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On November 21 and 22, 1972, twelve rounds in various test configurations of the five inch Guided Projectile were test fired. The purpose of these tests was to verify the motor case load and stress analysis so that a selection of wall thickness to support a sixty pound payload could be made. Analysis efforts previous to these tests had indicated that all motor cases with walls less than .190 would buckle and that the .190 wall motor case was marginal.

None of the test motor cases buckled. All the motor cases with walls less than .190 inches experienced a reduction in length (permanent strain) indicating an imposed longitudinal stress in excess of the compression yield. The thinnest motor case wall was .094 inches and this motor case experienced a reduction in length of .193 inches. A .106 wall motor case experienced a reduction in length of .313 inches.

This report presents the current thinking of the structure analysis effort and attempts to provide some guidance for future structural design of the guided projectile motor case.

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STRUCTURAL ANALYSIS STATUS REPORT
5 INCH GUIDED PROJECTILE

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I. INTRODUCTION

On November 21 and 22, 1972, twelve rounds in various test configurations of the five-inch Guided Projectile were test fired. The purpose of these tests was to verify the motor case load and stress analysis so that a selection of wall thickness to support a sixty pound payload could be made. Analysis efforts previous to these tests had indicated that all motor cases with walls less than .190 would buckle and that the .190 wall motor case was marginal.

—None of the test motor cases buckled. All the motor cases with walls less than .190 inches experienced a reduction in length (permanent strain) indicating an imposed longitudinal stress in excess of the compression yield. The thinnest motor case wall was .094 inches and this motor case experienced a reduction in length of .193 inches. A .106 wall motor case experienced a reduction in length of .313 inches.

The ability of the motor cases to resist buckling at stress above the yield point was not wholly unanticipated. The reliability of the method used for determining the strength of the motor cases was under suspicion as noted in Ref. 1, Page 15. Also the suspicion that the buckling strength might be significantly higher than the yield point was referred to in Ref. 2, Page 2.

This report presents the current thinking of the structure analysis effort and attempts to provide some guidance for future structural design of the guided projectile motor case.

II. TECHNICAL RELATIONS AND SYMBOLS

$A =$	$\pi(D-t)t$	Tube annular wall area (in^2)
D		Motor case tube outside dia (inches)
E		Modulus of elasticity (psi)
E_t		Tangent modulus of elasticity (psi)
e		Strain (inches/inch)
F_{tu}		Ultimate tensile stress (psi)
in		Inches
KIPS		Thousands of pounds
KSI		Thousands of pounds per square inch
L		Motor case length (inches)
P		Longitudinal compression load at station X (lbs)
R		Motor case tube outside wall radius (inches)
R_i		Motor case tube inside wall radius (inches)
t		Motor case tube wall thickness (inches)
V		Projectile muzzle velocity (ft/sec)
W		Projectile weight (lbs)
ΔD		Increase in D due to permanent strain (inches)
ΔL		Increase in L due to permanent strain (inches)
η		Load factor
μ		Poisson's ratio

III. DISCUSSION OF TEST RESULTS

A. General

Table 1 summarizes the more significant parameters and variables of the gun firing tests conducted on 21 and 22 November, 1972 of the five inch Guided Projectile at NWC, China Lake. Figures 2, 3, 4, and 6 present results of these tests in graphical form.

B. Motor Case Deformation

The longitudinal compression stress in the motor case cylinder wall exceeded the elastic limit for all motor cases with a wall thickness less than .190 inches. This was physically evidenced by an appreciable, measurable amount of decrease in length, with the more highly stressed motor cases experiencing the greatest amount of decrease. Also, these motor cases experienced a measurable increase in diameter, which is an expected result from having been compressed longitudinally beyond the yield point. Numerical values for the diameter increases are given in Table 1.

Figures 2 and 3 contains sketches depicting partial views of the test motor cases and the purpose of these figures is to illustrate the external effects on the motor case resulting from the tests. Each sketch is identified with the serial number of the round it represents and has listed with it values of parameters and variables applying to that round. The code letters Z, A, B, and C are explained in Figures 2 and 3 and are used to describe external effects caused by test firing. The regions designated by the code letters are to scale and indicate the external surface areas to which the code letters apply.

