

Effect of Shot Peening on Surface Fatigue Life of Carburized and Hardened AISI 9310 Spur Gears

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Dennis P. Townsend and Erwin V. Zaretsky
Lewis Research Center Cleveland, Ohio

Summary

Gear surface fatigue endurance tests were conducted on two groups of 10 gears each of carburized and hardened AISI 9310 spur gears manufactured from the same heat of material. Both groups were manufactured with standard ground tooth surfaces. The second group was subjected to an additional shot-peening process on the gear tooth surfaces and root radius to produce a residual surface compressive stress. The gear pitch diameter was 8.89 cm (3.5 in.). Test conditions were a gear temperature of 350 K (170° F), a maximum Hertz stress of 1.71×10^9 N/m² (248 000 psi), and a speed of 10 000 rpm.

The shot-peened gears exhibited pitting fatigue lives 1.6 times the life of the standard gears without shot peening. Residual stress measurements and analysis indicate that the longer fatigue life is the result of the higher compressive stress produced by the shot peening. The life for the shot-peened gear was calculated to be 1.5 times that for the plain gear by using the measured residual stress difference for the standard and shotpeened gears. The measured residual stress for the shotpeened gears was much higher than that for the standard gears.

Introduction

Shot peening has long been used as a method for improving the bending strength of gear teeth (refs. 1 to 3). However, shot peening has not been considered as a means of extending the surface fatigue life of gears. In essence, shot peening induces a residual compressive stress below the surface of the gear tooth. Studies of residual stresses in rolling-element bearings have shown that increased residual compressive stress will increase rolling-element (surface) fatigue life (refs. 4 and 5). There is always a need to improve the surface fatigue life of aircraft gears, especially in helicopter and V/STOL aircraft.

The objectives of the research reported herein were (1) to investigate the effects of shot peening of gear teeth on the surface fatigue life of standard ground, case-carburized, and hardened AISI 9310 spur gears, (2) to compare the life of shot-peened gears to that of non-shotpeened gears manufactured with the same material and specifications, and (3) to determine the residual stress produced by shot peening and its effect on the surface fatigue life.

To accomplish these objectives, 20 spur gears were manufactured from a consumable-electrode-vacuum-melted single heat of AISI 9310 material. Ten of these gears were shot peened after finish grinding. The gear pitch diameter was 8.89 cm (3.5 in.). Both the

shot-peened and non-shot-peened gears were then tested to fatigue by surface pitting under identical test conditions. These test conditions included a gear temperature of 350 K (170° F), a maximum Hertz stress of 1.71×10^9 N/m² (248 000 psi), and a speed of 10 000 rpm.

Apparatus, Specimens, and Procedure

Gear Test Apparatus

The gear fatigue tests were performed in the NASA Lewis Research Center's gear test apparatus (fig. 1). This test rig uses the four-square principle of applying the test gear load so that the input drive only needs to overcome the frictional losses in the system.

A schematic of the test rig is shown in figure 1(b). Oil pressure and leakage flow are supplied to the load vanes through a shaft seal. As the oil pressure is increased on the load vanes inside the slave gear, torque is applied to the shaft. This torque is transmitted through the test gears back to the slave gear, where an equal but opposite torque is maintained by the oil pressure. This torque on the test gears, which depends on the hydraulic pressure applied to the load vanes, loads the gear teeth to the desired stress level. The two identical test gears can be started under no load, and the load can be applied gradually, without changing the running track on the gear teeth.

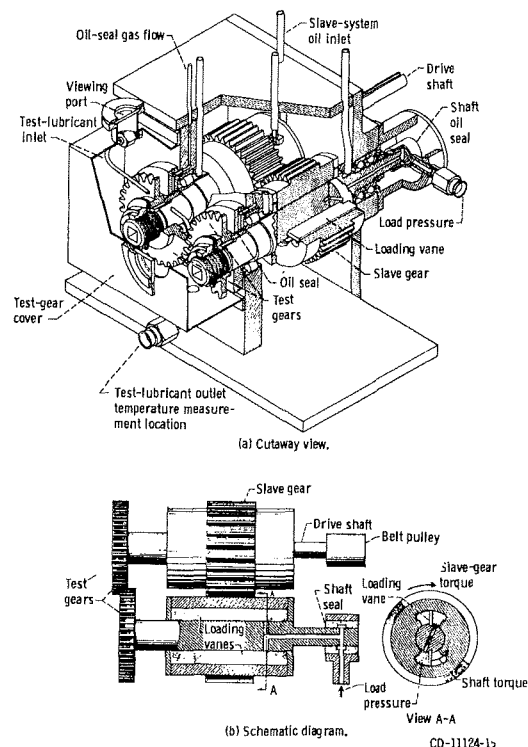


Figure 1. NASA Lewis Research Center's gear fatigue test apparatus.

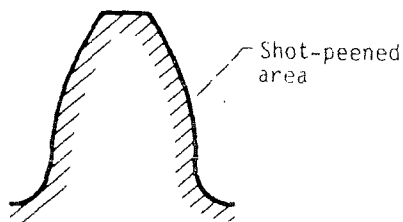
Separate lubrication systems are provided for the test gears and the main gearbox. The two lubrication systems are separated at the gearbox shafts by pressurized labyrinth seals. Nitrogen is the seal gas. The test gear lubricant filtered through a 5- μ m-nominal fiber-glass filter. The test lubricant can be heated electrically with an immersion heater. The temperature of the heater skin is controlled to prevent overheating the test lubricant. A vibration transducer mounted on the gear is used to automatically shut off the test rig when a gear surface fatigue occurs. The gearbox is also automatically shut off if there is a loss of oil flow to either the main gearbox or the test gears, if the test gear oil overheats, or if there is a loss of seal gas pressurization.

