An Evaluation of Structural Analysis Methodologies for Space Deployable Structures

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Benchmarks are introduced for evaluating the performance of numerical simulations of space deployable structures. These benchmarks embody the key challenges of interest to future large space deployable structures, including large angle motion, contact between flexible bodies, and the presence of both soft and stiff mechanical components. The benchmarks were used in companion studies to evaluate the ADAMS multibody dynamics code, the LS-Dyna nonlinear finite element code, and the Sierra large-scale parallel nonlinear finite element code. In the past, only multibody codes would have been considered for this application. This study found that all three codes could be used for these benchmarks, a finding that may lead to larger scale, higher fidelity simulations in the future.

Nomenclature

\[ L_p = \text{pendulum length} \]
\[ L_1 = \text{pendulum length from support to the barrier at initial contact} \]
\[ L(\theta) = \text{time varying pendulum length, from support to the barrier} \]
\[ \theta = \text{rotational angle of the pendulum} \]
\[ \theta_0 = \text{initial angle of the pendulum} \]
\[ \theta_1 = \text{angle of the pendulum at initial contact with the barrier} \]
\[ \alpha = \text{angle of the barrier} \]
\[ EI = \text{bending stiffness of the pendulum} \]
\[ \mu = \text{coefficient of friction between the pendulum and the barrier} \]
\[ m_b = \text{mass of the pendulum link} \]
\[ m_t = \text{mass of the pendulum tip} \]
\[ y(x, \theta) = \text{deflection of the pendulum along the y-axis} \]
\[ z(x, \theta) = \text{deflection of the pendulum along the z-axis} \]
\[ P(\theta) = \text{contact force between the pendulum and the barrier} \]
\[ U(\theta) = \text{strain energy of the pendulum} \]
\[ PE(\theta) = \text{potential energy of the pendulum} \]
\[ W_f(\theta) = \text{work done due to frictional force} \]
\[ g = \text{gravitation constant} \]
\[ E = \text{Young’s modulus of elasticity} \]
\[ G = \text{shear modulus} \]
\[ \rho = \text{density} \]
\[ \nu = \text{poison ratio} \]

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I. Introduction

There are many different types of deployable structures for space applications. The performance of these large lightweight deployable structures is determined by analysis and test for three distinct configurations: stowed, during deployment, and deployed. The analytical techniques for predicting stowed and on-orbit deployed performance are well established. However, the prediction of on-orbit deployment behavior is not well developed due to difficulties in analyzing these complex and highly nonlinear systems. In addition, analysis is playing a larger role in verification prior to flight, as an augmentation to test, and to help anticipate and understand anomalies. Also, ground test of these large flexible structures is often limited, since the effects of gravity and gravity off-loading mechanisms mask their true dynamic behavior [1]. Therefore, high fidelity deployment modeling and analysis for large deployable structures is an essential capability for predicting its structural behavior.

During the past year, an internal research and development (R&D) activity has been carried out at JPL to examine the capability of different analytical tools for analyzing and characterizing the deployment of these large flexible structures. The overall analytical challenge was to enable modeling of not only nominal behavior, but off-nominal, possibly anomalous behavior, so as to estimate the performance margins and uncertainties.

The class of deployable structures of interest exhibit features such as the lenticular joints used in MARSIS [1], mechanical joints, with or without free play (spherical, cylindrical, sliding), and soft goods (tapes, cable, mesh and fabrics). The deployment of these space structures is achieved by a combination of stored strain energy release (controlled by dampers, or uncontrolled and dynamic [1]), and motor actuation through gears, linkages, or cables [2]. These present challenges for a mechanical simulation of the deployment.

This paper and three companion papers [3,4,5] discuss the findings of this R&D to date and provide recommendations for further improvement.

Traditionally, these space deployable structures are analyzed using multibody dynamics solvers that allow large angle motion and large displacements. Also, a hybrid method has been used that incorporates empirical component nonlinear FE results or test data within a multibody dynamics solver as a simplified part [1,2].

In recent years, the distinction between multibody codes and FE codes has blurred. Multibody codes have started adding reduced order flexible bodies (derived from FE models), while nonlinear FE codes have improved their ability to allow large angle motions. Both types of codes have added contact as well. The remaining distinction between a multibody code and a FE code is perhaps that multibody codes define large structural deformations as small strains with respect to a network of body coordinate frames that can undergo large motions. In contrast, FE codes use fully nonlinear strain-displacement formulations that are valid for arbitrary motion with respect to a global coordinate frame. FE codes are thus the more general of the two, but possibly at the expense of computational speed on the types of problems for which multibody codes are intended. Multibody codes may solve their targeted simulations faster, but at the expense of incorporating reduced order structural models. However, because they are general purpose codes, nonlinear FE codes have also been developed recently that exploit large-scale parallel computing, opening the possibility of having both enhanced model speed with enhanced model fidelity. This is one of the questions that motivated the study reported here.

