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Evaluation of a Vibration Diagnostic System for the Detection of Spiral Bevel Gear Pitting Failures

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EVALUATION OF A VIBRATION DIAGNOSTIC SYSTEM FOR THE DETECTION OF SPIRAL BEVEL GEAR PITTING FAILURES

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ABSTRACT

A vibration diagnostic system was used to detect spiral bevel gear surface pitting fatigue in a closed loop spiral bevel gear fatigue test rig. The diagnostic system consists of a personal computer with an analog to digital conversion board, a diagnostic system unit and software. The diagnostic system performs time synchronous averaging of the vibration signal and produces a vibration image of each tooth on any gear in a transmission. Several parameters were analyzed including gear stresswave and raw baseband vibration, kurtosis, peak ratios of several parameters etc. The system provides limits for the various parameters and gives a warning when the limits are exceeded. Three spiral bevel gear tests were monitored with this system. The results show that the system is effective at detecting spiral bevel gear tooth surface pitting damage and other anomalies in the test rig.

INTRODUCTION

Aircraft gearbox failure can be a serious problem on aircraft such as turboprop, geared fans and helicopters. There are many times when transmissions are overhauled at a given service interval even when no failures have occurred simply because pre-set operating times to overhaul have elapsed. These transmissions may have considerable life remaining and could have been operated for a longer period of time providing a good failure detection system were available. Several methods of failure detection for aircraft gearboxes have been in use for several years with limited success. Chip detection will sometimes give an indication of failure provided there are enough chips that make it through the system to the chip detector. Spectrometric Oil Analysis Program (SOAP) has been used for failure detection for several years (Reynolds, 1967) with limited success. Unfortunately SOAP will not give any indication of failure when a five micron filtering system is used that is typical of most aircraft lubrication systems operating today.

Several methods using spectrum analysis of vibration or noise signals have been utilized for failure detection for several years with limited success. The spectrum analysis methods require an evaluation of the various gear meshing harmonics and sideband frequencies, to find a particular type of failure.

Recent vibration based failure detection methods have been developed using time synchronous averaging of the gear mesh vibration signal. (Zakrajsek, 1989; Stewart, 1977; Zakrajsek, et al., 1993; and Zakrajsek, et al., 1994). The vibration signal of a particular gear is found by time averaging the raw vibration signal using a timing signal synchronized to the rotation of the gear being analyzed. With this method the vibration which is noncoincident with the rotation of the gear will average out to a very low residual value leaving only the vibration from the gear being analyzed. This time averaged vibration signal usually depicts the meshing pulse for each tooth on the gear over one complete revolution.

Several parameters are calculated from the time-averaged signal, to find failure conditions such as fatigue spalls and cracks. In addition, the shape and amplitude of each gear tooth pulse is evaluated to determine whether a change has occurred over a given time period.

The object of the research reported herein is to determine the performance of the recently developed vibration based detection methods. These methods were applied to vibration data obtained from a spiral bevel gear fatigue test rig at NASA Lewis Research Center, where three sets of test gears were run until a fatigue failure occurred. The failure modes of test gears used in this program ranged from moderate pitting on two teeth in one test to spalling over several teeth in another test.

SPIRAL BEVEL GEAR TEST FACILITY

The spiral bevel gear test rig used for the testing conducted herein is one of two such facilities at NASA Lewis Research Center. A sketch of the bevel gear test rig is shown in Fig. 1 and a cross-section of the test rig showing the main features is shown in Fig. 2. The bevel gear test rig is a closed-loop, torque-regenerative system with a closed loop capacity of 560 kW (750 hp). Power for the rig is provided by a 75 kW (100 hp) variable speed drive motor. The torque is controlled in the closed loop by moving a helical gear with a hydraulically actuated thrust piston. A torque meter monitors the test torque and speed. The drive motor only needs to supply the power necessary to overcome the gear, bearing and drive V-belt losses. A lubricating system supplies the lubricant to the slave and test section of the bevel gear test rig. The lubricant temperature is controlled by an immersion heater and a water cooled heat exchanger. The test gear lubricant is filtered by a 5 μm depth type filter which removes wear particles from the lubricant.

As shown in Fig. 2 two sets of spiral bevel gears are tested simultaneously. The slave (right) side operates in a speed increaser mode, where the gear drives the pinion. The test (left) side operates in the intended fashion as a speed reducer. The gear ratio for the test rig is three to one with 12 teeth on the pinion and 36 teeth on the gear. A photograph of the test gears is shown in Fig. 3. The test hardware basic design parameters are given in Table I.

VIBRATION DATA

Accelerometer vibration measurements were taken on the bevel gearbox to find the best vibration accelerometer location, because vibration modes of the gearbox case can affect the gear vibration signal. The best location was found to be on the pinion bearing housing adjacent to the test gear. Figure 4 is a spectrum of the test gearbox showing the various vibration frequencies. The once per revolution of the pinion and the first and second harmonic of gear mesh frequency are clearly seen. The peak at approximately 1.5 times the pinion mesh frequency is the mesh frequency of the helical gears which have 55 teeth each.

GEAR DIAGNOSTIC SYSTEM

The diagnostic system used in this research program is a commercial gear diagnostic system. The system consists of a personal computer with an analog to digital board installed, a signal conditioning system, and a software package. A schematic of the system is shown in Fig. 5. The signal conditioning system includes a computer controlled tracking filter, computer controlled analog signal conditioning, a pulse rate multiplier/divider, and interfacing electronics. The software automatically sets and monitors the analog signal conditioning circuitry, performs the analysis and provides monitor displays of individual gear tooth vibration images. Gear condition status and alarms are issued when appropriate. The software can display current and past data in a variety of different formats. A flow chart of the system is shown in Fig. 6.

High pass and low pass filters are sometimes used in front of the system to remove the vibration frequencies above and below a frequency range of interest.

The gear diagnostic system uses a number of techniques to extract the condition of the gears from the raw vibration signal. The raw signal is time synchronous averaged over the meshing cycle of each tooth on the gear being monitored. A proximity probe was used to detect the passage of each gear tooth, with the resulting pulse used for the averaging process. Time synchronous averaging is used to remove the parts of the vibration signal not coincident with the meshing cycle of the gear being monitored.

