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DEVELOPEMENT OF PLASTIC GEARS FOR POWER TRANSMISSION*

(Power Transmission Mechanism of Plastic Gears)

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The plastic gears excel in the self-lubrication, the low noise and others over the steel ones. But they are not suitable for power transmission because of their low load carrying capacity and short lifetime.

In the present paper, the power transmission mechanism of the plastic gears is clarified theoretically and experimentally. And then the causes for the melting and the partially large wear which occurs on the tooth surface are analyzed through the mechanism.

As a result of these analyses, the plastic gears, whose contact ratio is more than two and whose tooth profile deflected by the load agrees with the normal involute, are recommended in order to increase the load capacity and the life time.

Key words : Gear, Plastic, Nylon, Tooth deflection, Power transmission mechanism, Contact ratio, Deformed tooth profile

1. Introduction

The plastic gears have many characteristics lacking in the metallic ones, which are lightness, higher corrosion resistance, self-lubrication and others. But their load carrying capacity and the wear resistance are inferior to those of the metallic ones. These characteristics become worse when the temperature is increased.

Because the plastic gears have many unknown properties, they are used only under the very light load, such as for the office machines, the audio equipment, the moving toys, etc. and have not been used as gears for the power transmission which is the most important function of gears.

If the plastic gears are made available for power transmission without losing their special features, many demands would be expected for them in other fields , such as the food machines, the chemical ones, the machines required to be corrosion-proof and the silent ones.

But, because of the preconception that they are not suitable for power transmission, few trials of use for such a purpose have been reported. In the present research, MC-nylon, a trade name of Nippon Polypenco Company, was selected for this experiment, because it is low in price and suitable for mass production.

One of the authors, N.Tsukamoto, has answered the questions which are raised when the plastic gears are used for power transmission, through many experiments (1) ∿ (5).

In this paper, through analysis of the conjugation mechanism of soft gears, the methods for increasing the load capacity and the life time of plastic gears are sought theoretically and experimentally.

In Table 1, the properties of plastic (MC-nylon)[6] used in this experiment are shown in comparison with those of steel.

2. Tooth Deflection and Profile Deformation of Plastic Gear

In order to make clear the power transmission mechanism of soft gears, the spring constants of plastic gear tooth are calculated, and then the deflection of plastic teeth and the rotational delay of steel gear due to the deflection are investigated.

The details of a gear pair used in the calculation examples and the experiments are shown in Table 2. The gear with $Z_1 = 17$ is made of plastic and used as driver. The gear with $Z_2 = 37$ is made of steel and used as follower.

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 ~ 120HRR	201~269H_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

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2.1 Spring constants and deflections of soft gear teeth

In order to analyze the meshing mechanism of soft gear, the spring constants at different loading positions f, i, p, w and t on the tooth profile (Fig.1) must be calculated for different combinations of pair gears.

The methods for calculating the spring constant of steel gear teeth (6) have been reported in some articles, but there are few examples for the plastic gear teeth. In the present investigation, Weber-Banaschek's method (7), Eq.(1), developed for calculating the deflection of the steel gear teeth was used for the plastic gear teeth.

$$\delta_c = K_c(P_n/E),$$

$$K_c = \cos^2 u \left\{ \frac{12(1-\nu^2)}{8} \int_0^h \frac{(h-y)^2}{x^3} dy + 1.2(1+\nu) \int_0^h \frac{1}{x} dy + \frac{(1-\nu^2)}{2} \tan^2 u \int_0^h \frac{1}{x} dy \right\}$$

$$\delta_r = K_r(P_n/E)$$

$$K_r = \cos^2 u \left\{ \frac{18(1-\nu^2)}{\pi} \left(\frac{h}{s}\right)^2 + 2(1+\nu)(1-2\nu)\frac{h}{s} + \frac{4.8(1-\nu^2)}{\pi}(1+\tan^2 u) \right\}$$

$$\delta_v = K_v \frac{P_n}{E}$$

$K_v = 2.13$: Plastic , 2.65 : Steel

$$\delta = \delta_c + \delta_r + \delta_v = K\frac{P_n}{E},\ K = K_c + K_r + K_v \qquad \cdots\cdots\cdots\cdots (1)$$

Table 2 Gears used in calculations and experiments

Items	Symbols	Dimensions
Module	m	5 mm
Pressure angle	α_c	20°
Plastic gear	Z_1	17 (Driver)
Steel gear	Z_2	37 (Follower)
Tooth width	b	10 mm
Normal pitch	t_n	14.761 mm
Contact ratio		1.61

Hobbed gears, JIS accuracy 4 grade,
Without lubricating fluid,
Backlash 0.6 mm

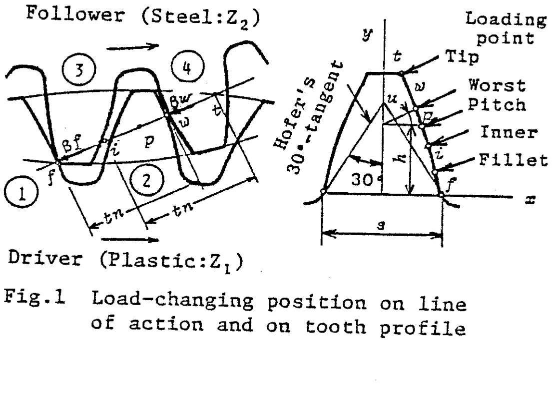

Fig.1 Load-changing position on line of action and on tooth profile

The symbols used in Eq.(1) are explained in Table 3, where δ_c is the deflection of tooth body by bending, shearing and radial component, δ_r is the deflection due to displacement and deformation at the tooth root, δ_v is the depth of elastic deformation on the tooth surface, which can be considered a kind of deflection. There are many theories for calculating δ_v, and therefore the spring constant K_v will be decided later by experiment.

