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TRANSPORTATION VIBRATION ANALYSIS OF THE XM982 PROJECTILE

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ARMAMENT RESEARCH, DEVELOPMENT AND ENGINEERING CENTER

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14. ABSTRACT

Vibration testing, simulating ground vehicle transportation was performed on the Excalibur and resulted in unexpected responses of the projectile. Modeling and simulation was used to replicate the vibration test and determine the merits of different mounting configurations to secure the round. The mounting configurations were compared based on levels of projectile rotation and torque.

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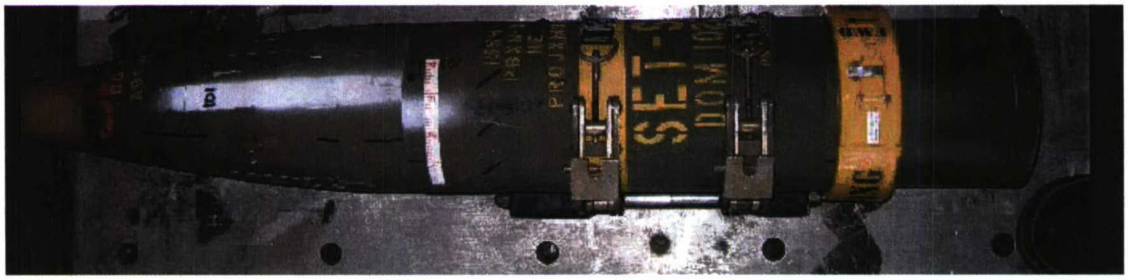
INTRODUCTION

The XM982 projectile must be designed to withstand the vibration environment normally encountered during ground vehicle transport. During laboratory vibration testing, which simulated the vibration of ground vehicle transport, the XM982 projectile experienced unexpected levels of rotation when secured with one strap. In that laboratory test, the vibration was applied in the vertical direction and the excitation was exaggerated to reduce test duration (and keep to a tight test schedule). Modeling and simulation as well as further laboratory experimentation was used to evaluate what mounting configuration and level of strap preload would diminish the projectile's rotation. Laboratory experimentation showed that the rotation of the projectile was reduced by using a two strap mounting configuration. However when two straps were used during the testing, a loose joint was discovered between the warhead and the guidance navigation unit [GNU) (fig. 1)].



Mismatched white stripe, following test, indicated joint rotation.

Figure 1
XM982 projectile secured with two straps for tactical vibration test (Yuma, June 2006).

A general purpose finite element package, ABAQUS Explicit, was used to simulate the vibration response of the projectile secured on the cradles. The type of input used in the laboratory test was a random vibration excitation [power spectral density (PSD) versus frequency]. However, an input with a frequency domain can not be used in a finite element simulation because the model contains contact interfaces. Therefore, for the finite element analysis, the input was converted from the frequency domain PSD to the time domain acceleration. A low pass filter (Butterworth filter) was used on the output data to avoid the potential aliasing problem and eliminate the high frequency data (noise) in the results.

Analyses were performed for the one strap and two strap mounting configurations with various preloads applied by the straps between the projectile and the cradles. The following items were analyzed for each mounting configuration and preload:

- Rotation of the projectile (angular displacement)
- Contact force between projectile and cradles due to the preload
- Torque due to the input excitation

TRANSPORTATION VIBRATION ENVIRONMENTS

Vibration-induced stress can cause items to fail to perform their intended function through material fatigue and wear accumulated over time. Two types of laboratory-simulated vibration tests (logistical and tactical) are used extensively in lieu of more time-consuming and less cost-effective methods of loading and installing equipment in various vehicles and operating them for required scenario distances over test courses. The first vibration test is a logistical vibration test which simulates the transportation of Army materiel as secured cargo during logistical shipments; the second vibration test simulates the tactical vibration environment experienced by equipment installed in or on ground vehicles.

The XM982 projectile needs to be subjected to the vibration tests at three orientations (vertical, longitudinal, and transverse). Vibration tests are also categorized into phases which are defined according to the vehicle speeds. Each phase has one specific test schedule (PSD amplitudes as a function of frequency). The test duration for each phase is determined on the basis of required scenario distances and vehicle speeds for field transport. The total test duration for the XM982 projectile is approximately 25 hrs for each orientation. The detailed procedures to develop the laboratory test times, vibration phases, and vibration schedules are defined in ITOP-1-1-050 (ref. 1).

