Load-Carrying Capacity of $HB \approx 340$ (Hardened and Tempered) Gears* (Influence of Gear Ratio and Surface Roughness on Surface Durability)

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The surface durability of $HB \approx 340$ (hardened and tempered) gears having a gear ratio of $u \neq 1$ was investigated by using a power circulating-type gear testing machine. The test results reveal that the gear ratio has a great influence on surface durability in the case of gears with a rough tooth surface, such as hobbled gears. The endurance limit of medium-hardness gears is summarized as follows: If the initial tooth surface is rough, (i) $P_{\max} \approx 0.1$ HB kgf/mm² when gear ratio u is not unity, and (ii) $P_{\max} \approx 0.2$ HB kgf/mm² when u equals unity. But, if the initial tooth surface is very smooth, (iii) $P_{\max} \approx 0.3$ HB kgf/mm², regardless of the gear ratio. In conclusion, the tooth surface should be finished as smoothly as possible in order to obtain a high load-carrying capacity, because the influence of roughness is more considerable in conventional gears having a gear ratio of $u \neq 1$.

Key Words: Gear, Medium-Hardness Gear, Load-Carrying Capacity, Surface Durability, Gear Ratio, Tooth Surface Roughness

1. Introduction

Recently, the requirement of higher load-carrying capacity has increased for power transmission gears. In the case of medium-hardness gears which are widely used in industrial work, harder gears ($HB = 300 \sim 400$) have come into use in order to obtain a high surface durability⁽¹⁾⁽²⁾. In fact, hardened and tempered gears with a Brinell hardness number of $HB = 400$ have been able to be produced by hobbing with the usual high speed steel (HSS) hobs⁽³⁾.

The surface durability of hardened and tempered gears ($HB = 300 \sim 360$) was studied experimentally^{(4)~(8)}. From the test results in the case of a gear ratio of $u = 1$, the authors emphasized that the initial tooth surface could significantly affect the surface durability of hardened and tempered gears. In the case of conventional gears having a gear ratio of $u \neq 1$, it

seems that the effect of surface roughness on surface durability appears more evidently because a pair of gears do not always engage with the same teeth.

The surface fatigue of gears is influenced by parameters such as material, hardness, tooth profile, tooth surface roughness, lubricant, operating speed, and contact stress⁽⁹⁾.

For the calculation of gear-tooth data, it is generally desirable to obtain the hunting ratio between gear and pinion teeth⁽¹⁰⁾. On the other hand, recent experimental studies show that in certain cases, surface durability is appreciably lower for gear pairs with a hunting ratio than for those with a nonhunting ratio^{(11)~(14)}.

In this paper, the influence of gear ratio, gear accuracy and tooth surface roughness on the surface durability of $HB \approx 340$ (hardened and tempered) gears is investigated, from the viewpoint of gear-manufacturing methods. The running tests are performed with a power circulating-type gear testing machine using hobbled gears and ground gears of the same hardness. The load-carrying capacity of gears whose gear ratio u is not unity ($u = 31/21$), is compared with that of gears whose gear ratio is unity.

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The results reveal that, for gears with a rough surface, the initial tooth surface roughness has a much greater influence on the surface durability of gears having a gear ratio of $u \neq 1$, than on gears having a gear ratio $u = 1$. Furthermore, the effect of initial tooth surface roughness on the surface durability of conventional gears having a gear ratio of $u \neq 1$ can be estimated quantitatively.

2. Experimental Conditions and Method

2.1 Test gears and material

The test gears used were spur gears having a gear ratio of $u = 31/21$ and $u = 1$. Specifications for the test gears are given in Table 1. The material of the test gears was a 1%Cr-0.2%Mo alloy steel (JIS SCM 435). Heat treatment (quenching and tempering) was conducted before hobbing. The hardness of the test gears was approximately HB=340. In order to investigate the influence of gear ratio and surface roughness on the surface durability, the tooth surfaces were finished by hobbing and grinding (Maag 0° method).

The tooth profile error of the hobbled gears was JIS⁽¹⁵⁾ class 3~6 (AGMA⁽¹⁶⁾ Quality number 11~8, ISO⁽¹⁷⁾ Accuracy grade 7~10), and that of the ground gears was JIS class 0~1 (AGMA 14~13, ISO 4~5). The single pitch error, pitch variation, accumulative pitch error, run out and lead error of all test gears were JIS class 0~3 (AGMA 14~11, ISO 4~7).

The tooth surface roughnesses along the tooth profile of the hobbled gears and ground gears had a

maximum height $R_{\max}$ of 8~19 μm and 1~2 μm , respectively.

2.2 Gear test apparatus, lubricant and test method

The test apparatus used was two power circulating-type gear testing machines with center distances of 156 mm and 155 mm. In these test rigs, each having the same mechanism, the test gears could be started under no load, and the load (maximum circulating powers were 590 kW and 150 kW) could then be applied gradually during running⁽¹⁸⁾. The operating speed of the test rig with a gear ratio of $u = 31/21$ was 1800 rpm for the pinion, and that with a gear ratio of $u = 1$ was 1450 rpm for the driver.

The properties of the test lubricants are given in Table 2. The oils contain no extreme pressure additives. In the case where the gear ratio u was 31/21, the lubricant (oil C) was supplied to the meshing zone by an oil jet. The inlet oil temperature was adjusted to $313 \pm 2\text{K}$ ($40 \pm 2^\circ\text{C}$) and the oil flow rate was held constant at $33 \times 10^{-6}\text{m}^3/\text{s}$ (2 l/min). The gears having a gear ratio of $u = 1$ were lubricated by splash in an oil bath (oil A), the oil temperature being about 323 K (50°C). Figure 1 shows the calculated elastohydrodynamic (EHD) film thickness $h_{\min}$ ⁽¹⁹⁾ on the pitch line of the test gears. The values of D ⁽²⁰⁾ ($D = \sum R_{\max} / h_{\min}$, $\sum R_{\max}$: total initial surface roughness, $h_{\min}$: oil film thickness) were greater than 10 for hobbled gears and less than 6 for ground gears^{*1}. The value of D increased considerably during the operation, because the tooth surface temperature was higher than the

Table 1 Specifications of test gears

	Pinion	Gear	Driver	Follower
Gear ratio	31/21		1	
Module	6			
Pressure angle	20°			
Number of teeth	21	31	26	26
Pitch diameter	126	186	156	156
Facewidth	15	15	10	10
Addendum modification coefficient	0	0	-0.08	-0.08

Table 2 Properties of lubricants

		Oil C *	Oil A *
Specific gravity	15/4 °C	0.888	0.882
Viscosity m^2/s	40 °C	153.9×10^{-6}	56.4×10^{-6}
	100 °C	15.0×10^{-6}	7.8×10^{-6}
Viscosity index		97	103
Sulphur	wt %	0.35	1.10
Phosphorus	wt %	0.038	—
Mean molecular weight		575	470

* These oils contain no extreme pressure additives.
(Oil C : ISO VG 150, Oil A : SAE 20)

Fig. 1 Calculated EHD film thickness (on the pitch line) in relation to oil temperature

Table 3 Experimental conditions and results

Test No.		GT-15		GT-16		GT-17		GT-1		GT-3		GT-14			
Number of teeth		21	31	21	31	21	31	26	26	26	26	26	26		
Brinell hardness number HB		331	331	331	331	331	332	334	330	343	341	352	341		
Tooth surface-finishing method		Hobbing				Grinding		Hobbing				Grinding			
Tooth surface roughness $R_{\max}$ μm		19	8	14	12	1	1	13	12	12	13	2	1		
Tooth profile error μm		29	18	22	23	7	7	31	28	23	34	6	7		
Max. Hertzian stress		MPa		490		690		1180		690		980		1180	
$P_{\max}$		kgf/mm ²		50 (0.15 HB)		70 (0.21 HB)		120 (0.36 HB)		70 (0.21 HB)		100 (0.29 HB)		120 (0.35 HB)	
Lubricant		Oil C (ISO VG 150)												Oil A (SAE 20)	
Cycles run N_1, N_2, N		10^7	0.7×10^7	2×10^6	1.4×10^6	10^7	0.7×10^7	2×10^7		10^7		10^7			
Pitting area ratio %		1.35	0.90	3.37	2.84	0.04	0.09	0.05	0.08	0.96	0.90	0.06	0.25		

inlet (or bath) oil temperature; and the EHD film became thin.

In order to evaluate the load-carrying capacity of the gears, the appearance of the tooth surfaces was observed at suitable intervals. In addition, the gear accuracy and tooth surface roughness was measured when the number of stress cycles in the pinion (or driver) reached N_1 (or N) = 10^6 , 5×10^6 and 10^7 .

3. Experimental Results and Discussions

3.1 Experimental results

The experimental conditions and results are given in Table 3. (i) For hobbed gears with a rough surface $R_{\max} \approx 13 \mu\text{m}$, tests [GT-15] and [GT-16] were performed with gears having a gear ratio of $u=31/21$, and tests [GT-1] and [GT-3] were performed with gears having a gear ratio of $u=1$. (ii) For ground gears with a smooth surface $R_{\max} \approx 1 \mu\text{m}$, tests [GT-17] and [GT-14] were performed with gears having a gear ratio of $u=31/21$ and $u=1$, respectively. In general, the surface durability of steel gears is quite closely related to the Brinell hardness number [$P_{\max} = (0.2 \sim 0.3) \text{ HB kgf}/\text{mm}^2$ ⁽¹⁾⁽⁹⁾]. Therefore, tests were performed in the range of pitch-line maximum Hertzian stress $P_{\max} = 490 \sim 1180 \text{ MPa}$ [(0.15 ~ 0.35) HB kgf/mm²].

In all tests, tooth surface fatigue "pitting" occurred. The pitting area of the test gears varied with the

experimental conditions.

3.2 Influence of gear ratio on pitting

Figure 2 shows the changes in the average pitting area ratio $\bar{p}^{*2}$ with the number of stress cycles. In Fig. 2, (a) is the case of test gears having a gear ratio of $u=31/21$, and (b) is the case of test gears having a gear ratio of $u=1$. Figure 3 shows the $P_{\max}-N_1$ (or N) curves for pitting in hobbed gears. In Fig. 3, $P_{\max}$ is the pitch-line maximum Hertzian stress, and N_1 (or N) is the number of stress cycles when the average pitting area ratio reaches $\bar{p}=0.1, 0.2, 0.5$ or 1.0% .

(i) In the case of hobbed gears, it is clear that the gear ratio has a great influence on the pitting of hardened and tempered gears. The pitting area ratio of hobbed gears when the gear ratio $u=31/21$ and $P_{\max}=490 \text{ MPa}$ ([GT-15]), is much the same as that of hobbed gears when the gear ratio $u=1$ and $P_{\max}=980 \text{ MPa}$ ([GT-3]). Consequently, the result shows that the tangential load of gears having a gear ratio of $u=31/21$ is approximately a fourfold reduction of that of gears having a gear ratio of $u=1$ for the same pitting area ratio, because the tangential load on the gear teeth is proportional to the square of the maximum Hertzian stress. In Fig. 3, it is indicated that the pitting life of gears having a gear ratio of $u=31/21$ is much shorter than that of gears having a gear ratio of $u=1$.

(ii) In the case of ground gears with a smooth surface, it is clear that the pitting area ratio decreases, regardless of the gear ratio, under heavier load conditions ($P_{\max}=1180 \text{ MPa}$ ([GT-17], [GT-14])).

3.3 Influence of surface roughness on pitting

Figure 4 shows typical examples of the appearance of the tooth surface after running in the case of gears having a gear ratio of $u=31/21$.

(i) In the case of hobbed gears, a lot of pits (0.1 ~ 0.7 mm in diameter) are observed when $P_{\max}=490 \text{ MPa}$ and $N_1=10^7$ ([GT-15]). Furthermore, when $P_{\max}=690 \text{ MPa}$, pits having a size of more than 1.0 mm in diameter are observed at $N_1=2 \times 10^6$ ([GT-16]).

*1) When the gear ratio u was 31/21 and the oil temperature was 313 K, the values of D for tests [GT-15], [GT-16] and [GT-17] were about 12, 13 and 1, respectively. When the gear ratio u was unity and the oil temperature was 323 K, the values of D for tests [GT-1], [GT-3] and [GT-14] were about 40, 44 and 6, respectively.

*2) The average pitting area ratio $\bar{p}$ is defined as follows: $\bar{p} = (p_1 z_1 + p_2 z_2) / (z_1 + z_2)$ where p is the pitting area ratio on one tooth surface, z the number of teeth, and suffixes 1 and 2 indicate the pinion (or driver) and gear (or follower), respectively.

Fig. 2 Changes in average pitting area ratio with the number of stress cycles

Fig. 3 $P_{\max}$ - N (or N) curves for pitting (in hobbed gears)

(ii) In the case of ground gears with a smooth surface, only small pits are observed at $N_1=10^7$ when $P_{\max}=1180$ MPa ([GT-17]), because a desirable condition of contact is kept during running due to the smooth tooth surface. In Fig. 2, it is indicated that the smoother the tooth surface, the smaller the rate of increase of pits with repeated contact stress.

3.4 Influence of gear ratio on changes in tooth profile and surface roughness

Figure 5 shows the changes in tooth profile with the number of stress cycles. In Fig. 5, the straight line indicates a perfect involute profile. Figure 6 shows typical examples of the changes in surface roughness

Fig. 4 Appearances of tooth surface after running (gear ratio 31/21)

Fig. 5 Changes in tooth profile with the number of stress cycles

Fig. 6 Changes in surface roughness along the tooth profile (near the pitch point)

along the tooth profile near the pitch point when $P_{\max} = 690$ MPa and 1180 MPa.

(i) In the case of hobbed gears, the tooth profile of gears having a gear ratio of $u=31/21$ changes considerably because many pits appear. Consequently, the gear ratio has a great influence on the changes in tooth profile. When $P_{\max}=690$ MPa, in the case of gears having a gear ratio of $u=31/21$, the tooth profile changes considerably on the tooth flank near the pitch line at $N_1=2 \times 10^6$ cycles ([GT-16]). On the other hand, the tooth profile hardly changes until $N=10^7$ cycles when the gear ratio $u=1$ ([GT-1]). When the gear ratio $u=31/21$, the asperities of the hobbed

tooth surface become flat by repeated contact stress before $N_1=10^6$ but a lot of pits appear ([GT-16]). This fact indicates that the decrease in surface roughness during the operation does not always have an advantageous effect on surface durability. In the hardened and tempered gears, the surface roughness is reduced with the number of stress cycles; but at the same time, a lot of fatigue-cracks develop on the tooth surface and finally pitting occurs⁽⁴⁾. On the other hand, when the gear ratio $u=1$, the changes in surface roughness are small at $N=10^7$ and few pits appear ([GT-1]). Therefore, in the case of gears with a rough tooth surface, the gear ratio has a considerable

Fig. 7 Relation between initial tooth surface roughness and average pitting area ratio

Fig. 8 Influence of gear ratio and surface roughness on the surface durability of medium-hardness gears (P_{max}/HB - R_{max} curves)

influence on the changes in tooth surface and on the pitting.

(ii) In the case of ground gears with a smooth tooth surface, the changes in tooth profile and surface roughness are small when $P_{max} = 1180$ MPa and N_1 (or N) $= 10^7$, regardless of the gear ratio ([GT-17], [GT-14]).

