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High Pressure Angle Gears: Preliminary Testing Results

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High Pressure Angle Gears: Preliminary Testing Results

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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 Glenn Research Center (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° pressure angle gears are the GRC baseline test specimen. Also, 2.12 module (12-diametral pitch), 42 tooth, 25° pressure angle gears were tested. Finally 1.59 module (16-diametral pitch), 56 tooth, 35° 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 good strength (load capacity) and therefore a long life before one of the many failure mechanisms initiates and results in component failure. When further adding very hostile environment conditions such as large temperature swings, very abrasive dust, and the use of non-traditional terrestrial lubricants, etc., further complicates the problem (Refs.1 and 2). In an attempt to improve gear performance, high-pressure angle gearing is going to be examined carefully 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 far the pressure angle can be increased as the design can eventually have a contact ratio approaching one and/or the tooth top land becomes pointed. The contact ratio is the average number of teeth in mesh for a given gear design. A way around this dilemma is to make the gear mesh helical and use the face contact ratio to boost the overall contact ratio greater than one.

Background

This idea comes from work being conducted by the technical universities in Germany (Refs. 7 and 8), where there was interest in a gearbox that could be run without a lubricant such as grease or oil circulation. Their idea showed promise as the results of their testing is shown in Figure 1.

From the data shown in Figure 1, the gear load-dependant losses could be reduced extensively by going from a conventional pressure angle (~20°) to very high-pressure angle (~40°). Also shown in Figure 1 is an approximation of how the tooth appearance differed, more but smaller teeth and lower module (or increased diametral pitch).

To understand what was happening, an analysis of the geared system testing at GRC for gear tooth contact fatigue was performed. A computer code developed at NASA (Ref. 9) was used to calculate various parameters including the sliding velocity of the teeth over one another, and the resultant gear meshing power losses. The difference in sliding velocity between the two designs is shown in Figure 2 for the 88.9 mm (3.5 in.) center distance (1:1 ratio) and for two gear types—the standard 3.18 module (8 diametral pitch), 20° pressure angle and the high pressure angle gears with 1.59 module (16 diametral pitch) and a 35° pressure angle.

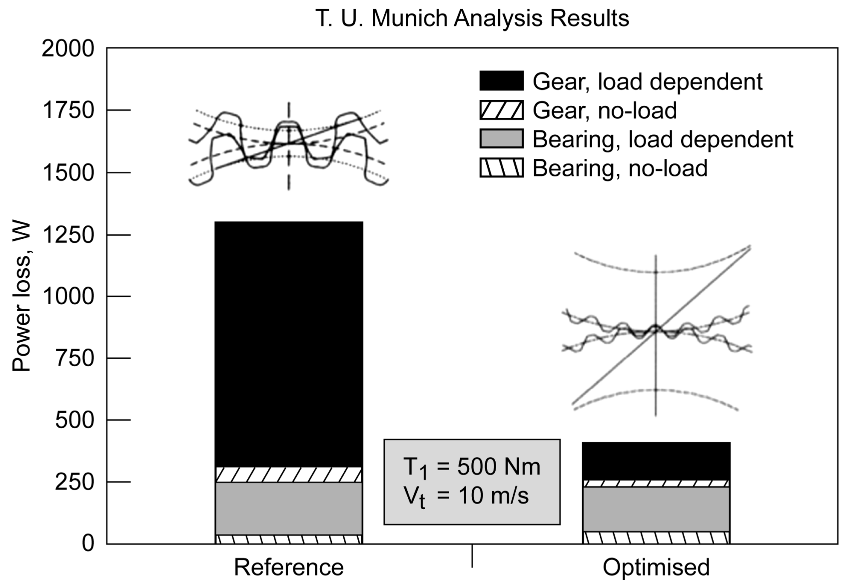


Figure 1.—High pressure angle results from testing (Ref. 7).