In order to reduce the test duration, an exaggeration factor can be used to increase the acceleration amplitudes applied which will thereby compress the many hours of field environment into fewer hours. The usage of an exaggeration factor is based on the damage equivalence theory of Miners' method. For random vibration environments, the relationship is defined as:

$$\left(\frac{W_1}{W_2}\right)^{\frac{b}{n}} = \frac{t_2}{t_1}$$

in which:

W_1 = real time amplitude (g^2/Hz)
W_2 = laboratory test amplitude (g^2/Hz)
t_1 = real time
t_2 = laboratory test time
b = 9 (generally used endurance curve constant)
n = 2.4 (generally used stress damping constant)

The ratio of W_2 to W_1 becomes the exaggeration factor (EF). For factors greater than 1, the laboratory test time is reduced, and conversely, for factors less than 1, the test time is increased.

The XM982 experienced some unexpected responses during the laboratory tactical vibration tests at phase three vertical orientation (V3) with exaggeration factor of 2. Therefore, the V3 test schedule used for this analysis used an exaggeration factor of 2.

XM982 MOUNTING CONFIGURATION

The current plan is for the XM982 projectile to be installed in the Paladin self-propelled howitzer in up to 10 locations. The location of interest for the testing and analysis is a rack mount consisting of two cradles connected by two bars. Each of the cradles is covered by a thin layer of Plastisol. Attached to the two bars is a latch mechanism that conforms around the outside diameter of the projectile. The latch mechanism consists of a metal latch and a nylon strap. When the latch and strap is engaged, the strap is stretched and applies a contact force between the projectile and cradles which secures the round. Figure 2 shows the single strap mount configuration.

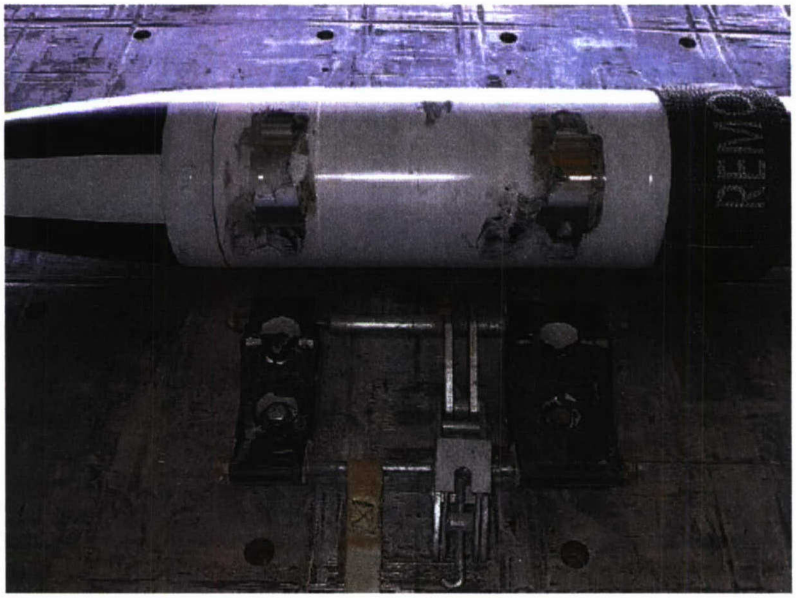


Figure 2
XM982 projectile besides single strap mount configuration used in laboratory vibration test (Yuma, September 2005)

FINITE ELEMENT MODEL

Material Properties, Interactions, and Damping

The projectile and rack mount (cradles, latch, bars, plastisol, and strap/straps) were modeled in Pro/Engineer and imported into the general purpose finite element package, ABAQUS Explicit. The projectile was modeled as a slug, with care being taken to preserve the correct weight, center of gravity, and stiffness of the actual round. 3D solid elements were used to model all the parts of the projectile and cradle with the exception of the nylon strap. The nylon strap has only tensile strength without flexure (bending) stiffness. Therefore, membrane shell elements, which have only tensile strength, were used to model the strap to simulate this unique structural behavior. A mix of tie and surface contact was used. Figures 3 and 4 show the finite element models of the one strap and the two strap mounting configuration, respectively. The coefficients of friction between the interfaces of the latch to the warhead and the strap to the warhead were specified as 0.3 and 0.1, respectively. Tables 1 and 2 summarize the material properties used for the projectile and the rack mount, respectively.

Figure 3
FE model of one strap configuration

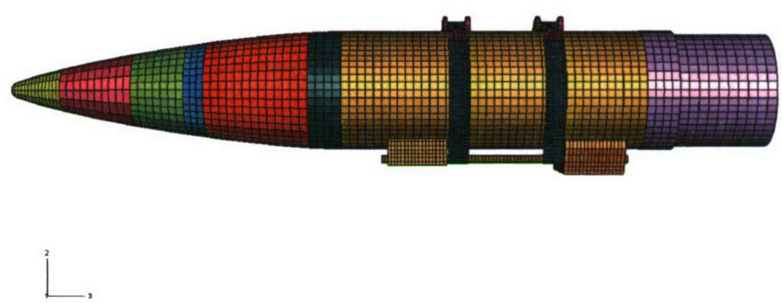


Figure 4
FE model of two strap configuration

Table 1
Material properties of the projectile

Part name/material	Density (lb/in.^{3})	Young's Modulus (psi)	Yield strength (psi)	Ultimate strength (psi)	Strain (%)	Poisson's Ration
Base, Unitary Warhead	0.125	3.0E+7	0.3	-	-	-
CAS, CAS shell	0.120	3.0E+7	0.3	-	-	-
GNC	0.090	3.0E+7	0.3	-	-	-