A time averaged signal was produced for two areas of interest, the low frequency band, or baseband vibration, and the high frequency band. The baseband vibration in this application is the portion of the signal up to about 10 kHz, which includes the primary meshing frequency (2.28 kHz) and its second, third and fourth harmonics. The baseband vibration was demodulated about the meshing frequency, and the amplitude modulation function, or envelope, was then constructed. The envelope of the baseband vibration depicts the vibration pattern of each tooth on the gear, using the low frequency components of the signal. Figure 7 shows examples of gear baseband and pinion stresswave plots. The stresswave plots are generated from a high frequency band. This high frequency band was demodulated about a harmonic of the meshing frequency, then the amplitude modulation function was constructed. The envelope of the high frequency band is also known in this system as the gear stresswave. The stresswave depicts the vibration pattern of each tooth on the gear, using the high frequency components of the vibration signal. In many cases the resolution of the tooth meshing pattern is considerably better in the stresswave, than the baseband envelope. This is due to the decreased effect of the vibration transfer path dynamics at higher frequencies.

A number of signal processing techniques were applied to the baseband envelope and stresswave to determine the condition of the gears. Each of these methods, designated M1 to M11, are designed to detect specific defects in the meshing pattern, corresponding to various gear tooth faults. An example of this is the M9 method, the normalized kurtosis of the baseband envelope or stresswave. The normalized kurtosis is defined as the fourth statistical moment of the signal about the mean of the signal, divided by the fourth power of the standard deviation. The normalized kurtosis detects single peaks in the vibration pattern caused by cracks or pitting on one or two teeth on a gear. It is a nondimensional parameter with a value of 3 for a signal with a normal gaussian distribution (i.e., a gear in good condition). A normalized kurtosis value of approximately 6 and higher signifies a possible cracked or pitted tooth on the gear being monitored.

TEST PROCEDURE

BEVEL GEAR TESTS

The test facility, after break-in, was operated at a constant gear speed of 3800 rpm, loop torque of 938 Nm (8300 lb-in.) which gives a Hertz stress of 2247 MPa (326 ksi) and a bending stress of 345 MPa (50 ksi). The oil-inlet temperature was 357 K (183 °F), and oil lubricating jet pressure was 552 MPa (80 psi) which gave a flow of 1.2 l/min (0.32 gpm) for each test gear. The closed loop power was 371 kW (497 hp). Test run A was operated at slightly different conditions of 4100 rpm and gear torque of 802 Nm (7100 lb-in). A single lubrication jet was used to lubricate each of the spiral bevel gear meshes. The single jet sprays lubricant on the gear at 90° before of the mesh position. The lubricant for the bearings is kept from splashing on the test components through the use of shields and drains. The lubricant used for this test was DOD-L-85734 with properties as shown in Table II. This lubricant is typically used in helicopter gearbox systems as well as the gas turbine engines. A summary of facility operating conditions, maintained during the testing, is given in Table III.

The tests were run continuously (24 hr/day) until a failure occurred. The pinions were operated at 11 400 rpm, except test A, and lubricant was supplied to the gear at 357 K (183 °F). The lubricant fling off temperature was nearly constant at 366 K (200 °F).

The gear vibration data was taken from an accelerometer mounted on the gearbox pinion shaft bearing housing, adjacent to the gear mesh. The accelerometer was mounted as close as possible to the bearing near the gear tooth mesh. A once per revolution signal and multiple pulse count which would normally be several counts per tooth, were used by the monitoring system. For these tests the multiple pulse was 36 or one per gear tooth. These were used along with the gear box vibration signal, by the gear diagnostic system to determine the various output parameters. Data samples were taken approximately every 5 min for the duration of the test. Plots of the time synchronous average for one revolution of gear pair stress wave, gear pair base band envelope, and gear pair raw base band were retained for each 5 min. data sample. A set of eleven feature plots showing different parameters (M1 to M11) of the data from beginning of the test to the end of the test was developed by the program for each of the three test gear data samples. These feature plots include raw vibration amplitude, noise ratio, harmonic ratio, nonmesh energy ratio, peak ratio, peak residual ratio, kurtosis, peak residual, and total energy. A limit was set on several of the feature plots to provide a warning when the limit was exceeded. Most of the test were continued after detection of a failure to determine what effect further damage would have on the various diagnostic parameters used in the system.

RESULTS AND DISCUSSION

Several gear surface fatigue tests were conducted using AISI 9310 spiral bevel gears. The tests were continued until a failure was detected, test run C ran a total of 527 hr. Some of the tests in this series were monitored using a commercial supplied gear vibration diagnostic system for the purpose of detecting the occurrence of surface fatigue spalls. One of the tests monitored by the diagnostic system (test A) developed a loose pinion hold down nut which allowed the pinion to run in an eccentric position. A second test (test B) monitored by the diagnostic system developed a fatigue spall in 107 hr test time. A third test developed a fatigue spall after 527 hr test time. The diagnostic results of these three tests are presented in this report.

Figure 8(a) shows a plot of the gear pair raw baseband and Fig. 8(b) shows a plot of the gear pair base band envelope from the test A with the loose pinion hold down nut which ran a total of 122 hr. These plots were taken near the end of the test. The horizontal scale covers one revolution of the gear and shows the condition of each of the 36 teeth on the gear, while the vertical scale is the amplitude of the signal and indicates the intensity of the tooth load. Figure 8(a) shows an increase and decrease in amplitude at three locations during one revolution of the gear which indicates the eccentric condition of the pinion which is running at three times the speed of the gear. The 36 teeth on the gear can be seen in these plots.

Figures 9(a) and (b) shows some of the feature plots, produced by the diagnostic system for the test A with the loose nut. These plots show the various parameters from the start until the end of testing and are generated from the data shown in Figs. 8(a) and (b). These plots show the total energy of the gear base band envelope (Fig. 9(a)) and kurtosis of the gear raw baseband (Fig. 9(b)). From these figures it appears that something happened approximately half way through the test, however there were no indications of a failure in the time averaged vibration plots at this time. Near the end of the test these feature plots still indicate that something has happened. The time averaged vibration plots show a three cycle wave giving a change in amplitude with each revolution of the pinion. This condition was caused by a loose pinion nut that allowed the pinion to run in an eccentric condition thereby producing the three cycle amplitude variation per gear revolution

Figures 10(a) to (d) shows the time averaged plots of the pinion stresswave and the pinion base band envelope from test B. Plots 10 a and b were taken at the start of the test and plots 10 c and d were taken at the end of the test. Near the end of the test the pinion stresswave and pinion baseband envelope show an increased amplitude for one tooth (tooth number 8). This indicates an increased loading for this tooth.

Figures 11 (a) to (c) are the time averaged plots of the gear stresswave, gear baseband envelope, and hunting tooth baseband envelope for test B. Each of these curves shows the gear teeth at locations 8, 20, and 32 to have a very high amplitude compared to the other teeth on the gear. These are exactly 12 teeth apart which is the number of teeth on the pinion. This increased amplitude of the three teeth on the gear is a result of the surface fatigue damage of one tooth on the pinion shown in Fig. 12.

