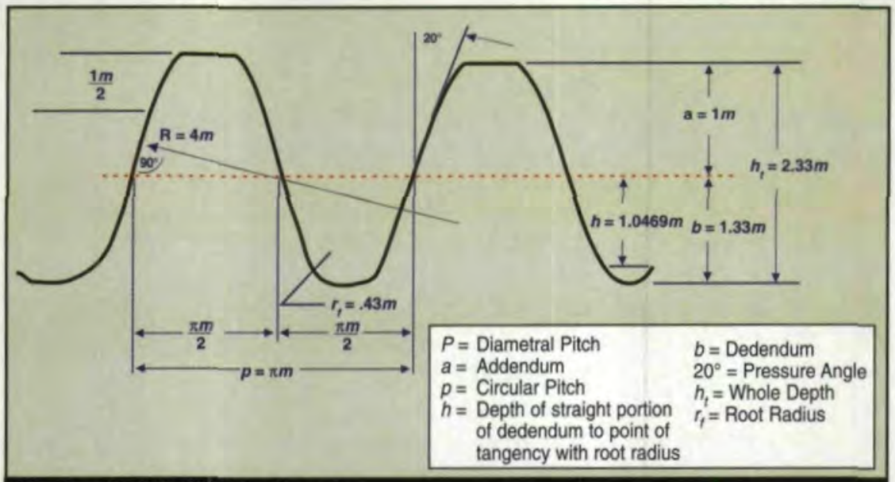

Fig. 2 — Modified ISO basic rack.

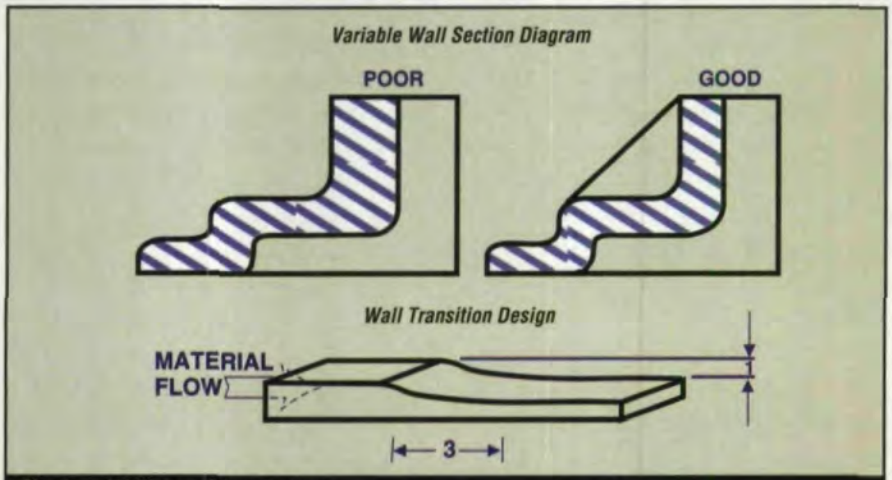

Fig. 3 — Wall transition design.

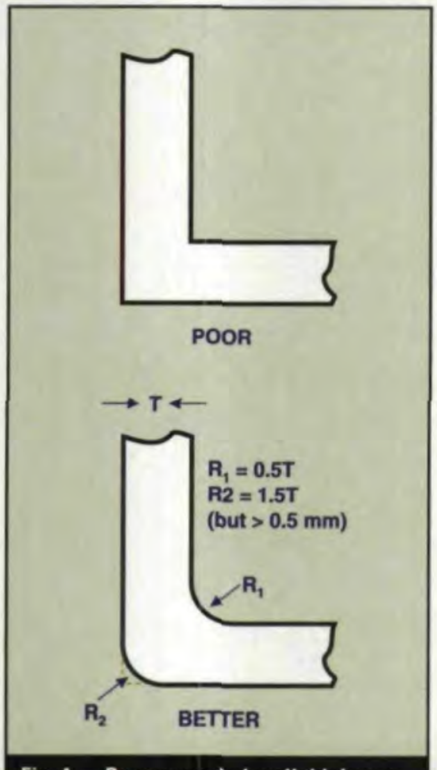

Fig. 4 — Recommended wall thickness design.

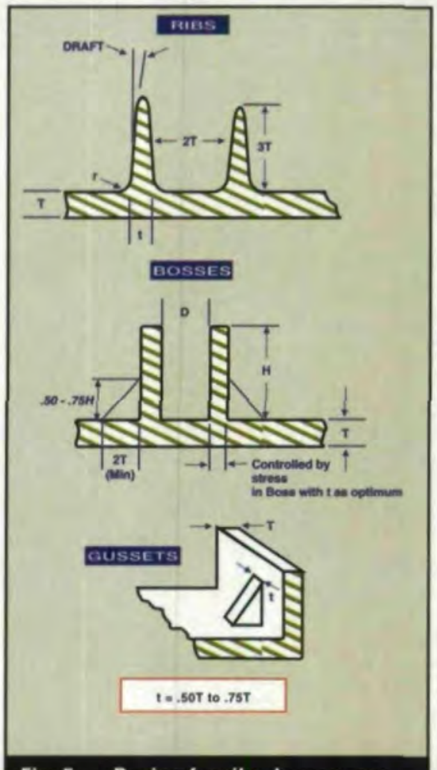

Fig. 5 — Design for ribs, bosses, gussets and projections.

tooth forms use diametral pitch, *P*. 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.