The region to the left on all sketches designated as "A" contains the most significant of the visible effects. This effect resulted from severe rubbing of the motor case against the gun bore, mashing the area flat, removing fabrication tool marks and leaving it smooth. The motor case expansion at this point is mainly due to the transfer of load from the aft closure piece to the motor case, which is an eccentric loading condition. This condition causes the aft end of the motor case to locally rotate at all points about the circumference which results in a bulging about the circumference. Once

the bulging starts, the inclination is augmented by the longitudinal loading. Restraint to this bulging is provided by fixity at the aft motor case joint, by the wall strength and rigidity, and by the physical obstruction of the gun bore. Region "A" supplies evidence that the bore provided appreciable restraint to prevent further bulging.

Motor cases F033, F035, F036 and F037 visually evidenced circumferential bands on the external surface forward of region "A". These circumferential bands, including the band formed by region "A", evidence the characteristic property of a cylinder wall to develop waves when a uniform bending load is applied locally about the circumference at some station, in this instance by the eccentric loading at the aft end. Although a series of bands was not observed on all the cases, band "A", which is found on all the cases, is accepted as evidence indicating that all the cases experienced the inclination to develop waves. These circumferential bands ahead of band "A" are not indicated in Figures 2 and 3 for motor cases F036 and F037 because they were difficult to define under close scrutiny with a magnifying glass. All the cases with a wall thickness of .150 or less experienced contact with the bore over a considerable portion of the case external surface forward of region "A" and this contact apparently resulted in poor delineation for the bands.

All the motor cases expanded to contact the gun bore and this occurred as the result of three influences:

1. The basic tendency of metals to expand laterally under a compressive load, quantitatively defined by Poisson's ratio.
2. Internal pressure created by dynamic loading of propellant grain.
3. Eccentric loading at the aft end.

The amount of internal pressure from dynamic loading of propellant grain is unknown at present but is suspected as being a considerable amount. The propellant's elastic modulus is very low, on the order of 1200 psi, and such a soft material can be expected to behave somewhat like a fluid under the applied load. A fluid of the same density as the propellant

will form a pressure of 9200 psi at the aft end of the motor case if subjected to 7000 g acceleration, and represents an upper limit to the possible pressure at the bottom. Some of the propellant is supported by shear at the sides and does not require pressure at the bottom to account for its loading. Internal pressure of this sort did not exist for rounds tested in earlier test firings. These rounds were of the fast burn motor configuration with a hole in the center of the propellant grain, which greatly reduces pressure build-up at the aft end.

It can be observed from Figures 2 and 3 that the tendency to expand becomes more pronounced as the thickness of the case wall decreases. This is consistent with the aforesaid three influences. The thinner walls are less able to resist the eccentric loading and have higher stress levels for both column loading and hoop loading from internal pressure. These higher stresses yield higher strains which mean greater motor case expansion.

Some concept of the expansion of the motor case may be achieved by calculating the expansion of a 4.970 inch dia cylinder under an axial load of 644 KIPS and an internal pressure of 9200 psi. This has been done with the cylinder considered to be elastic at all stress levels, and the results are presented in Table 2. The calculated diameter increase is enough for the motor case to contact the bore for all tabulated wall thicknesses, with the .190 wall being marginal. Since the longitudinal stress in all instances and the hoop stresses in two instances are beyond the yield point for 200 KSI HT TR 4340 steel, the actual diameter increases for this steel would be greater than those calculated. The motor case wall experiences its greatest load at the aft end and from this point forward the load decreases. Therefore, the tendency of the motor case to expand will diminish going forward. Further evidence of the case wall expanding to rub the bore is contained in Ref. 3, Paragraphs 4.4.2, 4.5.4, and 4.6.2. This document contains hardness test readings on the test motor cases and shows hardness readings in the area of contact with the bore higher than those which are specified for 200 KSI HT TR 4340 steel. Hardness in these areas was increased by either heating or cold working, resulting in either instance from forceful contact between the motor case wall and the bore. The hardness readings correlate well

with the evidence of Figures 2 and 3, with the highest hardness appearing on the cases evidencing the most extensive and forceful contact with the bore.

