Systematic Approach to Designing Plastic Spur and Helical Gears

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Abstract:
Plastic gears are being used increasingly in applications, such as printers, cameras, small household appliances, small power tools, instruments, timers, counters and various other products. Because of the many variables involved, an engineer who designs gear trains on an occasional basis may find the design process to be somewhat overwhelming. This article outlines a systematic design approach for developing injection molded plastic spur and helical gears. The use of a computer program for designing plastic gears is introduced as an invaluable design tool for solving complex gearing equations.

Introduction

As one of man's oldest mechanical devices, gears have been in use for thousands of years. It is no surprise that design techniques, manufacturing procedures and standards have been established based on the commonly available materials of the time, namely wood and metal. When plastic materials were first used as gears in the 1950's and 1960's, it was common to design them using existing AGMA standards for metal gears. For example, if a designer was considering trying plastic gears for one of his applications, he would usually take an existing product drawing specifying steel, change the material to nylon and leave the gear data intact. There were no standards at the time to guide the designer to do otherwise. It was also economical to consider cutting plastic gears with the available stock hobs and cutters.

Today, there are still no official "standards" available, but some tooth profiles have been developed especially for plastic gears and used successfully in industry. If the injection molding process is utilized, then the designer is free from the constraints of using stock hobs and cutters. Since special hobs are required regardless, (to compensate for mold shrinkages and EDM spark gap), the designer might as well incorporate additional modifications, such as full fillet radius, to optimize the strength of the gear teeth.

The modifications to the gear teeth mentioned in this article are not new to the gearing industry. Engineers familiar with gear design will recognize that the recommended modifications for strengthening the plastic gear teeth are the same ones specified for heavily loaded metal gears in critical applications.

In summary, this article describes the manufacturing process used to produce plastic gears, and explains inspection procedures.

Design of Plastic Gears

A designer must consider the pros and cons of plastic materials before choosing it as a gear material. The advantages are

- relative low cost
- resistance to corrosion
- reduction in weight (about 15% the weight of steel)
- low inerita
- self-lubrication capacity
- potential for noise reduction
- design freedom to obtain additional features in one product such as posts, cams, cluster gears, trunnions, palls, sprockets, metal inserts and spring arms
- color coding capability for fast, error-free assembly.

The disadvantages include

- lower strength
- greater thermal expansion/contraction
- limited heat resistance
- size change with moisture absorption.

COMMON PLASTIC MATERIALS FOR GEARS. Even though many new materials have been introduced in the past decade, the majority of gear applications still call for acetals and nylons. In special cases, polycarbonates, thermoplastic polyesters and thermoplastic polyurethanes can also be considered.[1,5]

<u>Acetal:</u> Strong, stiff plastic with exceptional dimensional stability due to low moisture absorption, resistance to creep and vibration fatigue; low coefficient of friction; high resistance to abrasion and chemicals; retains most properties when immersed in hot water; low tendency to stress-crack.

<u>Nylon:</u> Family of resins having outstanding toughness and wear resistance, low coefficient of friction and excellent electrical properties and chemical resistance. Resins are hygroscopic; dimensional stability is poorer than that of most other plastics.

<u>Polycarbonate:</u> Highest impact resistance of any rigid plastic; excellent stability and resistance to creep under load; fair chemical resistance; stress cracks in hydrocarbons; usually used with addition of glass fiber einforcement and PTFE lubricant.

<u>Thermoplastic polyester:</u> Excellent dimensional stability, electrical properties, toughness and chemical resistance, except to strong acids or bases; notch sensitive; not suitable for outdoor use or for service in hot water. Material is relatively soft and has the potential for tooth damage.

<u>Thermoplastic polyurethane:</u> Tough, extremely abrasive and

impact resistant material; good electrical properties and chemical resistance. Difficult to injection mold small parts due to the material's elastic properties.

Polyester elastomer: Sound dampening; resistance to flex-fatigue and impact.

All of the above base materials can be formulated with fillers, such as glass fibers, for added strength, and PTFE, silicone and molybdenum disulphide for added lubricity.

ESTABLISHING DESIGN GOALS. Arriving at the specifications for the best possible gears given an application is a time-consuming operation when considering the following variables.

- Load carrying capacity. Strength equations are used to avoid tooth fracture at the root. They are derived from the Lewis equations for beam strength and have been modified for plastic gear teeth.
- Smoothness. The gear mesh should be designed so that binding will not occur at extreme environmental conditions.
- Quiet continuity of action. Ensure that the contact ratio is greater than 1.0, preferably 1.2, or simply stated, that at least one tooth is in contact at all times.
- Cost. If the gears are to be molded in plastic, the task becomes even more difficult to handle, especially since there are literally hundreds of formulations from which to choose. In addition, the use of non-standard hobs allows the design engineer to consider a larger choice of tooth forms, such as the six listed in Appendix A. Because of all the options open much of the engineer's work must necessarily be a process of substitution and elimination. This process involves numerous calculations, some of them quite complex.

