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National Aeronautics and Space Administration
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Conservation and Renewable Energy
Office of Vehicle and Energy R&D

Prepared for
Conference on Deformation, Fracture, Wear, and
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New Orleans, Louisiana, November 23-24, 1981

NOTICE

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FAILURE ANALYSIS OF A TOOL STEEL TORQUE SHAFT

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ABSTRACT

A low design load drive shaft from an experimental diesel truck engine failed unexpectedly during highway testing. The shaft was driven by a turbine used to deliver power from an experimental exhaust heat recovery system to the engine's crankshaft. During design, fatigue was not considered a major problem because of the low operating cyclic stresses. An independent testing laboratory analyzed the failure by routine metallography. The structure of the hardened S-7 tool steel shaft was banded and the laboratory attributed the failure to fatigue induced by a banded microstructure. NASA was asked to confirm this analysis. Visual examination of the failed shaft plus the knowledge of the torsional load that it carried pointed to a 100 percent ductile failure with no evidence of fatigue. Scanning electron microscopy confirmed this. Torsional test specimens were produced from pieces of the failed shaft and torsional overload testing produced identical failures to that which had occurred in the truck engine. This pointed to a failure caused by a high overload and although the microstructure was defective it was not the cause of the failure.

INTRODUCTION

Any failure should be treated as a forensic study which encompasses all possible scenarios with a milti-disciplined analysis. In the case presented the original analysis was conducted by an independent laboratory using metallographic analysis only, fractorgraphy and structural analysis were not investigated.

Metallography examines metals, usually microscopically, to reveal its structure, hardness, and defects. It is a key tool when properly applied in failure studies.

Fractography is the discipline of examining a broken surface and deducing from the macroscopic and microscopic appearance of the structure what type of stress/strain environment existed and whether this environment produced a ductile or brittle failure.

Structural analysis will show when the loads needed to produce failures are, in fact, present. In this case structural analysis would consist of examining fatigue design values and torsional strength.

BACKGROUND

The failed shaft was turbine driven by an experimental 40 horsepower exhaust heat recovery system that delivered the power directly to the crankshaft of an experimental diesel truck. The truck was being highway tested at speeds below 50 mph. The exact time of the failure was unknown. Although the system had operated 20 000 miles it is conceivable that the shaft had failed earlier. The shaft was designed for infinite life.

Physically the shaft was 14" long and 1/2" in diameter with 3/4" splines on each end. The shaft had failed at the smallest diameter just as shaft began its transition to the 3/4 diameter spline. The shaft was manufactured from (S-7) tool steel and heat treated to RC 56-58. Figure 1

VISUAL OBSERVATIONS

1. The shaft was received in one long piece and several smaller pieces from which metallographic specimens had already been taken.
2. Although 80 percent of the fracture surface was destroyed due to rubbing at the time of failure there was strong evidence that the failure had been completely ductile in nature. The fracture surface was totally perpendicular to the shaft. There was no evidence that smaller pieces had broken away.
3. There was no evidence of fatigue "beach" marks.
4. There was no observable permanent twist in the shaft.
5. The independent laboratory report accompanying the shaft stated that the S-7 material failed due to a banded microstructure inducing fatigue.

TEST PROCEDURE

1. The shaft was Zyglo inspected for evidence of cracking.
   - There was no evidence of any flaw being present outside the actual fracture.
   - There was a secondary crack on the fracture surface.
2. A portion of the fracture was placed in the S.E.M. for examination.
   - All areas surveyed showed micro-void coalescence which is indicative of ductile failure. Figure 2.
3. A chemical analysis of the shaft was conducted.
   - The results showed the material to be S-7 tool steel. There was a rather high residual nickel content (0.22 percent), which is not regarded as significant to the failure.
4. Hardness tests were conducted.
   - The shaft hardness was RC 56-57.

5. Sections of the shaft both transverse and axial were mounted for microstructure and Tukon hardness. The transverse specimen also included the fracture zone.
   - The microstructure indicated a heavily banded structure. Tukon hardness indicated a possible 3 point difference in the bands (RC 52-55). There was no evidence of carburization or decarburization. Figure 3.
6. A section of the shaft was macro etch to reveal ingot quality.
   - The etched diameter did not show excessive inclusions or porosity.
7. Torsional strength of the actual shaft was measured.
   - Three specimens 2" long were cut from the shaft. A 7/16" hex was ground 1/2" long on each end taking care not to overheat the shaft. A center section approximately 1/2" long was cut to the diameter shown in Table 1. The specimen was mounted in a holder which prevented a bending moment from being applied to the test section while restraining one of the hex ends. Torque was applied to the other hex with an adjustable 400 foot pound torque wrench. Torque application was slowly incremented until failure occurred. Table 1 details the Test data.

DISCUSSION

The maximum design conditions for this shaft was 40 hp at 3000 rpm. From standard design formulas this can be converted to torque by:

$$\frac{\text{Torque}}{(\text{in-lb})} = \frac{63\ 000\ \text{hp}}{\text{rpm}}$$

The maximum design torque was, therefore, 70 ft-lb.