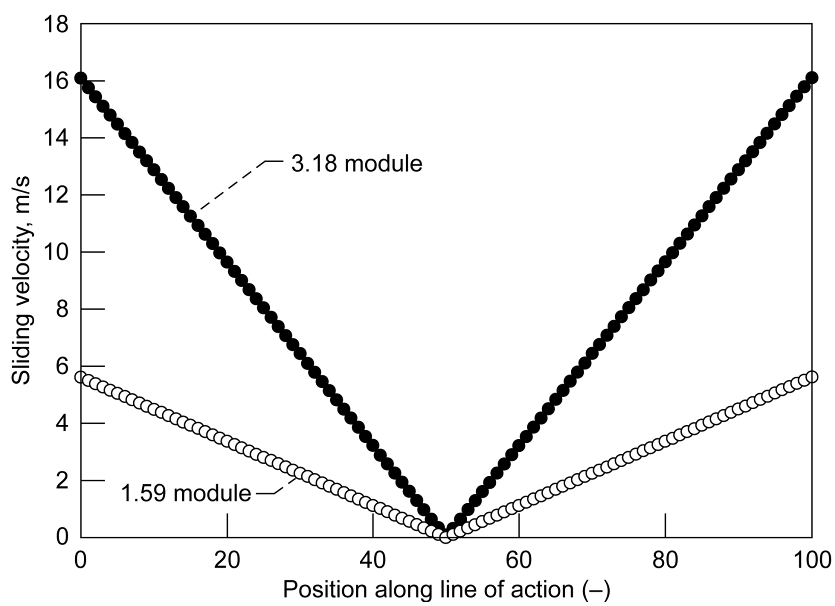


Figure 2.—Effect of gear design on sliding velocity at any point along the line of action between a 3.18 and 1.59 module (8 and 16 diametral pitch) gears for a 1:1 ratio.

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° 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 heat treatment process needed to be modified. Instead of using 9310 gear steel and carburizing, Nitralloy 135M material using nitriding heat treatment was chosen (very thin, hardened surface layer). With this material and heat-treat change the gears could be normally manufactured without the threat of tooth capping. Capping occurs when carburized surfaces come together at thin material region, such as at the top of the tooth, and the induced stress field, from the heat treating process, causes the material to fracture without any applied load.

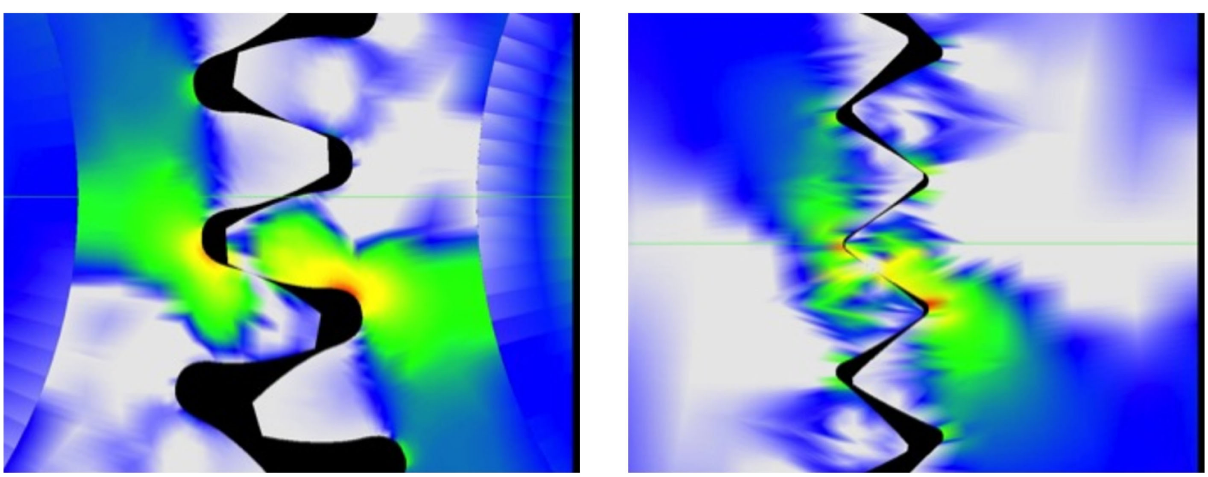
The design differences and the results of analysis are shown in Table 1. These calculations were made for the gears operating at 10,000 rpm using a synthetic aerospace lubricant. The one item to note in the table is that while the tangential load is the same (torque) the separating force between the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