The critical section s obtained by Hofer's 30 degree tangent method is chosen as a boundary of the tooth itself and its base, and it is made X axis. Poisson's ratio ν = 0.35 is used for the plastic.

The starting point f of tooth meshing at the tooth root is affected by the number of teeth of a mating gear. However, the spring constant at the starting point is approximately calculated assuming that the working pressure angle u = 0° and the radial load-position $R = R_o - m$ which is on the effective root circle.

The values of ($K_C + K_r$) = ($\delta_c + \delta_r$) / (P_n / E) calculated in such a way are given in Table 4. In this paper, $K = K_C + K_r + K_v$ is called the spring constant, although it is the tooth compliance.

The constant K_v relates to the depth of an elastic concavity produced by the load on the tooth surface. There are many theories for calculating K_v, e.g., it is reported that K_v-values obtained under the same condition are different by a factor of about ten (7).

The mean value of K_v is about 2.65 for steel as seen in Fig.2. This value was obtained experimentally using a pair of steel cylinders (S45C, Rmax 0.8 μm) with a length of l = 76 mm and radius R in the range of 21 ∿ ∞ mm. Normal loads in the range P_n = 78.4 ∿ 264.6 N/mm (8 ∿ 27 kgf/mm) are applied to the cylinder considering their relative curvature. K_v can be calculated from the approach $2\delta_v$ of two cylinders. A mean value of K_v = 2.65 is decided graphically from the K_v diagram of Fig.2.

The constant K_v for plastic gears is 2.13, which is obtained using a method shown in Fig.3. This is influenced by the kind of materials, the surface roughness of plastic cylinders used in the experiments, and especially by the temperature. Because the theoretically reliable value of k_v can not be obtained, such an experimental method is used to determine .

Table 3 Symbols used in Eq.(1)

δ	: Total deflection of tooth in mm
δ_c	: Deflection of tooth body in mm
δ_r	: Deformation in tooth fillet in mm
δ_v	: Concave on contact surface in mm
P_n	: Load along tooth trace in N/mm
E	: Young's modulus in MPa
ν	: Poisson's ratio
u	: Working pressure angle in degrees
h	: Loading height on tooth center line in mm
S	: Critical thickness on tooth fillet in mm
K, K_c, K_r, K_v	: Spring constants
x - axis	: Critical section line of fillet
y - axis	: Center line of tooth form

From the measurements of the concave depth for three cases a, b and c shown in Fig.3, it is found that the depths obtained in the cases b and c are 1.5 times the one in the case a.

Accordingly, when the tooth tip or the fillet is in contact, k_v = 2.13 x 1.5 will come close to the actual values. Spring constant K can be obtained by adding k_v to (K_c + K_r) given in Table 4. The values of (K_c + K_r) calculated by using Poisson's ratio ν = 0.3 (Steel) differ only several percent as compared with those of plastic as shown in Table 4. Hence, it will be understood that the value obtained by adding K_v = 2.65 to (K_c + K_r) for plastic shown in Table 4 can be used apporoximately as the spring constant K for steel.

2.2 Tooth deflection of soft gear and rotational delay of steel gear

The tooth deflection of a plastic gear with m = 5 and Z_1 = 17 are tabulated in Table 5, where P_n = 49 N/mm (5kgf/mm) is loaded on the load change point f, i, p, w or t (Fig.1) in meshing with a steel gear with Z_2 = 37.

Compared with the deflection of the plastic gear tooth, that of the steel one is very small and is only about 1.5 percent of the plastic one. Therefore the steel tooth deflection can be neglected.

Figure 4 shows schematically an apparatus used for measuring the plastic tooth deflection and the rotational delay of the

