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ARMY MATERIEL SYSTEMS ANALYSIS ACTIVITY
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TECHNICAL REPORT NO. 420

M9 ACE TRANSMISSION OUTPUT SHAFT FAILURES

R. DAVIS MENTZER, JR.

APRIL 1986

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Five M9 Ace transmission output shafts failed during FOE conducted at Ft. Hood, TX, (Mar-Jun 85). These failures were attributed to reverse torsional fatigue caused by repeatedly shifting from reverse to forward before the vehicle could come to a complete stop. Through vehicle modifications and a shaft redesign, the fatigue life of the shaft was effectively increased to meet the durability goals of the M9 Ace. The increased fatigue life and adequacy of the fix was proved through verification testing conducted Oct-Nov 85. This report documents

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered) the failure analysis, assessment of fix, and the verification testing which was conducted to resolve this problem.

M9 ACE Transmission Output Shaft Failures
Executive Summary

During the Follow on Evaluation (FOE) of the M9 ACE conducted at Ft Hood, TX in 1985, five transmission output shafts failed. These occurred during competitive side by side trials with the D7 bulldozer. During these trials, the M9 was repeatedly shifted from reverse to forward without stopping the vehicle as required in the operator's manual. It was determined by contractor testing and confirmed by the Physical Test Division, Engineering Directorate, Combat Systems Test Activity (CSTA), that the reversed torsional loading applied by this type of operation caused fatigue failure of the shafts. Failure occurred at approximately 30 percent of the required life of the shaft.

The vehicle contractor, Pacific Car and Foundry Company (PACCAR), along with the transmission manufacturer, Clark Equipment, proposed modifications to eliminate the fatigue failures of the shafts. These modifications included:

- Redesigning the shaft to increase the fatigue life by
  - changing the shaft material,
  - increasing the root fillet radius of the splines, and
  - shot peening the shaft.
- Changing the steer linkage to prevent the M9 from being driven in the geared steer guide mode in reverse to reduce the torque spikes when shifting from reverse to forward.
- Modifying the gear shift guide to increase the time to shift from reverse to forward allowing engine speed and torsional loading to drop.

The Mobility Analysis Branch of AMSAA was called upon to assess these modifications and determine if any additional testing would be required. Using information gathered from FOE, testing and analysis conducted by PACCAR and Clark Equipment, and the failure analysis conducted by CSTA, the following conclusions were drawn.

- Changing the steer linkage to prevent geared steer in reverse is probably the single most significant factor in extending the life of the shaft. This reduces maximum reverse speed which effectively reduces the loading on the shaft, increasing the life to nearly 100 times the required life according to fatigue testing conducted by Clark Equipment.
- Quantifying the effects of each of the changes of the redesigned shaft individually is difficult, but combined, an increase of 5-10 times the original shaft life is expected.
- Modifying the gear shift guide has a minimal effect on increasing the shift time or reducing the shaft loading. The benefit of this modification is considered to be relatively insignificant in comparison with the other changes.

Since the failure involved a major drive train component, AMSAA felt that some type of verification was necessary, but because of the low risk of attaining a successful fix, suggested not to delay the production decision for additional operational testing. Instead, verification could be accomplished by retrofitting existing vehicles and testing prior to production or during the scheduled Initial Production Test (IPT). During the DA Program Review presented to the Army Systems Aquisition Review Council (ASARC) in September 1985, a decision was made to conduct a verification test according to AMSAA's recommendations but delay the contract decision until the fix had been proved. AMSAA was tasked with developing the Test Design Plan (TDP) in September 1985.

Verification Testing of the M9's transmisson output shaft was designed to simulate, under controlled conditions, the circumstances which produced failures during FOE. The purpose of the test was to demonstrate the adequacy of the fix, specifically verifying the increased fatigue life as projected by the contractor. Two vehicles were tested, using four drive shafts; two shafts of the original design were tested to failure and two shafts of the new design, including modifications to the vehicles, were tested to a minimum of five times the life of the original shafts. The vehicles were cycled from forward to reverse, without stopping, to simulate the effect of digging fighting positions as was done in FOE. Government personnel monitored the test which was conducted at the PACCAR facility in Renton, Washington throughout the October - November 1985 time frame.

The modified shafts displayed an increased fatigue life of over five times that of the original shafts during verification testing. In addition, a failure analysis, conducted on the two original shafts by the Materials Branch, CSTA, indicated that the type and location of the failures was almost identical to those which occurred during FOE. This indicates that although the test procedures differed, the verification test accurately simulated the conditions which produced failure during FOE. No other drive train failures were reported, although both vehicles had experienced prior testing. The failure was not transferred to any other component in the drive train.

The results of the verification test substantiate the original data generated by the contractor when evaluating the failure mode and determining the fix. The proposed modifications significantly increase the fatigue life of the shaft and are judged to be an adequate fix. The shaft is not considered abuse proof, however, and operating procedures should still require the vehicle to be brought to a complete stop when shifting from reverse to forward. No further testing is considered necessary.

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1. BACKGROUND/OVERVIEW

The M9 ACE is a full tracked, armored, high speed earthmover, designed to perform mobility, countermobility and survivability tasks in support of light or heavy forces. Known as the Universal Engineer Tractor (UET) prior to October 1980, the M9 underwent engineering and service testing during the 1960's. Due to reliability problems, testing continued through the 1970's including field evaluation at Ft Hood (1975) and testing by the US Army Test and Evaluation Command (TECOM) in 1976. Type classification standard A was approved by the Department of the Army (DA) in February 1977 with procurement funding becoming available in fiscal year 1982-83.

Between the time of type classification and funding, many design changes were made, one of which included drive train components. A production contract for fifteen newly configured M9's was let to PACCAR in 1982 with a multi-year production contract planned for award in September 1984.

During 1983 it became apparent that data from the Operational Test (OT) would not become available before the production contract award. Therefore, DA directed the IPT scheduled for Aberdeen Proving Ground (APG) (April - September 1984) to be expanded to include concurrent Force Development Testing and Experimentation (FDTE). Following this test, a FDTE phase two was scheduled to be conducted during the September 1984 - March 1985 time frame. In August 1984, the Vice Chief of Staff of the Army (VCSA) directed a side by side comparison of the M9 and the D7 bulldozer and halted ongoing procurement. This FOE, conducted at Ft Hood, Texas (March - June 1985), took the place of the scheduled FDTE phase two.

During FOE, five transmission output shafts failed. These failures occurred while digging fighting positions during competitive side by side trials with the D7. During these trials, the M9 was repeatedly shifted from reverse to forward without stopping the vehicle as required in the operator's manual. No failures of this type were reported during IPT at APG.

Through contractor testing and a failure analysis by the Materials Branch, Physical Test Division, CSTA, it was determined that the reversed torsional loading, applied by shifting on the move, caused fatigue failure of the shafts. The vehicle contractor, PACCAR, in coordination with the transmission manufacturer, Clark Equipment, proposed modifications to the vehicle and the transmission output shaft to increase the fatigue life. The Mobility Analysis Branch, Combat Support Division, AMSAA, was tasked with assessing these modifications and later developing a Test Design Plan (TDP) to verify the fix. This report documents the analysis and testing which was conducted.

2. FATIGUE THEORY

Prior to the review of any testing or analysis, an understanding of the phenomenon of fatigue is essential. Fatigue is generally thought of as the apparent deterioration of a specimen which has been subjected to repeated loading and unloading. Since 1850, it has been recognized that a material subjected to alternating loads will fail at a stress significantly lower than its ultimate or yield strength. Fatigue failures are commonly seen in machinery containing rotating parts which are subjected to a large number of cycles at high loads. It has been estimated in various studies that 90 percent of all mechanical failures in machinery are attributed to fatigue.

Many variables have an effect on the fatigue strength of a component, but initially three factors are necessary to produce failure. These include: (1) cyclic loadings with a large variation in magnitude, (2) a sufficiently high alternating stress caused by this loading, and (3) a high number of applications or cycles.

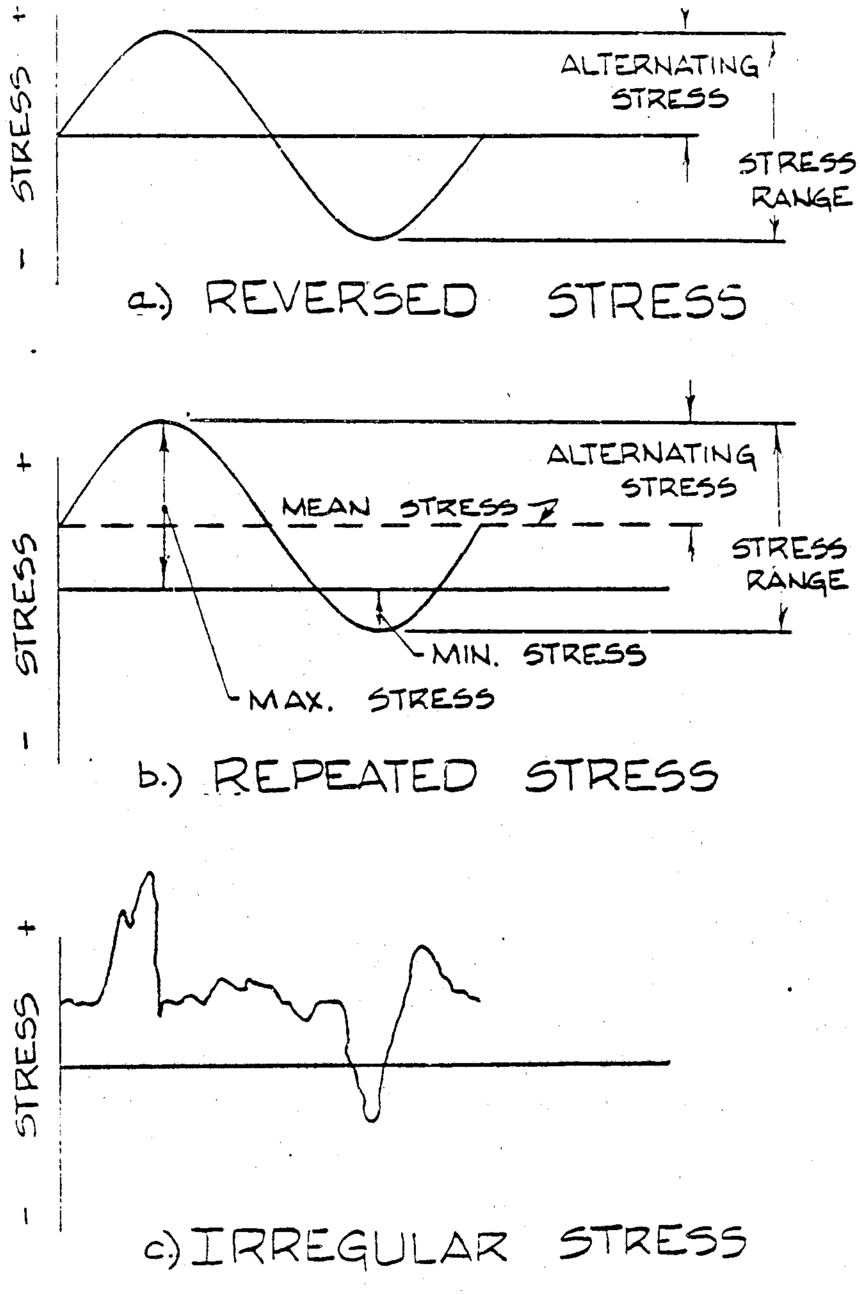
Cyclic loadings that lead to fatigue failures can be applied in a variety of ways. Figure 1 demonstrates typical fatigue stress cycles generated by various types of alternating loading. Figure 1A illustrates a sinusoidal, completely reversed cycle. This is an idealized situation but can be approached by rotating shafts, operating at a constant speed with no overloads. The magnitude of both the tensile and compressive stress for each cycle is equal producing a mean stress of zero. Most fatigue data available in literature is generated from this type of stress cycle.

A more common type of fatigue loading found in engineering practice is a situation where the maximum tensile and compressive stresses are not equal in magnitude, but alternate between constant values. This cycle, referred to as a repeated stress cycle, is illustrated in Figure 1B. This includes an alternating stress which varies around a mean or steady stress. As the mean stress becomes more positive, the allowable alternating stress decreases.

Probably the most common type of loading demonstrated in an operational environment is the irregular or random cycle (Figure 1C). This type of loading may exhibit several random fluctuations within one cycle. Estimating the fatigue life of a component can be extremely difficult when a component is exposed to random loadings.

The maximum load which occurs during any type of cycle usually produces a stress that when averaged over the entire cross sectional area is well below the yield strength of the material. However, very high localized stresses may be produced in the area of a stress concentration. Stress concentrations, which cause an irregular stress distribution across the cross sectional area, are due to abrupt changes in the geometry of a component. Holes, notches, small fillets, and even inclusions within the material are some of the more typical causes of stress concentration. The extremely high stresses produced in these small areas can exceed the yield strength of the material, causing localized plastic movement, resulting in the initiation of a microscopic crack. The crack propagates slightly with each additional loading cycle, reducing the cross sectional area until the load carrying capability of the component is exceeded. Failure then occurs suddenly without warning.

The failure often resembles a brittle type failure, with no deformation, yielding, or change of material away from the fracture. The fractured surface usually displays two distinct areas, a smooth region due to the rubbing action as the crack propagates, and a rough granular region where failure occurs in a ductile manner when the load carrying capability of the reduced cross section is exceeded. Most fatigue failures are due to cracking which is initiated on a free surface in the area of a high stress concentration.



TYPICAL FATIGUE STRESS CYCLES

FIGURE 1

The magnitude of the alternating stress which a specimen is subjected to has an enormous effect on the fatigue life. Increasing the magnitude of the alternating stress will decrease the number of cycles that a component can withstand before failure. At any given load, or stress, the component has a corresponding fatigue life of N cycles. This fatigue life (N) can be determined from a stress-cycle diagram commonly called a S-N diagram. This diagram is empirically generated, by testing several like specimens to failure, under a periodically varying or alternating load. Theoretically, the specimens should be identical to the component in question and subjected to loadings which duplicate the actual service conditions. However, fatigue testing is usually carried out under the fully reversed type loading previously described. Predictions of the fatigue life for fully reversed loading can be read directly from the diagram. The fatigue life for non-fully reversed loading can be estimated using the S-N diagram knowing the stresses produced in the component and the material properties. Care must be exercised when estimating fatigue life for non-fully reversed loading since actual stresses produced by stress concentrations are difficult to predict and fatigue testing usually shows considerable scatter (i.e., specimens fail at different times under identical conditions). The fatigue life in these cases is difficult to predict with extreme accuracy.

3. FAILURE ANALYSIS

Analysis was conducted by both the contractor and the government to determine the failure mode of the shafts. Two of the failed shafts were submitted to Clark Equipment, the manufacturer of the transmission, and two were submitted to the Materials Branch, Physical Test Division, CSTA.

3.1 Contractor Analysis.

A combined testing and analysis effort was conducted by Clark Equipment and PACCAR to determine the failure mode and develop a fix. The complete contractor analysis can be found in Appendix A. Testing included:

- Measuring torque in the transmission output shaft during various operational modes.

- Conducting fatigue testing of the old and newly modified shafts.

- Measuring torque in the shafts with modifications applied to the shaft and vehicle.

Results from these tests and additional analysis indicated that failure occurred due to fatigue, caused by the operators shifting from reverse to forward before stopping the vehicle. Torque spikes resulting from this type of operation along with torque levels produced by various operational modes can be seen in Table 1. Torque spikes were recorded when shifting from reverse to forward holding engine speed constant, in both steer modes, geared steer and clutch brake. These torques were measured for both the standard gear shift guide and a modified guide which was proposed to reduce loadings, by increasing shift time allowing engine speed to drop.

Table 1 shows that the maximum torque spikes occur when shifting from reverse two to forward two in the geared steer mode. By surveying M9 operators, it was found that this was the type of operation which occurred during FOE. Approximately 90 percent of the time the operators used geared steer when backing up and clutch brake when dozing. Shifting from geared steer to clutch brake occurred after shifting from reverse to forward on the forward move. Torque spikes of 7000 lbs-ft were recorded when shifting from reverse to forward in the geared steer mode, without applying the brakes, and holding engine speed constant at 2000 rpm. Torques higher than this can be expected if shifting occurs at maximum trottle, i.e. 2800 rpm. Table 1 also shows that typical bulldozing operations produce torques well below those generated by shifting on the move.

The required fatigue life of the shafts, and an estimation of the number of cycles which produced failure was calculated by PACCAR using data collected during FOE. According to the mission profile of the M9, 44.4 percent of the operational time is spent dozing, with the remainder consumed by travel, hauling, grading, scraping, swimming, and idle. During FOE, a total of 957 reversal cycles were counted in five hours while dozing tank traps. Using these numbers and a durability goal of 600 hours between major drive-train component failure, a total of 51,000 cycles was estimated for the required fatigue life of the shaft under conditions similar to FOE. During FOE, five vehicles were operated an average of 162 hours each before failure. This produced a maximum of 15,000 cycles on each shaft at the time of failure. This is approximately 30 percent of the required life of the shafts.

3.2 <u>Government Analysis</u>.

The analysis conducted by the Materials Branch, CSTA, confirmed that the failure mode was reversed torsional fatigue. Cyclic loading was attributed to reversing torsional loads which were produced as the vehicle changed direction without stopping. The complete lab report can be found in Appendix B.

Cracks initiated and propagated near the root fillet of each spline tooth with fracture occurring in the transverse plane of the shafts at spline runout. No other cracking or deformation was found away from the highly stressed end of the shafts.

Two additional shafts were submitted to the Materials Branch for evaluation which were taken from vehicles that had undergone IPT testing at APG. Although these shafts had not failed and no shaft failures were reported during IPT, the shafts were found to have longitudinal cracks in each fillet of the splines, extending through the case. These shafts were judged to be the same as the failed shafts in both material (AISI 8822) and hardness.

Recommendations offered by the Materials Branch to increase shaft life highlighted reducing the stress at the spline fillet. This could be accomplished by increasing the fillet radius of the splines and requiring the operators to bring the vehicle to a complete stop when shifting from reverse to forward to reduce loading.

BMY DEFENSE INDUSTRIES
A DIVISION OF HARSCO

M9 ACE TORQUE SPIKES

VEHICLE MOD	1400 RPM				2000 RPM			
	R2-F1 CB	R2-F1 GS	R2-F2 CB	R2-F2 GS	R2-F1 CB	R2-F1 GS	R2-F2 CB	R2-F2 GS
STD SHIFT	3200	5200	3800	5400	4600	6900	4900	7000
MOD SHIFT	3000	5000	3200	5200	4600	6900	4700	6800

R2 - REVERSE 2
F1 - FORWARD 1
F2 - FORWARD 2
CB - CLUTCH BRAKE
GS - GEARED STEER

OTHER TORQUE LEVELS

ENGINE MAXIMUM THEORETICAL OUTPUT	— 5200
TORQUE AT TRACK SLIP, 1ST GEAR	— 2500
TORQUE AT BRAKE SLIP, 1ST GEAR	— 3500
ENGINE AT 2000, SHIFT INTO 1ST GEAR	— 3700
MAXIMUM ACCELERATION, 2800 RPM 1ST THRU 4TH GEAR	— 2100
TYPICAL BULLDOZING	— 2900

ALL VALUES IN LB-FT MEASURED WITH 18000 LB BALLAST

TABLE 1

4. PROPOSED CONTRACTOR FIXES

The contractor proposed three modifications to the shaft and vehicle to increase the fatigue life of the drive shafts. These included:

- Shaft Redesign
  - Changing the shaft material from AISI 8822 H to AISI 8640 H to increase ultimate strength and fatigue life.
  - Increasing the root fillet of the splines from .008 to .031 inches to reduce the stress concentrations.
  - Shot peening the shaft to decrease the likelihood of surface cracks and their propagation.
- Modification of the Steer Linkage
  - Changing the steer linkage to prevent the M9 from being operated in the geared steer mode in reverse would reduce the torque spike when shifting from reverse to forward.
- Modification of Gear Shift Guide
  - Modifing the gear shift guide to preclude a straight shift motion when shifting from reverse to forward to increase the time required for shifting, allowing engine speed to drop. The new design requires a lateral motion around a protruding ear when shifting between reverse and forward.

Clark Equipment generated torsional fatigue S-N curves under fully reversed loading using the old and new shaft designs. From these curves (Figure 2), the fatigue life of each shaft can be estimated under various loadings. Using these curves, PACCAR estimated the fatigue life of the shaft in the modified M9 would be increased by over 100 times.

The cost impact of these modifications would be minimal. Approximately $100.00 per vehicle on a retrofit basis, replacing three minor parts, and no cost increase on future production.

Recommendations from PACCAR included retrofitting the existing fleet of vehicles with the improved shaft, modified shift guide, and modified geared steer link and approving these modifications for future production. No further testing was recommended.

5. ASSESSMENT OF FIX

The Mobility Analysis Branch, AMSAA, was called upon in August 1985 to assess the modifications proposed by the contractor. The AMSAA personnel visited PACCAR in Renton, Washington to review the analysis, testing and recommendations. The complete AMSAA assessment can be found in Appendix C.

