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Simulated Single Tooth Bending of High Temperature Alloys

Robert F. Handschuh
Glenn Research Center, Cleveland, Ohio

Christopher Burke
ASRC Aerospace Corporation, Cleveland, Ohio

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Abstract

Future unmanned space missions will require mechanisms to operate at extreme conditions in order to be successful. In some of these mechanisms, very high gear reductions will be needed to permit very small motors to drive other components at low rotational speed with high output torque. Therefore gearing components are required that can meet the mission requirements. In mechanisms such as this, bending fatigue strength capacity of the gears is very important. The bending fatigue capacity of a high temperature, nickel-based alloy, typically used for turbine disks in gas turbine engines and two tool steel materials with high vanadium content, were compared to that of a typical aerospace alloy—AISI 9310. Test specimens were fabricated by electro-discharge machining without post machining processing. Tests were run at 24 and at 490 °C. As test temperature increased from 24 to 490 °C the bending fatigue strength was reduced by a factor of five.

Introduction

In the future, space mechanisms will be required to operate in some very challenging environmental conditions. The conditions are far beyond normal Earth-bound requirements for mechanisms and at these conditions basic data for gear tooth failure mechanisms are non-existent.

Many materials can operate at extreme conditions but are not used in gear-driven mechanisms. A typical use of nickel-based alloys is in gas turbine engine disks, blades, and stator assemblies that require good strength at elevated temperatures. The operating temperature for the application being discussed in this paper is not an issue with a nickel-based alloy such as Inconel. The basic material properties of the four materials tested are shown in Table I. The two high-vanadium content tool steel materials tested are typically used for dies, punches, forming tools etc. These materials were compared to AISI-9310 aerospace gear steel (Refs. 1 to 4).

A hypothetical space mission was devised with the following environmental and design requirements: reduction ratio of ~ 2500:1; operational temperature from 20 to 470 °C; input speed ~ 2500 rpm, input torque ~ 0.18 N*m; and survive for at least 1 hr at the elevated temperature. The mechanism must also be extremely compact and lightweight. Under the mission expected, the components are expected to operate without typical lubrication, yet the wear of the mechanism must be minimized. Of the requirements identified, the need to operate at 450 °C and a 2500:1 ratio are the most challenging.

In previous space mechanisms, large reduction ratios have been achieved using harmonic drives (Ref. 5). In this study, however, other mechanism arrangements are being considered, as the deflection of the flex spline at extreme temperature conditions might be a greater challenge than gears operating at these elevated temperature conditions.

TABLE I.—PROPERTIES OF MATERIALS TESTED (REFS. 1 TO 4, AND 6)

	Nickel-based alloy	Tool steel #1	Tool steel #2	Aerospace gear steel
Elements (% by weight)	Nickel (plus cobalt) 50 to 55% Chromium 17 to 21% Niobium (plus tantalum) 4.75 to 5.5% Titanium 0.65 to 1.15% Aluminum 0.2 to 0.8% Other trace elements 2.1% Iron—balance	Carbon 2.45% Chromium 5.25% Vanadium 9.75% Molybdenum 1.3% Iron—balance	Carbon 3.4% Chromium 5.25% Vanadium 14.5% Molybdenum 1.3% Iron—balance	Carbon 0.1% Chromium 1.2% Manganese 0.55% Molybdenum 0.12% Nickel 3.2% Silicon 0.28% Iron—balance
Elastic modulus (GPa)	200	221	235	206
Density (g/cm^3)	8.19	7.42	7.25	7.8
Thermal conductivity (W/m*K) at 21 °C	133	20.4	20	61
Coefficient of thermal expansion, (mm/mm*°C) (21 to 93 °C)	14.6×10^{-6}	10.7×10^{-6}	10.5×10^{-6}	11.6×10^{-6}

Two possible configurations (Ref. 7) that could meet the input-output ratio challenge were investigated. The first, as shown in Figure 1, is a three-stage compound planetary-geared transmission. In all three stages the ring gear is fixed to the housing. The first stage is a simple planetary. The second and third stages are compound planetaries. The input is the sun gear and the output is the carrier that contains the planet gears. In the second and third stages the planets have an input gear that meshes with the sun gear and an output gear that meshes with the ring gear. The compound planetary has a much higher reduction ratio than the simple planetary gear train. By carefully choosing the numbers of teeth on the sun, planets, and ring gears, a reduction ratio of 24:1 for one stage can be attained. The equation for the gear ratio for a compound planetary is the following:

$$\text{Ratio} = \frac{(N_1)(N_2)}{(N_3)(N_4)} + 1$$

where: N_1 – number of ring gear teeth
N_2 – number of planet teeth (meshes with the sun gear)
N_3 – number of teeth on the sun gear
N_4 – number of teeth on planet gear that meshes with ring gear

Tooth numbers chosen must be checked for planetary gear train assembly in all three stages. However, it appears possible to have the required reduction ratio (2500:1) in three stages. The number of planets used will depend on the assembly requirements. In the sketch shown in Figure 1, overlapping of the planet gears may be required to produce the load capacity needed at the final stage.

Another possible arrangement is shown in Figure 2. This device is known as a planetary differential drive. The input shaft drives a carrier that contains shafts with gears on each end. Sun gear, S1, is fixed to ground or attached to the housing. The two planet gears, P1 and P2, are rigidly connected on a common shaft that rotates in the carrier. Sun gear, S2, is rigidly connected to the output shaft. The second sun-planet set (S2, P2) has a different ratio forcing the output to rotate.

