Failure Mechanisms in Plastic Gears

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Introduction

Plastics as gear materials represent an interesting development for gearing because they offer high strength-to-weight ratios, ease of manufacture and excellent tribological properties (Refs. 1–7). In particular, there is a sound prospect that plastic gears can be applied for power transmission of up to 10 kW (Ref. 6).

Typical plastics, such polyamide 66, more commonly known as nylon 66, have long been known as suitable materials for gearing. It was reported that polyamide gears experienced fatigue failure before significant wear was observed (Refs. 6–7), but crack initiation and propagation mechanisms were unclear. The failure mechanisms of polyamide 66 (PA66) gears when run in like pairs are still not clear (Refs. 6–15) and, as a result, PA66 users have to substantially underrate their designs for gears.

In a previous study (Ref. 15), a rolling-sliding test rig for other types of plastic gears has proved capable of measuring both friction and wear continuously during tests. This has enabled considerable detail to be obtained about the wear processes in polymer gears. The tests are thus a versatile way of studying wear mechanisms in order to contribute to accurate life prediction for gears.

This present work was thus initiated to study wear and friction mechanisms of PA66 and its composites. It was hoped that this investigation would enable the failure mechanisms of PA66 gears to be interpreted.

Experimental Apparatus and Procedure

The twin-disc wear testing machine used in our previous work (Ref. 15) was again employed in this investigation. With this machine, measurements of both the frictional force and wear between two discs in contact can be made continuously so that both the wear process and friction behavior during tests can be monitored. Experiments were carried out either at different slip ratios under a given normal load or at different loads with a fixed slip ratio.

Slip ratio is defined here as the ratio between sliding and rolling velocities. If the tangential velocities on both contact surfaces are v_1 and v_2 respectively, then the sliding velocity is (v_1-v_2) and the rolling velocity is $(v_1+v_2)/2$.

With this testing method, the typical loading and sliding conditions of plastic gears can be simulated (Ref. 15). Gears were tested using a back-to-back test configuration (Refs. 7–8, 13–14 and 16). The materials used were PA66 (R1000) and short-glass-fiber reinforced PA66 (RFL4036) (Ref. 17). The proportion of glass fiber added was 30% by weight. To ensure proper contact along the face width of the discs, all of the specimens were prepared by machining 20 µm from the molded surface and then polishing to a surface roughness of around 5 µm.

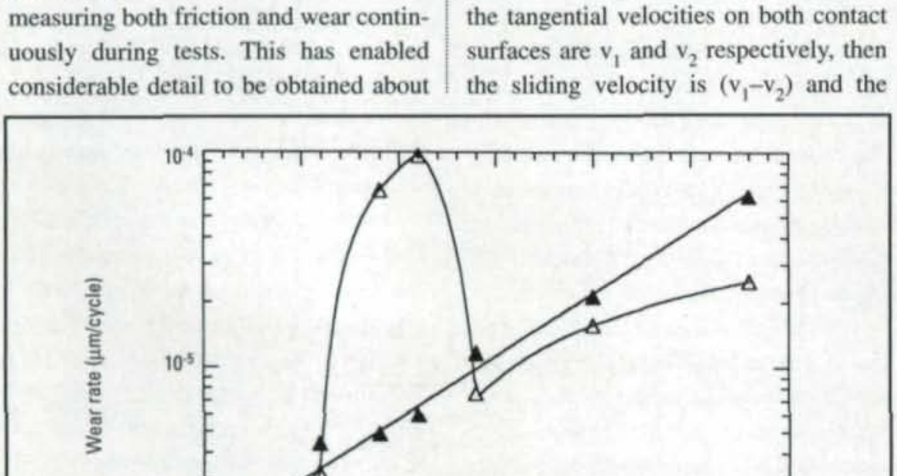

Figure 1—Wear rate versus slip ratio for PA66 at 200 N and 1,000 rpm and for 30% by weight short-glass-fiber reinforced PA66 at 300 N and 1,000 rpm.