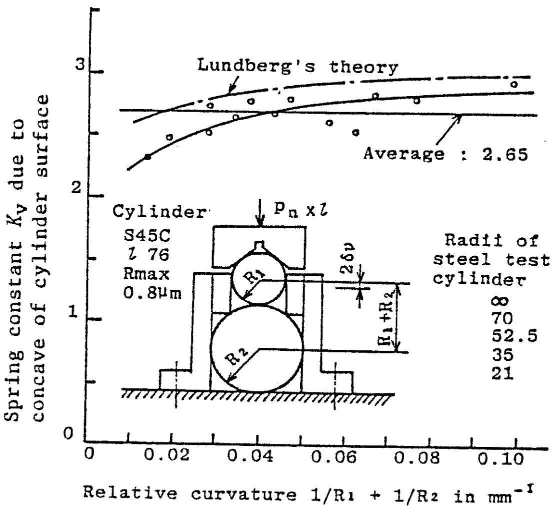

Fig.2 Approach δ_v of two steel cylinders due to compressive load

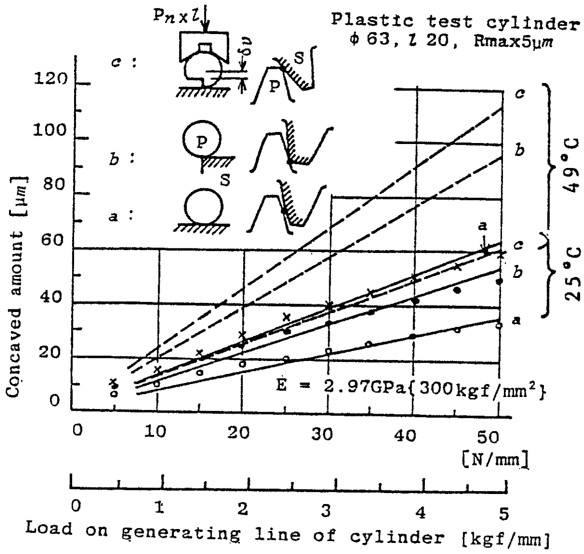

Fig.3 Approach δ_v of two plastic cylinders

Table 4 Spring constants $K_c+K_r=(\delta_c+\delta_r)/(P_n/E)$ of standard gear hobbed with pressure angle 20°, tip round 0.375m and clearance 0.25m hob

Z_1	Tip t	Z_2 U.Wst(Upper Worst Loading Position) w Z_2 17	22	30	34	37	57	60	75	100	150	300	Pitch p	I.Wst i	Root f	Z_1
17	13.315	6.198	5.963	5.722	5.636	5.583	5.353	5.331	5.242	5.149	5.052	4.949	4.797	3.982	1.532	17
18	13.157	6.075	5.843	5.605	5.520	5.467	5.240	5.217	5.129	5.037	4.940	4.837	4.686	3.879	1.523	18
19	13.022	5.969	5.740	5.504	5.420	5.368	5.142	5.120	5.032	4.940	4.843	4.741	4.590	3.794	1.517	19
20	12.907	5.876	5.649	5.416	5.333	5.281	5.057	5.034	4.947	4.856	4.759	4.658	4.506	3.721	1.510	20
21	12.807	5.794	5.570	5.338	5.256	5.205	4.982	4.960	4.873	4.782	4.686	4.584	4.433	3.660	1.506	21
22	12.719	5.722	5.500	5.270	5.188	5.137	4.915	4.893	4.807	4.716	4.620	4.519	4.368	3.607	1.502	22
24	12.574	5.600	5.381	5.154	5.074	5.023	4.803	4.781	4.696	4.605	4.510	4.409	4.258	3.523	1.494	24
26	12.459	5.500	5.284	5.060	4.980	4.930	4.712	4.690	4.605	4.515	4.420	4.319	4.169	3.459	1.488	26
28	12.365	5.418	5.204	4.982	4.903	4.853	4.636	4.615	4.530	4.441	4.346	4.245	4.095	3.409	1.483	28
30	12.289	5.348	5.137	4.916	4.837	4.788	4.573	4.551	4.467	4.378	4.283	4.183	4.033	3.370	1.479	30
34	12.171	5.237	5.029	4.811	4.734	4.684	4.471	4.450	4.366	4.278	4.184	4.084	3.934	3.313	1.473	34
37	12.104	5.172	4.966	4.750	4.673	4.624	4.412	4.391	4.307	4.219	4.126	4.026	3.876	3.284	1.470	37
43	11.992	5.067	4.864	4.651	4.575	4.526	4.317	4.296	4.213	4.126	4.033	3.933	3.784	3.243	1.465	43
50	11.890	4.975	4.774	4.564	4.489	4.441	4.233	4.213	4.131	4.044	3.951	3.853	3.704	3.213	1.462	50
57	11.829	4.911	4.713	4.504	4.430	4.382	4.176	4.155	4.074	3.987	3.895	3.797	3.649	3.198	1.460	57
60	11.809	4.889	4.692	4.484	4.409	4.362	4.156	4.136	4.054	3.968	3.876	3.778	3.630	3.195	1.459	60
75	11.746	4.811	4.616	4.410	4.336	4.289	4.085	4.065	3.984	3.898	3.807	3.709	3.561	3.187	1.458	75
100	11.698	4.738	4.545	4.342	4.268	4.222	4.019	3.999	3.918	3.833	3.742	3.644	3.497	3.192	1.456	100
150	11.664	4.670	4.479	4.278	4.205	4.159	3.958	3.937	3.857	3.772	3.682	3.584	3.437	3.209	1.455	150
300	11.645	4.608	4.419	4.219	4.147	4.101	3.901	3.881	3.801	3.716	3.626	3.529	3.381	3.241	1.454	300

steel gear. In the case of the root conjugation, for example, two gears are set to make contact each other at their pitch points, and the plastic gear is rotated by an angle of $\theta f = \cos^{-1}(Rg_1 / R_{o1}) - \cos^{-1}(Rg_1 / Rf)$, and then its shaft is fixed.

Then, the steel gear is rotated until its tooth surface contacts slightly with the tooth surface of the plastic one. Under such a state, a torque $WL = Rg_2 P_n b$ is applied about the steel gear axis and then a dial gauge reading Δs is taken quickly when the rotation of an indicator needle stops.