Figure 13 shows six of the 14 feature plots, for test B. These plots were generated from the data shown in Figs. 10 and 11. Figures 13(a) to (c) are the (a) peak residual ratio, (b) kurtosis and (c) total energy of the pinion stresswave. Figures 13(d) to (f) are (d) the peak residual and (e) total energy of the gear baseband envelope and (f) the kurtosis of the gear stresswave. From these figures it is clear that not much damage occurred until the last 15 percent of total test time. After about 85 percent of total test time all of the above indicators began a steady and continuing increase in amplitude until the test was stopped. The pinion time averaged plots, Figs. 10(c) and (d) show that one tooth (tooth number 8) on the pinion was producing a higher dynamic load than the other teeth. This tooth was found to have fairly sizable fatigue spall as shown in Fig. 12.

Another gear test run C completed a total of 527 hr and had some fatigue damage on one tooth. The plots for the time averaged pinion stresswave, gear stresswave and hunting tooth baseband envelope are shown in Figs. 14(a) to (c) respectively. A signal from each individual tooth cannot be seen on these figures, however these figures show an increase in amplitude for one area of the pinion near tooth number 3 and for three areas of the gear. This indicates that one location on the pinion is contributing to an increase in dynamic load. Figure 15 shows the fatigue spalls on the pinion and gear for test C.

Figures 16(a) to (c) are the feature plots for test C and show the kurtosis and peak ratio for the gear stresswave and the peak residual ratio for the gear base band envelope. These plots show an increase in these parameters near the end of the test C and are an indication of increased vibration activity caused by the pinion fatigue spall. The earlier peaks were probably the result of stopping and starting the test.

From the above discussion and results of the test it can be concluded that the gear diagnostic system used in this test program for spiral bevel gears was able to detect gear tooth fatigue spalls on a regular basis. This test program also determined that the diagnostic system used for detecting gear fatigue spalls has been shown to be an effective system for that purpose.

CONCLUSION

A vibration diagnostic system was used for the purpose of detecting spiral bevel gear surface pitting damage in a closed loop gear fatigue test rig. The diagnostic system consists of a personal computer with an analog to digital conversion board and a diagnostic system unit and software. The diagnostic system performs time synchronous averaging of the vibration signal and produces an image of each tooth on any gear in a transmission. Several data processing methods were compared including gear stresswave, raw baseband vibration images, and the kurtosis and peak residual ratio of those vibration signals. The system provides limits for the various parameters and gives a warning when the limits are exceeded. Three spiral bevel gear tests were conducted using this system. The vibration data were analyzed at five minute intervals.

From the above discussion and results of the test it can be concluded that the gear diagnostic system used in this test program for spiral bevel gears has the following capability.

1. The system was able to detect gear tooth fatigue spalls on a regular basis.
2. The system can identify which tooth is damaged on the pinion or gear.
3. The system was able to identify an eccentric pinion condition.
4. This test program determined that the diagnostic system used an effective system for detecting gear failures in spiral bevel gears.

Table I.—Test gear design parameters

Number of teeth pinion/gear	12/36
Module (diametral pitch)	4.94 (5.14)
Mean spiral angle deg.	35
Mean cone distance mm (in.)	81.1 (3.191)
Face width, mm (in.)	25.4 (1.0)
Normal pressure angle, deg	22.5
Shaft angle, deg	90
AGMA class	12

Table II.—Lubricant properties

Specification DOD-L-85734	
Kinematic viscosity 311 K (100 °F) 372 K (210 °F)	 27.6 5.18
22 Flash point, K (°F) Pour point, K (°F)	544(520) 211(-80)
Specific gravity at 289 K (60 °F)	 0.995
Total acid number (tan) Mg Koh/g oil	 0.40

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Table III.—Spiral bevel gear test facility parameters (100 percent test condition)

Pinion shaft speed, rpm	11 400
Pitch line velocity, m/sec (ft/min)	35.4(6969)
Pinion shaft power, kW (hp)	371(497)
Test section flow rate cm^3/sec (gpm)	20(0.32)
Oil inlet temperature, K (°F)	357(183)
Fling off temperature, K (°F)	366(200)
Oil pressure MPa (psi)	0.55(80)

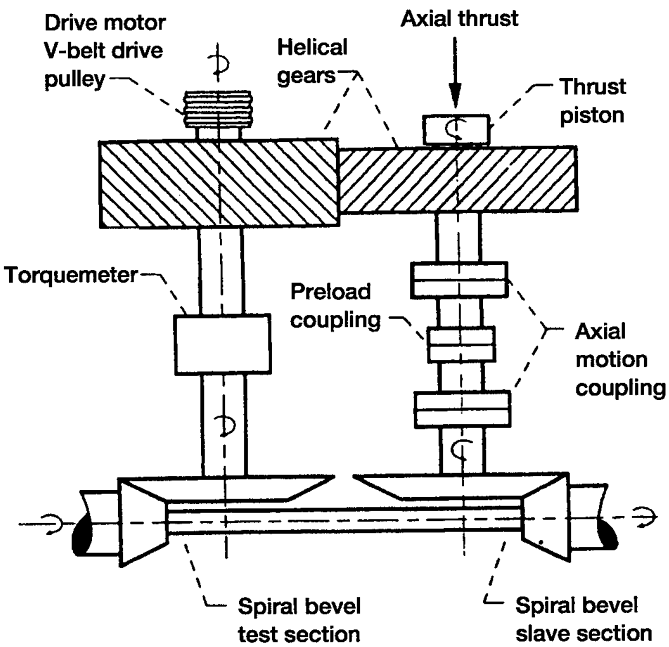

Figure 1.—Crossection view of test facility.

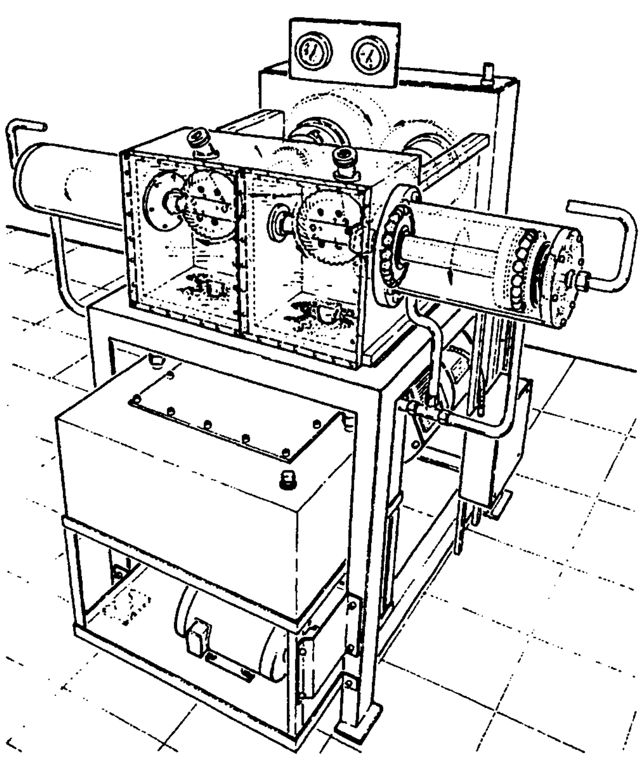

Figure 2.—Sketch of spiral bevel test facility.

