The Design and Manufacture of Machined Plastic Gears

Part 1

Design Parameters of Plastic Gears

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ABSTRACT

The use of Plastic Gearing is increasing steadily in new products. This is due in part to the availability of recent design data. Fatigue stress of plastic gears as a function of diametral pitch, pressure angle, pitch line velocity, lubrication and life cycles are described based on test information. Design procedures for plastic gears are presented.

Introduction

The use of machined plastic gears is increasing in industrial power transmission applications. Cast or extruded gear materials of 50 mm to over 2 meter (2" – 84") in diameter are machined to desired dimensions and gear tooth forms. Nylons and acetals are the most widely used thermoplastic gear materials. They offer resiliency, resistance to wear and corrosion, noise reduction, vibration suppression, lightweight and minimum maintenance. This increased use is due in part to the availability of new data in design parameters, manufacturing and installation techniques to optimize utilization of the materials and to prevent premature failures.

Some test data on plastic gears are the results of earlier publications.[1-4] Plastic gears are sensitive to frictional heat and hysteresis loss, which causes softening, expansion and deflection of the gear teeth. Tooth interference, caused by deflection and creep, can lead to different types of damage due to the viscoelastic nature of plastic gears.[5] For better heat dispersion and to reduce contact stress, plastic gears are usually mated with metal gears. To measure plastic gear temperature in operation, a technique utilizing an infrared radiometer was developed[6] and friction heat and hysteresis loss were investigated.[6, 7] Although plastic/steel contacts show rather low friction[8] and some plastic gears can be operated without lubrication, the lubrication also has a considerable effect on the performance of the plastic gears.[9] The tooth deflection of plastic gears and low Hertizan stress of plastic/steel contacts results in generally higher gear contact ratio and smooth rotation of plastic and steel gear mates. A model to assess gear contact ratios of plastic/steel gear mates was proposed.[10] A larger backlash allowance required for plastic/steel gear mates was suggested.[5] Factors which influence performance of plastic gears are, to an extent, clarified by the works mentioned above and others. However, design data available to meet power transmission of plastic gears is still insufficient for most applications. The purpose of this paper is to fill part of this gap by presenting some pertinent design parameters based on experiments, which were conducted at conditions close to typical field applications. Design parameters as functions of diametrical pitch, tooth form, lubrication, pitch line velocity, stress levels and life cycles using cast type 6 nylon and other plastic gears are presented.

Experiments

Test Gears

Industrial experience indicates that anionically polymerized cast type 6 nylon is one of the highest strength unreinforced thermoplastic gear materials available. The typical test material contains molybdenum disulphide for improved lubricity and wear resistance. Impact modified Nylon 6/6 and ultra high molecular weight polyethylene (UHMWPE) are known to be very tough and resistant to fatigue and notch sensitivity. The latter starting resin has a molecular weight of 2-5 million by solution viscosity test. Polyacetal (melt viscosity 3360 poise), which features strength and low moisture absorption was also selected for this test. Physical properties of these four thermoplastic test gear materials are summarized in Table 1.

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TABLE 1. PHYSICAL PROPERTIES OF TEST GEAR MATERIALS

Property	Test Method ASTM	Unit	Cast Nylon 6 With MoS_2	Nylon 6/6[3] Impact Modified	UHMWPE	Polyacetal
Tensile Strength	D638	Mpa (psi)	84 (12,000)	52 (7,500)	33 (4,800)	73 (10,400)
Elongation	D638	%	15	40	320	12-75
Tensile Modulus	D638	Mpa (psi)	2,800 (400,000)	1,760 (255,000)	630 (90,000)	3,290 (470,000)
Hardness, Rockwell	D785	R	116	112	64	120
Deformation Under Load 14MPa (2000 psi), 50°C	D621	%	.75	–	6-8 (6 hrs.)	0.7
Deflection Temperature 1.8 MPa (264 psi)	D648	°C	156	71	–	117
Melting Point	D789	°C	215-227	250-260	130	165-170
Tensile Impact	D1822	kJ/m^2 (ft. $lbs./in.^2$)	221 (105)	588 (280)	2,100 (1,000)	74 (35)
Specific Gravity	D792	–	1.16	1.09	0.94	1.42
Water Absorption						
24 Hours	D570[2]	%	0.9	1.2	.01	0.25
Saturation	D570	%	6.0	6.7	–	0.9

