Engagement of Metal Debris into Gear Mesh

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Management Summary

A series of bench-top experiments was conducted to determine the effects of metallic debris being dragged through meshing gear teeth. A test rig that is typically used to conduct contact fatigue experiments was used for these tests. Several sizes of drill material, shim stock and pieces of gear teeth were introduced and then driven through the meshing region. The level of torque required to drive the "chip" through the gear mesh was measured. From the data gathered, chip size sufficient to jam the mechanism can be determined.

Introduction

In some space mechanisms, the loading can be so high that there is some possibility that a gear chip might be liberated while in operation of the mechanism (Refs. 1–5). Also, due to the closely packed nature of some space mechanisms—and the fact that a space grease is used for lubrication—chips that are released can then be introduced to other gear meshes within this mechanism. In this instance, it is desirable to know the consequences of a gear chip entering in between meshing gear teeth. To help provide some understanding, a series of bench-top experiments was conducted to engage chips of simulated- and gear-material fragments into a meshing gear pair. One purpose of the experiments was to determine the relationship of chip size to the torque required to rotate the gear set through the mesh cycle. The second purpose was to determine the condition of the gear chip material after engagement by the meshing gears—primarily to determine if the chip would break into pieces and to observe the motion of the chip as the engagement was completed. This article also

Figure 1—Bench set-up for gear chip engagement testing using the NASA Glenn fatigue rig.

presents preliminary testing done with metal "debris"—other than chips from gears—namely, steel shim stock and drill bits of various sizes and diameters.

Test Equipment

The gear testing was done using a (Cleveland-based) Glenn Research Center spur gear fatigue test rig. This rig uses two identical spur gears engaged with one another. The shaft mounting is an overhung arrangement for the test gears, and the nearby bearings are roller bearings. The overhung distance from the bearing supports is approximately 32 mm (1.25 in.). Because the test machine was designed for experiments of gear fatigue, the shaft and bearing supports are relatively large and stiff. Figure 1 shows the test gearbox with a pair of test gears mounted.

Test Gears, Debris and Procedure

The gears used for this study were case-carburized and ground, and were manufactured from the steel alloy AISI 9310. The gears' design information is shown in Table 1. The gears were shot peened after hardening and before final grinding. The gears were representative of gears utilized in a space mechanism that was being simulated.

The gear design specifications and the actual mounted center distance determine the root clearance and backlash of a gear pair. The backlash and root clearances were determined for the first pair of test gears mounted on the test fixture. The backlash was measured as 0.15 mm (0.006 in.), using a standard measurement set-up (Fig. 2). The root clearance was measured as 0.94 mm (0.037 in.). The root clearance was determined by running a piece of soft material (solder) through the mesh and measuring the deformed material with a caliper.

Three types of "debris" were placed into the meshing gear teeth for testing—steel shim stock, drill bit shanks and pieces—or "chips"—from gear teeth. The shim stock materials used were steel and stainless steel. The shim thickness ranged from 0.13 to 1.88 mm (0.005 to 0.074 in.). The shims were cut to a length of approximately 2.5 to 3.8 mm (0.100 to 0.150 in.), representing approximately half of the tooth height. The shim lengths were wider than the faces of the gears and were placed to engage the full face width of the gear teeth (Fig. 3). The drill bit shanks used for these experiments had shank diameters ranging from 1.07 to 1.96 mm (0.042 to 0.077 in.).

The gear chip pieces used in this work were liberated from a spare test gear (case-carburized and ground AISI 9310 steel). Ten gear chips were used for testing. Eight of the

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Table 1—Basic Gear Dimensions	
Number of Teeth	42
Module, (Diametral pitch); mm (1/in.)	2.12 (12)
Circular pitch, mm (in.)	6.65 (0.2616)
Whole depth, mm (in.)	4.98 (0.196)
Addendum, mm (in.)	2.11 (0.083)
Chondal tooth thickness, mm (in.)	3.25 (0.1279)
Helix angle, (deg)	0
Pressure angle, (deg)	25
Pitch diameter, mm (in.)	88.9 (3.50)
Outside diameter, mm (in.)	93.14 (3.667)
Root fillet, mm (in.)	1.02 (0.04)
Measurement over pins, mm (in.)	93.87 (3.6985)
Pin diameter, mm (in.)	3.66 (0.144)
Backlash, mm (in.)	0.15 (0.006)
Tip relief, mm (in.)	0.015 (0.0006)

Figure 2—Set-up for backlash measurement using dial indicator.

Figure 3—Example of shim and placement used for engagement testing.