The gear ratio is found by the following equation:

$$\text{Ratio} = \frac{1}{\left(1 - \dfrac{N_{s1} N_{p2}}{N_{p1} N_{s2}}\right)}$$

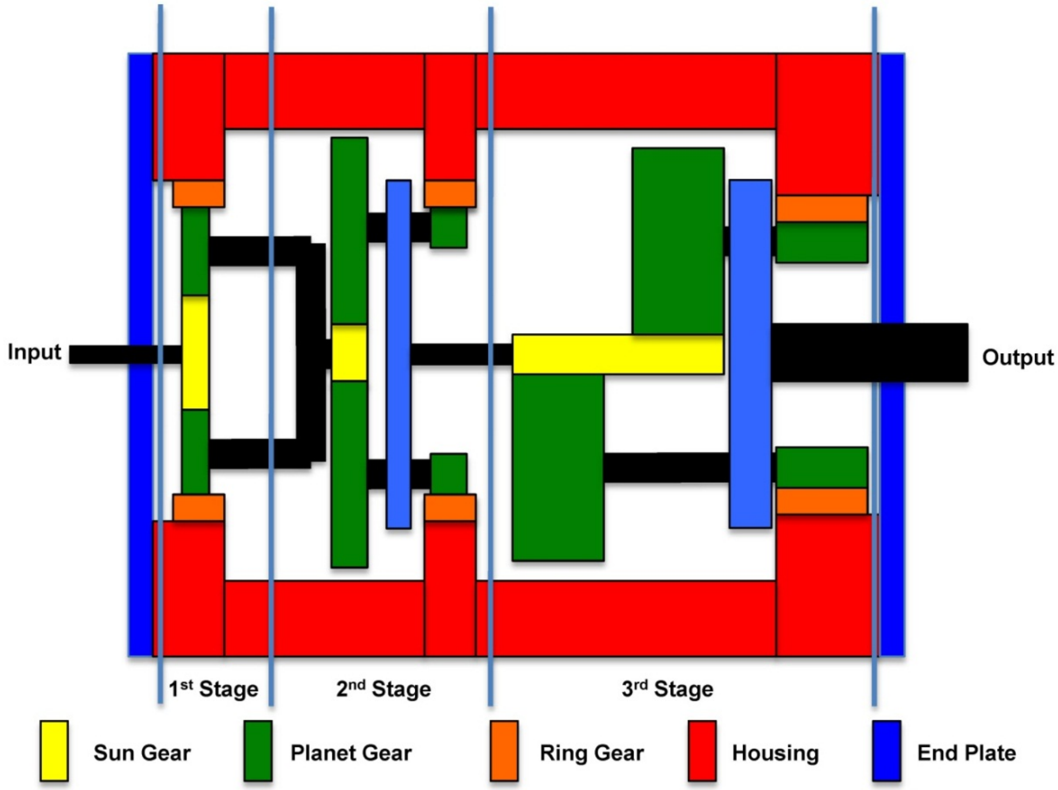


Figure 1.—Three-stage compound planetary mechanism.

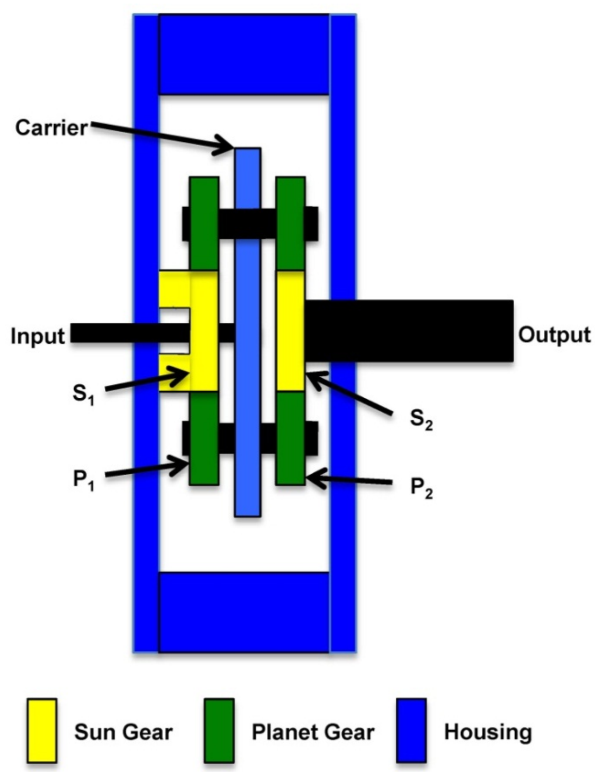


Figure 2.—Planetary differential drive.

As the term in the denominator approaches zero, or as the term $\left(\frac{N_{s1}N_{p2}}{N_{p1}N_{s2}}\right)$ approaches 1, the ratio becomes very large. As an example if N_{s1} = 102; N_{p1} = 50; N_{s2} = 100; and N_{p2}= 49, the ratio becomes 2500:1. This mechanism does pose some design issues, as the center distances of the two stages are required to be equal. Therefore, the module (or diametral pitch) of each gear mesh needs to be different for the tooth numbers shown in the example above.

As shown above, the first major requirement (2500:1 reduction ratio) can be achieved without using a harmonic drive. The main requirement remaining is to identify a gear steel capable of operating at 450 °C. The first consideration for using the nickel-based alloy for gear teeth is the tooth bending strength. A simulated specimen was developed for this purpose. A tensile test machine was used to conduct the tests. Tests were conducted at 24 and 490 °C.

Test Specimen, Test Fixture, Test Procedure

For several years, NASA Glenn Research Center has conducted single tooth bending tests using tensile test machines (Refs. 8 and 9). These machines are typically run at ambient temperatures with gears manufactured to the following specifications: 3.175 mm module (8 1/in pitch), 28 tooth, 6.35 mm face width, 20° pressure angle. This standard design has been used in all fatigue tests conducted at Glenn.

