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16. Abstract The OH-58 main transmission gearbox was run at varying output torques, speeds, and oil cooling rates. The gearbox was subsequently run to destruction by draining the oil from the gearbox while operating at a speed of 6200 rpm and 36 000 inch-pounds output torque. Primary cause of gearbox failure was overheating and melting of the planet bearing aluminum cages. Complete failure of the gearbox occurred in 28 $\frac{1}{2}$ minutes after the oil pressure dropped to zero. The gearbox air/oil cooler has sufficient cooling capacity margin for hot day takeoff conditions at a 117 percent power rating. The alternating and maximum stresses in the gearbox top case were approximately 10 percent of the endurance limit for the material. Deflection of the bevel gear at 67 000 inch-pounds output torque indicate a marginal stiffness for the bevel gear supporting system.		
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OH-58 HELICOPTER TRANSMISSION FAILURE ANALYSIS

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INTRODUCTION

The OH-58 is a U.S. Army Light Observation Helicopter (LOH) (fig. 1). The aircraft can carry up to five passengers and crew combined. It has a gross takeoff weight of 3000 pounds. It is currently powered by a 317 shaft horsepower (SHP) turbojet engine (Allison Model Specification 803-A).

The U.S. Army Aviation Systems Command (AVSCOM) has undertaken a program to upgrade the current OH-58 Light Observation Helicopter for utilization as an Interim Scout Helicopter (ISH) until such time as an Advanced Scout Helicopter becomes available. The mission of the Interim Scout will be primarily to acquire and designate targets for engagement by Army armed helicopters (AH-1Q Cobra-TOW and AH-1G Cobra) and secondarily to gather intelligence for the ground commanders. One of the key requirements for the accomplishment of this mission is survivability of the aircraft in hostile environments.

Helicopter operations in combat have shown that the transmission system is highly vulnerable to ballistic damage. One of the most vulnerable parts of the helicopter drive system is the lubrication and cooling system. Any type of ballistic hit in the lubrication system can result in a loss of lubricant to the gears and bearings in the transmission. If the supply of oil to the transmission is interrupted, the transmission may fail and seize thus limiting the time the pilot has to return safely to base or to make a forced landing.

The OH-58 Weapons System Manager requested NASA-Lewis Research Center's assistance in upgrading the current OH-58 main rotor transmission in performance and survivability. As a result of this request, a cooperative program was developed involving the OH-58 Systems Office, Corpus Christi Army Depot and Lewis Research Center. The NASA agreed to provide the following in support of this program:

1. Necessary professional and technical support personnel to instrument and perform testing on an OH-58 gearbox at Corpus Christi Army Depot to obtain baseline data on the gearbox.
2. Analysis of the data by personnel of the NASA Bearing, Gearing, and Transmission Section.

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3. Written recommendations by the Lewis Research Center to the OH-58 System Manager for the incorporation of technology necessary to upgrade the performance and survivability of the gearbox.

Tests of an OH-58 main rotor transmission were conducted by the NASA Lewis Research Center at the U.S. Army's Corpus Christi Army Depot under simulated field conditions. The results of these tests and recommendations are reported herein.

OH-58 HELICOPTER MAIN TRANSMISSION

The OH-58 main transmission (fig. 2) is rated for 270 horsepower continuous duty and 317 horsepower at takeoff for 5 minutes. The 100 percent input speed is 6200 rpm. The output speed is 354 rpm (ref. 1).

The input shaft drives a 19 tooth spiral bevel pinion. The pinion meshes with a 71 tooth gear. The input pinion shaft is mounted on triplex ball bearings and one roller bearing. The 71 tooth bevel gear is carried on a shaft mounted in duplex ball bearings and one roller bearing. The bevel gear shaft drives a floating sun gear which has 27 teeth. The power is taken out through the planet carrier. There are three planet gears of 35 teeth which are mounted on spherical roller bearings. The ring gear (99 teeth) is splined to the top case and therefore is stationary. The overall gear ratio is 17.44:1 reduction.

The planet bearing inner-races and rollers are made of AISI M-50 steel. The outer-races and planet gears, which are integral, are made of AISI 9310. The cage material is 2024-T4 aluminum. The gear shaft duplex bearing material is CVM 52CB. All other bearings are made of AISI 52100 with bronze cages. The sun gear and ring gear material is Nitalloy N (AMS6475). The input spiral bevel gear-set material is AISI 9310. Lubrication is supplied through jets located in the top case. An external air/oil cooler is used (fig. 2(b)).

TEST PROCEDURE

The OH-58 gearbox was placed in the Corpus Christi Army Depot OH-58 test stand after being fully instrumented (fig. 3). Figures 4 and 5 show the control room adjacent to the test cell. The test stand was calibrated and the gearbox filled with MIL L-23699 oil. The gearbox was operated for several hours to check out the instrumentation and to check for oil leaks, slipping operation and input shaft balance condition. This running time also gave additional break-in time for the new gearbox.

The gearbox was subsequently operated at 32 different test conditions with data taken after the oil temperature had stabilized. The temperature was considered stabilized when it changed less than 0.1° in 1 minute. The tests were run at output torque levels ranging from 2500 to 67 200 inch-pounds, oil-out temperatures ranging from 175° to 245° F, input speeds of 5580 (90 percent normal), 6200 (normal) and 6820 (110 percent normal) rpm. Oil-out temperature was controlled by blocking the airflow to the air oil cooler located on the transmission.

A cyclic test was conducted while continuously recording data. The test was started at zero speed and 2000 inch-pounds output torque. The speed was increased to 6200 rpm. Subsequently, the output torque was increased to 62 400 inch-pounds. While maintaining speed, the torque was decreased to 12 000 inch-pounds. The speed was reduced to zero rpm and then increased to 6200 rpm. The oil-out temperature was set at 200° F at the beginning of the test.

The final test run was the survivability test. The output torque was set at 36 000 inch-pounds (75 percent power rating), the input speed was 6200 rpm and the initial oil-out temperature was 200° F. The oil was drained from the bottom of the gearbox. The time from starting oil drain to zero oil pressure was approximately $1\frac{1}{2}$ minutes. It took another $1/2$ minute for all the oil to drain from the gearbox. The test was stopped at approximately 30 minutes from the time the oil drain was opened.

DATA ACQUISITION

Instrumentation provided for the OH-58 gearbox tests consisted of 23 iron-constantan thermocouples, four accelerometers, three strain gages and three proximity probes. The location of the instrumented components are shown in figure 2. Two silver brush sliprings were used to obtain data from the input and output shafts. Leads through the input shaft were used to obtain signals from a strain gage located on a pinion tooth and four thermocouples, one on a pinion tooth and three on the inner races of the three pinion shaft bearings. The output shaft slip ring was used to obtain signals from three thermocouples, one of which was located at the mast bearing inner race, the second was located close to the planetary support bearing inner race and the third was located close to the inside diameter of the sun gear. It was originally planned to locate a thermocouple on the planet bearing, however, the method of gearbox assembly did not permit a thermocouple at this location.

Thermocouples were also provided to measure temperatures at the following locations: the input shaft seal; the outer race of the two inner pinion bearings; the pinion roller bearing outer race; the oil splash off at the pinion mesh; the

outer races of the three bevel gear support bearings; the outside edge of the ring gear; three locations on the transmission housing; the oil inlet; the oil outlet; the oil sump; and the test cell ambient air. The 23 temperatures were recorded on a multichannel digital readout-recorder which would print all temperatures in approximately 20 seconds.

Accelerometers were mounted at three locations on the transmission case. One accelerometer was placed on the input shaft slipring. The accelerometer signals were recorded on magnetic tape for both the cyclic test and the survivability test.

The acceleration signals were examined at the Lewis Research Center. The raw time signals were played back from the magnetic tape. The time signals were complex waveforms due to many harmonic frequencies being present in the signals. Therefore, the raw signals were analyzed in two ways. First, the peak amplitude of the acceleration was plotted against time. Second, the frequency spectrum at four different points in time was determined.

Frequency spectrums were generated by using a real-time spectrum analyzer. Frequency spectra were obtained for the time periods of 0 to 1 minute, 7 to 8 minutes, 15 to 16 minutes, and 27 to 28 minutes. The data was averaged 256 times and 800 frequency points were used for 0 to 5000 Hz. The frequency resolution was better than 10 Hz. Spectrums for the ring gear and input pinion (No. 1 and 2 accelerometers) are reported herein.

The acceleration data can be helpful in determining the onset of failure in a transmission. The accelerometer responds instantaneously to mechanical vibrations in the gearbox which may be indications of failure in progress. Reliance on thermocouples as detectors of impending failure is not satisfactory because temperature response is too slow. The very short period of damage causing activity within the gearbox may be insufficient for temperatures to change. An additional drawback is that temperature gradients exist in the gearbox and hot spots are very much localized. Unless a thermocouple is very strategically placed, it will be too far away from the damaged area to record the effects of damage in the critical points of the transmission. On the other hand, vibrations are transmitted very rapidly and variations in frequency content of the signals give a measure of changes in the mechanical structure as fast as they occur.

Three inductance type proximity probes were located at different locations to measure various gearbox deflections and load conditions. One was mounted inside the gearbox, close to the top side of the bevel gear at the point where the pinion meshes with the gear, to measure bevel gear deflections under load. Another was mounted close to the transmission housing adjacent to the ring gear to measure housing deflections from the planet gear loads on the ring gear. A third

proximity probe was mounted close to the output shaft to measure output shaft motion.

The outputs of the three proximity probes were connected to drivers with a -18 V dc power supply and to differential amplifiers to amplify the signals and then to the 14 channel FM tape recorder with provisions to monitor the signals on a four channel oscilloscope.

Three strain gages with temperature compensating gages were also located on the various parts of the OH-58 transmission. One strain gage was located on the top case adjacent to and parallel to the ring gear and another was located on the slant section of the top case perpendicular to the ring gear. These gages were used to measure the case deflection and stress caused by the planet gears under load. The third strain gage was mounted at the small end of one of the pinion teeth, and connected through the input shaft slipring. The strain gages were connected to a bridge circuit and strain gage amplifier and to the 14 channel FM tape recorder with an additional connection to monitor the signals on the oscilloscope.

The 14 channel FM tape recorder was used to record the four accelerometer signals, three proximity signals, three strain gage signals, the input and output shaft torques and a 1 volt 1 kHz reference signal. The four channel oscilloscope was used to monitor the output signals in order to determine their condition. All wiring, including the thermocouple wiring, were shielded cables and were run from the test cell to the control room through shielded conduit to preclude the possibility of signal interference.