TABLE 1.—RESULTS FROM PERFORMANCE (EFFICIENCY) AND FINITE ELEMENT ANALYSIS CODES

	Standard Gear Geometry	High Pressure Angle
Number of teeth	28	56
Module, mm (diametral pitch, 1/in.)	3.18 (8)	1.59 (16)
Pressure angle, degrees	20	35
Contact ratio	1.64	1.31
Efficiency at 10,000 rpm, comparable torque, %	99.43	99.76
Maximum principal stress, GPa, (ksi)	0.612 (88.8)	0.667 (96.7)
Maximum contact stress, GPa (ksi)	1.816 (263.4)	1.611 (233.6)
Tangential load, N (lb)	12811 (2880)	12811 (2880)
Radial load, N (lb)	4662 (1048)	8972 (2017)

Also a 2-D finite element analysis was conducted on these two gear designs. Tooth bending stress results are shown in Figure 3 (Ref. 10). From Table 1, the results between the two designs were fairly comparable with the 3.18 module (8 diametral pitch) gears having a lower bending stress and the 1.59 module (16 diametral pitch) gears having a lower contact stress.

Finite Element Analysis (2-D)



28 by 28 gear mesh

56 by 56 gear mesh

Figure 3.—Finite element models for the two gear designs—maximum values for contact and bending stress are shown in Table 1.

Experimental Results

Test Procedure

As already mentioned, testing was done in two different modes for the gears evaluated in this study. 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, the gears and 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 (pressure on torque actuator) applied.

High-Speed Aerospace (Rotorcraft) Operation

In addition to the two gear systems, a third intermediate gear system was used for initial test validation. This third design has a 2.12 module (12 diametral pitch) with a 25° pressure angle. A photograph of the three designs is shown in Figure 4.

These gears were all tested in the same gearbox, at identical rotational speed, torque, and lubricant and inlet temperature. The test rig used is shown schematically in Figure 5. 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 relative to its shaft (shown in green). The test gears are shown in red in the figure. Tests were conducted at full-face width contact for all tests.

The gears were run at three different levels of lubricant jet pressure (flow). The lubricant inlet and outlet temperatures were monitored throughout the tests. The data taken from the three designs are shown in Figure 6. The lubricant used has a kinematic viscosity of 5 cSt at 93 °C (200 °F).

As can be seen from the data in Figure 6, there is a definite difference between the three gear designs. The lubricant temperature rise across the 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.

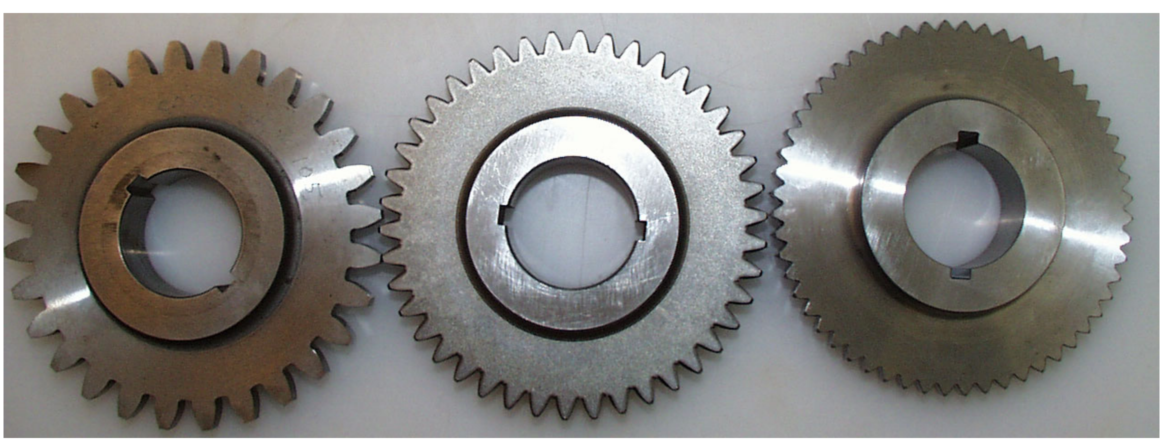


Figure 4.—Left to right; 3.18, 2.12, and 1.59 module (8, 12, and 16 diametral pitch) test gears.