SPUR AND HELICAL GEAR DESIGN — A SYSTEMATIC APPROACH.

Study of design parameters. The gear train design engineer is first required to make a study of all the information about the application available to him, such as the power to be transmitted, the ratio required, the speeds involved, the environment in which the gears will operate, the life expectancy and the space in which the gears are to be housed.

The designer then proceeds to make preliminary choices of numbers of teeth, diametral pitch, tooth form, type of material to be used and spur or helical design.

Determine load carrying capacity. The engineer now possesses all the information required to determine what power the gears can safely transmit. If the answer is too low, the effect of a coarser diametral pitch, a stronger tooth form, an increased face width, or plastic having higher allowable design stresses, singly or in any desired combination, must be investigated. The following is an example of an equation that can be used to calculate load capacity.[1]

$$HP = \frac{D \times F \times n \times J \times St \times K_T \times K_L}{126{,}000 \times P \times C_s \times K_R}$$

where:

HP = horsepower
D = operating pitch diameter
F = effective face width
n = speed, rpm
J = geometry factor
St = tensile strength of plastic
K_T = temperature factor
K_L = life factor
P = diametral pitch, or normal diametral pitch
C_s = service factor
K_R = factor of safety

If the calculations indicate that the gears would be pushed to their limits, developing prototype parts to perform actual life tests would be advisable.

Options available to optimize a gear mesh. In order to get the most out of the relatively low tensile strength of plastics, the tooth geometry can be altered several ways to be as strong as possible.

- Full fillet radius modification. The tooth form shown in Fig. 1 has a full fillet radius at the root to eliminate sharp corners that could result in high stress points. This modification can add up to 20% to the tooth strength. Teeth of heavily loaded metal gears are also designed with this feature.
- Tip relief modification. Tip relief starting halfway up the addendum should be another modification specified for plastic gears. When a tooth deflects under load, it can get in the way of the oncoming tooth of the mating gear. This can cause noise, wear and loss of uniform action. Tip relief is added to improve this situation. (See Fig. 2.)
- Elimination of undercut condition. If a standard gear has a small number of teeth, the lower portion on the tooth will be undercut as illustrated on the left in Fig. 3. This condition is undesirable in that it weakens the tooth, causes wear and inhibits continuity of action.

To eliminate undercutting, the circular tooth thickness should be increased over standard as shown in Fig. 3 on the right.

- Balance circular tooth thickness. If the gear teeth in a mesh are designed at standard, they will be the "weak link" in a power drive. The pinion tooth illustrated in Fig. 4 clearly shows the problem. (All dimensions are in inches.)

To design the gear teeth for equal beam strength, it is essential that the thickness at the root be equal, as illustrated in Fig. 5.

With the use of available formulas, the tooth thickness of the pinion was increased from a standard of 0.0491 to 0.0640, and the gear tooth thickness was decreased from 0.0491 to 0.0480 to get an equal thickness of 0.0660 at the root.

Expansion and error considerations. Once the tooth thickness of each gear has been established, the next step would be the calculation required to determine the increase in center distance above standard to cater to all the factors that could otherwise cause the gears to bind in operation. The designer considers the following equation:

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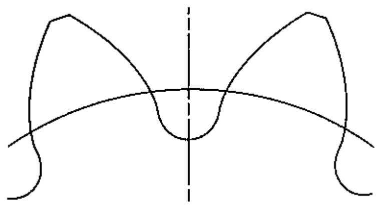