For the testing to be conducted in this project, a simulated gear tooth specimen was desired. To allow for a comparison with data gathered using actual gears, the test specimen was designed to have similar attributes. A finite element analysis result of the actual test gears meshing is shown in Figure 3 (Ref. 10). The maximum principal stress due to bending is shown as the red location in the tooth fillet. The testing to be done was to simulate the actual gears at the peak bending stress location due to the load at the highest point of single tooth contact, with the same contact and fillet geometry as the standard 28 tooth test gear used at Glenn for fatigue tests. The highest point of single tooth contact was determined (Ref. 11) along with the gear surface curvatures at this location.

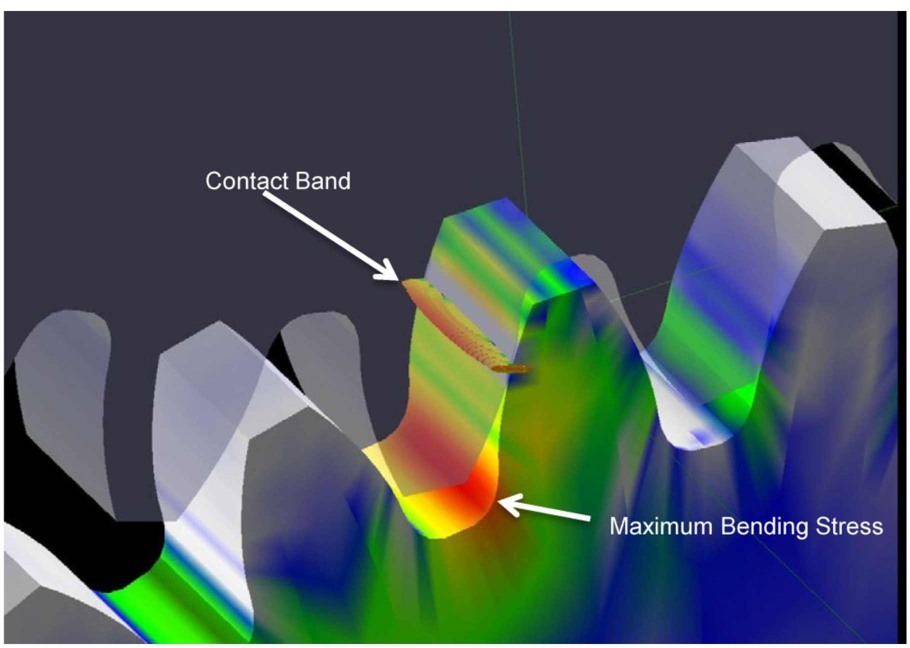


Figure 3.—Finite element analysis of gearing component to be simulated.

A simulated "gear tooth" was designed as a cantilevered beam, 6.35 mm. wide, with the same fillet radius as the NASA gear test specimen (0.84 mm.). The load was applied at a location along the cantilevered beam that was equivalent to the highest point of single tooth bending. This load location would be the point where the maximum bending stress would occur. The beam thickness was determined from the tooth thickness at the pitch point. The loading rod, that applied the normal load, had the same relative curvature as the actual contacting gears. The relative curvature was found via the following equation (Ref. 12):

$$R_{eq} = \frac{R_1 R_2}{(R_1 + R_2)}$$

where: R_{eq} – equivalent relative radius of curvature
R_1 – radius of curvature of gear 1 at the contact point
R_2 – radius of curvature of gear 2 at the contact point

The test specimen mounted in the tensile test machine is shown in Figure 4. The upper grip is rigidly mounted in the test machine and is locked to the machine frame. The upper grip locates the test specimen and provides a means for alignment of the load pin. The lower grip pushes vertically on the test specimen through the load pin.

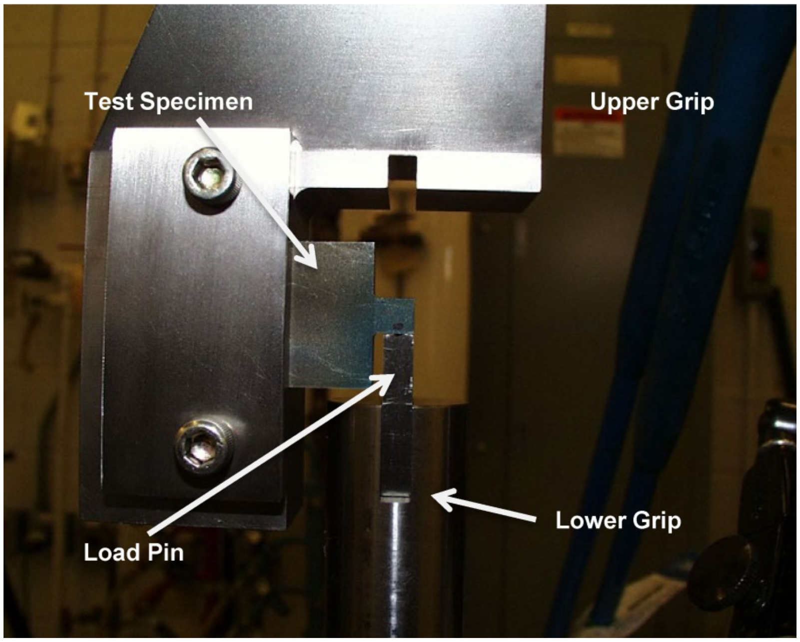


Figure 4.—Test setup showing main components.

Prior to conducting tests at the elevated temperature condition, a specimen with six thermocouples was installed in the test fixture. This specimen is shown in Figure 5.