Table 2
Material properties of the rack mount

Part name/material	Density (lb/in.^{3})	Young's Modulus (psi)	Yield strength (psi)	Ultimate strength (psi)	Strain (%)	Poisson's Ration
Frame (latch) - Carbon Steel, A36	0.283	2.90E+7	35,300	58,000	20	0.26
Link - Carbon Steel, A108 CD	0.284	2.97E+7	53,700	63,800	15	0.29
Hook - Alloy Steel, A331	0.283	2.90E+7	70,000	90,000	15	0.32
Pin - Alloy Steel, A322	0.284	2.07E+7	156,000	170,000	14.6	0.29
Strap - Nylon 66	0.05	4.00E+5	8,000	10,000	25	0.34
Cradle - Al 5083-H321	0.096	1.02E+7	31,000	44,000	12	0.33
Pad - Plastisol	0.0576	2.50E+5	-	-	-	0.34

Rayleigh damping properties were specified for the strap and the Plastisol pads. Two damping factors (mass proportional damping, α_R, and stiffness proportional damping, β_R) were used to specify the fraction of critical damping, ε_i. For a given mode I, the fraction of critical damping can be expressed in terms of the damping factors α_R and β_R as:

$$\varepsilon_i = \alpha_R / 2\omega_i + \beta_R \omega_i / 2$$

in which:

ω_i = Angular (circular) frequency

Table 3 shows the mass damping and stiffness damping and calculated critical damping ratio at a frequency of 125 Hz.

Table 3
Damping properties for strap and Plastisol pad

Part name	Mass damping Alpha, α_R	Stiffness damping Beta, β_R	Damping ratio ε_i, %
Strap - Nylon 66	125	5.0E-5	10
Pad - Plastisol	58	5.0E-5	6

Preload and Applied Loads

To simulate the stretching and thereby the preload in the strap restraining the projectile, the latch was slightly shortened using an artificial thermal load and coefficient of thermal expansion in an early step of the analysis (fig. 5). The preload was then measured from the resultant contact force between the projectile and cradles.