Fig. 1

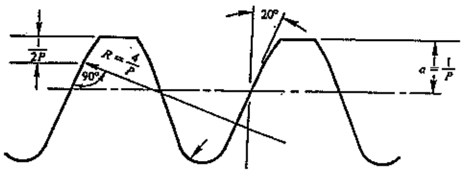


Fig. 2

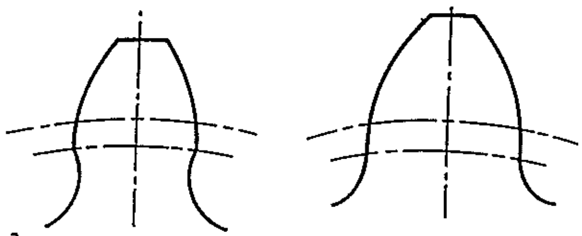


Fig. 3

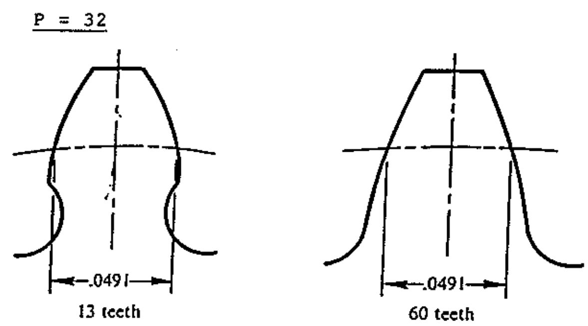


Fig. 4

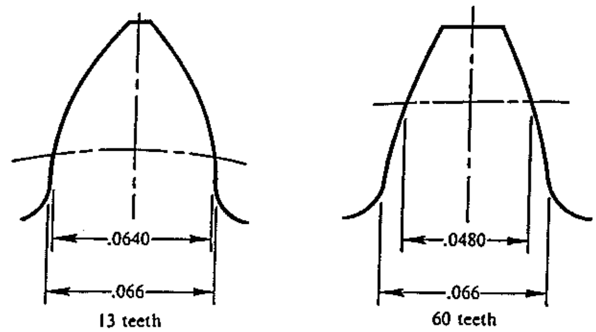


Fig. 5

$$\Delta_c = \frac{TCT_1 + TCT_2}{2} + C\left[(T-70)\left(\frac{COEF_1 \times N_1}{N_1+N_2} + \frac{COEF_2 \times N_2}{N_1+N_2} - COEF_H\right) + \left(\frac{M_1 \times N_1}{N_1+N_2} + \frac{M_2 \times N_2}{N_1+N_2} - M_H\right)\right] + \frac{TIR_1 + TIR_2}{2}$$

where:

- Δ_c = required increase in center distance
- TCT_1 = maximum total composite tolerance of 1st gear
- TCT_2 = maximum total composite tolerance of 2nd gear
- C = close mesh center distance
- T = maximum temperature to which gear will be subjected (°F)
- $COEF_1$ = coefficient of linear thermal expansion of material of 1st gear (in/in/°F)
- $COEF_2$ = coefficient of linear thermal expansion of material of 2nd gear (in/in/°F)
- $COEF_H$ = coefficient of linear thermal expansion of material of housing (in/in/°F)
- N_1 = number of teeth in 1st gear
- N_2 = number of teeth in 2nd gear
- M_1 = expansion due to moisture pick-up of material of 1st gear (in/in)
- M_2 = expansion due to moisture pick-up of material of 2nd gear (in/in)
- M_H = expansion due to moisture pick-up of material of the housing (in/in)
- TIR_1 = maximum allowable runout of bearing of 1st gear
- TIR_2 = maximum allowable runout of bearing of 2nd gear

The calculations enable the designer to specify tooth thicknesses and operational center distance.[1]

Calculate contact ratio. If one of the mating spur gears has a relatively small number of teeth, it is essential to check the pair's contact ratio, particularly if the gears are plastic. Plastics, in general, have high coefficients of linear thermal expansion. When the expansion differential between the gears and the housing in which they are mounted is great, and when the maximum environmental temperature is high, the center distance requires a significant amount of increase to prevent binding at those high temperatures. However when the same gears later contract at the lower temperatures, they may be out of mesh to an extent that the contact ratio becomes less than adquate.

One option available to get around this problem is to use a tooth form with longer teeth. (See Appendix A). For continuity of action the contact ratio must never be less than 1.0. It should preferably be at least 1.20.

The variables required to calculate the minimum contact ratio are the numbers of teeth for each gear, the diametral pitch, the minimum outside diameters, and the maximum operating center distance. If it is less than adequate, the designer can review the following available alternatives:

- tooth forms with longer teeth,
- non-standard diametral pitches ($P = 48.5526$, for example),
- materials with lower coefficients of linear thermal expansion,
- higher quality gears,
- tighter center distance tolerances,
- changes in the numbers of teeth in the gears.

distance tolerance from +0.003, −0.000 to +0.001, −0.000.

operating center distance	OCD	0.4773″
contact ratio	CR	1.13

He tries even more accurate gears — AGMA Quality No. 8, having maximum total composite tolerances of 0.0018.

operating center distance	OCD	0.4765″
contact ratio	CR	1.16

At this point, he decides to return to the beginning and determine what contact ratio he can achieve by switching the plastic he had first chosen from an acetal having a coefficient of linear thermal expansion of 6.8 x 10-5 in./in./deg. F, to a glass-filled variety having a coefficient of 1.5 x 10-5 in./in./deg. F. The operating center distance is now 0.4845-0.4855.

operating center distance	OCD	0.4750″
contact	CR	1.23

The designer has reached his goal, but only by switching to a more costly plastic, and has reduced the center distance tolerance to less than is desirable.