RESULTS AND DISCUSSION

Operating Temperatures

Figure 6 contains the temperatures of the various transmission components as a function of output torque. The respective numbers in the figure indicate the location of the thermocouples in figure 2(a). The oil-out temperature was maintained at 200° F and the input speed was 6200 rpm. The temperature recorded for the 67 000 inch-pound torque (140 percent power rating) was not stabilized due to the short time at which this overload condition (140 percent torque) was maintained. The highest temperature was recorded for the pinion seal which was 228° F at an output torque of 62 400 inch-pounds (130 percent power rating).

In order to determine the transmission oil-out stabilization temperature, the transmission was run with different output torque conditions while the oil cooler was either completely blocked, partially blocked or completely open. The results of these tests are summarized in table I and shown in figure 7. The maximum oil-out temperature obtained for zero percent and 50 percent oil cooler

blockage at 56 000 inch-pounds output torque (117 percent power rating) was 197° and 210° F, respectively. When the oil cooler was completely (100 percent) blocked, the transmission stabilized at an oil out temperature of approximately 246° F at an output torque of 48 000 inch-pounds (100 percent power rating). The 173° F oil outlet temperature at near zero output torque and 100 percent oil cooler blockage indicates excessive heat generation which is probably caused by excessive oil churning resulting from too much oil in the gearbox.

The ambient temperature condition was approximately 85° F so that for a hot day condition having a 125° F ambient temperature, the gearbox should stabilize at approximately 240° and 250° F oil outlet temperature for the completely open and 50 percent blocked conditions, respectively. These data indicate that the oil cooler has adequate margins for hot day take-off conditions at 117 percent power rating.

At a torque level of 36 000 inch-pounds and a speed of 6200 rpm the drain plug at the bottom of the transmission housing was removed and the oil in the transmission was allowed to drain out. The results of this test are shown in figure 8 wherein the component temperatures are shown as a function of time. Time zero is the time at which the oil drain plug was removed.

At 1.5 minutes after the oil began to drain from the transmission, the oil pressure dropped to zero. At this point, the temperatures of the various components began to rise reflecting a lack of cooling oil.

After approximately 7 minutes, there is an abrupt change in the rate of temperature increase which is more noticeable at the input pinion where temperatures reached 490° F on the pinion gear. This increased temperature was caused by the loss of the first planet bearing aluminum cage (fig. 9) causing a torque increase on the input pinion and some molten aluminum coating the pinion teeth (fig. 10).

Figure 8(c) shows an increase in temperature in the planet gear area at approximately 13 minutes.

The rate of increase of the pinion temperatures slowed until at approximately 14 minutes the rate began to increase again (figs. 8(a) and (b)). These increases in temperature were caused by additional planet bearing cages melting which allowed the spherical rollers in the planet gear bearings to tumble and slide within the bearings (fig. 11).

At approximately 27 minutes there is an additional increase in the rate of temperature rise as seen in the planet and (fig. 8(c)), the pinion area (figs. 8(a) and (b)) and to a lesser extent in the bevel gear bearing area (fig. 8(d)). At this point in time the gearbox is approaching complete failure. At these temperatures, the planet gears have very little strength. The lock nut on the planet bearing was being machined by the bevel gear bolt heads (fig. 12). The pinion

gear was approaching 600°F . The temperature near the sun gear and planetary carrier bearing was increasing rapidly indicating a much higher temperature in the sun gear and planet gears. Subsequent examination of the planetary bearings and gears indicates that they reached temperatures in excess of 1100°F .

Temperatures in the oil system were measured and are shown in figure 8(e). A thermocouple placed near the transmission recorded the ambient temperature of the test cell. It was $84 \pm 1^{\circ}\text{F}$ for the duration of the test. The thermocouple attached to the oil drain measured the slow temperature decay to ambient conditions from the time the drain plug was removed. The two thermocouples in the oil line measured the initial reduction in temperature that resulted from the disappearance of hot oil in the lines. The temperature later rose again in response to the heating by conduction from the hot gearbox.

Dynamic Failure Analysis

Figure 13 shows the record of the four acceleration peak amplitude levels as a function of elapsed time from the moment the lubricant drain was opened. The respective numbers in the figure indicate the location of the accelerometers, shown in figure 13. The accelerations are plotted on a logarithmic scale. Accelerations were measured (1) in the horizontal direction on the top case in the region of the ring gear, (2) in the horizontal direction on the lower case near the input spiral bevel pinion gear shaft, (3) in the horizontal direction on the bottom case in the plane of the spiral bevel gear, and (4) in the vertical direction on the outboard end of the slipring unit. The acceleration peak levels were lowest on the slipring unit and highest on the top case near the planetary ring gear. It was expected that the highest vibrations would be in the region of the ring gear on the top case. This is because the planet spur gears are the heaviest loaded gears. Also, the spur gears should have higher dynamic loads than the spiral bevel gears.

The first indication of failure occurrence in the gearbox was at zero plus 7 minutes. The vibration level increased very abruptly, by approximately 80 percent, at all three locations on the transmission. This rapid buildup and subsequent sustained level of vibration was an indication of a cage failure in the planet gear bearings. The thermocouples located at the spiral bevel mesh exit airstream showed a rapid increase to 360°F in the 2 minute time period following this event. The thermocouple located on the pinion tooth showed a much larger increase in temperature to 490°F in the same time period. The rapid increase in these temperatures indicated the sudden increase in the torque passing through the input gear mesh caused by the cage loss in the planet bearing. After 9 minutes, the vibration levels decreased briefly. Also, the input pinion

gear tooth temperature decreased to 455° F in the 9 to 13 minute time interval. This brief decrease in temperature was probably caused by the exiting of cage debris from the planet gear bearing cavity.

At 10 minutes, vibration amplitude began increasing again, with the peak level occurring at 14 minutes. At this time, a level of 40 g's was observed on the ring gear. This was about 230 percent of the level at time zero. This vibration activity was caused by the breakup of the second and third aluminum planet gear bearing cages. Once again, the vibration level decreased while the temperatures continued to rise at a steady rate. The accelerometer mounted on the lower case at location 3 dropped off at 18.5 minutes. The high temperature had weakened the mounting cement. During the time from 17 to 27 minutes the acceleration levels at the number 1 location were erratic and stayed between 10 and 16 g's. During this time, the rollers were skidding and their axes were badly skewed. The drag torque offered by the damaged bearings was sufficient to cause the inner races to turn on the planet bearing posts. The cotter key was sheared on two of the bearings and the shaft nut was being turned by the bearing inner race.

At 27 minutes the vibration levels again increased at the ring gear to the 30 to 40 g level. At that time the shaft nut was being struck by the bevel gear support bolts which were immediately below and on the gear shaft. The relative rotational speed of the planet carrier and the gear shaft was 1300 rpm. The retaining nut was rapidly machined away by the gear shaft bolts. Subsequently, an inner race of one of the spherical roller bearings came off the shaft allowing the spherical rollers to drop out of the bearing. One of the rollers wedged into the ring gear-planetary gear mesh. This caused the planet carrier to lock up. As a result, the teeth on the sun gear were destroyed by the torque overload. At 30 minutes after the oil drain was opened, complete transmission failure occurred.

Spectrum Analysis

The results of the spectrum analysis are shown in figure 14 for the input pinion location and in figure 15 for the ring gear location. Figures 14(a) and 15(a) show the frequency content of the normal transmission. The next three time periods at which a spectrum plot was obtained were selected in order to show the frequency content of the signal immediately after the several stages of high amplitude vibrations were observed. A comparison of the spectra for these time periods show how the transmission vibration signature changes as the shut-down failure point is approached.

The area under the curve on the spectrum plot is a function of the RMS value of the acceleration signals. But the most important items of interest are the

frequencies at which the major strength of the vibration is centered. The two dominant sources of vibration are the input spiral bevel gear mesh frequency (1963 Hz) and the planet-sun-ring spur gear meshing frequency (586 Hz). At time zero the strongest amplitude is caused by bevel gear mesh. There are first and second harmonics present. These are denoted by the circles on the plots. The spur mesh fundamental frequency and the next seven harmonics are visible also. These are denoted by the inverted triangles. The presence of higher harmonics is an indication of the degree of deviation from a sine wave type of acceleration signal. Also visible on the plots near the major frequencies are several subordinate peaks. These are the side-band frequencies. They are introduced by the modulating effect of the lower frequency components. The modulation is caused when the gear mesh vibration grows stronger as the planet approaches the accelerometer and then again diminishes as the planet passes by the accelerometer.

The spur gear mesh frequency is 586 Hz and the planet pass frequency is 18 Hz. Therefore, side band responses of 586 ± 18 Hz are expected. The higher harmonics may also have modulation sidebands. Another source of modulation is the rotational frequency of the planet gears which is 16 Hz. However, this will produce sidebands which are only 2 Hz away from the sidebands produced by the planet pass frequency. This difference is beyond the resolution of the spectrum analyzer. Modulation sidebands are also visible at 1963 ± 103 Hz. They are caused by the eccentricity of the pinion gear which makes the tooth impact loads vary in intensity as the gear rotates. It was known that the input shaft was not well balanced and this is the primary cause of these sideband responses. Similar sidebands also appear on the second harmonic of the spiral bevel gear mesh frequency ($2 \times 1963 \pm 103$ Hz).

After 7 minutes of operation two classes of change in the transmission signatures are noticed by comparing figures 14(b) and 15(b). The first is an increase in the amplitude of the 888 Hz frequency line. This is the frequency generated if it is assumed that one row of the rollers in the planet bearing has cage debris on it. The pulsating effect that this would have on the input mesh is seen by the presence of upper and lower side band response at 1963 ± 888 Hz. This is caused by the periodic variation in torque when the rollers roll over the cage debris.

The second general effect is that the percentage increase in the spiral bevel related amplitudes was approximately 75 percent. No significant increases in the spur mesh related amplitudes were observed. This indicates that gear dynamic loading at the spiral bevel mesh had increased, but the gear loads and meshing action at the sun planet and planet ring gear meshes were essentially normal. These changes in the transmission signatures indicate that planet bearing degradation began at the 7 minute mark. The sudden appearance of a frequency

component at 1550 Hz was also observed when comparing figure 14 (b) with figure 14 (a). It was speculated that this was a fourth lower side band of the bevel mesh fundamental frequency ($1963 - (4 \times 103)$). The immediate source of this component is unknown.