Note: (1) All values shown are based on as molded dry specimens
(2) Specimens 1/8" Thick, 2" Diameter (3) Published data of supplier

Test gears of 5, 8, 10, and 16 DP (5.1, 3.2, 2.5 and 1.6 modules) involute spur gears with 12.7 mm (½") face width were used. Tooth proportion was made according to AGMA standard 201.02 applying coarse-pitch, 20° full-depth involute form, except one series of gears with 14½° pressure angle. All test gears were cut with class A ground thread hobs. The test gears were "driven" gears. The "driver" steel pinion was hardened to Rockwell C44 and ground AGMA 390.03 Quality Number 13.

Test Equipment and Procedure

A four-square gear bench was utilized for fatigue testing of spur gears. The equipment consists of a variable speed electric motor, four-square mechanism, a torque-meter, a forced lubricating system with oil temperature control, a revolution counter, an electronic gear failure detection system and a digital panel instrument indicating torque or rotational speed. An overview of the test equipment is shown in Fig. 1.

Extreme pressure (EP) grade gear oil was heated to 120° ±2°F (49° ±1°C), while it circulated in a closed loop. After it reached this temperature, the oil was injected under pressure onto the departing side of the two meshing gears. The machine was started without load to run at the lowest speed for 10 minutes. Torque was then applied on the nylon gear,

Fig. 1—Overview of gear test equipment.

in the direction that it acts as a driven gear, before the machine speed was increased to test conditions. During the run, especially in the beginning, the torque had to be adjusted due to the creep of the plastic gear material. The adjustments were made, so that the torque was maintained within ±2% of its normal value throughout the test. The machine stopped automatically when the gear failed or when the load exceeded preset limits of high or low torque. The life of the gear could then be read from the revolution counter.

Test variables used are gear materials, diametral pitch, pressure angle, pitch line velocity and type of lubrication. Test parameters are listed in Table 2. At least four stress levels for each test parameter were applied.

The lubrication systems evaluated were continuous oil jet lubrication, initial greasing and unlubricated (dry). For the initial greasing test, a thin layer of calcium complex type petroleum base grease was brushed on the flanks of test thermoplastic gear and mating steel pinion before installation. For the unlubricated gear test, the flank surfaces were carefully wiped clean with acetone and then assembled on the test machine.

Test Results and Discussion

Root bending stress $\delta\beta$ vs. life cycles of the tested gears was calculated from the Lewis equation:

$$\delta\beta = \frac{F \bullet P}{f \bullet y} \qquad \text{Eq. (1)}$$

The form factor for thermoplastic gears, Table 3, was determined on the basis that the worst load conditions occur near the pitch point (1), due to multiple pairs of teeth sharing the load. The high contact ratio of thermoplastic gears can be observed by high speed photographic technique. The Lewis equation assumes that the gear contact ration is 1. This assumption presents a very practical way to assess fatigue strength of thermoplatic gears and serves design purposes, although it contains the factor of undetermined gear contact ratio.

Earlier test results indicate that thermoplastic gears, when properly designed and installed, fail eventually by root bending fatigue fracture.[5] Evaluation of fatigue life of thermoplastic gears for this paper was done by root stress using Equation (1).