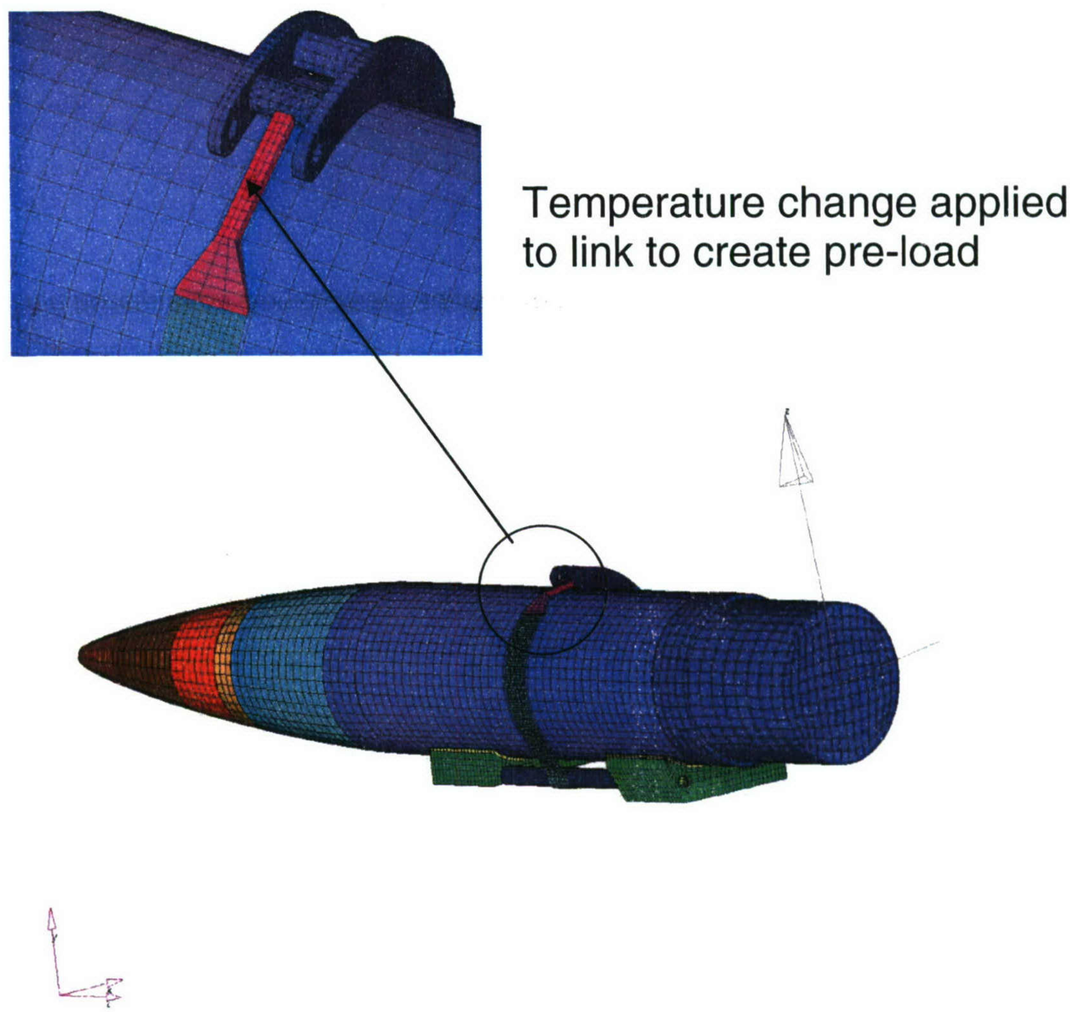


Figure 6
Preload - thermal load

The vertical phase 3, V3, vibration test schedule, figure 6, (frequency domain) was the excitation used in the laboratory test when the projectile responded in an unexpected level of rotation. To use this level of excitation in the analysis, the test schedule was converted to time domain acceleration (ref. 2), figure 7. In order to reduce the computer runtime, only 0.3 sec duration with a 2000 Hz sample rate was used to generate the time domain acceleration. Due to the sample rate and short duration, the actual PSD input, figure 8, generated by the time domain deviates from the test schedule. The time domain acceleration was applied to the base of the cradles (fig. 9) for the analysis.

