In many instances, plastic materials perform markedly better than do metals—especially in gears. Read on to learn the details that will allow you to make the best choice for your operation.

Plastic Materials

Properly selected plastic materials offer better performance than metals when all or some of the following requirements must be satisfied:

- Low maintenance
- Wear resistance when running dry
- Low noise
- Vibration dampening
- Corrosion-proof
- Low inertia due to low rotating mass, light weight
- Low manufacturing cost

Polyamides can be engineered toward one or more requirements, as is shown by the following listing of the plastic materials most commonly used for manufacturing gears:

1) PA 6 (Polyamide 6): this material is wear resistant and absorbs impact even under rough conditions, but it is less suitable for small precision gears.
2) PA 66 (Polyamide 66): compared to PA 6, this extruded polyamide offers better wear resistance (except against mating surfaces of high quality), absorbs less moisture and is dimensionally more stable, but it is also less suitable for small precision gears.
3) PA 6 G (Cast Polyamide 6): the high degree of crystallization makes PA 6 G especially wear resistant.
4) Calamid 612/612-Fe® (cast PA 6/12): this polyamide is engineered toward toughness against shock loads, with wear resistance similar to PA 6 G. (Calaumid is a Timco exclusive)
5) Calamid 1200/1200-Fe® (cast PA 12): a lower degree of moisture absorption gives better dimensional stability. It has excellent wear resistance and withstands high shock loads.
6) PA 6 G + Oil (Cast Polyamide 6 + Oil): the addition of lubricating oil into the PA 6 G provides very good dry running and wear resistant properties.
7) POM-C (Polyacetal-C): this Acetal absorbs very little moisture, which makes it suitable for precision gears, but it needs continuous lubrication under high loads.
8) UHMW-PE (ultra-high molecular weight polyethylene): PE absorbs no moisture, is dimensionally stable, resistant against chemicals, and dampens vibrations, but it is suitable only for low loads.

There are limitations of speed and load for plastic gears, of course. Metal gears usually operate well within the temperature limits of the material. The design of plastic gears, however, must always take into consideration the increase in temperature which is caused by friction, pressure, and speed. The following paragraphs will show how one’s design can safely stay within these limits.

Quality of Mating Gear:
The gear which mates with a plastic gear can also be of plastic, but only in slow and lightly loaded applications. Otherwise,
case-hardened steel is the best mating material because it dissipates friction heat quickly. The harder the steel, the slower the wear on the gear wheel and pinion. We recommend a surface quality of Rₐ = 8 to 10 μm, both for dry and lubricated applications.

The driving pinion is always subject to greater wear, therefore the pinion should always be of the more wear resistant material (e.g., steel pinion/plastic gear wheel; PA pinion/POM gear wheel).

Lubrication:
Automatic lubrication will improve the break-in performance and the overall lifetime considerably. Plastics which are engineered with oil additives—such as PA 6 G + Oil—provide much longer lifetimes than all other plastics. Continuous oil lubrication results in better friction heat dissipation and longer useful lifetime, or higher load bearing capacity. With grease lubrication, the circumferential speed should not be higher than 5 m/sec so that the gear does not sling off the grease. We do not recommend that you lubricate polyamides with water because the material tends to absorb water.

Noise Generation:
The excellent noise dampening properties of plastics result in a quieter-running gear. The graph below shows noise generation in dB of steel against steel “a” and of steel against plastic “b” at increasing speeds. The difference is as much as 9 dB. Steel against steel generates up to three times as much noise than does steel against plastic. Noise equals wear: When we replaced the noisy steel pinion gears in the drying section of a paper mill with steel against plastic, the noise disappeared, and one could stand a nickel upright on the housing of the running gear.

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UFE helped Hewlett-Packard print a best-seller.

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the only supplier who could meet their AGMA 11 specs – verified by an independent lab.
We’ve turned out millions of gears for H-P, at zero PPM. The story at Hewlett-Packard
has a happy ending, thanks to a little help from UFE.
contact line may be longer and several teeth engage simultaneously, but the load is distributed unevenly and the teeth get deformed. Helical plastic gears are calculated like metal gears, using a spur gear of similar size. $\beta = 10^\circ - 20^\circ$ is the preferred helix angle.

