Study on Load Bearing Capacity of Gears with Smaller Number of Teeth*

[2nd Report, Surface Durability of Tufftrided Spur Gears ($z = 13$)]

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The surface durability of tufftrided spur gears with a smaller number of teeth was investigated by carrying out a running test and compared with the case of the normalized steel spur gears. The surface durability of the tufftrided gear is found to be higher by 50.% than that of the normalized steel gear and a failure of the follower is hard to occur due to the effect of the residual compressive stress produced by tufftriding. The contact fatigue limit load (circumferential load) of the gear pair can be raised to a considerable extent by setting an operating pitch point in the range of double tooth pairs contact. Furthermore the range of applications of these experimental results was examined on the basis of the calculated results of Hertz stress and specific sliding.

Key Word : Gear, Fatigue, Tufftrided Gear, Smaller Number of Teeth, Load Bearing Capacity, Surface Durability, Spalling, Hertz Stress

1. Introduction

Recently with an increased demand for higher reduction ratio, size-down and lighter weight, planetary gear units have come increasingly into use. In the planetary gear units the profile shifted external gears with a smaller number of teeth than the minimum one to avoid undercut without addendum modification are often used as sun and planet gears to reduce the size and weight.

In the previous paper⁽¹⁾, the surface durability of normalized steel spur gears with number of teeth $z = 13$ with various amounts of addendum modification was investigated.

Steel members such as gear, bearing, cam and so on are often tufftrided in order to increase their surface durabilities. The studies on the load bearing capacity of tufftrided rollers were reported^{(2) ~ (6)}, but those on tufftrided gears are few.

In the present paper, the surface durability of tufftrided spur gears with a smaller number of teeth with various amounts of addendum modification was investigated by carrying out a running test

and compared with the case of the normalized steel spur gears⁽¹⁾. Furthermore the range of applications of these experimental results was examined on the basis of the calculated results of Hertz stress and specific sliding of spur gears with various numbers of teeth and addendum modification coefficients.

2. Experimental Method and Apparatus

2.1 Test gears

Material of test gears used in this experiment is S45C normalized steel (heat treatment condition : $850^{\circ}\text{C} \times 1.5\text{h}$ water quenched and $620^{\circ}\text{C} \times 2\text{h}$ air cooled). Dimensions of test gears used in this experiment are shown in Table 1. Test gears are hobbled to an accuracy of 4th class in JIS and then tufftrided (heat treatment condition : $570^{\circ}\text{C} \times 2\text{h}$ oil quenched). Values of addendum modification coefficient of test pinions $\alpha_1 = 0.24, 0.5, 0.75$ were selected considering the lower limit

Table 1 Dimensions of test gears

	Pinion	Gear
Tooth profile	Profile shifted	
Module m	3	
Pressure angle α_c	20°	
Number of teeth z	13	26
Face width b	8 mm	10 mm
Addendum modification coefficient α	G.A	0.24
	G.B	0.5
	G.C	0.75
		0
		-0.26
		-0.51

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for undercut and the upper limit for tip thickness. Figure 1 shows the hardness distribution in test gears. The tooth profiles of test gears before and after tufftriding are almost the same. The material and dimensions of test gears used in this experiment are the same as those in 1st report⁽¹⁾.

2.2 Testing method

A gear testing machine of power circulating type⁽¹⁾ (maximum circulating power 10 kW) is used in this experiment. The

Fig.1 Hardness distribution of test gear

(a) Tufftrided gears

(b) Normalized steel gears

$$1 \text{ N} = 0.102 \text{ kgf}$$

Fig.2 Contact fatigue test results

lubricating oil (FBK oil RO 150) is supplied to the engaging side at a rate of about 0.5 L/min at a temperature of 40°C.

The number of load cycles at which the failure area ratio of the pinion reaches 2% was regarded as the contact fatigue life. The failure area ratio (to the working surface area of a tooth) was obtained for a tooth which suffered the severest damage considering the accuracy of gears. The failure area was measured by magnifying the tooth surface ten times by using a TV camera, VTR and monitor TV. Pinion speed in this experiment is 1800 rpm and the contact fatigue test was carried out after the running-in operation [Hertz stress $\sigma_H = 0.4 \sigma_{Hu}$ (σ_{Hu} : surface durability), number of load cycles $N = 10^5$].

