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High-Pressure Angle Gears: Comparison to Typical Gear Designs

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Abstract

A preliminary study has been completed to determine the feasibility of using high-pressure angle gears in aeronautic and space applications. Tests were conducted in the NASA GRC Spur Gear Test Facility at speeds up to 10,000 rpm and 73 N*m (648 in.-lb) for 3.18, 2.12, and 1.59 module gears (8, 12, and 16 diametral pitch gears), all designed to operate in the same test facility. The 3.18 module (8-diametral pitch), 28 tooth, 20 degree pressure angle gears are the NASA GRC baseline test specimen. Also, 2.12 module (12-diametral pitch), 42 tooth, 25 degree pressure angle gears were tested. Finally, 1.59 module (16-diametral pitch), 56 tooth, 35 degree pressure angle gears were tested. The high-pressure angle gears were the most efficient when operated in the high-speed aerospace mode (10,000 rpm, lubricated with a synthetic turbine engine oil) and produced the lowest wear rates when tested with a perfluoroether-based grease. The grease tests were conducted at 150 rpm and 71 N*m (630 in.-lb).

Introduction

Gearing is chosen very carefully for any given application to have a sufficient strength (load capacity) and, therefore, a long life before one of the many failure mechanisms initiates and results in component failure. Operating in very hostile environment conditions, such as large temperature swings, very abrasive dust, the use of nontraditional terrestrial lubricants, etc., premature failure can result (Refs. 1 and 2). In an attempt to improve gear performance, high-pressure angle (HPA) gearing is examined for possible use in space mechanism applications. As the pressure angle of a gear mesh increases, the rate of sliding of the surfaces over each other is reduced (Refs. 3 to 6). There are limits to how much the pressure angle can be increased as the design can eventually have a contact ratio approaching one and/or the tooth top land can become pointed. An approach to circumvent this dilemma is to make the gear mesh helical and use the face contact ratio to boost the overall contact ratio to a value greater than one (Refs. 7 and 8).

Gear Design and Analysis

To understand the effects of pressure angle increase, an analysis used for gear tooth contact fatigue was utilized. A computer code developed at NASA (Ref. 9) was used to calculate various parameters, including the sliding velocity of the teeth, and the resultant gear meshing power losses. The difference in sliding velocity between two of the designs tested is shown in Figure 1. The sliding velocity as a function of meshing position is shown for the 88.9 mm (3.5 in.) center distance (1:1 ratio) and two gear types—the standard 3.18 module (8 diametral pitch) 20 degree pressure angle and the high-pressure angle gears with 1.59 module (16 diametral pitch) and a 35 degree pressure angle.

There are other design considerations when changing to a higher-pressure angle. When trying to modify a design the pressure angle can only be modified so far for any given diametral pitch before tooth pointing can be design limiting. In the design mentioned in this report, the 35 degree pressure angle required doubling the diametral pitch. This increased the tooth count from 28 to 56 teeth. Also, due to the tooth pointing issues and the size of the teeth, the gear material, and the heat treatment process employed required modification. Instead of using 9310 gear steel and carburizing, Nitralloy 135M material using a nitriding heat treatment was chosen (very thin, hardened surface layer). With this material and heat-treat