Gear Fatigue Strength

S-N curves of 4 different diametral pitches of cast nylon 6 (filled with MoS_2) gears at pitch line velocity of 10.2 m/s (2000 fpm) are illustrated in Fig. 2. Test points of these oil jet lubricated gears, for each of the four diametral pitches, can be connected approximately as a straight line on a semilogarithmic scale. The finest gears tested (16 P) could with-

TABLE 2. TEST PARAMETERS

Gear Materials	Diametral Pitch P	Pressure Angle, Degree	Gear		Gear		Pitch Line Velocity ft/min	Lubrication
			Dp	Teeth	Dp	Teeth		
Cast Nylon with MoS_2	5	20	3.6"	18	3.6"	18	2000	oil
	8	20 14½	4.5"	36	2.5"	20	2000	oil
	10	20	4.5"	45	2.5"	25	2000	oil & dry
	16	20	4.5"	72	2.5"	40	2000	oil
	10	20	4.5"	45	2.5"	25	680 1200 2000 4000	grease
Nylon 6/6, Impact Modified	10	20	4.5"	45	2.5"	25	2000	oil
UHMWPE	10	20	4.5"	45	2.5"	25	2000	oil
Polyacetal	10	20	4.5"	45	2.5"	25	2000	oil

Note: 1" = 25.4 mm 1000 ft./min. = 5.08 m/s

TABLE 3
TOOTH FORM FACTOR y

No. of Teeth	Pressure Angle 14½°	20° Depth	20° Stub
18	—	.521	.603
24	.509	.572	.664
28	.535	.597	.688
36	.559	.640	.721
45	.579	.681	.744
72	.611	.731	.788
150	.625	.779	.830
300	.650	.801	.855
Rack	.660	.823	.881

stand the highest root bending strength and vice versa. This can be explained by a higher contact ratio, i.e. more teeth to share the load, on fine pitch nylon gears.

For comparison, the endurance strength of 8 P, 14½° pressure angle gears were tested and the results charted with those of 8 P, 20° pressure angle gears (Fig. 3). The relationship between torque (load) and life cycle is linear and the two lines are approximately parallel. Gears with 20° PA are ca. 15% higher in load carrying capacity. Fatigue tangential force F at a life cycle is proportional to tooth form factor y as can be seen from Equation (1).

Root stresses of MoS_2 filled cast nylon 6 gears at 10 million cycles, from Fig. 2, are 6170, 4650, 3830 and 3180 psi for 16, 10, 8 and 5 pitches respectively. The root stress of other diametral pitches within the range tested can be illustrated as a curve (Fig. 4). Utilizing this curve, the fatigue life bending stress at 10 million cycles of other pitches can be obtained by interpolating the test points. Computer aided regression analysis show the root stress of other pitches can be expressed approximately as follows:

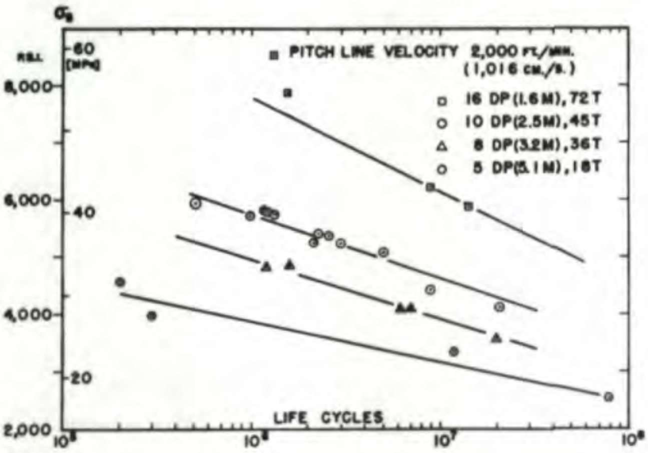

Fig. 2 – Bending fatigue stress of molybdenum disulphide filled cast nylon 6 gears at 2000 fpm (10.2 m/s). Standard 20° full-depth involute gears of different diametral pitches, oil jet lubricated (5).

