 Bulletin of JSME, Vol. 29, No. 250 April 1986

Paper No. 84—0454

Development of Plastic Gears for Power Transmission*

(Design on Load-carrying Capacity)

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Plastic gears have various merits such as silent running, self-lubrication, and so on, but they also have some demerits such as less load-carrying capacity and unsteady life time. In our sixty-two experiments, most of the failures of plastic gears were fractures of teeth from near pitch points. On basis of these results, a new design equation for bending strength, described by the bending stress at the pitch point, is proposed. In the equation, some coefficients: tooth thickness coefficient, load increasing factor, stress correction factor, etc., are introduced to realize higher load capacity economically by making the best use of the characteristics of plastic materials.

Key Words: Gear, Plastic, Nylon, Damage, Fracture, Design Equation

1. Introduction

Plastic gears are highly promising as gears for machines which call for silent running, lubricantless operation or corrosion resistance. However the use of plastic gears for power transmission is not popularized yet because of low load capacity, unsteady life and expensiveness.

The object of a series of these studies is to develop plastic gears for power transmission keeping good properties of plastics. In this report, the real state on failures of plastic gears is shown in terms of failure rates, kinds of failures and their percentages, using the results of sixty-two running tests performed by one of the authors, Tukamoto.

Ninety-two percent of the failures were tooth fractures and the starting points of fractures on the working tooth surface located between the worst point and the inner worst point. From this result, it can be known that the bending strength design equation for steel gears, which is derived from the idea that the weakest section exists at the tooth fillet, can not apply for plastic gears. So, a new design equation for load-carrying capacity is proposed in consideration of the actual state of failures and the characteristics of plastics.

2. Failures of Plastic Gears

The materials of the gear pair in the sixty-two experiments are MC-nylon 901 (improved 6-nylon, Table 1) and S45C. The specifications of gears and the experiment conditions range rather wide. Modules m range from 3 to 5 mm. Plastic gears are used mainly as the driver and the number of teeth Z_p ranges from 17 to 34, the number of teeth of driven steel gears Z_s ranges from 37 to 60 and the face width b ranges from 10 to 20 mm. The number of revolutions of plastic gears n_p ranges from 500 to 1500 rpm, the output torque T from 7.8 to 78 Nm and the room temperature from 15 to 35 °C.

In the sixty-two experiments, no-failures were 55% and failures were 45%. The failures can be classified as follows(1).

(1) Fractures near the pitch point 62%
(2) Fractures near the inner worst point 30%
(3) Scoring or pitting 8%

Most of the failures were fractures (92%). About starting points of the fracture, 62% were near the pitch point and 30% were near the inner worst point, as shown in Fig.1. Thus, it hardly happens in plastic gears unlike metallic gears that the fracture takes place at the tooth fillet.

The mechanism of the fracture of plastic teeth was discussed in detail in the previous report(3). The time and the position of the fracture occurrence can be considered as follows.

(1) Under overloads or low speed operations, the fracture occurs at the fillet of the tooth at the early stage of the operation.

(2) At the beginning of the growth of an abnormal wear of the root [$N_T=(2\sim3)\times10^6$ rev], the temperature near the inner worst point becomes high. So, the fracture initi-

Table 1 Properties of MC-nylon(2)

	Items	Units	MC Nylon
Physical properties	Melting point	°C	208
	Specific weight		1.16
	Rockwell hardness	HRR or HB	115～120 HRR
	Specific heat	J/(kg.K)	1670
	Heat conductivity	W/(m.K)	0.233
	Linear expansion coeff.	K^{-1}	0.9×10^{-4}
Mechanical properties	Tensile strength	MPa	78～96(23°C)
	Elongation	%	10～50
	Young's modulus	GPa	2.94～3.43
	Poisson's ratio		0.35
	Bending strength	MPa	96～110
	Compressive strength	MPa	92～103

* Received 26th October, 1984.
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ates from this point.

(3) Under a certain degree of the abnormal wear of the root growing [$N_T=(5\sim7)\times10^6$ rev], the area near the pitch point is rubbed strongly. Therefore, the fracture initiates from this point.

Fig.2 shows the results of the sixty-two experiments. Assuming that all the load acts on the tooth tip, the bending stress σ_p at the pitch point, at which most of the fractures occur, was calculated. In Fig.2, the relation between the bending stress at the pitch point σ_p and the total number of revolutions N_T is identified. Moreover, types of failures are shown by symbols ○∿▲ and modules are indicated by numerals 3∿5.

