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Development of Plastic Gear for Power Transmission*

(Economical Methods for Increasing Load-carrying Capacity)

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Plastic gears have many merits; silent running, self-lubrication, corrosion-proof, etc; but they have many demerits too; lower load-carrying capacity, lower thermal resistance, etc. Because the plastic is expensive, the plastic gears should be designed as small as possible. Then, plastic gears which have smaller module, pressure angle of 14.5°, contact ratio above two and helical trace are proposed in order to increase economically their load capacities. The load capacity of a gear pair whose bending strengths are balanced by applying thick teeth to the plastic gear and thin teeth to the steel gear is double that of the usual gear pair having the same thick teeth.

Key Words : Gear, Plastic, Nylon, Load-carrying capacity, Capacity increase, Economical design

1. Introduction

Plastic gears have many good points; silent running, no lubrication, light weight, corrosion-proof, fit for mass production, etc. However, for power transmission, they have some weak points; low load-carrying capacity,low thermal resistance, unstable life time, etc.

Moreover, the mechanical strength of plastic is about one-third that of steel and the price per unit volume of plastic is about three times that of steel, so plastic is nine times as expensive as steel for transmitting the same power. In order that plastic gears can display their good points as power transmission elements, economic problems should be thrashed out.

How the shape and dimension of plastic gears should be selected to increase economically the load-carrying capacity is studied in this research.

The way to extend the life time of plastic gears, whose dimensions and running conditions are already fixed is given in the reference (1).

In this experiment, the driver gear is steel (S45C) and the driven gear is MC-nylon (shown in Table 1). All of gears are hobbed and their accuracies are above 4th grade in JIS. The experiment was performed at constant room temperature of 25 °C and without lubrication. The suffixes s and p mean respectively steel and plastic and the letters D and F denote respectively the driver side and the follower side.

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2. Methods to Increase the Load Capacity

2.1 Selections of module m, number of teeth z and face width b

As the plastic is weak against heat, m, z, and b should be selected such as to suppress the heat generation and to dissipate quickly the generated heat, rather than to get a low cost by a small volume.

(1) Selections of the face width b

In steel gears, the prolongation of the face width to increase the load capacity is limited for reason of end tooth bearing. However, in plastic gears, as the tooth is soft, the load may be supported by deflected teeth or sunken tooth surfaces even if the end tooth bearing arises.

As shown on the line of width ratio ψ ($=b/m$) in Table 2, three plastic gears with $\psi=10$, 20 and 30 for $m=2$ were made and they were operated under the bending stress at pitch point $\sigma_p=34.3$ MPa (3.5 kgf/mm^2) and the tangential force on pitch circle $P=28.2$ N/mm (2.9 kgf/mm). For $\psi=10$ and 20, the total revolution $N_T=10^7$ was obtained. However, for $\psi=30$, the midlde part of the face width melted at $N_T=3.1\times10^6$. This implies that the heat was accumulated in the middle part of the face width because of the low thermal conductivity of plastic. Some ringed grooves whose depth is above the whole depth of the gear tooth should be cut

Table 1 Properties of gear materials

	Items	Units	MC Nylon	S45C Steel
Physical properties	Melting point	°C	208	1450
	Specific weight		1.16	7.8
	Rockwell hardness	HRR or H_B	115∼120 HRR	201∼269 H_B
	Specific heat	J/(kg.K)	1670	460
	Heat conductivity	W/(m.K)	0.233	75.3
	Linear expansion coeff.	K^{-1}	0.9×10^{-4}	0.12×10^{-4}
Mechanical properties	Tensile strength	MPa	78∼96(23°C)	above 686
	Elongation	%	10∼50	above 17
	Young's modulus	GPa	2.94∼3.43	206
	Poisson's ratio		0.35	0.3
	Bending strength	MPa	96∼110	above 686
	Compressive strength	MPa	92∼103	above 686

on the periphery of the gear at intervals of 10th to 20th times module.

(2) Selections of the number of teeth z

In plastic gears, the number of teeth z should be made as large as possible to reduce the sliding velocity at the root of tooth and prevent the heat generation.

(3) Selections of modules m

Figure 1 shows the results of sixty-two running tests[3] which were performed for various purposes. The classifications of damages and modules are given by symbols and numbers—respectively. Gears with m=3, which belong to the group A giving $N_T \geqq 10^7$ and the group C causing damages, exist in higher stress area than gears with m=5 which belong to the groups B and D for the same life time.