Figures 14(c), 14(d), 15(c), and 15(d) indicate that two primary changes in the nature of the spectra have occurred. The first is the large increase in the fundamental spur mesh frequency and its harmonics. This indicates that imperfect mesh action is occurring at the sun-planet-ring gear meshes. Secondly, there is a large amount of sideband activity around the spur mesh frequencies. The buildup of sideband amplitudes indicates that the planet bearings were in very poor condition. The increased runout and misalignment of the bearings caused by cage loss and roller skewing is the cause of the modulation. There is also a large level of broadband acceleration that is especially noticeable on figure 14(d). This is caused by the skidding of the badly misaligned rollers in the planet bearing. The amplitudes of acceleration are very low in the 2 to 5 kHz range in figure 15(d). This is attributed to the high temperature (500°F) softening of the cement used to mount the accelerometer.

Strain and Bevel Gear Deflection Measurements

Strain gages were placed on the top case in two places. Strain gage number 2 (SG2) was placed on the outside diameter in order to measure the circumferential strain. Strain gage number 3 (SG3) was placed on the sloping part of the top case near the edge. It measured the radial strain. Strain gage number 1 (SG1) was located at the root of one of the gear teeth on the spiral bevel input shaft. Unfortunately, this strain gage measurement was not successful.

The raw signals from the strain gage amplifiers were recorded on magnetic tape. Figure 16 shows a sample of the time trace of the strain gages. Strong pulsations at the planet pass frequency are clearly present in both of the plots. Also shown in figure 16 is the signal from the proximity probe which measured the deflection of the spiral bevel gear.

This deflection represents separation of the bevel gear teeth. Part of the separation is from the gear case deflection at the bevel gear mounting. The other cause of the separation is the deflection of the bevel gear shaft. The waviness of this signal is caused by the runout of the back surface of the gear. The frequency of the main pulsation from the probe signal agree with the rotational speed of the gear.

Figure 17 shows the bevel gear deflection as a function of output shaft torque. The deflection is very nearly a linear function of the torque with the spring rate being approximately 6.8×10^6 in.-pound/in. This deflection can cause reduced load capacity and life (ref. 2).

Figure 18 summarizes the stress values that were obtained from the strain gage measurements. The stress results are reported as having a mean value and a peak-to-peak fluctuating value. Under all parametric temperature and torque conditions, the mean stresses on the top case did not exceed 3350 psi compression. The fluctuating peak-to-peak values were less than 2700 psi.

For the circumferential or hoop stress on the outside diameter of the top case, the mean stress was always less than 800 psi compression. The fluctuation component of the stress was less than 2400 psi. The upper case was forged from 4032-T6 aluminum alloy. The room temperature tensile strength is 55 000 psi, the yield strength is 46 000 psi, and the endurance limit from R R Moore tests is 16 000 psi for a life of 5×10^8 cycles (ref. 3). The worst condition of fluctuating stress was less than 10 percent of the endurance limit of the material.

GENERAL OBSERVATIONS

The major weakness to the survivability of the OH-58 main transmission was the cages of the spherical roller bearings of the three planet gears of the planet carrier. These cages are made of 2024-T4 aluminum having a melting point of 1125° to 1250° F. The maximum useful temperature of this aluminum alloy is 300° to 400° F. As a result, the first and most mandatory change is to substitute silver-plated cages made out of vacuum remelted AMS 6414B which is a low carbon steel. These cages retain adequate strength to approximately 700° F. Cages made out of this material have been run continuously at temperatures of 425° F and at speeds of 3×10^6 DN (where DN is a speed parameter defined as the product of the bearing bore in mm and the bearing speed in rpm) for a cumulative time period of approximately 75 000 hours (ref. 4).

The lower gearbox assembly which comprises the input spiral bevel gear set was not damaged during the survivability test (fig. 19). The maximum temperatures within the lower gearbox assembly were in the range from 325° to 525° F. It would appear from the condition of gearing and bearing components after the survivability test that the materials were adequate for the time duration of the test which was 30 minutes. However, for longer time periods both the bearing and gear materials may be marginal. Consideration should be given to replacing all the bearing materials within the transmission which are AISI 52100, CVM 52 CM, and CVM AISI M-50 with vacuum-induction melted vacuum-arc remelted (VIM-VAR) AISI M-50 steel. This steel has high temperature bearing operating capability to 600° F (refs. 5 and 6) and rolling-element fatigue lives of approximately 20 times current design requirements (ref. 4). Manufacturing specifications incorporating the technology of reference 4 is contained in reference 7.

Insufficient gear data exists to recommend a high-temperature gear material in place of the AISI 9310 steel currently being used for the bevel gear set. Endurance testing of currently used and prospective gear materials are being performed at the NASA-Lewis Research Center. Out of those potential high-temperature materials studied, Super Nitralloy exhibited lives exceeding AISI 9310 (refs. 8 and 9). Nitralloy N (AMS 6475) is a material which also has high temperature characteristics but has not been endurance tested in rolling-element fatigue. However, this material has found acceptance as a gear material and is used for the sun and ring gear material in the OH-58 transmission. Consideration should be given to changing all the gear materials except as noted below to Nitralloy N (AMS 6475).

Because the outer race of the planet bearings is integral with the planet gear, there is a requirement to have a case depth of 0.040 inch prior to grinding the race surfaces. This results in a final case depth of 0.030 inch. Since the case depth of nitrided steels such as Nitralloy N is approximately 0.020 inch, this class of steel is precluded from use as the planet gear. AISI 9310 which is the current planet gear material does not have sufficient hot hardness characteristics to assure continuous operation as a bearing material at temperatures much above 300° F. As a result, another case carburized material must be considered. Because of a lack of rolling-element fatigue data with carburized steels having high-temperature capabilities no recommendations can be made for a substitute material at this time. Testing of various carburized materials are currently being performed by the NASA-Lewis Research Center.

Two steps can be taken to reduce heat generation at the planet gears. The first and most obvious step is to reduce the load on the planet gears. This step can be accomplished by replacing the three planetary system with a four planetary system. This will reduce the load for each planet gear-bearing combination by 25 percent.

The second would be to replace the spherical roller bearings with crowned cylindrical roller bearings. This can result in reduced heat generation within the bearing by as much as 30 percent (ref. 10). However, this step would require rigid straddle mountings in place of the overhung mountings currently on the planetary spider. However, there is a probability that gyroscopic induced thrust loads may be unacceptable for these bearing. As a result, further transmission bench testing would be required.

The cyclic stress levels in the top case do not appear to be of sufficient magnitude to cause fatigue of the top case. However, the transmission tests were not conducted under actual flight conditions. The amplitude of the cyclic stresses may be higher in flight than in the tests reported herein. It is therefore recommended that strain gages be placed on a transmission installed in an aircraft and data be recorded during flight operation.

A critical area within the transmission is the main mast bearing retention housing in the upper gearbox. This housing supports the weight of the helicopter. The casing of the upper gearbox is made out of 4032-T6 aluminum. At room temperature the tensile strength of this material is 55 ksi. At 300° F, the tensile strength is 37 ksi. It is 13 ksi at 400° F (ref. 11). Hence, at room temperatures above 300° F this material becomes marginal for the purpose for which it was intended. The maximum measured temperature of the mast bearing retention housing was approximately 348° F. This temperature is considered acceptable under the operating conditions reported for the material to assure survivability for 30 minutes.

Field experience has shown that the mast bearing which is made from CVM AISI M-50 has produced operating fatigue lives less than 1000 hours in many cases. The bearing life can be improved by using VIM-VAR AISI M-50 as previously discussed. However, should the transmission be uprated above 375 HP this bearing should be increased in size in order to increase its load capacity and life. In addition, the gearbox top cover should be strengthened in order to assure its capability of sustaining high loads.

Increased loading on the transmission above 375 HP will result in increased separation of the bevel gear set. As discussed in the text, this will result in a reduction in the load capacity of the bevel gear set, and thus life. As a result, consideration should be given to increase the stiffness of the bevel gear mounting and the bevel gear shaft.

SUMMARY OF RESULTS

The OH-58 main rotor transmission was run on the Corpus Christi Army Depot's OH-58 test stand at Corpus Christi, TX at output torques from 2500 to 67000 inch-pounds and at input speeds of 5580, 6200 and 6820 rpm with varying oil cooling rates. The gearbox was subsequently run to destruction by draining the oil from the gearbox while operating at a speed of 6200 rpm and 36000 inch-pounds output torque. The following is a summary of the results obtained.

1. The primary cause of failure in the survivability test was overheating and melting of the planet bearing aluminum cages with subsequent loss of one planet bearing and loss of teeth on the sun gear from excessive temperature and load.
2. The transmission air/oil cooler has sufficient cooling capacity margin for hot day takeoff conditions at a 117 percent power rating.
3. Deflection of the bevel gear for output torques up to 67000 inch-pounds (140 percent power) indicates a marginal stiffness for the bevel gear supporting system.

4. The alternating and maximum stresses in the gearbox top cover for output torques up to 67000 inch-pounds (140 percent power) and without helicopter loads were within approximately 10 percent of the endurance limit for the material at room temperature.

5. After oil pressure dropped to zero, the length of operating time until the first planet bearing cage failed was five and one half minutes. The remaining two cages failed in approximately 11 minutes. Complete failure of the transmission occurred in 28 1/2 minutes.

RECOMMENDATIONS

As a result of evaluation of the test data from the load and temperature survey tests and the survivability tests, the following recommendations are included to improve the gearbox survivability and load carrying capacity. These recommendations are listed in the order of importance and generally with the least potential cost impact on any proposed product improvement program.

1. Replace the planet bearing cages and the planetary support bearing cage with silver plated steel cages. This change will give a much needed increased temperature capability.

2. Change the three planet system to a four planet system. This change would reduce the load on each planet bearing by 25 percent resulting in reduced heat generation and increased life.

3. Change the spherical roller bearings to cylindrical roller bearings with rigid straddle mounting to replace the overhung mounting. This change would reduce the heat generation in the planet bearing up to 30 percent. Further testing would be required.

4. The material for all bearings within the transmission should be changed to VIM-VAR AISI M-50. This change will result in longer bearing fatigue life and improved elevated temperature operation.

5. Change the AISI 9310 gear material to a high-temperature material for longer life and survivability at high temperature. The specific material to be used should depend on existing endurance data and material availability.

6. Increase the stiffness of the bevel gear support shaft to reduce the amount of separation of the bevel gear under load and increase the power capability.

7. For uprating above 375 HP, the gear box top cover should be strengthened, the mast bearing increased in size, and the bevel gear mounting should be stiffened.

8. The improvements incorporated into an advanced transmission by the project office should be tested to determine the extent of the improvement and transmission reliability.