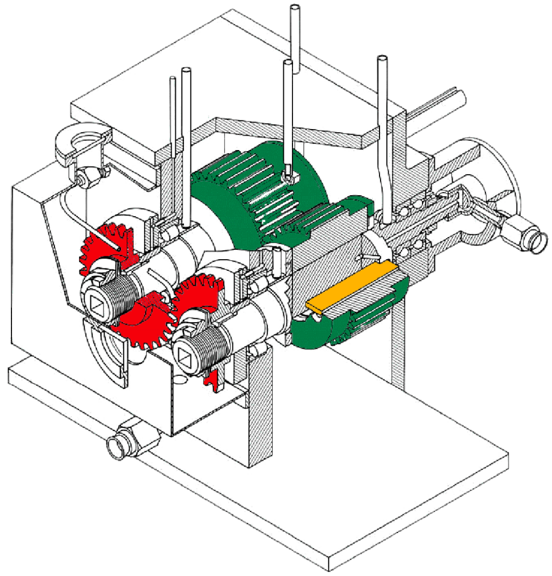


Figure 5.—Test facility at NASA Glenn Research Center.

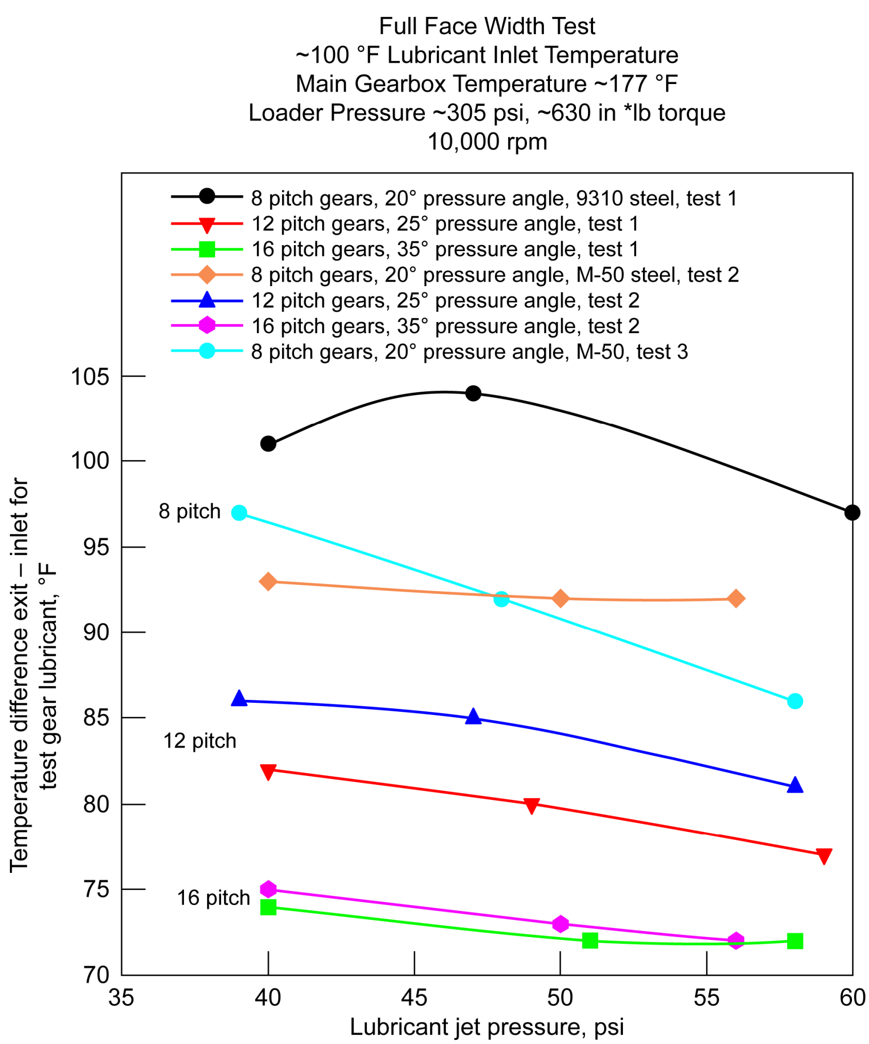


Figure 6.—Temperature results for three different gearing configurations. All test conditions were repeated.

Also, the high-pressure angle gears were tested 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 mentioned in Figure 6. These two tests were run non-stop at 10000 rpm, 73 N*m (648 in.*lb) torque, for 3×10^8 cycles. The gears are shown in Figure 7 before testing and in Figure 8 post-test conditions. The surfaces appeared to be in excellent condition as shown in Figure 8. The only anomaly occurred as a chip at the flank edge as can be seen in Figure 9. The chipped edge is thought to be due to installation interference that occurred between the two test gears.