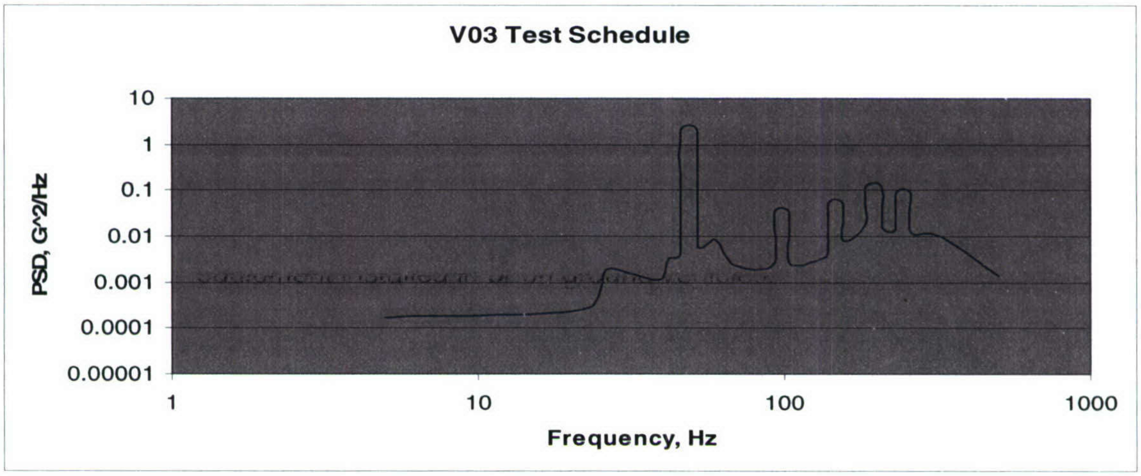


Figure 6
Vibration phase 3, V03, test schedule

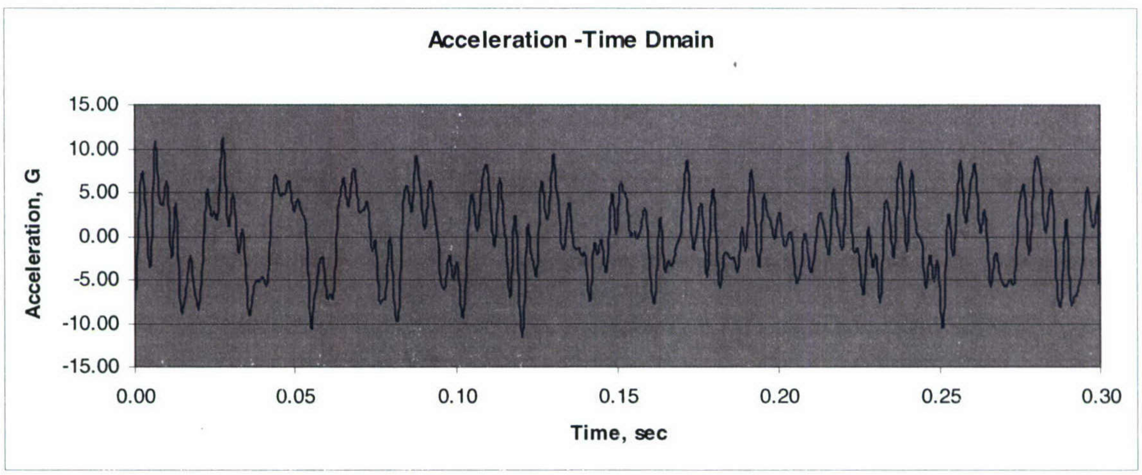


Figure 7
Input time domain acceleration

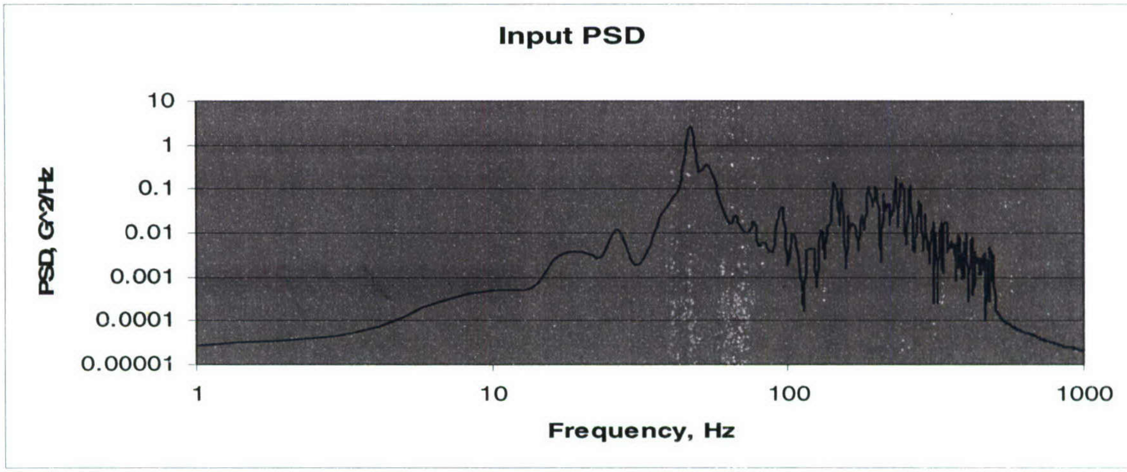


Figure 8
Input PSD generated by input time domain acceleration

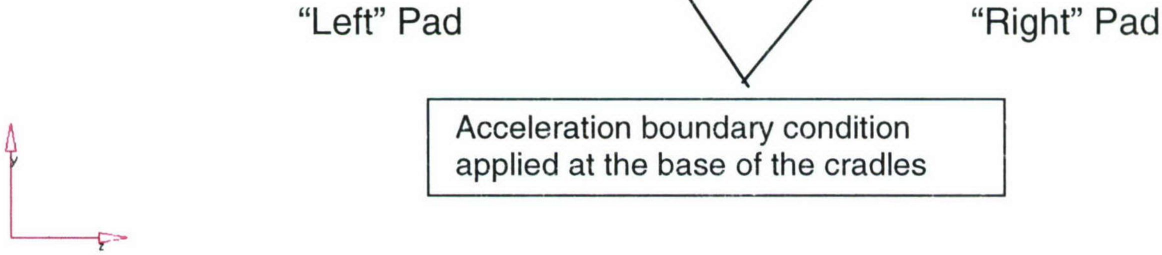


Figure 9
Acceleration applied on the base of cradles

ANALYSIS RESULTS

The primary purpose of this analysis is to determine the rotation of the projectile (angular displacement) and torque induced by the rotation of projectile. The quantitative value of the angular displacement and torque can not be directly extracted from the ABQUAS.odb (solution) file. However, these can be calculated based on the nodal responses of rectilinear displacement and acceleration from the ABAQUS resulting data. Appendix A shows the mathematics formulation to calculate the angular displacement and torque. This formulation assumes that the projectile moves as a rigid body with a small angle of rotation; otherwise the nodal deformation and acceleration would create artificial results.

The time history output of the displacements and accelerations on the projectile were requested from the ABAQUS.odb file. In order to avoid a potential aliasing problem in the resulting data, a higher sampling frequency of 1.E+6 (1 million) Hz was used to collect the time history data. Aliasing is a loss of valid results data. Aliasing will occur if the sampling frequency (the frequency at which the data are saved) is less than twice the highest frequency expected in the results. Then, a low-pass filter (Butterworth filter) was used to exclude the high-frequency noise (numerical and impact).

The contact force, angular displacement, and torque due to the input acceleration will be discussed in the following sections.