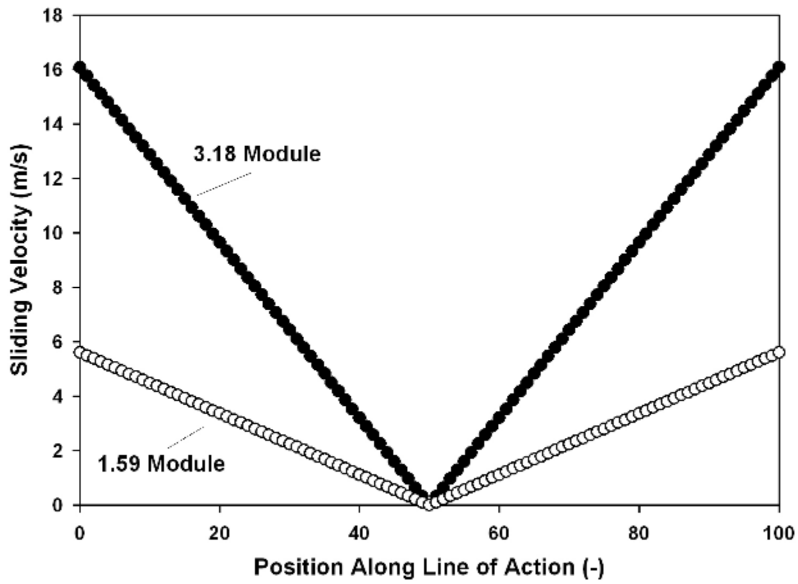

Figure 1.—Effect of gear design on sliding velocity at any point along the line of action between 3.18 and 1.59 module (8 and 16 diametral pitch) gears for a 1:1 ratio.

TABLE 1.—RESULTS FROM PERFORMANCE (EFFICIENCY) AND LOAD CAPACITY ANALYSIS CODES

	Standard gear geometry #1	Standard gear geometry #2	High pressure angle
Number of teeth	28	42	56
Gear material	AISI 9310	AISI 9310	Nitralloy 135M
Module, mm (diametral pitch, 1/in.)	3.18 (8)	2.12 (12)	1.59 (16)
Pressure angle, degrees	20	25	35
Addendum, mm (in.)	3.18 (0.125)	2.12 (0.083)	1.40 (.055)
Outside diameter, mm (in.)	95.25 (3.75)	93.17 (3.668)	91.69 (3.610)
Chordal tooth thickness, mm (in.)	4.85 (0.191)	3.25 (0.128)	2.41 (0.095)
Clearance, mm (in.)	0.79 (0.031)	0.79 (0.031)	0.19 (0.0075)
Face width, mm (in.)	6.35 (0.25)	6.35 (0.25)	6.35 (0.25)
Center distance, mm (in.)	88.9 (3.5)	88.9 (3.5)	88.9 (3.5)
Profile shift (-)	0	0	0
Backlash, mm (in.)	0.15 (0.006)	0.15 (0.006)	0.076 (0.003)
Contact ratio	1.64	1.53	1.16
Efficiency, load, and stress, at 10000 rpm, 71 N*m (630 in*lb) torque			
Efficiency, %	99.41	99.62	99.77
Tangential load, N (lbs)	1601.4 (360)	1601.4 (360)	1601.4 (360)
Radial load, N (lbs)	582.7 (131.0)	746.9 (167.9)	1121.4 (252.1)
Bending stress, GPa, (ksi)	0.246 (35.7)	0.353 (51.2)	0.308 (44.6)
Contact stress, GPa (ksi)	1.09 (158.1)	1.03 (149.3)	0.89 (129.1)

Note: Bending and contact stress found via ISO6336 (Ref. 10).

change the gears could be normally manufactured without the threat of tooth capping. Capping occurs when carburized surfaces come together at thin material regions, such as at the top of the tooth, and the induced stress field, from the heat-treating process, causes the material to fracture even without an applied load.

The designs and the results of analysis are shown in Table 1. Calculations were made for the gears operating at 10,000 rpm using a synthetic aerospace lubricant. The one item to note in the table is that while the tangential load is the same (torque), the separating force between the gears has nearly doubled with the high-pressure angle design. While the design changes reduced the gearing losses, they increased the load the bearings must carry.

ISO 6336 analysis was conducted on these three gear designs for tooth bending and contact stress. The results are shown in Table 1. From Table 1, the results between the three designs were fairly comparable with the 3.18 module (8 diametral pitch) gears having the lowest bending stress and the 1.59 module (16 diametral pitch) gears having the lowest contact stress.

