8 – Gears

Introduction

DELRIN® acetal resins and ZYTEL® nylon resins are used in a wide variety of gear applications throughout the world. They offer the widest range of operating temperature and highest fatigue endurance of any thermoplastic gear material, accounting for their almost universal use in non-metallic gearing.

The primary driving force in the use of plastic vs. metal gears is the large economic advantage afforded by the injection moulding process. In addition, cams, bearings, ratchets, springs, gear shafts and other gears can be designed as integral parts in a single moulding, thus eliminating costly manufacturing and assembly operations. Tolerances for plastic gears are in some cases less critical than for metal gears because the inherent resiliency enables the teeth to conform to slight errors in pitch and profile. This same resilience offers the ability to dampen shock or impact loads. The use of ZYTEL® nylon resin as the tooth surface in engine timing chain sprockets is an outstanding example of this latter advantage. In this case, timing chain life is extended because nylon dampens somewhat the transmission of shock loads from fuel ignition. ZYTEL® nylon resin and DELRIN® acetal resin have low coefficients of friction and good wear characteristics, offering the ability to operate with little or no lubrication. They can also operate in environments that would be adverse to metal gears. A summary of the advantages and limitations of plastic gears is given in Table 8.01.

Knowledge of the material performance characteristics and use of the gear design information to follow is important to successful gear applications in DELRIN® acetal resin and ZYTEL® nylon resin.

Gear Design

The key step in gear design is the determination of the allowable tooth bending stress. Prototyping of gears is expensive and time consuming, so an error in the initial choice of the tooth bending stress can be costly.

For any given material, the allowable stress is dependent on a number of factors, including:

– Total lifetime cycle.
– Intermittent or continuous duty.
– Environment – temperature, humidity, solvents, chemicals, etc.
– Change in diameter and centre to centre distance with temperature and humidity.
– Pitch line velocity.
– Diametral pitch (size of teeth) and tooth form.
– Accuracy of tooth form, helix angle, pitch diameter, etc.
– Mating gear material including surface finish and hardness.
– Type of lubrication (frictional heat).

Selection of the proper stress level can best be made based on experience with successful gear applications of a similar nature. Fig. 8.01 plots a number of successful gear applications of DELRIN® acetal resin and ZYTEL® nylon resin in terms of peripheral speed and tooth bending stress. Note that all of these applications are in room temperature, indoor environments. For similar applications operating at higher temperatures, the allowable stress should be corrected, see factor C_1 in Table 8.02. Since fatigue endurance is reduced somewhat as temperature increases, this effect must also be considered. Where very high temperatures are encountered, thermal ageing may become a factor.

Where suitable experience is not available, the allowable tooth stress must be based on careful consideration of all the factors previously outlined, and on available test data on the gear material of choice.

Table 8.01 **Advantages and Limitations of Plastic Gears**

Advantages	Limitations
Economy in injection moulding	Load carrying capacity
Combining of functions	Environmental temperature
No post-machining or burr removal	Higher thermal expansion coefficient
Weight reduction	Less dimensional stability
Operate with little or no lubrication	Accuracy of manufacture
Shock and Vibration damping	
Lower noise level	
Corrosion resistance	

A number of years ago, DuPont commissioned a series of extensive gear tests on gears of DELRIN® acetal resin and ZYTEL® nylon resin, which resulted in the information summarised in Tables 8.02 and 8.03. This data can be combined with environmental operating conditions to arrive at an allowable tooth bending stress.

Whether similar experience exists or not, it is essential that a prototype mould be built and the design carefully tested in the actual or simulated end-use conditions.

Table 8.02 **Allowed Fatigue Strength for DELRIN® and ZYTEL® 101**

$\sigma_n = \sigma_1\,[1 - C_n \, log\,(n)]$ (MPa)

where: σ_1 = fatigue strength for 10^6 cycles, see Table 8.03

C_n = 0,20 for ZYTEL® 101; = 0,22 for DELRIN®

n = number of cycles in million (industrial purposes: $n \geq 1000$)

$\sigma_{all} = c_1 c_2 c_3 \sigma_n$ (MPa)

where: $c_1 = 1 - 0{,}6\,(T - 20)/80$

T = temperature in °C

c_2 = factor for shock-load
no shocks: $c_2 = 1{,}0$
heavy shocks: $c_2 = 0{,}5$

c_3 = factor for velocity = $1/(1 + v)$

v = peripheral velocity; ≤ 5 (m/s)

$v = \pi\, d\, \omega / 60\,000$ (m/s)

d = pitch diameter (mm)

ω = rotational speed (rpm)

Table 8.03 **Fatigue Strength (σ_1) for DuPont gear materials for 10^6 cycles (MPa)**

Material	Mating Material	Lubrication Continuous	Initial
DELRIN® 100	Steel	48	27
DELRIN® 500	Steel	36	18
ZYTEL® 101	Steel	40	25
ZYTEL® 101	ZYTEL® 101		18

Once the admissible tooth bending stress has been determined, the designer can proceed with the selection of the other variables, for which an understanding of the basic terminology used in gear work is helpfull. The terms most commonly used to describe gears are:

– *Pitch diameter* (d) is the diameter measured at the pitch circle.

– *Diametral pitch* (P_d) is the number of teeth per inch of pitch diameter, commonly used in USA.

– *Module* (M) is pitch diameter divided by number of teeth (z). Thus: M = d/z.

For standard gears:

- external diameter = d + 2M;
- tooth thickness = 0,5 π M;
- tooth height = 2M + clearance.

– *Pinion* is the smaller of a pair of meshing gears.

– *Gear* is the larger of a pair of meshing gears.

– *Ratio* is number of teeth gear/number of teeth pinion.

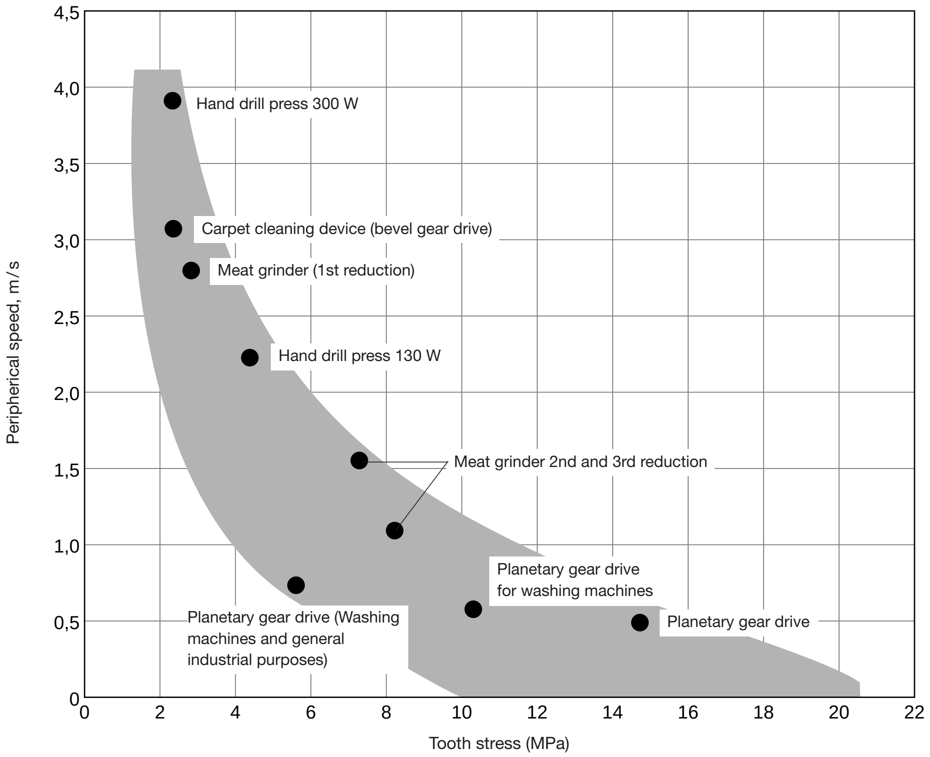

Fig. 8.01 **Speed versus stress: Typical gear applications placed on curve**

In a set of gears a torque is transferred by means of a peripheral force at the pitch diameter, with the following relation, (see also Fig 8.02):

$F = 2000\ T / d$

F =	peripheral (tangential) force	(N)
T =	torque, T = 9550 P/w	(N·m)
d =	pitch diameter	(mm)
P =	transmitted power	(kW)
ω =	rotational speed	(rpm)

In case both gear and pinion have a larger number of teeth, it is probable that multiple teeth are in contact while running. For smooth running, with as little production of vibration (noise) as possible, the number of theeth in contact should be independent of the rotation angle, and the take-over of the peripheral force by new sets of teeth in contact should also be smooth, for which reason gears are often made slightly helical.