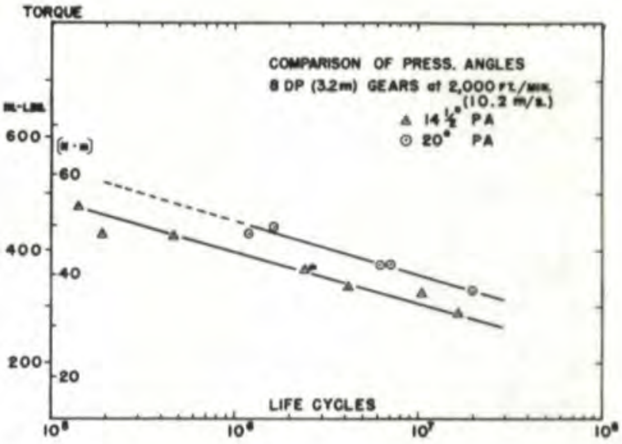

Fig. 3 – Torque vs. life cycle curves of gears with 14-1/2° and 20° pressure angles, oil jet lubricated (5).

$$\delta\beta = 28{,}992 - 25{,}637\ (1n\ P) + 9{,}578.47\ (1n\ P)^2 - 1{,}228.29\ (1n\ P)^3 \qquad \text{Eq. (2)}$$

where 1n P = natural logarithm of diametral pitch

Gears of 10 P, 45 teeth and ½" face width machined from Nylon 6/6 (impact modified), UHMWPE and polyacetal were

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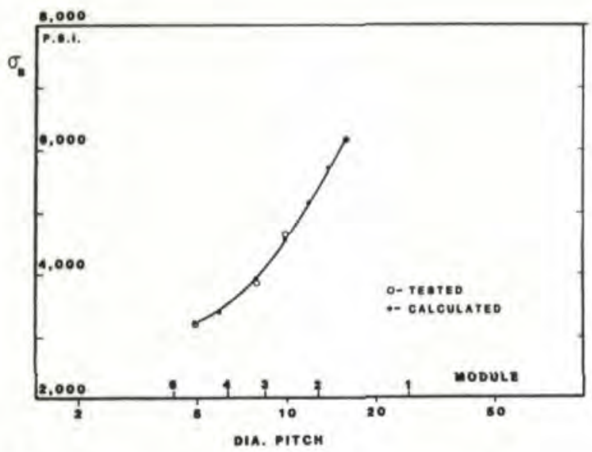

Fig. 4 – Bending fatigue stress of molybdenum disulphide filled cast nylon 6 gears vs. diametral pitch and module, oil jet lubricated.

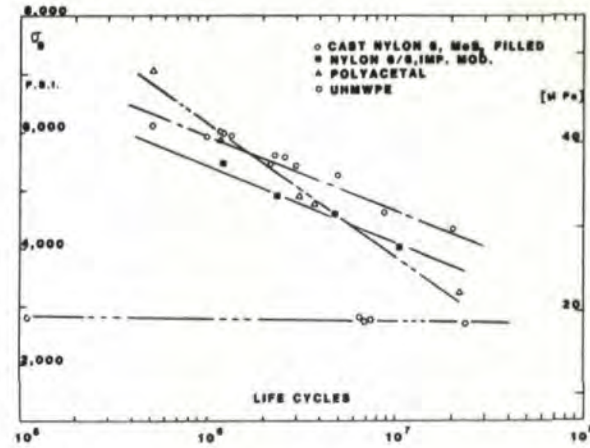

Fig. 5 – S-N curves of some thermoplastic gear materials, oil jet lubricated.

run at 2000 ft./min. for the endurance test and the results including those of cast Nylon 6 are illustrated in Fig. 5. The results of regression analysis of S-N curves are shown in Table 4. Root stresses at life cycles of 10×10^6 are 4650, 4084, 3856 and 2730 psi for cast Nylon 6, impact modified Nylon 6/6, polyacetal, and UHMWPE, respectively. A very moderate decline in the slope of S-N curve of UHMWPE indicates the material is extremely fatigue resistant, while the steep decline of the polyacetal curve indicates sensitivity of the material to fatigue. All coefficients of correlation for different thermoplastic gear materials are close to ± 1.0, which illustrates good approximation of S-N curves to straight lines. The small deviations are partially due to the high contact ratios of the gears. The slope of the S-N lines varies considerably with material.