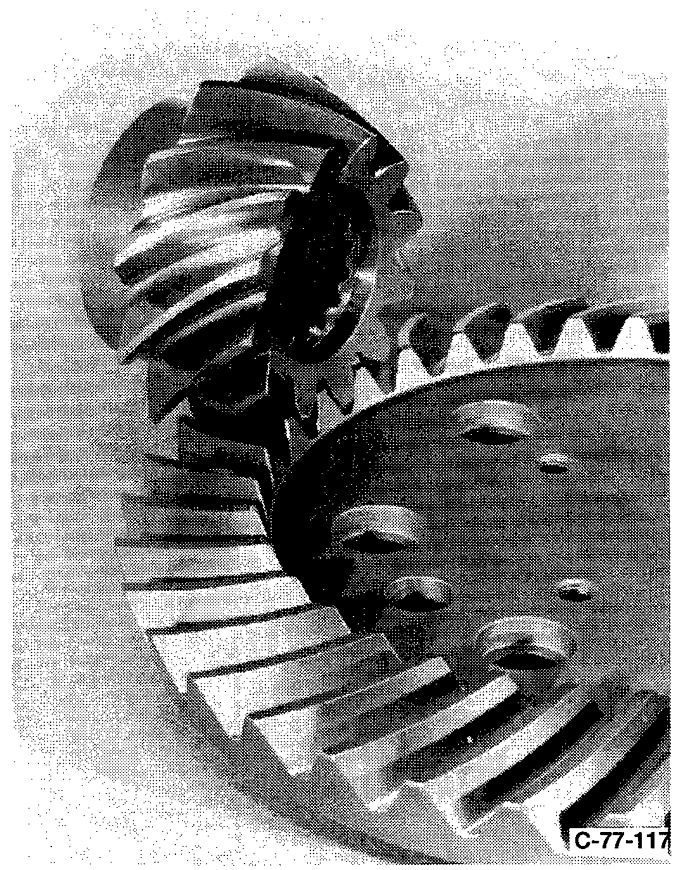

Figure 3.—Spiral bevel test gears.

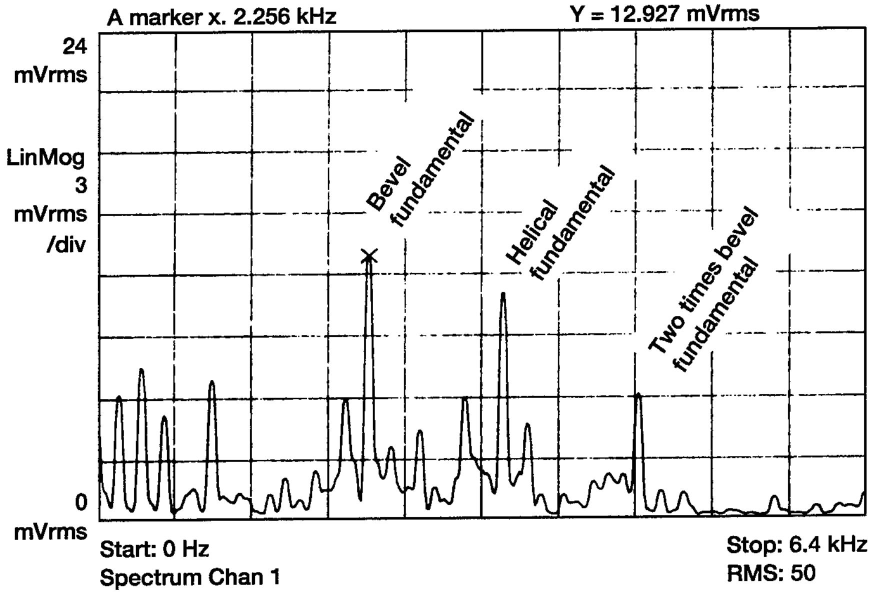

Figure 4.—Typical vibration spectrum from NASA Bevel Gear Rig Pinion speed 11 400 rpm, gear speed 3800 rpm, torque on gear shaft 938 N m (8300 lb/in.).

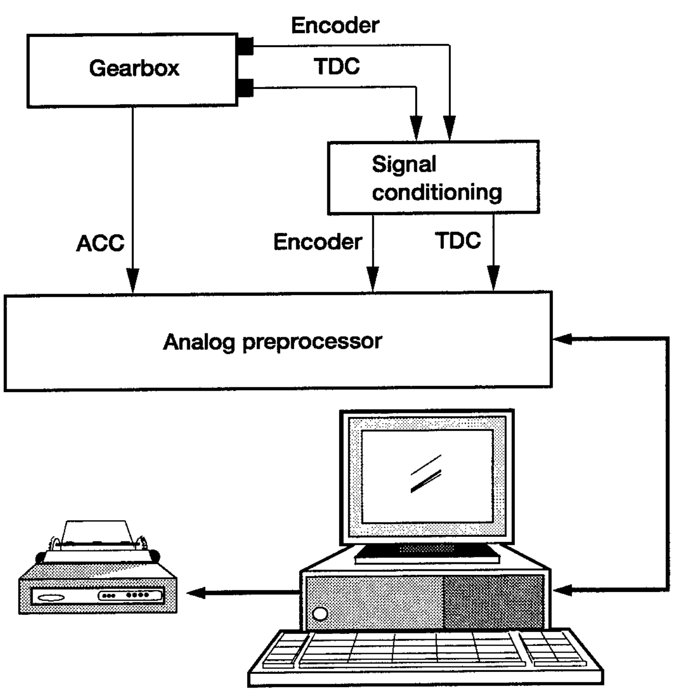

Figure 5.—Gear diagnostic system.