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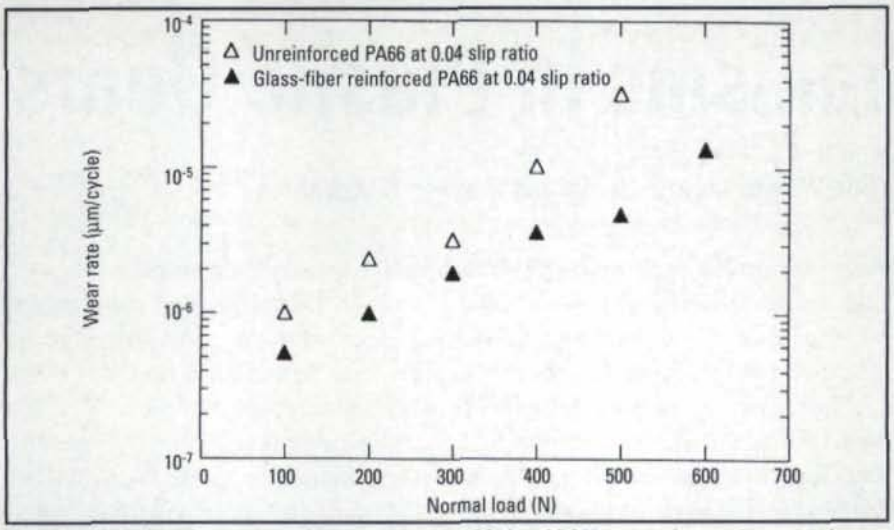

Figure 2—Variation of wear rate with normal load for PA66 and PA66 composite running at 1,000 rpm.

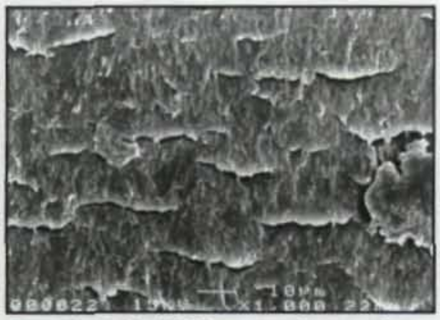

Figure 3—Typical PA66 worn surface running at 0.06 slip ratio, 32 MPa and 1,000 rpm; direction of friction force from bottom to top.

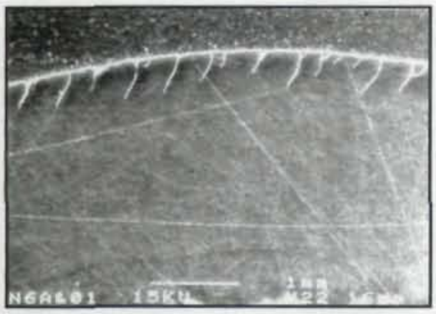

Figure 4—Transverse cross section through the disc parallel to direction of friction force running at 0.14 slip ratio, 32 MPa and 1,000 rpm; direction of friction force from left to right.

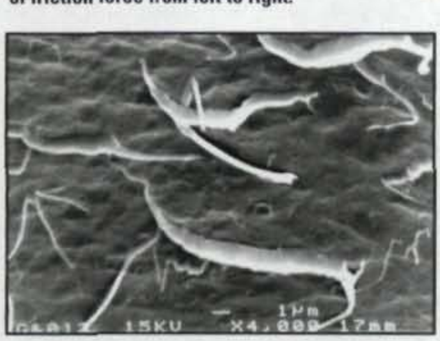

Figure 5—Geometry and dimension of typical debris to be produced on the PA66 worn surface running at a transition slip ratio of 0.11, 32 MPa and 1,000 rpm; direction of friction force from top to bottom.

Both the discs were 30 mm in diameter and 10 mm in face width.

Before testing, the samples were cleaned with methanol. They were then run at the test conditions for an extended period to remove the machining asperities and any subsurface layer affected by the manufacturing process. After running them, they were dried at 70°C for 15 hours to remove any absorbed water that might affect the measurement of wear and then weighed. Finally, they were left under atmospheric conditions for about two weeks to allow the water content to return to equilibrium conditions.

After this preliminary treatment, the specimens were remounted in the test rig in an identical position to that under which they had been run-in. Tests were run for running speed of 1,000 rpm and at a range of loads and slip ratios. The slip ratios used were between 0 and 0.28. Tests were performed under dry, unlubricated conditions at ambient temperature (22±1)°C until failure or for up to 10^7 contact cycles.

At the end of the tests, the discs were again dried at the same temperature and for the same time period as they were before they were tested. Then, the discs were weighed to measure the weight loss of each disc. With this drying procedure, the measurement of wear by weighing is accurate to about $\pm 10^{-5}$g. Finally, the worn surfaces were observed in detail by using a JEOL JSM6300 scanning electron microscope.

Experimental Results

Wear rate. Wear rate is defined here as the average depth of material removed from each disc per rolling cycle (Refs. 15 and 18–19) and was calculated by measuring the weight loss of the specimens.