To find the maximum shear stress the shaft was subjected to a standard formula for elastic torsional stress. The formula being:

$$\text{Maximum Shear Stress} = \frac{16 \text{ Torque (in-lb)}}{\pi \text{ Diameter}^3}$$

which shows the stress at the design torque of 70 ft-lb to be 35 000 psi.

The values shown in Table 1 were also calculated from the above formula assuming no plastic deformation. (This assumption is particularly valid in this case because the high Rockwell hardness places the yield close to the ultimate strength.) The minimum shear strength far exceeds the design requirements.

The number two specimen failure occurred at the hex (probably due to bending of the torque wrench) and traveled in a spiral through the specimen. The diameter had been increased to better approximate the actual design conditions.

The failure of the number one and number three specimens were indistinguishable from the actual failure surface. The surfaces were compared under the Scanning Electron Microscope (S.E.M.) and found to be predominately microvoid coalescence structures identical to the actual failure.

Further assuming that a fatiguing action had occurred on the original failure and taking 200 000 psi as the actual shear strength of the shaft the final failure would have occurred through only a 9/32" diameter. The failed shaft showed the failure had taken place through a minimum of 5/16" and almost certainly through the entire 1/2" diameter. (A 1/32 difference in shaft diameters is significant.)

The metallurgical structure of the number one and two specimens examined at the failure sight was shown to be heavily banded. The number three specimen was not examined metallographically. Figure 4.

The banded microstructure could not be related to either the original failure or the test speciment failures.

CONCLUSIONS

The torsional strength of this material exceeded the design criteria by a factor of greater than five.

Looking at the design conditions of 40 hp at 3000 rpm and comparing them in standard shafting formulas, it would seem that the shaft has adequate strength for all but the most severe loading conditions.

Although the banding of the microstructure is undesirable there is no evidence to support that it caused or even contributed significantly to this failure.

No evidence of fatigue was noted anywhere in the NASA analysis.

The cause of this failure was over stressing of the shaft well beyond its design limits.

TABLE 1. - SPECIMEN DATA AND RESULTS

[See discussion for explanations.]

Specimen number	Test section diameter, in.	Minimum torque at failure, ft-lb	Shear stress at failure, psi	Type of failure	Location of failure
1	0.378	180	200 000	Ductile	Mid test section
2	0.401	245	234 000	Brittle	Hex
3	0.375	205	237 000	Ductile	Mid test section

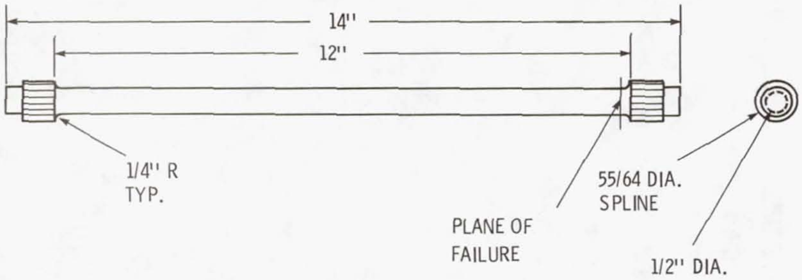

Figure 1. - Configuration of torque shaft showing the location of the failure.

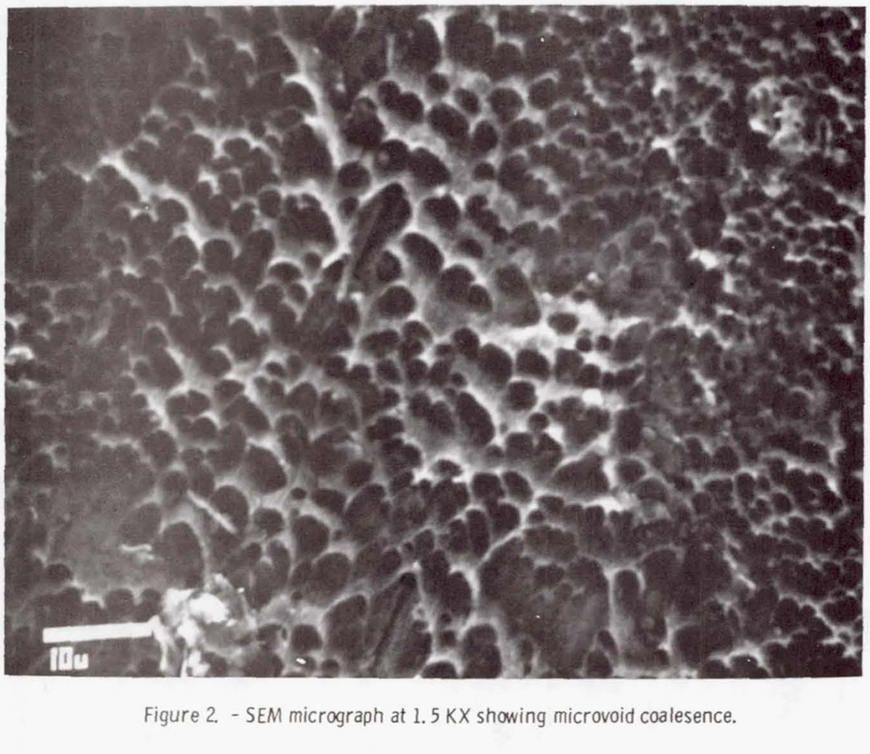

Figure 2. - SEM micrograph at 1.5 KX showing microvoid coalesence.