The failure mode of reversed torsional fatigue as concluded by PACCAR and Clark was verified by the Physical Test Division, CSTA. This is important due to the fact that any fix should not transfer the failure to the next weakest link in the powertrain. Because of the phenomenon of fatigue, components fail at a significantly lower stress than the yield point of the material. In increasing the fatigue life, the same loading is transferring through the shaft, however it is able to withstand more applications, or cycles, at this load. Each component throughout the powertrain will experience the same type of loading both before and after the fix. If failure had occurred due to a single cycle overload, where the component was stressed beyond the yield or ultimate strength of the material, a fix would be required that would increase the load carrying capability of the shaft. This would result in higher loads being transferred throughout the powertrain, possibly transferring the failure to the next weakest component. This is not expected in this situation and is verified by FOE results, wherein testing continued after replacing the failed shafts, with no other fatigue type failure reported throughout the powertrain.

It is interesting to note that no failures occurred during the IPT conducted at APG. Test personnel reported that the vehicles were brought to a complete stop when shifting from reverse to forward. Even so, minute cracking was found in the splined area of the shaft, indicating the effects of fatigue were developing.

The modifications to the shaft and vehicle which were proposed by the contractor are in the right direction and the data indicate a significant increase in the fatigue life of the shafts should be obtained. The impact of these modifications is outlined below.

- Shaft Redesign

The net improvement of the several shaft changes can be determined using Clark S-N curves (Figure 2) obtained from testing the original and new shaft configurations. The FOE testing is estimated to have produced, at most, 15,000 cycles of stress reversal, before shaft failure (about 30 percent of the life required). While peak torques experienced during FOE would vary with engine speed at gear change, failure at about 15,000 cycles would indicate a repeated torque of about 6000 lbs-ft. In fact, an instrumented drive shaft in PACCAR testing experienced 5400 lbs-ft torque when shifted on the move from reverse to forward at 1400 rpm engine speed. A similar shift at 2000 rpm produces about 7000 lbs-ft in the shaft. Regardless of whether the peak torques are 6000 lbs-ft, as indicated from FOE or 7000 lbs-ft, the maximum from instrumented testing, the redesigned shaft would be expected to have a fatigue life that is 5 to 10 times greater than the original shaft life.

- Geared Steer Link Change

This simple change prevents geared steer in reverse, requiring the vehicle to remain in the clutch brake mode when backing up, which would normally be used in dozing. The effect is to reduce the maximum reverse speed and consequently the torque spike induced with an on-the-move shift from reverse to forward gear. The instrumented shaft tests indicate that the 7000 lbs-ft previously measured are reduced to about 5000 lbs-ft when operating in the clutch brake mode. At this torque, the S-N curves indicate the new shaft would have nearly 100 times the life required.

CLARK COMPONENT COMPANY

10 AUG 1985

VEHICLE: M9-ACE
TRANSMISSION: CLARK 13.5HR3610
SHAFT: TRANSMISSION OUTPUT SHAFT

TORSIONAL FATIGUE S-N CURVES (IN FULL REVERSE CYCLES)

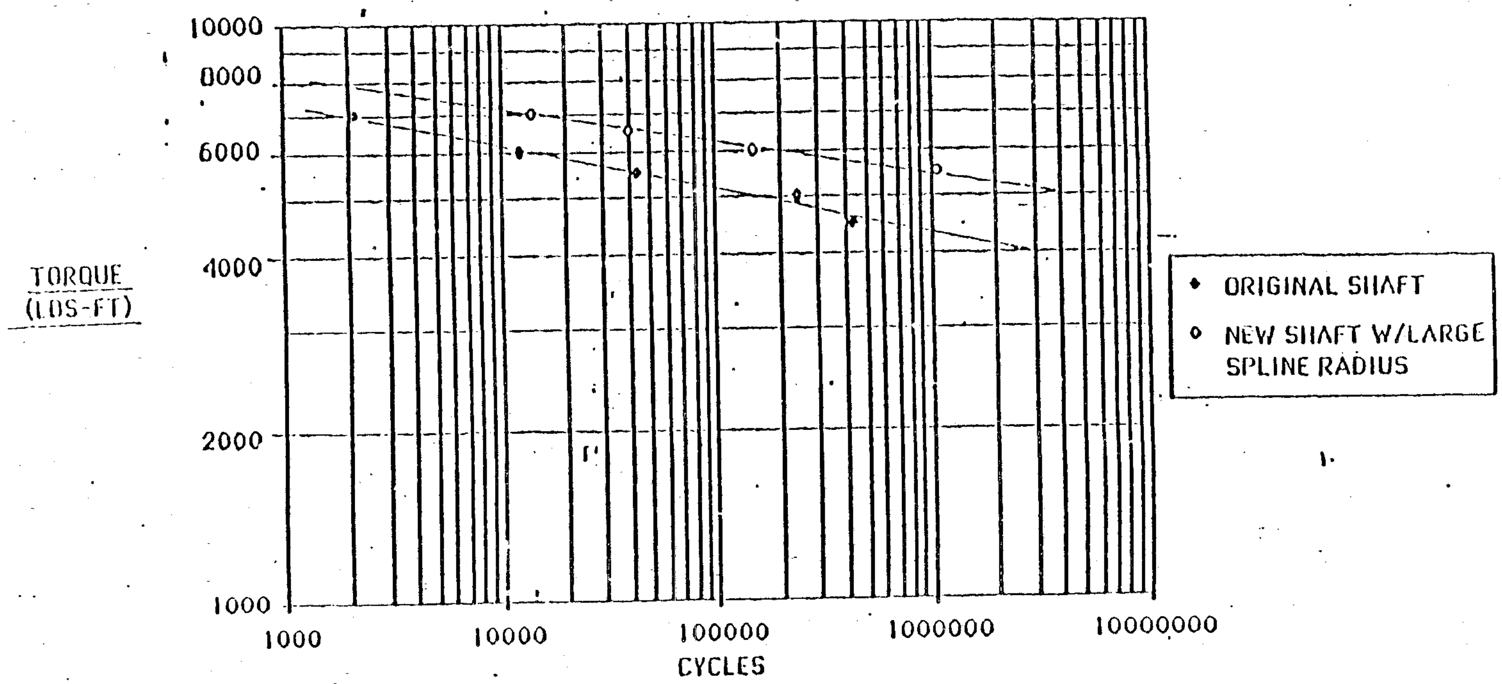


FIGURE 2

Calculations indicate that at an engine speed of 2000 rpm the speed in reverse will be reduced from 4.6 to 3.1 mph. This may produce a slight adverse effect on the productivity of the M9. The maximum impact of this change will be noticed in high stress dozing operations. Figure 3, Production Time Line, illustrates the minimum time required to doze a slot trench 90 feet in length. Acceleration time from 0 mph to the maximum reverse speed was calculated using tractive effort data and a soil strength of 120 RCI, dozing speed was assumed to be 2.0 mph, and shift times were assumed to be 0.5 seconds. These assumptions were substantiated by typical bulldozer performance and contractor testing. A comparison of cycle time was made using the geared steer mode, the preferred steer mode in reverse, and the clutch brake mode, the proposed steer mode to be used in reverse. An increased cycle time was estimated of not more than 10 percent, producing an equivalent decrease in productivity. At the maximum engine speed of 2800 RPM, maximum reverse speeds are reduced from 6.3 to 4.3 mph. This also produces a decrease in productivity of not more than 10 percent. This is a worst case estimate, which would only be expected during high stress operations. No productivity loss is anticipated during normal operations.

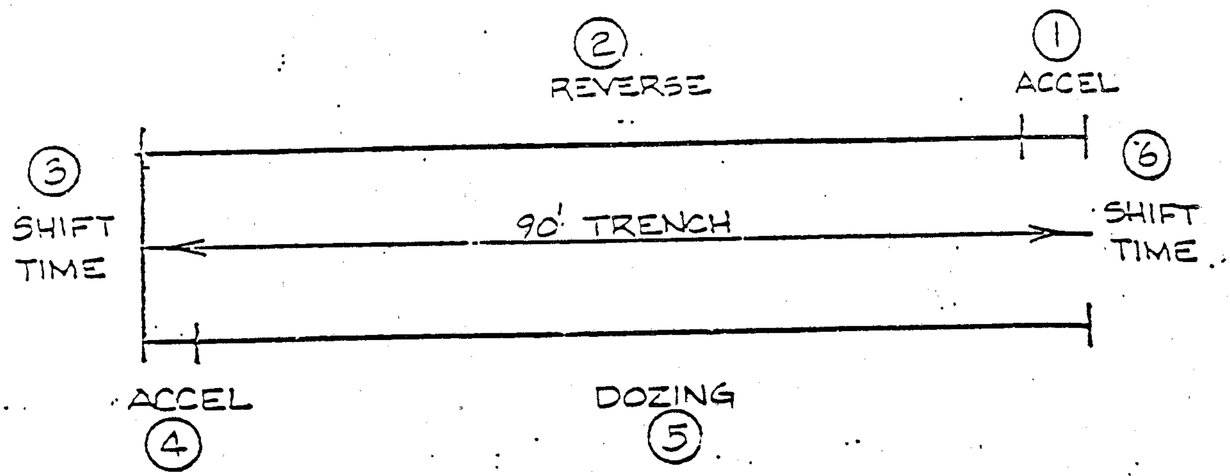
Because productivity is affected by many variables; operator experience, visibility, type of dozing, soil condition, etc; testing for losses of this magnitude would be difficult if not impossible. Operator experience and soil conditions each can account for up to 40 percent of the production of a tracked type dozer. It is therefore concluded that the risk of significant productivity loss is negligible.

- Gear Shift Guide Modification

This change is intended to increase the time required to shift between forward and reverse gear positions, and to increase awareness that a direction change is involved. However measured shift times and shaft torques with the original and modified shift guides showed very small differences. The effect of this change is therefore judged to be minimal and probably unnecessary in view of the effectiveness of the other changes.

The conclusion AMSAA drew from reviewing the available data is that the risk of a modified M9 not meeting a 600 durability goal under conditions experienced in FOE testing is virtually negligible. The fix significantly increases the fatigue life of the shaft and should not transfer the failure to another component in the drivetrain. However, as with any change to a critical component, some type of validation of the fix is required. This can be accomplished by either retrofitting an old vehicle prior to production or in IPT, but validation should not be required for a production decision.

Even with these modifications the new shafts are not abuse proof. Premature failure could still occur if the operator frequently shifted from reverse to forward at maximum engine speed (2800 rpm). Normal operating procedure should continue to require a stop before shifting from reverse to forward.



DOZING DIRECTION →

	CLUTCH BRAKE (REV)			GEARED STEER (REV)		
	SPEED (MPH)	TIME (SEC)	DIST. (FT)	SPEED (MPH)	TIME (SEC)	DIST FT
① ACCELERATE	3.1	1.1	3.1	4.6	2.9	11.6
② REVERSE	3.1	18.5	86.9	4.6	11.6	78.4
③ SHIFT	–	.5	–	–	.5	–
④ ACCELERATE	2.0	1.0	1.5	2.0	1.0	1.5
⑤ DOZING	2.0	30.2	88.5	2.0	30.2	88.5
⑥ SHIFT	–	.5	–	–	.5	–

* MIN. CYCLE TIME - CLUTCH BRAKE - 51.8 SEC
GEARED STEER - 46.7 SEC
DIFFERENCE < 10%

PRODUCTION TIME LINE

FIGURE 3

6. VERIFICATION TESTING

6.1 Test Outline.

In September 1985, during the DA Program Review presented to the ASARC for the M9 ACE, a decision was made to conduct a verification test according to AMSAA's recommendations. AMSAA was tasked with preparing the Test Design Plan (TDP). (Appendix D)

The purpose of the test was to demonstrate the adequacy of the recommended fix, specifically verifying the increased fatigue life of the shaft as projected by the contractor. The test was designed to simulate, under controlled conditions, the situation which caused the fatigue failures during FOE, namely, to duplicate the effect on the M9 when digging fighting positions competing with the D7 bulldozer. This was accomplished by cycling the vehicle from a reverse to forward motion without reducing engine speed, as was done during FOE. Dozing was not included due to the relatively low torsional spikes which are generated in the shaft as compared to shifting on the move.

The original test plan called for four drive shafts to be tested using two vehicles, two shafts of the old (FOE) design, and two shafts of the new design with the modifications applied to both the shafts and the vehicles. During testing a fifth, intermediate design shaft was tested. This shaft had several but not all of the modifications the contractor proposed.

During all testing, the vehicles maintained a constant engine speed of 2000 rpm before shifting from reverse two to forward two. Shifting was conducted as rapidly as possible with no reduction in engine speed. The TDP called for the vehicle to come to a complete stop before shifting from forward to reverse; however, during testing of the first original shaft this was changed to incorporate a shift on the move (2000 rpm) maneuver when changing both forward and reverse directions. A survey of the drivers prior to verification testing indicated that the vehicles were brought to a complete stop when shifting from forward to reverse. This was later determined to be not what actually occurred during FOE.

The unmodified vehicle containing the original shafts was shifted from reverse two to forward two in the geared steer mode. After shifting from reverse to forward, and on the forward move, the vehicle was shifted to clutch brake steer mode, the preferred dozing mode. Cycling continued until shaft failure, with the drivers rotated as necessary. Upon the first failure, the intermediate design shaft was mistakenly installed in the vehicle instead of the second original shaft. This shaft was tested in an identical manner and eventually replaced with the second original design shaft which was tested until failure.

The vehicle with the modifications and modified shaft, was tested similarly to the unmodified vehicle, shifting on the move. The modified geared steer link insured operation in the clutch brake steer mode in both the forward and reverse directions when shifting. Testing of this shaft continued for a minimum of five times the number of cycles required to produce failure in the original shaft. The second modified shaft was then tested in an identical manner.

Data recorded during testing included drive shaft torque, shift position, engine speed and a count of the cycles accomplished. A continuous record was maintained on magnetic tape with periodic monitoring accomplished using an oscillograph printout of all the measured values.

6.2 Test Results.

Verification testing of the M9 ACE transmission output shaft was completed in November 1985. Complete testing procedures and results are documented in the PACCAR Report M9 Armored Combat Earthmover Drive Shaft Report dated January 1986. Test results are summarized in Table 2.

Testing on the first shafts, both original and modified, included 6489 cycles of one way loading as called for in the original Test Design Plan. During this period, the vehicle was brought to a complete stop when shifting from forward 2 to reverse 2 and shifted from reverse 2 to forward 2 at an engine speed of 2000 rpm. This produced average one way torsional loadings or spikes of approximately 6800 lbs-ft for the original shaft and approximately 5000 lbs-ft for the modified shaft with no load reversals. The remainder of the cycles, shown in Table 2, were completed under reversed loading with the vehicle being shifted at an engine speed of 2000 rpm in both the forward and reverse directions. For the original shafts, the vehicle was operated in the geared steer mode when in reverse and clutch brake when moving forward. This produced average loads of approximately 6500-6700 lbs-ft when shifting from reverse to forward and -5000 lbs-ft when shifting from forward to reverse. In addition, shifting from geared steer to clutch brake after shifting from reverse to forward, produced a secondary torque spike of approximately -5000 lbs-ft. Under this type of reversed loading, the original shafts 1 and 2 withstood 2019 and 2136 cycles, respectively, before failing.

The modified design insured operation in the clutch brake mode, when operating in both forward and reverse. This effectively reduced the loading to approximately 5000 lbs-ft when shifting from reverse to forward and -4400 lbs-ft when shifting from forward to reverse. Also, since the vehicle was continuously operated in the clutch brake mode, the secondary torque spike was eliminated. Testing on the modified shafts was halted before failure because the test criterion to demonstrate five times the life of the original shafts was exceeded. The modified shafts 1 and 2 were subjected to 36051 and 11000 cycles respectively of reversed torsional loading.

Also tested during the verification test was an intermediate shaft. This shaft was mistakenly placed in the unmodified vehicle upon failure of the first original shaft. This shaft was tested with an average reversed loading of approximately 7300 lbs-ft to -4800 lbs-ft for 11388 cycles. It was removed from the vehicle prior to failure.

The total number of cycles accumulated on both vehicles were 33,032 and 42,450, respectively. Both original shafts, the intermediate shaft, and the second modified shaft were tested using the first vehicle. To reduce down time, the entire transmission with the shaft installed was replaced with the second original shaft. This transmission was used until failure of the shaft (2136 cycles) while the original transmission was operated throughout 30896 cycles. The first modified shaft was the only shaft tested in the second vehicle.

TABLE 2
TEST RESULTS

SHAFT DESCRIPTION	TOTAL CYCLES	CYCLES REVERSED TORSIONAL LOADING	AVERAGE LOADING REV/FWD (LB-FT)	AVERAGE LOADING FWD/REV (LB-FT)	AVERAGE LOADING GEARED/CLUTCH STEER/ BRAKE (LB-FT)	COMMENTS
1st ORIG	8,508	2,019	6768	-4839	-5470	FAILURE
2nd ORIG	2,136	2,136	6544	-5117	-5128	FAILURE
INTERMEDIATE	11,388	11,388	7320	-4791	-5225	NO FAILURE
1st MOD	42,540	36,051	4953	-4327	0	NO FAILURE
2nd MOD	11,000	11,000	5124	-4472	0	NO FAILURE

No major drive train failures were reported during testing even though the vehicles had been exposed to prior testing. Minor problems which occurred included replacement of the shifter bearing and the gear shift lever. The gear shift lever experienced significant wear due to the protruding ear on the modified gear shift guide. After completion of the test, a tear down of the transmission, steer unit, and final drive assembly was conducted on both vehicles. A thorough inspection revealed cracking of the reaction member of the torque converter and galling, or wearing, of various components within the final drive assembly.

6.3 Failure Analysis.

Upon completion of the verification test, three drive shafts were submitted to the Materials Branch, CSTA for a failure analysis. These included the two failed original shafts along with the intermediate shaft. The analysis was conducted to determine if the failure mode was similar to that which occurred during FOE and to determine how the intermediate shaft differed from the two original shafts. This analysis was similar to the one conducted on the FOE shafts.

The analysis concluded that the fracture features were nearly identical to those of the shafts that failed during FOE. Cracking initiated near the fillet of each spline tooth and propagated longitudinally and radially, identical to the FOE failures. The material properties of the failed shafts indicated that they were identical to the original FOE shafts. (AISI 8822)

A chemical analysis of the intermediate shaft revealed that it was constructed of a different alloy (AISI 8640). This is the material which was proposed for the new modified shafts. This effectively increased the case depth and the core hardness of the shaft. Although the shaft had been reported to have been shot peened, it was not possible to ascertain if this process had actually been performed. There was no significant change in the root radius of the splined region when compared to the original shafts.

The intermediate shaft did display a significant improvement in the fatigue life in comparison to the old design. No indications of cracking or any surface defects were seen from a magnetic particle inspection of the shaft.

6.4 Discussion.

The failures achieved during verification testing duplicated the failures in FOE. All failures were determined to be caused by reversed torsional fatigue, with the fractures occurring in the same location on each shaft, at spline runout. This indicates that the same type of loading was experienced in both the verification test and FOE.

Although the loadings were of the same type during both tests, they were more severe during the verification test because they were consistently repeated at the maximum level. Failure occurred within 2100 cycles during verification testing while an estimated 15,000 cycles were necessary to produce failure during FOE. This was probably due to the controlled nature of the verification test. Each shift was conducted at a constant engine speed, producing the consistant loading necessary to compare the new shafts to the old shafts. The FOE was not controlled in this fashion, shifting was conducted at random engine speeds, producing various loadings. Although not all of the torsional spikes during FOE were of the magnitude of verification testing, it is obvious that enough were experienced to cause failure.

Testing at these consistently higher loads during the verification test is thought to have produced somewhat conservative test results. In examining the S-N curve, it can be seen that the difference in fatigue life between the old and new shaft grows smaller as the loading is increased. If testing at high loads demonstrates an increase in fatigue life, as was demonstrated during verification testing, an increase of at least this much if not more, would be seen at lower loadings. Regardless of the magnitude of the load, the new shaft displayed a relative increase in fatigue life of over five times of the old shaft, without failure.

During testing of the first original shaft, the test procedure was changed to incorporate a shift on the move maneuver in both the forward and reverse directions. Two factors led to this change in procedure; a) it was determined that the original test plan did not accurately duplicate the conditions experienced in FOE, and b) the test time could be reduced drastically by shifting on the move in both directions.

A survey of the operators taken before the TDP was developed indicated that they had brought the M9 to a complete stop when shifting from forward to reverse. Even on quick shifts, it was felt that dozing resistance would slow the vehicle to a stop before shifting could be completed. Early in the verification test, it was determined that this was not what actually occurred during FOE. In further discussion with the operators, at leastone reported that a shift on the move type maneuver was used when shifting from forward to reverse to dislodge dirt from the blade of the M9. A review of the Materials Branch Failure Analysis of the FOE shafts showed cracking initiated on both sides of the spline root fillets, indicating that a significant amount of torque was applied in both directions. Therefore, it was concluded that shifting on the move in both forward and reverse would more closely simulate the effects of FOE testing.

The effect of shifting on the move in both forward and reverse produced a nearly fully reversed type of loading. This type of loading was expected to drastically reduce the test time required to fail the original shafts. Accurately predicting the fatigue life to determine actually how much test time could be saved, proved to be a difficult task. Variables which can lead to an inaccurate prediction include: a) the irregular stress distribution generated due to stress concentrations at the spline fillets; b) a large scatter in fatigue data; and c) a complex loading cycle which resembles a fully reversed type of cycle but approaches a random type of loading. Although accurately predicting the fatigue life expected during the test would be difficult if not impossible, an estimation of shaft life was calculated using several assumptions. This produced a relative basis between the old and new test procedure to be used as a comparison.