Contact Force

The contact force between the projectile and the mount, and thereby the preload of the strap, was measured using the "total force due to contact pressure" history output from the analysis. A Butterworth filter, with a cut off frequency of 100 Hz, was used to filter the output data. A low cut off frequency was used for the filter, based on the assumption that all movement of the projectile while the strap is being tightened is rigid body movement which is characterized by low frequencies.

As was described previously, the tightening and preload of the strap was simulated by slightly shortening the metal latch using an artificial thermal load and coefficient of thermal expansion. Four cases were included in this study: combinations of one strap and two straps and shortening the metal latch by 0.173 in. and by 0.183 in.

The resultant contact forces from preloading the strap or straps can be seen in table 4. A mounting configuration with two straps, more than doubled the total contact force, in essence a doubling of the preload of the straps. In addition, shortening the latch by an additional 0.010 in. from 0.173 in. to 0.183 in., simulating pulling the strap tighter, and significantly increased the total contact force.

Table 4
Effect of number of straps and strap preload on the contact force between the projectile and the mount

Number of straps	Thermal (displacement) load (in.)	Total contact force (lbs)
1	0.173	95
2	0.173	195
1	0.183	123
2	0.183	310

Angular Displacement

The projectile was secured to the cradles by the strap/straps with an applied preload. The analysis results showed that the projectile rotated and lifted off from the cradles for all the cases (fig. 10). This indicated that the preload was not enough to overcome the force induced by the input acceleration. High-amplitude and high-frequency accelerations, caused by the impact between the projectile and cradles, were noticed on the acceleration time history output.

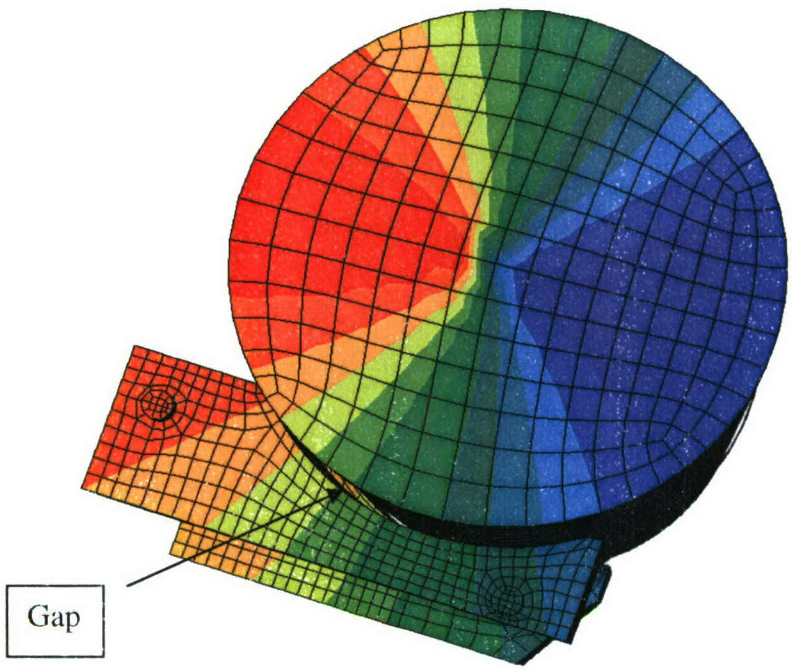


Figure 10
Appearance of gap between projectile and cradle forewarning projectile impact with cradle

Angular displacement of the projectile was calculated from the difference of linear displacement of opposing nodes on the projectile circumference divided by the projectile diameter (app. A). The following two figures, Figures 11 and 12 show the time history of the angular displacement for the one strap configuration with a 0.173 in. strap preload and a 0.183 in. strap preload and the two strap configuration with a 0.173 in. strap preload and a 0.183 in. strap preload. The two strap configuration shows lower angular displacement than the one strap configuration.