The belt-driven test rig can be operated at several fixed speeds by changing pulleys. The operating speed for the tests reported herein was 10 000 rpm.

Test Materials

The test gears were manufactured from consumable-electrode-vacuum-melted (CVM) AISI 9310 steel from the same heat of material. Both sets of gears were case hardened to a case hardness of Rockwell C 58 and a case depth of 0.97 mm (0.038 in.). The nominal core hardness was Rockwell C 40. One set of the gears was shot peened, after finish grinding, on the tooth root and the tooth profile according to the specifications given in table I. The chemical composition of the material is given in table II. Both sets of gears were

Table I. Shot Peening Specification



Specification	MIL-S-131658 BPS FW 4409
Shot size	070
Shot type	Cast steel
Intensity (height of Almen strip, type A), mm (in.)	0.18 to 0.23 (0.007 to 0.009)
Coverage (sides and root only), percent	200

Table II
Nominal Chemical Composition of CVM AISI 9310 Gear Material

Element	Composition, wt%
C	0.10
Mn	.63
Si	.27
Ni	3.22
Cr	1.21
Mo	.12
Cu	.13
P	.005
S	.005

Table III
Heat Treatment for AISI 9310

Step	Process	Temperature		Time, hr
		K	°F	
1	Preheat in air	----	----	-----
2	Carburize	1172	1650	8
3	Air cool to room temperature	----	----	-----
4	Copper plate all over	----	----	-----
5	Reheat	922	1200	2.5
6	Air cool to room temperature	----	----	-----
7	Austenitize	1117	1550	2.5
8	Oil quench	----	----	-----
9	Subzero cool	180	-120	3.5
10	Double temper	450	350	2 each
11	Finish grind	----	----	-----
12	Stress relieve	450	350	2

case carburized and heat treated in accordance with the heat treatment schedule of table III. Figure 2 is a photomicrograph of an etched and polished gear tooth surface showing the case microstructure of the AISI 9310 material.

Test Gears

Dimensions of the test gears are given in table IV. All gears have a nominal surface finish on the tooth face of 0.406 μ m (16 μ in.) rms and a standard 20° involute profile with tip relief. Tip relief was 0.0013 cm (0.0005 in.), starting at the highest point of single-tooth contact. Surface traces of the standard gear and the shot-peened gear are shown in figure 3.

Test Lubricant

All the gears were lubricated with a single batch of synthetic paraffinic oil. The physical properties of this lubricant are summarized in table V. Five percent of an extreme-pressure additive, designated Lubrizol 5002 (partial chemical analysis given in table V), was added to the lubricant.

Test Procedure

After the test gears were cleaned to remove the preservative, they were assembled on the test rig. The 0.635-cm (0.25-in.) wide test gears were run in an offset condition with a 0.30-cm (0.12-in.) tooth-surface overlap to give a load surface on the gear face of 0.28 cm (0.11 in.), thereby allowing for the edge radius of the gear teeth. If both faces of the gears were tested, four fatigue tests could be run for each set of gears. All tests were run in at a pitch-line load of 1225 N/cm (700 lb/in) for 1 hour, which gave a maximum Hertz stress of 0.756×10^9 N/m² (111 000 psi). The load was then increased to 5784 N/cm (3305 lb/in), which gave a pitch-line maximum Hertz stress of 1.71×10^9 N/m² (248 000 psi). At this pitch-line load the tooth root bending stress would be 0.21×10^9 N/m² (30 000 psi) if plain bending were assumed. However, because there was an offset load, an additional stress was imposed on the tooth bending stress. Combining the bending and torsional moments gave a maximum stress of 0.26×10^9 N/m² (37 000 psi).