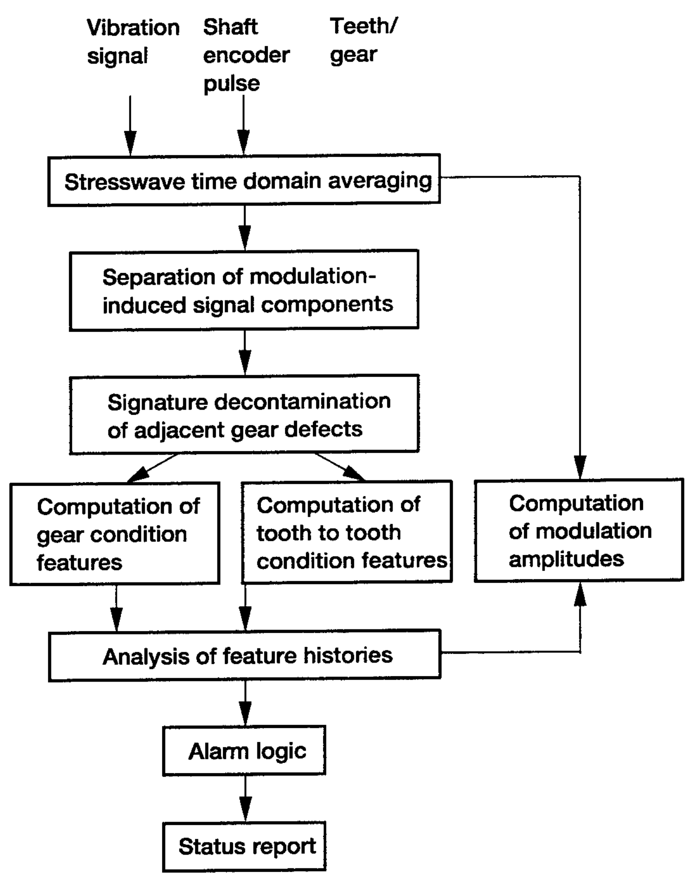

Figure 6.—Gear diagnostic system flow chart.

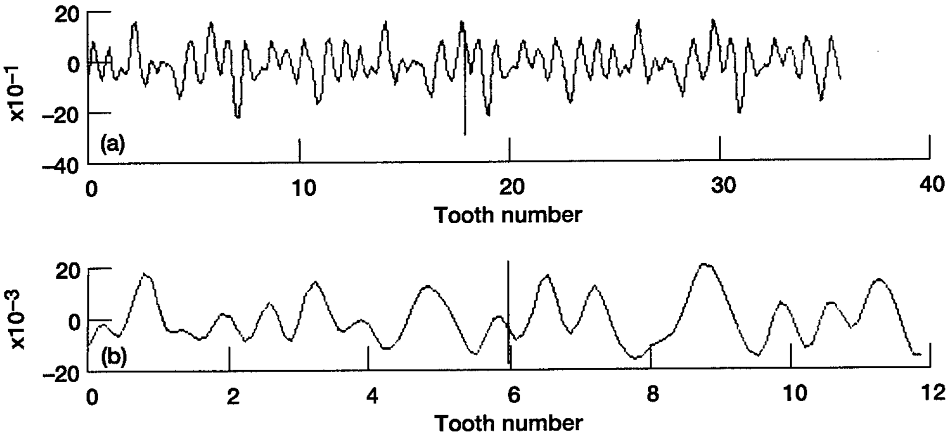

Figure 7.—Typical time synchronous average plots for spiral bevel test rig. Pinion speed 11 400 rpm, gear torgue 938 N m (8300 lb/in.). (a) Gear baseband plot. (b) Pinion stresswave.

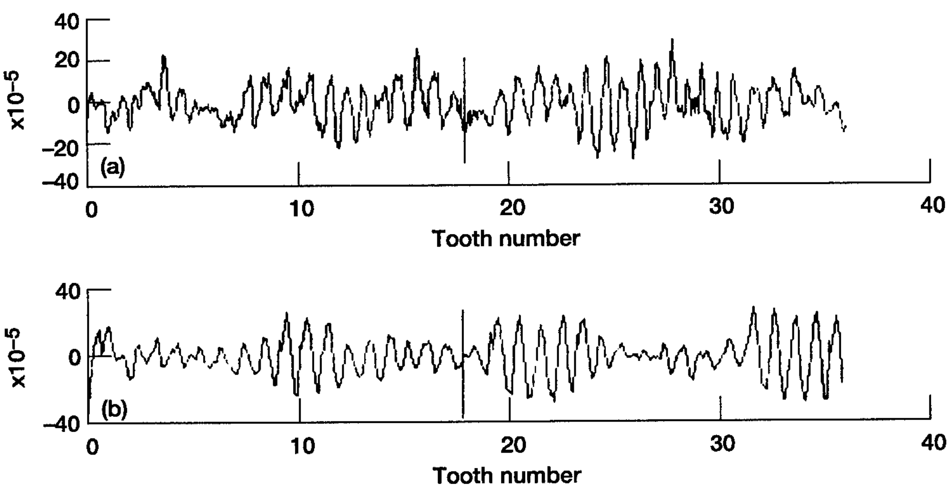

Figure 8.—Time synchronous average plots of gear for test run A. (a) Gear raw baseband plot. (b) Gear baseband envelope.

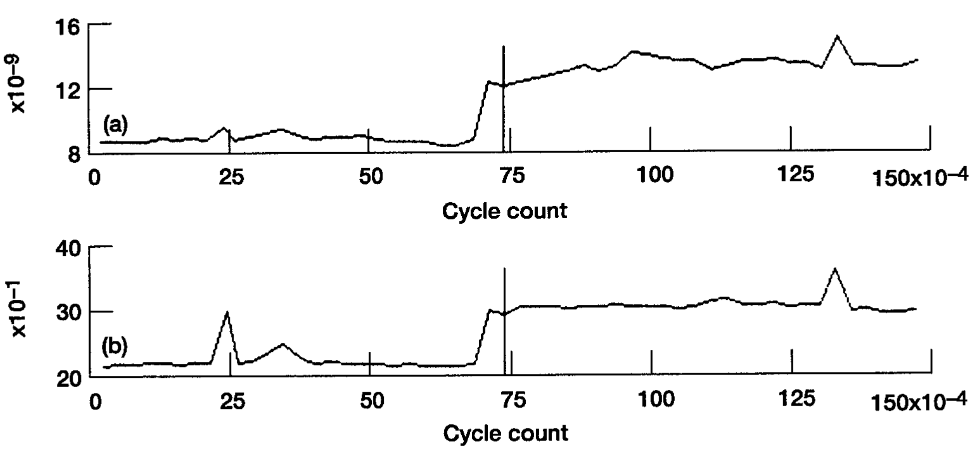

Figure 9.—Features plots of gear for test run A. (a) Total energy gear baseband envelope. (b) Kurtosis of gear baseband envelope.