Figure 1 shows how the wear rate of unreinforced PA66 varies with slip ratio for a fixed normal force of 200 N and a constant running speed of 1,000 rpm. It can be seen that slip ratio has a significant effect on wear rate. The wear rate rises slightly with an increase of slip ratio when the slip ratio is less than 0.09, at which point discolored material appears on the contact surfaces during the tests. The wear rate starts to increase sharply from 2.0×10^{-6} μm/cycle to 7.0×10^{-5} μm/cycle as the slip ratio increases from 0.06 to 0.09 and the wear rate reaches its highest value of 10^{-4} μm/cycle at a slip ratio of 0.11.

The unique characteristic property of this material is that a further increase in slip ratio from 0.11 results in a dramatic decrease in wear rate. When the slip ratio increases to 0.14, the wear rate decreases rapidly from 10^{-4} μm/cycle to 8.0×10^{-6} μm/cycle and discolored material returns to the contact surfaces. The difference between the two wear rates is more than tenfold while that between the two slip ratios is approximately 27%. After this considerable decrease, the wear rate increases slowly to 1.0×10^{-5} μm/cycle as slip ratio increases to 0.21. It is suggested that the slip ratio of 0.11 is a critical one corresponding with maximum wear rate.

Figure 2 shows the effect of normal load on the wear rate at a given slip ratio and running speed (at 1,000 rpm with a slip ratio of 0.04). It can be seen that the wear rate varies in the range from 2.0×10^{-6} μm/cycle to 4.0×10^{-6} μm/cycle. When the load is between 300 N and 500 N, the wear rate starts to increase significantly from 2.0×10^{-6} μm/cycle to 3.0×10^{-5} μm/cycle.

The effect of short-glass fibers. Reinforcement with short-glass fibers has significant effects on wear and friction. As shown in Figure 1, both wear and friction are dominated by the ability or inability of a thin layer of self-lubricating film to be formed continuously and to be retained on the surfaces in contact (Ref. 18). Similar to the case for PA66 at low slip ratios, when the film was found on the contact surface, wear debris could hardly be observed and friction coefficient was less than 0.1. Unlike PA66, once the film was disrupted, the friction coefficient of short-glass-fiber reinforced PA66 varied between 0.25 and 0.3 while that of unreinforced PA66 was in the range of 0.42 to 0.72.

Figure 2 shows the effect of normal load on the wear rate of 30% short-glass-fiber reinforced PA66 composites at a fixed slip ratio of 0.04 and a constant running speed of 1,000 rpm. Here, the wear rate increases uniformly on a logarithmic scale from 10^{-7} µm/cycle to 1.35 x 10^{-5} µm/cycle as the normal load increases from 100 N to 500 N.

Figure 1 shows the wear rate of 30% short-glass-fiber reinforced PA66 composites as a function of slip ratio for a fixed normal force of 300 N and constant running speed of 1,000 rpm. The self-lubricating film on the contact surfaces exists during all of Figure 1's wear-rate measurements. It can be seen that the wear rate increases nearly logarithmically from 10^{-6} µm/cycle to 10^{-5} µm/cycle as the slip ratio increases from 0.04 to 0.21. The wear rate is of the same order of magnitude as that of unreinforced PA66, apart from the unique peak in the wear rate.

Discussion

The wear mechanisms of discs can be used to explain wear behavior of gear teeth. Figure 3 shows the early stage of a general surface damage on a disc after running for 5.8 x 10^{6} cycles when the slip ratio was 0.06 and less than its critical value of 0.11. It can be seen that the length of these cracks varies from a couple of microns to tens of microns. These cracks eventually propagated across the whole width of the disc and were perpendicular to the direction of the friction force that moves from the bottom to the top.

Figure 4 shows a section through the disc parallel to the direction of the friction force when the slip ratio was 0.14, which is greater than its critical value of 0.11. It can be seen that the typical width between two fully developed cracks is 200–400 µm and that the distance between two sub-cracks is less than 100 µm. This can be compared with this material's typical Hertzian contact width of 400–980 µm (Ref. 19). It can be seen that, as shown in Figure 4, the main cracks initiated in the radial direction, perpendicular to the friction force. Then, they propagated in a direction at an angle to the friction force instead of perpendicular to the friction force. The depth to which the main cracks propagated varied from 150 µm to 450 µm.

Between these main cracks, there

were some sub-cracks. The main cracks propagated up to 450 μm, and the small cracks propagated to about a quarter of the depth of the main cracks. These sub-cracks eventually joined the main cracks. As shown in Figure 4, a sub-crack initiated from the contact surface and propagated in an arc towards the adjacent main crack. Since the sub-crack is wider at the contact surface than beneath it, it is suggested that the small crack was initiated from the contact surface rather than from beneath it (Ref. 19). Some cracks propagated to join their neighbor, and as a result, the material between the two cracks was fractured, severe spalling occurred and debris was formed. It was noted that the crack propagated and fractured gradually rather than suddenly. This compares well with the wear observed on a PA66 gear tooth surface, as shown in Figure 8.

Figure 5 shows a general view when the slip ratio reached its critical value, that is 0.11. The worn surface shows unique characteristics but is not discolored. A large quantity of roll-like debris is attached to the surface and tends to accumulate on the surface. It can be seen that roll-like debris consists of the middle part of the roll and two tails on both sides. The length of roll-like debris varies from tens of microns to a couple of hundred microns. The diameter of the middle part is about 1.0–10.0 μm. A tail has a very small diameter (about 0.1 μm) and is very long (about 100 μm). Most tails were broken and separated from their body during the friction process.

It was observed that before roll-like debris on the worn surface was formed, the surface suffered a very deep shear deformation and surface material moved in the direction of friction. This deformed material was gradually rolled in the direction of friction. As a result, the body of a piece of roll-like debris was formed, as shown in Figure 5. It was noted that there was, overall, a great deal of debris collected during the test and that it appeared very thin and long. Since the slip ratio along gear tooth profile in contact varies and covers the slip ratio range of 0–0.28, it is suggested that failure mechanisms of PA66 gears are severe wear due to a critical slip ratio and tooth fracture due to macrotransverse cracks on the contact surface.

The reinforcement of PA66 with short-glass fiber also has an effect on transverse cracks, which occur when PA66 is in non-conformal rolling-sliding contact. Although the wear mechanisms are very complicated, as shown in Figures 6–7, no transverse cracks on the worn surfaces were observed under a variety of test conditions.

Because of the film on the surface, as shown in Figure 7, the friction coefficient can be below 0.1, and as a result, the shear stress at the interface between the films should be low. This low shear stress may play an important role in the self-lubricating property of the composite. Also, because of the low friction coefficient, the maximum shear stress is not at the contact surface but on the subsurface, and there is little chance for a transverse crack to initiate on the contact surface (Ref. 18).

Figure 6 shows a surface in the severe wear stage obtained from a disc run at 1,000 rpm, with a 300-N load and a 0.28 slip ratio. It can be seen that the fibers on the surface were highly aligned and approximately parallel to the friction force. Figure 6 also indicates that both long and short broken fibers can move on the surface, causing alignment from an initially random distribution. It is observed that the aligned broken fibers do not remain on the surface indefinitely but are expelled out as debris. It appears that, in the rapid wear stage, wear is due to the removal of unworn, but fragmented fibers rather than to their gradual abrasion. This also compares well with the wear observed at the composite gear tooth surface, as shown in Figure 9.

These test results clearly explain why PA66 gear teeth fracture near the pitch line area with little debris, where the sliding ratio between gear teeth in mesh is very small (Refs. 6–8 and 13). From the above study of non-conformal, unlubricated rolling-sliding contact, one of the dominating factors in surface failure at low and high slip ratios is transverse crack propagation on the surfaces in contact. Also, there was not much debris before cracks fully developed and spalled. Since the slip ratio on the gear teeth in contact varies and is around 0–0.21 near the pitch line (Refs. 6–8 and 13), the behavior of crack propagation on a gear tooth surface should be similar to that on disc surfaces. In other words, these cracks on a gear tooth propagate not only across the gear face width but also into the subsurface of the gear tooth.

After a certain number of cycles, the cracks will propagate down to a depth of 0.5 mm near the dedendum of a gear tooth—in the region near initial contact on a driving gear tooth—where a high slip ratio is expected and near the pitch line area on the gear tooth where a low slip ratio occurs. For a gear of module 2 mm, a reduction of 0.5 mm of the tooth thickness near the dedendum is considerably significant. Stress concentrations at the tip of cracks are severe. Therefore, the bending stress on the dedendum of the tooth could be much higher than the tooth was designed for. Near the pitch line area on a gear tooth, corresponding to lower slip ratios, bending stresses are high since this is the position where only a pair of gear teeth are in contact. Crack propagation at lower slip ratios will also cause severe stress concentrations at the tip of cracks if the tooth is subjected to a reasonably higher bending moment. Under high bending stresses, it is suggested that the severe stress concentration results in the tooth fracture near the pitch line area on the gear tooth. Therefore, it is suggested that tooth fracture in PA66 gears is due to initiation and propagation of transverse cracks rather than creep as shown in Figure 8.