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, 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 modifications 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 plastic gear teeth will be attached to some plastic part, these rules must also apply to the overall design.

One of the most important features in a good plastic design is the nominal wall. The nominal wall is the feature 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"–0.200" thick. Although there is no such thing as an average wall thickness for an injection- molded plastic part, 0.125" 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 (Fig. 3). *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.

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% and a maximum of 75% of the nominal wall. 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% (Fig. 4).

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 $2^{1}/_{2}$–3 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 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 (Fig. 5).

Gear Layout

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 gated in the center. This gear will not have any 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.

When designing a plastic gear that has a hub and a rim, careful consideration must be given to the thickness of the various parts. Tooth thickness and height have 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 a wall (the rim), the thickness

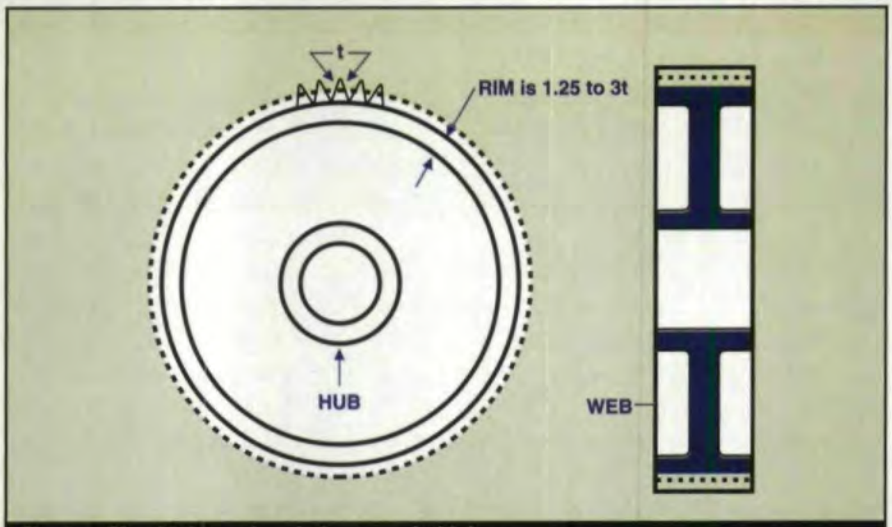

Fig. 6 — Rim thickness in relation to web thickness.

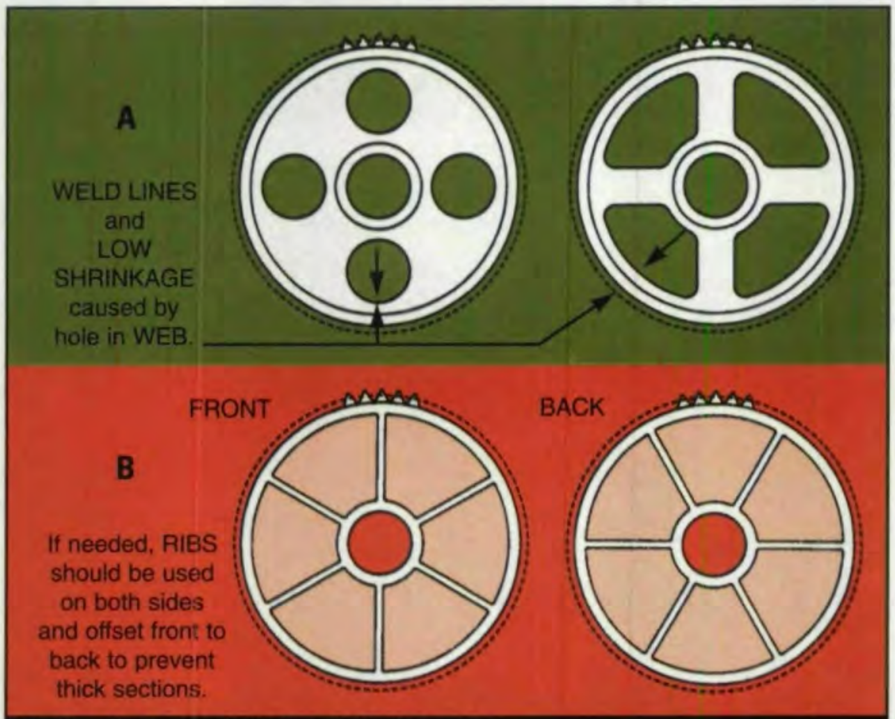

Fig. 7 — Recommended hole and rib placement.