In this case, the revolution angle $\Delta\theta$ of the steel gear is given by $\tan^{-1}(\Delta s / D)$ and this corresponds to the rotational delay of the steel gear. $Rg_2\Delta\theta$ is the deflection of the plastic one. Full line in Fig.5 shows the plastic tooth deflection tabulated in Table 5, and the mark ∘ shows the deflection measured by the method in Fig.4. Because the values obtained from the measurement agree fairly well with those from the calculation, the deflection in this research can be called reliable.

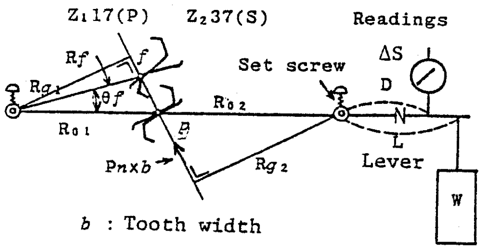

Fig.4 Method for measuring tooth deflection and rotational delay

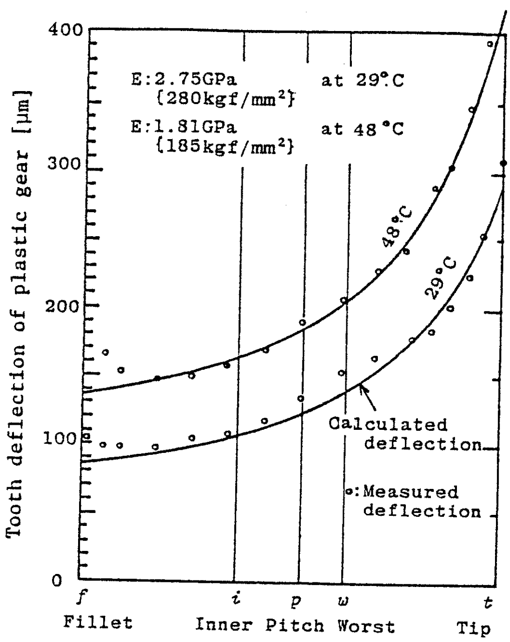

Fig.5 Theoretical and experimental deflections

2.3 Deformation of tooth profile and displacement of contacting point due to tooth deflection

When the plastic tooth is bent by load, its tooth profile deforms, its pressure angle increases and its contact point on the tooth surface displaces from the line of action. As shown in Fig.6, the plastic tooth contacts at a point B which is on the line of action and also on the circle with a radius of R_b. When the tooth bends by δ due to loading, the tooth surface displaces from B to C on the line of action.

$$\omega = \angle AO_1C = AO_1B + \angle BO_1C = \mathrm{inv}\cos^{-1}(Rg_1/R_b) + \cos^{-1}(Rg_1/R_b) - \tan^{-1}\{\sqrt{R_b^2 - Rg_1^2} - \delta)/R_{g1}\} \quad \cdots\cdots\cdots\cdots (2)$$

Table 5 Calculations of tooth deflection using Table 4 $\delta = K P_n/E$

S.C : Spring constant, L.P : Loading position

Z	L.P / S.C	Tip t	U.Worst w	Pitch p	I.Worst i	Root f
Z_1 (P) 17	$Kc + Kr$	13.315	5.583	4.797	3.982	1.532
	Kv	3.192	2.128	2.128	2.128	3.192
	K	16.507	7.711	6.925	6.110	4.724
	δ [μm]	δt 295	δw 138	δp 124	δi 109	δf 84
Z_2 (S) 37	$Kc + Kr$	12.104	5.172	3.876	3.284	1.470
	Kv	3.975	2.650	2.650	2.650	3.975
	K	6.079	7.822	6.526	5.934	5.445
	δ [μm]	δt 3.8	δw 1.9	δp 1.6	δi 1.4	δf 1.3

E : Plastic 2740 MPa (280 kgf/mm^2)
Steel 205800 MPa (21000 kgf/mm^2)

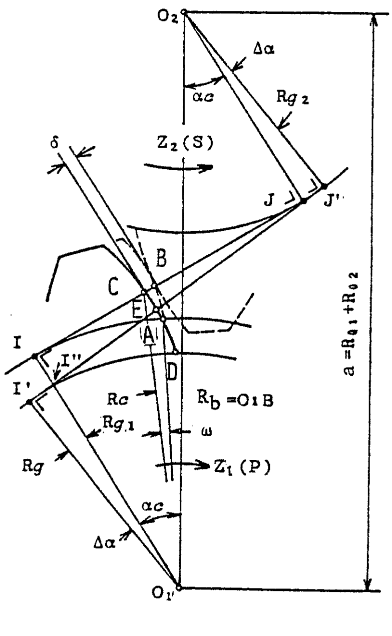

Fig.6 Deformation of tooth profile and displacement of contact point due to tooth deflection

The deformed tooth profile can be considered an imaginary involute which starts from point D on the new base circle and passes through the points A and C. When Rg is a radius of the new base circle and $\Delta\alpha$ is an increase in pressure angle of the new involute, the relations among R_g, $\Delta\alpha$, $w = \angle AO_1C$ and $R_C = O_1C$ are as follows, where α_c is the tool pressure angle.

$$\left.\begin{aligned} &R_g = 0.5 m Z_1 \cos(\alpha_c + \Delta\alpha) \\ &\omega \fallingdotseq \angle DO_1C - \angle DO_1A \\ &\quad = \mathrm{inv}\cos^{-1}(R_g/R_c) - \mathrm{inv}\cos^{-1}(R_g/R_{g_1}) \\ &R_c = \sqrt{(\sqrt{R_b^2 + R_{g_1}^2} - \delta)^2 + R_{g_1}^2} \end{aligned}\right\} \quad \cdots\cdots\cdots\cdots(3)$$

ω_1 is calculated by substituting R_b and δ into Eq.(2). When ω_2 obtained by substituting the same R_b, δ and an arbitrary $\Delta\alpha$ into Eq.(3) becomes equal to ω_1, then the values of $\Delta\alpha$ and Rg obtained are the increase of pressure angle and the radius of the new base circle, respectively.