Table 2—Mass of the Test Gear Chips	
Chip Identification Number	Mass, (grams)
1	0.090
2	0.100
3	0.106
4	0.116
5	0.139
6	0.254
7	0.342
8	0.045
9	0.098
10	0.032

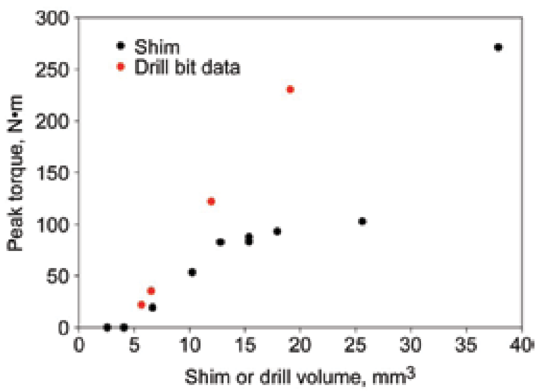

Figure 4—Relationship of volume of engaged objects to the peak torque required to rotate the gears for the NASA GRC spur gear rig using 12-pitch case-carburized steel gears.

Figure 5—Example of deformed shim stock after engagement of meshing gear teeth.

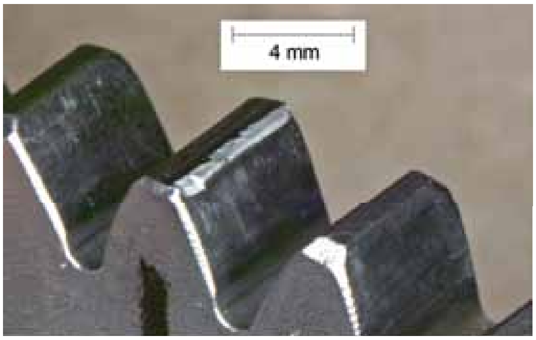

Figure 6—Example of damaged tooth tip after engagement of 1.96 mm (0.077 in.) diameter drill bit shank.

ten chips were created by scoring a mark on the gear tooth, using a small rotating cutting wheel and then striking the score line with a cold chisel. This created chips with irregular shapes, and the chips were of varying sizes (mass). Two of the gear chips were made by cutting the gear tooth using a metals lab cutting wheel. These two chips have a more-regular shape and (as compared to chips made by striking) relatively smooth edges.

To quantify the sizes of the chips, the mass of each chip was determined (Table 2). The figures that follow show at least one image with a scale marker for each chip used for testing.

Testing using the shims and drill bits was done with no lubrication on the gear teeth. For tests done with gear chips, a generous amount of a Teflon-based, space-qualified grease was applied with a brush to the teeth prior to testing. During testing, one of the gear shafts (the driven gear) was free to rotate—i.e., there was no resistive torque applied. The driving gear was rotated with a torque wrench by hand in a very deliberate manner. The peak torque reading recorded by the torque wrench was recorded as the engaged debris was rolled through the mesh.

Test Results

The results for the shim stock and drill bit testing will be discussed first.

To relate the peak torque required to roll the object through the gear mesh to the size of the object, the volume of the object was calculated—assuming the full face width of the gear was engaged and using a nominal height of 3.18 mm (0.125 in.) for the shim objects. The resulting relationship of peak torque to the object volume is provided in Figure 4. For the smallest shims tested, only a very small, nominal drag torque of the rig set-up was required to rotate the shim through mesh. The drill bit shanks required a significantly greater peak torque for an equivalent volume of shim stock material. This can be explained by the higher hardness of the drill bit shank, relative to shim stock—thereby requiring larger peak torque before any plastic deformation (or shaft deflection) of the bit shank or gear teeth takes place.

For the drill bit test data, a straight line can be passed through the data. In the data for the much softer shim stock, there initially appears to be a linear relation—up to a certain volume. Then a region of this data is nearly non-varying—and only increasing torque slightly as the volume was increased. Finally, for this data, a rather large increase in torque was recorded as the shim volume was in excess of 35 mm^3.

A plausible explanation for this can be stated as follows:

Initially, the shim stock plastically compresses and an increase in volume causes a proportional change in torque. As the volume is increased, enough of the tooth tip root clearance is still available to plastically compress the shim material, resulting in little increase in torque. Finally, as the volume of the tooth-tip-to-root-clearance is used up, the torque required to "extrude" the shim increases substantially.