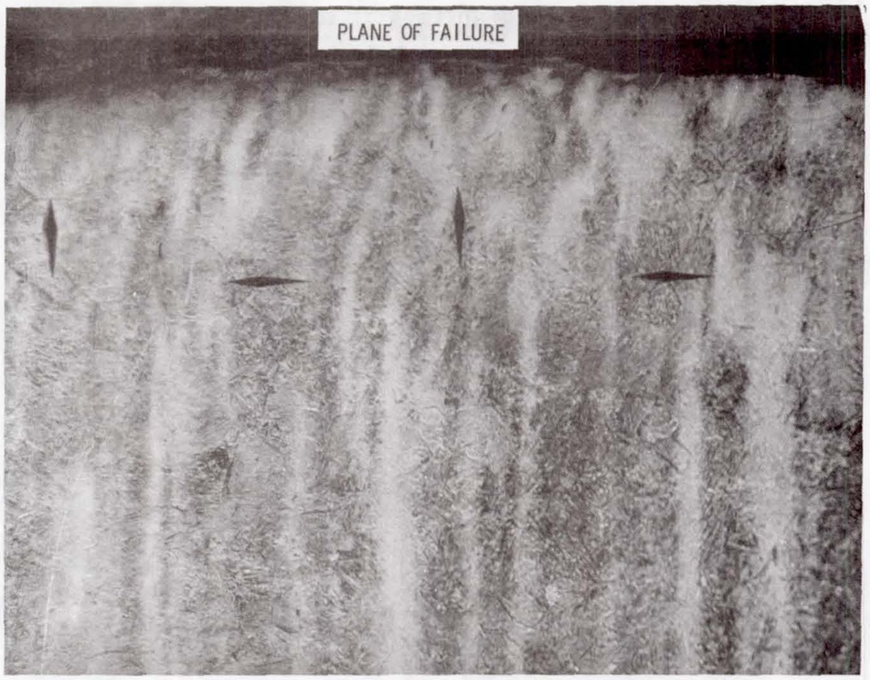

Figure 3. - Banded microstructure at 100X of failed shaft showing failure surface and tukon hardness indentations.

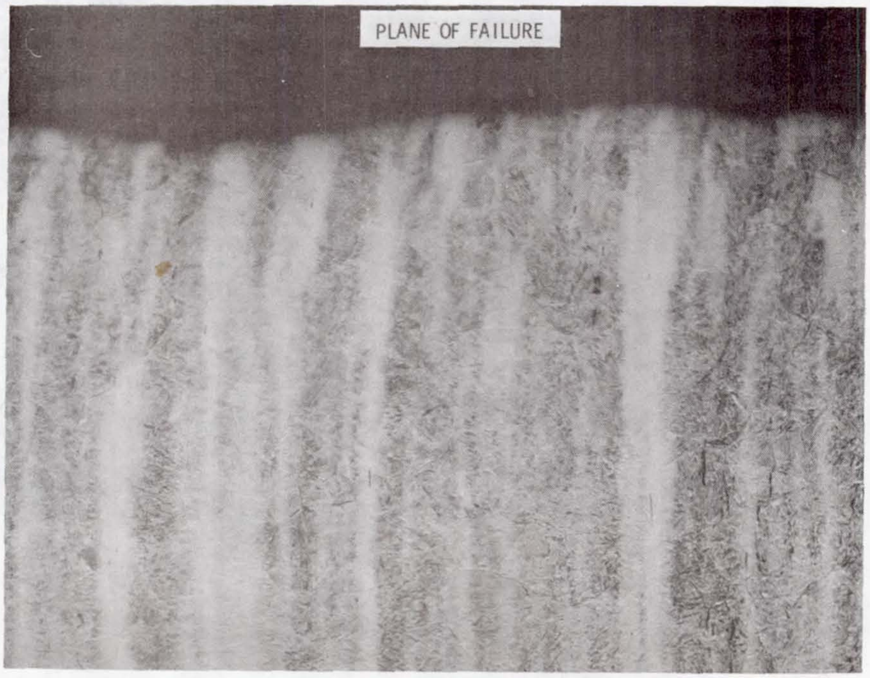

Figure 4. - Typical banded structure at 100X from No. 1 and 2 test specimen.

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