For one set of (standard shaped) teeth in contact the following formula has been derived for the bending stress in the teeth of a spur gear:

σ =	F / (y M f)	(MPa)
y =	tooth form factor, $y = 0{,}25\ z^{0{,}25}$	
f =	tooth width	(mm)
z =	number of teeth	

The computed bending stress should always be lower than the allowable bending stress according Table 8.02.

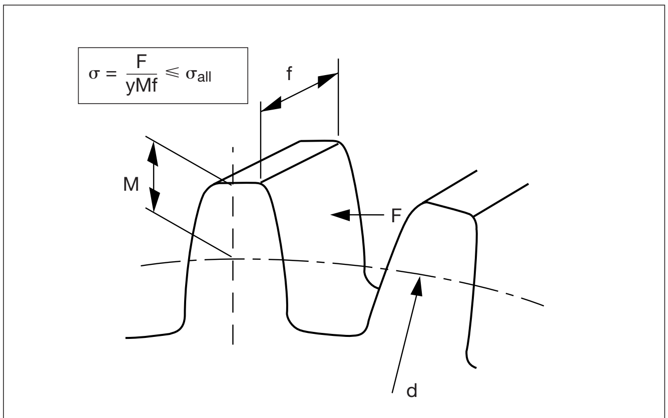

Fig. 8.02 **Gear design**

From a strictly functional and technical point of view, there is no reason to choose a larger tooth size than required. In the case of plastic gear design, the tooth size is often chosen smaller than necessary for the following reasons:

– Smaller teeth, for a given diameter, tend to spread the load over a larger number of teeth.
– Less critical moulding tolerances.
– Less sensitivity to thermal variations, post moulding shrinkage, and dimensional stability.
– Coarse module teeth are limited by higher sliding velocities and contact pressures

Designing for Stall Torque

There are many applications where the gear must be designed to withstand stall torque loading significantly higher than the normal running torque, and in some cases this stall torque may govern the gear design. To determine the stall torque a given gear design is capable of handling, use the yield strength of the material at expected operating temperature under stall conditions. Only a small safety factor (S = 1,3 – 1,5) needs to be applied if the material to be used is either ZYTEL® nylon resin or DELRIN® 100, as the resiliency of these materials allows the stall load to be distributed over several teeth.

Some applications, like window lift gears for cars, use a steel worm and a plastic helical gear (= worm gear), where the tooth thickness of the steel worm has been reduced in favour of the tooth thickness of the plastic gear. In this case the gear strength may be limited by the shear strength of the loaded teeth, as given by the equation:

F_{max}	= n f t τ	(N)
n	= number of teeth in (full) contact	
f	= tooth width	(mm)
t	= tooth thickness	(mm)
τ	= shear strength = σ_y / (1,7 S)	(MPa)
σ_y	= yield strength at design temperature	(MPa)

Again, adequate testing of the moulded prototype is necessary.

Gear Proportions

Once the basic gear design parameters have been established, the gear design can be completed. It is very important at this stage to select gear proportions which will facilitate accurate mouldings with minimum tendency for post moulding warpage or stress relief.

An ideal design as far as moulding is concerned is shown in Fig. 8.03. For reasons of mechanical strength, it is suggested that the rim section be made 2 times tooth thickness “t”.

The other sections depend both on functional requirements and gate location. If, for some reason, it is desirable to have a hub section ‘‘h’’ heavier than the web, then the part must be centre gated in order to fill all sections properly, and the web ‘‘w’’ would be 1,5 t. If the gate must be located

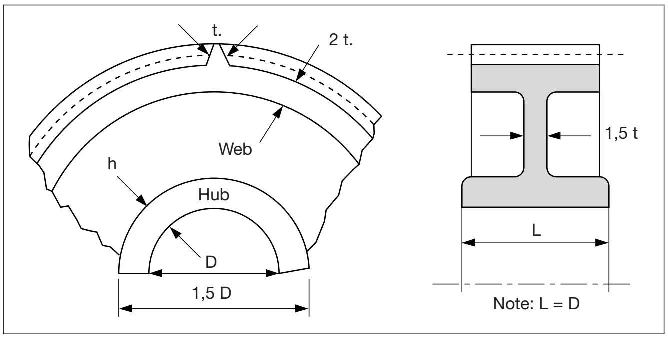

Fig. 8.03 **Suggested gear proportions**

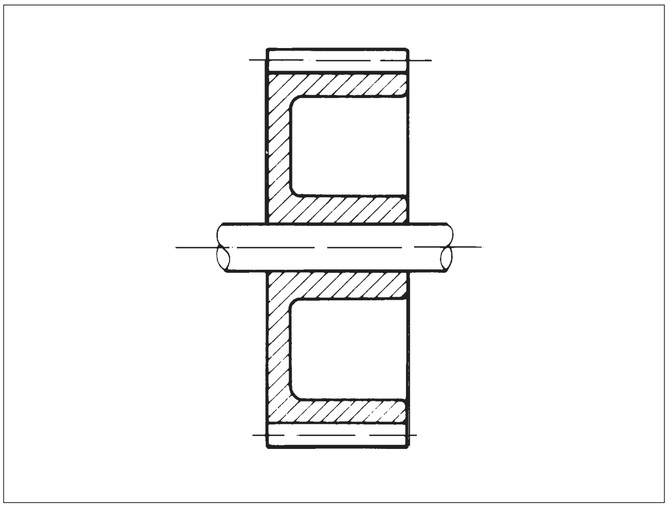

Fig. 8.04 **Gear with off-centre web**

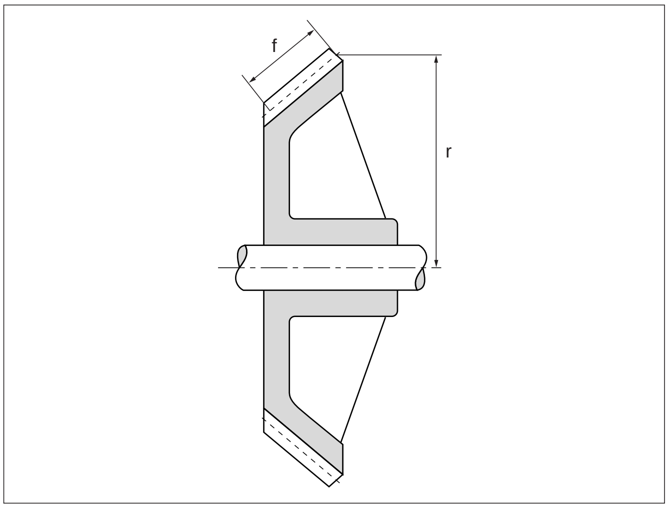

Fig. 8.06 **Ribbed bevel gear**

in the rim or the web, then web thickness should equal hub thickness, as no section of a given thickness can be filled properly through a thinner one.

The maximum wall thickness of the hub should usually not exceed 6 mm. For minimum out-of-roundness, use centre gating.

On gears which are an integral part of a multifunctional component or which have to fulfill special requirements as shown in Fig. 8.20-8.25, it could be impossible to approach the ideal symmetrical shape as shown in Fig. 8.03, in which case the assembly must be designed to accept somewhat less accuracy in the gear dimensions.

The following additional examples illustrate a few more gear geometries which could lead to moulding and/or functional problems:

- Relatively wide gears which have the web on one side will be rather difficult to mould perfectly cylindrical, especially if the centre core is not properly temperature controlled. If the end use temperature is elevated, the pitch diameter furthest from the web will tend to be smaller than the pitch diameter at the web (Fig. 8.04).
- Radial ribs which support the rim often reduce accuracy and should be provided only when strictly required due to heavy axial load. Helical gears are often designed this way, even when the resulting axial load is negligible (Fig. 8.05).
- On heavily loaded large bevel gears, thrust load on the tooth crown may become considerable and ribs cannot always be avoided. The basic principles for good rib design apply (Fig. 8.06).
- The same is valid for worm gear drives where the stall torque may produce severe thrust load requiring axial support.

For instance, it has been found on windshield wiper gears that ribbing may be necessary to prevent the worm gear from deflecting away from the worm under stall conditions (Fig. 8.07).

- It must also be understood that any large opening in the web, especially if located close to the teeth, will show up on a gear measuring device and may cause noise or accelerated wear on fast running gears (Fig. 8.08).