Conclusions

The failure mechanisms of polymeric (PA66 and PA66 composite) gears have been investigated by testing plastic against plastic in counter-conformal, unlubricated, rolling-sliding contact over a wide range of slip ratios, loads and running speeds. Comparisons between tests on discs at varying slip ratios and the results of gear tests under comparable conditions have been very favorable.

The wear and friction behavior of PA66 was dominated mainly by three major features: a critical slip ratio under a fixed load and running speed, macrotransverse cracks and a layer of film on the contact surface. These results corresponded closely to the failure phenomena of PA66 gears. It is suggested that the transverse cracks caused the plastic gear teeth to fracture, even near the pitch line. The macrotransverse cracks in the gear teeth on the contact surfaces are a serious disadvantage of PA66 gears.

To remedy this, the effect of reinforcements of short-glass fiber on the wear and friction behavior has been studied. Both the wear and friction properties of

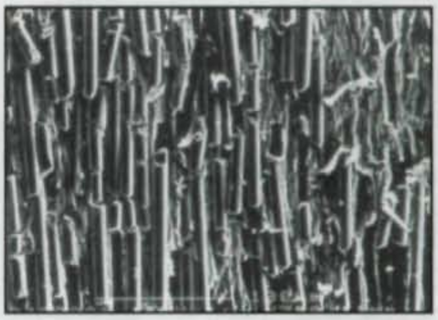

Figure 6—Aligned broken fibers on the composite surface after disruption of the surface layer (0.28 slip ratio, 63 MPa and 1,000 rpm); direction of friction force from bottom to top.

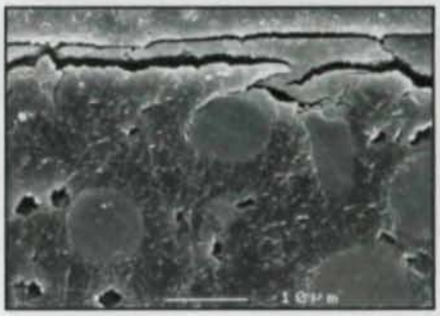

Figure 7—Transverse cross section of typical surface film on the contact surface of short-glass-fiber reinforced PA66 running at 1,000 rpm, 0.04 slip ratio and 63 MPa; direction of friction force from left to right.

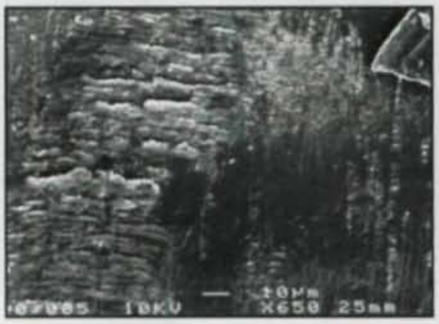

Figure 8—Typical surface topography of a PA66 gear running at 1,500 rpm and maximum contact stress of 39 MPa after 2.25×10^6 cycles.

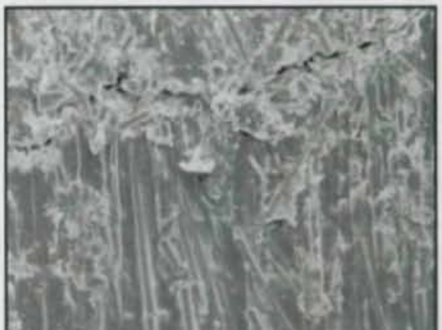

Figure 9—Typical surface feature on the contact surface of short-glass-fiber reinforced PA66 driving gear running at 1,500 rpm and 66 MPa after 2.25×10^6 cycles. (Pitch line not in photograph.)

unreinforced PA66 were improved considerably by the reinforcement of 30% by weight short-glass fiber. This reinforcement prevented both the initiation and propagation of transverse cracks on the contact surfaces that occurred in the unreinforced material. Also, it decreased both the wear rate and the friction coefficient substantially. A thin film on the contact surfaces was observed and played a dominant role in the "self-lubricating" behavior of the composite and in suppressing the transverse cracks. These results offer the prospect of enhanced applicability of polyamide 66 in gears.

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