Supposing that the contact point B on the tooth surface moves to an intersection E of the deformed tooth profile and the inner common tangent I'J' of the base circles Rg and R_{g2}, the displacement CE is calculated approximately through ΔJII".

$$CE = (R_{g_1} - R_g)(a \sin\alpha_c - \sqrt{R_c^2 - R_{g_1}^2})/(a \sin\alpha_c) \quad \cdots\cdots\cdots\cdots(4)$$

In the example of Table 2, the deflection δ for the pitch point contact is 0.124 mm as shown in Table 5. In this case, $\Delta\alpha$ = 1.69°, Rg = 39.5 mm and CE = 0.31 mm.

2.4 Factors affecting tooth deflection

The measured results of plastic tooth deflection are poor in their reproducibility even when the measurements are conducted with full care. This is caused by the following facts, that is, the mechanical properties of plastic shown in Table 1 change depending on the directional qualities of materials, the temperature, the humidity, the loading velocity, the surface hardening and many other factors.

A full line in Fig.7 shows Young's modulus E measured at 20° and 50°C according to the JIS K 6810-1977 testing method. A broken line shows the dynamic elastic modulus E measured under a vibration of 3.5 Hz and loads ranging from 18.7 to 15.5 N using a viscoelasticity-spectrometer VIBRON. Values of Young's modulus differ depending on the above factors. E-values measured by the JIS method are suitable for the purpose of present investigation.

The mark ° in Fig.5 shows the tooth deflection measured at 29° and 48°C. The deflection at 48°C is 1.5 times the one at 29°C and this value is equal to a ratio 2.75/1.81 ≒ 1.5 of Young's moduli at 48°C and 29°C. The concave depth in Fig.3 also is influenced by the temperature.

In this paper, the effect of temperature is considered at first because the temperature is the most important factor. However, it is necessary to investigate many problems caused by other factors, such as the moisture, the loading velocity, the surface hardening, the material creep, the permanent deflection of teeth and others.

As is evident from Table 5, the deflection of the plastic tooth is about ten times that of the steel one; additionally this deflection fluctuates periodically and violently due to the change of one or two pair conjugation.

The thick line curve in Fig.8 shows the measured result of plastic tooth deflection. This curve is smooth as compared with the theoretical one, and the top of curve becomes a high point (peak) due to the temperature rise. When the temperature rises further, the peak becomes lower because the tooth deflects easily.

3. Power Transmission Mechanism of Soft Gear

The dynamic conjugation of soft gear is considered through the example of a gear pair, where the driver is a plastic gear (m = 5, Z = 17) and the follower is a steel one (Z = 37).

3.1 Conjugation under dead load

As shown in Fig.1, the conjugation starts from the tooth root f of the plastic gear. In this case, the next tooth ahead meshes at its worst loading point w. When β_f and β_w are the load share ratios, we have $\beta_f + \beta_w = 1$. Neglecting the deflection of steel gear tooth, we have $\delta_f\beta_f = \delta_w\beta_w$ because the tooth root deflection $\Delta\delta_f$ must be equal to the worst loading deflection $\Delta\delta_w$, where δ_f and δ_w are already calculated as given in Table 5.

β_f, β_w, $\Delta\delta_f = \Delta\delta_w$ and $\Delta\theta_f = \Delta\theta_w$ of the rows 1 and 6 in Eq.(5) can be obtained by solving the above two equations, where suffixes f, i, p, w and t indicate respectively root, inner worst loading, pitch, outer worst loading and tip points, and $\Delta\theta$

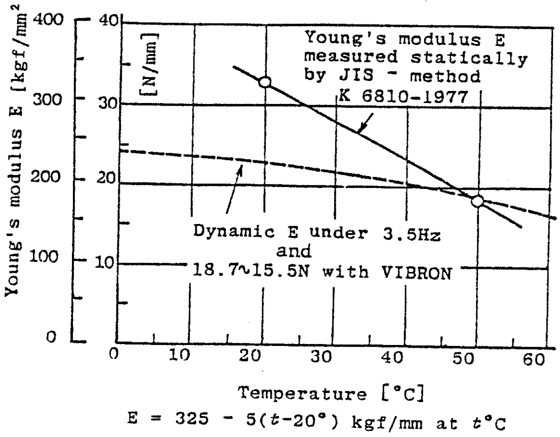

Fig.7 Young's modulus of MC-nylon according to temperature and loading velocity

is the delay angle of steel gear rotation. β, $\Delta\delta$ and $\Delta\theta$ in the other rows can be calculated in much the same way.