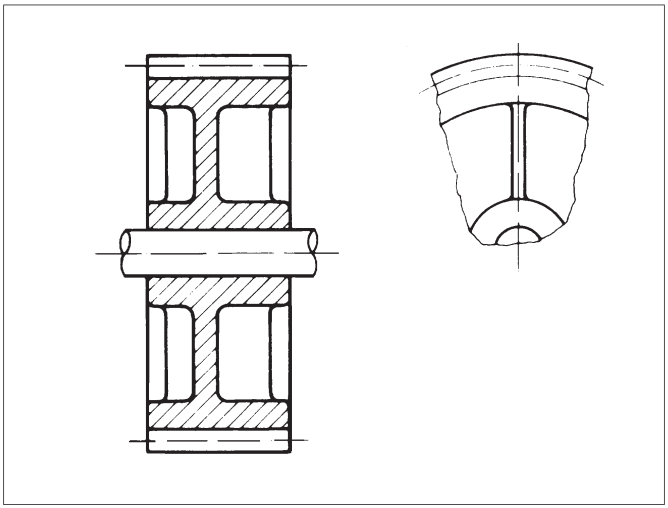

Fig. 8.05 **Effect of radial ribs**

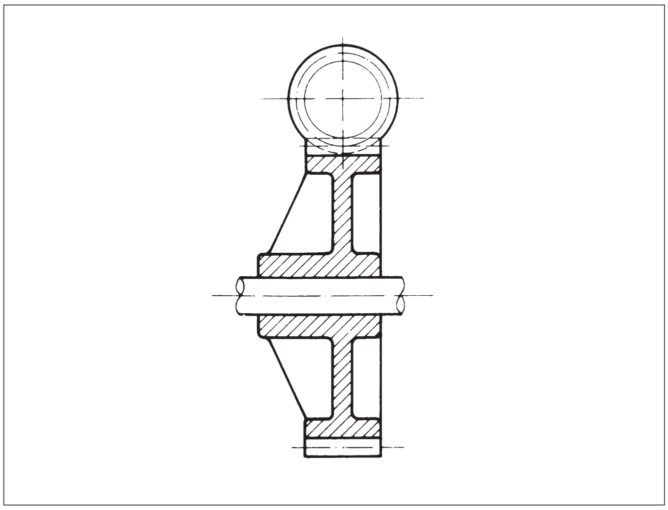

Fig. 8.07 **Ribbed worm take-off gear**

– Fig. 8.09 and 8.10 demonstrate how design and gating sometimes can determine whether a gear fails or performs satisfactorily. Both are almost identical windshield wiper gears moulded onto knurled shafts. The gear on Fig. 8.09 is centre gated and does not create a problem.
– The gear shown in Fig. 8.10 is filled through 3 pin-point gates in the web. In addition, the three holes provided for attaching a metal disc are placed close to the hub. As a result, the thicker hub section is poorly filled and the three weld lines create weak spots, unable to withstand the stresses created by the metal insert and the sharp edges of the knurled surface.

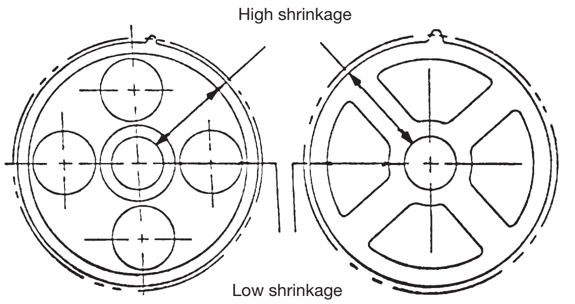

Fig. 8.08 **Holes and ribs in moulded gears**

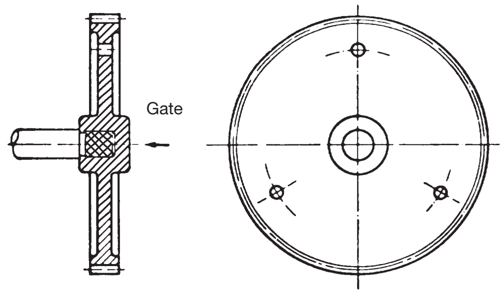

Fig. 8.09 **Centre gated gear**

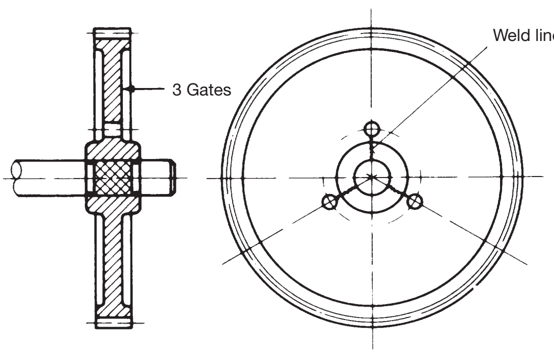

Fig. 8.10 **Web gated gear**

Accuracy and Tolerance Limits

As discussed previously, plastic gears, because of their resilience, can operate with broader tolerances than metal gears. This statement should not be taken too generally. Inaccurate tooth profiles, out-of-roundness and poor tooth surfaces on plastic gears may well account for noise, excessive wear and premature failure. On the other hand, it is useless to prescribe tolerances which are not really necessary or impossible to achieve on a high production basis.

The main problem in producing accurate gears in plastic is, of course, mould shrinkage. The cavity must be cut to allow not only for diametral shrinkage but also the effect of shrinkage on tooth profile on precision gears, this fact must be taken into account, requiring a skillful and experienced tool maker.

With the cavity made correctly to compensate for shrinkage, moulding conditions must be controlled to maintain accuracy. The total deviation form the theoretical tooth profile can be measured by special equipment such as used in the watch industry. An exaggerated tooth profile is shown in Fig. 8.11-a. It includes the measurement of surface marks from the cavity as well as irregularities caused by poor moulding conditions.

In practice, the most commonly used method for checking gear accuracy is a center distance measuring device as shown in Fig. 8.11-b.

The plastic gear meshes with a high precision metal master gear, producing a diagram of the center distance variations as shown in Fig. 8.11-c.

This diagram enables the designer to evaluate the accuracy of the gear and to classify it according to AGMA or DIN specifications.

AGMA specifications No. 390.03 classifies gears into 16 categories, of which class 16 has the highest precision and class 1 the lowest. Moulded gears usually lie between classes 6 and 10 where class 10 requires superior tool making and processing.

Similarly, DIN specification No. 3967 classifies gears into 12 categories, of which class 1 is the most precise and class 2 the least. Moulded gears range between classes 8 and 11 in the DIN categories.

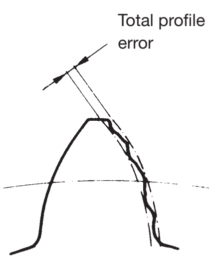

Fig. 8.11-a **Measurement of tooth profile error**

The total error as indicated in Fig. 8.11-c may be due in part to an inaccurate cavity, inadequate part gating or poor processing.

If several curves of a production run are superimposed, as in Fig. 8.11-d, the distance “T” between the highest and the lowest indicates the moulding tolerances.

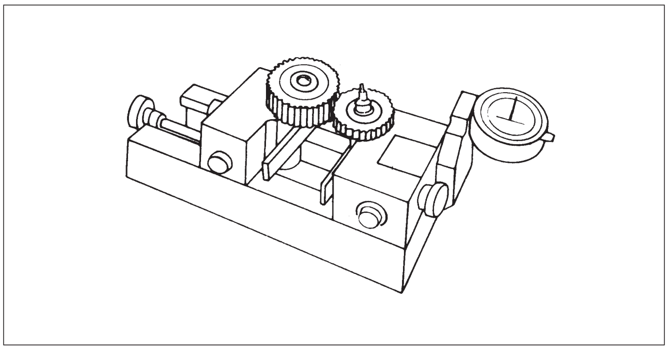

Fig. 8.11-b **Center distance measuring instrument**

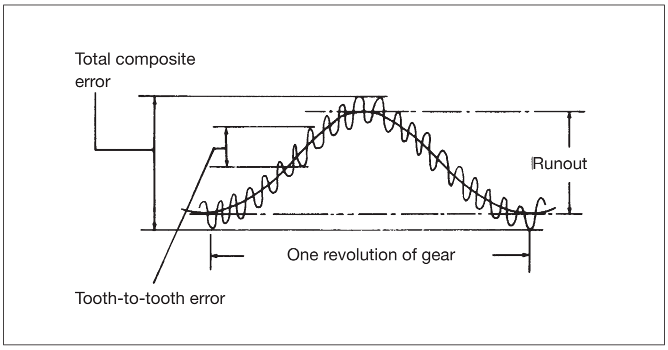

Fig. 8.11-c **Center distance variation diagram**

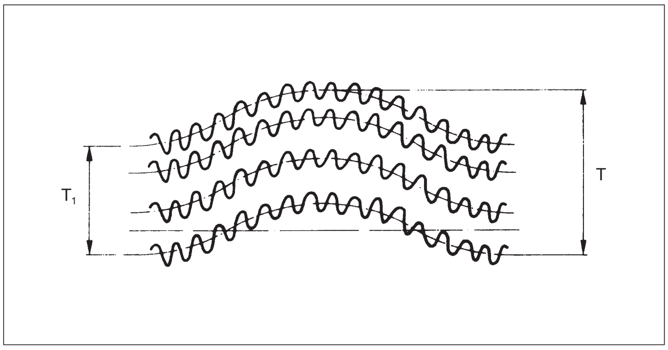

Fig. 8.11-d **Moulding tolerances from center distance diagram**

Backlash and Centre Distances

As shown in Fig. 8.11-e backlash is the tangential clearance between two meshing teeth. Fig. 8.12 provides a suggested range of backlash for a first approach.