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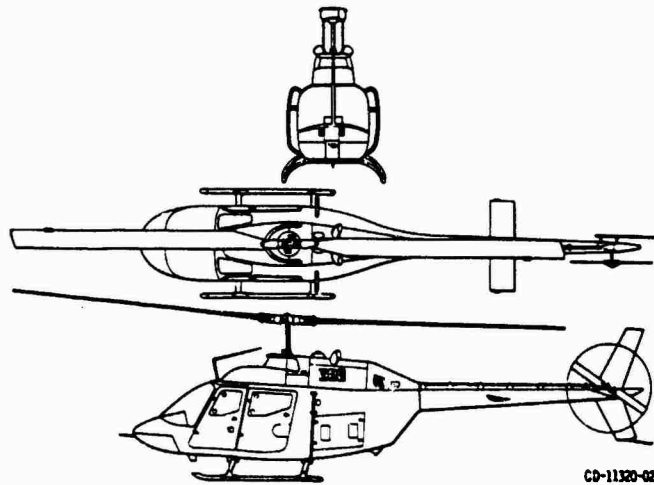
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TABLE I. - OIL OUTLET TEMPERATURES AS
A FUNCTION OF OIL COOLER BLOCKAGE WITH
VARYING SPEED AND OUTPUT TORQUE

[Ambient temperature, 85° F.]

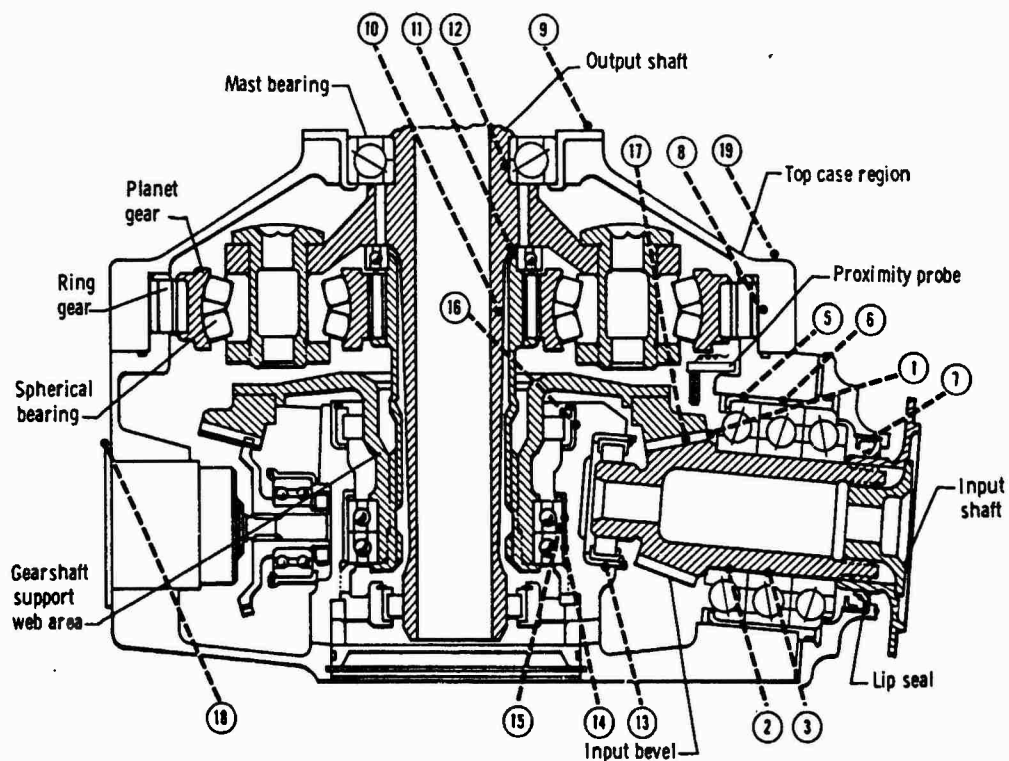
Speed, rpm	Output torque, in. -lb	Oil cooler blockage, % area	Oil-out temperature, °F
5580	48 000	70	204
	56 000	55	204
6200	2 500	100	173
	12 500	100	200
		80	174
	24 000	100	225
		90	200
		70	179
		50	173
		0	167
	36 000	100	244
		75	200
		50	186
		45	183
	48 000	100	244
		87	230
		50	199
		0	191
6820	56 000	90	246
		80	230
		40	210
		0	197
	24 000	75	199
	36 000	60	200
	48 000	50	200
	56 000	50	200

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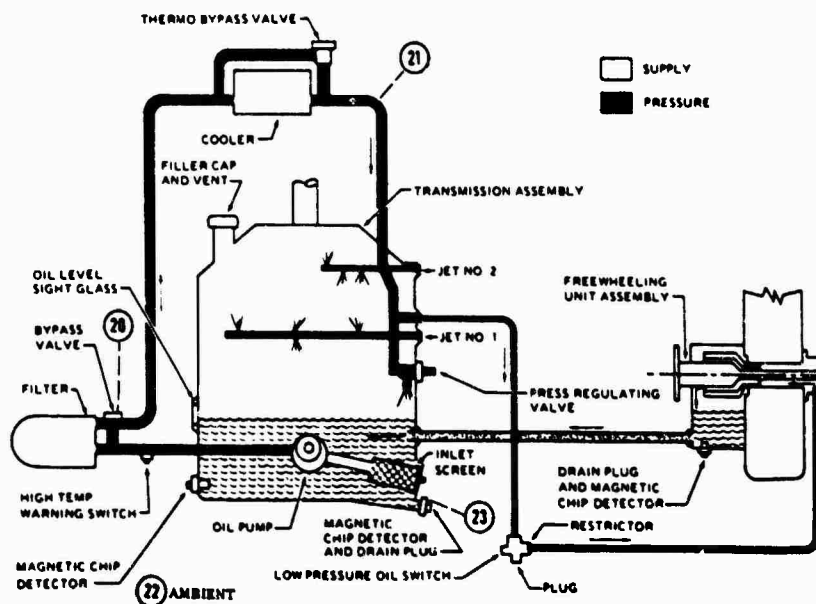


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Figure 1. - OH-58A helicopter.



(a) Cross section of transmission showing location of instrumentation.



(b) Transmission lubrication system.

Figure 2 - OH58A main transmission.

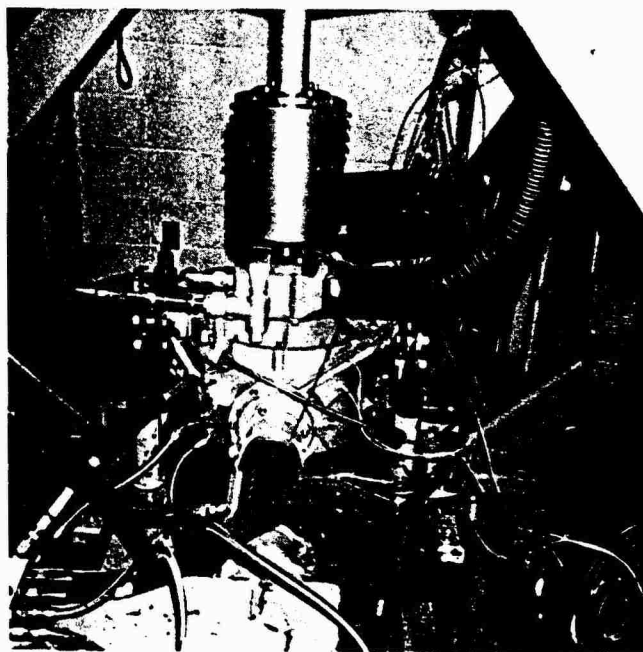


Figure 3. - Transmission installed in Corpus Christi Army Depot test stand.

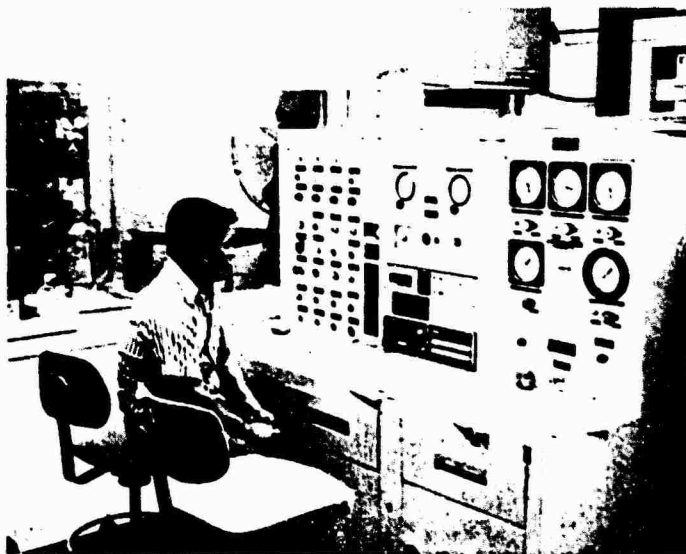


Figure 4. - Control room at Corpus Christi Army Depot transmission test facility.



Figure 5. - View of NASA instrumentation in transmission test control room.

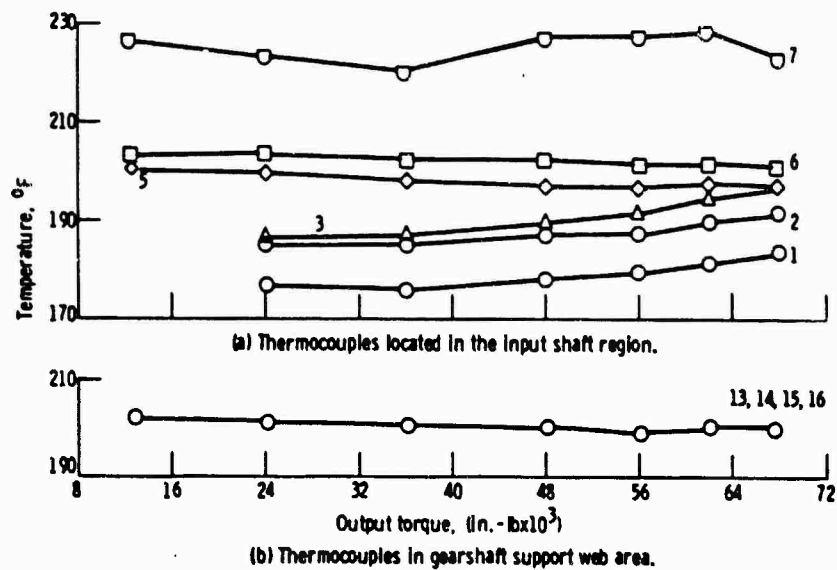


Figure 6. - OH-58 transmission stabilized temperatures as a function of output torque. Input speed, 6200 rpm; oil-out temperature, 200° F.