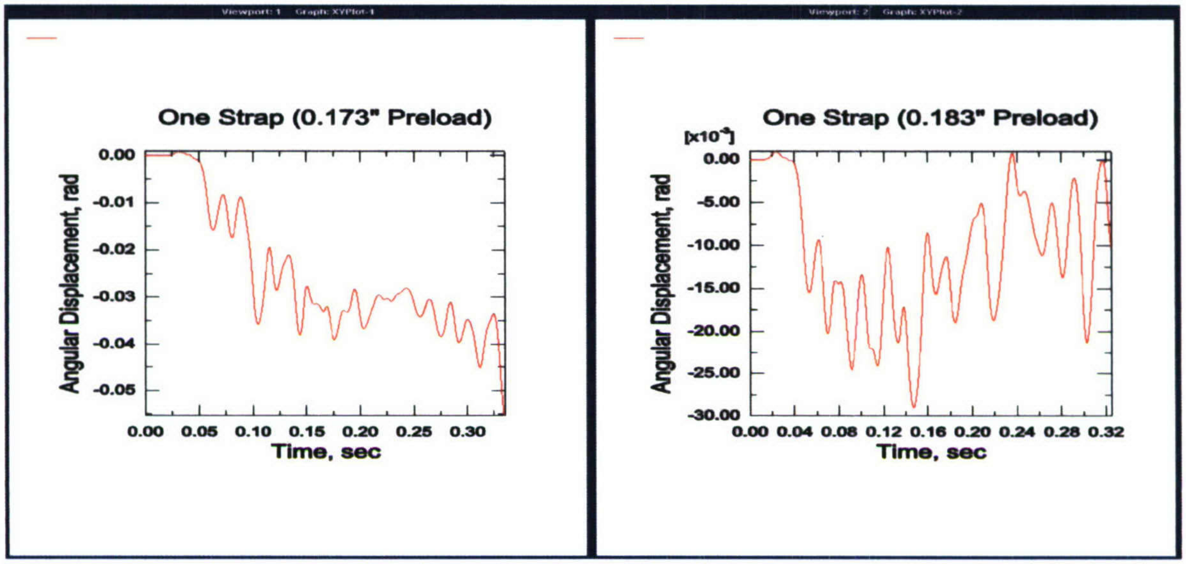


Figure 11
Angular displacement (radians) versus time (s) for projectile secured in one strap configuration with 0.173 in. strap preload and 0.183 in. strap preload

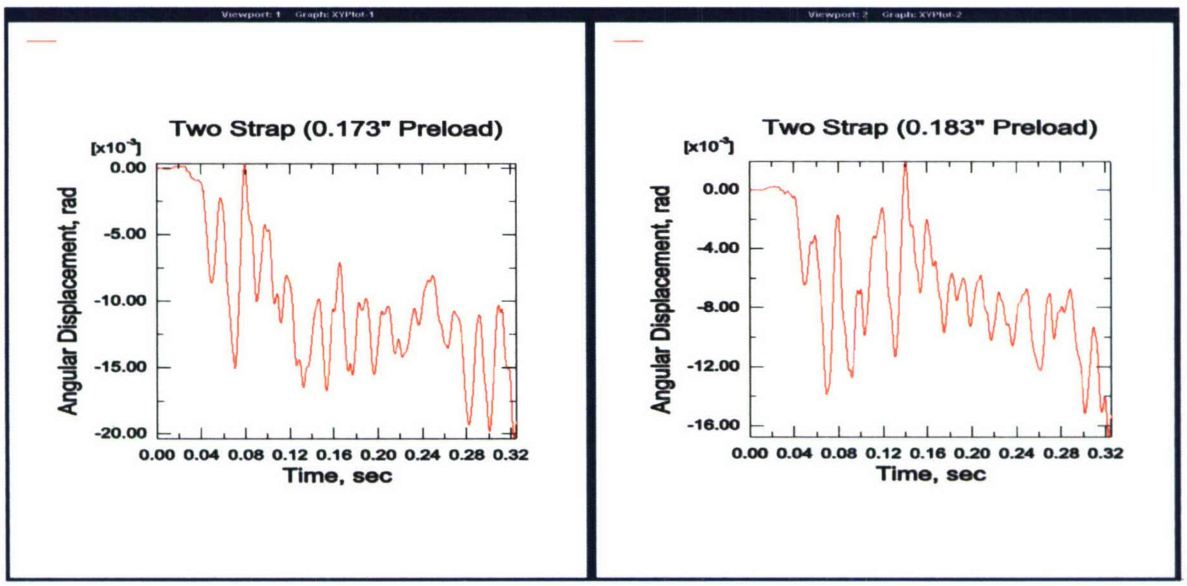


Figure 12
Angular displacement (radians) versus time (s) for projectile secured in two strap configuration with 0.173 in. strap preload and 0.183 in. strap preload

Torque

The torque of the projectile, as mentioned previously, could not be extracted directly from the ABAQUS output file. Therefore, it was calculated based on the nodal responses of four equidistant nodes on the projectile circumference in the area of the warhead and the guidance navigation unit (the area of joint loosening during testing). The torque was calculated using the difference of linear acceleration in the horizontal direction between two opposing nodes (a_1-a_2) and the difference of linear acceleration in the vertical direction between two opposing nodes (a_3-a_4). See appendix A for the torque equation.

The torque was calculated initially without filtering the output data. The resulting torque values were much higher than expected and the torque value in the vertical direction was about double the value in the horizontal direction. This was due to the inclusion of impact which was primarily in the vertical direction. Once the data was filtered, to remove the effects of the high frequency impact, the torque more closely matched between the horizontal and vertical directions and had a more reasonable value (table 5).