From these results, it is known that small modules should be selected provided the beam strength of teeth and the surface durability permit. When using small modules, the relative sliding velocity between tooth surfaces becomes small, the tooth meshing period per tooth is shortened and the generated heat becomes low.

2.2 Adoption of the contact ratio more than two

If the contact ratio ε is set at more than two for spur gears, more than two tooth pairs share the transmitted load on the line of action and the maximum load acting on a pair of teeth reduces to 50~60 %. In this case, the load capacity can be 1.4~1.6 times at least in comparison with the case of the contact ratio being more than one and less than two. In order to get $\varepsilon \geqq 2$, the combination of α_c=14.5° and $z \geqq 35$ must be used. To get $\varepsilon \geqq 3$, helical gears can be used.

In Table 3, the standard gear ① was designed as a popular spur gear with small module, many teeth, 20°-pressure angle and 1.7-contact ratio.

Regarding the load P in the experiment ① as unity, the load magnification f shows how many times the load can be imposed in the experiment ②~⑤.

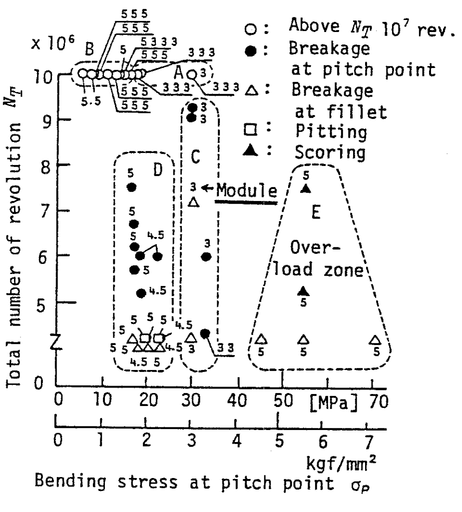

Fig.1 Relations between m and N_T

The specification of gear ② is almost the same as that of gear ①. In gear ②, ε=2.2 and the load-increasing factor[3] β_m=1.4 can be obtained by using α_c=14.5°. The factor β_m gives the magnification of load in two pair or more meshing to keep the maximum load applied to one tooth the same in one pair meshing.

Gear ② was tested by 1.4 times the tangential force on pitch circle in comparison with gear ① to get the same bending stress of teeth as gear ①. Just after the beginning of operation, the tooth surface temperature rose to 55°C for the plastic gear and to 67°C for the steel gear. So, a cooling fan was attached to the steel gear shaft. By this cooling, the tooth surface temperature dropped to 46°C and $N_T=10^7$ was obtained. 1.4 times the load capacity was obtained with the same volume as gear ①.

2.3 Adoption of helical gears with a constant contact line length

In helical gears, ε can be increased by increasing the face width b. When b is selected as integral multiple of the axial pitch $t_a=\pi m_n/\sin\beta_o$, the length of the simultaneous contact line on the tooth surface becomes constant and the fluctuation of the tooth deflection (the dynamic load) decreases. Consequently, an increase of the load capacity may be expected as well as silent operation.

Table 2 Relations between the limit of ψ and m--z

Room temp. 25°C * MC-Nylon

Items		Ratio ψ of tooth width	Small m large z	Medium m small z
Module	m	2	3 ○	5 ●
Pressure angle	α_c	20°	20°	20°
Number of teeth	z	67 : 68	30 : 60	17 : 37
Driver : Follower		D : F	D : F	D : F
Diameter of P.C.	D_0	134 : 136	90 : 180	85 : 185
Tooth width (Width ratio)	b (ψ)	60 : 20 (10) 60 : 40 (20) 60 : 60 (30)	10 (3.3)	10 (2)
Gear material		S45C : MCN*	MCN*: S45C	MCN*: S45C
Gear accuracy		Hobbed JIS 4	Same to left	Same to left
Transmission force	P_n	28.2 N/mm	31 ~ 38	51 ~ 62
Bending stress	σ_p	34.3 MPa	31 ~ 37	25 ~ 30
Number of rpm	n	761 : 750	1000 , 1500	1000 , 1500
Integrated rev.	N_T	10^7(ψ10) 10^7(ψ20) Scoring (ψ30)	(4 ~ 10) x 10^6	(2 ~ 10) x 10^6