But he still has another option open. He can use a tooth form having a longer addendum. One such tooth form used successfully in plastics gearing is the tooth form with an addendum of 1.35/P shown in Appendix A. This tooth form will add little to the overall tool cost.

To eliminate undercut, the pinion will now be required to have a circular tooth thickness of 0.0424 — 0.0434. The minimum outside diameters of the pinion and gear will be 0.3796 and 0.6725, respectively. Returning to the cheaper unfilled acetal, a center distance tolerance of +0.003, −0.000 and AGMA Quality No. 7 gears, the designer tries again. The maximum center distance is now 0.4882

outside diameter	Do1	0.3796″
outside diameter	Do2	0.6725″
operating center distance	OCD	0.4882″
contact ratio	CR	1.20

Step 7. Check the load carrying capacity. Even though the loads in this application are light, the designer checks the allowable horsepower to determine what difference the new tooth form has made. If any change to the gear design is necessitated, it will amount to no more than a slight increase to the face width.

Step 8. Establish manufacturing and inspection data. Finally, the designer arrives at the manufacturing and inspection data to go on the production drawings. The complete data for the 15-tooth and 30-tooth gears used in this example are shown in Appendix E.

Manufacturing of Plastic Gears

Description of Molding Dies and the Injection Molding Process. A molded gear, of course, can be no more accurate that the molding die which produces it. Basic considerations for gear molding dies are the same as for any precision molded product. Since molded gears are usually small in relation to the size of average moldings, they lend themselves admirably to being molded in small, high speed automatic injection molding machines (20 - 80 ton range). This allows for compact dies, usually with one to eight cavities.

Several factors can be built in to a molding die to produce the most accurate gear possible.

- The increased strength of case-hardened die frames will enable them to withstand the abuses of the molding process and can maintain their accuracy throughout extended useful life.
- The elimination of wear bushings allowed by hardened steel surfaces in the bores removes the additional errors resulting from less than perfect concentricity of these items.
- The mold should have a balanced runner system to provide equal pressure drops into all cavities.
- Adequate venting in the cavities will allow air to be displaced by the flow of the plastic.
- Adequate ejection systems ensure minimum distortion of the product when ejected from the die.
- Adequate interlocks between the die halves remove the misalignment of running fits provided in the guide system.

Figs. 6 and 7 are cross sections of a gear molding die.

When the molding die is closed, the melted plastic resin is injected under high pressure into the gear cavities. When the resin solidifies, the molding die opens, and the plastic gear is ejected from the die. Cycle times usually range from 10 to 30 seconds, depending on the part geometry and the plastic resin used.

Several factors in a molding die can contribute to the gear's runout:

- The .0001/.0002″ running fit required between the core pin and the stationary half of the die;
- An additional .0001 to .0002″ running fit will be present if sleeve ejection is required around the core pin because the solid anchor is lost;
- Because the gear cavities themselves are a series of concentric rings, the potential for runout error exists.
- Uneven plastic flow resulting from the off-center gating position on the part will introduce additional runout in the gear.

The selection of a molder should be given careful consideration. True, given a correctly designed and accurately made molding die, any competent molder can produce gears; however, not all molding companies can be classified as gear manufacturers.

A good plastic gear manufacturer should have the following capabilities:

- experience in molding a wide range of gears;
- ability to advise on product design and selection of materials;
- availability of molding presses most suitable for gear molding;
- availability of inspection equipment necessary to maintain proper quality control (such as gear checking equipment).

Gear Cavities. Popular wisdom holds that molded gears are not as accurate as machined gears. This is inaccurate. A molded gear held to the tolerance of AGMA Quality Number Q8 is just as accurate as a machined gear of the same quality number. It is true that gears have not yet been molded to the highest precision obtainable by machining, but the number of gears requiring such precision represent only a small percentage of all gears made. In general, plastic gears are usually specified AGMA quality Q6 or Q7.