Experimental Results

Test procedure: Testing was done in two different modes for the gears evaluated in this study. The basic properties of the two lubricants used are provided in Table 2. In the high-speed test mode, the gears were lubricated with a synthetic turbine engine lubricant. The gear mesh was lubricated with the jet pointing into mesh. The lubricant is gravity drained and returned to the lubricant reservoir. Lubricant temperature was measured just prior to the jet and at the exit region (drain) of the test gear cover. The load applied was measured statically using a torque wrench and was proportional to the torque actuator pressure applied. For these high-speed tests, the facility was brought up to full speed (10,000 rpm) prior to increasing the load to the maximum conditions tested.

In the low-speed, grease-lubricated mode, both the gears and the amount of grease applied were weighed prior to testing, and the gears were also weighed post test. Gears were rotated up to the 150 rpm condition where tests were run prior to increasing to the maximum load applied (pressure on torque actuator).

High-Speed Aerospace (Rotorcraft) Operation

Three-gear designs were tested in the same gearbox, at identical rotational speed, torque, lubricant, and inlet temperature. A photograph of the three designs tested is shown in Figure 2. The test rig used in this study is shown schematically in Figure 3. In this test rig, the drive motor only needs to provide enough power to overcome the losses within the geared system. A rotating torque actuator uses fluid pressure to apply torque by rotating one of the slave gears (shown in green) relative to its shaft. The test gears are shown in red in the figure. Tests were conducted at full-face width contact for all tests.

TABLE 2.—BASIC PROPERTIES OF THE TWO LUBRICANTS USED IN THIS STUDY

	Turbine engine lubricant	Space grease
Mixture	Synthetic ester blend	Base oil-perfluorinated polyether
Pour point, °C (°F)	–62 (–80)	–73 (–100)
Specific gravity	1	1.85
Viscosity, cSt. at 38 °C, 100 °F	29.2	148
Viscosity, cSt. at 100 °C, 210 °F	5.3	45

Figure 2.—Left to right: 3.18, 2.12, and 1.59 module (8, 12, and 16 diametral pitch) test gears.

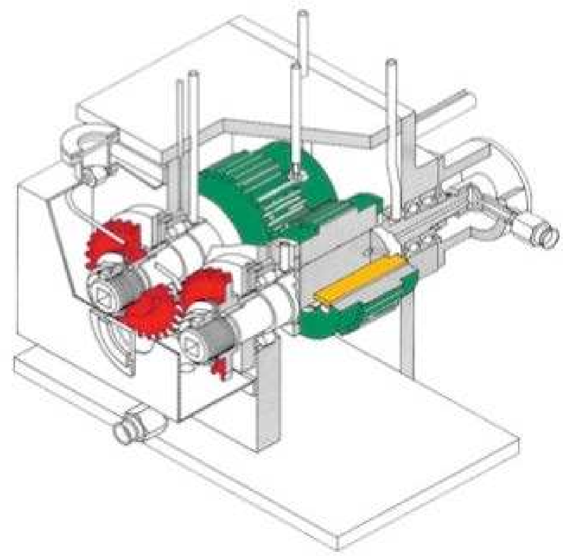

Figure 3.—Test facility at NASA Glenn Research Center.