Table 3 Methods for increasing load capacity and their effects

Tooth width of steel gears : 18.5 mm , Steel gear : Driver , Room temp. 25°C

Experiment purpose		① Standard	② Large ε	③ Const. B	④ Thick tooth τ	⑤ B and τ
Sorts of gear		Spur gear	Spur gear	Helical g.	Spur gear	Helical g.
Normal module	m	2.25	2.25	2.25	2.25	2.25
Pressure angle	α_c	20°	14.5°	14.5°	14.5°	14.5°
Number of teeth	z	60 : 60	60 : 60	54 : 54	60 : 60	54 : 54
Whole depth	h	2.25 m	2.25 m	2.25 m	2.25 m	2.25 m
Helix angle	β_o	0°	0°	25.842°	0°	25.842°
Face width	b	16.5 mm	16.5	16.5	16.5	16.5 mm
Tooth thickness coeff.	τ	0	0	0	0.3	0.3
Contact ratio	ε	1.7	2.2	3.0	2.2	3.0
Load-increasing factor	β_m	1.0	1.4	1.6	1.4	1.6
Volume of gear blank	V	252 cm³	252	252	252	252 cm³
Number of rpm	n	755	755	755	755	755
Transmission power	H	2.40 Kw	3.36	3.83	4.64	5.30 Kw
Load on P. circle	P	449 N	628	718	868	992 N
Bending stress	σ_p	29.4 MPa	29.4	29.4	29.4	29.4 MPa
Load maltiples	f	1.0	1.4	1.6	1.9	2.2
Fan for cooling			Fan	Fan	Fan	Fan
Running-in oil				Used	Used	Used
Surface temp.	°C	41	46	54	48	43
Noise level(Dist.15 cm)	dB	82	84	85	86	84
Lifetime [rev.]	N_T	Over 10^7	〃 10^7	6.7 x 10^6	〃 10^7	Over 10^7

In Table 4, the calculating equations of the dimension and the load capacity are shown of a helical gear, which has the specification shown by the symbols surrounded by circles, z, β_o, b and tooth thickness coefficient τ (explained in section 2.4), and which is cut by the hob having the specification shown also by the symbols surrounded by circles, normal module m, cutter pressure angle α_c.

In Table 3, helical gear ③ has m=2.25 and α_c=14.5°. Furthermore, it is designed for β_o=25.842°, the transverse module m_s=2.5 and z=54 to make the pitch radius r_{os} coincide with that in gear ①. The face width b=16.5 mm is taken almost equal to the axial pitch t_a=16.2 mm. Consequently, the transverse contact ratio ε_s=2.0, the overlap ratio ε_p=1.0 and the total contact ratio ε= 3.0 are obtained. By substituting τ=0, β_m=1.6 (for ε=3 in Table 4) and σ_p=29.4 MPa (3kgf/mm²) into the equation for $\overline{P}$, $\overline{P}$ was obtained first and then P=718 N was calculated. In this case, the load capacity magnification becomes f= 718/499=1.6 in comparison with gear ①.

As a higher heat generation than in the experiment ② was forecast in this experiment, fans were attached to both gears but the tooth surface temperature could not be kept under 50°C. So, the lubricating oil for running-in was spread thinly on the tooth surface with a brush.

The slight lubrication showed a great effect and the gear temperature dropped to 42 °C. As the effect of the lubrication remained for a few hours, the oil was applied several times at intervals of six hours. After that, the oil was wiped off by using alcohol and the running test was continued. Then, the tooth temperature did not drop below 54 °C and breakages took place at N_T=6.7×10⁶ rev. If the slight lubrication had been continued, the life time N_T=10⁷ rev might have been achieved.