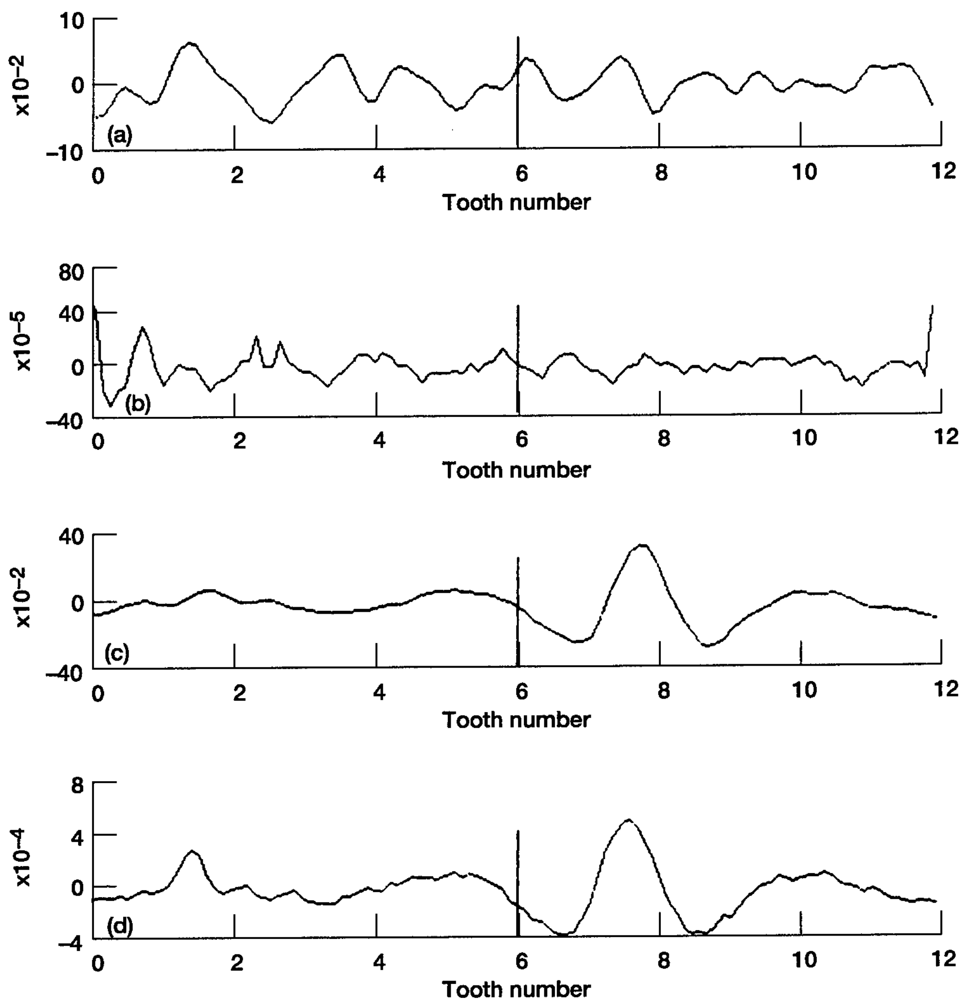

Figure 10.—Time synchronous average plots of pinion for test run B. (a) Pinion stresswave start of test. (b) Pinion baseband envelope start of test. (c) Pinion stresswave end of test. (d) Pinion baseband envelope end of test.

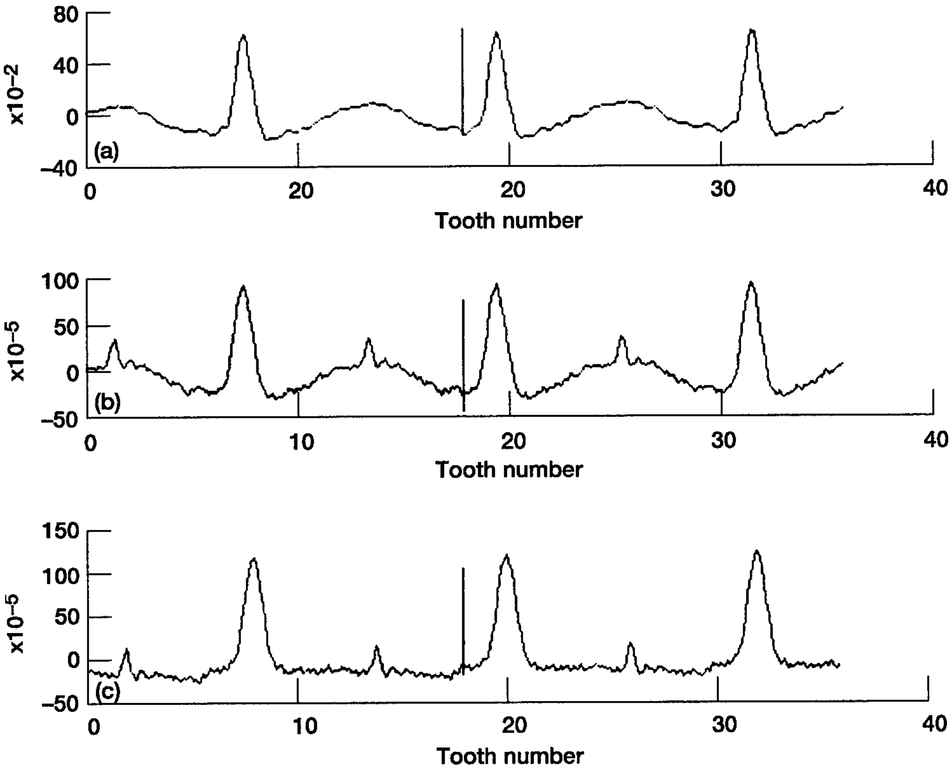

Figure 11.—Time synchronous average plots of gear for test run B. (a) Gear stresswave. (b) Gear baseband envelope. (c) Hunting tooth baseband envelope.