The thermocouples were monitored to determine the temperature of the specimen in the tooth and fillet regions. In all high temperature tests, thermocouples were used to actively control the quartz lamp heating source. Figure 6 shows the test fixture with quartz lamps in operation.

Tests were conducted using load control procedure. In this procedure a small static load is applied, typically a small percentage of the full load, and then cycled sinusoidally to the maximum load and returned to the initial load. Tests were conducted at approximately 10 Hz. The test conditions (maximum applied load and rate) were maintained until the nominal deflection of the “test tooth” increased by about ~2 percent. A deflection of 2 percent is indicative of a crack in the fillet region that signifies the end of the test.

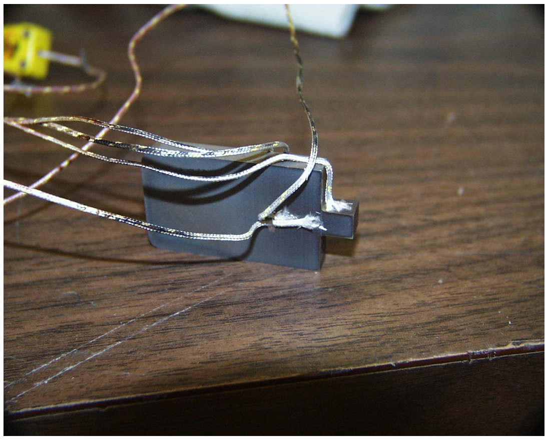


Figure 5.—Specimen used for quartz lamp temperature calibration.

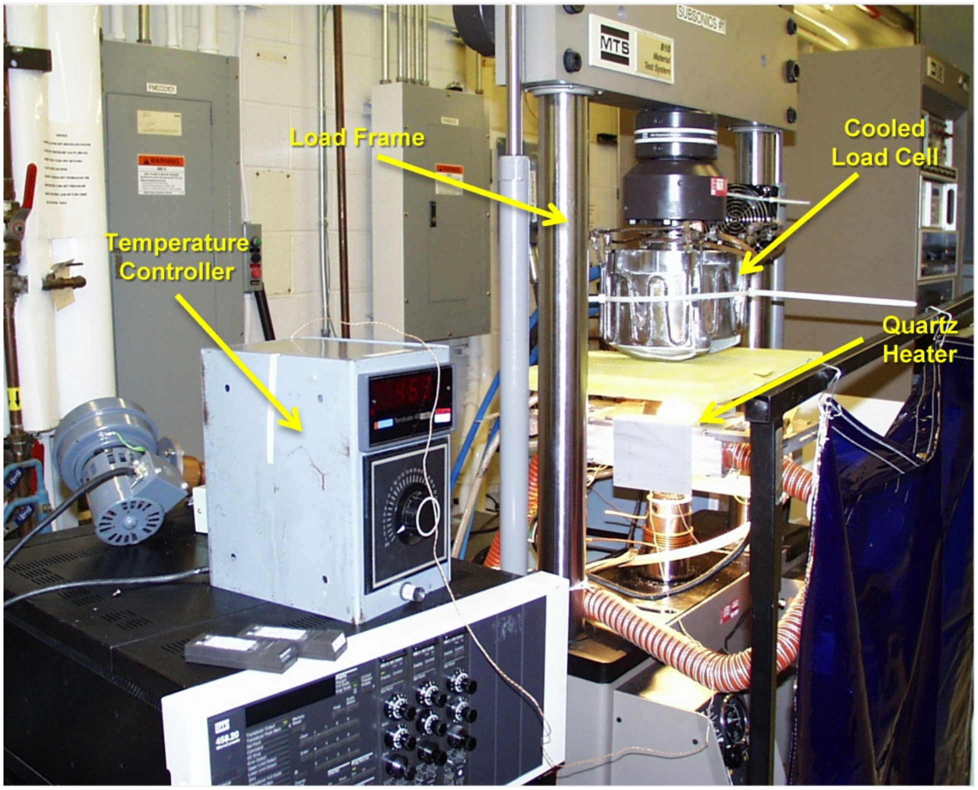


Figure 6.—Tensile test machine in operation during high temperature test.

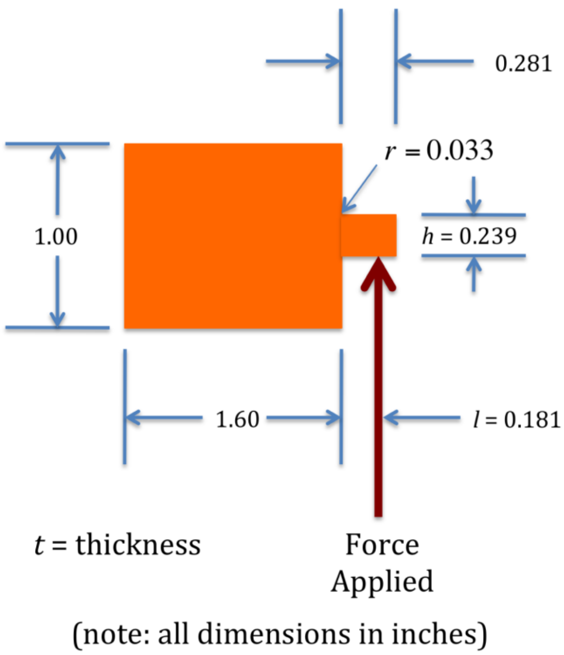
Test Results

Stress Analysis

To analyze the resultant stress on the "test tooth" two procedures were used. The first was to use a handbook calculation typically used for stress concentration effects from area changes subjected to a perpendicular and concentrated load (Ref. 12). The calculations were done for a 4448.2 N (1000 lb_f) load. For this level of load, located at the highest point of single tooth contact, resulted in the bending stress maximum in the fillet being 1.048 GPa (152.1 ksi). The calculation procedure is shown in Figure 7.