3. Experimental Results and Discussions

3.1 Contact fatigue test results

Figure 2(a), (b) show the contact fatigue test results for tufftrided and normalized steel⁽¹⁾ spur pinions respectively. The contact fatigue life and limit load of the normalized steel gear pair G.C are governed by the failure of gear ($x_2 = -0.51$). The surface failure of the normalized steel gear is pitting and that of

(a) G.A ($P = 1.2 P_u$, $N = 1.8 \times 10^6$)

(b) G.B ($P = 1.1 P_u$, $N = 1.3 \times 10^6$)

(c) G.C ($P = 1.2 P_u$, $N = 2.4 \times 10^6$)

Pinion

Gear

P: Pitch point

C: Inner point of single tooth pair contact

Fig.3 Failure of tufftrided gears

the tufftrided gear is spalling. The contact fatigue limit loads (circumferential loads) P_u of pinions of G.A ($x_1 = 0.24$) and G.B ($x_1 = 0.5$) are almost equal and much smaller than those of G.C ($x_1 = 0.75$) in both the tufftrided and normalized steel gear. P_u of the tufftrided pinions are found to be higher by about 100 % than those of the normalized steel pinions.

Figure 3 shows the tooth surface on the tufftrided gear at the contact fatigue life at which the spalling area ratio of pinion is 2 %. The spalling occurs only on the pinion (driver) and the contact fatigue life of the tufftrided gear pair is governed by the spalling of the pinion. In the case of the normalized steel gear pair, the contact fatigue life is governed by the pitting of a driver or follower depending on the Hertz stress and specific sliding⁽¹⁾. Figure 4 shows a transverse section of the failure of the tufftrided and normalized steel pinions. The spalling cracks of the tufftrided gear are found to occur at the subsurface and they propagate almost parallel to the tooth surface and then to the contacting surface. The pitting cracks of the normalized steel gear occur on the tooth surface and propagate into the subsurface.

Figure 5 shows the Hertz stress and specific sliding over the range of engagement of each gear pair. The abscissa denotes the radius ρ of curvature of pinion tooth surface and the ordinate the Hertz stress and specific sliding. The Hertz stress is given as the ratio to that at pitch point calculated considering that the load sharing factor is unity. In this figure, A, B, C, D, P denote the beginning of engagement, inner and outer points of single tooth pair contact, end of engagement and pitch point of pinion respectively.

The pitch point lies in the range of single tooth pair contact in G.A, G.B, and in the range of double tooth pairs contact in G.C. The surface failure is seemingly apt to occur at the pitch point of the

driver when this point lies in the range of single tooth pair contact, which is common in practice. This might be because the Hertz stress is fairly high at this point and the directions of sliding are such that sliding takes place away from the pitch point on the driver, which affects the occurrence of the surface failure.

In both the tufftrided and normalized steel gear pairs, the failures of G.A, G.B occur first at the pitch point of the driver and the contact fatigue lives of the gear pairs are governed by this failure, since the pitch point lies in the

(a) G.A

(b) G.B

(c) G.C

Fig.5 Hertz stress and specific sliding of test gear pairs

Fig.4 Transverse section of failure (G.A)

range of single tooth pair contact as shown in Fig.5(a), (b). In G.C, the pitch point lies in the range of double tooth pairs contact. Therefore the contact fatigue limit loads of the driver of G.C are likely to become larger than those of G.A, G.B, since the Hertz stress at the pitch point of G.C is much lower. In the case of normalized steel gear pair⁽¹⁾, the contact fatigue life of G.C is governed by the failure at the inner point of single tooth pair contact of the follower where both the Hertz stress and specific sliding are considerably higher than those at the pitch point. In the case of tufftrided gear pair, the Hertz stress and specific sliding are the same as in the case of the normalized steel gear pair, but the failure occurs only on the driver. This might be explained as follows.

It is reported⁽⁵⁾ that a fairly large residual compressive stress (about -200 MPa) occurs at the hardened layer of the tufftrided roller of which material and heat treatment condition are considered almost the same as in the case of the tufftrided gears used in this experiment. In the case of the follower of the tufftrided gear pair, the occurrence of the failure is restrained by this residual compressive stress and the failure does not occur in this experimental condition. In the case of the driver, the effect of the change of the directions of sliding at the pitch point on the occurrence of the failure is considered larger than the restraining effect of the residual compressive stress against the occurrence of

(a) Tufftrided gears

(b) Normalized steel gears

$$1 \text{ MPa} = 0.102 \text{ kgf/mm}^2$$

Fig.6 $\sigma_H - N$ curves

the failure. Therefore the failure of the tufftrided gear pair is considered to occur only on the driver.

Consequently the contact fatigue limit load of the tufftrided gear pair can be raised considerably by designing a gear pair such that the pitch point may locate in the range of double tooth pairs contact.

Figure 6 shows the contact fatigue test results arranged with the Hertz stress at the positions of failure as the ordinate and number of load cycles as the abscissa on the basis of the results of Figs.2 and 5. The surface durability σ_{Hu} of each gear is found almost equal regardless of addendum modification coefficient and σ_{Hu} of tufftrided gear is higher by 50 % than that of the normalized steel gear.

3.2 Change of tooth profile with number of load cycles

Figure 7(a), (b) show the typical tooth profiles of tufftrided and normalized steel⁽¹⁾ gears before test and at the contact fatigue life (after test). It is found that the local high spots on the tooth surface which are observed before test are reduced at the contact fatigue

(a) Tufftrided gear ($P = 1.2 P_u$) (b) Normalized steel gear ($P = 1.2 P_u$)

Fig.7 Change of tooth profile (G.A)

life, but a large change of tooth profile is not observed.