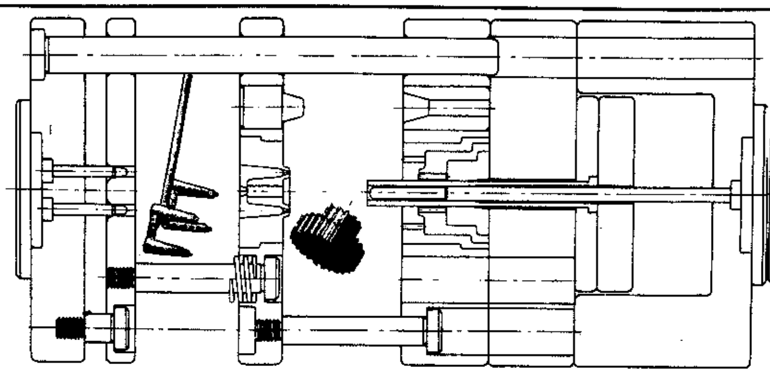


Fig. 6 Die Open

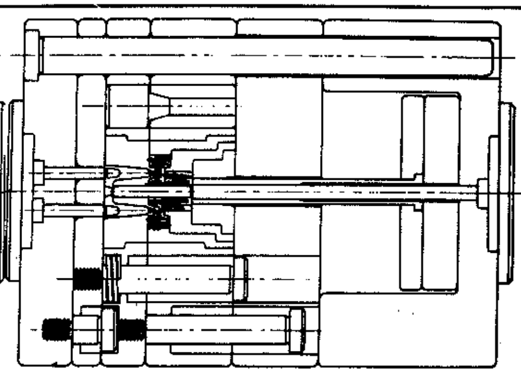


Fig. 7 Die Closed

The idea that molded plastic gears need not be as accurate as metal gears is based upon the contention that, because of the yielding nature of plastic, runout and tooth-to-tooth errors do not have the same ill effects. This is incorrect because plastic gear teeth that are flexing to an excessive degree because of inaccuracies will fail through fatigue and wear much earlier than accurate teeth.

All plastics shrink when changing from a molten to a solid state. As a consequence, all mold cavities must be made larger than the product specification. For example, if a molded gear is to have an outside diameter of 1.200", and the plastic has a mold shrinkage of .025 in/in, then the outside diameter of the cavity will be required to be 1.2308".

In making a gear cavity, however, it is not sufficient to take a generating hob and machine an oversize gear. Compensation must also be made for the pressure angle of the hob. If it isn't, the result will be a molded gear with a serious profile error. It will have a larger than acceptable tooth-to-tooth error. For example, in Fig. 8a an enlargement of a 32 D.P., 20 degree P.A., gear tooth is shown, and superimposed upon it is the profile of an oversize gear tooth cut with a standard 32 D.P., 20 degree P.A. hob. Fig. 8b shows the standard gear tooth, and this time superimposed upon it is the profile of the molded tooth (after shrinkage) that would be obtained from the oversize cavity.

Note that the tooth of the molded gear departs considerably from standard. It is thicker at the root and thinner at the tip. (It has a pressure angle much in excess of 20°.)

The formula used to calculate the correct pressure angle is:

$$\cos \phi_2 = \frac{D \cos \phi_1}{D(1 + S)}$$

where:

D = Pitch circle diameter
ϕ_1 = Pressure angle of hob
ϕ_2 = Pressure angle of molded gear
S = Shrinkage

The teeth in the cavity must be carefully compensated for shrinkage so that, when the molded gear solidifies and becomes stable, the teeth will have the correct profile. This design work is further complicated in the case of the helical gear, because the axial shrinkage is usually different from the shrinkage across the diameter.

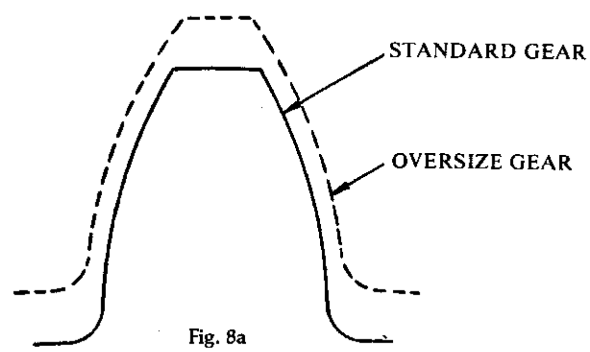


Fig. 8a

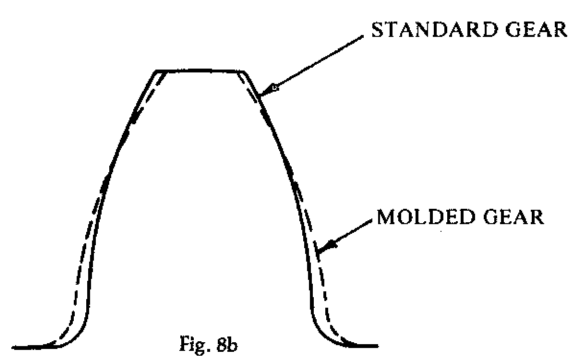


Fig. 8b

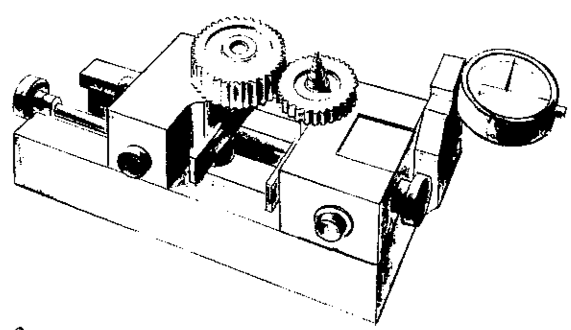


Fig. 9

Compensating correctly for shrinkage in a gear requires that the mold designer have a thorough understanding of gear geometery, plus considerable experience in the shrinkage