The same geometry was then put into a finite element analysis (Ref. 13) and statically loaded to 4448.2 N (1000 lbf). The finite element mesh is shown in Figure 8 and the results from the analysis are shown in Figure 9.

The resultant fillet bending stress from the finite element analysis was approximately 1.034 GPa (150 ksi) and agreed well with the method of Reference 14 as shown in Figure 7. The rest of the stress values displayed in this paper are based on the handbook calculation and are linearly dependent on the applied load.



From Lipson and Juvinal Reference 13:

$$\frac{r}{h} = \frac{0.033}{0.239} = 0.138$$

$$\frac{l}{h} = \frac{0.181}{0.239} = 0.757$$

from these ratios the stress concentration factor, K_T is:

$$K_T \approx 2.0$$

$$\sigma_{nom} = \frac{6Pl}{th^2} = \frac{6(0.181)P}{(.25)(.239)^2} = 76.05P$$

$$\sigma_{act} = K_T \sigma_{nom} = 152.1P$$

$$\therefore$$

for

$$P = 1000 lbf (4448.2N)$$

$$\sigma_{act} = 152.1 ksi (1.05 GPa)$$

Figure 7.—Calculated values for simulated gearing arrangement.

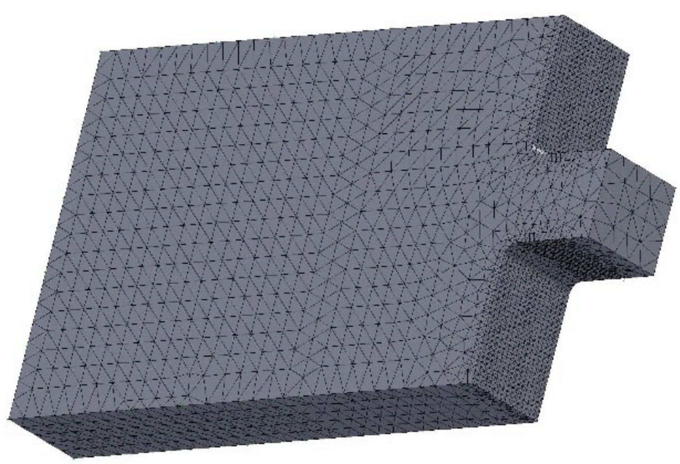


Figure 8.—Finite element model for bending stress simulation.

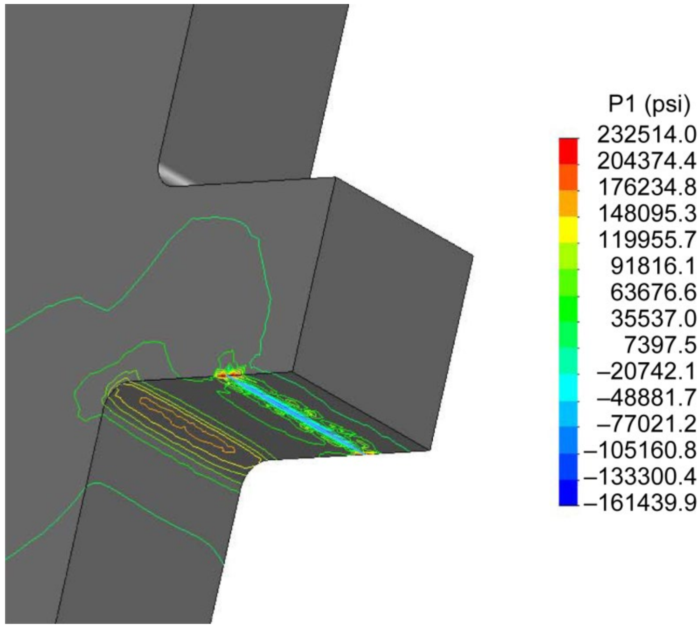


Figure 9.—Finite element stress result.

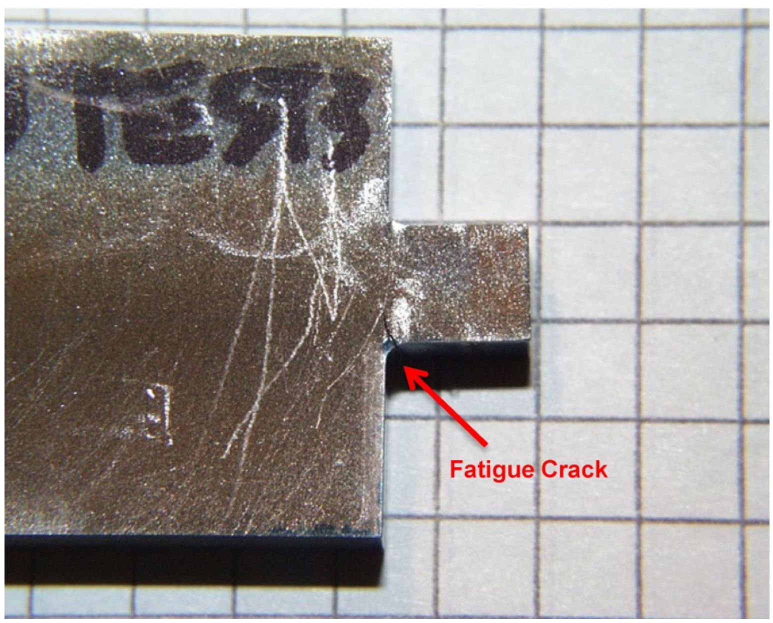


Figure 10.—Side view of a test specimen with crack, post-test.