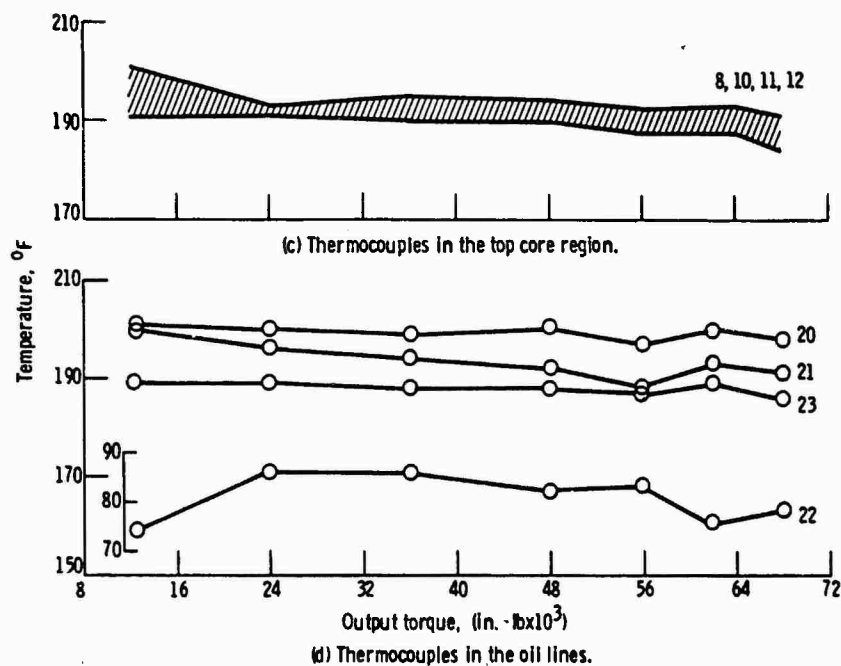


Figure 6. - Concluded.

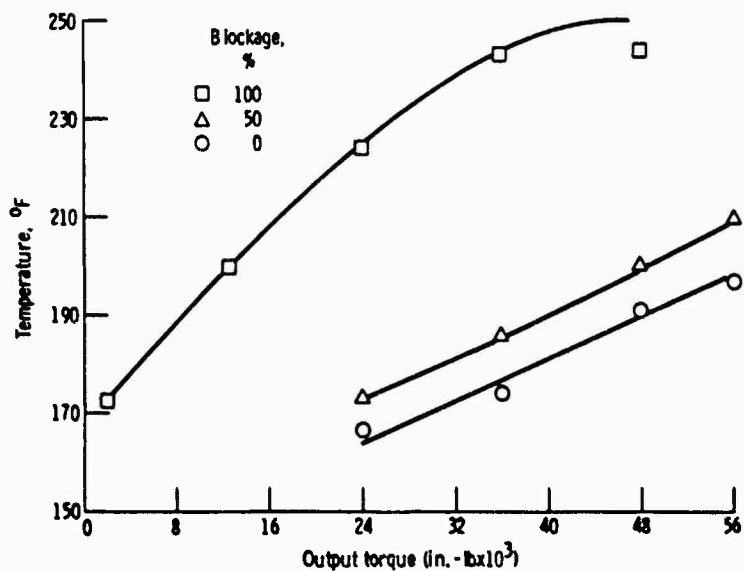


Figure 7. - Stabilized oil-out temperatures as a function of output torque for different amounts of oil cooler blockage. Input speed, 6200 rpm.

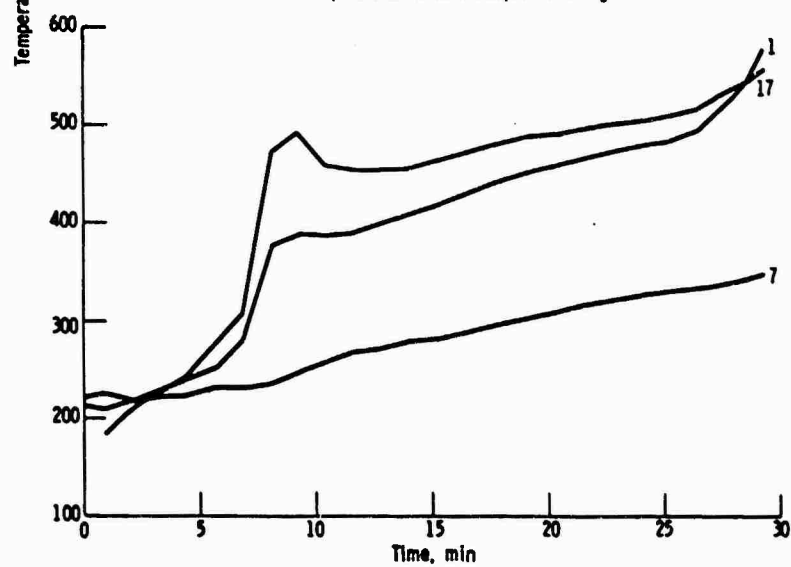
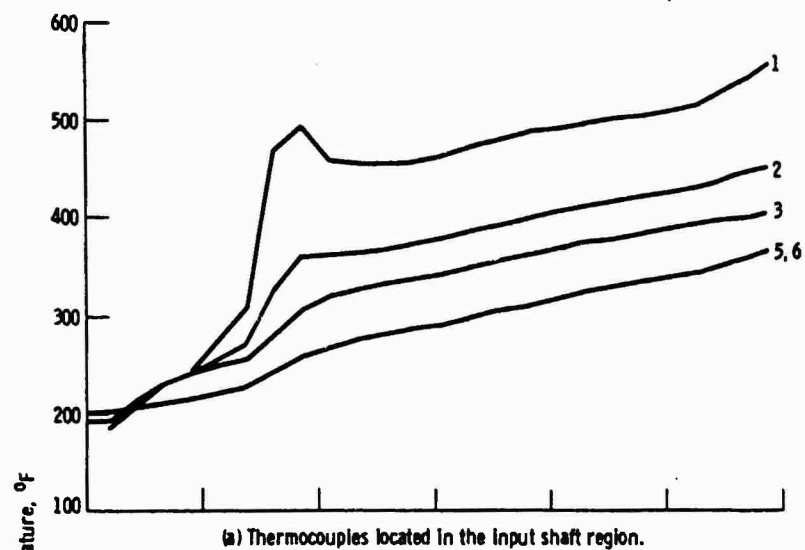


Figure 8. - Temperature as a function of time for survivability test of OH-58 transmission. Input speed, 6200 rpm; output torque, 36 000 in. - lb.

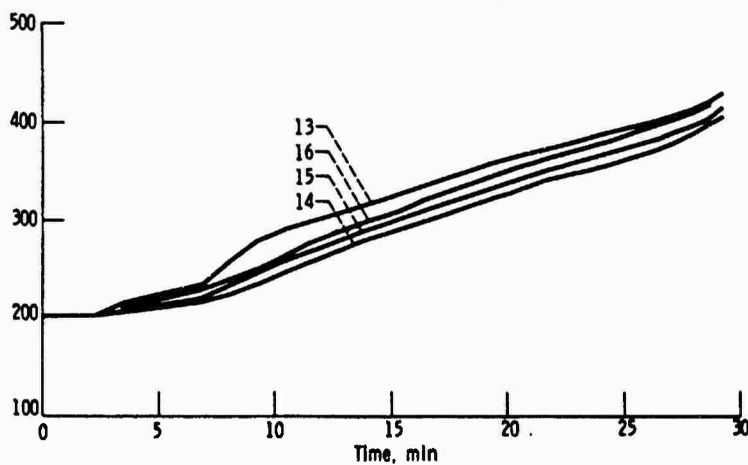
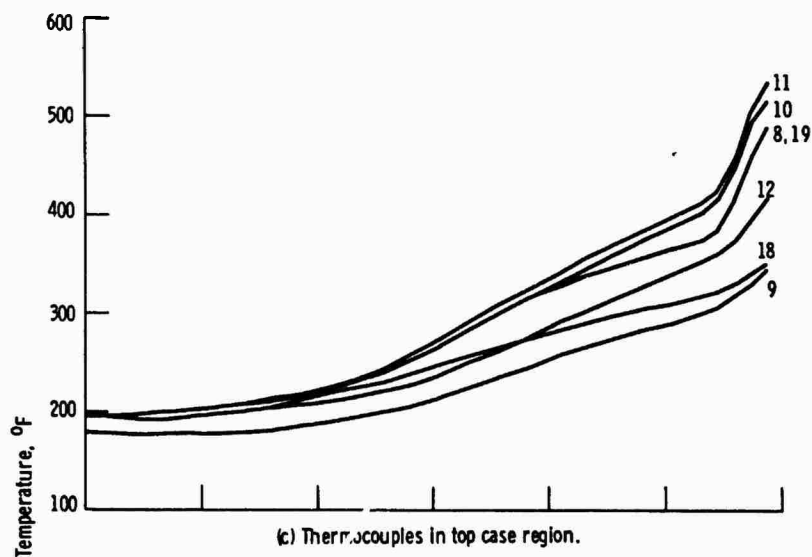


Figure 8. - Continued.