The assumptions used to calculate the fatigue life included:

a. The shaft was treated as a solid round shaft when calculating the torsional stresses. The stress concentration due to the splines were neglected.

b. The loadings were applied in a sinusoidal cycle. Figure 4 illustrates the difference between a sinusoidal cycle and the cycles which were actually produced during testing both before and after the test plan was changed. It is apparent that the actual test data approach a random type cycle.

The scatter in the fatigue data (S-N curve) is inherent in any type of fatigue testing. Although this can be minimized by closely controlled testing, it cannot be eliminated due to the statistical nature of the material.

The fatigue life of a component exposed to non-reversed loading can be estimated using the S-N curve and a Goodman Diagram. The Goodman Diagram (Figure 5) illustrates the effect of the mean stress on the alternating stress levels and the stress range. It can also be used to predict an equivalent fully reversed loading which produces the same effects as a non-reversed loading. The estimated fatigue life can be read directly from the S-N curve using this equivalent fully reversed loading.

Knowing the stresses generated and the material properties of the component, a Goodman Diagram can be constructed. The stress range is plotted on the ordinate and the mean stress is plotted on the abscissa. A line drawn at 45 degrees passes through the zero point of the diagram and out to the ultimate strength of the material. The maximum and minimum stress for each cycle is then plotted at the corresponding mean stress. Because of the relationship between the stress range and the alternating stress, these points will fall at an equal distance above and below the mean stress line drawn at 45 degrees. Lines then projected from the ultimate strength, through these points and back to the ordinate will reveal the equivalent, fully reversed range of stress. Since the S-N curve is plotted as a function of load instead of stress, the corresponding load which produces this stress must be calculated. The fatigue life can then be read directly from the S-N curve.

The same results can be obtained by solving for σ_e in the equation which describes the Goodman Diagram.

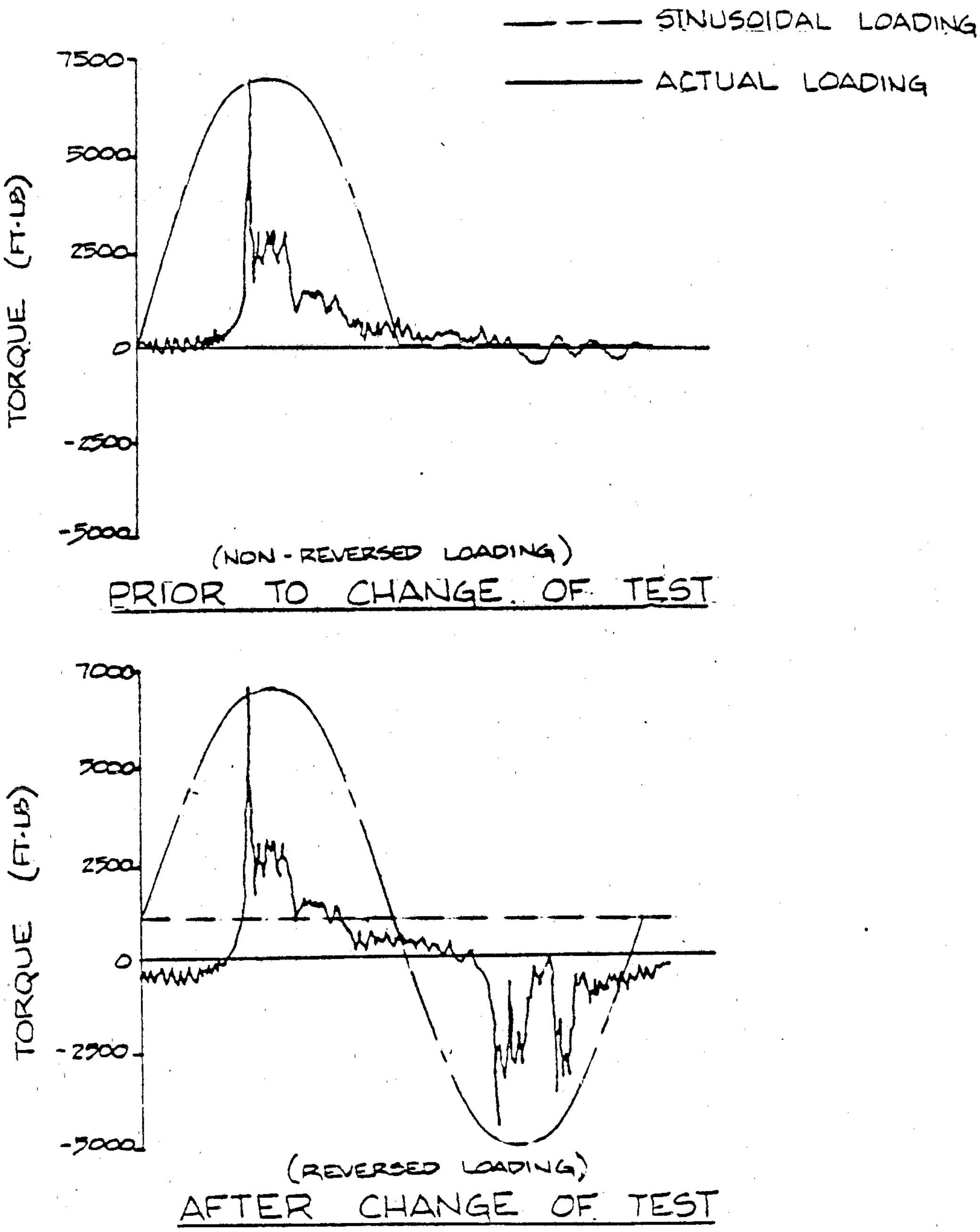
$$\sigma_a = \sigma_e \left[1 - \left(\frac{\sigma_m}{\sigma_u}\right)\right]$$

where: $_a$ = alternating stresses

σ_e = fatigue limit for completely reversed loading
σ_m = mean stress
σ_u = ultimate shear stress (torsional loading)

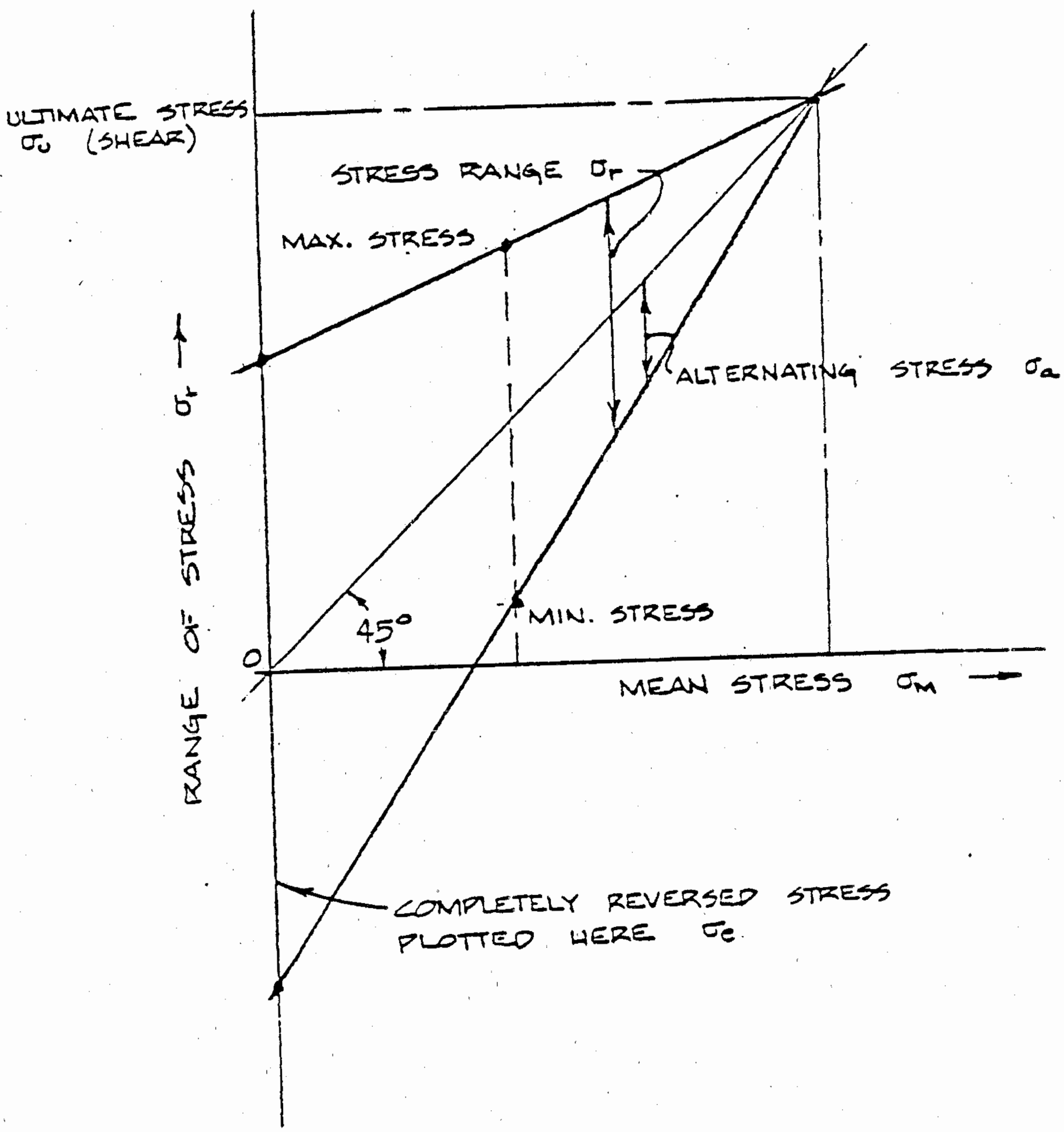
Both this equation and the Diagram can be solved using either stresses or the corresponding loads. Since the fatigue data generated by Clark Equipment is in terms of the applied loading, these are more easily handled using loads in lieu of stresses. Using this methodology an estimate of the number of cycles which the shafts could withstand was calculated for shifting on the move in reverse only and shifting on the move in forward and reverse.

Results of this analysis indicated that if the test procedure was left unchanged, the number of cycles the original shaft could withstand could exceed 60,000. In comparison, by changing the test to incorporate a shift on



TYPICAL LOADING CYCLES

FIGURE 4



GOODMAN DIAGRAM

FIGURE 5

the move maneuver in both directions, test time could be reduced to approximately 3000 cycles. In view of these estimates and the conclusion that on the move shifts in both directions were indeed made during FOE, the test procedure was changed.

Changing the test plan after the start of the test raised an obvious question - what effect did the 6489 cycles which were conducted under the original test plan have on the overall results of the test? It was analytically predicted and verified by the test that these cycles had very little impact on the fatigue life of the shafts.

The first 6489 cycles produced a different alternating stress and stress range than that which was produced after the test was changed. When more than one stress level is present over the life of the component, it becomes necessary to combine the overall effects of the various stresses to estimate the life. The total life can be determined by adding the percentage of lives consumed at each stress level. If n_1 and n_2 represent the number of cycles a specimen is subjected to at different stress levels, and N_1 and N_2 are the respective fatigue life of the specimen at this corresponding stress, then:

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} = 1$$

This equation, known as the cumulative damage rule or Miner's rule, can be used to assess the impact of various loadings on the overall fatigue life of the shaft. The actual test results showed that the life consumed by the original 6489 cycles was small compared to the life under the fully reversed loading test. The first original shaft which experienced the 6489 cycles, failed at 2019 cycles of reversed loading. The second original shaft which only experienced reversed loading failed at 2136 cycles. Using Miner's Equation the effect of the original loading was estimated to be $\frac{6489}{118,465}$.

This is approximately 5.4 percent of the total life of the shaft. The 6489 cycles of nonreversed loading were considered to have very little impact on the overall test, therefore 2136 cycles of reversed loading was used as a baseline for the modified shafts. However, to relieve any concerns, the first modified shaft was tested for 42,540 cycles (including the 6489 non-reversed cycles) which is five times the total number of cycles on the first original shaft.

The results obtained from the intermediate shaft provide a great deal of insight as to the effectiveness of the fix. This shaft was tested in place of an original shaft, in an unmodified vehicle. This produced higher loadings than the modified shafts would be exposed to due to the absence of the modified geared steer link. Also, the intermediate shaft did not have the increased fillet radius to reduce the stress concentration in the splined area. These are the two most prominent features in increasing the fatigue life of the shaft. Regardless, the intermediate shaft endured over five times the baseline life of the original shafts with no indication of cracking or fatigue.

Since only one intermediate shaft was tested, and that inadvertently, it should not be considered a replacement for the proposed fix. All of the proposed modifications are still deemed necessary and are required. The fully modified shafts are expected to have a fatigue life much greater than the intermediate shaft at no additional cost.

The cracking of the reaction member of the torque converter and the galling of components within the final drive assembly may be attributed to the excessive amount of consistently high torsional loads encountered during the verification test. However, these problems are not expected in operation within durability goal of the vehicle for the following reasons:

- It is not clear how much of this was induced from testing prior to the verification test.
- Two vehicles were used to simulate the life of five shafts. These vehicles experienced significantly more high torque applications than would be expected during the normal life of the vehicle.
- The fix actually reduces the magnitude of the torsional spikes. Maximum torsional loadings on the modified vehicle will be lower than those experienced during FOE, where no indication of this problem was seen.

7. CONCLUSION

The following conclusions were drawn from this study:

a. Failure of the transmission output shaft was caused by reversed torsional fatigue due to the operator shifting from reverse to forward and vice versa before bringing the vehicle to a complete stop.

b. The fix reduces the maximum reverse speed of the M9, slightly decreasing productivity (less than 10 percent). This will be most apparent in highly stressed situations with very little or no deterioration in productivity expected under normal operating conditions. In comparison to other factors which influence productivity, ie; operator experience, visibility, soil conditions, etc, this degradation is considered to be insignificant.

c. The modifications to the M9 ACE and the transmission output shaft are predicted to increase the fatigue life of the shaft by more than 100 times. During verification testing the shafts were tested in excess of five times the fatigue life without failure. Durability goals will be satisfied by this increased life.

d. The proposed modifications do not transfer failure to any other drive train components.

e. The modified design is not considered abuse proof. Shifting on the move at maximum engine speed (2800 rpm) may still produce premature failures. The vehicle should continue to be operated according to the operator's manual, i.e. bring the vehicle to a complete stop before shifting from reverse to forward or forward to reverse under normal operations.

f. Verification testing simulated the conditions which caused initial failure during FOE. The failure type and location were identical to the original failures.

g. Adequacy of the transmission shaft fix has been sufficiently demonstrated. No further operational testing is considered necessary.

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APPENDIX A

ORIGINAL CONTRACTOR EVALUATION

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SUMMARY SHEET
M9 ACE DRIVE SHAFT SOLUTION

DESCRIPTION OF FAILURE

DURING THE M9 ACE FOLLOW-ON-EVALUATION (FOE), FIVE TRANSMISSION OUTPUT DRIVE SHAFTS FAILED IN FATIGUE. THE FAILURE WAS CAUSED BY OPERATORS SHIFTING FROM REVERSE TO FORWARD WITHOUT FIRST STOPPING THE VEHICLE AS REQUIRED IN THE MANUAL.

DESCRIPTION OF FIX

IMPROVE THE FATIGUE STRENGTH OF THE SHAFT AND AT THE SAME TIME REDUCE THE MAGNITUDE OF TORQUE SPIKES INDUCED IN THE SHAFT, BY INCORPORATING:

1. A SHAFT WITH HIGHER FATIGUE STRENGTH
2. A MODIFIED GEAR SHIFT GUIDE (INCREASE TIME TO REVERSE),
3. A MODIFIED GEARED STEER LINK (PREVENT GEARED STEER IN REVERSE)

TESTING

- MEASURED TORQUE IN THE SHAFT AT VARIOUS OPERATIONAL MODES.
- CONDUCTED FATIGUE TESTING OF OLD AND NEW SHAFTS.
- MEASURED TORQUE IN THE SHAFT WITH MODIFICATIONS APPLIED.

FINDINGS

- MAXIMUM INDUCED TORQUE IN THE SHAFT OCCURS AS THE M9 IS QUICKLY SHIFTED TO FORWARD (WITHOUT REDUCING POWER) AFTER MOVING REARWARD IN GEARED STEER AT MAXIMUM SPEED. 7000 LB FT.
- IMPROVED SHAFTS HAVE 6 TIMES THE LIFE OF THE OLD SHAFTS
- THE MODIFIED GEARED STEER LINK AND SHIFT GUIDE REDUCE THE MAXIMUM INDUCED TORQUE TO 4700 LB FT.
- MAXIMUM NUMBER OF CYCLES AT THE WORST CASE IS 57,000.

RISK ASSESSMENT

CHANGES TO THE M9 ARE MINOR. THE RISK OF INDUCING ANOTHER METHOD OF FAILURE IS MINIMAL. THE RISK THAT THE CHANGES DO NOT "FIX" THE PROBLEM IS LOW.

1. SHAFT LIFE 6 TIMES GREATER.
2. MAXIMUM INDUCED TORQUE REDUCED BY 33%.
3. TOTAL SHAFT LIFE (CONSIDERING REDUCED TORQUE) INCREASED FROM 2100 CYCLES TO OVER 1,000,000 CYCLES.
4. INDUCED TORQUE SPIKES ON ALL OTHER COMPONENTS REDUCED 33%.

COST IMPACT

- ON FUTURE PRODUCTION - NO CHANGE.
- ON RETROFIT BASIS - 3 MINOR PARTS ARE REPLACED (LESS THAN $100)

CONCLUSION

THE DESCRIBED FIX WILL IMPROVE THE LIFE OF THE TRANSMISSION SHAFT FROM 2100 CYCLES TO OVER 1,000,000 CYCLES. (WORST CASE IS 57,000 CYCLES). SHAFT LIFE HAS BEEN PROVEN BY TEST. ANY FURTHER TESTING IS OF MARGINAL BENEFIT.

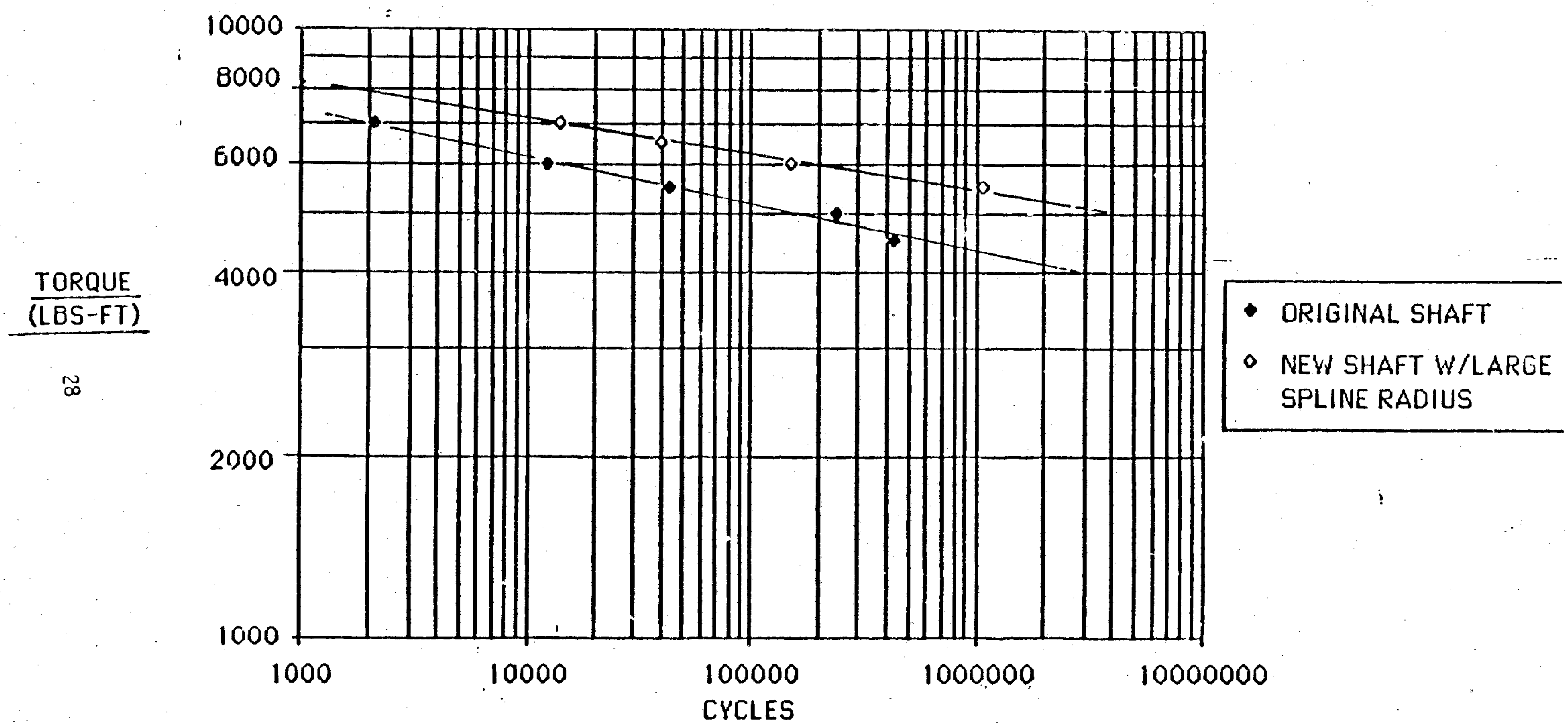
RECOMMENDATIONS.