① Tooth root : Two pair meshing

$$\beta_f=\delta_w/(\delta_f+\delta_w),\ \Delta\delta_f=\delta_f\beta_f,\ \Delta\theta_f=\Delta\delta_f/R_{g2}$$

② Inner worst : Two pair meshing

$$\beta_i=\delta_t/(\delta_i+\delta_t),\ \Delta\delta_i=\delta_i\beta_i,\ \Delta\theta_i=\Delta\delta_i/R_{g2}$$

③ Inner worst : One pair meshing

$$\bar{\beta}_i=1,\ \Delta\bar{\delta}_i=\delta_i,\ \Delta\bar{\theta}_i=\Delta\bar{\delta}_i/R_{g2}$$

④ Pitch point : One pair meshing

$$\beta_p=1,\ \Delta\delta_p=\delta_p,\ \Delta\theta_p=\Delta\delta_p/R_{g2}$$

⑤ Outer worst : One pair meshing

$$\bar{\beta}_w=1,\ \Delta\delta_w=\delta_w,\ \Delta\bar{\theta}_w=\Delta\bar{\delta}_w/R_{g2}$$

⑥ Outer worst : Two pair meshing

$$\beta_w=1-\beta_f,\ \Delta\delta_w=\Delta\delta_f,\ \Delta\theta_w=\Delta\theta_f$$

⑦ Tooth tip : Two pair meshing

$$\beta_t=1-\beta_i,\ \Delta\delta_t=\Delta\delta_i,\ \Delta\theta_t=\Delta\theta_i \quad \cdots\cdots\cdots\cdots(5)$$

Table 6 Tooth deflection of plastic gear and rotational delay of steel gear

Loading position	Meshing pair	Load share coeff.	Deflection [μm]	Delay [min.]
f : Root	Two pair	βf 0.62	$\Delta\delta f$ 52	$\Delta\theta f$ 2.1′
i : I.Wst		βi 0.73	$\Delta\delta i$ 80	$\Delta\theta i$ 3.2′
i : I.Wst	One pair	β 1.00	$\Delta\bar{\delta} i$ 109	$\Delta\bar{\theta} i$ 4.3′
p : Pitch			$\Delta\delta p$ 124	$\Delta\theta p$ 4.9′
w : U.Wst			$\Delta\bar{\delta} w$ 138	$\Delta\bar{\theta} w$ 5.5′
w : U.Wst	Two pair	βw 0.38	$\Delta\delta w$ 52	$\Delta\theta w$ 2.1′
t : Tip		βt 0.27	$\Delta\delta t$ 80	$\Delta\theta t$ 3.2′

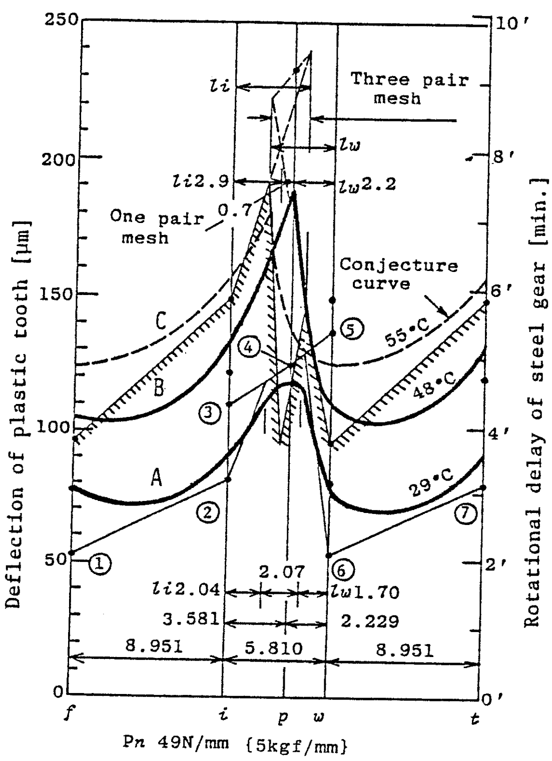

Fig.8 Deflection of plastic tooth and rotational delay of steel gear

Deflection $\Delta\delta$ and delay $\Delta\theta$ calculated from Eq.(5) using the data of Table 5 are shown in Table 6, which are plotted using numbers ①∿⑦ in Fig.8. Because the value of Young's modulus E of plastic decreases with a temperature rise, the tooth deflection increases by a reciprocal of E as curves A ∿ C shown in Fig.8.

As is evident from Table 5, the deflection of the plastic tooth is about ten times that of the steel one; additionally this deflection fluctuates periodically and violently due to the change of one or two pair conjugation.

The thick line curve in Fig.8 shows the measured result of plastic tooth deflection. This curve is smooth as compared with the theoretical one, and the top of curve becomes a high point (peak) due to the temperature rise. When the temperature rises further, the peak becomes lower because the tooth deflects easily.

3.2 Two or three pair conjugation in one pair meshing range

Numerical deflection curve②∿⑥ in the range of one pair meshing shown in Fig.8 does not agree with the measured one in their shapes. The peak of the curve becomes sharper with the temperature rise at first, and subsequently the curve becomes gradually flat. This causes a severe wear of the tooth root.