It is essential to measure and adjust the correct backlash at operating temperature and under real working conditions. Many gears, even though correctly designed and moulded fail as a result of incorrect backlash at operating conditions.

In particular, the designer must be aware that backlash may be adequate after a device is assembled, but may change in time and under working conditions due to the following reasons:

– Thermal variations.
– Post moulding shrinkage.

If the gear box is moulded in a plastic material as well, the same considerations apply. Centre distance may vary and influence backlash, thus the dimensional stability characteristics of the housing material must be considered.

Increased backlash causes the gears to mesh outside the pitch diameter resulting in higher wear. Insufficient backlash may reduce service life or even cause seizing and rapid part destruction.

It is often easier to determine centre distance after having produced and measured the gears. It must be kept in mind that this procedure may produce more wear as the gears may no longer mesh exactly on the theoretical pitch circle.

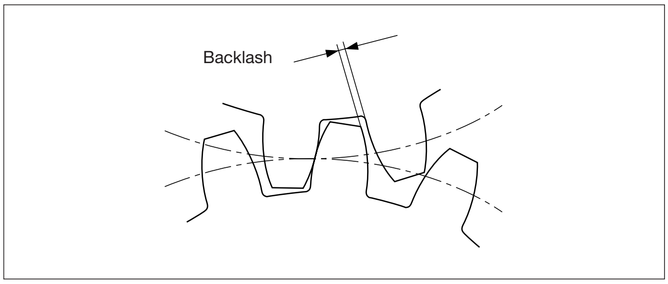

Fig. 8.11-e **Measurement of backlash**

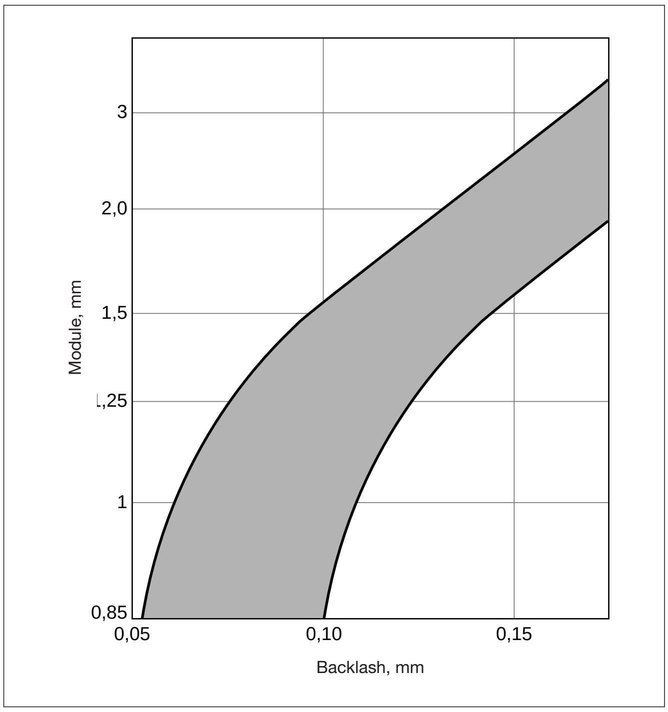

Fig. 8.12 **Suggested backlash for gears of Delrin® and Zytel®**

Mating Material

The coefficient of friction and wear factor of DELRIN® acetal resin on DELRIN® acetal resin are not as good as DELRIN® acetal resin on hardened steel, see Table 7.01. Even so, a great number of commercial applications have entire gear trains made of DELRIN® acetal resin (especially appliances and small precision speed reducers for clocks, timers, and other mechanical devices).

– If two meshing gears are moulded in DELRIN® acetal resin, it does not improve wear to use different grades, as for instance, DELRIN® 100 and DELRIN® 900F or DELRIN® 500CL.

– In many cases wear can, however, be significantly improved by running DELRIN® acetal resin against ZYTEL® nylon resin. This combination is especially effective where long service life is expected, and shows considerable advantage when initial lubrication cannot be tolerated.

– In all cases, where two plastic gears run together, allowances must be made for heat dissipation. Heat dissipation depends on the overall design and requires special consideration when both materials are good thermal insulators.

– If plastic gears run against metals, heat dissipation is much better, and consequently a higher load can be transmitted. Very often, the first pinion of a gear train is cut directly into the fast running motor shaft. Heat transmitted through the shaft from the bearings and electric coils can raise the gear teeth temperature above that which might be expected. The designer should pay particular attention to adequate motor cooling.

– Gear combinations of plastic and metal may perform better and show less wear than plastic on plastic. This is, however, only true if the metal gear has a hardened surface.

Table 8.04 **Suggested mating material choice for spur gears of DELRIN® acetal resin**

Driving Gear	Meshing Gear	
DELRIN® 500	DELRIN® 500	General purposes, reasonable loads, speed and service life, such as clock and counter mechanisms.
DELRIN® 100	DELRIN® 100	Applications requiring high load, fatigue and impact resistance, such as: hand drill presses, some appliances, windshield wiper gears, washing machine drives (esp. reversing). Gears combined with ratchets, springs or couplings.
DELRIN® 500 soft metals	DELRIN® AF	Small mechanism gear drives requiring non slip-stick behaviour and less power loss (e.g., measuring instruments, miniaturized motor speed reducers). This combination does not necessarily give better performance as far as wear is concerned.
Hardened Steel (surface hardness approx 50 Rc)	DELRIN® 100	Excellent for high speed and load, long service life and low wear. Especially when used for the first reduction stage of fast running motors where the pinion is machined into the motor shaft (e.g., appliances, drill presses and other electric hand tools).
Soft steel, non-ferrous metals	DELRIN® 500CL	Combined with soft metals, DELRIN® 500CL gives considerably better results in wear than all other grades. In addition, it has little effect on the metal surfaces. Recommended for moderate load but long service life (e.g., high quality precision gear mechanisms).

Table 8.05 **Suggested mating material choice for gears of ZYTEL® nylon resin**

Driving Gear	Meshing Gear	
ZYTEL® 101L	ZYTEL® 101L	Very common usage in light to moderate load applications.
Hardened Steel	ZYTEL® 101L	Recommended for high speed, high load applications. Best sound and shock absorption. Longest wearing.
ZYTEL® 101L	DELRIN® 100, 500 and 900	Lowest friction and wear compared to either material running against steel 500, 900 or against itself. Highly recommended for moderate duty. Best where no lube is permissible. Either material can be the driver, however, the better dimensional stability of DELRIN® acetal resin makes it the logical choice for the larger gear.

Lubrication

Experience has shown that initial lubrication is effective for a limited time. Units disassembled after completion of their service life showed that all the grease was thrown on the housing walls; hence the gears ran completely dry. Initial lubrication does not allow a high load, it should be considered as an additional safety factor. *It should, however, always be provided as it helps greatly during the run-in period.*

On applications where lubricant cannot be tolerated, the combination of DELRIN® acetal resin and ZYTEL® nylon resin offers great advantages. Even under dry conditions such gear trains run smoothly and with little noise.

Where continuous lubrication of gears in DELRIN® acetal resin and ZYTEL® nylon resin is practical, and where surface pressure on the meshing teeth is not excessive, wear is negligible and service life is determined exclusively by fatigue resistance.

Testing Machined Prototypes

Though it would appear that the easiest way to determine whether a proposed gear will show the expected performance would be to test machined prototypes, results thus obtained must be interpreted with great care. A designer has no guarantee that a subsequently moulded gear will have the same performance characteristics. Therefore no final conclusion can be drawn from test results using machined gears. Making a trial mould is the only safe way to prototype a gear design. It allows not only meaningful tests but also the measurement of shrinkage, tooth profile, pitch diameter and overall accuracy.

It is highly recommended to check tooth quality on a profile projector which enables detection of deviation from the theoretical curve.

Prototype Testing

The importance of adequate testing of injection moulded prototype gears has been emphasized. Here are some guidelines:

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

The following examples further explain the need for meaningful end-use testing.

– Gears under a high load (e.g., in appliances) which operate only intermittently should not be tested in a continuous run, but in cycles which allow the whole device to cool down to room temperature between running periods.
– Infrequently operated, slow-running gears (such as window blinds) can be tested in a continuous run but at the same speed, providing temperature increase on the tooth surfaces remains negligible.
– Other applications like windshield wiper gears reach their maximum working temperature quickly, and operate most of their service life under these conditions. They should therefore be tested on a continuous-run base.