- RETROFIT THE EXISTING FLEET WITH AN IMPROVED SHAFT, A MODIFIED SHIFT GUIDE AND A MODIFIED GEARED STEER LINK.
- APPROVE ECP PCF J9390 FOR ALL FUTURE PRODUCTION.
- NO FURTHER VEHICLE TESTING IS RECOMMENDED.
- PROCEED WITH PRODUCTION.

16 AUG 1985

VEHICLE: M9-ACE
TRANSMISSION: CLARK 13.5HR3610
SHAFT: TRANSMISSION OUTPUT SHAFT

TORSIONAL FATIGUE S-N CURVES (IN FULL REVERSE CYCLES)



PCF DEFENSE INDUSTRIES
A DIVISION OF PACCAR

M9 ACE TORQUE SPIKES

	1400 RPM				2000 RPM			
VEHICLE MOD	R2-F1 CB	R2-F1 GS	R2-F2 CB	R2-F2 GS	R2-F1 CB	R2-F1 GS	R2-F2 CB	R2-F2 GS
STD SHIFT/BLADDER	3200	5200	3800	5400	4600	6900	4900	7000
MOD SHIFT/BLADDER	3000	5000	3200	5200	4600	6900	4700	6800
STD SHIFT/SPRING	3200	5300	3400	5500	4600	6800	4900	6800
MOD SHIFT/SPRING	3200	5200	3000	5400	4400	6800	4500	6700

OTHER TORQUE LEVELS

ENGINE MAXIMUM THEORETICAL OUTPUT — 5200
TORQUE AT TRACK SLIP, 1ST GEAR — 2500
TORQUE AT BRAKE SLIP, 1ST GEAR — 3500
ENGINE AT 2800, SHIFT INTO 1ST GEAR — 3700
MAXIMUM ACCELERATION, 2800 RPM 1ST THRU 4TH GEAR — 2100
TYPICAL BULLDOZING — 2900

ALL VALUES IN LB-FT MEASURED WITH 18000 LB BALLAST

8 85 005

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME T. SCHUMAKER REFERENCE M9 ACE

DATE 6 AUG 1985 PAGE ______ OF ______

M9 ACE REVERSAL CYCLES

FOR COMPARISON PURPOSES ESTIMATE THE NUMBER OF REVERSAL CYCLES IN THE LIFE OF AN M9 ACE.

DURING FDE AT FT HOOD TEXAS, WHILE BUILDING TANK TRAPS A TOTAL OF 957 REVERSAL CYCLES IN 5 HOURS WERE COUNTED.

ASSUMPTIONS

AS A WORST CASE CONDITION ASSUME THAT ALL DOZING CONSISTS OF BUILDING TANK TRAPS.

THE DUTY CYCLE OF THE M9 HAS BEEN ESTABLISHED AS 44.4% DOZING AND THE REMAINDER AS TRAVEL, HAULING, GRADING, SCRAPING, SWITCHING AND IDLE.

THE LIFE, PRIOR TO REBUILD, OF THE M9 HAS BEEN ESTABLISHED AT 600 HOURS.

PCF-RN-1284

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME ______________________ REFERENCE ______________________

DATE ______________________ PAGE __________ OF __________

CALCULATIONS

WORST CASE NUMBER OF REVERSALS IN 600 HR.

$$C_y = \frac{(600)(.444)(957)}{5} = \underline{\underline{51,000 \text{ CYCLES}}}$$

ESTIMATED NUMBER OF CYCLES AT DRIVE SHAFT FAILURE DURING FOE

AVERAGE TEST HOURS (5 VEHICLES) - 162 HR

ASSUME 44.4% TANK TRAP

∴ CYCLES AT FAILURE

$$C_y = \frac{(162)(.444)(957)}{5} = 13800 \text{ CYCLES.}$$

DESIGN CYCLES - $\underline{\underline{51000}}$

PCF-RN-1284

PACIFIC CAR AND FOUNDRY COMPANY
ENGINEERING DEPARTMENT

NAME T. SCHUMAKER REFERENCE ____
DATE 8 AUG 85 PAGE ____ OF ____

FROM PM 179

RESULTS OF QUESTIONS TO FT HOOD 179 ACE OPERATORS — 6 WERE QUESTIONED

1. WHAT STEER MODE DID THEY USE WHEN DOZING?

ANS - CLUTCH BRAKE - 90% OF THE TIME.

2. WHAT STEER MODE DID THEY USE WHEN BACKING UP?

ANS - GEARED STEER - 90% OF THE TIME.

3. WHEN DID THEY RESHIFT TO CLUTCH BRAKE AFTER BACKING UP?

ANS - AFTER SHIFTING AND DURING FORWARD MOVE.

PCF-RN-1284

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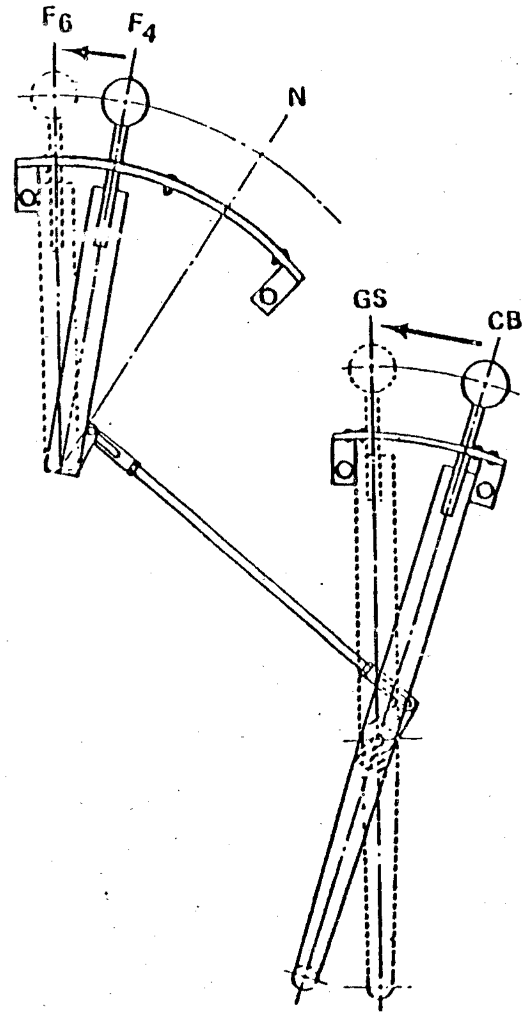
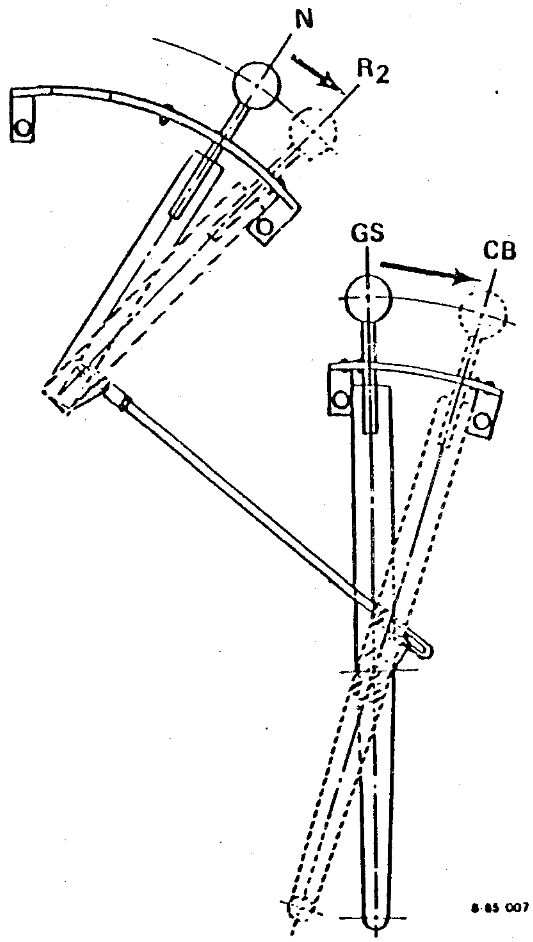
HIGH SPEED OPERATION



MOVEMENT OF TRANSMISSION SELECTOR FROM F4 THROUGH F6 SHIFTS STEER SELECTOR FROM CB TO GS.

DOZING OPERATION



MOVEMENT OF TRANSMISSION SELECTOR FROM NEUTRAL THROUGH R2 SHIFTS STEER SELECTOR FROM GS TO CB.

APPENDIX B

MATERIALS BRANCH FAILURE ANALYSIS ON

FOE SHAFTS

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ENGINEERING DIRECTORATE
PHYSICAL TEST DIVISION
REPORT NO. 86-M-2

STECS-EN-PM   Date of Report 2 JUL 1985

Title of Report: M9 ACE Transmission Output Shaft Failure

TECOM Project Title: First Article: Initial Production Test

TECOM Project No: 8-EG-785-ACE-001   W.O. No. 330-21470-50

Conducted For: Mr. L. Weaver, Wheeled Veh. Br., Support Eq. Div., Close Combat Sys. Directorate

References: Metals Handbook, Ninth Edition, Volumes 1 and 4, Eighth Edition Volume 10; ASTM E45-84; SAE Handbook J498b

TEST OBJECTIVE:

The objective of this investigation was to determine the cause of failure of the transmission output shaft of the M9 ACE.

INTRODUCTION:

1. Testing was conducted from 14 May 85 to 2 Jul 85.

2. During Follow On Evaluation of the M9 ACE, the transmission output shafts of five test vehicles failed. (This type of failure disables the vehicle and requires replacement of the power pack for repair.) The shafts from two vehicles were evaluated by Clark Equipment (the manufacturer of the transmission). The other two shafts, numbered 9 and 11 were submitted to the Materials Branch for an analysis of the failure. Because there were no failures during the Initial Production Test, output shafts from two of the three IPT vehicles number 3 and number 4 were submitted to be compared with the failed shafts.

PROCEDURE:

1. Fracture surfaces were examined and photographed.

2. The splined portion of the failed shaft number 11 and the two unfailed shafts were magnetic particle inspected to reveal the extent of cracking. The two unfailed shafts were sectioned in the spline runout area.

3. All shafts were chemically analyzed by x-ray fluorescent spectroscopy; carbon and sulfur levels were determined by IR absorption of combustion gases.

4. Rockwell hardness measurements were taken on the shaft surface, at 3/4 radius, at 1/2 radius, and at the center.

5. A laboratory fracture was produced in the longitudinal radial plane by putting a circumferential shell about 10 mm thick in bending. The hardened case was on the tension side; the section was from shaft 9 near the threaded end.

6. Microhardness was measured as a function of distance from the root or fillet surface on all four shafts and from the crown surface on shaft number 9.

7. Longitudinal metallographic specimens were examined for inclusion content according to ASTM E45, Method A.

8. The microstructure in the case and core regions of each shaft was examined on transverse specimens.

RESULTS:

1. The fracture surfaces are shown in Figures 1 and 2, Appendix I. Cracks, which were initiated near the fillet of each spline tooth, propagated longitudinally and radially. At the spline runout and the end of the engagement with the mating splines, the cracks propagated transversely. The final fracture occurred in the transverse plane at the spline runout. Both fractures were identical except that the longitudinal cracks propagated deeper radially on shaft number 9 than shaft 11.

2. Magnetic particle inspection revealed no cracks away from the highly stressed end of shaft 11. Both unfailed shafts have longitudinal cracks in each fillet. The cracks range from 5 to 10 mm in length; at the end of engagement and the beginning of runout, branches propagate across roots and up teeth at 45° to longitudinal axis of the shaft (see fig. 3, app. I). These cracks all extend through the case as shown in Figure 4, Appendix I.

3. The chemical analyses of all four shafts conform to specifications for AISI 8822 (See table 1, app. II).

4. All four shafts had similar macrohardness profiles (See table 2, app. II).

5. The laboratory fracture showed a mixture of cleavage and microvoid coalescence in the hardened case area (See fig. 5, app. I). The core fracture appeared "woody" by visual observation. Examination in the SEM reveals microvoid coalescence initiated at long stringers and large round inclusions. Figure 6, appendix I shows an example of each. The stringers were identified as MnS; the round inclusions were identified as calcium modified alumina.

6. The microhardness profiles of the four shafts near their fillets were similar. The case was deeper as measured from the spline crown. (See graph 1, app. III.)

7. The inclusion content was rated at 3 for type A thin series; it was rated at 1 for type D thin, 1 for type D thick. Several type D inclusions ranging from 15 to 25 μm in diameter (thick series are in the 12 μm range) were found. Examples are shown in Figures 7, 8, 9, Appendix I.

8. The microstructure of the case is martensite with some retained austenite (see fig. 10, app. I). The microstructure of the core is predominantly bainite (see fig. 11, app. I).

DISCUSSION:

1. The failure occurred by reverse torsional fatigue. The cracks originated and propagated near the root fillets of the splines in the longitudinal planes in the radial direction. The exact origin of the cracks is unclear because the surfaces rubbed during propagation. If there were no existing cracks or surface flaws, fatigue cracks should initiate at the case/core interface because this area has the most stress for its strength. But it is uncertain how the stress concentration varies with depth into the shaft. The existence or absence of quench cracks prior to loading the shaft cannot be determined. But an intergranular crack found through the case in a microspecimen taken 20 mm from the threaded end of the shaft indicates that transformation induced stresses are assisting the crack formation (see fig. 12, app. I).

2. The chemical composition of the core is satisfactory.

3. The similar microhardness and macrohardness profiles of the four shafts indicates that they all received similar heat treatments. The necessity of such a hard case in the splines is questionable. A softer case would certainly have more fracture toughness and lower transformation induced stresses.

4. The high nonmetallic inclusion content of these shafts is not a cause of failure. But the high content is accelerating the rate of crack growth in the core. This is evident both from the branching of the fatigue cracks as seen in Figure 4, Appendix I and from the laboratory fracture as seen in Figure 6, Appendix I.

5. The microstructure of the core is appropriate. The microstructure of the case consists of martensite which hasn't been tempered enough to be ductile at room temperature (see fig. 5, app. I), and it contains austenite which is prone to transformation which induces additional stresses and introduces a brittle phase. untempered martensite.

6. The manufacturer has initiated a fix, which includes shot peening the shaft and using AISI8640 steel. The use of the harder steel should increase the shaft life by slowing crack propagation in the core, but this doesn't address the problem of crack initiation. Shot peening should help prevent crack initiation, but considering the consistent initiation in these shafts, it is questionable whether shot peening will prevent all crack initiation.

CONCLUSION:

The fatigue failure of the M9 ACE transmission output shafts occurred from reversing torsional loads as applied upon changing direction of the vehicle. Contributors to the failure include a case with low toughness, which aided in crack initiation, and a high inclusion content, which accelerated crack growth in the core.

RECOMMENDATIONS:

1. Lowering the stress at the spline fillet will increase life. Increasing the fillet radius or using a fillet root spline will decrease both transformation induced stresses and operating stresses. Requiring operators to come to a complete stop will lower the amplitude of the stress reversals and thereby increase shaft life.

2. The metallurgical fixes of the manufacturer are marginal. A more fracture tough case in the splines could be obtained in conjunction with a very hard surface on the bearing race (further up the shaft) by selective carburizing or induction tempering.

3 Enclosures
Appendix I - Figures
Appendix II - Tables
Appendix III - Graph

SUBMITTED:

B. C. HENDRIX
Materials Branch

REVIEWED:

CHARLES R. KLARICH
Chief
Materials Branch

APPROVED:

KERSEY A. JONES, JR.
Chief
Physical Test Division

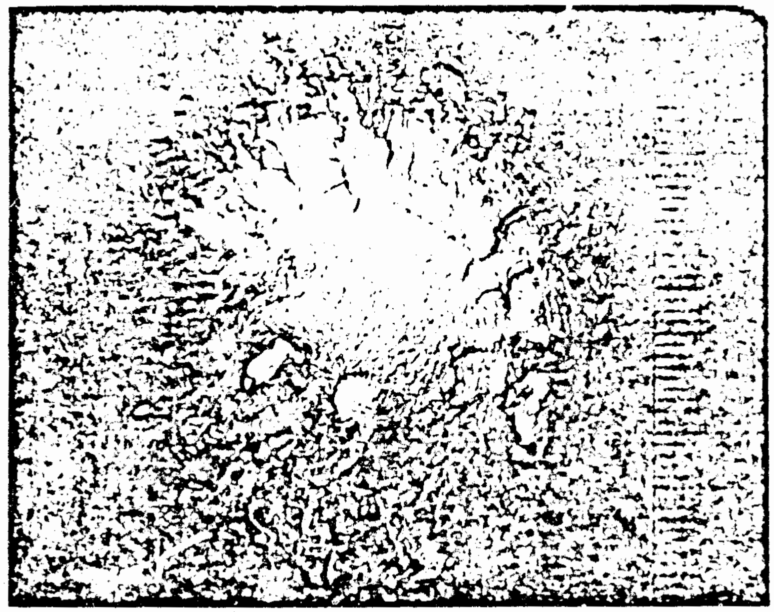


Figure 1. Fracture surface of shaft 9.

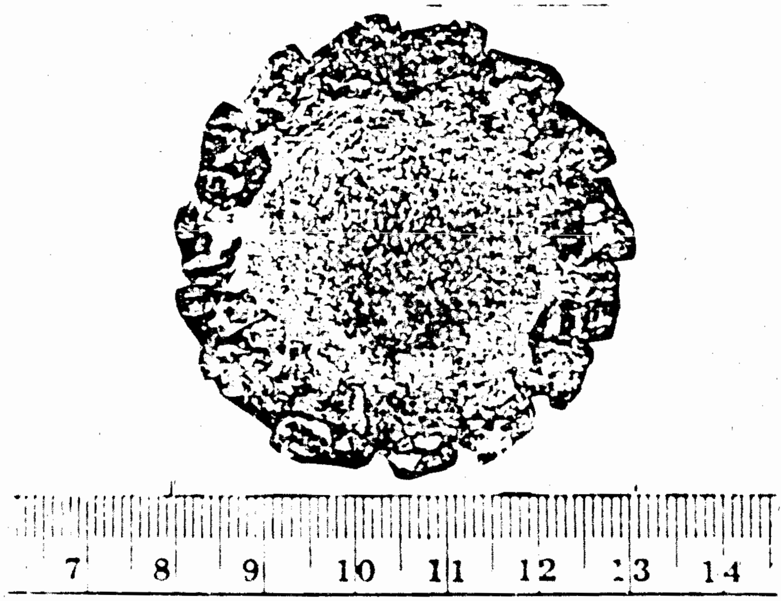


Figure 2. Fracture surface of shaft 11.

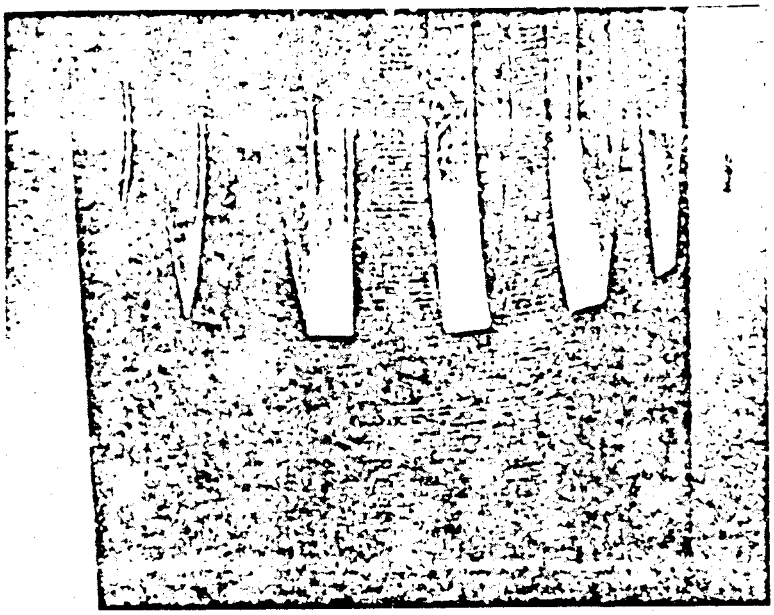


Figure 3. Magnetic particle indications on shaft 4. An identical pattern was found on shaft 3.