In the case of the shim stock material, the main effect was elastic deformation of the support structure (shaft and bearings) and plastic deformation of the shim stock. Essentially, the shim stock was extruded. The shim stock took the shape of the root-fillet region of the driven gear, with intrusion by the tip of the driving gear (Fig. 5). The peak torque was roughly a linear function of the engaged shim volume. By visual examination, it was judged that the gear teeth were not damaged by engagement of the shim stock. For the case of the drill bit shanks, the main effect was elastic deformation of the supporting structure and some plastic deformation of the teeth. The drill bit shanks permanently deformed and damaged the tooth tip that made contact with the bit shank during the meshing process (Fig. 6).

Next, the results of testing with the gear chips are presented.

The peak torque required to rotate the gears was, to a good approximation, a linear function of the mass of the engaged chip (Fig. 7). For chip #2 of Table 2, the peak torque exceeded the measuring capacity of the torque wrench, and, since the peak torque was not known precisely, the data point was not included in the plot of Figure 7. A larger-capacity torque wrench was used for subsequent testing.

It was noted that immediately after testing, the disengaged chips were hot—indicative of the tremendous friction forces and plastic deforming work being done during the meshing process. Some of the smaller chips tended to remain in the root of the gear after the engagement. Larger chips were forced out of the mesh and were found on the table top, just below the gear mesh. The gear on the left had been rotated clockwise with the torque wrench (the input-torque direction), so the motion tended to throw and/or allow the chip to drop to the tabletop. For the two largest chips (#6 and #7 of Table 2), the engaged gear teeth were deformed and sufficiently damaged that the gears could no longer be rotated with hand torque past the damaged tooth. An image of such damage is provided in Figure 8.

Images of the chips—both before and after

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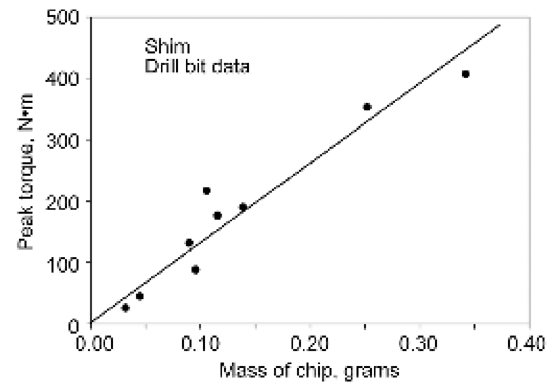

Figure 7—Peak torque recorded during chip engagement testing on GRC bench test set-up as a function of the mass of the engaged chip. The line is a linear regression of the data.

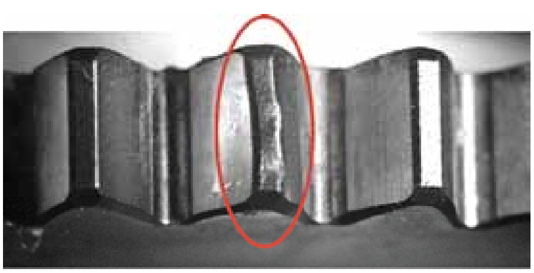

Figure 8—Severely deformed gear tooth as a result of engagement with largest chip tested, chip #7 having a mass of 0.342 grams. The gears can no longer be rotated to engage and pass through this tooth after using hand torque.

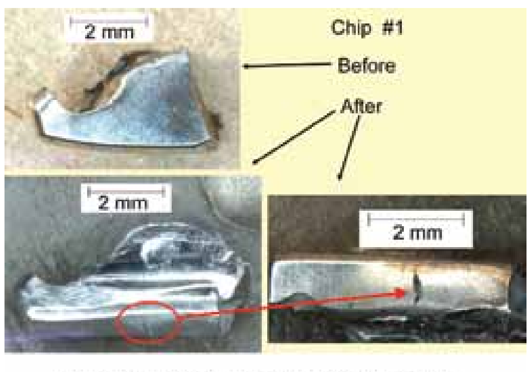

Figure 9—Chip #1, mass of chip 0.090 grams.

Figure 10—Chip #2, mass of chip 0.100 grams.

Figure 11—Chip #3, mass of chip 0.106 grams.

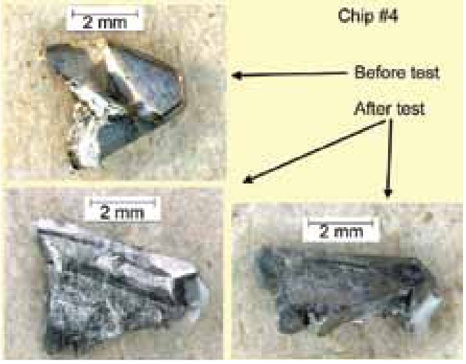

Figure 12—Chip #4, mass of chip 0.116 grams.