Effects of Lubrication and Life Cycles

Fatigue root bending stress and life cycles of oil jet lubricated, initial greasing only and unlubricated (dry) cast nylon 6 gears are depicted in Fig. 6. The S-N curves form straight lines in the range of long cycle fatigue failures.[11] The results of regression analysis are given in Table 4. Test points of oil lubricated and initially greased gears form approximate parallel lines and they show small deviations from each of the straight lines, while a much larger scattering of the points resulted with dry gears. Under dry conditions, the slope of the S-N line toward longer life cycles is steeper than those of two other lubricated types.

As indicated in Table 4, the root stresses of initially greased and unlubricated cast nylon 6 at life cycles of 10 million are

TABLE 4.
S-N CURVE REGRESSION ANALYSIS OF SOME THERMOPLASTIC GEARS

Gear Materials	Lubrication	Root Stress, psi at Life Cycles of 10×10^6	Slope S-N Line psi	Coefficient of Correlation
Cast Nylon 6	Oil jet	4650 (100%)	−1240	−.9723
Nylon 6/6,	Oil jet	4084 (87.8%)	−1490	−.9973
Polyacetal	Oil jet	3856 (82.9%)	−2324	−.9949
UHMWPE	Oil jet	2730 (58.7%)	−80	−.9978
Cast Nylon 6	Initial greasing	3380 (72.7%)	−1063	−.9929
Cast Nylon 6	Unlubricated	1810 (38.9%)	−1659	−.9334

TABLE 5.
LIFE FACTOR K_1 OF MoS_2 FILLED CAST NYLON 6 SPUR GEARS EXTERNALLY LUBRICATED

	Diametral pitch or module			
No. of Cycles	16 P 1.6 m	10 P 2.5 m	8 P 3.2 m	5 P 5.1 m
1 million	1.26	1.24	1.30	1.22
10 million	1.0	1.0	1.0	1.0
30 million	0.87	0.88	0.89	0.89

72.7% and 38.9% of that of oil jet lubricated gears. These values can be used as lubrication factors.

Life factor K_1 of greased and oil lubricated cast nylon 6 spur gears at 1, 10 and 30 million cycles are shown in Table 5.

Effects of Pitch Line Velocity

10P cast nylon gears were examined for fatigue strength at pitch line velocities of 680, 1200, 2000 and 4000 fpm (3.5, 6.1, 10.2 and 20.3 m/s) under initially greased conditions. Test points of each velocity can be connected as straight lines (Fig. 7). S-N lines of 680, 2000 and 4000 fpm are close to parallel. Root fatigue stress of 1200 fpm at 2 to 4 million cycles are higher than those run at 680 fpm. This may be related to properties peculiar to the test machine, such as natural resonance. At 10 million life cycles and over, where the gears are rated, the root stress appears to correlate with other tested pitch line velocity.

Relative root stress at 20 million cycles compare gear strength run at different pitch line velocities. Root stresses illustrated in Figs. 2 (except as noted), 4 and 5 are obtained at pitch line velocity of 2000 fpm. Correction factors for pitch line velocities are listed below:

Pitch Line Velocity, fpm	680	1200	2000	4000
Root Bending Stress, psi	3856	3442	3030	2851
Velocity Correction Factor	1.27	1.14	1.00	0.94
Calculated Factor	1.27	1.11	1.00	0.92

The velocity correction factors can be calculated approximately by

$$K_v = 394/(200 + v) + 0.825 \qquad \text{Eq. (3)}$$

where v = Pitch line velocity, fpm

$$= .262 \times D_p\, n$$

Thermoplastics Gear Design – A Summary

The test results of fatigue bending stress described are used to obtain safe tangential force, safe torque and horsepower capacity of thermoplastic gears under various operational conditions. Conversely, knowing the required torque or horsepower to be transmitted, a gear design under specified conditions can be made.