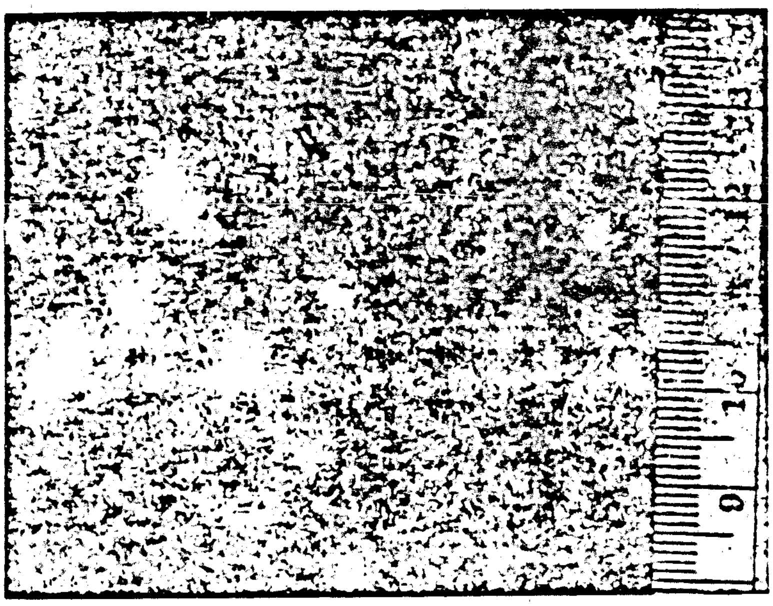


Figure 4. Cross section of shaft 3 near spline runout. Note that all cracks extend through the case. A similar situation was found in shaft 4.

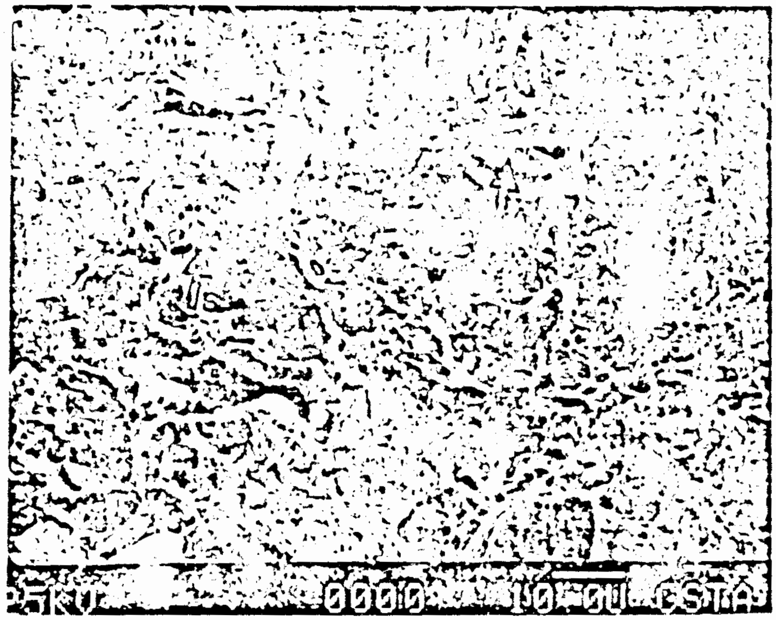


Figure 5. Laboratory fracture of shaft 9. In the case, the fracture was dominated by transgranular cleavage (A) and had interspersed microvoid coalescence (B). Secondary electron image.

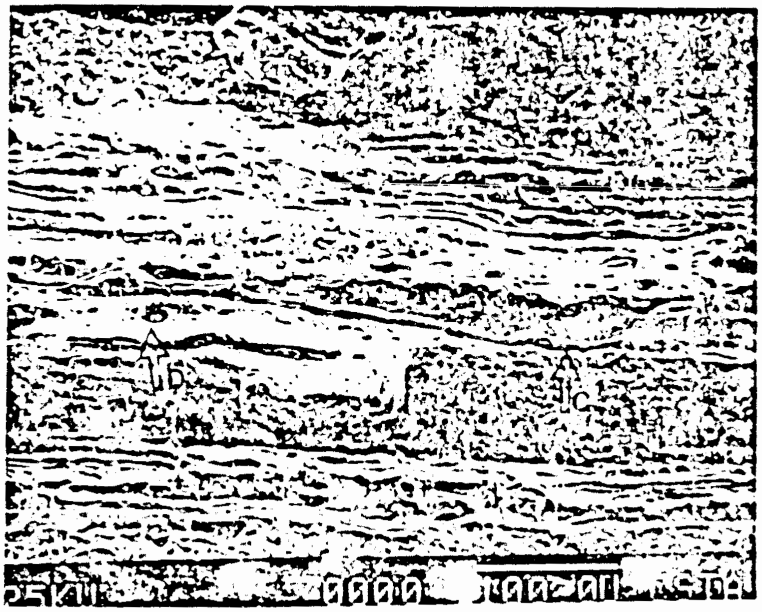


Figure 6. Laboratory Fracture of shaft 9. In the core, the fracture was entirely microvoid coalescence. Void origins were predominantly MnS as at C with some origins at calcium modified alumina as at D.

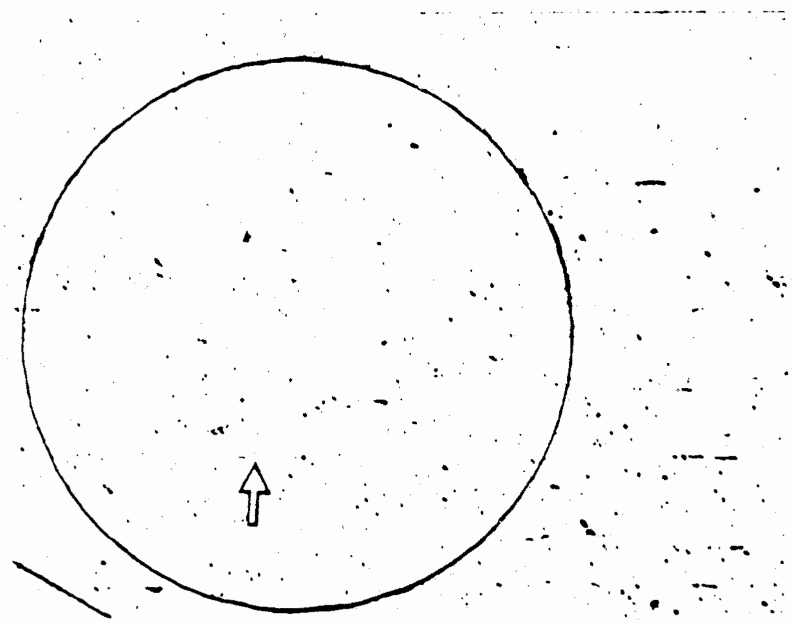


Figure 7. Manganese sulfide inclusions, type A stringers. MAG: 100X

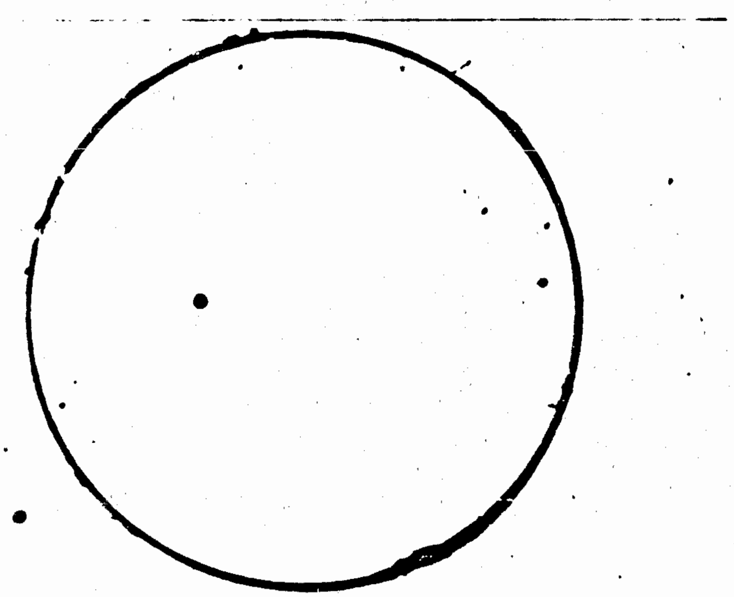


Figure 8. Type D inclusions. See figure 9. MAG: 100X

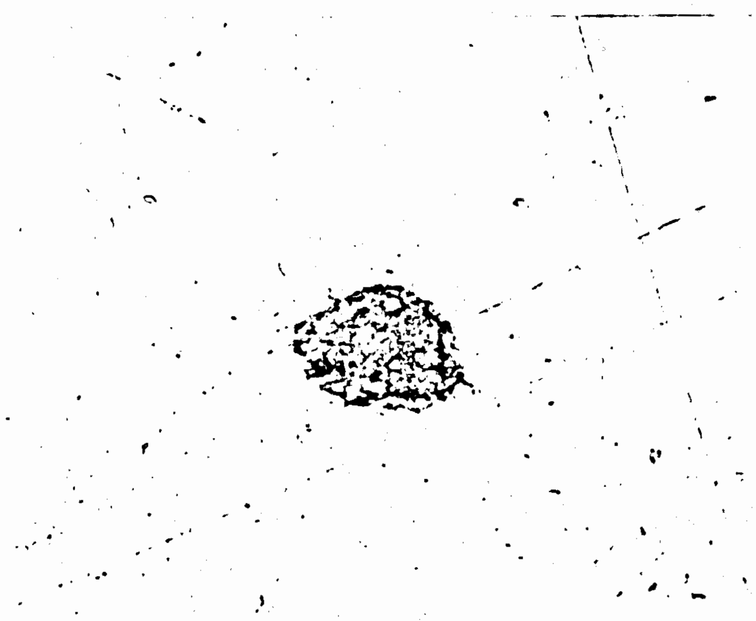


Figure 9. Calcium modified alumina inclusion with a shell of calcium modified manganese sulfide MAG: 1000X

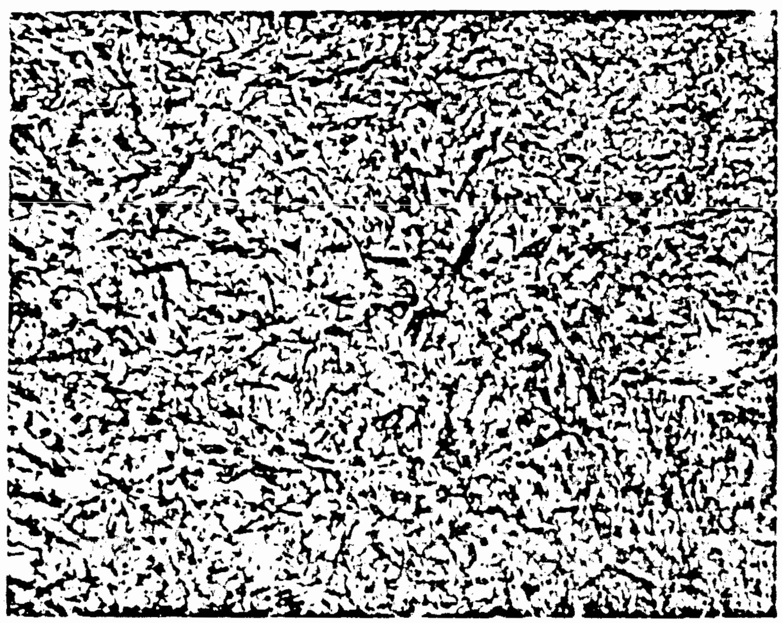


Figure 10. Microstructure of hardened case. Gray and dark constituents are plate martensite; white constituents are retained austenite

MAG: 800X
ETCHANT: PICRAL

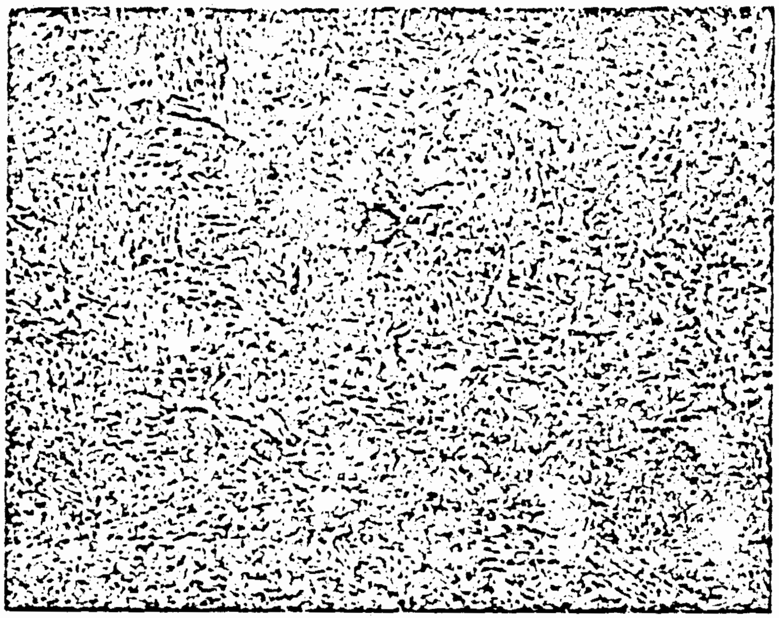


Figure 11. Bainitic Structure of shaft core. MAG: 1000X ETCHANT: PICRAL

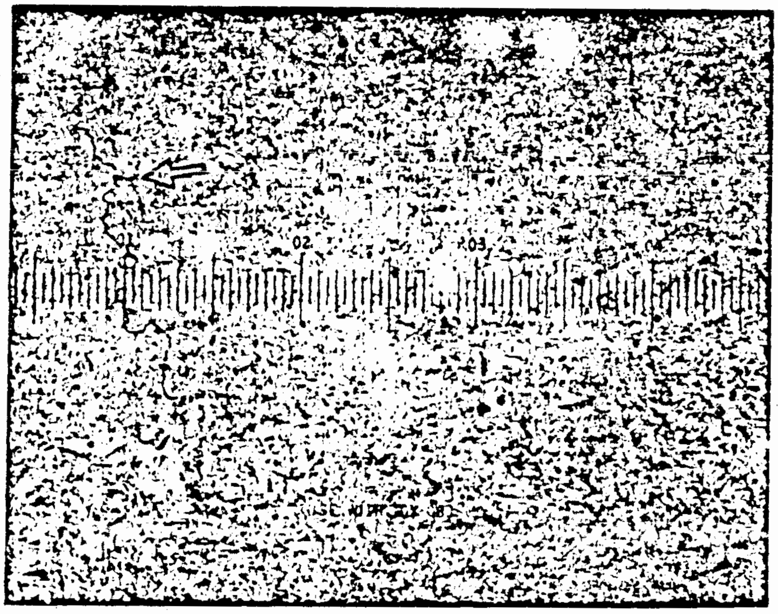


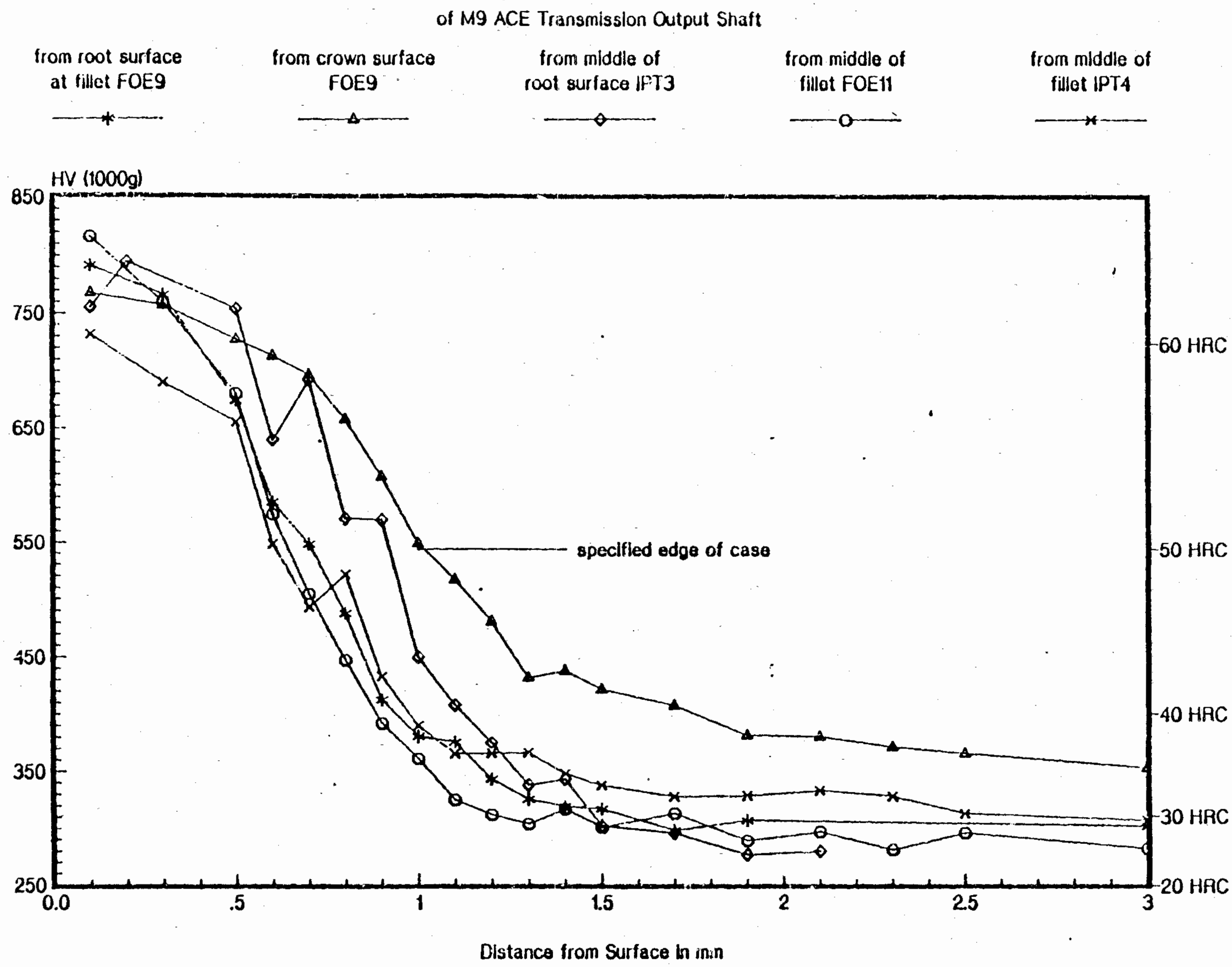
Figure 12. Transverse microsection taken 20 mm from threaded end. Note intergranular fracture in center of root. Scale is in inches. MAG: 1000X ETCHANT: PICRAL

TABLE 1. CHEMICAL ANALYSIS OF SHAFTS.

	C	Mn	P	S	Si	Cr	Ni	Mo
#3 IPT	.22	.82	.002	.016	.15	.47	.42	.32
#4 IPT	.22	.83	.005	.018	.25	.49	.42	.31
#9 FOE	.23	.81	.002	.018	.15	.46	.41	.32
#11 FOE	.21	.83	.007	.017	.36	.49	.42	.31
AISI 8822	.20-.25	.75-1.00	.035max	.040max	.15-.30	.40-.60	.40-.70	.30-.40

TABLE 2. ROCKWELL HARDNESS, "C" SCALE

	Center	1/2 Radius	3/4 Radius	Surface
#3 IPT	27.6	27.0	29.0	63.2
#4 IPT	26.1	27.1	30.0	63.1
#9 FOE	25.8	25.7	26.3	62.8
#11 FOE	25.7	26.8	31.3	63.5



APPENDIX III

APPENDIX C

MAB ASSESSMENT

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DEPARTMENT OF THE ARMY
U. S. ARMY MATERIEL SYSTEMS ANALYSIS ACTIVITY
Aberdeen Proving Ground, Maryland 21005-5071

REPLY TO ATTENTION OF

AMXSY-CM

SEP 1985

SUBJECT: M9 ACE Drive Shaft Fix

Commander
US Army Tank-Automotive Command
ATTN: AMCPM-M9
Warren, MI 48397-5000

1. Per your request, AMSAA, with TECOM assistance, has assessed the proposed fix for the M9 transmission output shaft and the risk associated with a production decision prior to validation of the fix. The enclosed memorandum contains that assessment.

2. As with any change to a critical component, some validation of the fix is required. However, having reviewed the proposed fix for this drive shaft and the supporting contractor data already available, it is concluded that the risks associated with this fix are sufficiently low that no additional testing is required prior to a contract decision. Final validation of the fix can be accomplished either by retrofitting an old vehicle prior to production or in the scheduled Initial Production Test, but should not be required for a production decision.

3. AMSAA point of contact is Mr. William A. Niemeyer, AUTOVON 298-6450.

FOR THE DIRECTOR:

Encl

ARREND H. REID
Chief
Combat Support Division

AMXSY-CM 4 September 1985

MEMORANDUM FOR RECORD

SUBJECT: M9 ACE Drive Shaft Fix

1. Issue: Whether additional operational testing is necessary to demonstrate that the recommended fix for drive shaft failures occurring during FOE is satisfactory.

2. Background: Five transmission output drive shafts failed during conduct of FOE, while none failed during IPT. Probably the key factor differentiating operation in the two tests occurred in the competitive side-by-side FOE trials with the D7, digging anti-tank ditches. During these trials, the M9 was repeatedly shifted from reverse to forward without stopping the vehicle as required in the operators manual. Standard procedure in IPT was to bring the vehicle to a stop before shifting from reverse to forward.

3. Assessment of the FOE Test Procedure: Even though the manual specifically prohibits shifting from reverse to forward on the move, it is our opinion that the highly stressed competitive situation that induced the operators to use this technique is not unlike the stress induced by combat conditions. Therefore it is concluded that a fix must be incorporated in the M9 to preclude premature shaft failure under these operating conditions.