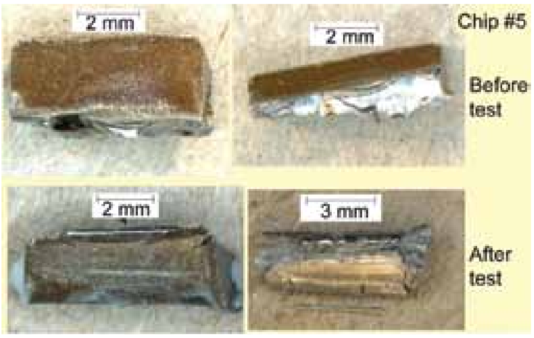

Figure 13—Chip #5, mass of chip 0.139 grams.

engagement testing—are provided in Figures 9–18. The images show that the main effect on the chips was deformation of the chips to conform to the available tooth root spaces. In general, the chips remained intact. For the case of chips #5 and #8, small pieces were liberated from the main piece, as can be seen in the figures. Chip #10 especially shows that the case-carburized material, even though generally described as brittle, can be significantly deformed and exhibit some toughness. Although the chip experienced some "tearing," the chip did not "shatter" or otherwise exhibit extreme brittleness.

Finally, all the data generated is shown in Figure 19. The drill bit and shim stock mass data was found using the data from Figure 4. The drill bit data was very comparable to the tooth chip data following the same trend. The shim stock data had a lower torque level over all the data taken. This must be an artifact of the nominal material hardness. The scatter in the tooth chip data is possibly due to non-symmetric shapes in comparison to the symmetry of the drill bit data.

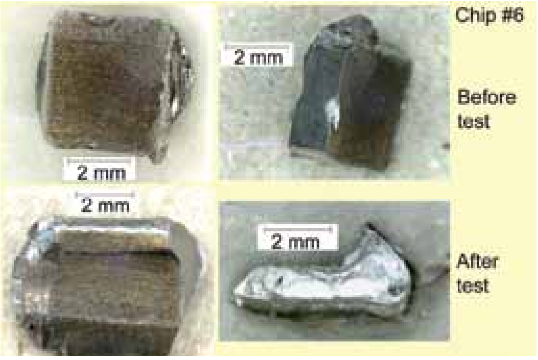

Figure 14—Chip #6, mass of chip 0.252 grams.

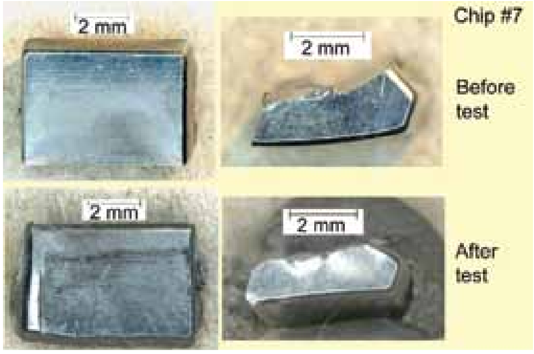

Figure 15—Chip #7, mass of chip 0.342 grams.

Conclusions

A series of bench-top experiments was conducted to provide an understanding of the engagement of metal debris into a gear mesh. The gears used for testing were case-carburized, 12-pitch spur gears made from the steel alloy AISI 9310. The metal debris that was engaged into the pair of meshing test gears was shim stock, drill bit shanks and chips of gears liberated from a test gear.

It was found that the peak torque required to rotate the gears with the object engaged was proportional to the size (mass) of the engaged object. Put another way, engaging objects of higher hardness required a significantly greater peak torque relative to an equivalent-sized object of lesser hardness.

For the largest chip sizes tested, sufficient deformation occurred to the gear teeth to prevent smooth motions of the gear when the damaged tooth is engaged.

During this study, no new chips or obvious tooth fracture occurred.

Even though case-carburized steel hardened in the manner of the aerospace gears is generally described as "brittle," the chips exhibited significant deformation and a degree

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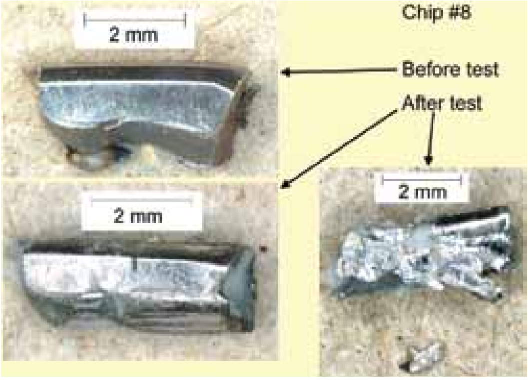

Figure 16—Chip #8, mass of chip 0.045 grams.