Valuable conclusions can often be drawn from the static torque at which a moulded gear fails. If breaking torque proves to be 8-10 times the operating load, it can usually be taken as an indication that the gear will provide a long service life in use. However, plastic gears often operate very close to the endurance limit, and the above relation should not be considered as valid in all cases.

In any event, backlash must be checked during all tests. Once a gear has failed, it is almost impossible to determine whether incorrect backlash was partially or entirely responsible.

Helical Gear Design

Whenever possible, helical gears should be used in preference to spur gears. Among other advantages they run more smoothly and have less tendency to squeak.

However, they require not only perfect tooth profiles but also exactly matching helix angles. This requirement is sometimes difficult to fulfill, especially if the plastic gear meshes with a metal gear.

Helical gears generate axial thrust which must be considered. It is advisable to use helix angles not greater than 15°. Compared to a spur gear having the same tooth size, a helical gear has slightly improved tooth strength. Since small helix angles are most commonly used, this fact can be neglected when determining the module and it should be considered as an additional safety factor only.

Worm Gear Design

Most machined worm gears are provided with a throated shape which provides a contact line of a certain length on the worm. Because this system cannot easily be applied on moulded plastic gears, a simple helical gear is normally used. Consequently, the load is transmitted on contact points which increases surface pressure, temperature and wear.

Various attempts have been made, aimed at improving wear and increasing power transmission, to change the contact points to contact lines. The following examples of practical applications demonstrate some possibilities along this line.

Fig. 8.13 shows a one-piece moulded worm gear in DELRIN® 100 meshing with a worm in ZYTEL® 101L for a hand-operated device. The undercut resulting from the throated shape amounts to about 4% and can therefore be ejected from the mould without problems. This principle of moulding and ejecting a one-piece worm gear is used in quite a number of applications even though it requires experience and skilled tool making. It is interesting to note that this particular worm with 7 leads cannot be moulded in a two-plate mould with the parting line on the centre.

The lead angle of 31° being greater than the pressure angle (20°), results in an undercut along the parting line. Therefore, the worm must be unscrewed from the mould.

Fig. 8.14 shows a windshield wiper gear produced in a different way. Because of the undercut of about 7% and the rigid structure, ejecting becomes impossible. The tool is therefore provided with 9 radial cores, each of which covers 6 teeth. This procedure produces an excellent worm gear but is limited to single cavity tools. Tooling cost is of course higher.

The worm gear in Fig. 8.15 is again for a windshield wiper drive and based on an intermediate solution. It is composed of a half-throated and a helical gear portion.

The tooth contact takes place on the curved section whereas the helical part merely improves the tooth strength and therefore the stall torque. Even though this solution is not ideal, it nevertheless offers a significant advantage over a simple helical worm gear.

A full throated worm gear is shown in Fig. 8.16 in the shape of a split worm gear. The two halves are designed in such a way that components in the same cavity can be fitted together, centered and the teeth perfectly aligned by means of lugs fitting into corresponding holes (see also Fig. 8.17). Thus, a single cavity provides a complete gear assembly which is held together by means of snap-fits, ultrasonic welding or rivets. As required by production, multi cavities can be added later. The gears can be made as wide as necessary limited only by proper meshing.

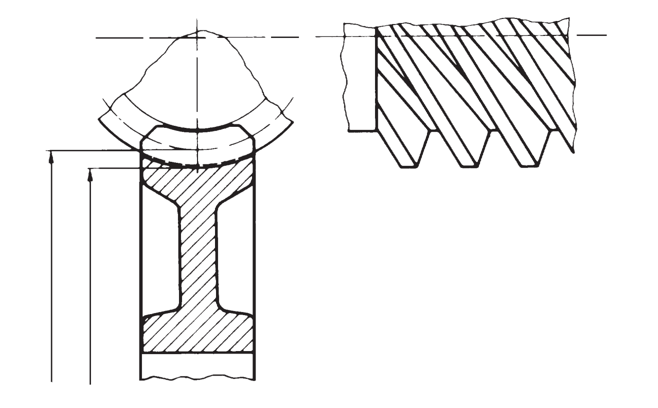

Fig. 8.13 **One piece worm gear**

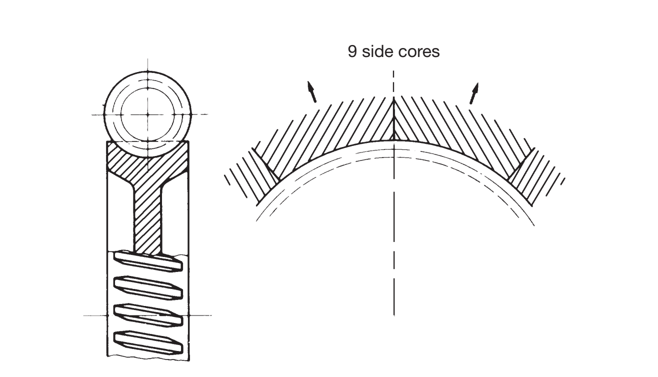

Fig. 8.14 **Side-cored worm gear**

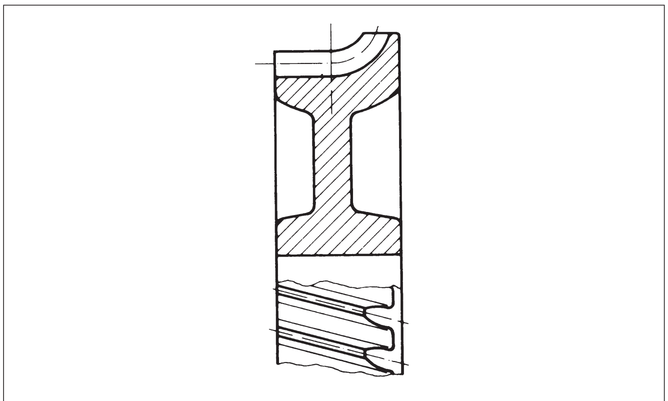

Fig. 8.15 **Half throated worm gear**

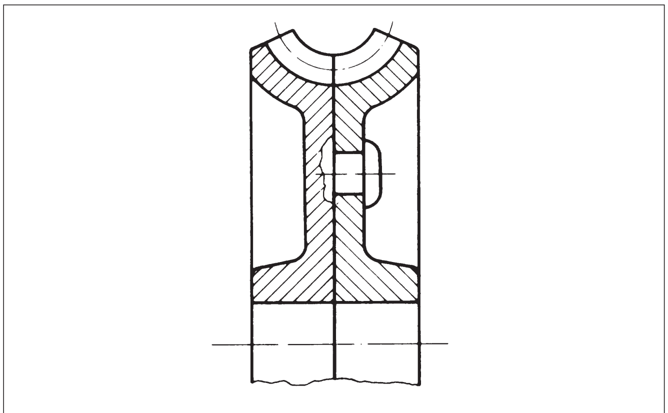

Fig. 8.16 **Split worm gear**

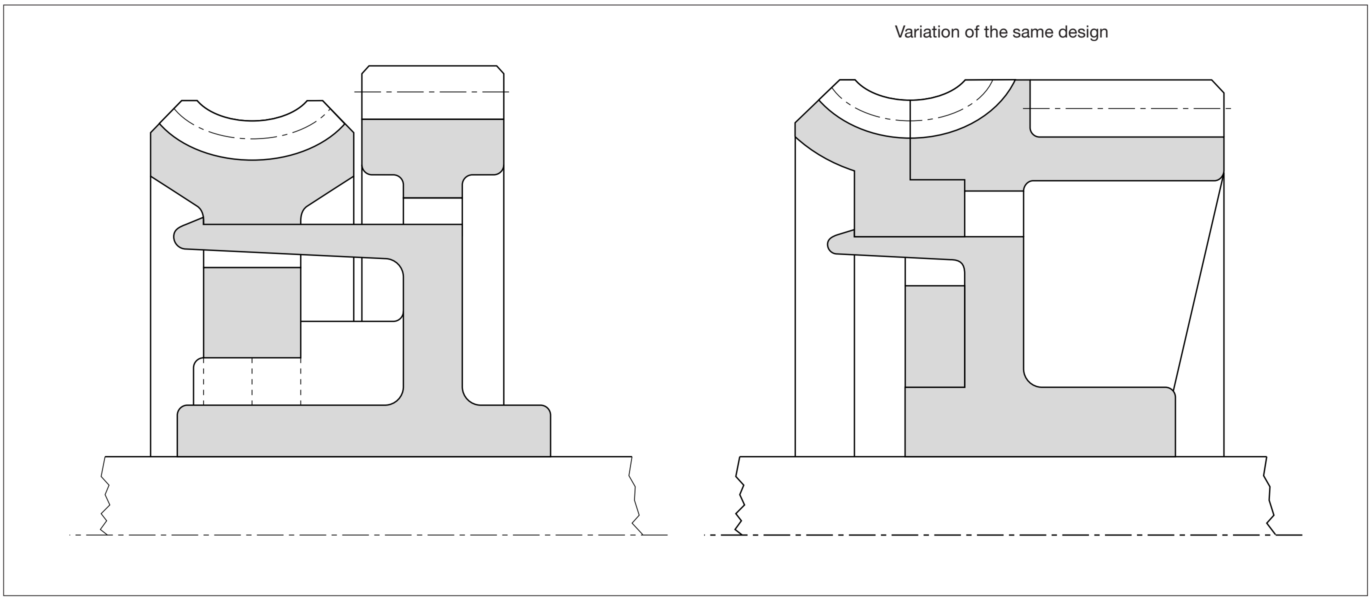

Fig. 8.17 **Snap-fit split worm gear**

This design comes closest to the classic machined metal full throated worm gear, and tooling costs are no higher than for the gear shown in Fig. 8.15. Split worm gears are especially recommended for larger worm diameters, where performance is improved considerably.