4. Description of Fix: Three modifications have been proposed to eliminate fatigue failures in this shaft.

a. The shaft has been redesigned with a material change from AISI 8822H to AISI 8640H, the radius at the spline root has been increased from .008 inches to .031 inches, and the shaft will now be shot peened.

b. Through a linkage change, the M9 now cannot be driven in the geared steer mode when in reverse. This reduces the maximum speed in reverse from 6.3 mph to 4.3 mph at maximum rpm for the clutch brake mode (or from 4.6 mph to 2.1 mph at 2000 rpm).

c. The gear shift guide has been modified to preclude a straight shift motion from forward gears to reverse gears. The new guide requires a lateral motion around a protruding ear when changing direction.

5. Assessment of Fix: Both failed and unfailed shafts were analyzed by the Physical Test Division, Engineering Director, CSTA, at APG, confirming that the failures

occurred by reverse torsional fatigue. This is significant to the retest issue because a single cycle overload failure would not necessarily be fixed with a redesigned shaft which merely transferred the load to the next weakest point where failure may again occur. In that case a repeat of FOE testing would be strongly indicated. No such "fusing" situation exists with this shaft however.

a. Shaft redesign - Individual contributions of the several shaft changes cannot be quantified, though all are in the right direction. The net improvement is shown with the attached Clark S-N curves obtained from tests of the original and new shaft configurations. FOE testing is estimated to have produced, at most, 15,000 cycles of maximum stress reversal, during the competitive test, to the average shaft failure point (about 30 percent of the life required). While peak torques experienced during FOE would vary with engine speed at gear change, failure at about 15,000 cycles would indicate a repeated torque of about 6000 ft-lb. In fact, an instrumented drive shaft in PACCAR testing experienced 5400 ft-lbs torque when shifted on the move from reverse to forward at 1400 rpm engine speed. A similar shift at 2000 rpm produces about 7000 ft-lbs in the shaft. Regardless of whether the peak torques are 6000 ft-lbs as indicated from FOE or 7000 ft-lbs, the maximum from instrumented testing, the redesigned shaft would be expected to have a fatigue life that is 5 to 10 times greater than the original shaft life.

b. Geared steer link change - This simple change prevents geared steer in reverse, requiring the vehicle to remain in the clutch brake mode, which would normally be used in dozing, when backing up. The effect is to reduce the maximum reverse speed and consequently the torque spike induced with an on-the-move shift from reverse to forward gear. The instrumented shaft tests indicate the 7000 ft-lbs previously measured are reduced to about 5000 ft-lbs when operating in the clutch brake mode. At this torque, the S-N curves indicate the new shaft would have nearly 100 times the life required.

c. Gear shift guide modification - This change is intended to increase the time required to shift between forward and reverse gear positions, and to increase awareness that a direction change is involved. However measured shift times and shaft torques with the original and modified shift guides showed very small differences. The effect of this change is therefore judged to be minimal, though clearly unnecessary in view of the effectiveness of the other changes.

6. Risk Assessment: The risk that the new shaft design will not meet the 600-hour durability requirement of the M9 ACE under conditions experienced in FOE testing is virtually negligible. However, even the new shaft is not abuse-proof. Should the operator shift from reverse to forward without stopping at maximum throttle, i.e. 2800 rpm, it is anticipated that torque peaks will be sufficiently high, even with all fixes in place, to cause premature shaft failure. Normal operating procedure should continue to require a stop before shifting from reverse to forward gears.

The only other element of risk associated with the proposed fix is that associated with a productivity loss in reducing the reverse speed by eliminating geared steer in reverse. The maximum impact of this change would be in the high stress dozing operation previously described. Calculated times for equivalent tasks are increased less than 10 percent for such tasks and under normal operation there is no loss at all. It is concluded that the operational risk of significant productivity loss is negligible.

7. <u>Conclusion</u>: As with any change to a critical component, some validation of the fix is required. However, having reviewed the proposed fix for this drive shaft and the supporting contractor data already available, it is concluded that the risks associated with this fix are sufficiently low that no additional testing is required prior to a contract decision. Final validation of the fix can be accomplished either by retrofitting an old vehicle prior to production or in the scheduled IPT, but should not be required for a production decision.

William A. Niemeyer
WILLIAM A. NIEMEYER
Chief, Mobility Analysis Branch
USAMSAA

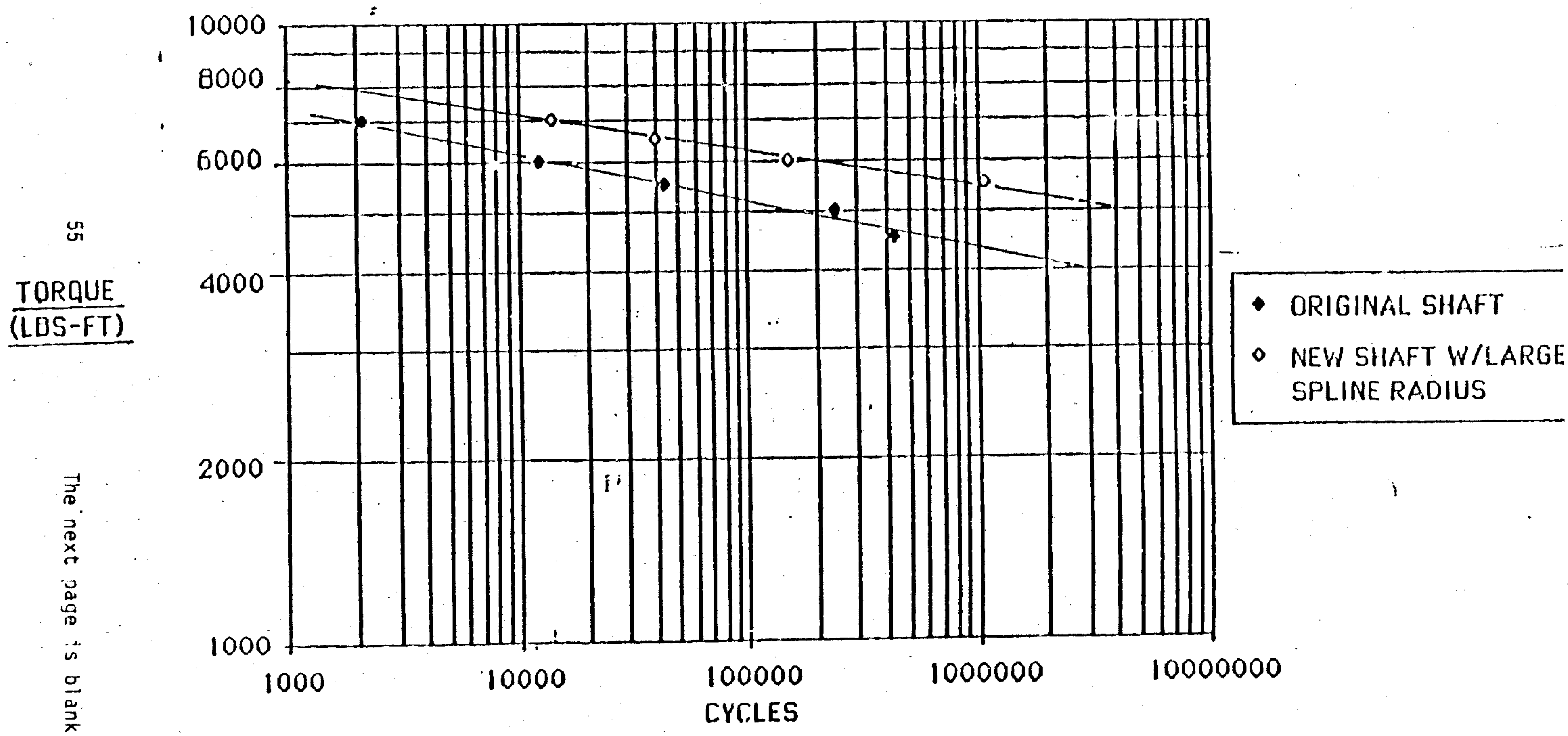
R. D. M
R. DAVIS MENTZER
AMSAA Analyst

Larry Harrison
LARRY HARRISON
TECOM Analyst

10 AUG 1985

VEHICLE: M9-ACE
TRANSMISSION: CLARK 13.5HR3610
SHAFT: TRANSMISSION OUTPUT SHAFT

TORSIONAL FATIGUE S-N CURVES (IN FULL REVERSE CYCLES)



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APPENDIX D

TEST DESIGN PLAN

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AMSAA TDP

M9 ACE DRIVE SHAFT FIX VERIFICATION TEST

PURPOSE OF TEST -

The purpose of this test is to demonstrate the adequacy of the recommended fix for the M9 ACE Drive Shaft which failed during FOE. Specifically, the increased fatigue life of the shaft, as projected by the contractor, will be verified.

BACKGROUND -

Five transmission output drive shafts failed during conduct of FOE, while none failed during IPT. Probably the key factor differentiating operation in the two tests occurred in the competitive side-by-side FOE trials with the D7, digging fighting positions. During these trials, the M9 was repeatedly shifted from reverse to forward without stopping the vehicle as required in the operators manual. Standard procedure in IPT was to bring the vehicle to a stop before shifting from reverse to forward.

The peak torsional loading which resulted from the cyclic motion of shifting from reverse to forward was found to have caused the fatigue failures in the drive shafts. Failure occurred at an estimated one-third of the required durability of major components within the M9.

TEST CONCEPT -

Four drive shafts will be tested using two vehicles. Testing will consist of cycling the vehicle from reverse to forward motion, at constant engine speed, to simulate the effect of digging fighting positions. Since peak loading on the shaft occurs while shifting from reverse to forward, a test which simulates this cyclic behavior will verify the adequacy of the fix.

Vehicle #1 will be identical to the vehicles used in FOE testing, including the drive shaft. The vehicle will be run forward and backward in a cyclic pattern until shaft failure. This shaft will be replaced with an identical shaft and again run until failure. These results should produce a baseline from which to judge improvement resulting from proposed changes.

Vehicle #2 will be modified to incorporate the fixes the contractor has proposed. These include:

(1) A new shaft with a higher fatigue life.

(2) A modified geared steer linkage to prevent geared steer in reverse.

(3) A modified gear shift guide to preclude a straight shift motion from reverse to forward.

This vehicle, with each of the two new shafts, will be operated through a cyclic pattern identical to that of vehicle #1 for a minimum of five times the life of the original shafts or until shaft failure, whichever occurs first. Since the vehicles available for this test have previously experienced extensive operation, failures other than to the drive shaft may be expected and should not be considered relevant to this issue.

DATA REQUIREMENTS -

The vehicles and drive shafts will be instrumented to record:

(1) Shift Position

(2) Drive Shaft Torque (FT-LBS)

(3) Engine Speed (RPM)

(4) Drive Shaft Speed (RPM)

(5) Continuous Count of Cycles Accomplished

TEST PROCEDURE -

Testing shall be conducted on a hard, level surface. The M9 shall be loaded to maximum gross vehicle weight (18,000 lb bowl load). The suspension shall be locked in the sprung mode.

Testing will consist of cycling the vehicle from reverse motion to forward motion to simulate the effect of digging fighting positions. The vehicle shall achieve a constant road speed in reverse, at an engine speed of 2000 RPM, before shifting to forward. Shifting from reverse to forward will be conducted as rapidly as possible and with no reduction in engine speed. Shifting from forward to reverse will be accomplished after the vehicle has come to a complete stop.

Vehicle #1 (unmodified) shall be shifted from Reverse 2 to Forward 2 in the geared steer mode to produce maximum torque levels experienced in FOE (6000-7000 FT-LBS). Cycling shall continue until shaft failure, with the driver being rotated as necessary. Upon failure, the second unmodified shaft will be installed and tested in a similar manner.

Vehicle #2 (modified) shall be shifted from Reverse 2 to Forward 2. The action of the modified geared steer linkage will insure operation in the clutch brake steer mode. Measured torsional loads on the shaft should be approximately 5000 FT-LBS. This test will run to shaft failure or for a minimum of five times the life of the orginal shaft as determined by Vehicle #1 testing. A second modified drive shaft will then be installed and the test repeated.

CRITERION FOR SUCCESS -

The proposed fix will be considered successful if the new transmission output shaft design demonstrates a fatigue life approximately three times as long as that of the original configuration.

JOINT MESSAGEFORM

UNCLASSIFIED

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01 of 01	301500Z	OCT	85	PP	PP	UUUU		AT	ZYUW	303140OZ

BOOK: NO

MESSAGE HANDLING INSTRUCTIONS

FROM: DIR USAMSAA APG MD//AMXSY-CM//

TO: CDR USATACOM WARREN MI//AMCPM-M9//

INFO CDR AMC ALEX VA//AMCDE-SS-V//

CDR USAOTEA FALLS CHURCH VA//CSTE-FSS-A//

UNCLAS

A. LTR, AMSAA, AMXSY-CM, 12 SEP 85, SUBJ: TRANSMISSION OUTPUT SHAFT CHECK TEST.

1. A REVISION TO REFERENCE A DRAFT TEST PLAN IS RECOMMENDED. THE ORIGINAL PLAN CALLED FOR THE VEHICLE TO BE SHIFTED FROM REVERSE TO FORWARD "ON THE MOVE", BUT FOR SHIFTS FROM FORWARD TO REVERSE THE VEHICLE WOULD BE STOPPED THIS PLACES TORSIONAL STRESS ON THE TRANSMISSION OUTPUT SHAFT IN ONE DIRECTION ONLY AND RECENT ANALYSIS INDICATES THIS WILL REQUIRE EXTREMELY LONG TEST TIMES TO INDUCE FAILURE. AT LEAST ONE FOE DRIVER HAS SINCE REPORTED THAT ON-THE-MOVE SHIFTING FROM FORWARD TO REVERSE WAS ALSO USED TO DISLODGE DIRT FROM THE BLADE. THE APG LABORATORY ANALYSIS OF THE FAILED SHAFTS ALSO CONCLUDES THAT THE FAILURES OCCURRED BY REVERSE TORSIONAL FATIGUE, INDICATING SIGNIFICANT TORQUE WAS APPLIED IN BOTH DIRECTIONS.

2. BASED ON THE ABOVE FACTORS, IT IS RECOMMENDED THAT ALL REMAINING

DISTR

DRAFTER TYPED NAME, TITLE, OFFICE SYMBOL, PHONE

SPECIAL INSTRUCTIONS

RELEASER

TYPED NAME, TITLE, OFFICE SYMBOL AND PHONE: AREND H. REID, C, CSD, AV298-6598

SIGNATURE

SECURITY CLASSIFICATION: UNCLASSIFIED

DATE TIME GROUP

DD FORM 173/2 (OCR)

PREVIOUS EDITION IS OBSOLETE

JOINT MESSAGEFORM

SECURITY CLASSIFICATION: UNCLASSIFIED

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02 of 02	301500Z	OCT	85	PP	PP	UUUU		AT	ZYUW	30314002

BOOK: NO

MESSAGE HANDLING INSTRUCTIONS

TESTING (EXCEPT THAT IDENTIFIED IN PARA 3) BE ACCOMPLISHED WITH FROM:

ON-THE-MOVE SHIFTS FROM FORWARD TO REVERSE AS WELL AS FROM REVERSE TO:

TO FORWARD. ENGINE SPEED SHOULD BE HELD AT 2000 RPM THROUGHOUT THE TEST CYCLE AS BEFORE.

3. IT CAN BE SHOWN THAT THE IMPACT OF TESTING ACCOMPLISHED TO DATE WITH LOADING IN ONE DIRECTION ONLY WILL HAVE MINIMAL IMPACT ON THE FATIGUE LIFE RESULTING FROM FULL REVERSE CYCLES. NEVERTHELESS, THE SHAFTS CURRENTLY IN TEST (ONE OLD DESIGN AND ONE NEW DESIGN) SHOULD BE BROUGHT TO AN IDENTICAL NUMBER OF TEST CYCLES UNDER THE OLD TEST PROCEDURE BEFORE CONVERTING THE TEST TO LOAD THE SHAFT IN BOTH DIRECTIONS.

4. AMSAA POC IS MR. WILLIAM NIEMEYER, AV 298-6450, OR MR. DAVE MENTZER, AV 298-6445.

5. AMSAA - PROVIDING LEADERS THE DECISIVE EDGE.

DISTR:

DRAFTER TYPED NAME, TITLE, OFFICE SYMBOL, PHONE

SPECIAL INSTRUCTIONS

RELEASER TYPED NAME, TITLE, OFFICE SYMBOL AND PHONE: ARENDH. REID, C, CSD

SIGNATURE

SECURITY CLASSIFICATION: UNCLASSIFIED

DATE TIME GROUP

PACCAR DETAILED TEST PLAN

9 OCTOBER 85

M9 ACE DRIVE SHAFT FIX VERIFICATION TEST

PURPOSE OF TEST

THE PURPOSE OF THIS TEST IS TO DEMONSTRATE THE EFFECTIVENESS OF THE DRIVE SHAFT AND TRANSMISSION CONTROLS DESIGN CHANGES FOR INCREASING THE LIFE OF THE TRANSMISSION OUTPUT DRIVE SHAFT.

BACKGROUND

FIVE FAILURES OF THE M9 DRIVE SHAFT OCCURRED AT FOE WHILE NONE FAILED AT IPT. THE PRIMARY DIFFERENCE BETWEEN THE IPT AND THE FOE WAS THE COMPETITIVE NATURE OF THE TESTING AT FOE. THE M9 WAS IN DIRECT, SIDE-BY-SIDE COMPETITION WITH THE D7 SO THE OPERATORS WERE RUNNING THE M9 IN GEARED STEER (A HIGHER SPEED THAN CLUTCH BRAKE) AND NOT BRINGING THE VEHICLE TO A STOP BEFORE SHIFTING FROM REVERSE TO FORWARD. THIS TYPE OF OPERATION CAUSED HIGH TORQUE SPIKES IN THE DRIVE SHAFT WHICH RESULTED IN THE FATIGUE FAILURES OF THE SHAFT.

TEST PROCEDURE

FOUR DRIVE SHAFTS WILL BE TESTED BY TWO VEHICLES. ALL FOUR SHAFTS WILL HAVE A SERIAL NUMBER ETCHED IN THEM ON BOTH ENDS BEFORE THEY ARE INSTALLED IN THE VEHICLES. THESE NUMBERS WILL BE RECORDED IN A WRITTEN LOG. THE PREVIOUS HISTORY OF THE SHAFTS WILL BE DOCUMENTED (HOURS OF PREVIOUS OPERATION AND THE VEHICLES THEY HAVE BEEN USED IN). ALL FOUR SHAFTS WILL BE MAG PARTICLE INSPECTED BEFORE INSTALLATION.

THE VEHICLES WILL BE OPERATING FOR TWO SHIFTS TO COMPLETE THE TEST AS QUICKLY AS POSSIBLE. TESTING WILL BE ON A HARD, LEVEL SURFACE AT PCF DEFENSE INDUSTRIES WITH BOTH VEHICLES TRAVELING BACK AND FORTH ON A STRAIGHT COURSE OF APPROXIMATELY 50 FEET IN LENGTH. EACH VEHICLE WILL BE LOADED WITH 18,000 LB IN THE BOWL TO BRING IT UP TO MAXIMUM GROSS VEHICLE WEIGHT. THE SUSPENSION WILL BE LOCKED IN THE SPRUNG MODE. IF THERE ARE NO FAILURES THE OPERATORS SHOULD BE ABLE TO COMPLETE APPROXIMATELY 2000 CYCLES PER DAY.

THREE CHANNELS OF DATA FROM EACH VEHICLE WILL BE RECORDED CONTINUOUSLY ON MAG TAPE: SHAFT TORQUE, TRANSMISSION SHIFT LEVER POSITION AND ENGINE SPEED. THE DATA WILL BE RECORDED PERIODICALLY ON A VISICORDER TO ASSURE THAT THE DATA IS CONSISTENT WITH THE EXPECTED VALUES. A CYCLE COUNTER WILL RECORD THE NUMBER OF CYCLES OF EACH VEHICLE AND THIS DATA WILL BE NOTED PERIODICALLY ON A VOICE TRACK. THE TAPE RECORDER, VISICORDER, SIGNAL CONDITIONING AND COUNTERS WILL BE LOCATED IN AN INSTRUMENT VAN. DATA WILL BE TRANSMITTED TO THE VAN ON UMBILICAL CORDS. THERE WILL BE ONE M9 ON EACH SIDE OF THE VAN.

DESCRIPTION OF THE DATA CHANNELS:

- SHIFT POSITION - POTENTIOMETER LOCATED ON THE SHIFT QUADRANT
- PROP SHAFT TORQUE - STRAIN GAGES WIRED TO A SLIP RING. BOTH SHAFTS WILL BE INSTRUMENTED AND CALIBRATED BY CLARK EQUIPMENT COMPANY.
- ENGINE SPEED - VEHICLE TACH GENERATOR
- COUNTER - DIGITAL GAGE TRIGGERED WHEN SHIFT POSITION DATA INDICATES VEHICLE HAS BEEN SHIFTED INTO REVERSE.

THE OPERATORS WILL REPORT DIRECTLY TO BILL BOYCE. BYRON BAKKE WILL BE THE SECOND ENGINEER IN CHARGE OF THE TEST. THE FIRST SHIFT WILL TEST FROM 0800 TO 1600 AND THE SECOND SHIFT WILL BE FROM 1600 TO 2400. THE OPERATORS WILL BE USING EAR PLUGS AND HEADSET HEARING PROTECTION. THIS WILL BE VERIFIED EACH TIME THE DRIVER ENTERS OR LEAVES THE VEHICLE.