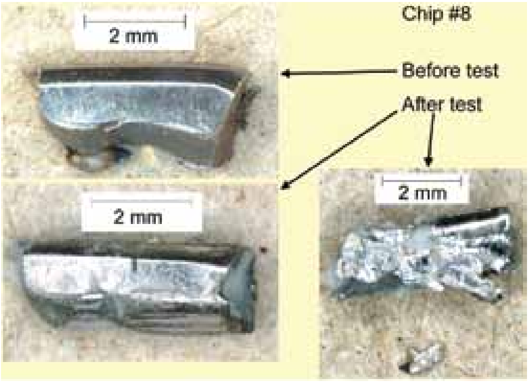

Figure 17—Chip #9, mass of chip 0.096 grams.

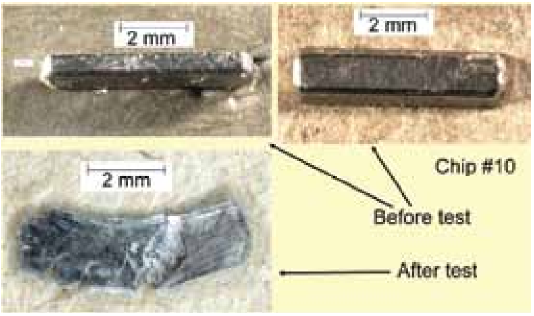

Figure 18—Chip #10, mass of chip 0.032 grams.

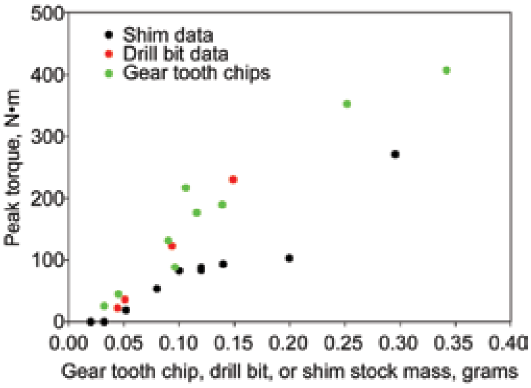

Figure 19—Debris mass versus peak torque required to rotate the gear mesh.

of toughness. Chips did not "shatter" or otherwise exhibit extreme brittle behavior.

Although large chips caused significant damage to the test gear teeth, the data generated in this study could be used for benchmarking the minimum torque needed to drive a gear chip of a certain mass for the tested system.

Therefore, the torque required to drive a gear chip through mesh would be expected to be system-dependent.

References:

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Dr. Robert F. Handschuh *possesses more than 25 years of experience with NASA and the Dept. of Defense in rotorcraft drive system analysis and experimental methods. He has served as the Drive Systems Team leader for the Tribology & Mechanical Components Branch at NASA Glenn Research Center for over 15 years. The Drives Team Leader is responsible for the technical work conducted by the Drives Team within the Mechanical Components Branch. This includes managing, advocating and directing work in this area for rotorcraft and advanced geared turbofan engine technology. Other recent activities include drive system configuration assessment for the multi-fan drive system for the Blended Wing Body aircraft, propulsion lead for the NASA Heavy Lift Rotorcraft Program and mechanical system investigation as part of the Space Shuttle Return to Flight efforts. He is currently leading research in high-speed gearing, including windage and loss-of-lubrication technology. Handschuh has developed analytical processes for conducting thermal analysis of spiral bevel gears from a fundamental, geometrical model development for gear geometry in analyzing the transient, thermal environment using a time-and-position, varying finite element modeling technique. He has successfully developed many experimental research test facilities, including: high-temperature, ceramic-seal erosion, blade-shroud seal rub test, planetary gear train test facility, spiral bevel and face gear test facility, high-speed helical gear train facility, single tooth bending fatigue test facility and high-speed windage. He is the author of many papers within the scope of his expertise.*

Tim Krantz *has worked since 1987 as a research engineer at the NASA Glenn Research Center—first as an employee of the U.S. Army and presently as an employee of NASA. He has researched many topics to improve power transmission components and systems, with an emphasis on helicopter gearbox technologies. He has also helped investigate several issues for the NASA Engineering Safety Center, including the space shuttle rudder speed brake actuator, space shuttle body flap actuator and the International Space Station solar alpha rotary joint mechanism. He is the current vice-chair of the ASME Power Transmission and Gearing Committee.*