Table 4 Equations for load capacity of helical gears [3]

Module	$m_s = m / \cos\beta_o$
Pressure angle	$\tan\alpha_{os} = \tan\alpha_c / \cos\beta_o$
Number of teeth	Z
Helix angle (PC)	β_o
Whole depth	$h = (2+k)m$
Face width	b
Rad. of pitch circle	$r_{os} = 0.5 m_s Z$ [mm]
Rad. of add. circle	$r_k = r_{os} + m$ [mm]
Rad. of base circle	$r_g = r_{os}\cos\alpha_{os}$ [mm]
Transverse pitch	$t_{os} = \pi m_s$ [mm]
Normal pitch	$t_{es} = t_{os}\cos\alpha_{os}$ [mm]
Helix angle on BC	$\tan\beta_g = \tan\beta_o \cos\alpha_{os}$
Axial pitch	$t_a = t_{es} / \tan\beta_g$
Contact length a : Center distance	$d = \sqrt{r_{k_1}^2 - r_{g_1}^2} + \sqrt{r_{k_2}^2 - r_{g_2}^2} - a\sin\alpha_{os}$
Transv. cont. ratio	$\varepsilon_s = d / t_{es}$
Over-lap ratio	$\varepsilon_p = b / t_a$
Contact ratio	$\varepsilon = \varepsilon_s + \varepsilon_p$
Load-increasing factor	β_m = 1 for ε = 1∼2 1.4 ε = 2∼3 1.6 ε = 3∼4
Transmission power	H [Kw]
Number of rpm	n [rpm]
Pitch circle vel.	$v_s = 2\pi r_{os} n / 60000$ [m/s]
Peripheral force (PC)	$P = 10^3 \cdot H / v_s$ [N]
Equivalent spur gear	
Peripheral force	$P = \overline{P}\cos\beta_o$ [N]
Load capacity (PC)	$\overline{P} = (0.411 + 0.524\tau) \cdot b\beta_m m \sigma_p / \cos\beta_o$ [N]
Bending stress of tooth on pitch point	$\sigma_p = P / \{(0.411+0.524\tau) \cdot b\beta_m m\}$ [MPa]

2.4 The improvement by balancing the bending stress

The idea[4] that the load capacity can be increased by the combination of a thick tooth plastic gear and a thin tooth steel gear as shown in Fig.2(a) is realized in the gear pair with the pressure angle of 14.5°.

Assuming that the weakest section of the gear teeth exists at the pitch point for the plastic gear teeth, at this point many breakages occur, and at the effective dedendum point for the steel gear teeth, and that P is applied at the tooth tip, as shown in Fig.2(b), τ for balancing the bending strength in rack tooth profiles can be calculated approximately by the following equations.

$$S_p = \zeta m, \quad \zeta = 0.5\pi + \tau$$

$$S_s = \xi m, \quad \xi = 0.5\pi - \tau + 2\tan\alpha_c$$

$$P m = \sigma_p b S_p^2 / 6, \quad 2Pm = \sigma_s b S_s^2 / 6$$

$$\zeta / \xi = \sqrt{\sigma_s / (2\sigma_p)} = k \qquad \text{-----(1)}$$

$$\tau = \frac{0.5\pi(k-1) + 2k\tan\alpha_c}{k+1} \qquad \text{-----(2)}$$

The greater the value of τ is, the more sharpened the tip of steel gears is. So, there exists a limit value of τ. When the allowable minimum value of the tooth tip thickness $\widehat{S}$ mm is given, τ can be calculated as follows.

$$R_o = 0.5 m Z_s, \quad R_k = R_o + m, \quad R_g = R_o\cos\alpha_c$$

$$\phi_k = 0.5\pi / Z_s + \mathrm{inv}\,\alpha_c - \mathrm{inv}\cos^{-1}(R_g / R_k)$$

$$\widehat{S_k} = 2\phi_k R_k, \quad \widehat{S_k} - \widehat{S} \doteqdot \tau(R_k / R_o) m$$

$$\tau = (\widehat{S_k} - \widehat{S})(R_o / R_k) / m \qquad \text{------(3)}$$

As shown in Fig.2(c), the tip of the hob teeth becomes an arc (full round) at a certain value of τ. Then the thickness of the hob tooth can not be decreased any more.