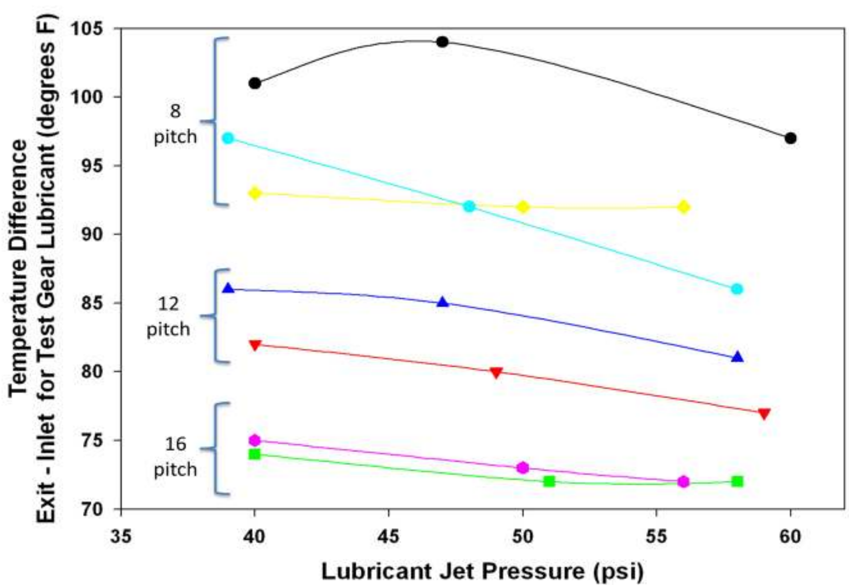

Figure 4.—Temperature results for three different gearing configurations. All tests conducted at 10,000 rpm, 71 N*m (630 in.-lb). Flow rate at 40 psi 8.1 ml/s (0.13 gpm).

The gears were tested at three levels of lubricant jet pressure (flow). The lubricant inlet and outlet temperatures were monitored throughout the tests. The data acquired for each gear design is shown in Figure 4. Seven different tests were conducted to generate the data shown in Figure 4. A comparison of the results indicates there is a definite difference between the three gear designs.

The lubricant temperature rise across a gearbox is a function of the gear meshing losses and any resultant windage losses. Pitch line velocity for these tests was 48.8 m/s (9160 ft/min). Finer pitch teeth with high-pressure angle showed an improvement resulting in a lower temperature change across the gearbox. Lubricant jet pressure (flow) into the gearbox had a lesser effect for each of the gear configurations, but there was a definite trend of lower temperature difference as the jet pressure was increased. It would be expected that heat removed from the gearbox should be related to $\dot{m}C_p\Delta T$ (lubricant flow rate, heat capacity of the lubricant, and the temperature difference between inlet and outlet for the lubricant) if other effects are neglected.

The high-pressure angle gears were included in two extensive operational tests on the same pair of gears. The gears were run full-face width at the same conditions as the lubricant tests previously discussed. Each test was run continuously at 10,000 rpm, 73 N*m (648 in.-lb) torque, for 300 million cycles. The gears are shown in Figure 5 before testing and in Figure 6 after testing. The post-test gear surfaces appeared to be in excellent condition, as shown in Figure 6. The only anomaly that occurred was a chip at the flank edge shown in Figure 7. The chipped edge is thought to be due to installation interference that occurred between the two test gears.

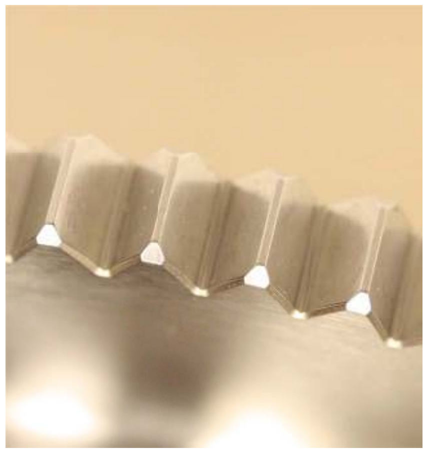

Figure 5.—Photograph of HPA gears before testing.

Figure 6.—Photograph of HPA gears after testing.

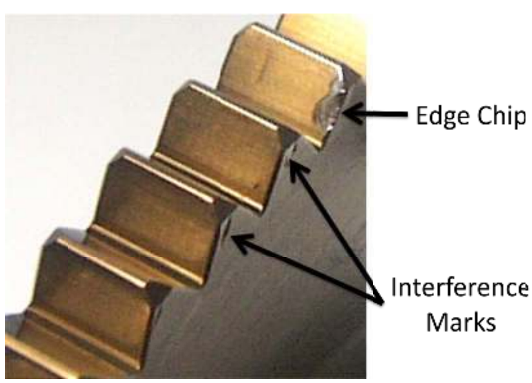

Figure 7.—Photograph of HPA gear flank end chip.