The advantage of throated over helical worm gears stems primarily from the load being spread over a larger area of the tooth, resulting in lower localized temperature and reduced bending stress. Tests with the split worm gear show it to be superior to a helical worm gear by a factor of 2-4.

An example of a snap-fit helical gear is shown in Fig. 8.17.

Certain limitations should be kept in mind in the design of these throated worm take-off gears when compared to simple helical gears, as follows:

- Higher tooling cost.
- Requirement of perfect centering of worm and worm gear. Even small displacements cause the load to be carried by only a portion of the tooth width, resulting in increased wear or rapid failure.
- The worm gear drive is more sensitive to discrepancies of the lead angles which must match perfectly.
- The worm and the gear must be assembled in a certain way. If for instance the worm is mounted first into the housing a throated gear can be introduced only in a radial direction, whereas a simple helical gear (or a gear as shown in Fig. 8.13) can be mounted sidewise.

Worm gear drives allow high speed reductions with only two elements. Consequently, they often are used in connection with fast running motors, with the worm cut directly or rolled into the metal shaft.

Since the requirements are quite different in various applications, the performance and limitations of worm gears in DELRIN® acetal resin and ZYTEL® nylon resin will depend upon the specific application.

For instance, a windshield wiper gear may run repeatedly for a considerable length of time and at high temperature. It can, in addition, be submitted to severe stalling torques if the windshield wiper blades are frozen. Since this happens at low temperatures, the gear diameter is smaller due to thermal contraction, causing the teeth to be loaded closer to the tip and thus increasing stress. Often it is this condition which controls the gear design.

On the contrary, an electric car window drive works under normal conditions only a few seconds at a time, with long intervals. Consequently, where is no time for a temperature rise and the gear can thus withstand higher loads. Since the total service life, compared to a wind-shield wiper, is very short, wear is rarely a problem. Many of these power window mechanisms will lock up substantial torque with the window closed. The gear must be strong enough that creep does not cause significant tooth distortion, particularly under closed car summer conditions.

In appliances, requirements are again quite different. The working time is usually indicated on the device and strictly limited to a few minutes at a time. This allows the use of smaller motors which are overloaded, heating up very fast and transmitting temperatures to the shaft and the worm. If the appliance is used as prescribed, temperature may not exceed a reasonable limit. If, however, the devices are used longer or at frequent intervals, temperature can reach a level at which high wear and premature failure can occur.

These examples demonstrate the necessity of carefully defining the expected working conditions, and of designing and dimensioning the worm and gear accordingly.

In addition to the limitations mentioned previously, other factors to be carefully considered are:

– Metal worms cut directly into motor shafts usually have very small diameters. Unless supported at both ends, overload and stalling torques cause them to bend, and lead to poor meshing.
– Under the same conditions, insufficiently supported plastic worm gears are deflected axially, with the same results.
– In cases where worms are cut into small-diameter motor shafts, tooth size is considerably limited. Actually, many worm gear drives, especially on appliances, work only satisfactorily as long as the initial lubrication is efficient. With regard to the relatively short total operating lifetime, performance may nevertheless be acceptable.

Even though initial lubrication has limited efficiency in time, it is highly recommended for all worm gear drives, since friction is the main problem. Moreover, whenever possible, proper steps should be taken to keep the lubricant on the teeth. It is also advisable to choose a grease which becomes sufficiently liquid at working temperature to flow and thus flow back onto the teeth. In cases where severe stalling torques are applied on the gear, bending stresses and shear strength must be checked as well. As noted previously, load is concentrated on a very small area of helical take-off gears which causes uneven stress distribution across the width. Consequently, the tooth face *f* used for determining stall torque stresses should not be more than approximately two times the tooth size. For a metric module of 1, f = 4 mm.

It is advisable that bending stresses should not exceed approximately 30 MPa at room temperature (accounting for a safety factor of S = 1,5).

Some manufacture machine worm take-off gears from moulded blanks. If there is a valid reason for making gears this way, it is important that the tooth spaces be moulded in the blank in order to prevent voids in the rim section. Many plastic worm gears fail due to tiny voids in the highly stressed root area of the teeth because the rim was moulded solid. (The same is valid for other types of gears.)

The majority of worm gear applications use single threaded worms with helical worm gears. The teeth of the helical gear are weaker than the threads of the worm, thus output will be limited by the torque capacity of the gear.
A liberal safety factor (3-5) should be applied to take into account the stress concentration due to theoretical point contact as well as the high rubbing velocity. With ZYTEL® nylon resin as the worm and DELRIN® acetal resin as the gear, heat dissipation is limiting, as both materials are not good conductors of heat. Thus, it is recommended rubbing velocities be less than 0,125 m/s. With a steel worm, heat dissipation is markedly improved, and rubbing velocities as high as 1,25 m/s can be tolerated with initial lubrication. With continuous lubrication or intermittent operation, speeds as high as 2,5 m/s are possible.

The equation used to determine rubbing velocity is:

$$v = \frac{0{,}001\ \pi\ d\ \omega}{60\ cos(\alpha)} \qquad \text{(m/s)}$$

d = worm pitch diameter (mm)
ω = worm speed (rpm)
α = lead angle (Fig. 8.18)

Table 8.06 **Worm gear mating material**

Worm Material	Gear Material	Possible Applications
Soft steel (machined DELRIN® 500CL or rolled)	DELRIN® 500CL	Excellent wear behaviour; can be considered for small devices (e.g. appliances, counter, small high quality mechanical speed reducers).
Soft steel and hardened steel	DELRIN® 100	Less wear resistant, improved fatigue and impact strength, for high stalling torques (e.g. windshield wipers, car window mechanisms, heavily loaded appliances like meat grinders where impact load must be expected). Worms in hardened steel give far better wear results.
Non ferrous metals (brass, zinc-alloys)	DELRIN® 500CL	DELRIN® 500CL has proven to give superior wear results compared to all other DELRIN® acetal resin grades even though heat build-up is not better. (Used in speedometers, counters and other small devices).
ZYTEL® 101L (66 Nylon resin)	DELRIN® 500 DELRIN® 100	Excellent for hand-operated or intermittent use, low speed devices (e.g. window blinds; car window mechanisms; continuously running, small-speed reducers where load is negligible, such as speedometers, counters). Very good dry-running behaviour.
DELRIN® 500	DELRIN® 500	Should be avoided on the basis of unfavourable wear behaviour and high coefficient of friction. It is nevertheless used in many small slow-running mechanism where load is extremely low.

Mating Material

Generally speaking, all worm gear speed reducers are inefficient due to high sliding velocity, which converts a large percentage of the power into heat. Therefore, it is important to choose mating materials which have low wear and friction. Along this line, a worm in ZYTEL® 101 running against a gear in DELRIN® acetal resin is a good combination. Because heat dissipation is poor, this combination is limited to light duty applications.
The can opener shown in Fig. 8.18 is a good example of a commercial design using this combination of materials.

The motor speed of 4000 rpm is reduced in the first stage with a pinion and an internal gear before driving the worm in ZYTEL® 101. Working cycles are, however, so short that no significant heat build-up can take place.

Bevel Gear Design

The formula for the calculation of the bending stress in a spur gear has to be corrected to the following:

σ = {r / (r-f)} F / (y M f)
r = pitch radius (r = 0,5 d_{max}, see Fig. 8.06)
f = tooth width

With plastic materials, support of the rim is very important and supporting ribs are almost always necessary.