The following can be seen from Fig.2.

(1) No failure takes place at $\sigma_p \leqq 19$ MPa for m=5 and at $\sigma_p \leqq 30$ MPa for m=3.

(2) The value of σ_p being above these stresses, failures take place frequently. Moreover, the life extends widely from the beginning of the operation to 10^7 rev under the same stress. This is the reason why plastic gears are not reliable as power transmissions.

(3) Many of the failures are fractures from near the pitch point and the fracture from near the inner worst point occurs at the early stage of the operation.

(4) The smaller the modules are, the higher the allowable bending stresses are.

Failures take place in the groups C∿D in Fig.2. If these groups are eliminated, the unsteady life can be improved. If overload (in this case, σ_p>45 MPa) is removed, the fracture in the group E can be prevented. The failure in the groups C and D must be able to be removed because some gears have achieved a life of $N_T \geqq 10^7$ rev under the same stress σ_p. To remove these failures, some ideas to prevent the abnormal wear at the root and to suppress the tooth temperature will be necessary.

3. Load Capacity of Plastic Gears

The failure of plastic gears shown in Fig.2 is greatly different from that of steel gears. This is the reason why the mechanical and physical properties or the dynamic behaviors of plastic gears are very different from those of steel gears because plastics are highpolymeric and viscoelastic materials. As 92% of the failures are fractures, it is sufficient to check the bending strength of teeth.

3.1 Present equations for the bending strength

Present design equations for the bending strength of plastic gears are based on

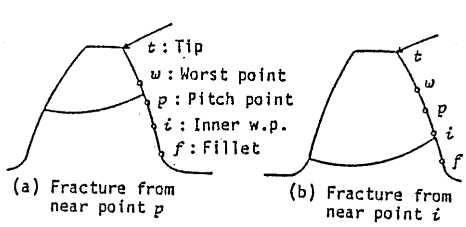

Fig.1 Position of tooth fracture

Lewis' equation for metallic gears. In Hachmann's equation[4], the allowable bending stress at the fillet σ_b is treated as a function of the temperature and the total number of revolutions. In Masuzawa's equation[5], the allowable bending stress is treated as a function of the number of repeated stresses and the module, moreover the correcting factor for the temperature is added.

All of these equations are based on the assumption that the weakest section exists at the fillet of the tooth and the lubricant is supplied. This assumption contradicts the fact that the fracture of plastic gears takes place between the worst point and the inner worst point. Moreover, dry operation, which is the great merit of plastic gears, is not considered in these equations.

3.2 A new equation for bending strength

In this section, a new equation is proposed in which the weakest section of the tooth is considered to locate at the pitch point.

The reason why the weakest section of the tooth locates at the pitch point is as follows. On the assumption of all loads acting on the tooth tip, comparing the bending stresses at the pitch point and at the fillet about a tooth with a pressure angle of 14.5°, in which the tooth is weaker against bending stress than at 20°, there is no difference between both stress values for the number of teeth more than 40.

Moreover, the temperature at the pitch point is higher than at the tooth fillet by 15∿20 °C. Therefore, the bending fatigue limit of the plastic gear material is lower about 20% at the pitch point than at the fillet (Fig.4). From these facts, it can be understood that many of the failures are fractures initiated from the pitch point.

Figure 3 is illustrates how to derive the equation for the bending strength of plastic gears. In this figure, the tooth thickness is increased by τm in comparison with the standard thickness $0.5\pi m$ as one of the methods to increase the load capacity. We call the coefficient τ the tooth thick-

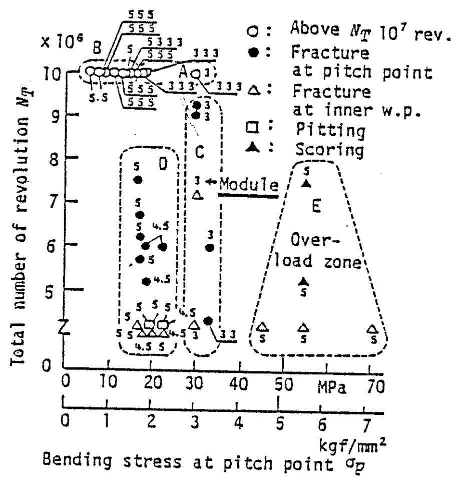

Fig.2 Relation between σ_p and N_T

ness coefficient.

