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Letter to the Editor

Dynamic characteristics of artillery shells

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1. Introduction

This paper presents the dynamic characteristics of a M549 155MM artillery shell. The scope of this study involved finite element modelling of an artillery shell and its validation with experimental results. Artillery shells filled with mock explosives and propellents were used during hardware evaluations. A detailed finite element model was developed that adequately represented structural stiffness and mass in the frequency range of interest. Natural frequencies and modeshapes calculated from the model were validated with test results, and good agreement was obtained for most modes; however, contact non-linearities affected the correlation of modes that involved localized relative motions at component interfaces. Component modal energies were calculated to understand correlation discrepancies and to estimate energy dissipation loss factors. A consistent behavior was seen when other types of artillery shells were tested and analyzed.

2. Model description

The artillery shell was modelled as an assembly of seven components, namely fuse, shell, nozzel, support, mock high explosives (HE), and mock top and bottom propellents. A Young's modulus (Y) of 69 GPa and a density (ρ) of 2770 kg/m^3 was used for aluminum components: fuse, nozzle and support, and $Y = 207$ GPa and $\rho = 7850$ kg/m^3 for the steel shell. Mock explosive properties ($Y = 2.6$ GPa, $\rho = 1607$ kg/m^3) from Ref. [1] were used to represent the HE and propellent. Viscoelastic affects were ignored for the real eigenvalue analyzes. The Poisson's ratio of 0.3 was used for all components. A finite element model with a total of about 30,000 solid elements was constructed. A full contact was modelled between all components assuming that they were welded together. A real eigenvalue analysis [2] was performed to calculate natural frequencies and modeshapes of the assembled system.

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Table 1
Natural frequencies (Hz) of a M549 artillery shell

Mode	Test	Model		
		A	B	C
1B[a]	1200	1113	1197	1209
1O	2100	3278	2097	2109
2B	2400	2204	2415	2422
1T	2620	2442	2591	2598
1A	3270	2885	3218	3226
2O	3310	4137	3301	3309
3B	3660	3481	3933	3938
Weight (*lb*)	93	95	73	95

[a] B = bending, O = ovaling, T = torsion, A = axial.

Table 2
Modal damping ratios from test (% ζ)

Mode						
1B	1O	2B	1T	1A	2O	3B
2.10	1.67	0.56	0.29	0.98	0.31	0.99

3. Model validation

A modal survey was conducted on a freely supported artillery shell with an inert fuse and mock HE and propellent. Eighteen accelerometers were used to measure the response of the shell and an instrumented hammer was used to apply the excitation. To obtain sufficient spatial density of the modeshapes, accelerometers were moved to various locations on the artillery shell. Multiple impact points were used to assure all modes of interest were excited. Frequency response functions (FRFs) were estimated over a frequency range of 0−4000 Hz. The natural frequencies and the damping ratios extracted from the test data are listed in Tables 1 and 2, respectively. Three bending modes, two ovaling modes, a torsion and an axial mode were found within the frequency measurement band of interest. Orthogonal bending and ovaling modes were also extracted but are not presented in Table 1. In addition to frequencies and damping ratios, modeshapes were also extracted from the test data and were visually correlated with the analysis modeshapes, and then quantitatively compared using modal assurance criteria.

4. Calculation of natural frequencies

The natural frequencies of the system were calculated using a real eigenvalue solver [2] and are listed under Model A in Table 1. In comparison with the test, the model under-predicted most

Table 3
Percent modal energy distribution

Component	Mode						
	1B	1O	2B	1T	1A	2O	3B
Mock HE	1,12[a]	69,28	3,15	1,9	5,17	52,26	6,20
Shell	97,68	31,72	94,62	98,86	92,68	48,73	86,53
Fuse	0,16	0,0	2,19	0,1	0,5	0,0	8,22
Support	0,0	0,0	0,0	0,0	0,1	0,0	0,0
Nozzle	0,1	0,0	0,0	0,0	0,1	0,0	0,0
Mock T.Prop	1,1	0,0	1,3	1,2	2,3	0,0	1,4
Mock B.Prop	0,1	0,0	0,0	0,2	0,4	0,0	0,1

[a] Each pair indicates % strain energy, % kinetic energy.

modes but grossly over-predicted the ovaling modes. To better understand this discrepancy, strain and kinetic energies were analytically calculated for each mode.