THE TWO M9S WILL BE PRODUCTION #003 AND PRODUCTION #007. VEHICLE #003 WILL BE TESTED IN THE SAME CONFIGUREATION AS THE M9S WERE AT FOE. THE TRANSMISSION OUTPUT SHAFT FROM PRODUCTION #006 WILL BE INSTALLED IN #003 AND TESTED TO FAILURE. THIS VEHICLE WILL BE SHIFTED FROM REVERSE 2 TO FORWARD 2 AT 2000 RPM WITH THE STEER UNIT IN THE GEARED STEER MODE. THE OPERATOR WILL MAKE THE SHIFTS AS QUICKLY AS POSSIBLE THIS TYPE OF OPERATION WILL PRODUCE THE SAME HIGH PEAK TORQUES AS OCCURRED AT FOE (APPROXIMATELY 7000 LB-FT). THE FORWARD TO REVERSE SHIFTING WILL BE ACCOMPLISHED AFTER THE VEHICLE HAS COME TO A COMPLETE STOP. THIS TYPE OF FORWARD-REVERSE CYCLING WILL CONTINUE TO FAILURE WITH THE OPERATOR ROTATED AS NECESSARY. WHEN THE FIRST SHAFT FAILS IT WILL BE REPLACED WITH THE SHAFT FROM #003 AND TESTED IN THE SAME WAY TO FAILURE.

FOLLOWING THIS FAILURE, #003 WILL BE TESTED WITH A REDESIGNED TRANSMISSION OUTPUT SHAFT, REDESIGNED STEER UNIT LINKAGE AND REDESIGNED SHIFT QUADRANT. THE ACTION OF THE MODIFIED SHIFT LINKAGE WILL ASSURE OPERATION IN THE CLUTCH BRAKE MODE FOR THE REVERSE 2 TO FORWARD 2 SHIFTS. THESE DESIGN CHANGES WILL LOWER THE PEAK TORQUES CONSIDERABLY (TO APPROXIMATELY 5000 LB-FT) AND SIGNIFICANTLY INCREASE THE LIFE OF THE SHAFT. THE VEHICLE WILL BE OPERATED THROUGH THE IDENTICAL CYCLIC PATTERN AS DESCRIBED FOR UNMODIFIED #003 FOR A PERIOD OF 5 TIMES THE LONGEST LIVED SHAFT OF THE FAILURES IN UNMODIFIED #003 OR UNTIL A SHAFT FAILURE (WHICHEVER COMES FIRST).

VEHICLE #007 WILL ALSO BE TESTED WITH A REDESIGNED TRANSMISSION OUTPUT SHAFT, REDESIGNED STEER UNIT LINKAGE AND REDESIGNED SHIFT QUADRANT. IT WILL BE TESTED WITH THE SAME PROCEDURE AS MODIFIED #003.

THERE WILL BE VISITORS FROM AMSAA, OTEA, TECOM AND POSSIBLY OTHER ORGANIZATIONS THAT WILL COME TO WITNESS PARTS OF THE TEST.

A WEEKLY SUMMARY OF RECORDED TORQUE LEVELS AND CYCLES ON EACH VEHICLE WILL BE GIVEN TO THE M9 PM. A REPORT WILL ALSO BE GIVEN TO THE PM WHEN A SHAFT FAILS. ADDITIONAL A LOG WILL BE KEPT OF ALL FAILURES OCCURRING TO OTHER SYSTEMS DURING THIS TEST. A FINAL REPORT OF THE RESULTS AND CONCLUSIONS WILL BE WRITTEN WHEN THE TEST IS COMPLETE.

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APPENDIX E

MATERIALS BRANCH FAILURE ANALYSIS ON VERIFICATION TEST SHAFTS

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ENGINEERING DIRECTORATE
PHYSICAL TEST DIVISION
MATERIALS BRANCH
REPORT NO. 86-M-44

STECS-EN-PM

Date of Report 19 Dec 85

Title of Report Metallurgical Analysis of Special Study ACE Shafts
TECOM Project Title Special Study of the M9 Armored Combat Excavator (ACE)

TECOM Project No. 8-EG-785-ACE-004 Work Order No. 330-23602-60

Conducted For Lawrence Weaver, Wheeled Vehicle Branch, Close Combat Systems Directorate, Bldg 400

References P.T. Division Report No. 86-M-2; ASM Metals Handbook, Vol. 1

TEST OBJECTIVE:

The object of this investigation was to compare the failure mode of two ACE transmission shafts to previous failed ACE shafts and to examine an unfailed shaft.

INTRODUCTION:

1. Testing was conducted from 21 Nov 85 to 13 Dec 85.

2. It was requested that Materials Branch conduct an analysis of three M9 ACE transmission output shafts that were used during testing at the contractor's (PACCAR) facility during the peri . 16 Oct to 5 Nov 1985. Two of the shafts failed as expected during the .est. The third shaft was still intact and was inadvertently introduced during the contractor's test. The two failed shafts were compared to previous failures documented in Physical Test Division (PTD) Report No. 86-M-2. The third shaft was analyzed to determine how it differed from the two failed shafts. The unfailed shaft, SN 002, was labeled "A". The failed shafts, SN 005 and SN 001, were labeled "B" and "C" respectively.

PROCEDURE:

1. Fractured surfaces were examined and photographed.

2. All three shafts were inspected using non-destuctive magnetic particle techniques. The shafts were sectioned in the spline runout area.

3. The shafts were chemically analyzed using an X-ray fluorescent spectrometer and a carbon-sulphur IR determinator.

4. Rockwell hardness measurements were taken on the shaft surface, at 1/2 radius, and at the center.

5. Microhardness was measured as a function of distance from the root of the spline surface on the three shafts.

6. The microstructure of the shafts was examined using chemical etching techniques and a metallograph.

7. An optical comparator was used to measure the fillet radius at the base of the splines of each sectioned shaft.

RESULTS:

1. A comparison of the fracture location of shafts B and C and the nonfractured shaft A is shown in Appendix I, Figure 1. Shaft B and C fractured initially near the fillet of each spline tooth and the cracks propagated longitudinally and radially (app I, fig. 1 and 3). The fracture features were nearly identical to the documented failures of PTD Report No. 86-M-2. SEM examination revealed quasi-cleavage features along the outside splined fracture surface of the shafts (app I, fig. 4). The core of the shafts consisted of microvoid coalescence which indicated ductile fracture (app I, fig. 5). Some particles around which microvoids initiated were found to contain high concentrations of manganese and some calcium.

2. Magnetic particle inspections of the shafts revealed extensive cracking in the splines of the failed shafts (app I, fig. 6). The cracking was limited to the region of final failure. Shaft ~~C~~A showed no discontinuities indicative of cracks or other defects (app I, fig. 7).

3. The chemical analysis of the shafts is given in Appendix II, Table 1. Shafts B and C corresponded to AISI 8822 alloy steel. The silicon level of shaft B was slightly higher than the nominal concentration specified for 8822. Shaft A met specifications for AISI 8640 alloy steel.

4. The macrohardness of the shafts is listed in Appendix II, Table 2. The surface hardness measurement was the same for the three shafts. The through hardness of shaft A was 15 to 20 Rockwell hardness points above the through hardnesses of shafts B and C.

5. Microhardness profiles of the three shafts are shown in Appendix III, Figure 1. The hardness measurements were made close to the fillets and progressed from the surface to the core of the shafts. The microhardness profiles of shafts B and C were similar to previous failures. The shafts of this investigation were compared directly to previous failures for the same region. The microhardness data are given in Appendix III, Figure 2 and 3. The hardness profile of shaft A indicated a significant increase in case depth. Shafts B and C and the previous shafts possessed a case depth of approximatly 1.0 mm. The case depth of shaft A was 1.8 mm.

6. The inclusion content of all three shafts was measured. A longitudinal cross-section of shaft B and C revealed maganese sulfide inclusions with a rating of 2, thin series (app I, fig. 8). Shaft A was cleaner and revealed no ratable inclusions.

7. The microstructure of shafts B and C was similar to the previous failures. The carburized case of shaft A consisted of martensitic lathes in a matrix of retained austenite (app I, fig. 10). This structure resulted from the high carbon content induced by the carburizing atmosphere and the subsequent quench. The core of shafts B and C consisted of bainite and ferrite (app I, fig. 11). The core of shaft C was a mixture of tempered martensite and bainite.

8. The optical comparator revealed a fillet radius of 0.010 inch in the splines of shafts B and C, and a fillet radius of 0.015 inch in shaft A in the root of the splines. The change in radius appeared to be a result of tooling and not a deliberate change in design of the fillet radius.

DISCUSSION OF RESULTS:

1. All material properties indicated shafts B and C were identical to the failed shafts of PTD Report No. 86-M-2. The fracture features indicated shafts B and C failed in a similar manner to the previously failed shafts.

2. The primary change to shaft A was the introduction of a new alloy, AISI ~~8822~~ 5820 alloy steel, and a corresponding increase in case depth and core hardness. The carburized case was almost doubled and both the microhardness and macrohardness indicate a consistent increase in hardness through the cross-section of the shaft.

3. Metallurgically the case of shaft A was altered from the tempered martensitic structure of shafts B and C to a structure with a substantial amount of retained austenite. This structure, brought on by the carburizing process, is mechanically weaker and tends to introduce microcracks into the carburized case.

4. The fillet radii of all three shafts were essentially the same. Although shaft A was supposed to be shot peened, it was not possible to ascertain that this process had been performed.

5. Shaft A seemed to represent a significant improvement in fatigue strength from the design of the old shafts. This is a limited sample and further testing of more than one shaft is needed to insure the validity of the improvements.

CONCLUSIONS:

It is concluded that:

1. The transmission shafts B and C of the ACE failed due to reverse torsional loading. The failure mode was similar to previous failures of the ACE transmission shaft.

2. Transmission shaft A revealed no surface cracks or discontinuities after magnetic particle inspection.

3. Shaft A was composed of AISI 8640 alloy steel. There was no significant change in the fillet radius of the spline region compared to shafts B and C. The presence of shot peening on shaft A was not ascertained.

4. The carburized case depth of shaft A was significantly increased compared to shafts B and C.

RECOMMENDATIONS:

It is recommended that:

1. The carburizing process of shaft C be altered to prevent retained austenite in the case and reduce the chance of microcracks in the case.

2. Further testing of the improved shaft be instituted to ascertain the reliability of the design changes.

4 Enclosures

Appendix I - Photographs
Appendix II - Chemistry and Hardness Tables
Appendix III - Microhardness Graphs

SUBMITTED:

Will C. Simmons

WILL C. SIMMONS
Materials Branch

REVIEWED:

Will C. Simmons

for CHARLES R. KLARICH
Chief
Materials Branch

APPROVED:

Kenneth A. [illegible]

for KERSEY A. JONES, JR.
Chief
Physical Test Division

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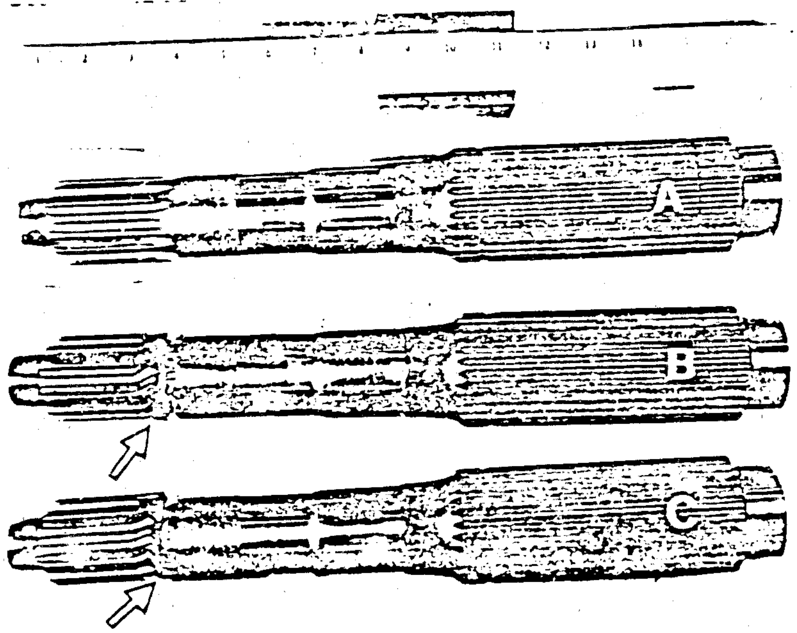


Figure 1. Shaft A was undamaged. Fracture path of shafts B and C in the splines indicated by arrows.

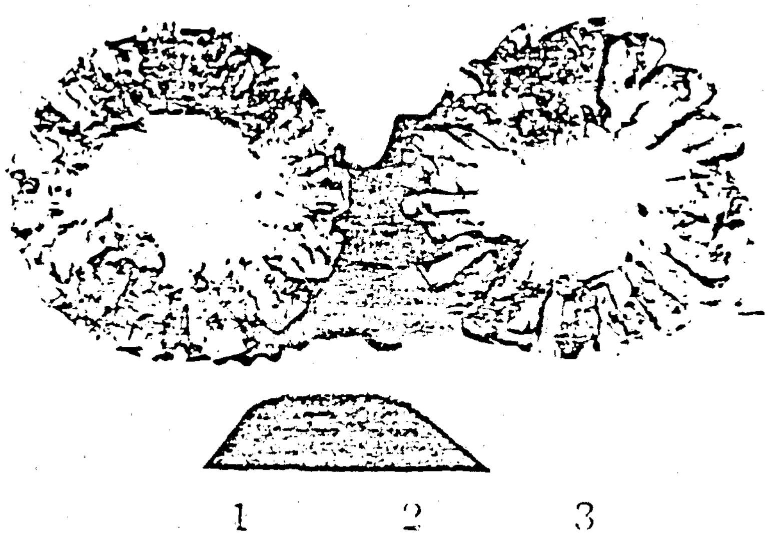


Figure 2. Mating fracture surfaces of shaft B.

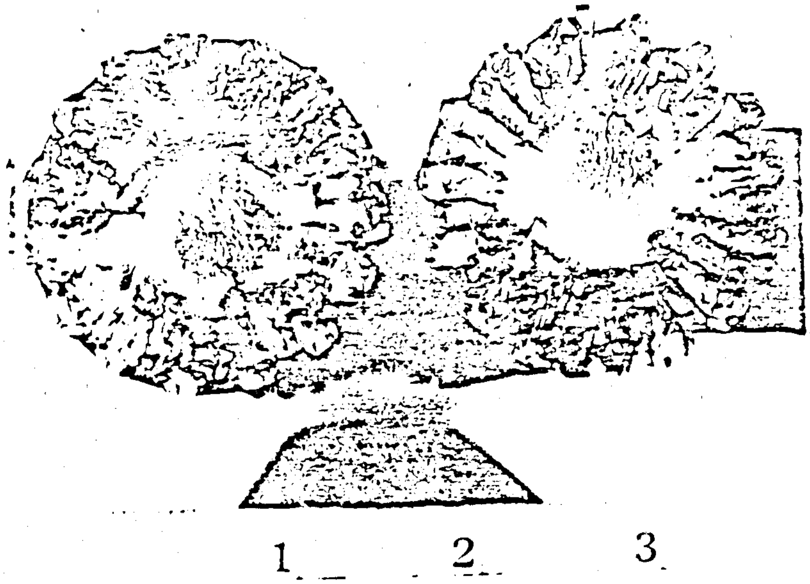


Figure 3. Mating fracture surfaces of shaft C.

Figure 4. Fracture surface typical of case hardened region in the splines of the shaft. Features are indicative of quasi-cleavage fracture.

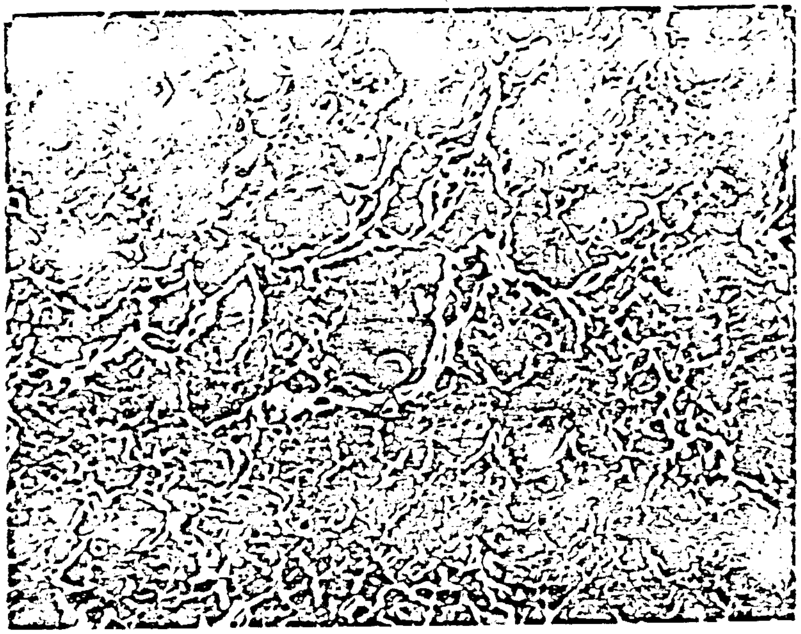


Figure 5. Typical microvoid coalescence found in the core of the fractured shaft. Particles around which microvoids begin are apparent (arrows). Magn: 1500 X

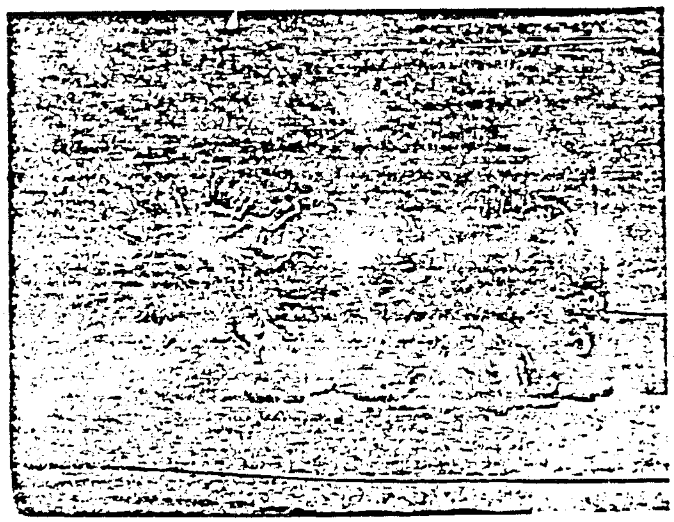


Figure 6. Magnetic particle testing reveals extent of cracking in shaft B.

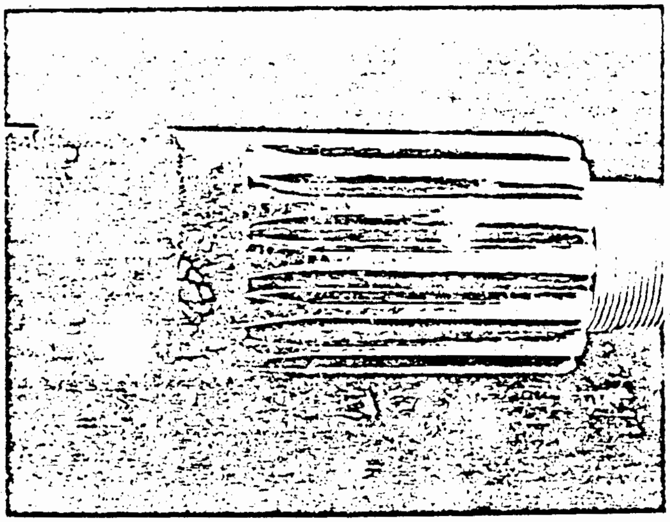


Figure 7. Magnetic particle testing revealed no discontinuities indicative of cracks or other defects.