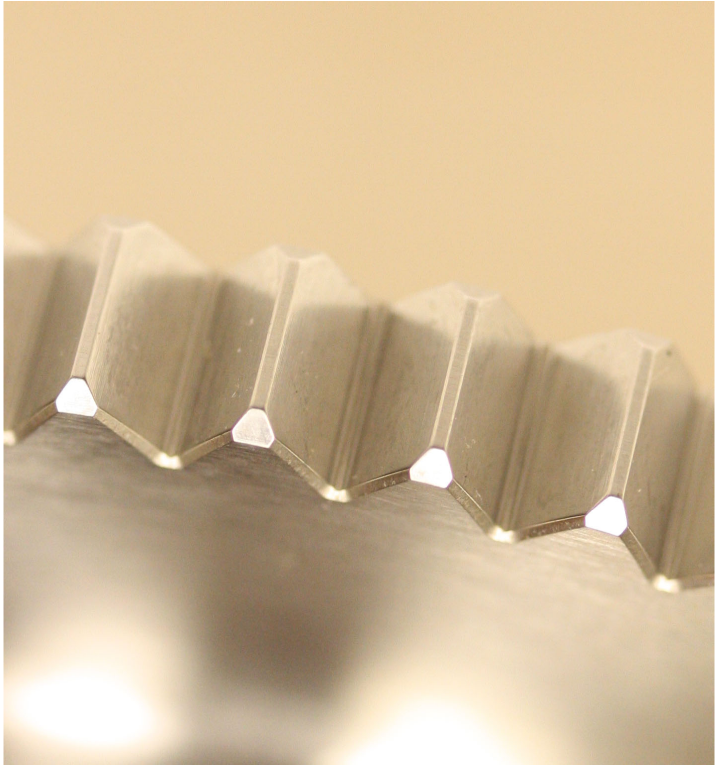


Figure 7.—Photograph of HPA gears before testing.

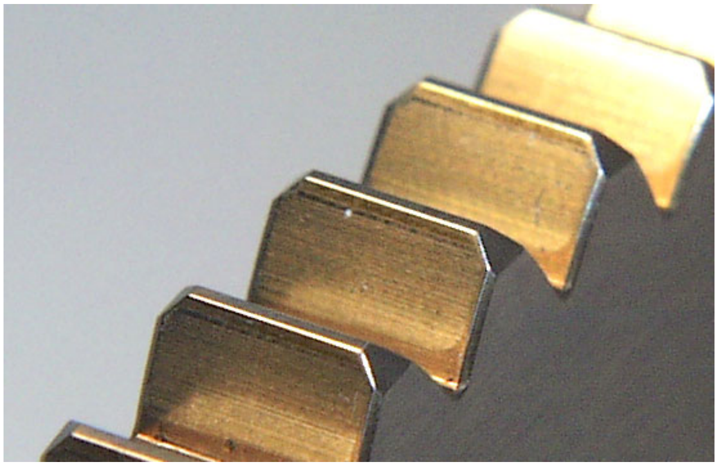


Figure 8.—Photograph of HPA gears after testing.

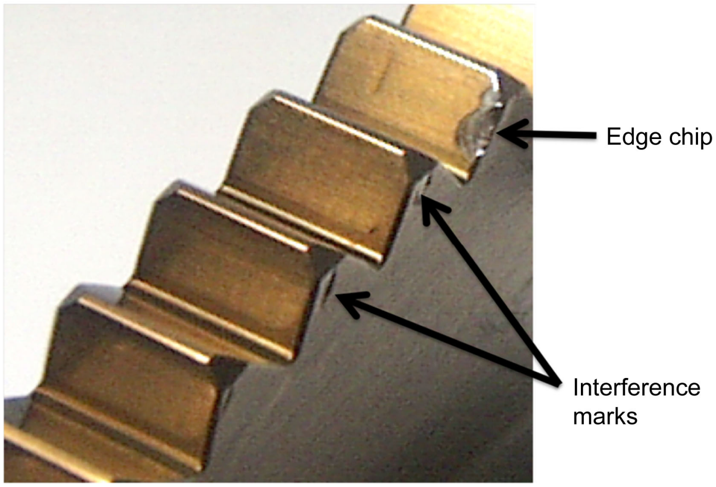


Figure 9.—Photograph of HPA gear flank end chip.