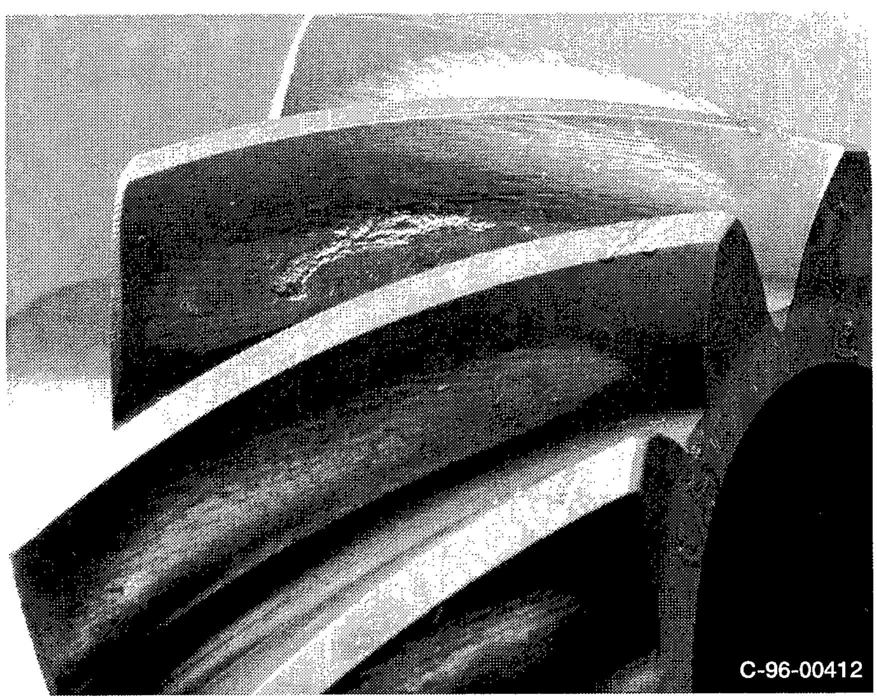

Figure 12.—Pinion tooth fatigue spall test run B.

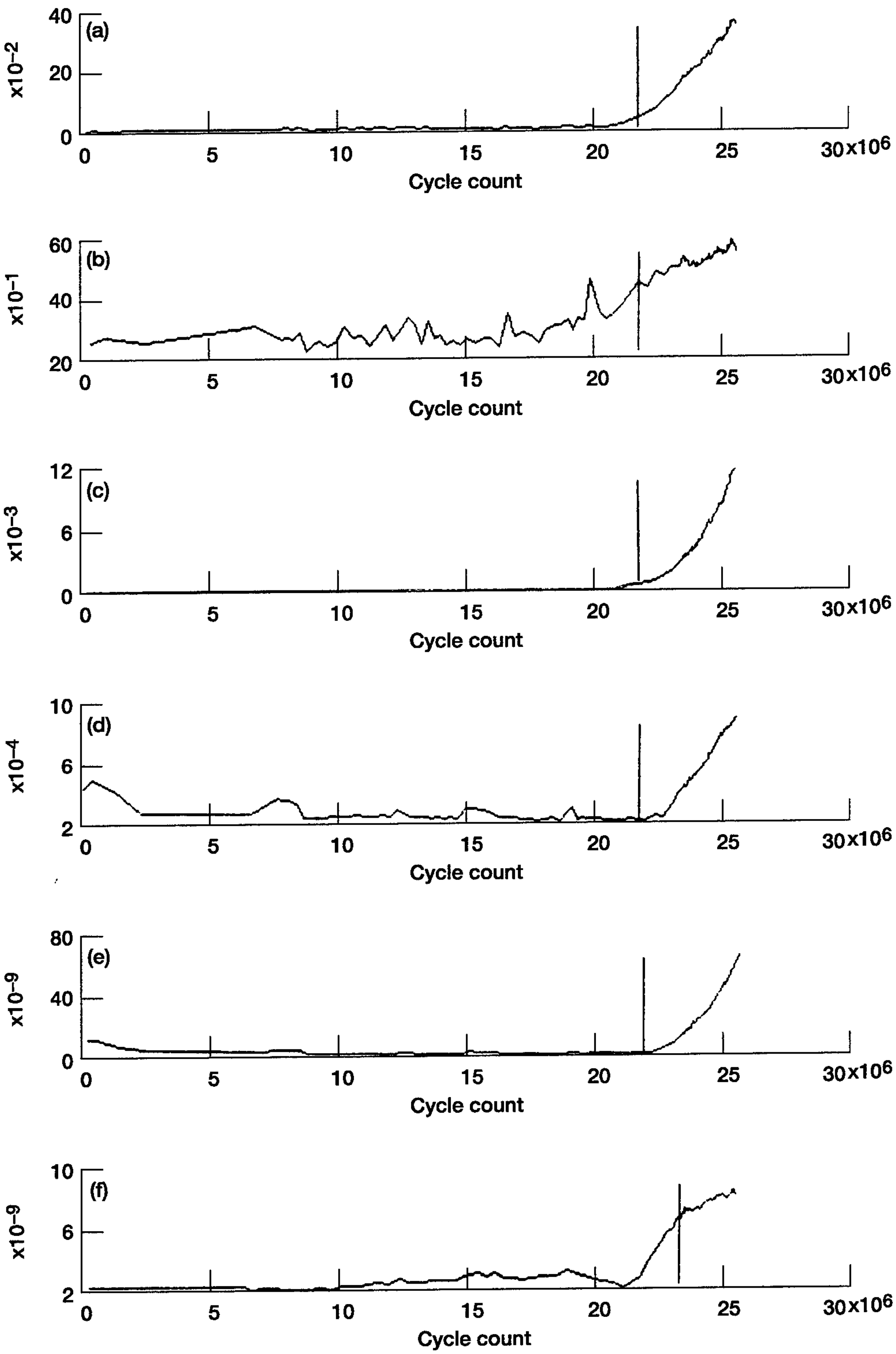

Figure 13.—Features plots for test run B. (a) Peak residual pinion stresswave. (b) Kurtosis pinion stresswave. (c) Total energy pinion stresswave. (d) Peak residual gear baseband envelope. (e) Total energy gear baseband envelope. (f) Kurtosis gear stresswave.