Test Results

Photographs of the test specimen, post-test, are shown in Figure 10 and Figure 11. Due to the sensitivity of the tensile test machine being used, the test was stopped after the deflection changed by approximately 2 percent from the nominal condition. Therefore all tests were stopped when the crack length was approximately 30 percent of the "tooth" thickness. Figure 10 shows the entire specimen, post-test, with the crack starting at the fillet. This type of result is similar to that seen in other gear tooth bending fatigue tests conducted at Glenn. Figure 11 is a photograph showing the failed tooth with the applied load region (contact zone) and bending fatigue crack. Damage found on tested specimen that failed was similar.

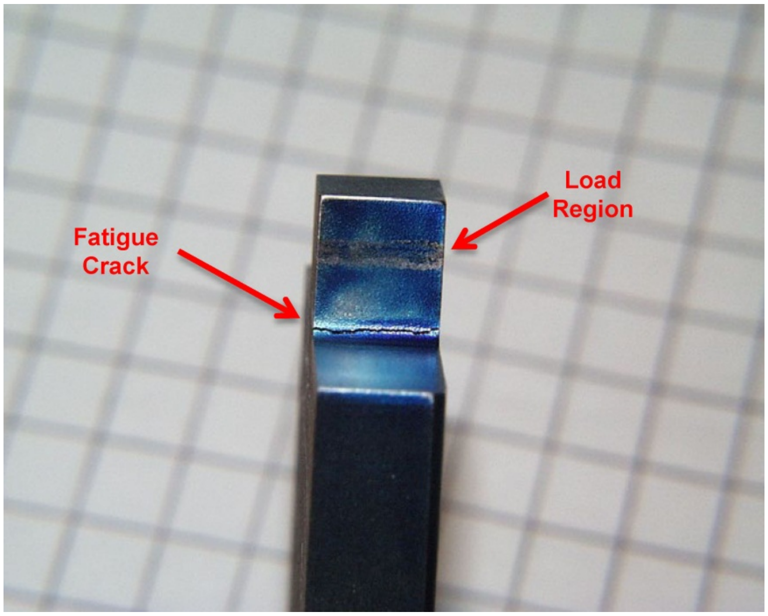


Figure 11.—Test specimen post-test indicating bending fatigue failure.

The test data accumulated during this study are shown in Figure 12 to Figure 14. The data plotted in these figures are the bending stress versus cycles to failure. As mentioned earlier, the number of tests at any one condition/material, was rather limited, but there are some general trends that can be discussed.

Three main levels of applied bending load were tested along with a few other tests at intermediate and higher levels of load. The three main loads applied were 6672 N (1500 lb_f), 4448 N (1000 lb_f) and 3225 N (725 lb_f). This corresponded to a bending stress of 1.67 GPa (228.2 ksi), 1.05 GPa (152 ksi) and 0.76 GPa (110.3 ksi), respectively, based on the stress concentration factor or finite element analysis results.

The data from tests run at room temperature (24 °C) are shown in Figure 12. The data clustered closer at each of the two higher loads. At the lowest load, two tests were suspended as they accumulated over 4 million cycles without failure. All four materials were within the usual scatter found in fatigue testing.

The tests shown in Figure 13 were run at 490 °C. The nickel-based alloy (Inco 718) performed better than the tool or aerospace gear steels at high load (stress). At the lowest load tested, the tool or aerospace materials did very well with these tests being suspended after 4 million cycles without failure.

A small number of tests were conducted at an intermediate and higher stress levels (Figure 12 (at ~ 190 ksi) and Figure 13 (at ~ 190 and 268 ksi)). These tests followed a similar trend as the data obtained at the three main load levels.

In Figure 14 data from all the tests are shown. Seven of the approximately 30 tests were suspended. The lowest load case produced all but one of the run-out test points.