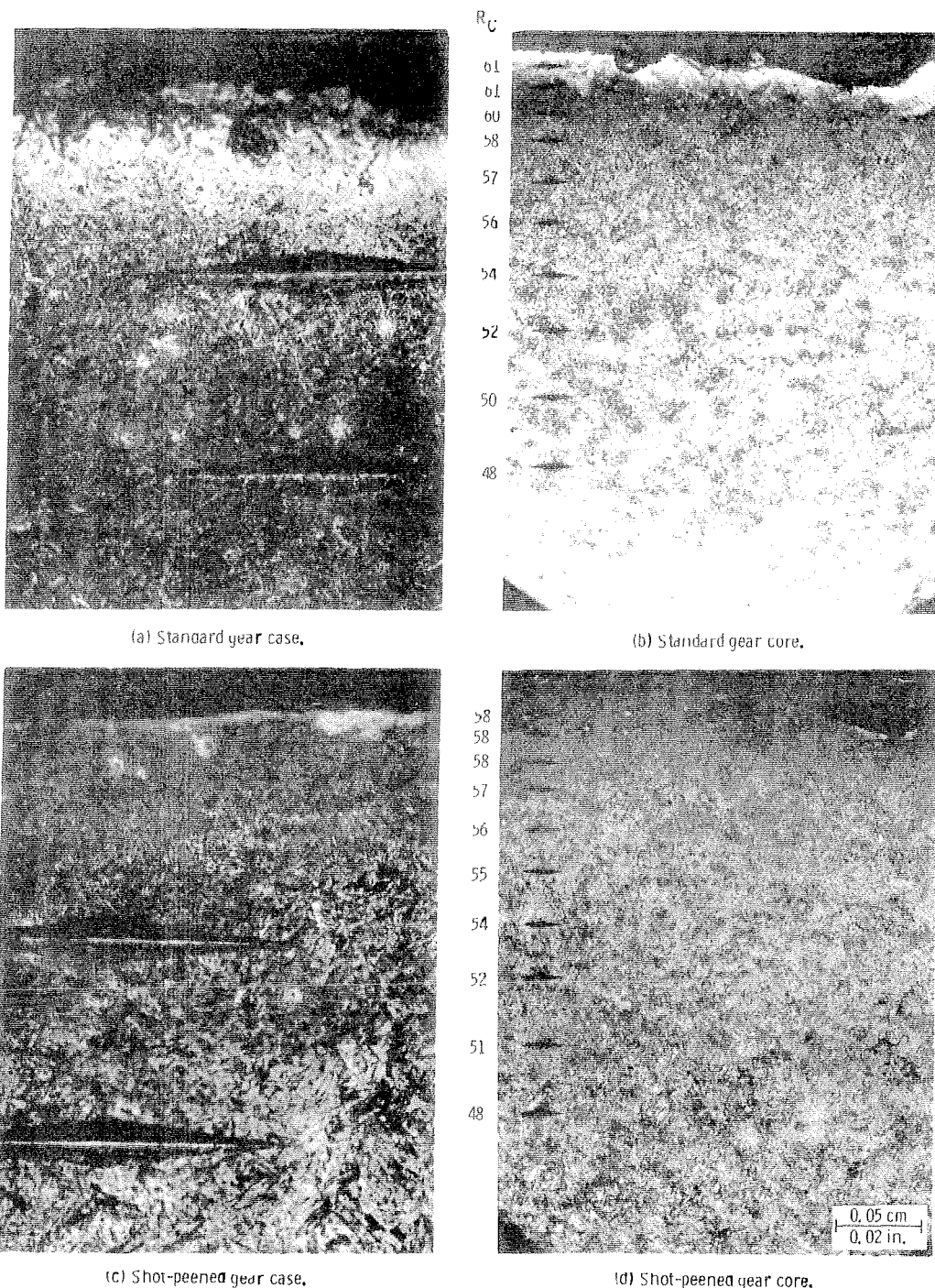


Figure 2. Photomicrographs of case and core for standard and shot-peened spur gears.

This bending stress does not include the effects of tip relief, which would also increase the bending stress.

Operating the test gears at 10 000 rpm gave a pitch-line velocity of 46.55 m/sec (9163 ft/min). Lubricant was supplied to the inlet mesh at 800 cm³/min at 319 ± 6 K (116° ± 10° F). The lubricant outlet temperature was nearly constant at 350 ± 3 K (170° ± 5° F). The tests ran continuously (24 hr/day) until they were automatically shut down by the vibration detection transducer, located on the gearbox adjacent to the test gears. The lubricant circulated through a 5- μ m fiberglass filter to remove wear particles. After each test the lubricant and the filter element were discarded. Inlet and outlet oil temperatures were continuously recorded on a strip-chart recorder.

The pitch-line elastohydrodynamic (EHD) film thickness was calculated by the method of reference 6. It was assumed, for this

film thickness calculation, that the gear temperature at the pitch line was equal to the outlet oil temperature and that the inlet oil temperature to the contact zone was equal to the gear temperature, even though the inlet oil temperature was considerably lower. It is possible that the gear surface temperature was even higher than the outlet oil temperature, especially at the end points of sliding contact. The EHD film thickness for these conditions was computed to be 0.33 μ m (13 μ in.), which gave an initial ratio of film thickness to composite surface roughness h/σ of 0.55 at the 1.71x10⁹-N/m² (248 000-psi) pitch-line maximum Hertz stress.

Results and Discussion

Gears manufactured from CVM AISI 9310 material were tested in pairs until failure or for 500 hours. One-half of the gears were

shot peened on the tooth root and profile. Nineteen tests were run with standard-finish ground test gears, and 24 tests were run with standard finish ground gears that had been shot peened. Test results were analyzed by considering the life of each pair of gears as a system.

Surface (pitting) fatigue results for the standard-finish AISI 9310 gears are shown in figure 4(a). These data were analyzed by the method of reference 7. The 10- and 50-percent fatigue lives were 18.8×10^6 and 46.1×10^6 stress cycles (31.3 and 76.8 hr), respectively. These results are summarized in table VI. The failure index (i.e., the number of fatigue failures out of the number of sets tested) was 18 out of 18. A typical fatigue spall is shown in figure 5(b). A cross section of a typical fatigue spall is shown in figure 5(a). The surface pitting failure occurs slightly below the pitch line in the area of highest Hertz stress and is of subsurface origin.