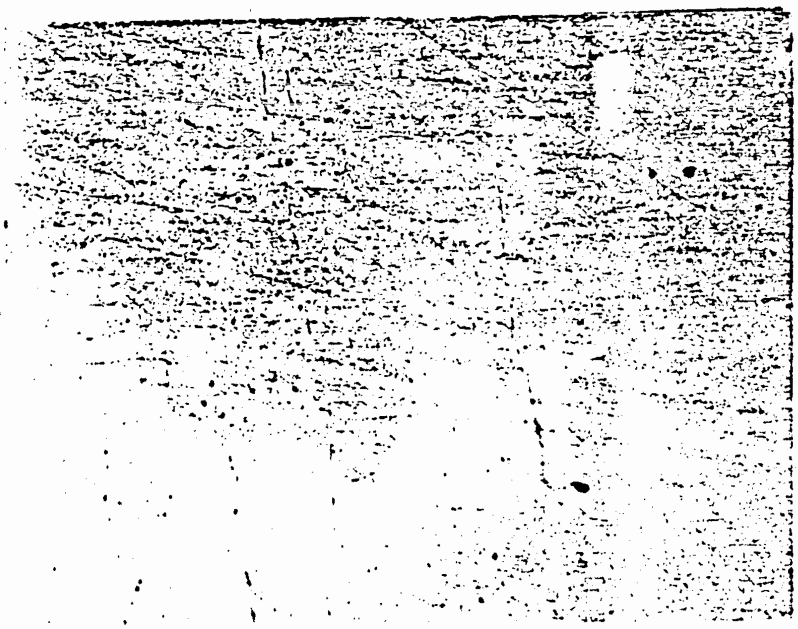


Figure 8. Longitudinal micrograph of shaft B.
No Etchant. Magn: 100 X

Figure 9. The microstructure along the outside case of shafts B and C consisted of quenched and tempered martensite. Etchant: 2% Nital Magn: 400 X

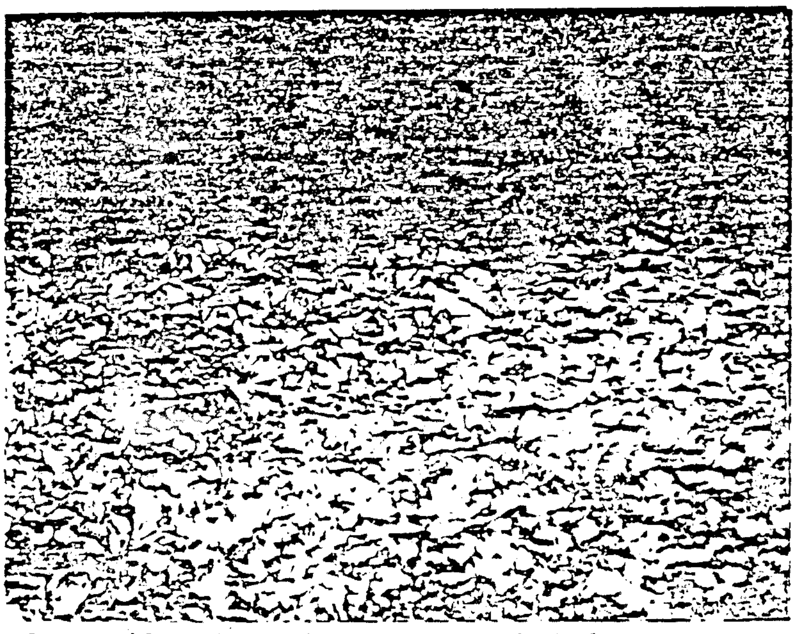


Figure 10. The carburized case of shaft A consisted of a layer of martensitic lathes in a matrix of retained austenite (white). Etchant: 2% Nital Magn: 400 X

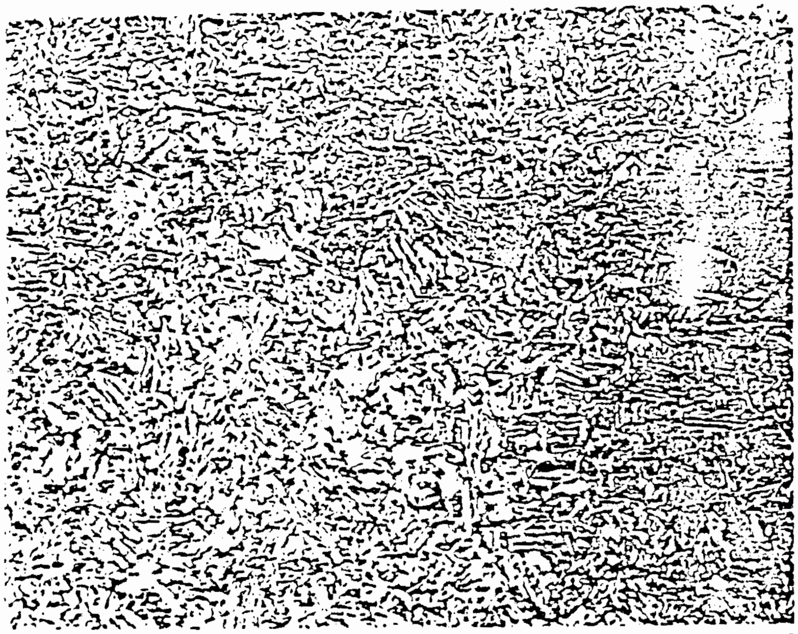


Figure 11. Microstructure of a transverse section of the core of shaft C consisted of bainite (dark regions) and ferrite (white, equiaxed regions).
Etchant: 2% Nital Magn: 400 X

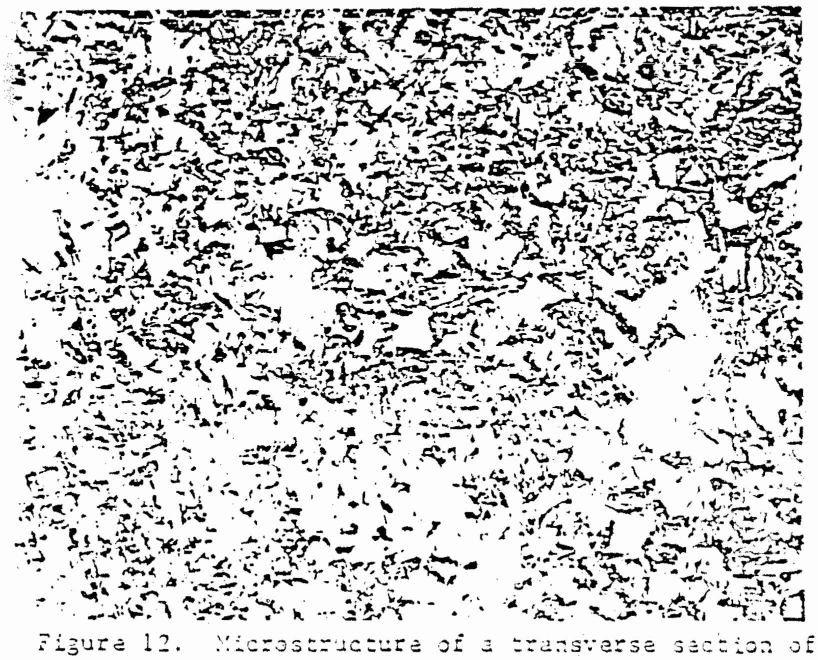


Figure 12. Microstructure of a transverse section of the core of shaft A consisted of tempered martensite (gray) and bainite (dark areas). Etchant: 2% Nital Magn: 400 X

TABLE 1. CHEMICAL ANALYSIS OF ACE TRANSMISSION SHAFTS

% Elemental Composition by Weight

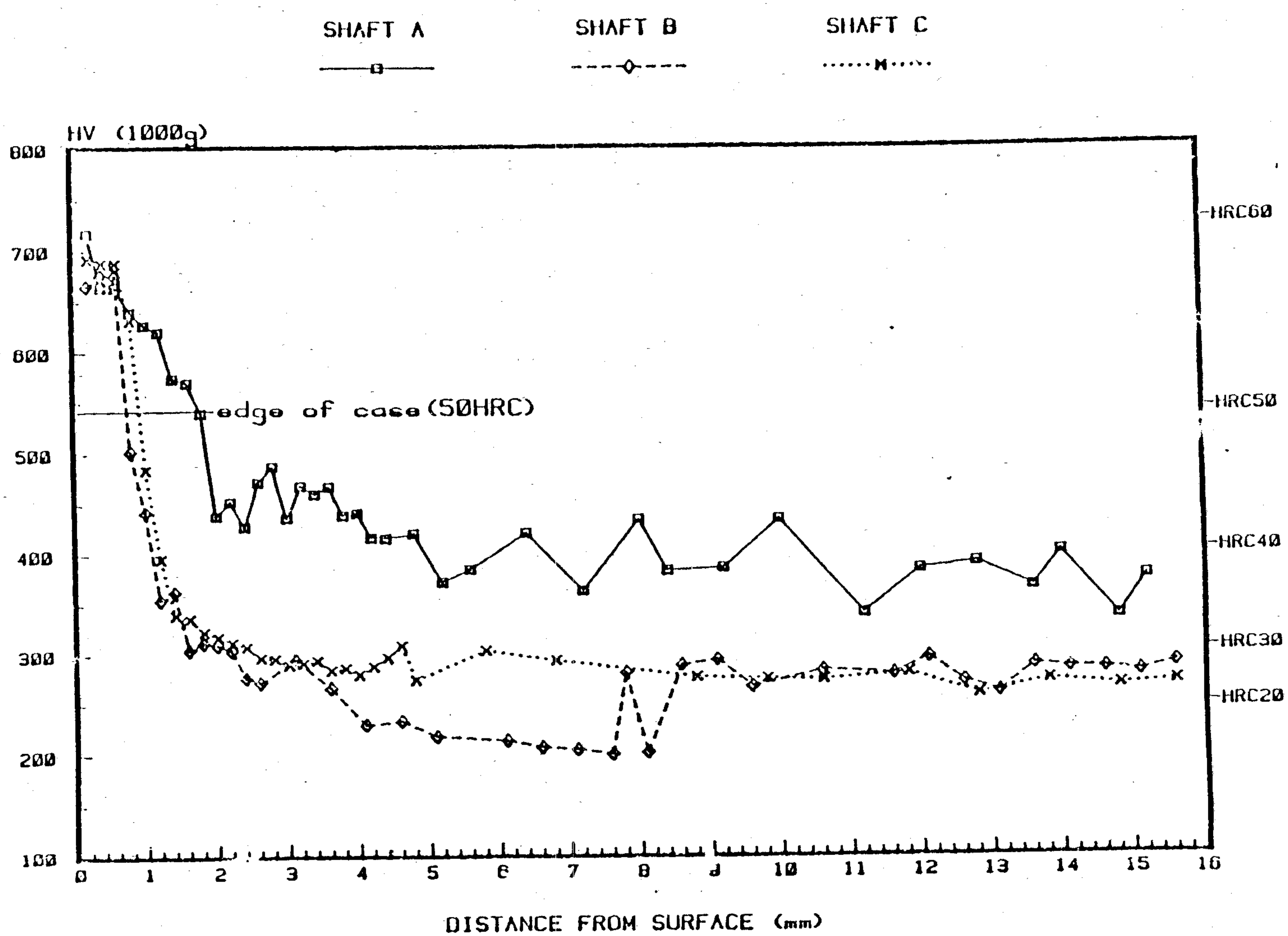
	C	Mn	P	S	Si	Cr	Ni	Mo	Fe
	0.42	0.94	.007	.017	0.33	0.59	0.45	0.20	Remainder
	0.24	0.84	.013	.018	0.37	0.57	0.42	0.32	Remainder
	0.22	0.82	.010	.016	0.29	0.56	0.43	0.32	Remainder
22 max	0.25	1.00	.035	.040	0.30	0.60	0.70	0.40	---
min	0.20	0.75	--	--	0.15	0.40	0.40	0.30	---
40 max	0.43	1.00	.035	.040	0.30	0.60	0.70	0.25	---
min	0.38	0.75	--	--	0.15	0.40	0.40	0.15	---

TABLE 2. ROCKWELL "C" HARDNESS OF ACE SHAFTS

Center	1/2 Radius	3/4 Radius	Surface
37	42	45	61
23	23	27	61
22	25	26	61

GRAPH 1. Microhardness Traverse

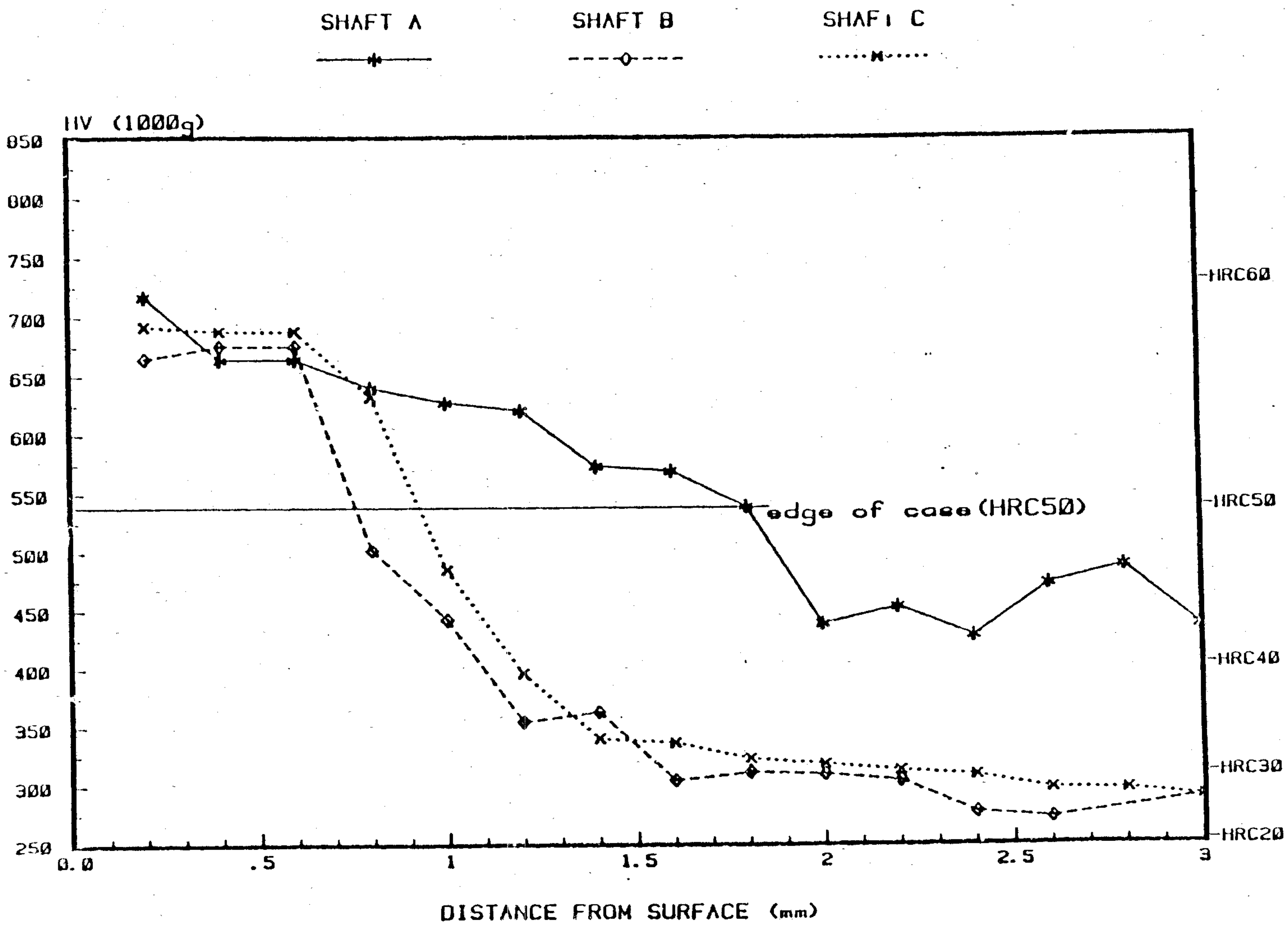
M9 ACE Transmission Output Shaft



APPENDIX III

GRAPH 2. Microhardness Traverse

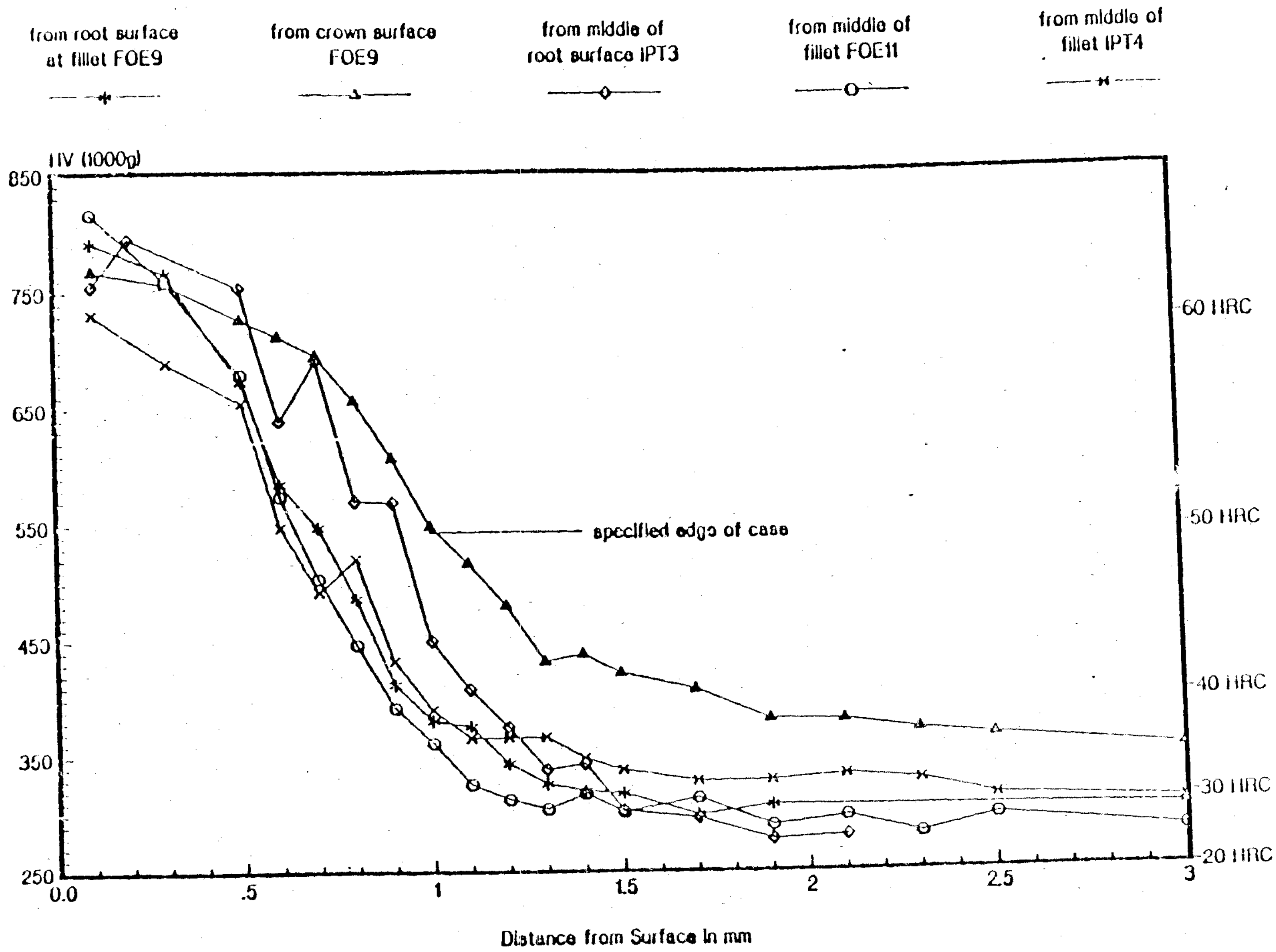
M9 ACE Transmission Output Shaft



APPENDIX III
Page 2

GRAPH 3. MICROHARDNESS TRAVERSE FOR PREVIOUSLY FAILED M9 ACE SHAFTS

(Physical Test Division Report No. 86-M-2)



APPENDIX III
Page 3

DISPOSITION FORM

For use of this form, see AR 340-15; the proponent agency is TAGO.

REFERENCE OR OFFICE SYMBOL	SUBJECT
STECS-EN-PM	Revision of Physical Test Division Report No. 86-M-44, Metallurgical Analysis of Special Study ACE Shafts

TO Chief, Close Combat Systems Directorate
ATTN: L. Weaver

FROM Chief, Materials Br

DATE 21 Jan 86
Mr. Simmons/jab/34461

CMT 1

1. In light of further information supplied to this office concerning the Special Study of the M9 Armored Combat Excavator (ACE), the final recommendations of Physical Test Division Report No. 86-M-44 should be revised. Two improved output transmission shafts from the ACE were tested. The improvements included shot peening of the surface of the shafts and increasing the fillet radius of the spline region of the shafts. The improved shafts had a significant improvement in fatigue life. It is therefore appropriate to eliminate the second recommendation of Physical Test Division Report No. 86-M-44 which reads: "It is recommended that further testing of the improved shaft be instituted to ascertain the reliability of the design changes."

2. Two additional corrections should be made as follows:

   (a) page 2, RESULTS, para 2, line 3 change: Shaft C
   to: Shaft A

   (b) page 3, DISCUSSION OF RESULTS, para 2, line 2 change: AISI 8822
   to: AISI 8640

3. For clarification, Appendix I, Figure 7, should be labeled "Shaft A."

Charles R Klarich

CHARLES R. KLARICH

Mr. Weaver/SW/210-5125

7 APR 1986

STECS-CC-SW

SUBJECT: Special Study of the M9 Armored Combat Earthmover (ACE), TECOM Project No. 8-EG-785-ACE-004

Commander
U.S. Army Test and Evaluation Command
ATTN: AMSTE-TE-T

1. Reference letter, USACSTA, STECS-CC-SW, 8 Jan 86, SAB.

2. In light of further information furnished this office, the final recommendations of the Physical Test Division Laboratory Report No. 86-M-44 should be revised. It was learned through AMSAA that three of the improved transmission shafts (Type A) were tested to five times the fatigue life of the original type shafts. Because of this significant improvement in fatigue life it is therefore appropriate to eliminate the second recommendation of Physical Test Division Report No. 86-M-44 which reads: "It is recommended that further testing of the improved shaft be instituted to ascertain the reliability of the design changes."

3. USACSTA - Providing Leaders the Decisive Edge.

FOR THE COMMANDER:

Original Signed

JEROLD L. HOCK
Director, Close Combat Systems
Directorate

CF:
Cdr, USATACOM
ATTN: AMCPM-LCV-QT
AMCPM-M9
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