An overview of such capabilities is provided through multiple preselected analytical benchmark problems relevant for space deployable structures using three available solvers at JPL. The selection of the three codes is based on available expertise for particular codes and does not preclude that other codes have the same or better capabilities. The codes used in this paper are: (1) MSC/ADAMS (MacNeal-Schwendler/Corporation Automated Dynamic Analysis of Mechanical Systems) [6]; (2) LS-Dyna (Livermore Software Technology Cooperation) [7]; and (3) the government developed Sierra Solid Mechanics (SM) code (Sandia National Laboratories) [8,9]. Details of each implementation are presented in the companion papers, references [3,4,5].

Four benchmark problems are presented to evaluate the capabilities of the chosen codes. Some of those problems have known closed-form solutions, while others have only numerical solutions. The benchmarks are: (1) rotation of a simple flexible pendulum impacting an inclined deformable bar, (2) a flexure sliding over a bump, (3) dynamics of three masses connected with highly flexible straps, and (4) fabric contacting flexible straps. Note that this paper presents primarily the benchmark problem definitions and an overview of typical results. The companion papers [3,4,5] provide detailed information about the formulation specifics within each of the three codes.

II. Selected Software and Numerical Features

The following sections discuss some key distinctions between the codes and the simulations implemented for this study.
A. Element Types

Multibody codes, such as ADAMS, do not directly incorporate finite elements. While they can represent small deformation flexibility with respect to body fixed coordinate frames, they rely upon imported FE model results in order to formulate their equations of motion. In the case of ADAMS, this is ordinarily done using NASTRAN component mode and/or modal models. Hence the entire NASTRAN library of elements is implicitly available to ADAMS.

Nonlinear FE codes, such as LS-Dyna and Sierra, include extensive element libraries as well, including most of the shell, beam and solid formulations included in NASTRAN. In addition, these codes use elements with unique nonlinear features that stabilize the solution under large strains and motions. Unlike linear codes, these nonlinear FE codes will ordinarily implement reduced order integration formulations for the elements. This is done to expedite the computation at every iteration and every time step. Both LS-Dyna and Sierra, however, include full integration elements as well. In the benchmarks implemented here, the reduced element formulations are used except when noted.

B. Contact Algorithms

ADAMS uses a contact algorithm based on the IMPACT function which is a nonlinear contact formulation. LS-Dyna uses the penalty method for contact. Sierra uses a parallel contact search algorithm in conjunction with an augmented lagrangian penalty method, with particular differences for the implicit and the explicit solvers iteration [8,9].

C. Damping Formulations

It is a common practice to use artificial damping in order to stabilize numerically unstable models and remove high frequency chattering. Artificial damping should be used with special care and may require additional checks to validate the results. Viscous and Raleigh (mass and/or stiffness proportional) damping are used in all three codes considered in this study.

D. Solvers

ADAMS uses an implicit time integrator coupled to a Newton-Raphson nonlinear iterator. LS-Dyna includes both implicit and explicit solvers, though only the explicit solver was used in these benchmarks. Sierra includes both implicit and explicit solvers as well, and the comparison of the two is a major focus of reference [5].

Theoretically, one would always prefer implicit solvers because solution accuracy can be controlled through the selection of a suitable tolerance, while relatively large time steps are allowed. Explicit solvers control stability and accuracy through the use of time steps constrained by the element level elasto-mechanical wave speed, and can be slow for low rate problems. However, implicit solvers can be inefficient for problems involving high velocity impact or high frequency vibration, and for problems with poorly conditioned contact stiffnesses.

III. Benchmark Problems

Modeling a very complex deployable structure in any software requires confirmation that small subsystems from within the large assembly can be modeled properly. These subsystem models should accurately capture the physics of different parts and their interactions. Such a building blocks approach should highlight the strength and shortcoming of software/tool on a very specific problem.

In the following described case studies, a simple subsystem of deployable structures has been separated and studied. In some cases, there are multiple ways of modeling the same physical problem with different methods, even within one software package. The focus of comparison between different modeling methods has been mainly on the feasibility of using it in large-scale models considering accuracy, setup time, and run time.