Perfluoroether (Space Qualified) Grease Tests

The spur gears were lubricated with a grade 2, perfluoroether based grease commonly used in space (see Table 2). In space applications, perfluoroether-based grease is typically used because of its extremely low vapor pressure, to minimize out-gassing in a vacuum. Here, the wear study was conducted on the same test rig as previously described, except instead of spraying synthetic lubricant oil onto the meshing gears, they were lubricated with the space grease. The purpose of this part of the study was to investigate gear surface wear and to develop methods for keeping grease on the profile of the gear teeth.

The specifics of this test involve the previous three gear designs, 3.18, 2.12, and 1.59 module (8 pitch, 12 pitch, and 16 pitch) gears (refer to Fig. 2). The study consisted of 500,000 cycles for each set of gears with and without a grease retention shroud. The gears where rotated at 150 rpm at a torque equal to 71 N*m (630 in.-lb). The grease was applied to the gears using a syringe before testing. The weight of the grease applied was measured before each test. Without the gear shrouds, significant wear was found in the 3.18 and 2.12 module (8 and 12 pitch) gears, with rust and black colored debris found in the gearbox housing cover (Fig. 8).

The 1.59 module (16 pitch) gears generated little debris, less than one half of the mass loss of the 2.12 module (12 pitch) gears and about one fourth of the mass loss of the 3.18 module (8 pitch) gears (Table 3). The teeth of the 3.18 and 2.12 module (8 and 12 pitch) gears indicated severe wear, primarily due to the meshing teeth squeezing out the perfluoroether-based grease allowing metal-to-metal contact.

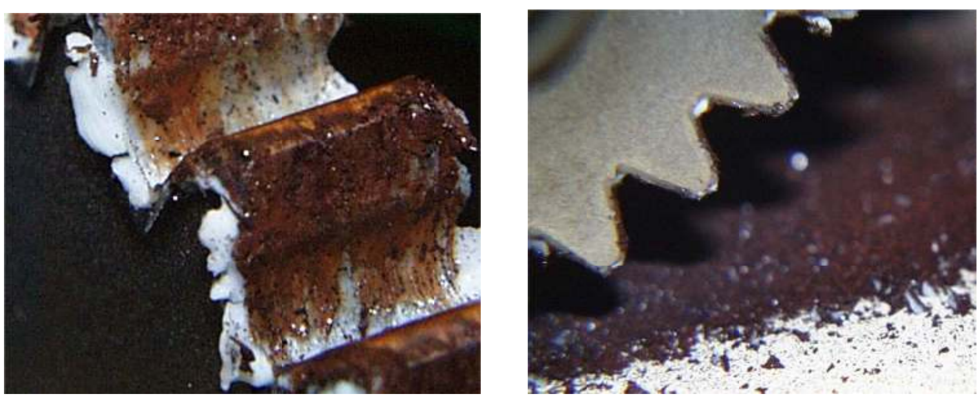

Figure 8.—Left: shows rust forming on the gear profile. Right: shows rust debris from metal on metal contact. (Both photos from a 2.12 module (12-diametral pitch) gear test.)