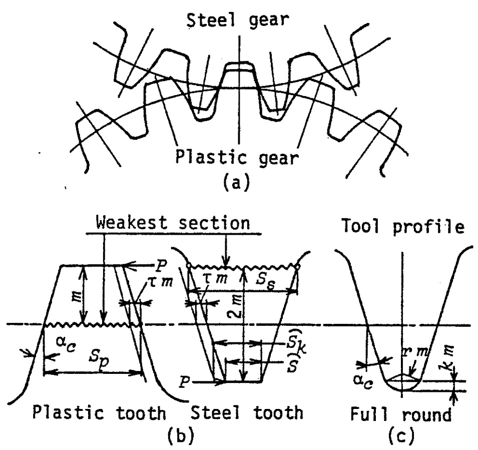

Fig.2 Methods for balancing the bending strength

In the normal section of hob teeth, denoting the radius of the full round tip and the bottom clearance by rm and km respectively, the limit value of τ is as follows.

$$r = k / (1 - \sin\alpha_c)$$

$$\tau = 0.5\pi - 2\tan\alpha_c - 2r\cos\alpha_c \quad \text{-------(4)}$$

In a 20°-pressure angle, $\tau \leqq 0.13$ being obtained from Eq.(4), the improvement of the load capacity can not be expected. At σ_p= 29.4 MPa (3kgf/mm²) and σ_s=58.8~117.6 MPa (6 ~12 kgf/mm²), τ is 0.26~0.30 from Eq.(2). Assuming that $z_s \geqq 40$ and $\widehat{S}=0.6m$ ($\widehat{S}$=1.35mm for m=2.25), $\tau \leqq 0.33$ is obtained from Eq.(3). At α_c=14.5° and k=0.25, $\tau \leqq 0.41$ is obtained from Eq.(4). In this research, in consideration of the above results, τ is selected as 0.3.

The hobbing of the gears with a thin or thick tooth thickness was performed as follows. At first, both the plastic and the steel gear were cut by the hob with a tooth thickness smaller by τm than the standard tooth thickness of $0.5\tau m$ and then only the steel gear was cut additionally by shifting the hob along the hob axis by ΔS shown in Eq.(5) to decrease the tooth thickness.

$$\Delta S = (2\tau + c)\, m / \cos\gamma \quad \text{-------- (5)}$$

where, c:circumferential backlash on a pitch circle, γ:lead angle of hob. The gear ④ in Table 3 was cut by the hob with m=2.25, τ= 0.3 (decreased tooth thickness) and γ=2.412° and additionally the steel gear was cut by shifting the hob by ΔS=1.576 mm to give the backlash c=0.1.

2.5 The method of improving the load capacity through increased tooth thickness

When the above mentioned methods for increasing the load capacity are used independently or in combinations, additive or multiplied effects appear. For evaluations of these effects, Eq.(6)[3] can be used.

$$P = (0.411 + 0.524\tau)\, b\, \beta_m\, m\, \sigma_p$$

$$H = \frac{(0.411 + 0.524\tau)\, Z_p\, n_p\, b\, \beta_m\, \sigma_p\, m^2}{267.3^3 \cos\beta_o} \quad \text{-------- (6)}$$

For the gears ④ and ⑤ in Table 3, 1.9 times and 2.2 times load capacities were expected respectively by increasing the tooth thickness by 0.3m in comparison with the gears ② and ③.

Cooling fans were attached to these gear pairs and a small amount of running-in oil was supplying every six hours to prevent the heat generation. In the gear ④, the tooth temperature was stabilized by supplying the oil several times, after that, $N_T=10^7$ rev was achieved without lubrication. In the gear ⑤, when oil supply was stopped, the tooth temperature rose to 60°C or more, so the oil supply was continued till $N_T=10^7$ rev.

3. Conclusions

The improvement of the load capacity of plastic gears can be achieved by good design and suitable operation management without employing large sized gears.

(1) As the temperature rise is suppressed by using small modules, many teeth and a wide face width, the load capacity can be increased.

(2) In wide face widths, it is recommended to cut the ringed groove along the axis at intervals of 10~20m on the periphery of the gear.

(3) When α_c=14.5° and $z \geqq 35$ are combined, ε= 2~3 is obtained. So, 1.4 times the load capacity can be obtained in comparison with that in the case of ε=1~2.

(4) In helical gears, which have a constant contact line length obtained by selecting the face width as an integral multiple of the axial pitch, 1.6 times the load capacity can be expected.

(5) The design procedure of a gear pair with the bending strength balanced by thick plastic teeth and thin steel teeth is shown. At α_c=14.5°, τm=0.3m is optimal. In this case, 2 times the load capacity can be obtained.

(6) In plastic gears, the effect of lubrication is kept for a few hours with a small amount of oil.

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