A. Benchmark Problem 1 – Flexible Pendulum

One of the basic behaviors in a complex deployable structure is rigid body motion of flexible members in addition to their flexible displacement and their interaction with other members. A simple flexible pendulum and a stationary beam-like structure at its path of motion are selected for this study (see Figure 1). This simple time variant structure is selected to study the ability of nonlinear FE software when there is large rigid body motion combined with elastic and localized contact deformations. These types of problems were once problematic for nonlinear FE software [10,11], and were a major reason for the development of multibody codes.
In this benchmark, a pendulum is released from its initial angle \( \theta_0 \) and makes contact to a flexible beam-like barrier at \( \theta_1 \) at its mid-point, \( L_1 \). The barrier is tangent to the pendulum motion at \( \theta_1 \) and has an inclined angle \( \alpha \) with respect to pendulum plane of motion resulting in slow down, stop, and then reversal of the pendulum motion.

In order to capture the physical behavior of the system, the pendulum and the barrier are modeled with flexible elements. These flexible elements allow the pendulum to deform (to the extent of its flexibility) and slide on the barrier after it makes contact. This problem can be considered as a relatively fast motion at the time of contact. Note in this problem a shallow barrier angle \( \alpha \) was selected to prevent the bouncing of the pendulum after the initial impact. This was done so a simple closed form solution could be derived (see Appendix). This type of modification (if needed) is recommended in order to validate the model behavior and gain confidence in the simulation approach.

1. **Element Types**

   The problem is solved using three different element types in FE software: beam, shell, and solid elements. One of the shortcomings of the ADAMS solver is that a modal neutral file (MNF) with shell elements cannot be used where there is contact with another part, therefore only beam elements and solid elements are used in ADAMS.

2. **Physical Data**

   The pendulum physical properties and dimensions are:
   
   \[ L_p = 0.691 \text{ m}, \quad L_1 = 0.338 \text{ m}, \quad EI = 88.96 \text{ N.m}^2, \quad m_b = 0.024 \text{ kg}, \quad m_s = 0.454 \text{ kg}, \quad \theta_0 = 30 \text{ degrees}, \quad \theta_1 = -150.2 \text{ degrees}, \quad \bar{\alpha} = 5 \text{ degrees}, \quad \mu = 0.2. \]

3. **Results and Discussion**

   The results are presented in Table 1 and a typical contact force time history is shown in Figure 2. The maximum travel and contact force varies between different models mainly due to localized contact deformation depending on the element used. If the localized deformation is important due to material behavior or local geometry, then shell or solid elements should be used. A hybrid technique can also be developed with modifying the contact stiffness in the beam elements by determining the local deformation from a FE model using shell or solid elements. This approach could be used to reduce the degrees of freedom (DOF) and computation time. In general, the FE implicit method is robust when there is no contact and becomes computationally difficult during the contact phase [5]. The opposite is true for FE explicit methods. Explicit FE solutions excel during the high frequency impact, but are slow during the pre-contact phase. All three solvers compared favorably with the closed-form solution. References [3,4,5] note additional information about the relative performance of each code as a function of selected solver options.

![Figure 1. Pendulum Model.](image-url)
### Table 1. Pendulum Friction Contact Results Summary.

<table>
<thead>
<tr>
<th></th>
<th>ADAMS</th>
<th>LS-Dyna</th>
<th>Sierra</th>
<th>Closed-Form Solution</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Element Type</strong></td>
<td>Beam (FE_Part)</td>
<td>Beam</td>
<td>Beam</td>
<td>Beam Theory</td>
</tr>
<tr>
<td>Max rotation after contact (deg)</td>
<td>24.7</td>
<td>24.9</td>
<td>25.3</td>
<td>25.3</td>
</tr>
<tr>
<td>Contact Force [N], initial peak</td>
<td>54.1</td>
<td>55.1</td>
<td>60.9</td>
<td>62.4</td>
</tr>
</tbody>
</table>

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**Figure 2. Pendulum Contact Force with Friction – LS-Dyna solver with shell elements.**

**Figure 3. Flexure-Bump.**

### B. Benchmark Problem 2 – Flexure-Bump

The purpose of this benchmark problem is to evaluate the large relative motion between two parts during contact. To study this behavior a single Flexure-bump is considered for analysis. In this model, the flexure is a cantilevered beam with an inclined 55-degree tip, and a moving bump with a 60-degree inclined surface that moves toward the flexure free tip as shown in Figure 3.

1. **Physical Data**
   
   The flexure and bump material are titanium. The flexure is 25 mm long with a 1.3 mm square cross section. The coefficient of friction between the two surfaces was 0.2 for the ADAMS and LS-Dyna simulations, and 0.3 for the Sierra simulations.

2. **Quasi-Static Motion (Slow Moving)**
   
   These types of contact between mechanical parts usually happen over a long period of time, therefore the nature of most cases are going to be quasi-static where it is minimally influenced by mass/inertia properties of moving parts. Because of that, most of the simulations have been done with a fairly slow moving bump of 4 mm/s.