TABLE 3.—RESULTS FROM SPACE GREASE TESTING: 71 N*M (630 IN.-LB) TORQUE AT 150 RPM

Module (pitch)	Material	Revolutions or cycles	Grease applied (g)	Gear mass loss (g)	Average initial grease loss (g)	Shroud	Shroud grease applied (g)
3.18 (8)	M50	400000	1.425	0.15	-------	No	0
3.18 (8)	M50	400000	1.425	0.15	-------	No	0
3.18 (8)	9310	500000	1.19	0.95	0.305	No	0
3.18 (8)	9310	500000	1.19	0.44	0.305	No	0
2.12 (12)	9310	500000	1.02	0.57	0.04	No	0
2.12 (12)	9310	500000	1.02	0.51	0.04	No	0
1.59 (16)	Nitralloy 135M	500000	1.005	0.27	0.795	No	0
1.59 (16)	Nitralloy 135M	500000	1.005	0.22	0.795	No	0
1.59 (16)	Nitralloy 135M	500000	0.755	0	0	Yes	1.075
1.59 (16)	Nitralloy 135M	500000	0.755	0	0	Yes	
2.12 (12)	9310	500000	1.285	0.25	0	Yes	2.215
2.12 (12)	9310	500000	1.285	0.3	0	Yes	
3.18 (8)	9310	500000	0.66	0.46	0	Yes	2.805
3.18 (8)	9310	500000	0.66	0.37	0	Yes	

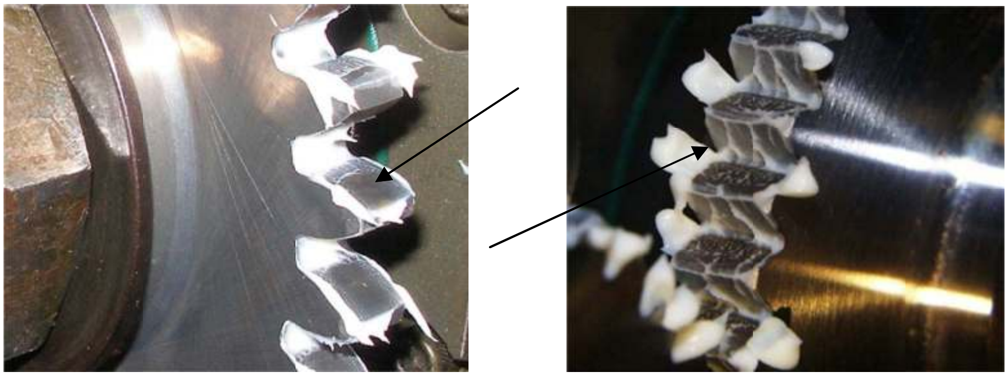

Figure 9.—Left: 20 degree pressure angle gear (3.18 module (8 Pitch)). Right: High-pressure angle gear 1.59 module (16 Pitch).

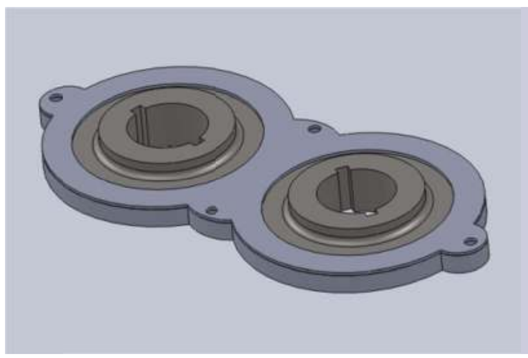

Figure 10.—Grease retention shroud CAD model.

In Figure 9(a) photograph of the trapped grease on the tooth profile of the 3.18 module (8 diametral pitch) and high-pressure angle gear can be seen. Both photographs in Figure 9 show the gears right after grease application. The static roll through mesh with the grease just applied demonstrates the need for some type of gear shrouding, as the grease is pushed to the root area and axially off the gear tooth-meshing surface.

As a result of losing the grease, the gears experiencing accelerated wear. Metal-to-metal contact between the two mating gears left debris, discoloration, and rusting wear particles in the grease. As soon as the gears started rotating the grease would be lost to the outer walls of the housing, thus not serving the lubrication purpose.

Shrouding was needed to catch the grease for redistribution into the mesh. The shrouds were designed with a fairly tight clearance to the gears in axial and circumferential directions. The radial clearance to the tooth tips was 0.51 mm (0.020 in.), and axial clearance was approximately 0.20 mm (0.008 in.) on each side. The gears performed quite differently with the use of the grease retention shrouds. Each gearing configuration lost less mass due to wear than the previous non-shrouded gears.

A detailed model of the shrouds for each gear mesh was developed using a commercially available computer-aided design software (Ref. 11) and then produced using stereolithography (Fig. 10). As shown in Table 3, the high-pressure angle gears had zero mass loss for 500,000 cycles with the shrouds. The other two gear designs 3.18 and 2.12 module (8 and 12 pitch) gears had reduced mass loss. The shroud's ability to hold the grease, that is, flung off the gears and redistribute it to the gear system, proved that this concept will reduce gear wear. In Figure 11(a) detailed explanation of the shrouds in action is shown. Future investigation should be directed toward optimizing shrouding configurations to promote grease re-application and retention to promote good lubricating conditions.

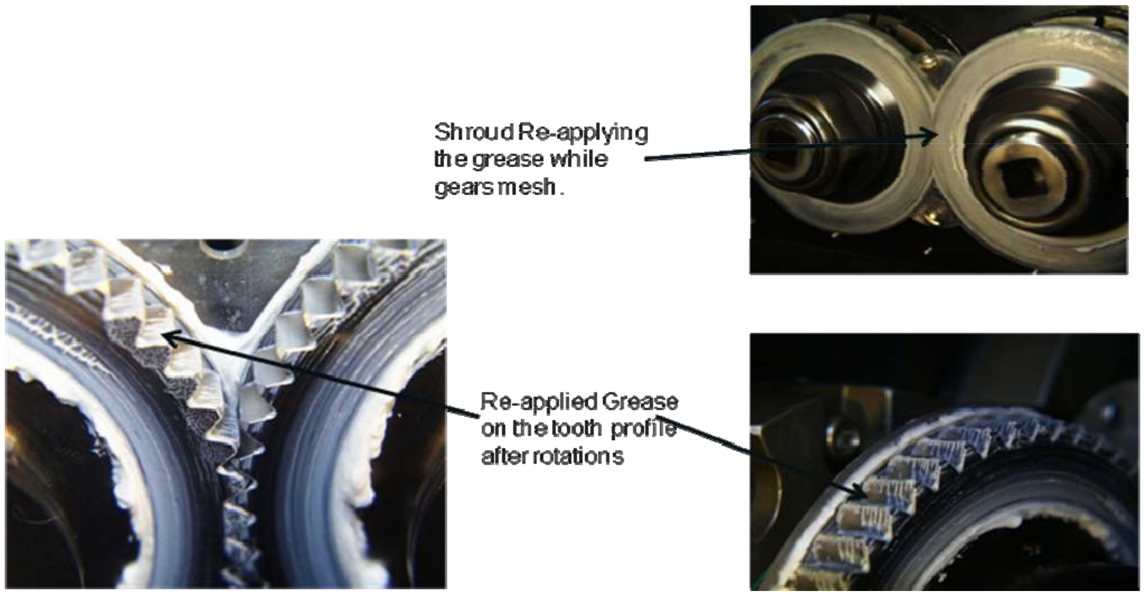

Figure 11.—Shrouds on the high-pressure angle gears during tests.

Conclusions

Based on the results obtained in this study, the following conclusions can be drawn:

1. High-pressure angle spur gears (35 deg pressure angle) running at high-speed provide improved performance with similar bending and contact stress over more traditional gear pressure angles (20 deg). This was verified via analytical computer codes for efficiency, bending and contact stress analysis, as well as through experimental testing.
2. A general trend found from the experimental testing at identical conditions in the aerospace, jet-lubricated configuration was that the higher the pressure angle, the lower the temperature increase of the lubricant across the gearbox. This is an indication of the improved efficiency.
3. The space grease-lubricated tests conducted at 150 rpm and high load requires shrouding of the gear mesh to produce lower wear rates.
4. The high-pressure angle gears appeared to be better suited to the low-speed, high load, grease-lubricated conditions compared to the 3.18 and 2.12 module (8 and 12 pitch) gears with perfluoroether-based space grease.

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