Perfluoroether (Space Qualified) Grease Tests

The spur gears were lubricated with a grade 2, perfluoroether-based grease commonly used in space. 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 space grease. For space applications, perfluoroether-based grease is typically used because of its extremely low vapor pressure, to minimize out-gassing in a vacuum. The purpose of this part of the study was to investigate gear surface wear and 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 as shown in Figure 4. 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 amount of grease applied was determined (weighed) 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. 10).

The 1.59 module (16 pitch) gears generated little debris, less than half of the mass loss of the 2.12 module (12 pitch) gears and about a fourth of the mass loss of the 3.18 module (8 pitch) gears (Table 2). The teeth of the 3.18 and 2.12 module (8 and 12 pitch) gears showed scaling, crowning, and pitting wear. This is mainly due to the meshing teeth squeezing out the perfluoroether-based grease resulting in metal-to-metal contact.

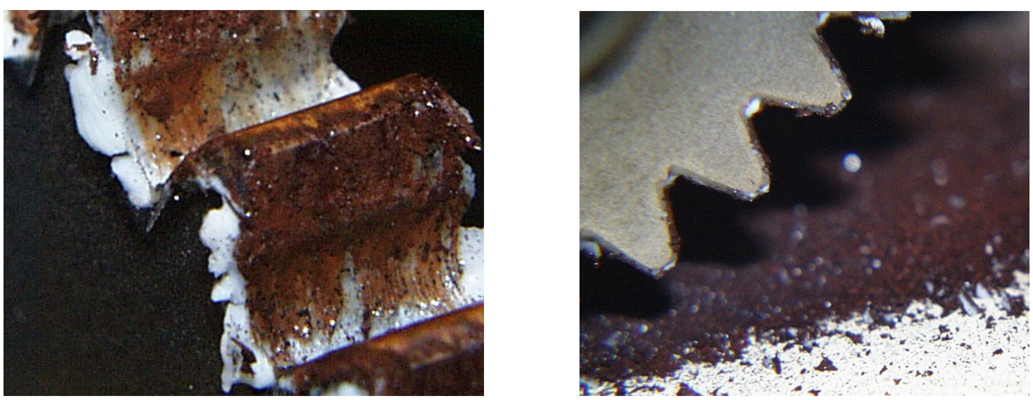


Figure 10.—Left shows rust forming on the gear profile. Right shows rust debris from metal on metal contact. (Both photos from a 2.12 module (12-diametral pitch) gear test).

TABLE 2.—RESULTS FROM SPACE GREASE TESTING. 1.72 MPa (250 psi) LOADER PRESSURE 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	

In Figure 11 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 11 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 the grease being forced out of the contact, 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 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.

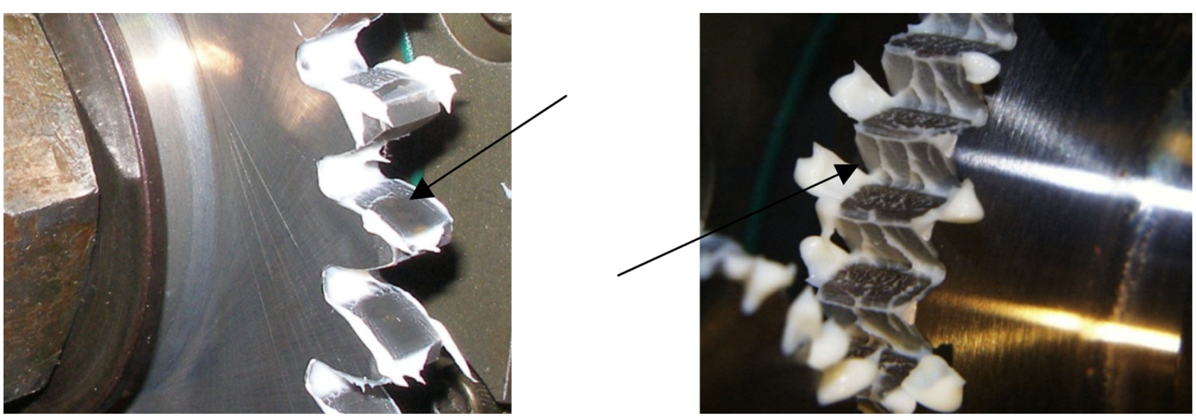


Figure 11.—Left, 20° pressure angle gear (3.18 module (8 pitch)). Right, High-pressure angle gear 1.59 module (16 pitch).