As shown in Fig.9, when the right profile of one tooth comes to a point L which is at a distance lw from the worst loading point w, then the next tooth profile comes at a position M which is at a distance lw from the starting point f of meshing. Two segments ST and SN are tan-

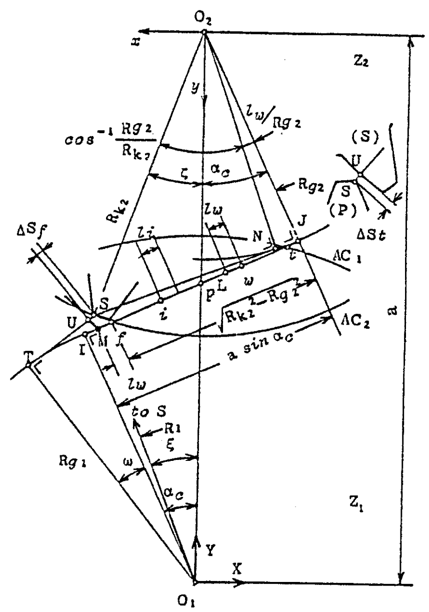

Fig.9 Two pair meshing in original one pair zone

gent from the tooth tip S to the base circles of driving and following gears respectively. Point U is an intersection of tangent ST and tooth profile.

The clearance ΔS_f = SU can be calculated from Fig.9 as follows.

$$\left.\begin{aligned}
&\zeta=\angle SO_2N-(\angle O_1O_2J-\angle NO_2J)\\
&\quad=\cos^{-1}(R_{g2}/R_{k2})-(\alpha_c-l_w/R_{g2})\\
&x=R_{k_2}\sin\zeta,\ y=R_{k_2}\cos\zeta \quad \text{: Coordinate of point S}\\
&R_1=SO_1=\sqrt{x^2+(a-y)^2}\\
&\xi=\angle SO_1O_2=\tan^{-1}\{x/(a-y)\}\\
&\omega=\angle IO_1T=\angle SO_1T+\angle SO_1O_2-\angle O_2O_1I\\
&\quad=\cos^{-1}(R_{g1}/R_1)+\xi-\alpha_c\\
&ST=\sqrt{R_1^2-R_{g1}^2}\\
&UT=\overline{MI}+\overline{IT}=a\sin\alpha_c-\sqrt{R_{k_2}^2-R_{g_2}^2}\\
&\quad-l_w+R_{g_1}\omega\\
&\Delta S_f=ST-UT
\end{aligned}\right\} \quad \cdots\cdots\cdots\cdots(6)$$

If the clearance ΔS_f calculated by substituting l_w into Eq.(6) is smaller than the tooth deflection δ_L* at the position L, the tooth tip S of steel gear cuts into the tooth surface U of plastic gear. (* : When the contact point L is near the worst point w, $\Delta\delta_w$ in Table 6 is used as δ_L ; $\delta_L = \Delta\delta p$ for L near p.)

The steel tooth tip S begins to scrape the plastic tooth surface from position L where clearance ΔS_f is equal to deflection δ_L. Similarly the clearance ΔS_t at the plastic tooth tip t can be calculated by using Eq.(6), where the symbols l_w, R_{g1}, R_{g2}, R_{k2}, xy and ΔS_f are replaced respectively with l_i, R_{g2}, R_{g1}, R_{k1}, XY and ΔS_t.

The distances l_w and l_i satisfying $\Delta S_f = \Delta\delta_w$ and $\Delta S_t = \Delta\delta_i$ in the example are 1.70 and 2.04 mm respectively. Thus, as shown at the foot of Fig.8, two pair meshing ranges increase inwards by 1.70 and 2.04 mm from both ends of one pair meshing range 5.81 mm. The remainder 2.07 mm is actual range of one pair meshing.

Curve A in Fig.8 shows the measured result for theoretical deflection ①∿⑦. The peak-shaped curve B occurs in the case of $l_i + l_w = \overline{iw}$ where the plastic tooth is easy to bend. The curve C will occur in the case of ($l_i + l_w$)> $\overline{iw}$ because three pair teeth engage at the same time in the range ($l_i + l_w - \overline{iw}$).

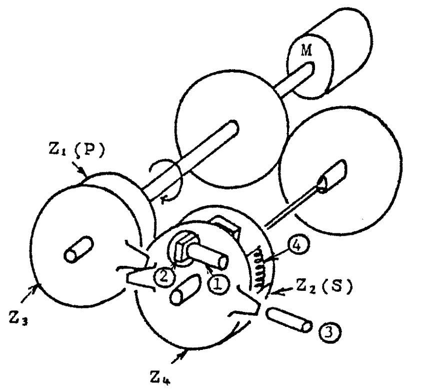

Fig.10 Method of measuring rotational delay of steel gear

3.3 Dynamic conjugation of soft gear

A method of measuring the rotational delay of the steel gear during power transmission is illustrated in Fig.10, where a power-circulation type gear test machine is used. Z_1 and Z_2 are test gears made respectively of plastic and steel. Steel gears Z_3 and Z_4 are the same size as Z_1 and Z_2 respectively.

The gears Z_3 and Z_1 are fixed with a key after agreeing precisely with these tooth profiles in the direction of tooth trace. In order to remove a backlash between the gears Z_3 and Z_4, the gear Z_4 without the key is pulled with the coil spring④ toward the periphery of gear Z_2. Projector ② is fitted on the gear Z_2 and it projects itself through a hole of gear Z_4. Gapsenser (Midimeter)① is mounted on the gear Z_4.