3.3 Hertz stress and specific sliding of gears with number of teeth $z \leq 12$

The Hertz stress and specific sliding of gear pairs of module $m = 3$ with various numbers of teeth ($z_1 \leq 12$) and addendum modification coefficients were calculated in order to examine the position of failure occurrence. Figure 8(a)~(d) show the Hertz stress and specific sliding over the range of engagement of gear pairs with $z_1 = 12, 11, 9, 8$ (gear ratio $i = z_2/z_1 = 2$) respectively. In Fig.8(a), (b), the addendum modification coefficients of pinion x_1 and gear x_2 are selected as lower limit for undercut and upper limit for tip thickness respectively to maximize the Hertz stress and specific sliding at the root (point A) of the pinion. In Fig.8 (c), (d), x_1 and x_2 are selected as upper limit for tip thickness and lower limit for undercut respectively to minimize the Hertz stress and specific sliding at point A. It is seen from these figures that the Hertz stress shows the highest value at the inner point of single tooth pair contact of pinion (point B) in each gear pair

but is almost equal to that at the pitch point, and the specific sliding at point B is fairly small. Therefore it might be considered that the failure at point B does not occur earlier than that at pitch point in each gear pair. In the case of $z_1 = 12$, the Hertz stress at point A is almost equal to that at pitch point and the specific sliding is almost equal to that in the case of $z_1 = 13$. In this case the failure is considered to occur first at the pitch point irrespective of the addendum modification coefficient. In the case of $z_1 = 11$, the Hertz stress at point A is much larger than that at pitch point and the specific sliding is much larger compared with the case of $z_1 = 13$. Since the Hertz stress and specific sliding at point A increase with a decreasing number of teeth, the failure is considered to occur first at point A in the case of $z_1 \leq 11$ with the addendum modification coefficients for which Hertz stress and specific sliding at point A become maximum. In the case of $z_1 = 9$, the Hertz stress at point A is smaller than that at pitch point and the specific sliding is smaller than in the case of $z_1 = 13$. The failure is considered to occur first at pitch point in the case of $z_1 \geq 9$ with the addendum modification coefficients for

(a) $z_1 = 12$

(c) $z_1 = 9$

(b) $z_1 = 11$

(d) $z_1 = 8$

Fig.8 Effect of number of teeth on Hertz stress and specific sliding

which Hertz stress and specific sliding at the point become minimum. In the case of $z_1 = 8$, the Hertz stress at point A is much larger than that at pitch point and the specific sliding is larger than in the case of $z_1 = 13$. The failure is considered to occur first at point A irrespective of addendum modification coefficient in the case of $z_1 \leq 8$.

It might be considered from these results that the failure occurs first at pitch point irrespective of addendum modification coefficient in the case of $z_1 \geq 12$, and in the case of $9 \leq z_1 \leq 11$ at pitch point or root point depending on the addendum modification coefficient, and in the case of $z_1 \leq 8$ at root point irrespective of addendum modification coefficient.

4. Conclusions

The main results obtained from this investigation are summarized as follows.

(1) The contact fatigue limit load of the tufftrided gear pair can be raised considerably by designing a gear pair such that the pitch point may locate in the range of double tooth pairs contact, but for normalized steel gear pair the limit load cannot be raised.

(2) The surface durability of the tufftrided gear is higher by 50 % than that of the normalized steel gear.

(3) The contact fatigue life of the normalized steel gear pair is governed by pitting of a driver or follower depending on the Hertz stress and specific sliding, but that of the tufftrided gear pair in this experiment is governed by spalling of a driver because the failure of a follower

is hard to occur due to the effect of the residual compressive stress produced by tufftriding.

(4) The failure occurs first at pitch point irrespective of addendum modification coefficient in the case of the number of teeth $z_1 \geq 12$, and in the case of $9 \leq z_1 \leq 11$ at pitch point or root point depending on the addendum modification coefficient, and in the case of $z_1 \leq 8$ at root point irrespective of addendum modification coefficient.

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References

- (1) Oda, S. and Koide, T., Bull. JSME, Vol.28, No.236(1985), p.337.
- (2) Nakamura, H., et al., Trans. Jpn. Soc. Mech. Eng., Vol.39, No.328 (1973), p.3519.
- (3) Nakamura, H., et al., Trans. Jpn. Soc. Mech. Eng., Vol.41, No.342 (1975), p.351.
- (4) Fujita, K., et al., Trans. Jpn. Soc. Mech. Eng., Vol.47, No.417, C(1981), p.683.
- (5) Fujita, K., et al., Trans. Jpn. Soc. Mech. Eng., Vol.47, No.423, C(1981), p.1509.