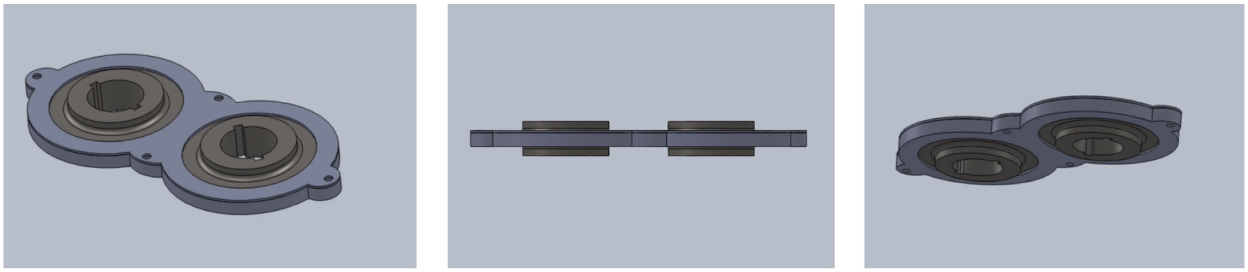


Figure 12.—Grease retention shroud CAD model.

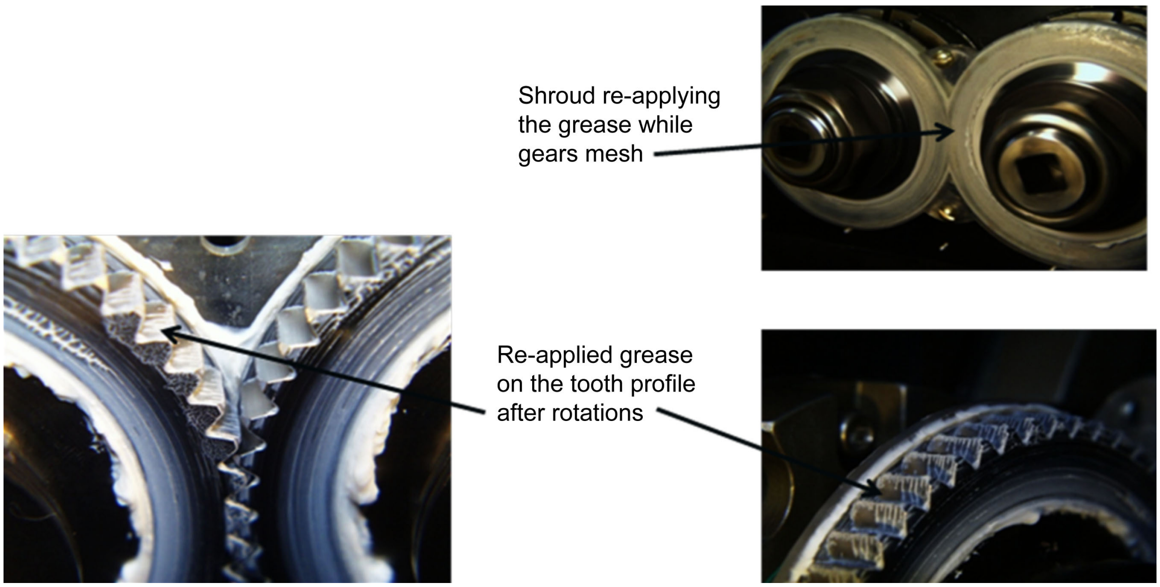


Figure 13.—Shrouds on the high-pressure angle gears during tests.

A detailed model of the shrouds was made for each gear mesh in commercially available computer aided design software (Ref. 11) and then produced using stereo-lithography (Fig. 12). As shown in Table 2, 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 in the grease that is flung off and redistribute it to the gear system proved that this concept will reduce gear wear. In Figure 13 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.

Conclusions

Based on the results attained from this study the following conclusions can be drawn:

(1) High-pressure angle spur gears (35° pressure angle) running at high speed provide improved performance with similar bending and contact stress over more traditional gear pressure angles (20°). This was verified via analytical computer codes for efficiency, finite element analysis, and experimental tests.

(2) A general trend found in the experimental testing was that the higher the pressure angle, the lower the temperature-increase of the lubricant across the gearbox while being tested at identical conditions in the aerospace, jet-lubricated configuration. 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 this 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 greases.

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Andrew J. Zakrajsek, NASA Glenn Research Center, summer intern.

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