**Backlash and Clearance:**

The material-specific designing of backlash and clearance is important due to the high thermal expansion rates of plastic materials. A minimum backlash must be guaranteed. We recommend a minimum face clearance of about $0.04 \times$ modul. Thus, the backlash at start-up is:

\[
S_c = S_{co} + 2 \cdot l \cdot \sin(\alpha) \cdot (k_\alpha \cdot k_f) \ [\text{mm}]
\]

$S_{co}$ = minimum face clearance in mm

$l$ = total distance in the plastic material between the rotating axes in mm

$\alpha$ = pressure angle

$k_\alpha$ = factor of elongation

$k_f$ = factor for moisture absorption

We recommend an initial clearance of $0.3 \times$ modul. This allows for temperature variations of $\pm 20^\circ\text{C}$ as well as for tolerance variations.

**Power Transmission:**

Key and keyway connections are used for plastic gears, too.

The load bearing side of the keyway must be dimensioned so that the permissible contact pressure of the plastic is not exceeded. This maximum contact pressure is:

\[
P_F = \frac{M_d \cdot 10^3}{i \cdot r_m \cdot h \cdot b} \ [\text{MPa}]
\]

$M_d$ = torque in Nm

$i$ = number of keys

$r_m$ = radius from the shaft center to the center of the key way in the plastic, in mm

$h$ = height of the key in the plastic gear which engages the plastic in mm

$b$ = length of key in the plastic gear which engages the plastic in mm

The resulting value must be compared with Diagram 1. It may not exceed the values which are permissible for the specific plastic material. However, this value does not include a safety factor for shock loads or other reserves. We recommend a safety factor of between 1.5 and 4. Keyways in plastic should be machined with a radius in the corners, if at all possible,
because of the notch sensitivity of the plastic material.

There are several other designs for connecting a steel hub to the plastic gear. A press-fitted steel hub secured with bolts allows for keyways and keys designed according to traditional guidelines for metals.

Another very good alternative design is the material Calaumid 612 Fe® or Calaumid 1200 Fe®. The metallic core is securely connected with the plastic gear. Keyways and keys can again be designed according to traditional guidelines for metal.

Calculating Thermoplastic Gears

Thermoplastic gears fail prematurely for mostly the same reasons, and with the same symptoms, as do gears made of steel. Therefore, the calculation of plastic gears does not use different methods. The only difference is the inclusion of material-specific properties in the form of correction factors. The calculations for torque, circumferential force, and circumferential speed use the formulas which apply to steel gears.

Estimated Tooth Temperature in Continuous Operation:

Temperature plays a major role in establishing the load capacity of plastic gears. The tooth body temperature sets the permissible loading and deformation of the tooth base. The tooth face temperature allows an approximation of the rate of wear. An accurate determination of both temperatures is difficult. The heat transmission coefficient on a rotating gear wheel can only be estimated approximately. Thus, the calculation of the tooth face temperature may sometimes result in a very high number, higher even than the melting temperature of the plastic.

However, we have not yet observed a melting of plastic gear tooth faces.

The following formula presents an estimate of the temperature of the gear. Because it is an estimate, the calculated temperature may sometimes result in a higher value than the actual temperature. This is used as an extra safety factor. The calculations below take into account the friction heat, the rate of heat dissipation from the gear, and from the gear housing to the outside. We arrived at the following formula:

\[ \theta_{1,2} = \theta_U + P \cdot \mu \cdot 136 \cdot \frac{i+1}{z_{1,2} + 5i} \left( \frac{k_2 \cdot 17100}{b \cdot z_{1,2} \cdot (v \cdot m)^{3/4} A} \right) + 7.33 \cdot \frac{k_1}{A} \]°C

Where:

Index 1 for the pinion
Index 2 for the wheel
ϑ_U = ambient temperatures in °C

P = power in kW

µ = coefficient of friction

z = teeth

i = transmission ratio \( z_1/z_2 \) with \( z_1 \) = number of teeth in pinion

b = width of the tooth face in mm

v = circumferential speed in m/sec

m = module in mm

A = surface of the gear box in mm²

k_2 = material-related factor

k_3 = gear-related factor in m²K/W

For factor \( k_2 \), the following must be included depending on the temperature to be calculated:

**Factor \( k_2 \):**

\( k_2 = 7 \) for mating components steel/plastic

\( k_2 = 10 \) for mating components plastic/plastic

\( k_2 = 0 \) in the case of oil lubrication

\( k_2 = 0 \) at \( v \leq 1 \) m/sec

Factor \( k_3 \) and the coefficient of friction \( \mu \):

\( k_3 = 0 \) for completely open gear m²K/W

\( k_3 = 0.043 \) to 0.129 for partially open gear in m²K/W

\( k_3 = 0.172 \) for closed gear in m²K/W

\( \mu = 0.04 \) for gears with permanent lubrication

\( \mu = 0.07 \) for gears with oil mist lubrication

\( \mu = 0.09 \) for gears with automatic lubrication

\( \mu = 0.2 \) PA/steel

\( \mu = 0.4 \) PA/PA

\( \mu = 0.25 \) PA/POM

\( \mu = 0.18 \) POM/Stahl

\( \mu = 0.2 \) POM/POM

**Tooth Temperature in Intermittent Operation:**

The load capacity of plastic gears increases if they operate intermittently because less friction heat is generated. The relative duty cycle \( ED \) is considered in the following equation by a correction factor \( f \). This relative duty cycle is defined as a percentage of the load time \( t \) and the overall cycle time \( T \).
A total cycle time $T = 75$ min has been established for thermoplastic gears. The sum of all individual operating times under load during these 75 minutes is the total operating time $t$. Using the value which resulted from this calculation, the correction factor $f$ can be determined from Diagram 2.

Please note that each operating time which exceeds 75 minutes, even if this happens only once, must be considered as continuous operation. Considering the correction factor $f$, we can calculate the tooth face temperature and the tooth body temperature:

$$\theta_{1,2} = 9_i + P \cdot \mu \cdot f \cdot 136 \cdot \frac{i+1}{z_{1,2} + 5i} \cdot \left( \frac{k_2 \cdot 17100}{b \cdot z_{1,2} \cdot (v \cdot m)^{3/4}} + 7.33 \cdot \frac{k_1}{A} \right) [^\circ C]$$

The values for $k_2$ and $k_3$, and $\sigma_F$ and $\sigma_{zul}$ and the coefficient of friction $\mu$ can be taken from the preceding paragraph ("Continuous Operation").
Calculation of the Root Strength of the Teeth:

If the tooth root stress $\sigma_f$ under load exceeds the permissible stress $\sigma_{zul}$, breakage of teeth is likely. Therefore the tooth root stress $\sigma_f$ must be calculated and compared with the permissible values. The calculation must be made separately for either part where both pinion and gear are made of plastic. The tooth root stress is:

$$\sigma_f = \frac{F_U}{b \cdot m} \cdot K_B \cdot Y_F \cdot Y_\beta \cdot Y_\varepsilon \text{ [MPa]}$$

$F_U$ = circumferential force in N
$b$ = tooth width in mm (use smaller width + m for the wider gear when widths of pinion and gear are different.)
$m$ = modul in mm
$K_B$ = operating factor for variations in drive operation
$Y_F$ = tooth shape factor from Diagram 3
$Y_\beta$ = helix factor to allow for the increased load capacity of helical gears

For plastic gears, use $Y_\beta = 1.0$
$Y_\varepsilon$ = contact ratio factor from table 1; $Y_\varepsilon = 1/\varepsilon_\alpha$ and $\varepsilon_\alpha = \varepsilon_\alpha 1 + \varepsilon_\alpha 2$

For profile-corrected gears, the factor $Y_\varepsilon$ must be adjusted accordingly:

$$\varepsilon_\alpha = \frac{Z_1}{2 \cdot \pi} \cdot (\tan \alpha_{1} - \tan \alpha_{A1})$$

There are many things to consider when placing an order with a gear manufacturer, such as trust, confidence and level of integrity. Spending money correctly is also a primary concern, but so is getting a reliable product on-time that fits YOUR particular gear needs.