Table 5
Effect of number of straps and strap preload on the projectile torque

Configuration	Cut-off-frequency (Hz)	Torque (a_1 -a_2) (ft-lbs)	Torque (a_3 - a_4) (ft-lbs)
1 strap - 0.173 in. preload	No filter	1070	2060
1 strap - 0.173 in. preload	62500	329	270
1 strap - 0.183 in. preload	No filter	920	2050
1 strap - 0.183 in. preload	62500	460	417
2 straps - 0.173 in. preload	No filter	725	1665
2 straps - 0.173 in. preload	62500	243	198
2 straps - 0.183 in. preload	No filter	653	1690
2 straps - 0.183 in. preload	62500	200	185

As can be seen from the data in table 8, the two strap configuration resulted in a lower torque than the one strap configuration. In addition, a higher preload with the two strap configuration lowered the torque as well. It was unexpected for a higher preload configuration to have a higher torque, as is seen in a few cases in the above table, however, this may be attributed to a larger angle of rotation in those instances which would have violated the initial assumptions for the torque equation to be valid.

DISCUSSION

Replicating and predicting experimental results for this type of problem was exceedingly difficult due to inherent uncertainties, such as material properties, damping coefficients, friction coefficients, and level of applied preload. For this reason, results of this analysis should only be used for comparison between models and should not be used to predict actual experimental results.

Based on a model comparison, results for contact force, angular displacement and torque all indicated more favorable outcomes for the two strap mounting configuration with a higher preload than for the one strap configuration with a lower preload. As expected, the two strap configuration with a higher preload exhibited a significant increase in the total contact force over the one strap configuration with a lower preload. In addition, the two strap configuration showed a lower angular displacement than the one strap configuration and the two strap configuration resulted in a lower torque than the one strap configuration. A higher preload with the two strap configuration lowered the torque as well.

That the two strap configuration had a lower torque than the one strap configuration would seem to contradict the actual experimental results, where the issue with a joint loosening only occurred in the two strap configuration. However, as mentioned, the specific material properties of the strap and cradle and the strap preload in the experiment was not measured and remains an unknown variable.

CONCLUSIONS

In conclusion, results from this model comparison show that projectile movement and torque can be reduced by increasing the preload and adding more straps. Experience with vibration indicates that to further decrease projectile movement it would be recommended to stiffen the straps; increase the interface friction; and use soft and highly damped material for the cradle cushion to attenuate input. Caution however should be exercised whenever modifying the boundary conditions of an object undergoing vibration testing. Stiffening the mounting configuration may shift the natural frequency of the projectile up, moving the natural frequency to a frequency corresponding with a random-on-random narrow band in the input vibration schedule. This would create more vibration problems for the projectile. Therefore, further testing is recommended to prove out any configuration changes.

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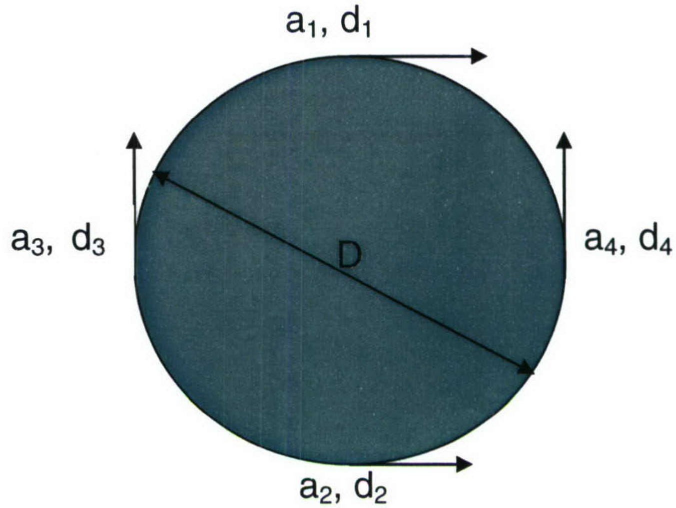
1. FR/GE/UK/US International Test Operations Procedure (ITOP) 1-1-050 Development of Laboratory Test Schedules, 6 June 1997.

2. Irvine, T., A Method for Power Spectral Density Synthesis Revision B, 2000.

APPENDIX A
CALCULATION OF ANGULAR DISPLACEMENT, ANGULAR ACCELERATION, AND TORQUE

Linear Acceleration: a_1, a_2, a_3, a_4
Displacement: d_1, d_2, d_3, d_4
Diameter: D
Weight: W
Gravity Acceleration: g
Mass: $m = W / g$
Angular Displacement: $\theta = (d_1 - d_2) / D$
or $(d_3 - d_4) / D$
Angular acceleration: $\alpha = (a_1 - a_2) / D$
or $(a_3 - a_4) / D$
Torque: $T = \alpha * m * (D^2 / 4)$



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