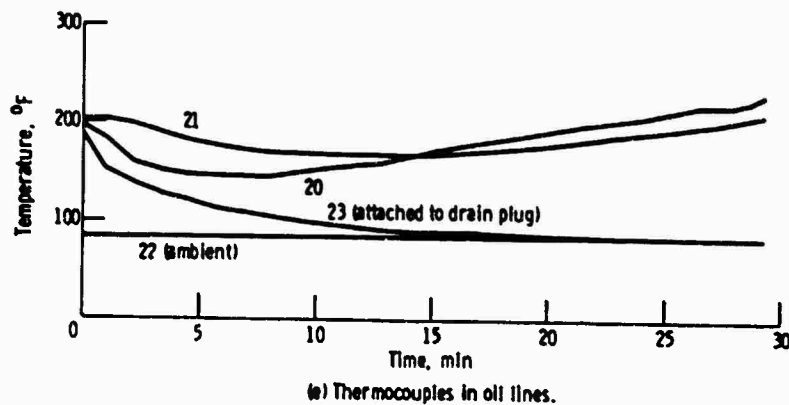
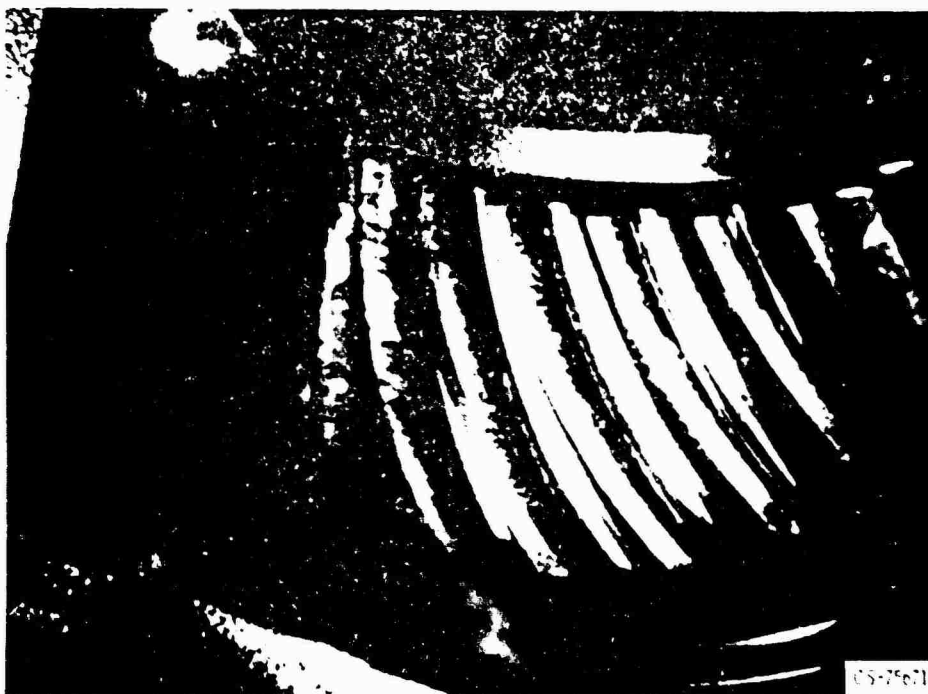


Figure 8. - Concluded.



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Figure 9. - Planet gear support bearing. Bearing cage has melted.



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Figure 10. - Inboard planet. Planetary gear teeth visible on gear teeth.



Figure 11. - Planet gear support bearing. Badly misaligned and skewed rollers caused high vibration and temperature levels.

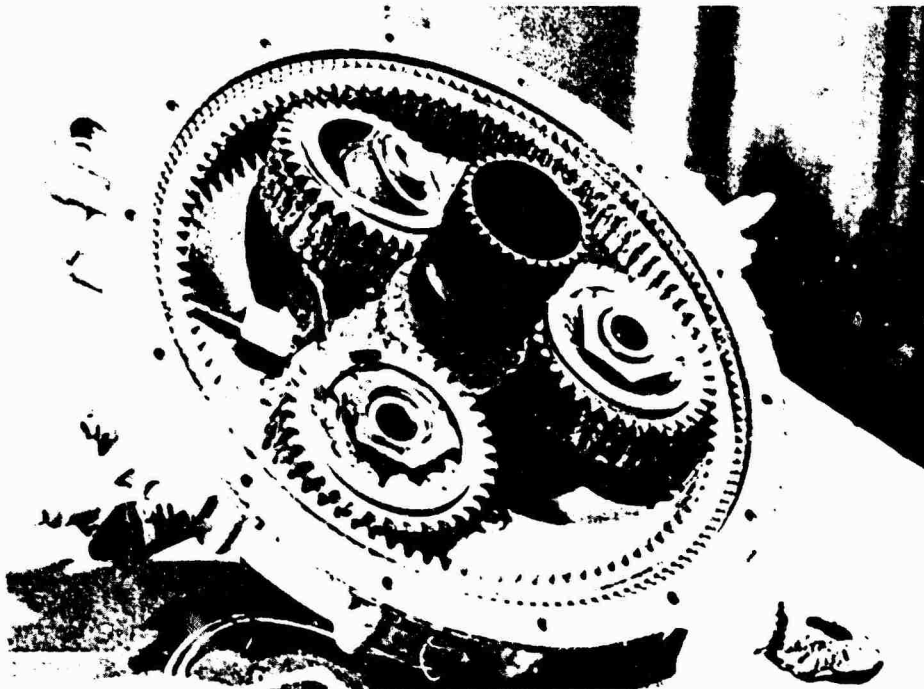


Figure 12(a). - Transmission top case with planet gear assembly. Planet lock nut (lower right) was machined to one-fourth original thickness.

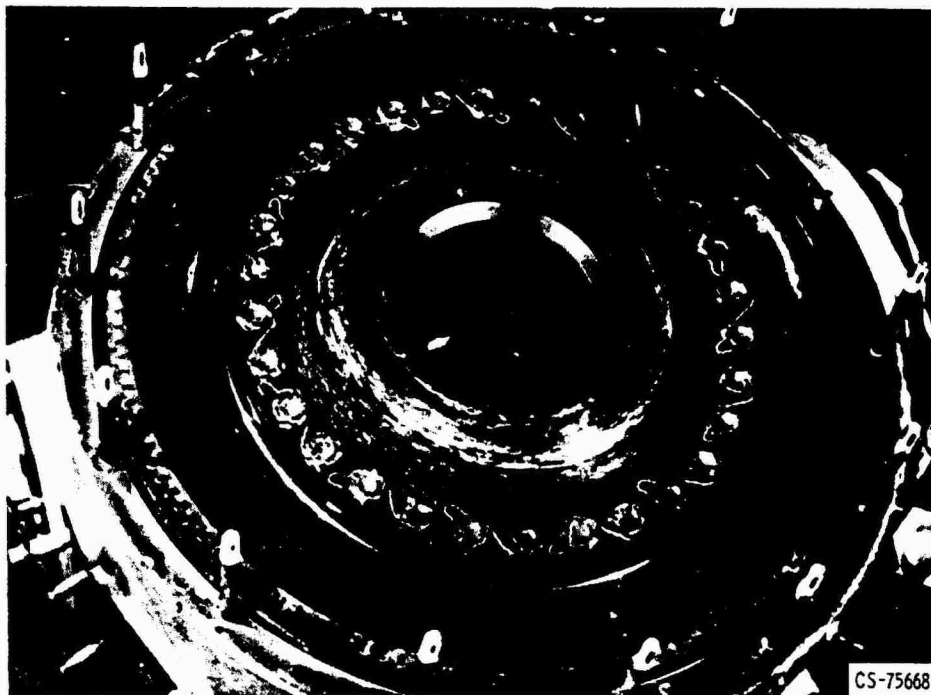


Figure 12(b). - Transmission lower case with bevel gear shaft and bevel gear mounting bolts showing. Scored area caused by scraping lock-nut is visible.

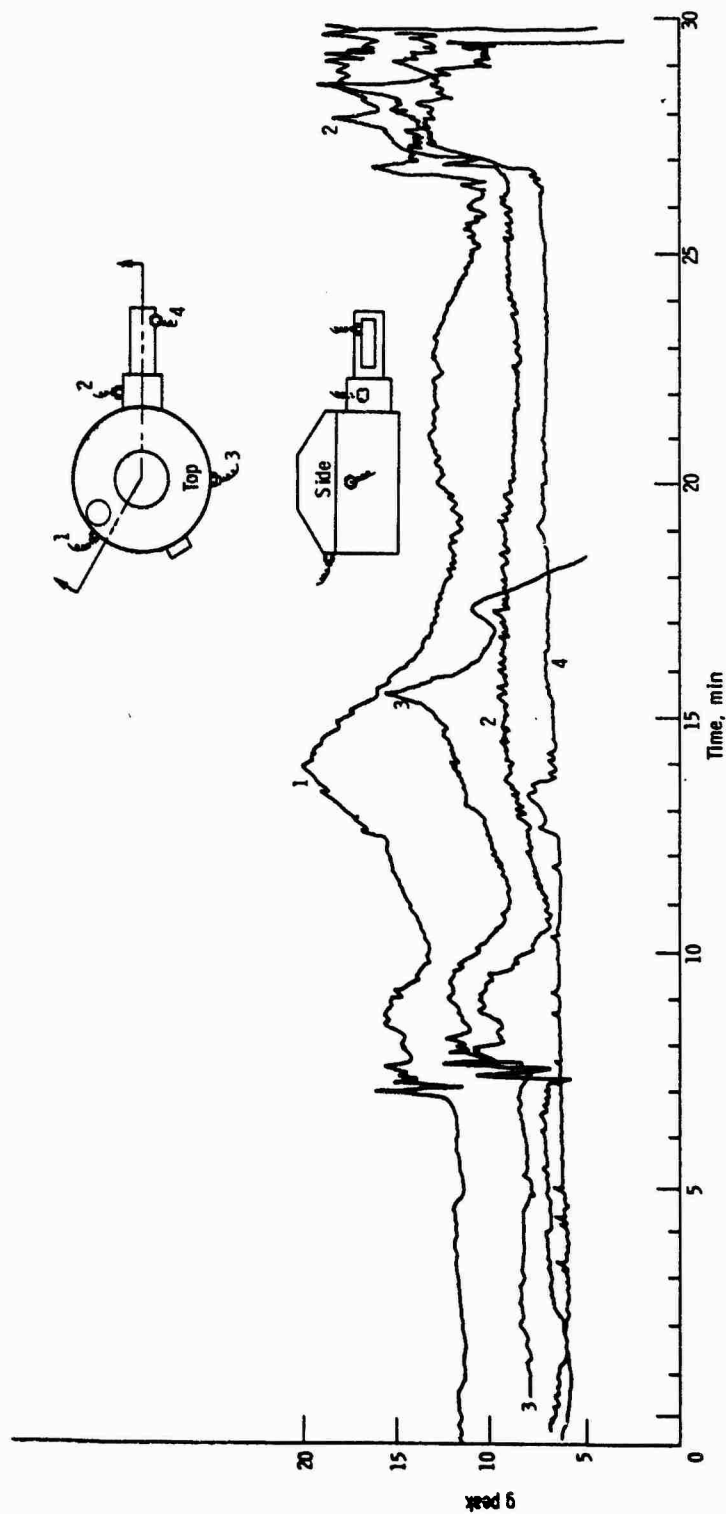


Figure 13. - Time history of acceleration measured on transmission during survivability test. Input speed, 6200 rpm; output torque, 36 000 in. -lb.