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The value $\tan \alpha_{E1}$ is dependent on correction value: 

$$D_1 = \frac{d_{K1}}{d_{G2}}$$

Where:

- $d_{K1}$ = outside diameter of pinion
- $d_{G2}$ = base diameter of large wheel

And:

$$\tan \alpha_{A1} = \tan \alpha_{sw} \cdot \left[1 + \frac{Z_2}{Z_1}\right] \cdot \frac{Z_2}{Z_1} \cdot \tan \alpha_{A2}$$

The value $\tan \alpha_{A2}$ is dependent on the correction value

$$D_2 = \frac{d_{K2}}{d_{G1}}$$

Where:

- $d_{K2}$ = outside diameter of pinion
- $d_{G1}$ = base diameter of pinion

The values of $\tan \alpha_{E1}$ and $\tan \alpha_{A2}$ can be taken from Diagram 5. The effective pressure angles $\alpha_{tw}$ and $\tan \alpha_{sw}$ are calculated from the profile correction $x_{1,2}$ and the number of teeth $z_{1,2}$ where Index 1 stands for the pinion and Index 2 for the large gear. The effective pressure angles for spur gears are shown in Diagram 4.

**Calculation of Gear Face Load Capacity:**

Excessive gear face loading may cause pitting or excessive wear. This wear is concentrated at the root and crest of the tooth. It leads to a change in the tooth profile and, consequently, to uneven motion transmission.

The tooth face pressure $\sigma$ must be checked to determine whether it is still within the permissible face pressure $\sigma_{H}$. This will preclude premature failure due to pitting and/or wear.

For plastic gears the material factor $Z_M$ is added to the standard formula in order to consider the plastic material:

$$\sigma_{H} = \sqrt{\frac{K_h \cdot Z_1 + Z_2}{b \cdot d_0 \cdot z_1}} \cdot K_h \cdot Z_e \cdot Z_{A1} \cdot Z_M \ [\text{MPa}]$$
\[ Z_M = \sqrt{0,38 \cdot E'} \quad \text{and} \quad E' = \frac{E_1 \cdot E_2}{E_1 \cdot E} \]

\( E_1 = \) modulus of elasticity for the pinion material
\( E_2 = \) modulus of elasticity of gear wheel material

This takes into consideration the different moduli of different materials of pinion and gear. The factor \( Z_M \) for mating of plastic with steel may be taken from diagram 6.

The mating of two plastic gear wheels results in \( Z_{M(K/K)} \)

\[ Z_{M(K/K)} = \frac{1}{\sqrt{2}} \cdot Z_{M(K/S)} \]

If the gear wheel and pinion are made of different plastics, the factor \( Z_{M(K/S)} \) which is valid for the softer plastic should be used.

**Safety Factors:**
We recommend the following minimum safety factors,
Please contact Timco concerning the diagrams for permissible tooth root stresses and tooth face pressures (contact information at end of article).

**Plastic Materials Engineered for Gear Manufacturing:**
The three grades of polyamide listed at the beginning of this article are particularly applicable for manufacturing gears:

1) **PA 6 G + Oil:** a polyamide with an additive of lubricating oil which makes it most suitable for dry-running applications.
2) **Calaumid 612/612-Fe®:** a tough polyamide for shock-load applications. It has a metal core which allows the use of standard steel keyways and keys.
3) **Calaumid 1200/1200-Fe®:** a pure PA-12 tough-hard polyamide with less moisture absorption and good dimensional stability that is suitable for shock loads, with metal core to permit use of standard steel keyways and keys.

**ABOUT THE AUTHOR:**
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