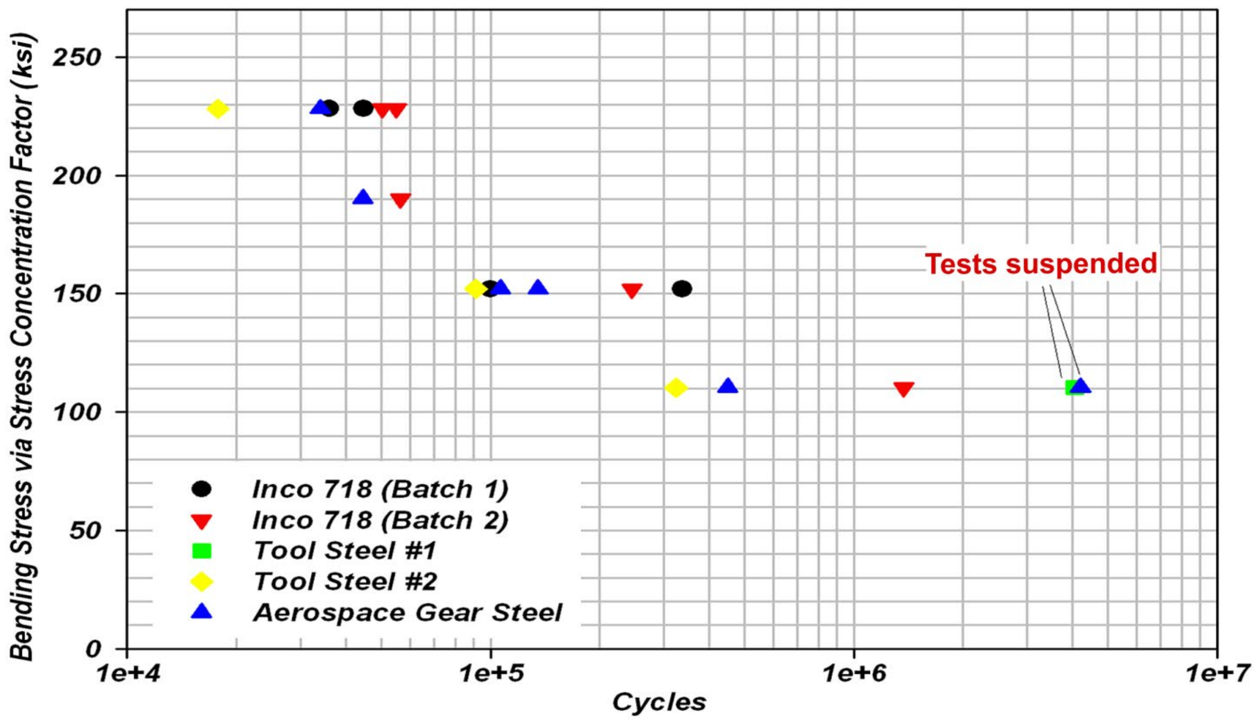


Figure 12.—Single tooth bending simulation results at 24 °C.

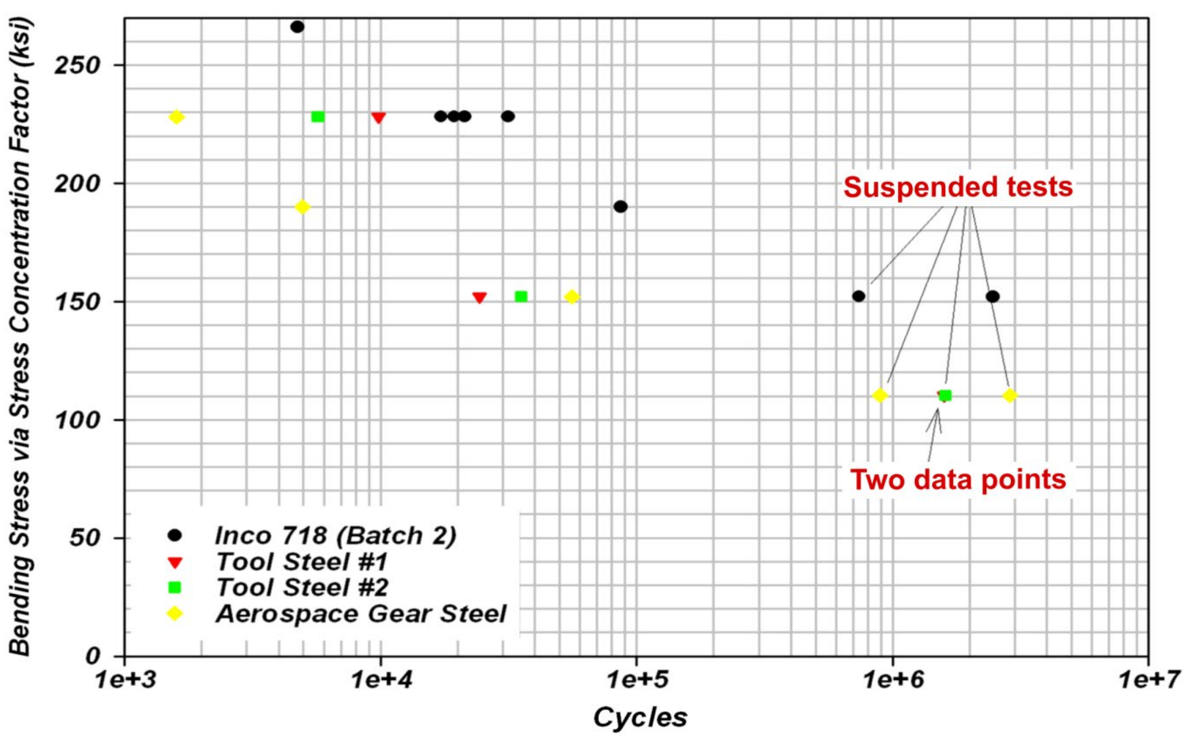


Figure 13.—Single tooth bending simulation results at 490 °C.