Fillet Radius

Most gear materials are not sensitive, including DELRIN® acetal resin and ZYTEL® nylon resin. Thus, the importance of proper fillet radius cannot be over emphasized. Standard fillets have proved satisfactory in most applications. It has been found that the use of a full rounded fillet will increase the operating life of gears of DELRIN® acetal resin approximately 20% under continuously lubricated conditions. Full rounded fillets may also prove advantageous where shock or high impact loads are encountered.

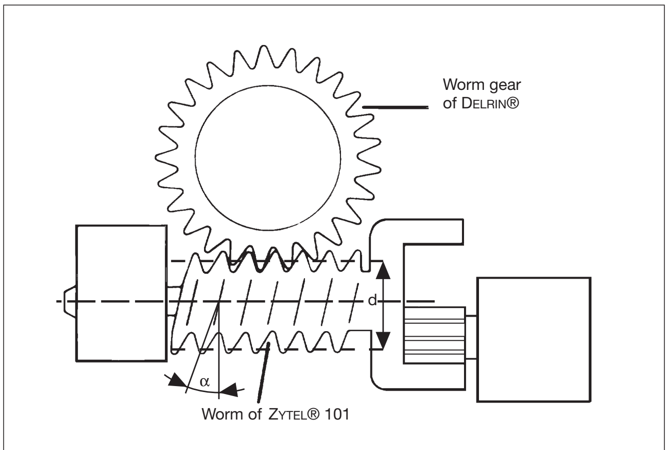

Fig. 8.18 **Can opener with worm drive**

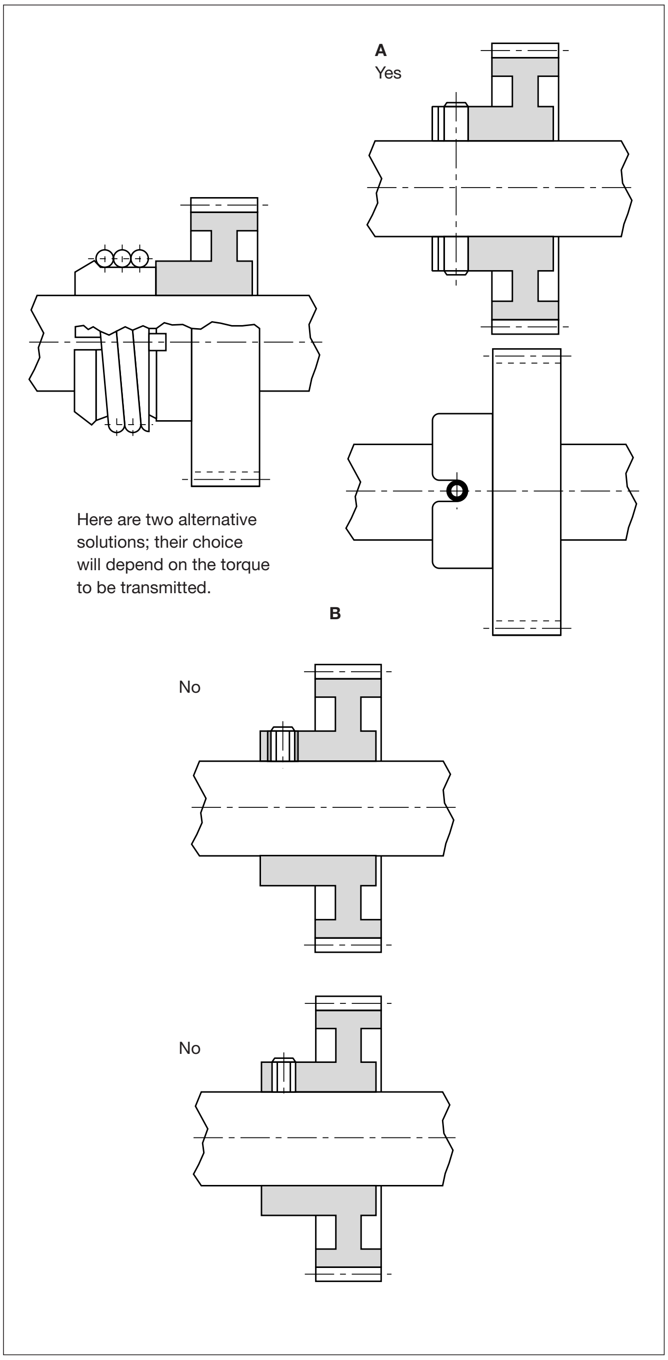

Fig. 8.19 **Alternatives for grub screws**

Methods of Fastening

The involute spline (Fig. 8.19 A) is by far the best method of fastening plastic gears to shafts.

Keys and set screws, although they have been used successfully, should be avoided as they require an unbalanced geometry in the hub. If set screws are used, they must bottom in a recess in the shaft. Interference fits can be used providing the torque requirements are low. Stress relaxation of the plastic material may result in slippage. Knurling the shaft can be helpful. Also grub screws (Fig. 8.19 B) should not be used to transmit even low torques. Plastic may break during assembly or creep in use.

The use of moulded-in inserts has been successful in gears of DELRIN® acetal resin and ZYTEL® nylon resin. The most common insert of this type is a knurled shaft.

Circumferential grooves in the knurled area can be used to prevent axial movement with helical, worm or bevel gears. Stamped and die cast metal inserts have also been used successfully. Engine timing sprockets mentioned previously use a die cast aluminium insert with incomplete teeth. ZYTEL® nylon resins is moulded over the insert to form the teeth. This is a good example of using the best properties of both materials to achieve a dimensionally stable, low cost, improved timing gear. Screw machine inserts have been used. It is important that materials with sufficiently high elongation be selected for use with moulded-in metal inserts so that the residual stress resulting from mould shrinkage will not result in stress cracking around the insert. Any of the ZYTEL® nylon resins are suitable in this regard. DELRIN® acetal resins have generally lower elongation than ZYTEL® nylon resins and have higher creep resistance, thus latent cracking can occur over moulded-in inserts with resins such as DELRIN® 500 and 900. However, DELRIN® 100ST has very high elongation and is recommended for use with moulded-in inserts. Inserts can be pressed or sonically inserted to reduce residual stress.

Stampings have been employed in the form of plates fastened to the web of the gear with screws or rivets, or by sonic staking.

With any fastening method, it is important to avoid stress risers. Fillets on the splines, inserts, etc., are extremely important.

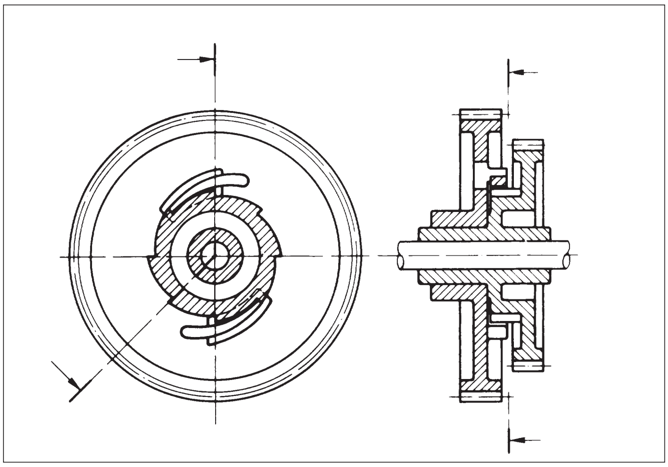

Fig. 8.20 **Ratchet and gear combined**

Combined Functions – Design Examples

As mentioned previously, plastic gears offer large economic advantages over metal gears, and the greatest cost savings can come from combining an almost unlimited number of elements and functions in one single part.

Fig. 8.20–8.25 demonstrate a few design examples along this line:

– In this example (Fig. 8.20), a gear in DELRIN® acetal resin is provided with moulded-in springs acting on a ratchet wheel in ZYTEL® 101 nylon resin, which in turn is combined with another gear. There exist many various types of ratchets in DELRIN® acetal resin which function properly, providing the springs are not held in the loaded position for a long period of time.