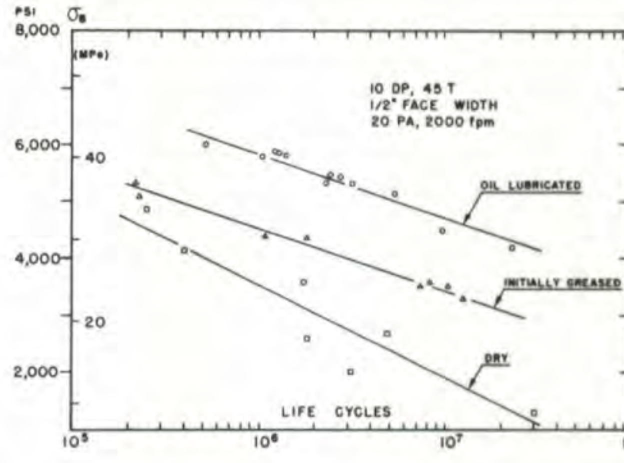

Fig. 6 – S-N curves of filled cast nylon 6 gears effected by types of lubrication (9).

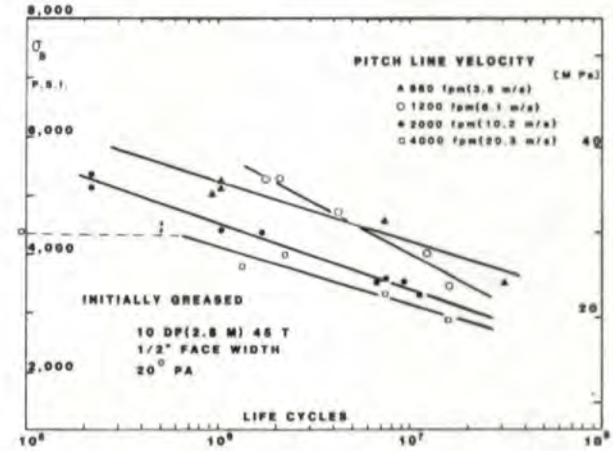

Fig. 7 – S-N curves of filled cast nylon 6 gears using different pitch line velocity as parameter.

Fatigue bending stresses under different test parameters are illustrated in Figs. 2, 4, 5, 6 and 7 and Table 4 using Eq. (1).

1. $\delta\beta$ – Fatigue Bending Stress

$$\delta\beta = \frac{F\,P}{f\,y}$$

2. F_t – Safe Tangential Force

$$F_t = \frac{S_{at}}{P}\, f\, y\, L_u\, K_v\, K_1 \qquad \text{Eq. (4)}$$

3. T – Safe Torque

$$T = \frac{D_p}{2}\, F_t \qquad \text{Eq. (5)}$$

4. HP – Horsepower Capacity

$$HP = \frac{F_t v}{33{,}000} \text{ or } \frac{T_n}{63{,}000} \qquad \text{Eq. (6)}$$

In obtaining the fatigue bending stress and velocity factor of gears, interpolation of the results using Fig. 5 and Eq. (3)

are permissible. However, extrapolation of the results to untested pitch as well as velocity ranges can be very misleading.

To determine the allowable bending stress S_{at} in Eq. (4), 75% of fatigue root bending stress is used for externally lubricated cast nylon 6 and impact modified Nylon 6/6. Tooth form factor y, in Table 3, is used for pressure angle or tooth form variables. For initial greased unlubricated cast nylon 6 (filled with molybdenum disulphide) gears lubrication factors L_u of 72.7% and 28.9%, respectively, are to be applied. Life factors in Table 5 apply to grease and oil jet lubricated cast nylon 6 spur gears.

NOMENCLATURE

D_p = pitch diameter
F = tangential force at pitch circle
f = gear face width
F_t = safe tangential force
HP = horsepower cavity
K_l = life factor
K_v = velocity factor
L_u = lubrication factor
m = module
n = gear rotational speed, rpm
P = diametral pitch
S-N = root stress vs. life cycle
S_{at} = allowable root bending stress
T = safe torque
v = pitch line velocity
y = tooth form factor for load applied near pitch circle
δ_B = root fatigue bending stress

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