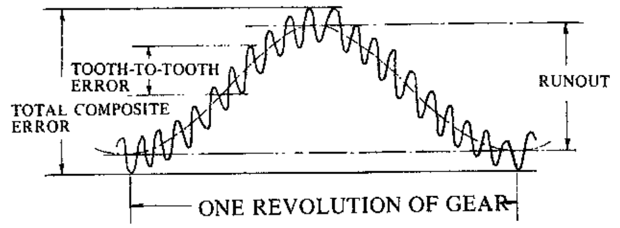


Fig. 10

behavior of all types and grades of plastics.

The actual manufacture of accurate gear cavities is accomplished by using conventional or wire EDM machines.

Inspection of Plastic Gears

One practical method of checking gear accuracy is to rotate the manufactured gear in close mesh with a master gear using a center distance measuring instrument. The model shown in Fig. 9 is equipped with a dial indicator and requires that the operator note the radial displacements as the gear is rotated manually through one complete revolution.

The more sophisticated models trace the radial displacements on a moving chart (See Fig. 10.) through an electronic device.

The errors present in a gear are runout, lateral runout (wobble), pitch error, and profile error.

The pitch error plus the profile error add up to the tooth-to-tooth composite error, and this, plus the total runout, is known as the total composite error.

As mentioned previously, the AGMA quality number is used to classify the quality of metal gears as well as plastic gears. The number specifies the gear accuracy in terms of maximum tooth-to-tooth and total composite tolerances allowed. The AGMA quality numbers and the corresponding maximum tolerances are listed in the AGMA "Gear Manual 390.03".

Summary

The injection molding process allows the gear designer to freely modify the tooth profile to obtain maximum tooth strength. These modifications will add little to the final tooling cost.

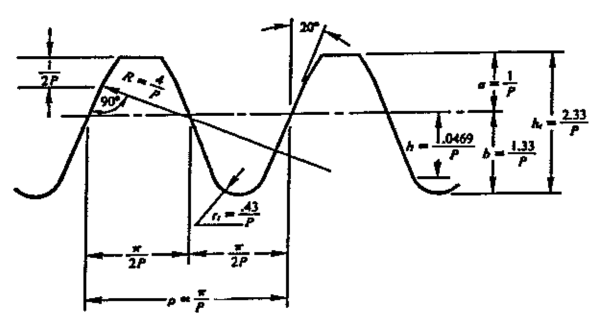
The gear design example was chosen to point out the many variables involved when designing plastic gears. The use of a systematic design approach allows the overall design problem to be broken down into smaller blocks which can be analyzed individually. Given another design with different constraints (fixed operating center distance, for example), the sequence of events in the design process would differ, but the problem would still be broken down into smaller blocks and analyzed one step at a time.

Correctly used, the systematic approach will provide a design for plastic gears having load carrying capacity, adequate contact, as close to balance tooth thickness as possible, and sufficient clearance so that no binding will occur at the extreme operating conditions.

APPENDICES

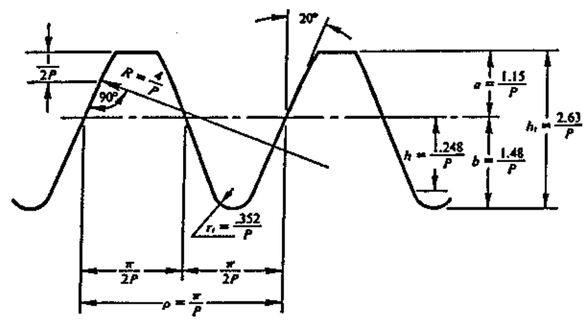
Appendix A – Tooth Forms for Plastic Gears.

(PGT-1 TOOTH FORM)

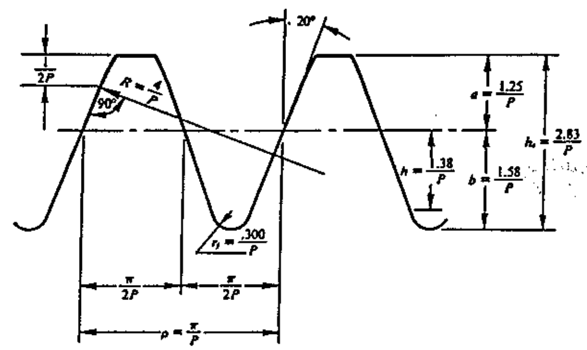


P = diametral pitch p = circular pitch 20° = pressure angle
a = addendum b = dedendum h_t = whole depth r_f = root radius
h = depth of straight portion of dedendum to point of tangency with root radius.