3. **Dynamic Loading (Fast Moving)**
   
   The dynamic behavior of the flexure-bump is also studied to see the effect of speed on impact forces. The linear speed of the bump was increased by an order of magnitude from 4 mm/s to 40 mm/s.

4. **Contact Definition**
   
   Three types of contacts are considered based on the stiffness of the parts in contact: (1) Flex on flex, (2) Flex on rigid, and (3) Rigid on rigid.
5. Results and Discussion

The results for this study with 4 mm/s speed are summarized in Table 2 for all three solvers. Note the higher speed of 40 mm/s had minimal effects on the peak contact force.

ADAMS: Flex on flex: In this method both flexure arm and bump are flexible bodies. Flex on rigid: Moving bump was replaced with a rigid part. Rigid on rigid: Moving bump and flexure tip are rigid parts. The ADAMS solver is much more efficient when dealing with rigid on rigid contact compared to other methods.

LS-Dyna: Both flexure and bump are represented by deformable solid elements for the first model. As shown in Figure 3, both parts have geometry features that require a very fine mesh which resulted in long run time. The small time-step due to very small elements can be effectively mitigated by use of mass scaling which is outside of the scope for this paper. Instead of mass scaling, different combinations of deformable and rigid elements are studied. The shortest run time was achieved by having beam element for flexure, rigid elements for flexure tip and rigid elements for the bump.

Sierra: The Sierra simulation implemented a variable density mesh to study the effects of mesh density and conditioning on the solution speed. The implicit quasi-static solver was found to be 400 times faster on 4 processors than the explicit solver, even with 24 processors. In addition, as reported in [5], the Sierra model was extended by copying the mesh 120 times so as to assess the scaling of the solvers for larger scale parallel solution.

<table>
<thead>
<tr>
<th>Table 2. Flexure-Bump Contact Force with Friction..</th>
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<tbody>
<tr>
<td>Element Type</td>
</tr>
<tr>
<td>Maximum Contact Force [N]</td>
</tr>
</tbody>
</table>

C. Straps and Fabric

There are many types of deployable mechanisms made of thin wall flexible structures, e.g., Lenticular Joint [1] or a network of structural straps making the structural surfaces of reflector mesh antenna [2]. Also, structures with membrane components, e.g., sunshield structures or the reflective surface of a deployable mesh antenna are often made of soft fabric attached to a network of structural straps. Usually each strap is a long flat beam that deflects and stores bending energy similar to compressing a spring during the stowing process. The following two cases are developed to study these types of structures. Note these types of material could have a wide range of axial and bending stiffness. When there is minimal stiffness in absence of a tension field, they are treated as soft goods, such as cable or fabrics. In general, these types of soft goods are modeled accurately when there is a tension filed and have been omitted when there is no tension. This was mainly due to software limitations and their secondary contributions. However, in some recent applications their representation in the deployment model became necessary, mainly for their stored strain energy contributing to bloom force [2] and also studying snag issues.

The next two benchmark problems are selected for this study with following load cases representing stowing and deployment:

1. Quasi-Static Stowing/Deployment With Gravity

   The process of stowing all deployable structures happens very slowly with gravity present and monitored carefully. This is done to protect the sensitive fabric and also to make sure that snagging of straps or strings does not happen. The main reason for running the quasi-static analysis in both stowing and deployment is to make sure the results are repeatable and similar loads are generated in both directions.

2. Dynamic Deployment With and Without Gravity

   During actual deployment of the structure the restraints that hold the stowed structure together are released very quickly. The energy stored in bending and compressing straps and fabric together during the stowing process gets released and pushes out the outer components of the structure. This phenomenon is called “bloom” and is a highly dynamic event. Capturing the behavior of the structure during this event both with gravity and without gravity representing ground and on-orbit deployment is crucial in predicting the internal component loads and also overall satellite attitude.
D. Benchmark Problem 3 - Double Straps

A simple model consisting of two thin elastic straps and three moving carts was built in order to understand their behavior under gravity and no gravity (see Figure 4). Each strap end is connected with a pivot joint to the block above it. The strap connections are 1/4 of the carts’ width away from its edge. This was done to exhibit contact between the straps and the carts.

![Two Straps Setup Model](image)

**Figure 4. Two Straps Setup Model.**

1. **Physical Data**

   Strap dimensions are 0.432 m long and 9.525 mm wide and 0.18 mm thick. The strap material properties are: \( E=41.8 \text{ GPa}, G=16.1 \text{ GPa}, \nu=0.3, \rho=2657.271 \text{ kg/m}^3 \).

2. **Results and Discussion**

   The results for this study are summarized in Table 3 for all three solvers.

<table>
<thead>
<tr>
<th>Table 3. Two Straps Setup – