Clarification of Tribological Behavior on Tooth Surface of Resin Worm Gear for Electric Power Steering

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We have clarified the tribological behavior on tooth surface by devising a simplified sliding test which simulates a resin worm gear. It was confirmed that the main factor of tooth surface deformation was creep deformation, and increase of molecular mass of the polyamide contributed to long-term durability. In addition, surface roughness and temperature dependence of the friction coefficient were gathered through variation of film thickness ratio using EHL theories and applied to rotating torque reduction by controlling tooth surface roughness.

Key Words: *polyamide, worm gear, simplified test, abrasion, creep deformation, EHL, torque reduction*

1. Introduction

Demands for weight reduction (energy conservation, reduced CO_2 emissions) in automotive parts are rising[1), 2)] as a result of energy-related problems such as global warming. In addition, requirements for high-level quietness associated with electronization and hybridization of automobiles have caused an upsurge in the proportion of resin usage in automotive components with sliding mechanisms. Amidst these situations, resin gears have in recent years become widely utilized in automotive parts[3)] due to the high level of quietness, self-lubricity, and reduction in mass[4)] they achieve within such parts. The usage of resin worm gears (hereafter "worm gears") is especially high for the reduction gear used in electric power steering (EPS), which is illustrated in **Fig. 1**. EPS is an important safety part which assists steering force by supporting the load on the front axle of the automobile, and is also a tribological element for which high reliability is expected. Furthermore, the downsizing of automotive parts in recent years, coupled with the high demand for application of EPS on large vehicles due to its effects on improved fuel efficiency, has emphasized the necessity of high-output support in worm gears.

The necessary characteristics of worm gears include suppression of rattling caused by decrease in tooth thickness, known as backlash, in addition to assurance of durability life and reduction of steering torque. **Figure 2** illustrates the contact form of the resin worm gear, which during usage is easily affected by sliding in addition to rolling, and upon which heat generation and cooling are repeated due to the process of sliding. This combination

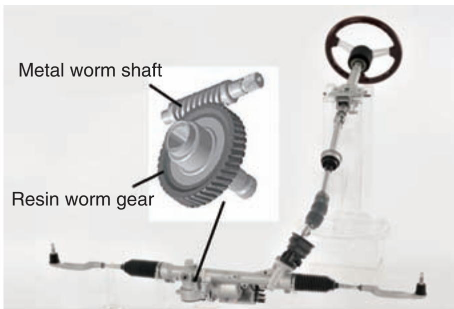


Fig. 1 Basic structure of JTEKT EPS and internal structure of reduction gear

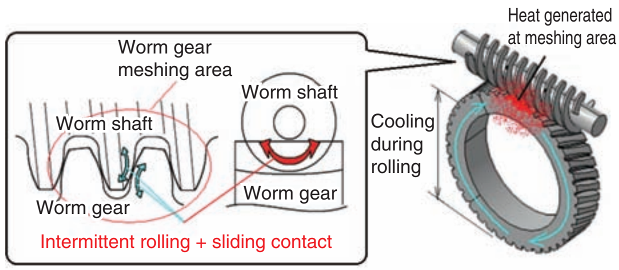


Fig. 2 Contact form of resin worm gear

creates a complex mechanism of tribological behavior on the tooth surface. Wear and creep deformation are factors which influence decrease in tooth thickness, however it is difficult to isolate wear and creep deformation from the tooth surface for evaluation after gear operation. Furthermore, high output exacerbates the decrease in tooth thickness by increasing load on the tooth surface, which in turn reduces durability life. Consequently, it is essential to ensure wear resistance and creep resistance within material development and the design of the worm gear, as these properties affect the decrease in tooth thickness. It is also necessary to lower rotation torque in the worm gear in order to improve steering feeling. However, as there are multiple agents of sliding resistance on the tooth surface which affect rotation torque, it is impossible to give a simple estimation of which of these agents is the primary cause. It is therefore important to conduct integrated analysis on the factors affecting sliding resistance on the tooth surface, and apply the results in the design of the worm gear and the conditions of the gear teeth.

This paper describes, as a measure for the above mentioned issues, the development of a basic testing method simulating the worm gear to clarify the tribological behavior acting upon the gear tooth surface, and the application of the test results to the development of worm gears with long-term durability and low torque.

2. Basic testing method[5)-7)]

2. 1 Establishment of reproducible basic testing method for sliding mode of worm gear

Issues that have occurred up until now within the development of worm gears have been reproduced within the basic test in the complex sliding mode of the worm gear. The worm gear is employed under conditions of high surface pressure and high sliding speed which exceed the PV limit of general resin materials, and it is known that the temperature of the sliding contact face is low. This is because the lubrication conditions of the sliding contact surface due to grease and the suppression of heat generation in the intermittent contact cause the heat generated on the sliding contact surface to dissipate, thereby suppressing temperature rise. Such a unique sliding mode was difficult to reproduce in conventional basic tests such as the Suzuki frictional wear test, and required a durability test using the actual worm gear during actual development. Nevertheless, the durability test necessitated a large amount of time which in turn lengthened development time, and in addition, the mechanism of tribological behavior was difficult to identify.

To resolve this issue, we developed a basic testing method simulating the sliding mode of the actual worm gear. An outline of this test is shown in **Fig. 3**. By substituting the disk of an ordinary disk-on-ring test with a metal roller, we were able to reproduce the contact form with high surface pressure seen on the tooth surface of the worm gear. In addition, by changing the diameter of the metal roller, we were able to simulate different contact forms originating from the elastic modulus and presence or absence of reinforced fibers in the gear material, as shown in **Fig. 4**. Also, by designing a wedge-shaped contact area and grease reservoir on the fixture, we reproduced stable flow of grease on the friction surface as well as intermittent contact. This, combined with the introduction of an intermittent process and adjustment of its interval, suppressed temperature rise and improved the conformity of the simulated results to the tooth surface temperature of the actual worm gear. The developed method is also able to accurately identify the ratios of wear and creep deformation from the changes in weight and height of the resin ring before and after the test.

The above enable the simulation of the contact form of the actual worm gear tooth within the developed basic testing method. As seen in **Fig. 5**, it has also become possible to estimate the decrease in tooth thickness of the worm gear from changes in height during the basic test[5)-7)].

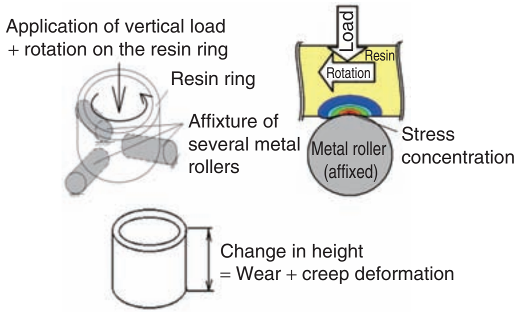


Fig. 3 Outline of basic test

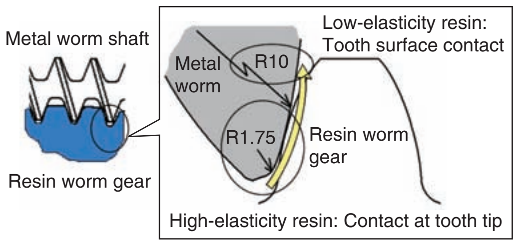


Fig. 4 Contact form of worm gear

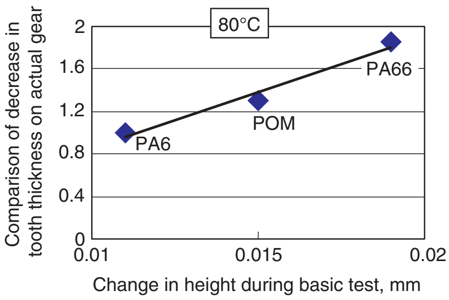


Fig. 5 Change in height during basic test and comparison of decrease in tooth thickness on actual gear

2. 2 Test conditions

For the test specimen, we used Polyamide 66 (hereafter “polyamide”), which has excellent self-lubricity. The conditions of the test are shown in **Table 1**. The basic test was conducted using the method described in **2.1**. After verifying the correlation between decrease in tooth thickness and gear life by comparing the basic test with a durability test, we evaluated wear resistance and creep resistance which both affect tooth surface deformation and durability life, and the friction coefficient which influences steering torque.

Table 1 Basic test conditions

Evaluated items		Wear resistance/ creep resistance evaluation	Friction coefficient evaluation
Roller specimen	Roller R	1.75mm	10mm
	Roller material	S45C Induction heat treatment	S45C
Resin ring Conforms to JIS K7218A	Shape dimensions	O.D. : 25.6mm I.D. : 20mm Height : 12mm	
Test conditions	Sliding speed	1.016 m/s	0.166 m/s
	Load surface pressure	90～160 MPa	23 MPa
	Testing time	240 min. (with intervals)	3 min.
	Atmospheric temperature	25℃ to 120℃	
	Lubrication conditions	Grease lubrication (Barium complex)	

3. Technology for lengthening long-term durability of worm gears

Figure 6 shows the observed tooth images from SEM back-scattered electron images of the non-reinforced polyamide worm gear and the glass fiber (GF) reinforced polyamide worm gear after durability tests on the actual gears. No remarkable wear was found on the non-reinforced polyamide, although there were slight sliding marks caused by contact with the worm shaft. In contrast, a large amount of flaking of the resin due to wear and cohesion of GF which had broken off was observed on the GF reinforced polyamide. From the observations of tooth surface post durability testing, however it is unclear how much wear and creep deformation had contributed to these flaws, and impossible to pinpoint measures for material development.

The results of the developed basic test are shown in **Fig. 7**. Using this test, it was possible to isolate the amounts of wear and creep deformation by measuring the changes in weight and height before and after the test. The results showed that the non-reinforced material had a smaller overall decrease in tooth thickness. In addition, it was found that, while creep deformation was the main factor in decrease in tooth thickness of the non-reinforced material, wear was the main factor in decrease in tooth thickness of the GF reinforced material. In regard to these findings, a cross section of the tooth surface of an actual damaged worm gear is shown in **Fig. 8**, which clearly shows that creep deformation has progressed, and that a crack has occurred near the bottom of the tooth, damaging the gear. We believe that the creep deformation of the resin is caused by the plastic flow of the molecular chain due to sliding, and that a high molecular weight increases creep-rupture elongation due to the strength of the molecular bond, thereby suppressing crack progression. **Figure 9** shows the results of an evaluation on the effect of resin viscosity (molecular mass) on the failure life of the polyamide. These results identify a strong correlation between the molecular mass and durability life of the worm gear, confirming that the polyamide with high molecular mass contributes to longer gear life[9].

Furthermore, as shown in **Fig. 10**, by tracking the time-based change in the sliding contact face after the basic test, it is now possible to estimate the wear mechanism of the GF reinforced polyamide[6]. When the GF on the surface of the tooth make contact with the metal worm at a high surface pressure, the GF breaks off from the resin, adheres to the surface of the gear, and acts as an abrasive. The GF then causes flaking of the resin due to wear, after which stress concentrates in the holes from which the GF has broken off and causes cracks to occur, which eventually leads to flaking over a large area.

In consideration of the above, we have developed a non-reinforced high molecular mass polyamide material with long-term durability and low level of decrease in tooth thickness. We then applied the developed polyamide as the material used for resin worm gears, which are now currently being mass-produced.

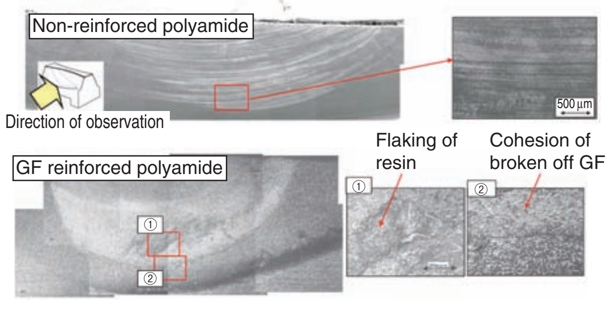


Fig. 6 Observed results of tooth surface after durability test of actual worm gear

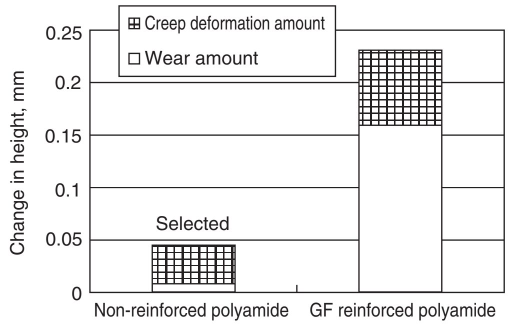


Fig. 7 Results of measurement of change in height due to basic test

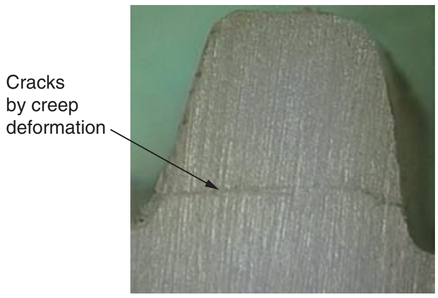


Fig. 8 Observed results of tooth surface after durability test of high molecular mass polyamide

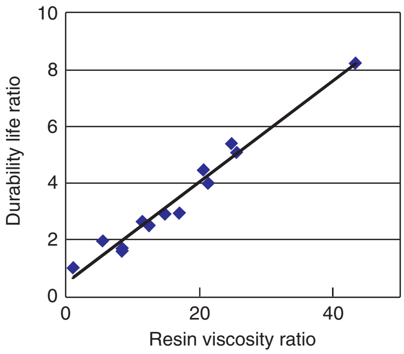


Fig. 9 Relationship between viscosity (molecular mass) of non-reinforced polyamide and durability life

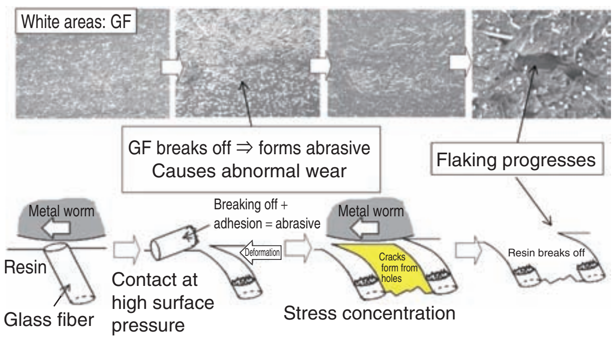


Fig. 10 Wear mechanism of GF filler

4. Torque-lowering technologies for the worm gear[8), 9)]

4. 1 Friction coefficient

There exist many factors (atmospheric temperature, tooth surface roughness, resin elastic modulus, grease base oil viscosity, oil film thickness, etc.) which act upon the steering torque of a steering system. If these factors can be categorized and steering torque (=friction coefficient) estimated, then these factors and steering torque can be included in machining conditions to reduce torque from the aspect of the material. The research described in this paper identifies the active mechanism of steering torque through the basic testing method we have developed, thus establishing a method of estimating steering torque.

Figures 11 and **12** respectively show the non-reinforced high molecular mass polyamide material that we have developed, and the relationship of the sliding contact surface temperature and sliding surface roughness of the abovementioned GF reinforced polyamide material to the friction coefficient. Both materials showed that increased surface roughness and higher sliding contact surface temperature lead to increased friction coefficient[8), 9)]. These factors were verified by film thickness ratio (a comparison of minimum oil film thickness of grease to synthetic surface roughness of sliding contact surface), which is an indicator of grease lubrication conditions.

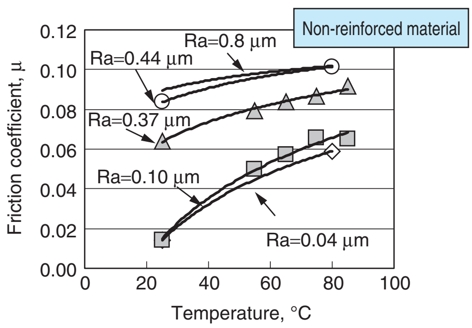


Fig. 11 Relationship of temperature and surface roughness of non-reinforced high molecular mass polyamide material to friction coefficient

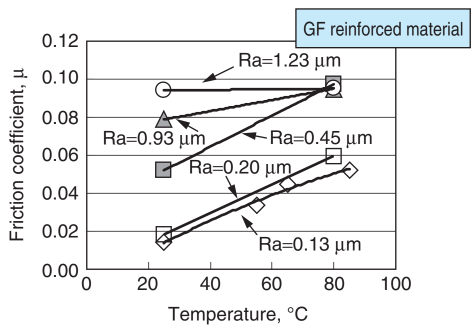


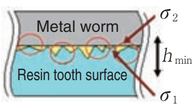
Fig. 12 Relationship of temperature and surface roughness of GF-reinforced polyamide material to friction coefficient

4. 2 Calculation of oil film thickness[10)]

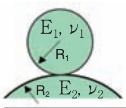
The following lubrication theory was used in order to calculate film thickness. The map of the lubrication region on line contact (Hooke's chart) is shown in **Fig. 13**. From the calculated results of Johnson's elasticity parameter g_E and viscosity parameter g_V, shown in Formula (2), in the sliding conditions of the developed non-reinforced high molecular mass polyamide, it is evident that the lubrication conditions of this study are within the IE lubrication region (equivalent viscosity - region of elasticity). In other words, it is not necessary to consider the rise in pressure seen in hard EHL (PE region) as the surface pressure is low (approximately 20 MPa), however it is necessary to take into account elastic deformation during sliding since the resin used has high viscoelasticity. The minimum oil film thickness h_{min} of the given region was calculated by the Dowson-Higginson indication by Herrebrugh, as shown in Formula (3). **Figure 14** shows the results of the minimum oil film thickness calculated from the tensile modulus of elasticity of the resin and base oil viscosity of the grease at each temperature. When compared with the effect of reduced elastic modulus of the resin associated with a rise in temperature, decreased base oil viscosity shows a higher amount of contribution, and therefore it was found that minimum oil film thickness is reduced as temperature rises.

$$\Lambda = \frac{h_{min}}{\sqrt{\sigma_1^2 + \sigma_2^2}} \quad (1)$$

Λ: Film thickness ratio
h_{min} : Minimum oil film thickness
⇒Calculated from theoretical formula
σ_1 : Surface roughness Ra of T/P resin
σ_2 : Surface roughness Ra (=0.0945 µm) of metal roller



$$g_E = \left(\frac{w^2}{\eta_0 uER} \right)^{1/2} \qquad g_v = \left(\frac{\alpha^2 w^3}{\eta_0 uR^2} \right)^{1/2} \quad (2)$$



Effective radius R: $\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}$

Effective elasticity coefficient E: $\frac{2}{E} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}$

R_1 = Roller radius
R_2 = ∞ (Flat contact)
E: Elastic modulus
ν: Poisson's ratio

E: Effective elasticity coefficient
η_0: Base oil viscosity
w: Load per unit length
α: Pressure-viscosity coefficient
R: Effective radius
u: Friction speed

$$h_{min} = 3.10 \left(\frac{\eta_0 u}{ER} \right)^{0.6} \left(\frac{w}{ER} \right)^{-0.2} R \quad (3)$$

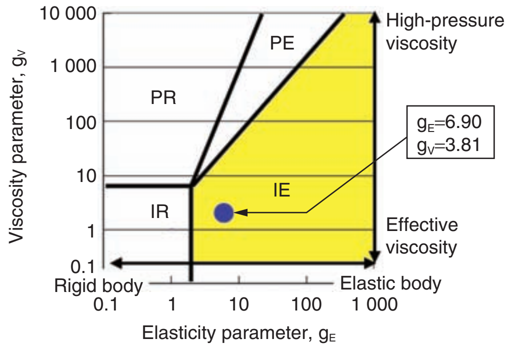


Fig. 13 Map of lubrication region (Hooke's chart)

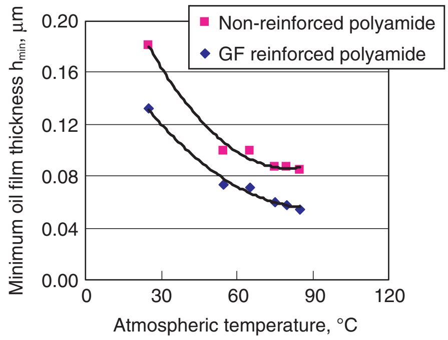


Fig. 14 Calculated minimum film thickness

4. 3 Study on friction control using film thickness ratio

Film thickness ratio was obtained by substituting the minimum oil film thickness calculated in **4.2** within Formula (1), and the relationship of this ratio to the friction coefficient was examined. The film thickness ratio-friction coefficient relationship within the developed non-reinforced high molecular mass polyamide is shown in **Fig. 15**. The film thickness ratio-friction coefficient relationship within the GF reinforced polyamide, which was used as a comparative material, is shown in **Fig. 16**. It was found that in both cases, the friction coefficient did not depend on sliding contact surface temperature nor surface roughness, but did depend on film thickness ratio, and decreased as film thickness ratio increased. As illustrated in **Fig. 17**, the friction coefficients of both materials are equivalent at no lubrication when the film thickness ratio is in the smaller range, and it is thought that this is a result of boundary lubrication, where the ratio of direct solid contact between two objects is large. On the other hand, it is also estimated that friction coefficient decreases due to the slightly mixed lubrication conditions, since the film thickness ratio increases along with the decrease in the ratio of direct solid contact. From these results, the dependence of the friction coefficient on surface roughness and temperature is clearly visible through the change in film thickness ratio.

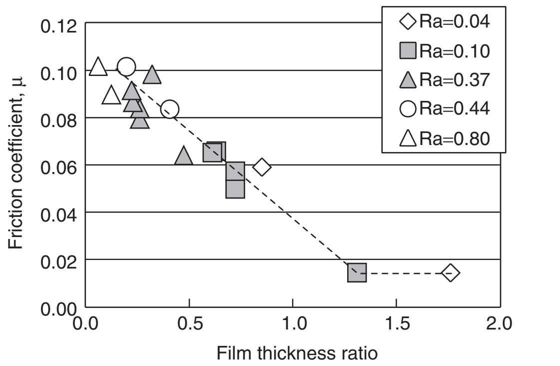


Fig. 15 Relationship between film thickness ratio and friction coefficient (Non-reinforced high molecular mass polyamide material)

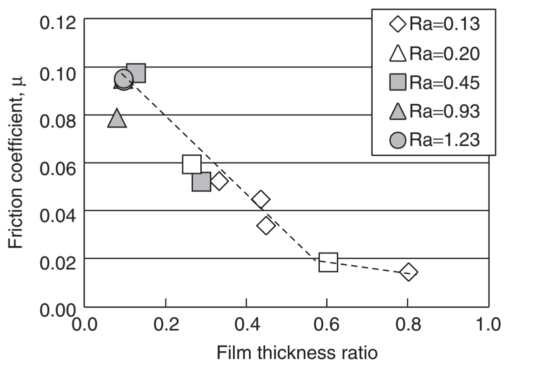


Fig. 16 Relationship between film thickness ratio and friction coefficient (GF-reinforced polyamide material)

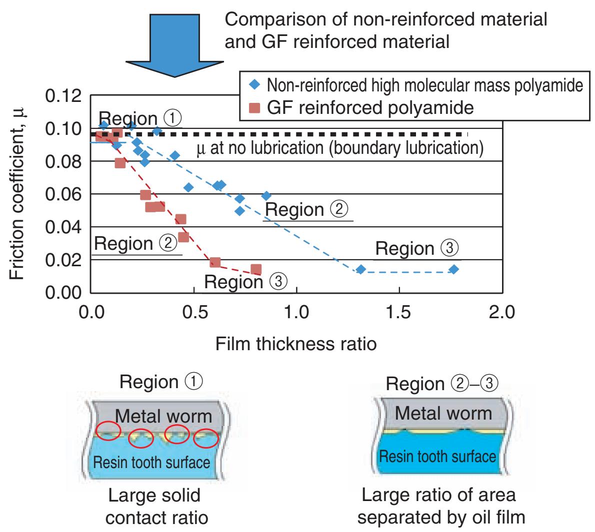


Fig. 17 Relationship between film thickness ratio and friction coefficient

4. 4 Application in actual EPS worm gear

The knowledge obtained from **4.3** is described below in an example of the practical application of the developed resin in torque reduction of actual worm gears. **Figure 18** shows the relationship between the friction coefficient, obtained from a basic test using four materials with different friction coefficients, and the rotation torque ratio of an actual worm gear. A strong correlation can be seen between the friction coefficient of the resin material and rotation torque. From this relationship, we can estimate the tooth surface roughness that will enable us to obtain the target rotation torque under prescribed temperature conditions. For a high surface roughness, it is possible for rotation torque to increase at high temperatures (80°C: temperature inside the vehicle), however it is also possible to suppress torque by maintaining the film thickness ratio. As **Fig. 18** shows, a friction coefficient at or less than 0.07 and a film thickness ratio at or above approximately 0.6 (calculated from the relationship shown in **Fig. 17**) are both necessary in order to obtain the target torque value. To obtain such a film thickness ratio, Formulas (1) and (2) show that it is necessary to set the tooth surface roughness as Ra=0.2 or less by performing an inverse operation substituting each physical property value at 80 °C within the formulas. We applied the surface roughness obtained in the gear formation conditions of hobbing and gear break-in, which enabled torque reduction in the worm gear of the reduction gear used in EPS.

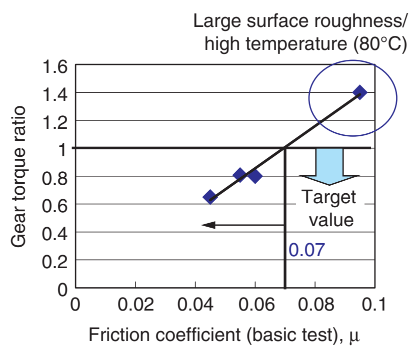


Fig. 18 Relationship between friction coefficient and gear torque ratio

5. Conclusion

We have established a basic testing method simulating a resin worm gear, identified the sliding mode of the tooth surface, and applied the gathered results to achieve a resin worm gear with long-term durability and low torque for usage in electric power steering.

1) We have developed a basic testing method that enables the isolation of wear and creep deformation, which are the causes of tooth surface deformation. We have

applied this testing method to the development of a new resin material. In addition, by clarifying creep deformation as the main factor in polyamide tooth surface deformation, we have improved the durability life of the resin worm gear by utilizing a high molecular mass polyamide.

2) The basic test has clarified that the lubrication conditions of the resin worm gear are within the IE lubrication region. We have also confirmed that within this lubrication region, the dependence of the friction coefficient on surface roughness and temperature depends in turn on the change in film thickness ratio, and that it is possible to estimate surface roughness that will achieve the target rotation torque. These technologies, which lower rotation torque by controlling the roughness of the resin tooth surface, have been applied to the resin worm gear for electric power steering.

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