C. Elastic Stress versus Strain

Figure 4 shows a compression stress-strain curve based upon the data given in Table 1 and Ref. 4. Also shown is a tension stress-strain curve from MIL-HDBK-5 and it is to be observed that the test compression curve is markedly higher and steeper than the tension curve. Published information to corroborate this shape of the compression test curve is difficult to obtain. No compression values for 200 KSI HT TE steel are given in MIL-HDBK-5 beyond the compression yield point.

However, MIL-HDBK-5 does provide a compression stress-strain curve for 260 KSI HT TR steel, and such a curve has also become available from the recent one inch dia. cylinder tests (February 15 through February 20). These two curves and a tension stress-strain curve from MIL-HDBK-5 are shown in Figure 5. As can be seen, the compression curves are higher and steeper than that for tension. This indicates that the curve in Figure 4 based on the five inch gun tests is basically correct.

The properties tabulation for low-alloy steels in MIL-HDBK-5, Table 2.3.1.0, lists compression yield as significantly higher than tension yield for all the listed heat treated steels. This suggests that the compression curve is higher and steeper than that for tension for these steels.

Further support for the argument that the compression curve is higher and steeper is derived from the fact that stress is defined using the original, unstressed, cross sectional area of the material (MIL-HDBK-5, 1.4.4.5). If the stress-strain curve based on the true area is the same for compression and tension, then the use of the stress as normally defined would cause the compression curve to be higher than the true curve, and the tension curve lower. Thus, the compression curve would be relatively higher and steeper than the tension curve.

Figure 6 shows a tangent-modulus curve for 200 KSI HT TR 4340 steel taken from Figure 4 curve, and two

other curves for the purpose of comparison. The values for the tangent-modulus derived from the gun test data shows a rising trend at higher stresses and is logically consistent with the preceding discussion. When a tension test specimen starts to neck down, its defined stress drops off sharply causing the stress curve to flatten out, in turn causing the tangent-modulus to approach zero. Conversely, with a compression test specimen the defined stress rises when the cross section starts to bulge pronouncedly, and causes the tangent-modulus to rise also.

The foregoing observations and discussion argue convincingly that the basic shape of the curve in Figure 4 derived from the gun tests is correct.

This observation about the compression tangent-modulus has interesting analysis implications. The tangent-modulus is used in the tangent-modulus formula for calculating column strength (Ref. 5, eq. 1.6.2.3) and a higher tangent-modulus value will yield a higher calculated value for strength. Using the tangent-modulus as given by Figure 6 will result in a calculated failure stress for all the test motor cases of about 260,000 psi. At present this figure isn't considered reliable, but it does indicate that more work can be done on this subject.

D. Motor Case Strength

The most significant result of the tests is that none of the cylinders buckled.

The analytical method per Ref. 6, Page 528 was used to anticipate the buckling strength of the test motor cases and this method indicated that cases with a wall thickness less than .190 would buckle and that the .190 wall was marginal. The two reasons which are considered to be mainly responsible for the discrepancy are:

1. The analytical method used does not really apply to the parameter range in which the motor cases are defined. The value of the buckling-stress coefficient, C_c , is not defined by the analytical method for R/t less than 100 and at most R/t for the test motor cases is 28. This aspect is discussed in Ref. 2, Page 2.

2. Supporting influences existed which were not accounted for by the analysis. These influences were the restraint offered by the gun bore and the propensity of the case to expand per the three items listed in Section III.B. The bore prevents buckling to the outside and buckling to the inside is resisted by the aforesaid propensity of the motor case to expand.

The results of the analysis predicting motor case buckling received deceptive reassurance from earlier gun tests in which several motor cases did buckle while passing through the gun barrel. However, these failures have been investigated and it has been ascertained that in the tests in which these failures occurred that there was gas leakage past the obturator ring, subjecting the motor cases to a buckling influence from external pressure. Photos of these buckled motor cases show a buckling mode more similar to that for external pressure than for longitudinal loading. Contrasted to this observation is the fact that no buckling failures to date have occurred in gun firings in which there was no gas leakage. It is therefore probable that the experienced buckling was due in part to external gas pressure and possible that the external pressure experienced was enough alone to buckle the motor cases without the combined effect of axial compression. Figure 8 gives external buckling pressure as a function of wall thickness as determined by analysis. The five inch cases which buckled while passing through the barrel had a .150 wall and the buckling pressure read from the curve is 4300 psi. It is believed that pressure from gas leakage is higher than this value.