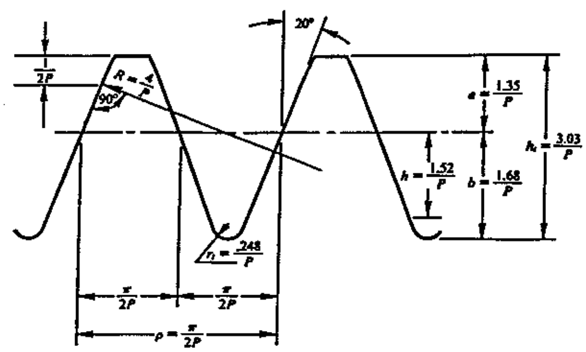
(PGT-2 TOOTH FORM)



(PGT-3 TOOTH FORM)



(PGT-4 TOOTH FORM)



(ISO 53-1974 (E) Tooth Form)

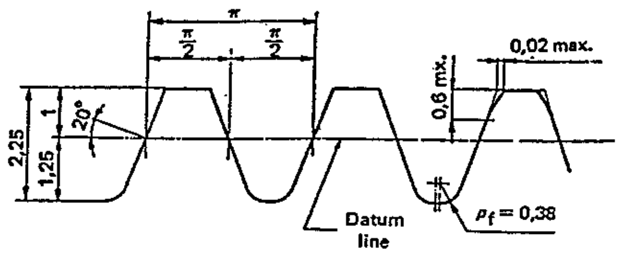
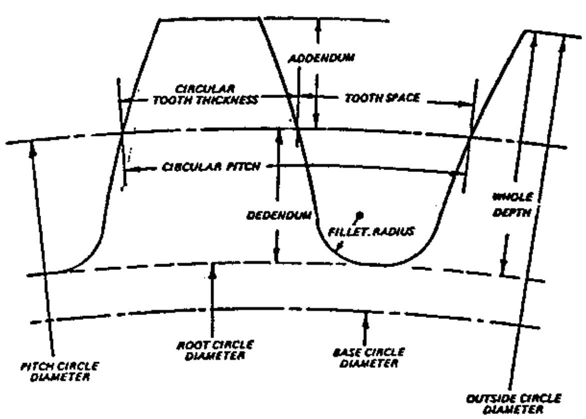


FIGURE — Representation of the profile of module $m = 1$ or of diametral pitch $P = 1$

(AGMA Tooth Form)



	Coarse-Pitch 19.99 and Coarser	Fine-Pitch 20.00 and Finer
Addendum	$\frac{1.000}{P}$	$\frac{1.000}{P}$
Dedendum	$\frac{1.250}{P}$	$\frac{1.20}{P} + .002$
Whole Depth	$\frac{2.250}{P}$	$\frac{2.20}{P} + .002$
Circular Tooth Thickness	$\frac{\pi}{2P}$	$\frac{\pi}{2P}$

P = Diametral Pitch

Appendix B — Computer Software.

If the design engineer has a personal computer and a gear software program, the time involved in designing a train of gears is reduced from many hours to a matter of minutes. It can prove to be an invaluable tool for a designer who wants to try many options in obtaining optimal gear meshes. Appendices B, C, and D describe one of the commercially available software programs for plastic gears.

Because mathematical and interpolation errors are eliminated, the designer is free to spend more time designing gears and less time solving mathematical equations.

Summary of Computer Program Features[6]

1) Program runs on IBM PCs and compatibles.
2) Design can be for either a spur or helical gear.
3) Designer can choose either English or Metric Units.
4) One gear can be designed alone, or two meshing gears can be designed simultaneously.
5) Gear data can be saved, retrieved, printed or deleted.
6) When an input value is entered in one of the sub-program selections, it is also entered in the remaining sub-programs. For example, if the diametral pitch of 48 is entered in sub-program "A", the user does not have to enter it again in sub-program "G"; it is already displayed there.
7) The calculations in the program have been designed similar to a spreadsheet. The user can enact "what if" variations by changing input values. The output values are immediately recalculated and displayed.

The program is a series of sub-programs which perform various calculations used in the process of designing spur and helical gears. (See Appendix C for a listing of the 12 sub-programs available in the "Main Menu"). Recognizable descriptions and symbols are employed throughout the program. The operational procedure can be mastered in less than an hour.