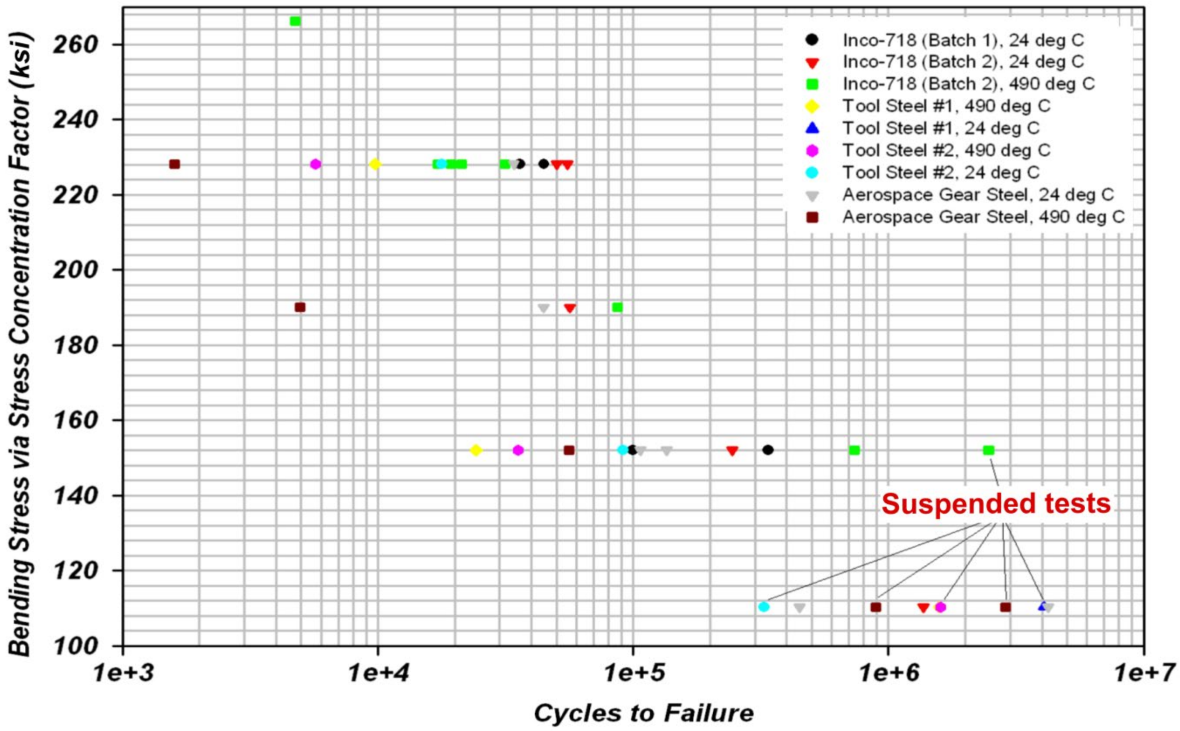


Figure 14.—All results at 24 and 490 °C.

Discussion of Results

The test results shown are preliminary in nature, and more data is needed at each condition, for each material to make the results statistically significant. However some trends in the data presented did occur. It appears that if the bending stress were limited to approximately 0.69 GPa (100 ksi), the mechanism to be developed would not experience a bending fatigue failure at the 490 °C temperature condition. This information is extremely important in designing this limited-life component, as the highest cycled component would have less than 1 million stress cycles during the expected 1 hr of operation. It appears if the high temperature condition was the only problem constraint and that the bending stress was limited to 0.69 GPa (100 ksi), any of these materials would satisfy the design requirements.

If there is a corrosive environmental requirement, the nickel-based material might be better suited. If wear of the surfaces becomes an issue, then possibly one of the tool steels would be better for this application.

Another item not known at this time is the effect of post-processing that would normally be performed on any high performance gear. The specimens used in this study were electro-discharge machined and then tested. While it is expected that the EDM process would put some residual stress in the material at the surface where it is needed, the exact amount is unknown. Also, there are many other processes that could have been utilized to improve the bending fatigue, such as heat treatment or shot peening.

Conclusions

Four materials were tested in a simulated single tooth bending fatigue test. A nickel-based alloy, two tool steels, and an aerospace gear material were tested. Tests were conducted at 24 and at 490 °C. A limited number of tests were conducted at any one condition or with any one material. Based on the results of this study the following conclusions can be drawn:

1. All four materials performed similarly for the room temperature (24 °C) tests at the four load levels tested. Also, the fatigue cycles at the highest stress level were more closely concentrated.
2. At 490 °C, the nickel-based alloy appeared to perform better (larger number of cycles to failure) at the higher bending stress conditions (1.67 GPa (228.2 ksi) and 1.05 GPa (152 ksi)).
3. At the lowest load condition, 0.76 GPa (110.3 ksi), nearly all tests resulted in suspended tests at either test temperature.
4. The effect of temperature change from 24 to 490 °C resulted in roughly an order of magnitude decrease in the fatigue life of the samples that failed.

References

1. Inconel Alloy 718, Publication Number SMC-045, Special Metals Corporation, September 2007.
2. Crucible CPM10V, Crucible Materials Corporation, Syracuse, NY.
3. Crucible CPM15V, Crucible Materials Corporation, Syracuse, NY.
4. Aerospace Structural Metals Handbook, Vol. I, Ferrous Alloys, AFML-TR-68-115, Syracuse University Press, 1973.
5. Shafer, I., and Slatter, R.: Precision Pointing and Actuation Systems for UAV’s Using Harmonic Drive Gears, Harmonic Drive AG, Limburg, Germany.
6. Holman, J.P.: Heat Transfer, Fourth Edition, McGraw-Hill Book Company, 1976.
7. Wilson, C. and Michels, W.: Mechanisms: Design Oriented Kinematics, American Technical Society, 1969.
8. Handschuh, R.; Krantz, T.; Lerch, B.; Burke, C.: Investigation of Low-Cycle Bending Fatigue of AISI 9310 Steel Spur Gears, NASA/TM—2007-214914, ARL-TR-4100, July 2007.
9. Krantz, T. and Tufts, B.: Pitting and Bending Fatigue Evaluations of a New Case-Carburized Gear Steel, NASA/TM—2007-215009, ARL-TR-4123, December 2007.
10. Vijayakar, S. and Abad, S.: Helical 3-D User’s Manual, Advanced Numerical Solutions, Hilliard, OH, February 2005.
11. Anderson, N. and Lowenthal, S.: Spur Gear System Efficiency at Part and Full Load, NASA TP–1622, February 1980.
12. Boresi, A., Sidebottom, O., Seely, F., and Smith, J.: Advanced Mechanics of Materials, John Wiley & Sons, 3rd Edition, 1978.
13. Solidworks Simulation 2009 x64 SP0.0. (2008). Dassault Systèmes. Concord, MA.
14. Lipson, C. and Juvinall, R.: Handbook of Stress and Strength, Design and Material Applications, 1st Edition, Macmillion Co., New York, 1963.

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14. ABSTRACT
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