IV. TEST CONCLUSIONS

- A. Motor cases with a wall thickness less than .190 inches experienced longitudinal stresses beyond the compression yield point and experienced appreciable reductions in length.
- B. There is an eccentric loading at the aft end of the motor case of considerable magnitude resulting in restraint by the gun bore with high compression loading between the motor case and gun bore and high accompanying friction.

- C. No motor cases to date have buckled solely under the influence of compression loading in gun firing tests. The buckling failures which have occurred have all been accompanied by gas leakage past the obturator ring with buckling occurring while the wall was subject to loading from both longitudinal compression and external pressure.
- D. A significant amount of pressure was developed in the propellant at the aft end of the motor case at the time of maximum acceleration which exerted force on the motor case wall and contributed to the expansion of the motor case to contact and exert force against the bore.
- E. The compression stress-strain curve for 200 KSI HT TR 4340 steel beyond the yield point provides higher values for stress and the tangent-modulus than does the tension stress-strain curve.

V. DESIGN CONSIDERATIONS

A. Design Problems

The following questions indicate problems needing further attention:

1. Should an attempt be made to design a motor case which is compression stressed beyond the compressive yield point at the limit load?
2. How much external pressure must the motor case be able to support?
3. How much surface area contact and how intense a bearing force between the motor case and the bore are permissible resulting from motor case deformation?
4. Would it be desirable to eliminate the pressure build-up in the propellant so as to help minimize motor case expansion?

B. Aft End Loading Eccentricity

The aft end eccentric loading condition can be improved by contouring the case wall so that the centerline of the wall forms a more gradual transition from the load input point to the full motor case diameter. This

principle is illustrated in Figure 7. The more gradual the transition, the less severe is the effect of the eccentricity.

C. Motor Case Wall Thickness

Analysis work has been performed regarding determination of the motor case wall thickness and the results are presented by the curves shown in Figures 8 and 9. The Figure 8 curves are based on the strength exhibited in the recent gun tests and the effect of external pressure is achieved by extrapolations from the test results. These curves are most reliable if pressure is allowed to build up in the propellant since this pressure provided support against buckling in the tests.

The Figure 9 curves are based on conservative analytical assumptions. It is believed that a laterally unsupported motor case with a wall thickness determined by these curves will not buckle under a static load equal to the maximum applied load. These curves do not include the effect of internal pressure and are considered most reliable with internal pressure minimized. Also the inducement to buckling provided by eccentric loading must be neutralized or compensated for in order for the cylinder walls to resist buckling without support from the bore.

In both Figures 8 and 9 the 7000 g curve applies for the maximum loading condition and the 1400 g curve for the conditions at the beginning of projectile motion when gas leakage may occur due to improper seating. A curve derived by analysis for strength without longitudinal loading is also included for reference.

VI. RECOMMENDATIONS

Contingent upon design decisions, the following items are recommended:

- A. That urgent consideration be given to the questions in Section V.A concerning design problems.
- B. That an effort be made, probably involving testing, to find the stress-strain relation and the value of Poisson's ratio beyond the compression yield point for the motor case steel.
- C. That further effort be exerted, probably involving testing, to determine the strength of the motor case under combined compression and external pressure loads.

- D. That an analytical study be made to determine the transition shape for the wall in order to minimize the effect of eccentric loading, as discussed in Sections III.B, and V.B.
- E. That in future tests the test specimens be carefully documented as to dimensions, heat treatment and other pertinent items before the tests. Exact dimensions and hardness readings on the motor cases taken before testing would have helped in analyzing the results of the recent five inch gun tests. The values for these items given on the manufacturing drawing have tolerances which sometimes make the judgement of small effect impossible.