In plastic teeth, an ideal meshing like in metallic gear teeth can not be expected because of the wear or the deflection of the tooth. So, the load is assumed to act on the tooth tip. When the tangential force on pitch circle, the normal force and its horizontal component are denoted respectively by P, P_n and P' [N], the arm length of bending moment at pitch point by h [mm], the circular tooth thickness on pitch circle by S [mm] and modules by m [mm], we can obtain

$$P' \fallingdotseq P,\ h \fallingdotseq m,\ S = (0.5\pi + \tau)m.$$

The bending stress at the weakest section AA', which is caused by the bending moment $P'h$, being denoted by σ_p [MPa], we get

$$P'h = \sigma_p (bS^2/6)$$

$$\therefore\ P = (0.5\pi + \tau)^2 m b \sigma_p / 6$$

$$\fallingdotseq (0.411 + 0.524\tau) m b \sigma_p \quad \cdots\cdots\cdots\cdots (1)$$

Equation (1) is a calculating equation for the load capacity P of gears with contact ratio more than one and less than two. The equation can be used for plastic gears with any pressure angle.

For the contact ratio more than two, the load is shared between more than two teeth. The load sharing ratio β varies with the position on the tooth profile. Therefore it is very difficult to obtain an exact value of the ratio about teeth which can be deflected easily and are deformed by the wear. Approximate values of the ratio are given in Table 2[6].

In helical gears the treatment becomes more complicated, so the value about the equivalent spur gear is used.

For contact ratio ε above two, unlike gears with ε more than one and less than two, the load per tooth is reduced to β times (β<1). Therefore, $1/\beta$ times as heavy as total load can be imposed. The theoretical tooth meshing can not be expected because of the deformation of the tooth profile due to the deflection and the wear. So, assuming that the load sharing ratio increases by 20% and it is equal to 1.2β, $1/(1.2\beta) \fallingdotseq \beta_m$ times as heavy as loads can be imposed. This factor β_m is called here the load increasing factor. The value of β_m corresponding to ε is also given in Table 2.

Considering this factor β_m, Eq.(1) is rewritten as follows:

$$P = (0.411 + 0.524\tau) b \beta_m m \sigma_p \quad \cdots\cdots\cdots\cdots (2)$$

where, m is the normal module.

For plastic materials, the fatigue limit σ_k can be considered to be 30 ∿ 40% of their tensile strength[7]. The value of σ_k is influenced by temperature.

Figure 4 shows the relation between the tensile strength f_t of plastic materials and the temperature[2]. One third (33%) of the tensile strength is marked on the right hand ordinate. This can be considered to be the fatigue limit σ_k of plastic materials under consideration of the temperature.

The generated heat in plastic gears consists of the heat remaining on the plastic side of the frictional heat caused by the relative slide between tooth surfaces, the heat based on the hysteresis loss of viscoelastic materials and the conductive heat from steel gears whose temperature is usually higher by 5 to 15 °C than that of plastic gears. This heat is radiated into the surroundings with the rotation of gears. The temperature of gears is determined by the relation between the generated heat, the radiated heat and the temperature of surroundings.

As there are many factors which influence these heats, it is difficult to estimate the temperature of gears by calculation. Therefore, from the experimental information that the smaller the module is, the lower the tooth face temperature in operation is, it is assumed that the allowable bending fatigue stress σ_p [MPa] can be approximately given by the following equation using the tensile fatigue limit σ_k [MPa] and the factor K_O which is a function of modules. We call K_O the stress collection factor.

$$\sigma_p \leqq K_O \sigma_k \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (3)$$

Hall-Alvord[8] showed the relation between the number of repeated stress N_T and the allowable stress σ_b at the root using modules as a parameter as shown in Fig.5. Considering the value of σ_b at N_T=10^7 and m=5 in Fig.5 to be unity as a standard, the value of σ_b for other modules is shown by a symbol × in Fig.6.