Pitting fatigue life results for the gears that were shot peened are shown in figure 4(b). The failure index was 24 out of 24. A typical fatigue spall for the shot-peened gears is shown in figure 6(a). A cross section of a typical fatigue spall for the shot-peened gears is shown in figure 6(b). The 10- and 50-percent surface pitting fatigue lives were 30.1×10^6 and 67.5×10^6 stress cycles (50.3 and 112.6 hr), respectively. These results are summarized in table VI. The shot-peened gears exhibited a 10-percent fatigue life 1.6 times that of the standard ground AISI 9310 gears. The confidence number for the difference in life was 83 percent. The mean life ratio for the shot-peened over the standard AISI 9310 gears was 1.5, with a confidence number of 98 percent. The confidence number indicates the percentage of time the relative lives of the material will occur in the same order. The 90-percent confidence bands for the standard ground and shot-peened AISI 9310 test gears are shown in figure 4. A confidence number of 95 percent is equivalent to a 2σ confidence level.

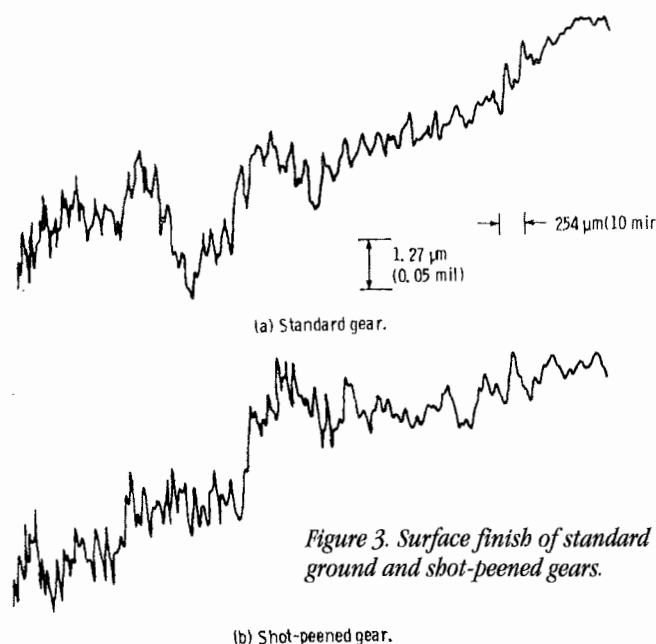
It is well known that shot peening produces residual subsurface stresses in steel in addition to the residual stresses produced by case carburizing, hardening, and grinding. It was theorized that the additional residual stresses induced by shot peening should account for the increased life of the shot-peened gears. Therefore two shot-peened and untested gear teeth and two standard ground and untested gear teeth were subjected to X-ray diffraction residual stress measurements to determine the magnitude of these residual stresses. Residual stress measurements were made near the pitch point at the surface and at nominal subsurface depths of 5, 13, 25, 76, 127, and 254 μm (0.2, 0.5, 1.0, 3, 5, and 10 mil).

Material was removed for subsurface measurement by electropolishing in a sulphuric-phosphoric-chromic acid electrolyte in order to minimize possible alteration of the subsurface residual stress distribution as a result of material removal. All data obtained as a function of depth were corrected for the effects of the penetration of the radiation employed for residual stress measurement into the sub-surface stress gradient and for stress relaxation, which occurred as a result of material removal. The method used for the X-ray stress measurements and the calibration procedures used are described in references 8 and 9.

Figure 7(a) shows two corrected X-ray diffraction residual stress measurements as a function of depth below the surface for the standard ground AISI 9310 gear teeth that had not been shot

Table IV. Spur Gear Data. Gear Tolerance per ASMA class, 12.

Number of teeth	28
Diametral pitch	8
Circular pitch, cm (in.)	0.9975 (0.3927)
Whole depth, cm (in.)	0.762 (0.300)
Addendum, cm (in.)	0.318 (0.125)
Chordal tooth thickness	0.485 (0.191)
(reference), cm (in.)	
Pressure angle, deg	20
Pitch diameter, cm (in.)	8.890 (3.500)
Outside diameter, cm (in.)	9.525 (3.750)
Root fillet, cm (in.)	0.102 to 0.152
	(0.04 to 0.06)
Measurement over pins,	9.603 to 9.630
cm (in.)	(3.7807 to 3.7915)
Pin diameter, cm (in.)	0.549 (0.216)
Backlash reference, cm (in.)	0.0254 (0.010)
Tip relief, cm (in.)	0.001 to 0.0015
	(0.0004 to 0.0006)
Tooth width, cm (in.)	0.635 (0.25)



peened or tested. The high compressive stress on the surface of the gear tooth is the result of grinding and has a very shallow depth that has very little effect on the surface durability of the gear. The lower compressive stress, which has much greater depth, is from the case carburizing and hardening of the gear tooth surface. This compressive residual stress has a definite beneficial effect on surface fatigue and bending fatigue life.

Figure 7(b) contains two plots of corrected X-ray diffraction residual stress measurements as a function of depth below the surface in the ground and shot-peened AISI 9310 gear teeth that had not been tested. The high grinding compressive stress on the surface was reduced. A hook in the curve shows a high compressive stress 1.3 μm (0.5 mil) below the surface as a result of the shot peening. The compressive stress at greater depths below the surface was also increased as a result of the shot peening. It is the increased compressive stress at the greater depths that has the major effect on the surface fatigue life. The depth to the maximum shear stress for the

load conditions reported herein was 178 μm (7 mil).