Table 1 — Common Allowances for Moisture

Material	M(in/in)
Acetal	0.0005
Nylon 6/6	0.0025
Nylon 6/6 + 30% glass fiber	0.0015
Polycarbonate	0.0005

Formula For Increasing Center Distance

$$\Delta_c = \frac{T_{ct_1} + T_{ct_2}}{2} + C\left[(T-70)\left(\frac{\alpha_1 N_1 + \alpha_2 N_2}{N_1 + N_2} - \alpha_H\right) + \left(\frac{M_1 N_1 + M_2 N_2}{N_1 + N_2} - M_H\right)\right] + \frac{TIR_1 + TIR_2}{2}$$

of the rim should be 1¼–3 times the thickness of the gear tooth (Fig. 6). 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. Once again, the web should not be more than 1¼–3 times thicker than rim and hub. If the hub must be thick, 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 50–75% 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 (Fig. 7a). 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 (Fig. 7b).

Assemblies

The four modifications to standard gear and basic thermoplastic part design guidelines outlined above will provide stronger injection molded plastic gears. However, the gears must remain in 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. 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 can be calculated with the equation displayed on page 38,
where:

- Δ_c = 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
- α = 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, 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 nylon, the expansion may be more important. Some common allowances are shown in Table 1. For materials not shown on this table, you can use the polycarbonate numbers for low moisture materials and the Nylon 6/6 numbers for hygroscopic materials.

Part Consolidation

One of the most useful features of thermoplastic injection molded gears is the ability to consolidate a number of parts into one multi-functional design. 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 called 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. Fig. 8 shows the difference between a good design and one that is too thick.

Gear Testing

Plastic and metal gears fail in the same ways if the gear materials design limits are exceeded. All new applications should be prototype tested near or at operational conditions. 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 the normal working temperature may cause rapid failure, whereas under normal operating conditions, the gear may work well. Test conditions should always come as close to actual conditions as possible.

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 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, PTFE-lubricated gears will have a 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 load on a tooth with a reduced cross-section. 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 ones. 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 12–15 μin. 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. Design for abrasive wear should be avoided.

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 plastic, 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 temperature (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.

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 ar"e usually due to overload. Fractures high on the tooth are usually wear related.

Thermal Cyclic Fatigue. Unlubricated and lubricated gears may fail due to thermal cyclic fatigue. Tooth bending stresses always result i 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).

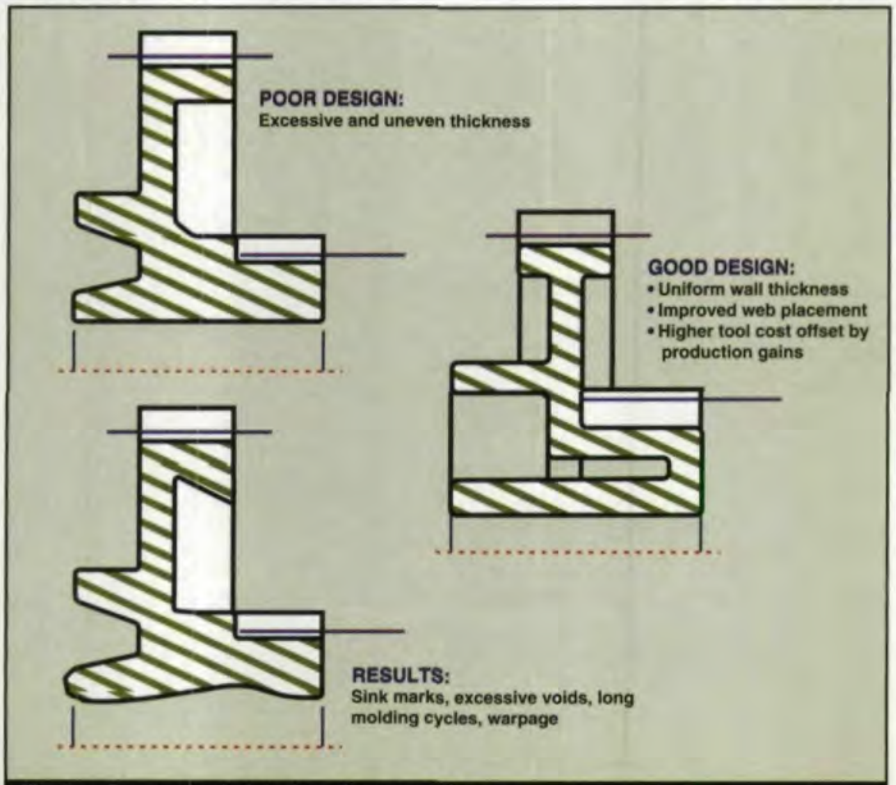

Fig. 8 — Recommended design of compound gears.

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