The rotational delay of gear Z_2 is observed from a small clearance between ① and②. The marker ③ records the time of meshing at the pitch point, on the oscilloscope.

Figures 11 and 12 show the rotational delay of gear Z_2 measured under loads of

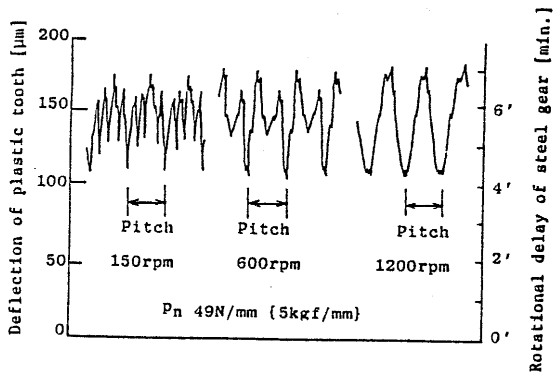

Fig.11 Deflections due to loading velocity

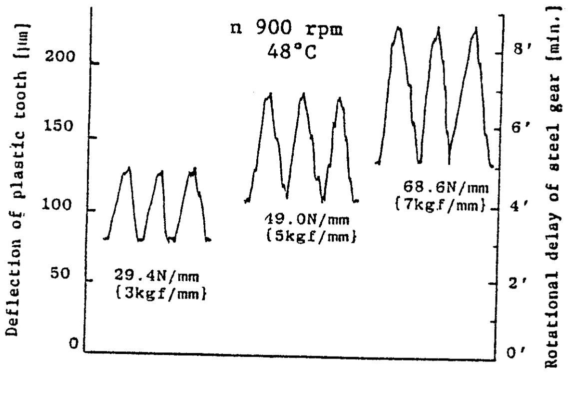

Fig.12 Deflections due to load magnitude

P_n 29.4, 49.0, 68.6 N/mm (3, 5, 7 kgf/mm) and at speeds of n_1 150 ∿ 1200 rpm, where A-form deflection in Fig.8 occurs in the case of low speeds and C-form at medium speeds. Figure 12 shows that the dynamic deflection also changes according to the load.

These deflection-curve shapes include various problems caused by the factors other than the tooth deflection, such as the tooth wear, permanent tooth deformation, elastic deformation of key, torsional vibration due to the fluctuation of tooth deflection, etc.

3.4 Basic measures for increasing load carrying capacity and lifetime of plastic gear

The steel tooth tip scrapes deeply the tooth dedendum of plastic gear. Because of large tooth deflection, the range of two pair meshing expands inward from both ends of one pair meshing. The narrow tooth surface near the pitch point is struck hard with the mating tooth surface due to a sudden change of load. These strikes will become the cause of wear which occurs near the pitch point of plastic gears.

In order to use satisfactorily the plastic gears for power transmission, improvements of mechanical properties are naturally required on the material side. From the design side, various ideas other than an increase in the gear module and/or the tooth width are necessary.

New ideas, such as the load-sharing by using gear pairs with a contact ratio above 2, the applications of V-0 profile, the profile modifications, etc. will increase the load capacity and the lifetime of plastic gears.

4. Conclusions

Present investigation to develop the plastic gears for power transmission can be summarized as follows:

(1) The tooth deflection of plastic gear and the rotational delay of the mating gear can be calculated easily using the spring constants provided in Table 4.

The results of computation and experiment make it clear that the plastic tooth deflections are as large as 0.1 ∿ 0.2 mm and account for 98 % of the total deflection of the two meshing teeth.

(2) The severe wear which occurs on the dedendum of plastic teeth is caused by the rotational delay of steel gear due to the large deflection of plastic gear teeth. Moreover the additional two pair meshing occurs inevitably in the original one pair meshing range. In the more deflectable case, three pair meshing also occurs near the pitch point of teeth.

(3) Measures for increasing the load carrying capacity and the lifetime are proposed other than the adoption of wide tooth width and thick tooth thickness; e.g. the contact ratio greater than 2, the tooth profile modification and the deflected tooth profile equal to the true involute.

References

(1) Tsukamoto, N., Trans. Jpn. Soc. Mech. Eng. Ser. C (in Japanese), Vol.45, No.391 (1979), p.382.
(2) Tsukamoto, N., Trans. Jpn. Soc. Mech. Eng. Ser. C (in Japanese), Vol.46, No.409 (1980), p.1116.
(3) Tsukamoto, N., Trans. Jpn. Soc. Mech. Eng. Ser. C (in Japanese), Vol.47, No.416 (1981), p.474.
(4) Tsukamoto, N., Trans. Jpn. Soc. Mech. Eng. Ser. C (in Japanese), Vol.47, No.416 (1981), p.484.
(5) Tsukamoto, N., Trans. Jpn. Soc. Mech. Eng. Ser. C (in Japanese), Vol.48, No.425 (1982), p.95.
(6) Terashima, K., Preprint of Jpn. Soc. Mech. Eng. (Kyushu branch), (in Japanese), (1966-2), p.95.
(7) Semba, M., Gears Vol.3 (in Japanese) (1962), p.607, Nikkankogyo.
(8) Nippon Polypenco Company, Technical Report of MC-nylon.