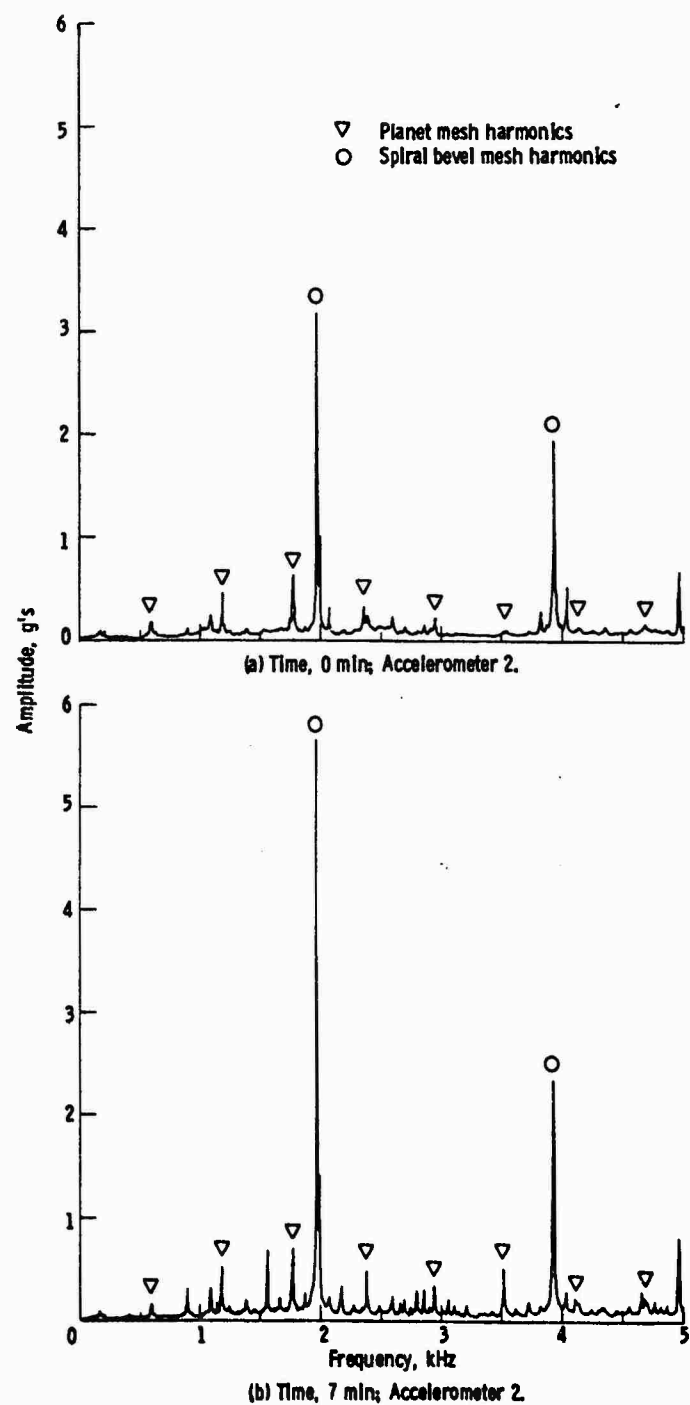


Figure 14. - Input pinion amplitude spectrum for survivability test. Input speed, 6200 rpm; output torque, 36 000 in.-lb.

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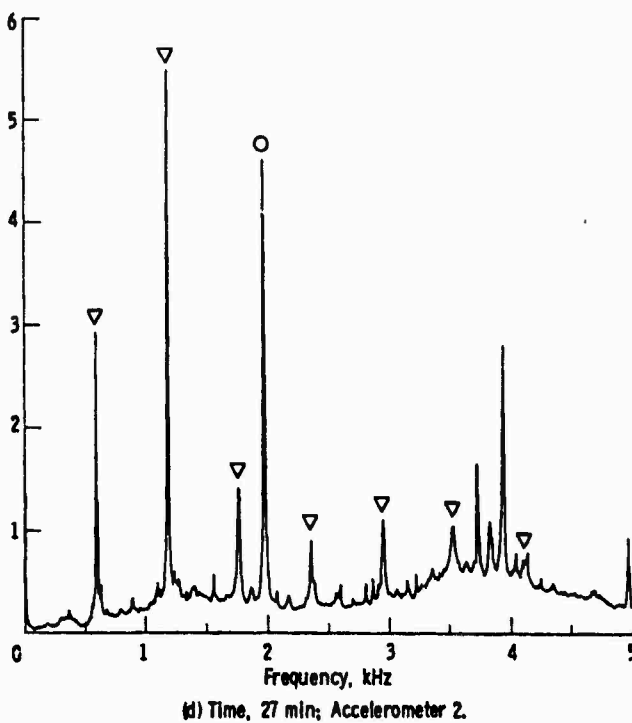
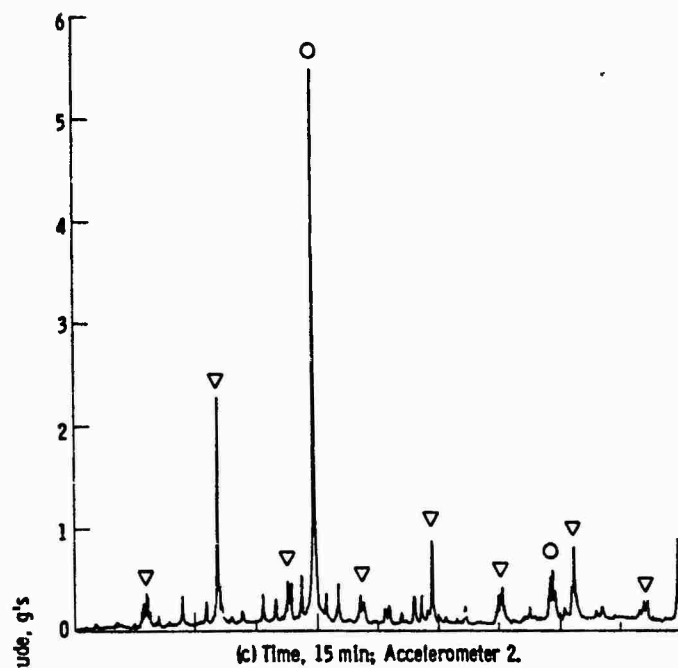


Figure 14. - Concluded

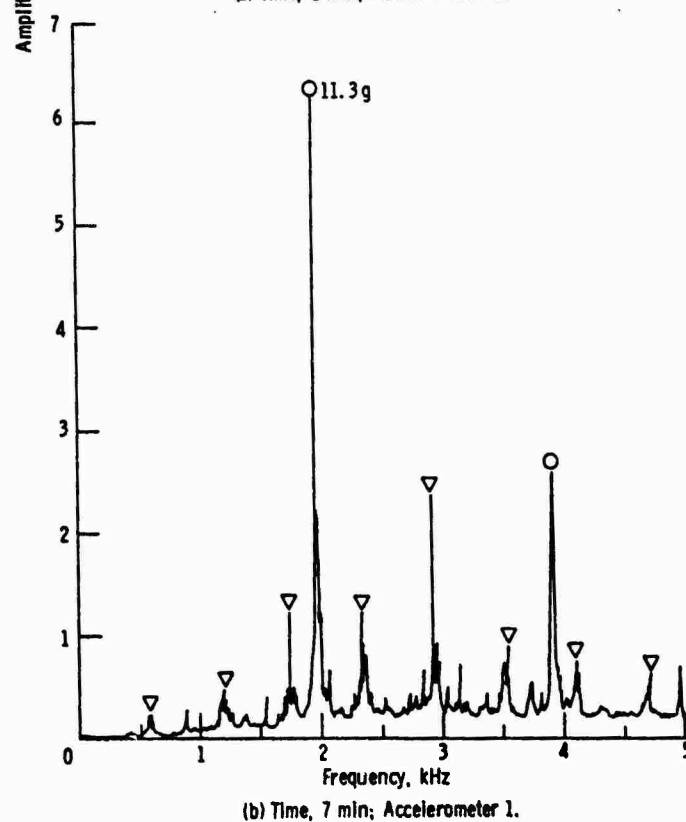
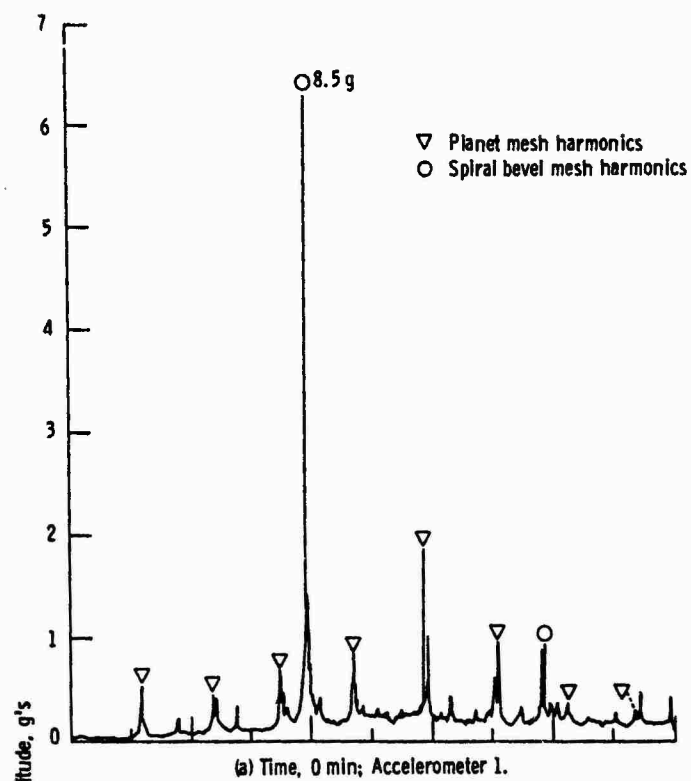


Figure 15. - Ring gear amplitude spectrum for survivability test. Input speed, 6200 rpm; output torque, 36 000 in.-lb.