Figure 7(c) contain plots of the average of the two X-ray residual stress measurements as a function of depth below the surface for both the standard ground and shot-peened gears. This figure shows the average increase in the residual compressive stress due to shot peening. At the maximum shear stress depth of 178 μm (7 mil) the average residual compressive stress was increased from $0.186 \times 10^9 \text{ N/m}^2$ (27 000 psi) in the standard ground AISI 9310 gear to $0.26 \times 10^9 \text{ N/m}^2$ (37 700 psi) in the ground and shot-peened AISI 9310 gear. From equation (A11), taken from the analysis given in the appendix for maximum shear stress and residual stress,

$$(\tau_{\max})_r = -3.15 \times 10^6 \left(\frac{P_N}{L S_{\max} R} \right) - \frac{1}{2} S_{ry}$$

where

$$R = 7.62 \text{ mm (0.3 in.)}$$

$$S_{\max} = 1.71 \times 10^9 \text{ N/m}^2 \text{ (248 000 psi)}$$

$$\frac{P_N}{L} = 578 \text{ 375 N/m (3305 lb/in)}$$

Therefore for peened gears, in SI units,

$$(\tau_{\max})_r = -21.74 \times 10^9 \frac{578 \text{ 375}}{0.00762 \times 1.71 \times 10^9} - \frac{1}{2} (0.26 \times 10^9) = -0.835 \times 10^9$$

in U.S. customary units,

$$(\tau_{\max})_r = -3.15 \times 10^6 \frac{3305}{0.3 \times 248 \text{ 000}} - \frac{1}{2} (-37 \text{ 000}) = -121 \text{ 080}$$

And for standard gears, in SI units,

$$(\tau_{\max})_r = -21.74 \times 10^9 \frac{578 \text{ 375}}{0.00762 \times 1.71 \times 10^9} - \frac{1}{2} (0.186 \times 10^9) = -0.872 \times 10^9$$

in U.S. customary units,

$$(\tau_{\max})_r = -3.15 \times 10^6 \frac{3305}{0.3 \times 248 \text{ 000}} - \frac{1}{2} (-27 \text{ 000}) = -126 \text{ 430}$$

Table V.
Properties of Synthetic Paraffinic Oil

Additive	^a Lubrizol 5002
Kinematic viscosity, cm^2/sec (cS) at-	
244 K (-20° F)	2500×10^{-2} (2500)
311 K (100° F)	31.6×10^{-2} (31.6)
372 K (210° F)	5.7×10^{-2} (5.7)
477 K (400° F)	2.0×10^{-2} (2.0)
Flashpoint, K (°F)	508 (455)
Fire point, K (°F)	533 (500)
Pour point, K (°F)	219 (-65)
Specific gravity	0.8285
Vapor pressure at 311 K (100° F),	0.1 mm Hg (or torr)
Specific heat at 311 K (100° F),	676 (0.523) J/kg K (Btu/lb °F)