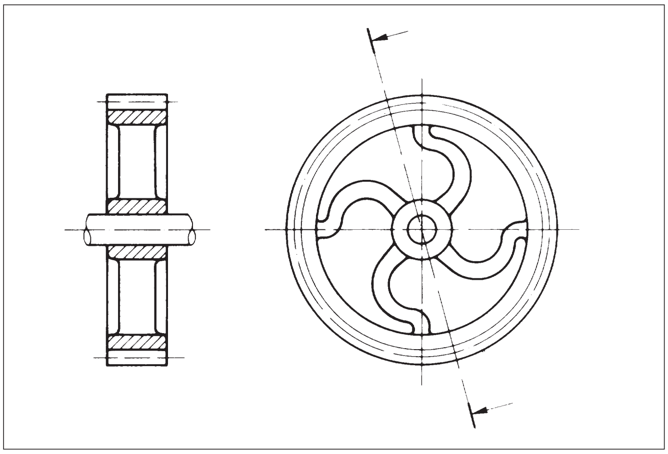

Fig. 8.21 **Impact resistant web**

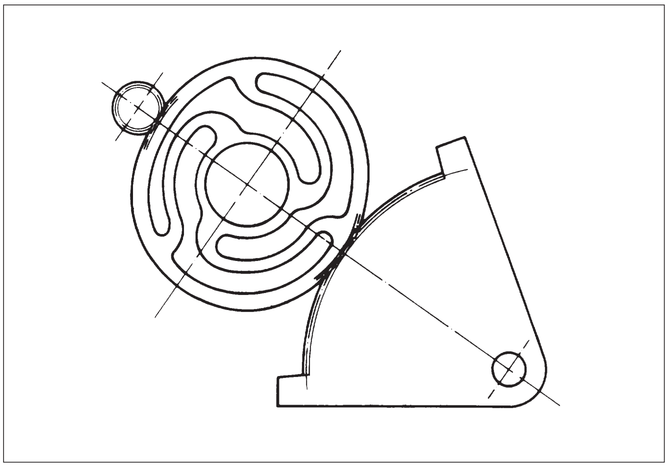

Fig. 8.22 **Backlash-free gear**

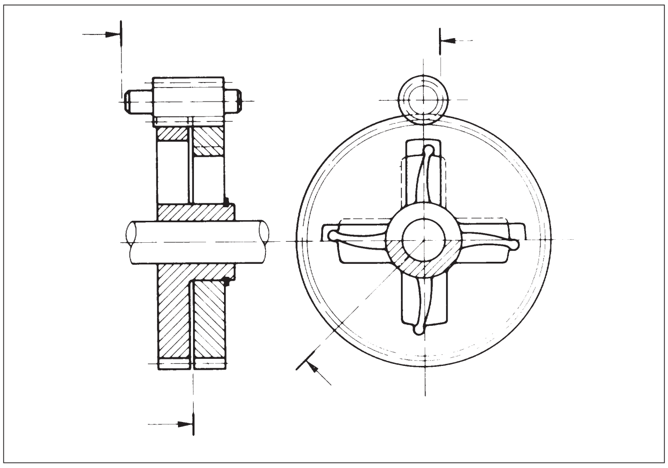

Fig. 8.23 **Backlash-free gear**

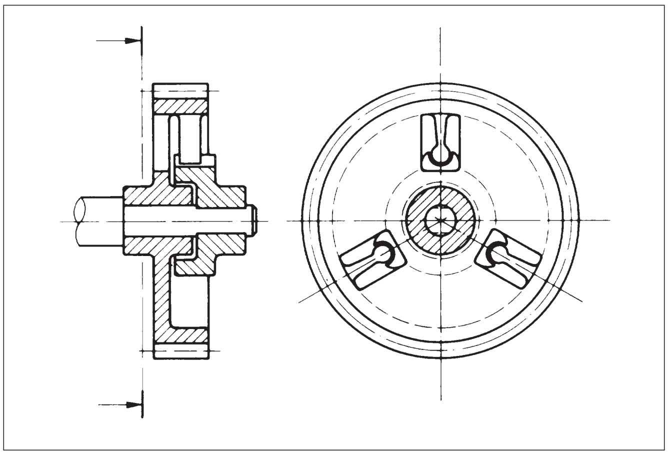

Fig. 8.24 **Torque limiting gear**

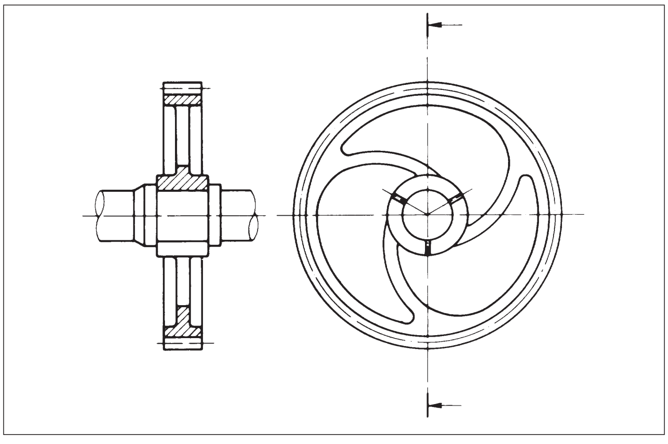

Fig. 8.25 **Gear with sliding coupling**

– In many cases, it is highly desirable to protect the teeth against impact load. This can be achieved, as is shown in Fig. 8.21, in connecting the hub and the rim by properly dimensioned flexible elements. Sometimes, this solution also is used on printing wheels in order to obtain consistent printing results without the need for precision tolerances.

– Fig. 8.22 is a backlash-free adjusting arrangement for an automobile clock. The motion is transmitted from the pinion to the segment by means of a flexible gear. When assembled, the gear is ovalized, thus applying a certain preload onto the pinion and the segment. As the transmitted torque is very small, stress relaxation does not jeopardize proper function. In addition, each time the mechanism is actuated, the oval gear moves into another position, thus increasing the preload and allowing the previously loaded section to recover. As an unusual additional feature, the segment is provided with two stops for limiting its movement. When a stop comes into contact with the gear, the pinion can be turned further without causing any damage to the system because the teeth slip over the flexing gear.

– Fig. 8.23 is another design suggestion for a backlash-free motion transmission between two gears. The main gear is equipped with moulded-in springs fitting into corresponding slots on the second gear. When assembled with the pinion, the two tooth crowns are slightly offset, causing the springs to be loaded and thus suppress any backlash. Here again, stress relaxation causes the force to decrease. The principle is adequate for only small torques such as those found on instrument dials or clock adjusting mechanisms.

– Torque limiting devices are quite often very useful for plastic gears in order to prevent tooth damage when overloading occurs (for instance on high torque transmissions like meat grinders, can-openers and hand drill presses). Fig. 8.24 shows one solution out of many other possible designs. It is essential that the springs do not remain accidentally in the loaded position. In the example shown, this is achieved by providing three pivoting springs.

– For specific requirements, it is also possible to combine gears with sliding couplings. The example shown in Fig. 8.25 is a gear in DELRIN® acetal resin snapped onto a shaft in ZYTEL® 101, in which the split hub acts as a coupling for a small dial setting device. If, as in this case, the required torque is very low, stress relaxation in the springs will not endanger perfect functioning for a sufficient length of service life. If, on the contrary, a constant torque must be transmitted for a long period of time, an additional metal spring around the hub would be required to keep the force constant.

When to use DELRIN® Acetal Resin or ZYTEL® Nylon Resin

ZYTEL® nylon resin and DELRIN® acetal resin are excellent gear materials, used extensively in a variety of applications. The choice of one over the other may at first seem unclear, but as one examines the specific requirements of the application, it becomes relatively easy. Although the two materials are similar in many ways, they have distinct property differences, and it is these differences upon which the selection is made. Some guidelines are as follows:

ZYTEL® Nylon Resin
- Highest end-use temperature
- Max. impact and shock absorption
- Insert moulding
- Max. abrasion resistance
- Better resistance to weak acids and bases
- Quieter running

DELRIN® Acetal Resin
- Best dimensional stability
- Integrally moulded springs
- Running against soft metals
- Low moisture absorption
- Best resistance to solvents
- Good stain resistance
- Stiffer and stronger in higher humidity environment

As previously pointed out, running DELRIN® acetal resin and ZYTEL® nylon resin against each other results in lower wear and friction than either material running against steel (not always true when high loads are encountered and heat dissipation is controlling). Some designers have used this combination in developing new, more efficient gear systems.

When properties of DELRIN® acetal resin are needed, DELRIN® 100 is the preferred gear material. As previously stated, DELRIN® 100 outperforms DELRIN® 500 by about 40%. DELRIN® 100 is the most viscous in the melt state, and cannot always be used in hard to fill moulds. DELRIN® 500 and 900 have been used successfully in many such cases.

When ZYTEL® nylon resin is the chosen material, ZYTEL® 101L is the most common material used. ZYTEL® 103HSL, a heat stabilized version of ZYTEL® 101L, should be specified if the service life and end use temperature are high.

Glass reinforced versions of either material should be avoided. The glass fibres are very abrasive and the wear rate of both the plastic gear and the mating gear will be high. Gears which operate for extremely short periods of time on an intermittent basis have been used with glass reinforcement to improve stiffness, strength or dimensional stability. Very careful testing is mandatory.
The moulding conditions must be controlled carefully, not only for the usual purpose of maintaining gear accuracy, but also because glass reinforced resins will exhibit large differences in surface appearance with changes in moulding conditions, particularly mould temperature. It is possible to vary mould temperature without changing dimensions, by compensating through adjustments in other process variables. Thus, establish surface smoothness specifications to be sure the type of gear surface tested is reproduced in mass production.