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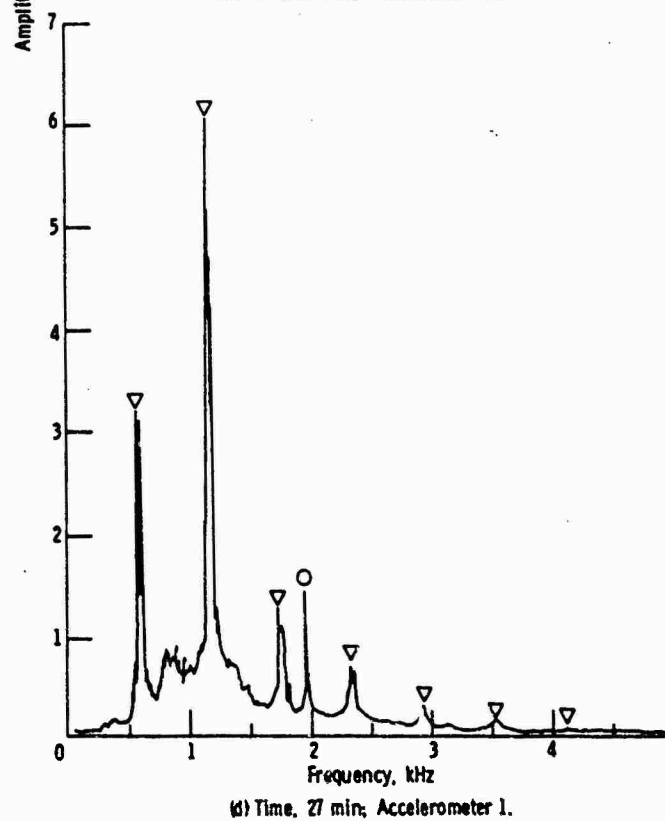
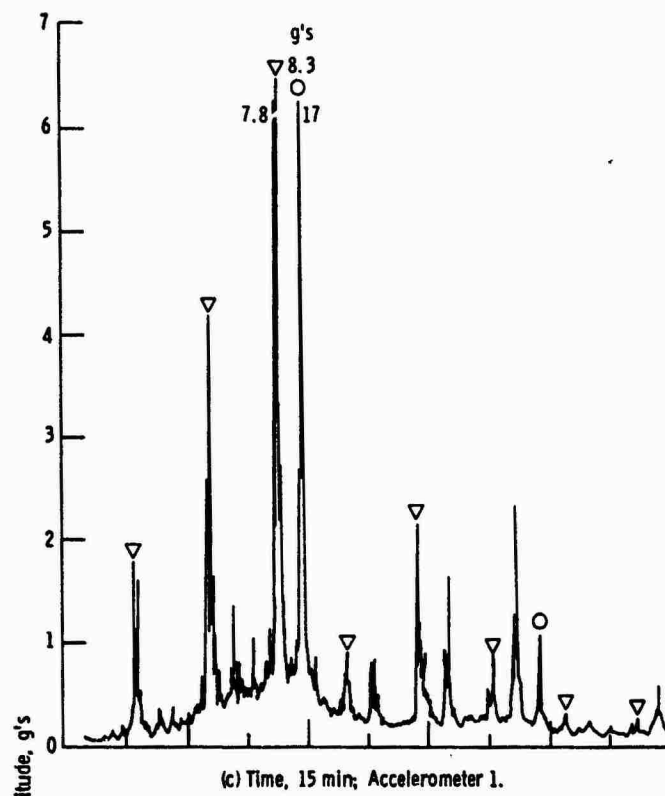


Figure 15. - Concluded.

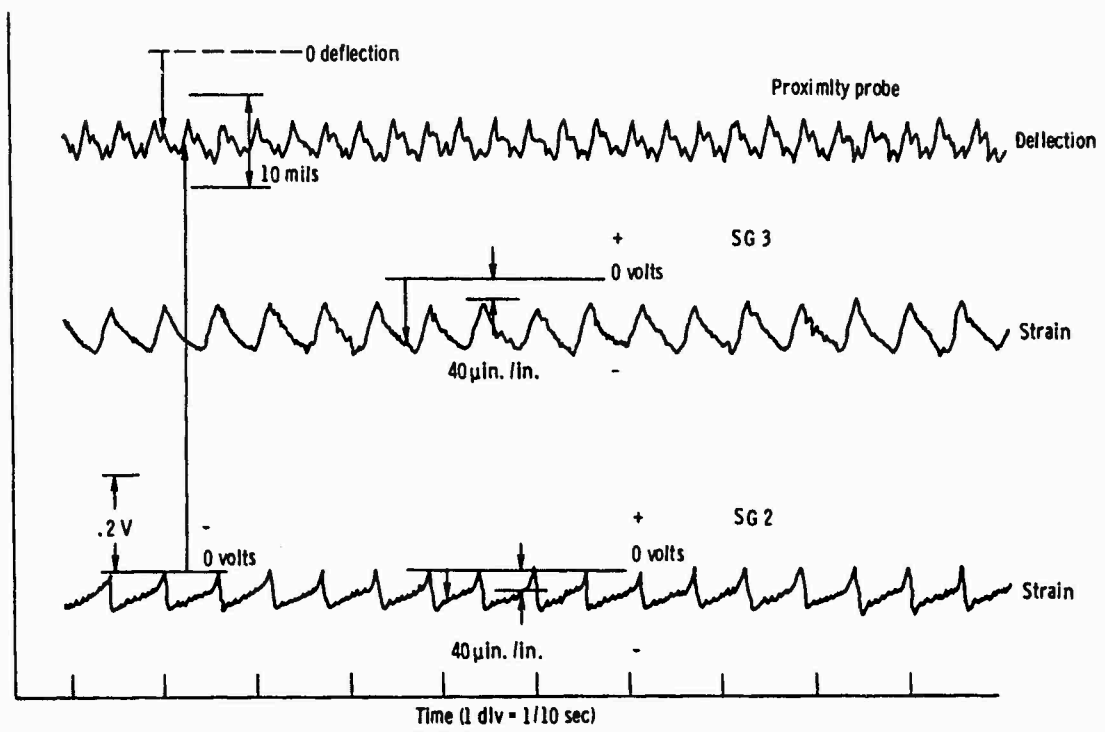


Figure 16. - Typical time traces played back from magnetic tape recording of bevel gear deflection and top case strains. Input speed, 6200 rpm; output torque, 56 000 in. - lb; oil-out temperature, 200° F.

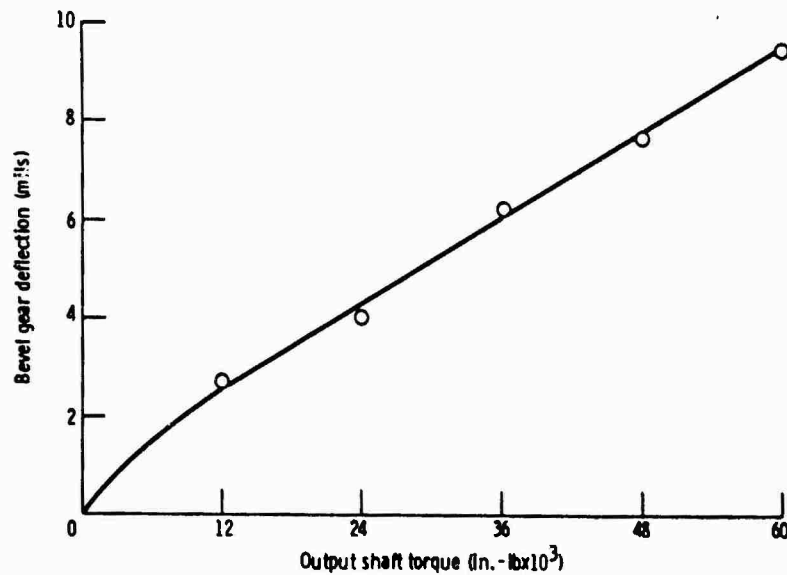


Figure 17. - Bevel gear deflection as a function of output shaft torque. Input speed, 6200 rpm.

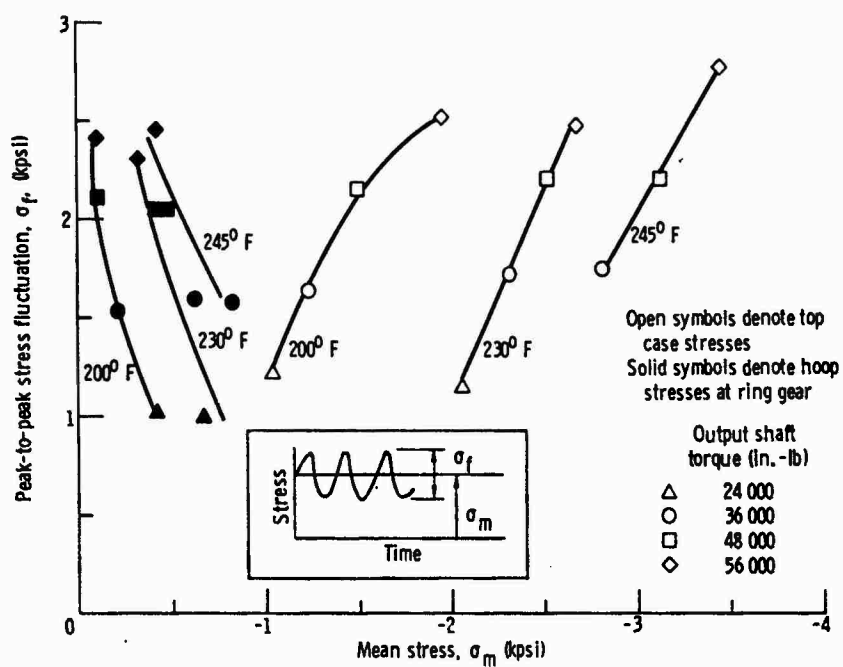


Figure 18. - Variation of measured stresses with output shaft torque for different oil out temperatures. Input speed, 6200 rpm; oil-out temperature, 200° F.

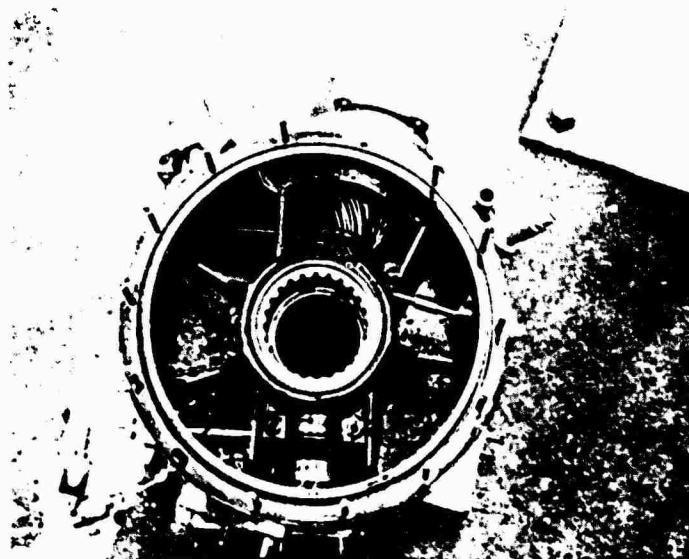


Figure 19. - Lower gear assembly after survivability test.