The modal energy is essentially a measure of structural stiffness and mass associated with a particular mode. The modal strain energy, E_{MSE} is calculated as $\frac{1}{2}\{\phi_i\}^{\mathrm{T}}[K]\{\phi_i\}$, and modal kinetic energy, E_{MKE} as $\frac{1}{2}\omega_i^2\{\phi_i\}^{\mathrm{T}}[M]\{\phi_i\}$, where $[K]$ is the structural stiffness matrix, $[M]$ is the structural mass matrix, and ω_i and $\{\phi_i\}$ are the natural frequency and the modeshape of the ith mode, respectively. Table 3 shows the strain and the kinetic energy contribution of each component as a percentage of the total energy of the structure, and are listed together as pairs for each mode. Because these are percent energies, their sum total for all components adds up to 100 for any given mode, although the sum of energies in Table 3 might be slightly less than 100 because of the roundoffs. The following conclusions could be drawn from the energy distribution in Table 3. The mock HE, shell and fuse are the three most active components in the structure and that most of the strain energy is in HE for ovaling modes, and in the shell for all other modes.

A second case was run without any mock material components i.e., with no HE, top and bottom propellents. The results are listed under Model B in Table 1. The removal of the mock material affected all the modes and this case compared very well with test frequencies. This closer correlation of Model B to the test (as compared to Model A) led to an important conclusion that mock HE and propellents were not strongly coupled with the rest of the structure in the test hardware, and hence violated the modelling assumption of a full contact at interfaces between mock material and the shell. Physically, this decoupling made sense because the mock material had a waxy characteristic and did not adhere well to the shell walls. The full contact assumption made Model A overly stiff for ovaling modes, as well as slightly heavier for all other modes.

The frequencies predicted by Model B agreed well with the test frequencies, although the total model weight, as shown in the last row of Table 1, was off by 22 pounds as compared to the actual physical weight of the test assembly. To overcome this discrepancy, a third model, Model C was constructed with mock HE and propellents included back into the model, but connected to the shell with soft springs to represent a loose coupling at shell–mock interface. Not only Model C predicted frequencies close to the test but also had the total weight comparable to the actual physical weight.

Table 4
Contribution of material damping (% ζ)

Approach	Mode						
	1B	1O	2B	1T	1A	2O	3B
Strain energy	0.1059	3.4391	0.1891	0.0995	0.3709	2.6059	0.3303
Eigensolution	0.1059	3.4347	0.1893	0.1000	0.3694	2.6042	0.3297

5. Calculation of loss factors

The loose coupling of mock material not only affected the modal mass and stiffness of the model but also the modal damping extracted from the test. The test damping ratios shown in Table 2 predominantly accounted for energy dissipation due to Coulomb effect at joints and not so much due to the viscoelastic effect of the mock material, which would add to the overall damping of the system depending on how well the mock material bonds with the steel shell. The contribution of mock material damping to the overall damping was analytically calculated using the following two approaches.

The first approach used strain energies to calculate viscoelastic effect of mock components and its contribution to each mode. The material energy loss contribution was estimated by the following strain energy relationship [3]:

$$\eta_{mode} = \eta_{mock} \frac{(E_{MSE})_{mock}}{(E_{MSE})_{TOTAL}}, \tag{1}$$

where η_{mode} is the loss factor associated with a mode for a given mock material loss factor η_{mock}, $(E_{MSE})_{mock}$ is the sum total of the modal strain energies of mock components i.e., HE, top and bottom propellents, and $(E_{MSE})_{TOTAL}$ is the total strain energy of the mode. $(E_{MSE})_{mock}$ was calculated from Table 3 and a value of $\eta_{mock} = 10\%$ was used from Ref. [1]. Loss factors calculated using the above equation were converted to modal damping ratios using the relationship $\zeta_{mode} = \eta_{mode}/2$ and are shown in Table 4. These results were confirmed by a second method that used a complex eigensolution approach [2]. For this approach, a structural damping (g) input of 10% was used for the mock components, ignoring any other form of damping effects for other components and their interfaces. It can be seen from Table 4 that the damping ratios calculated from the two methodologies agreed well, and as expected the material damping contributed mostly to the ovaling modes. Therefore, for a system with strong coupling between mock material and steel shell, the material damping (in Table 4) could be added to the test modal damping (in Table 2) to obtain the total modal damping of the assembly.

6. Conclusions

This paper summarizes the dynamic modelling and validation of a M549 155MM artillery shell. The correlation of natural frequencies and mode shapes was affected by the loose coupling of mock material filled inside the steel shell. Component modal energies were calculated to understand modal stiffness and mass distribution in the structure and to estimate energy loss factors. The analysis and testing procedure outlined in this paper when repeated for other types of artillery shells showed similar dynamic characteristics.

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