Table 2 Load increasing factor to contact ratio

Contact ratio ε	Max. load sharing ratio β[6]	1.2β	Load increasing factor β_m ($\fallingdotseq 1/1.2\beta$)
1 ∿ 2	1	–	1
2 ∿ 3	0.6	0.7	1.4
3 ∿ 4	0.5	0.6	1.6

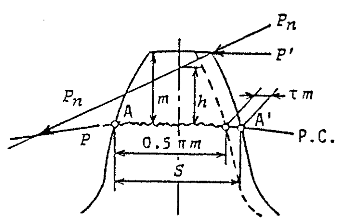

Fig.3 Tangential force on pitch circle and bending stress at pitch point

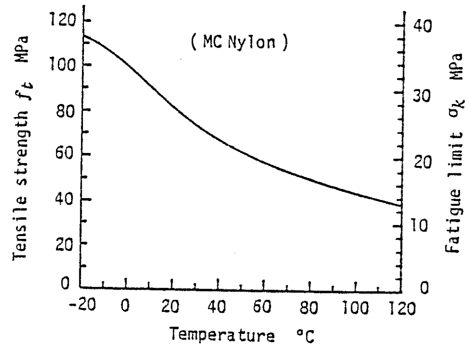

Fig.4 Fatigue limit of MC-nylon

The smaller the friction coefficient μ between tooth surfaces is, the lower the generated heat is, so, K_O can be considered to be proportional to $1/\mu$. Takanashi and Syoji[9] show that the friction coefficient μ is proportional to the relative sliding velocity V between tooth surfaces to the 0.23 power. V can be considered to be proportional to module m, therefore $K_O=Km^{-0.23}$ is obtained. When the value of the constant K is taken as $K_O=1$ for $m=5$, the following equation is obtained.

$$K_O=1.45m^{-0.23} \qquad \cdots\cdots\cdots (4)$$

Symbols ○ in Fig.6 denote the value of K_O to m. The full line in Fig.6 is drawn in consideration of the above.

An example of calculation by the new equation is given in Table 3. The specifications from m to n_p in Table 3 being given, the load capacity P of this plastic gear is 868N. But the value of σ_k at 45°C being taken, some device to keep the gear below this temperature is necessary.

This gear pair is a special one in which the tooth thickness of the plastic gear is increased by $0.3m$ and that of the steel gear is decreased by the same amount to balance the bending strengths in the pair. In the operation of this gear pair, the steel gear was cooled by a fan attached on its shaft. At the room temperature of 25°C, the plastic gear temperature was 48°C and we had $N_T=10^7$.

4. Conclusions

By the analysis of the results of sixty-two experiments about plastic gears for power transmission, the actual state of the failures of plastic gears is shown and a new equation for calculating the load capacity is proposed. A summary of this research is as follows:

(1) In plastic gears for power transmission, sometimes the failure took place at the early stage of the operation and at other stage the life of 10^7 rev was obtained. Thus, the life ranged very widely.

(2) 92% of the failures were fractures. Fractures from near the pitch point were 62%, those from near the inner worst point were 30% and scorings or pittings accounted for 8%.

(3) A new equation for calculating the load capacity using the bending stress at the pitch point, where many of the fractures take place, is proposed.

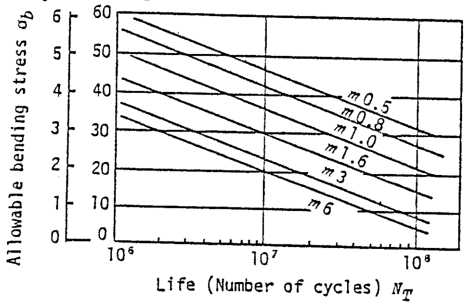

Fig.5 Allowable bending stress of polyacetal gears

(4) A method for estimating by use of modules the allowable bending fatigue stress, which is greatly influenced by the temperature, is demonstrated.

(5) For stabilizing the life, it is necessary to estimate the running temperature of gears in design and to manage the operation such as to keep the actual running temperature below the estimated one.

We are indebted to Nippon Polypenco Company for providing the plastic material.

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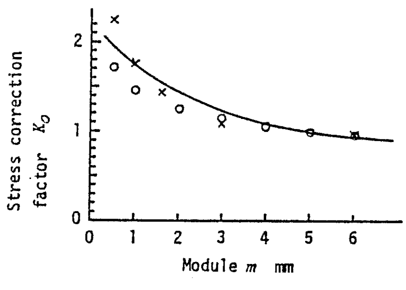

Fig.6 Stress correction factor

Table 3 Example

Item	Value	Memo.
m	2.25 mm	Given spec.
$Z_p : Z_g$	60 : 60	
α_c	14.5°	(Running
b	16.5 mm	temp. 45°C)
τ	0.3	
n_p	755 rpm	
ε	2.2	Calculated
β_m	1.4	From Table 2
K_O	1.4	From Fig. 6
σ_k	21 MPa	From Fig. 4
σ_p	29.4 MPa	From Eq. (3)
P	868 N	From Eq. (1)