From the above results, it is observed that when the gear ratio $u \neq 1$, the influence of the accuracy and surface roughness of gear pairs on surface durability is quite considerable, except when gears with smooth tooth surfaces are used, because a pair of gears do not always engage with the same teeth.

4. Influence of Gear Ratio and Surface Roughness on Surface Durability

Figure 7 shows the relation between the initial tooth surface roughness and the average pitting area ratio at N_1 (or N) $= 10^7$ cycles. In Fig. 7, which includes the data obtained in previous investigations^{(4)~(8),(21),(22)}, (a) and (b) represent the results when gear ratio $u = 31/21$ and $u = 1$, respectively. There is a tendency that pitting is avoided by using a smooth tooth surface, though the tooth profile error of each test gear is different. Consequently, Fig. 7 confirms that the initial surface roughness has considerable influence on the surface durability of $H_B \approx 340$ steel gears.

Figure 8 shows the relation between the initial

surface roughness ($R_{\max}$) and the endurance limit ($P_{\max}/HB$). It is assumed, for estimating the endurance limit, that the average pitting area ratio is $\bar{p} = 0.1\%$ (there is no trouble in operation) in Fig. 7. From Fig. 8, which represents $P_{\max}/HB$ - $R_{\max}$ curves, the influence of gear ratio and surface roughness on the surface durability of hardened and tempered gears can be estimated quantitatively. For example, the results show that the values of the endurance limit obtained from this investigation are approximately (i) $P_{\max} \approx 0.1 HB \text{ kgf/mm}^2$ ($R_{\max} \approx 10 \mu\text{m}$) or $0.3 HB \text{ kgf/mm}^2$ ($R_{\max} \approx 2 \mu\text{m}$) when the gear ratio u is not unity, and (ii) $P_{\max} \approx 0.2 HB \text{ kgf/mm}^2$ ($R_{\max} \approx 10 \mu\text{m}$) or $0.3 HB \text{ kgf/mm}^2$ ($R_{\max} \approx 2 \mu\text{m}$) when u equals unity. It is noticed that the tangential load of gears with a smooth tooth surface ($R_{\max} \approx 2 \mu\text{m}$) is approximately (i) ten times under a hunting gear ratio ($u \neq 1$) and (ii) two times under a unity gear ratio ($u = 1$) that of gears with a rough initial tooth surface ($R_{\max} \approx 10 \mu\text{m}$).

Therefore, the tooth surface should be finished as smoothly as possible in order to obtain a high load-carrying capacity of medium-hardness gears, because the influence of roughness is much more evident in conventional gears having a gear ratio of $u \neq 1$ than in gears having a gear ratio of $u = 1$.

5. Conclusions

The influence of gear ratio and surface roughness on the surface durability of $HB \approx 340$ (hardened and tempered) gears was investigated by using a power circulating-type gear testing machine. The test gears used were spur gears having a gear ratio of $u = 31/21$ and $u = 1$, and the tooth surfaces were finished by hobbing and grinding (Maag 0° method).

The endurance limit of medium-hardness gears is summarized as follows: If the initial tooth surface is rough, (i) $P_{\max} \approx 0.1 HB \text{ kgf/mm}^2$ when gear ratio u is not unity, and (ii) $P_{\max} \approx 0.2 HB \text{ kgf/mm}^2$ when u equals unity. But, if the initial tooth surface is very smooth, (iii) $P_{\max} \approx 0.3 HB \text{ kgf/mm}^2$, regardless of the gear ratio. Therefore, the above results reveal that the gear ratio has a great influence on surface durability in the case of gears with a rough tooth surface, such as hobbed gears.

In conclusion, the tooth surface should be finished as smoothly as possible in order to obtain a high load-carrying capacity, because the influence of roughness is more considerable in conventional gears having a gear ratio of $u \neq 1$.

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