Appendix C — Main Menu Screen

A: (dsk)	SME__15T (gr #1)	SME__30T (gr #2)	English (unit)	Spur (type)	Mode 1	New

Plastics Gearing Technology, Inc. — Gear Design Program
****Main Menu****

A) Oper Phi, stand t, stand a, ht, and PD
B) Outside dia, root dia, and Domax (given t)
C) Minimum circular tooth thickness to avoid undercutting
D) Circular tooth thicknesses for balanced tooth strengths
E) CMCD, minimum operating center distance
F) Delta C
G) t1 + t2
H) Contact ratio, RA, AA, %RA
I) Gear testing radius (min and max)
J) Measurement over pins (min and max)
K) Horsepower ratings
L) Check and/or print gear data specified on drawing
M) Exit

Enter menu selection —>> M

F1	Help	F2	Type Unit	F3	Mode Mon	F4	Disk	F5	Dir Swap	F6	ID1 ID2	F7	New Del	F8	Clr1 Clr2	F9	Ret1 Ret2	F10	Sav1 Sav2

Description of abbrevations used in the above Main Menu

Oper Phi .... operating pressure angle
stand t .... standard circular tooth thickness
stand a .... standard addendum
ht .... whole depth
PD .... pitch diameter
Domax .... maximum allowable outside diameter
t circular tooth thickness

CMCD .... close mesh center distance
Delta C .... required increase in center distance
t1 + t2 .... sum of the tooth thicknesses for a given CMCD
RA .... recess action
AA .... approach action
%RA .... percent recess action

Appendix D — Example of a Sub-program Screen (For Contact Ratio Calculation).[6]

C:	SME__15T	SME__30T	English	Spur	Mode 1	Calc	Page 1 of 1

Contact ratio, RA, AA, %RA

Parameter Description	Symbol	Gr#	Value Units
number of teeth	N	1	15
number of teeth	N	2	30
diametral pitch	P		48.0000
outside diameter	Do	1	0.3796 inches
outside diameter	Do	2	0.6725 inches
operating center distance	OCD		0.4882 inches
recess action	RA		0.0501
approach action	AA		0.0235
contact ratio	CR		1.196
percent recess action	%RA		68.11
User Note: Gear 1 is the drive gear and Gear 2 is the driven gear.			

F1	Help Equa	F2	F3	Mode Mon	F4	Calc	F5	Dir Swap	F6	ID1 ID1	PgUp PgDn	Prev page Esc Exit Next Page

Appendix E — Manufacturing and Inspection Data For The Pinion and Gear

PINION DATA

Basic Specifications	
number of teeth	15
diametral pitch	48.0000
pressure angle	20.00°
standard pitch diameter	0.3125″
PGT tooth form	4
standard addendum	0.0281″
standard whole depth	0.0631″
circ tooth thickness max at PCD	0.04340″
circ tooth thickness min at PCD	0.04240″
Manufacturing and Inspection	
gear testing radius max	0.1716″
gear testing radius min	0.1677″
AGMA quality number	Q7
max total composite tol of gear	0.0026″
max tooth-to-tooth comp tolerance	0.0016″
number of teeth in master gear	96
circ tooth thickness of master gear	0.03270″
testing pressure	8.0 oz
diameter of measuring pins	0.0400″
measurement over two pins (max)	0.3920″
measurement over two pins (min)	0.3903″
max allowable outside diameter	0.3826″
outside diameter tol (+ tol)	0.0000″
outside diameter tol (− tol)	0.0030″
max root diameter	0.2718″
Engineering References	
mating gear part number	30T gear
number of teeth in mating gear	30
max operating center distance	0.4882″
min operating center distance	0.4852″

GEAR DATA

Basic Specifications	
number of teeth	30
diametral pitch	48.0000
pressure angle	20.00°
standard pitch diameter	0.6250″
PGT tooth form	4
standard addendum	0.0281″
standard whole depth	0.0631″
circ tooth thickness max at PCD	0.03100″
circ tooth thickness min at PCD	0.03000″
Manufacturing and Inspection	
gear testing radius max	0.3114″
gear testing radius min	0.3074″
AGMA quality number	Q7
max total composite tol of gear	0.0026″
max tooth-to-tooth comp tolerance	0.0014″
number of teeth in master gear	96
circ tooth thickness of master gear	0.03270″
testing pressure	8.0 oz
diameter of measuring pins	0.0400″
measurement over two pins (max)	0.6851″
measurement over two pins (min)	0.6828″
outside diameter	0.6765″
outside diameter tol (+ tol)	0.0000″
outside diameter tol (− tol)	0.0040″
max root diameter	0.5503″
Engineering References	
mating gear part number	15T GEAR
number of teeth in mating gear	15
max operating center distance	0.4882″
min operating center distance	0.4852″

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