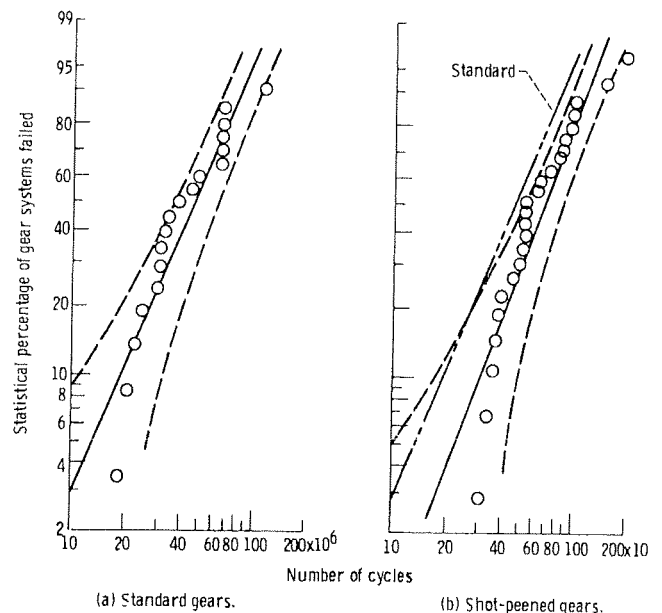


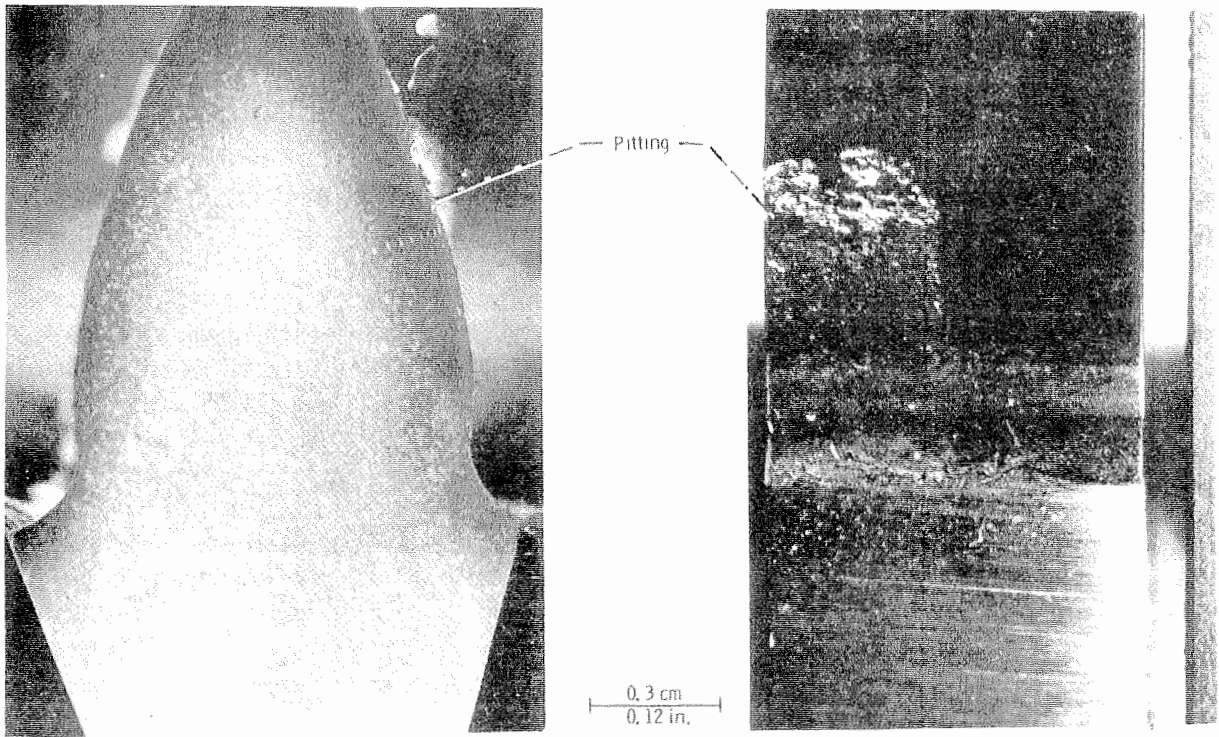
Figure 4. Comparison of surface (pitting) fatigue lives of standard ground and shot-peened carburized and hardened CVM AISI 9310 steel spur gears. Speed, 10 000 rpm; lubricant, synthetic paraffinic oil; gear temperature, 350 K (170°F); maximum Hertz stress, $1.7 \times 10^9 \text{ N/m}^2$ (248 000 psi).

Table VI.
Fatigue results with AISI 9310 Standard and Shot-Peener Test Gears

AND SHOT-PEENED TEST GEARS

Gears	10-Percent life, cycles	50-Percent life, cycles	Slope	Failure index ^a	Confidence number, ^b percent
Standard	19×10^6	46×10^6	2.1	18/18	--
Shot peened	30	68	2.3	24/24	83

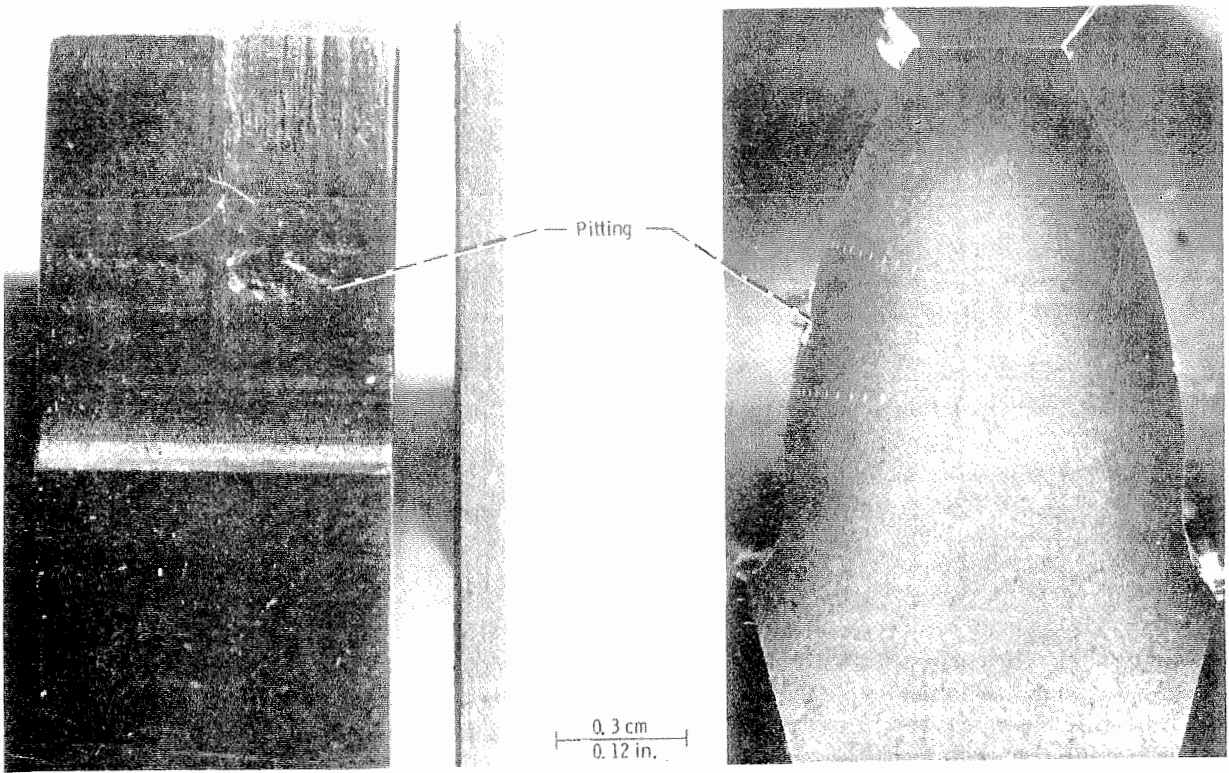
^aIndicates numbers of failures out of total number of tests.
^bprobability, expressed as a percentage, that the 10-percent life with the baseline AISI 9310 gear is either less than, or greater than, that of the particular lot of gears being considered.



(a) Cross section of typical fatigue spall.

(b) Typical fatigue spall.

Figure 5. Fatigue spall for standard ground gear.



(a) Typical fatigue spall.

(b) Cross section of typical fatigue spall.

Figure 6. Fatigue spall for shot peened gear.

The surface fatigue life from reference 10 for gears is inversely proportional to the maximum shear stress to the ninth power. The calculated life ratio from measured residual stress is therefore

$$\frac{L_{10p}}{L_{10s}} = \left[\frac{(\tau_{max})_{rs}}{(\tau_{max})_{rp}} \right]^9 = \left(\frac{0.872 \times 10^9}{0.835 \times 10^9} \right)^9 = 1.5$$

This calculated ratio of the fatigue life of the shot-peened gears to that of the standard gear compares favorably with the experimental fatigue life ratio of 1.6.

Summary of Results

Gear surface fatigue endurance tests were conducted on two groups of carburized and hardened AISI 9310 steel spur gears manufactured from the same heat of material. Both groups were manufactured with a standard ground tooth surface. One group was subjected to an additional shot-peening process on the gear tooth surface and root radius to produce a residual compressive surface stress. The gear pitch diameter was 8.89 cm (3.5 in.). Test conditions were a gear temperature of 350 K (170° F), a maximum Hertz stress of 1.71×10^9 N/m² (248 000 psi), and a speed of 10 000 rpm. The lubricant was a synthetic paraffinic oil with an additive package.

The following results were obtained:

1. The 10-percent surface (pitting) fatigue life of the shot-peened gears was 1.6 times that for the standard test gears that were not shot peened. This was shown to be statistically significant.
2. The calculated 10-percent surface (pitting) fatigue life for the shot-peened gears determined from measured residual subsurface stress was 1.5 times that for the standard gears that were not shot peened.
3. Measured residual stresses for the standard and shot-peened gears show an increase of 40 percent for the maximum shear stress in addition to a 350 percent increase at a depth of 13.0 μm (0.5 mil).

Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio, March 5, 1982

Appendix - Derivation of Residual Stress Effect on Maximum Shear Stress

It is well known that classical rolling-element fatigue begins in the subsurface zone of maximum shear stress (refs. 11 and 12). Therefore, to determine the effect of residual stress on rolling-element fatigue in gears, it is necessary to analyze the effect of residual stress on the maximum shear stress below the surface. The maximum shear stress at any point in a stressed volume below a rolling line-contact load is

$$(\tau_{max})_y = \frac{1}{2} (S_z - S_y) \quad (A1)$$

$$(\tau_{max})_x = \frac{1}{2} (S_z - S_x) \quad (A2)$$

where S_z is the principal compressive stress in a direction normal to the contact area, S_y is the principal compressive stress parallel to the direction of rolling (ref. 13), and S_x is the principal stress normal to the direction of rolling.