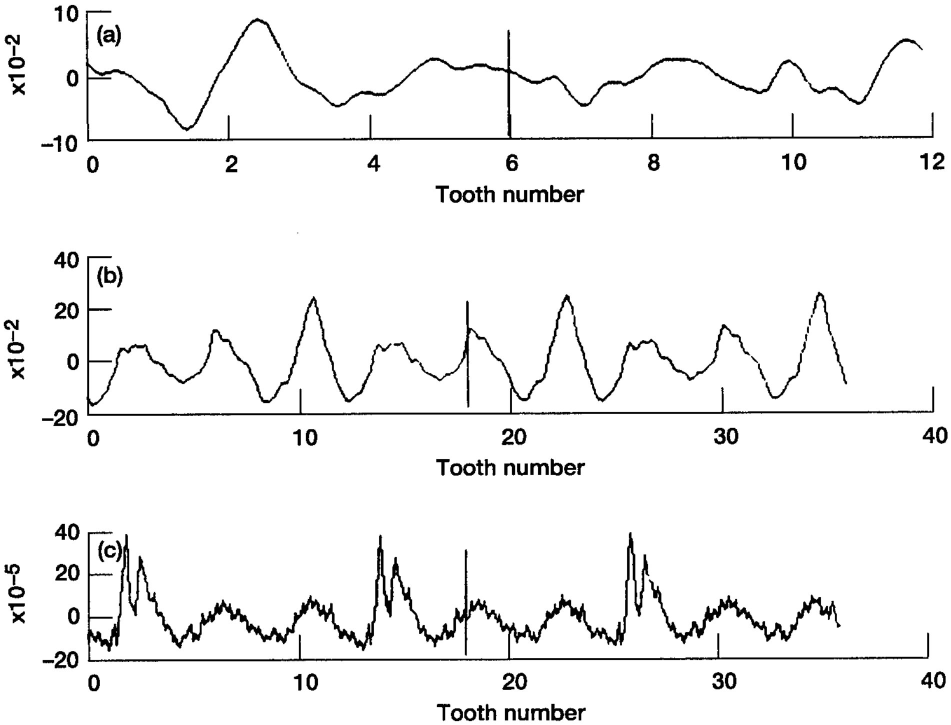

Figure 14.—Time synchronous average plots of test run C. (a) Pinion stresswave. (b) Gear stresswave. (c) Hunting tooth baseband envelope.

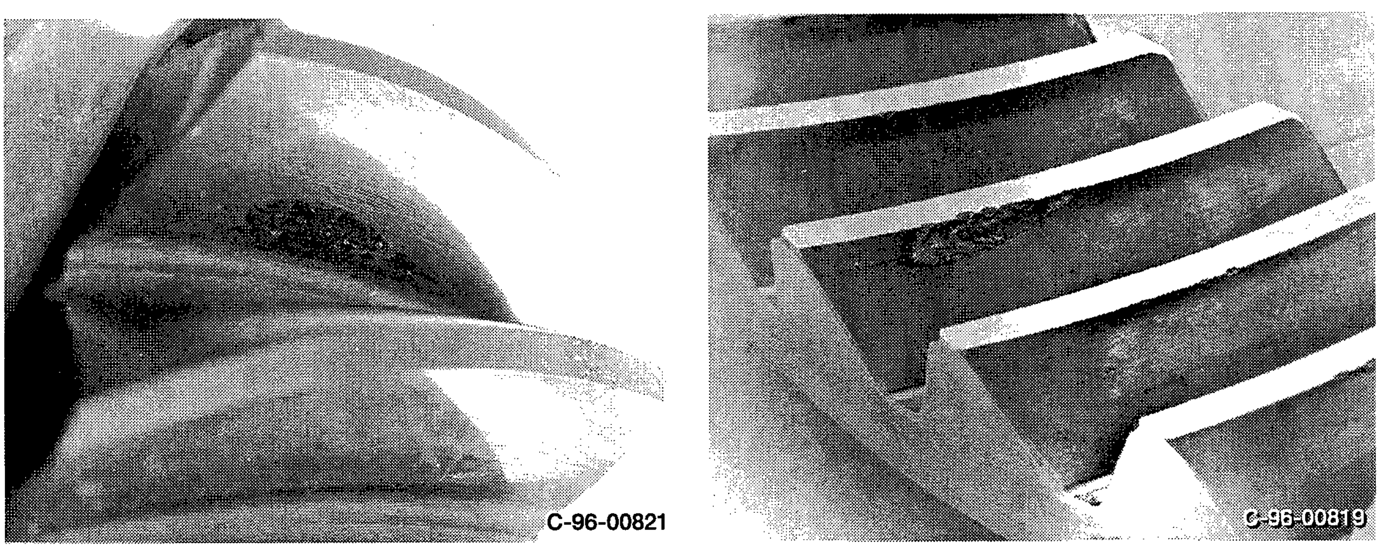

Figure 15.—Gear and pinion fatigue spall for test run C.

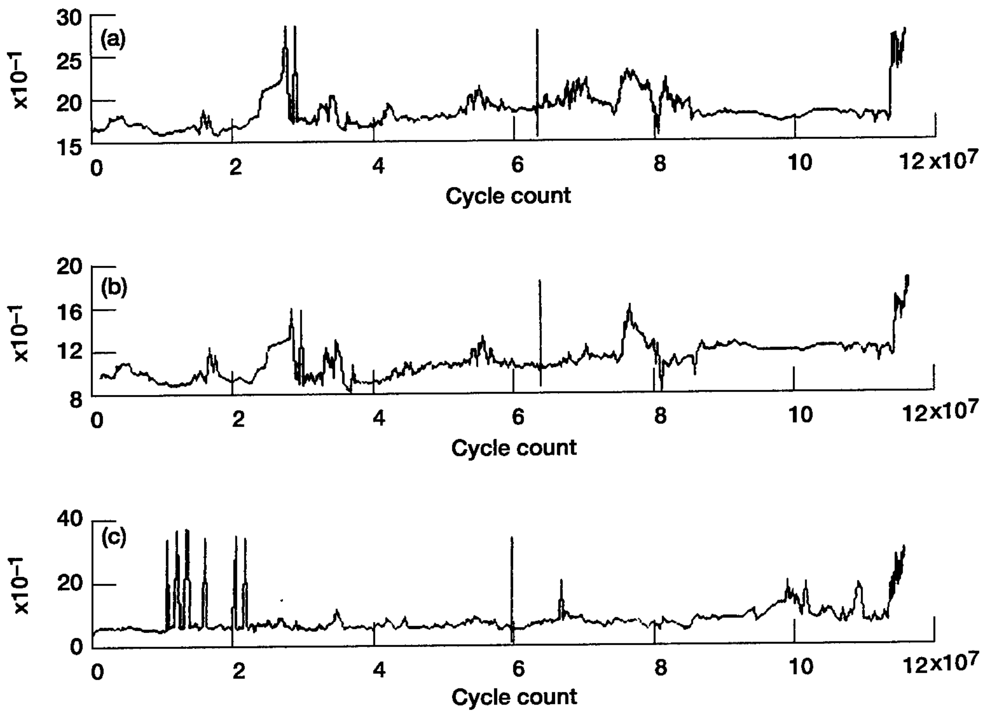

Figure 16.—Features plots of test run C. (a) Kurtosis gear stresswave. (b) Peak ratio gear stresswave. (c) Peak residual ratio baseband envelope.

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13. ABSTRACT ***(Maximum 200 words)***

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