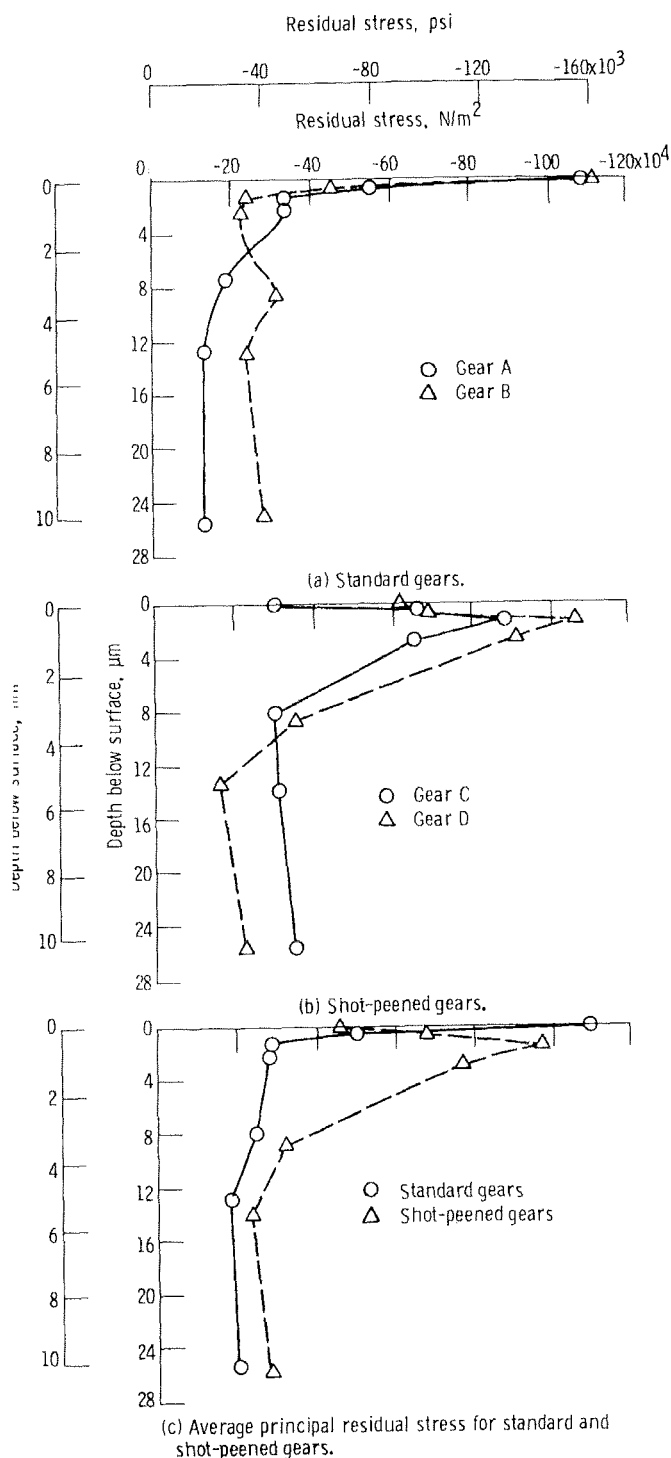


Figure 7. Principal residual stress as a function of depth below the surface of carburized, hardened, ground and untested AISI 9310 steel spur gear teeth.

For rollers or gear teeth loaded statically, the maximum theoretical shear stress occurs in the y - z plane since the stress in the y , or rolling, direction is less than the stress in the x direction. Therefore the maximum shear stress is

$$\tau_{\max} = \frac{1}{2} (S_z - S_y) \quad (\text{A3})$$

If the residual stresses are equal in the x and y directions, for the line contact in the y - z plane the maximum shear stress including the residual stress is

$$(\tau_{\max})_r = \frac{1}{2} [S_z - (S_y + S_{ry})] \quad (\text{A4})$$

$$(\tau_{\max})_r = t_{\max} - \frac{1}{2} S_{ry}$$

where S_{ry} is the residual stress in the y direction and is positive for tensile stress and negative for compressive stress. From reference 13 for line contact of rollers

$$\tau_{\max} = -0.30025 \frac{b}{\Lambda} \quad (\text{A5})$$

where b is the half width of the Hertzian contact,

$$b = \frac{2P_N}{\pi L S_{\max}} \quad (\text{A6})$$

and Λ for a contact of two rollers of the same material is

$$\Lambda = 4 \frac{R_1 R_2}{R_1 + R_2} \left(\frac{1 - \delta^2}{E} \right) \quad (\text{A7})$$

If the rollers are of the same radius,

$$\Lambda = 2R \left(\frac{1 - \delta^2}{E} \right) \quad (\text{A8})$$

where

- P_N normal load, N (lb)
- S_{\max} maximum Hertz stress, N/m² (psi)
- R_1, R_2 radius of curvature of the two rollers, m (in.)
- δ Poisson's ratio
- E Young's modulus, N/m² (psi)

Substituting equations (A6) and (A8) into equation (A5) for τ_{\max} results in

$$\tau_{\max} = -0.30025 \frac{P_N}{\pi L S_{\max}} \left[\frac{E}{R(1 - \delta^2)} \right] \quad (\text{A9})$$

If equation (A9) is substituted into equation (A4),

$$(\tau_{\max})_r = -0.30025 \frac{P_N}{\pi L S_{\max}} \left[\frac{E}{R(1 - \delta^2)} \right] - \frac{1}{2} S_{ry} \quad (\text{A10})$$

For steel gears $E = 207 \times 10^9$ N/m² (30×10^6 psi) and $\delta = 0.30$; therefore equation (A10) becomes for SI units

$$(\tau_{\max})_r = -21.74 \times 10^9 \frac{P_N}{L S_{\max} R} - \frac{1}{2} S_{ry} \quad (\text{A11a})$$

and for U.S. customary units

$$(\tau_{\max})_r = -3.15 \times 10^6 \frac{P_N}{L S_{\max} R} - \frac{1}{2} S_{ry} \quad (\text{A11b})$$

where S_{ry} can be either compressive or tensile. When gears are shot peened, the residual stress is compressive and therefore reduces the maximum shear stress. Since the rolling-element fatigue life of gears is inversely proportional to the maximum shear stress to the ninth power (ref. 14),

$$\left. \begin{aligned} L &\sim \left(\frac{1}{\tau_{\max}} \right)^9 \\ \text{or} & \\ L &\sim \left[\frac{1}{(\tau_{\max})_r} \right]^9 \end{aligned} \right\} \quad (\text{A12})$$

From equation (A4), where $(\tau_{\max})_r = \tau_{\max} - 1/2 (S_{ry})$,

$$L \sim \left[\frac{1}{\tau_{\max} - 1/2 (S_{ry})} \right]^9 \quad (\text{A13})$$

using a life ratio of L_1 and L_2

$$\frac{L_1}{L_2} = \left[\frac{\tau_{\max} - 1/2 (S_{ry})_2}{\tau_{\max} - 1/2 (S_{ry})_1} \right]^9 \quad (\text{A14})$$

When the residual stress developed by the shot peening the gear teeth is known, the change in life produced shot peening can be determined from equation (A14).

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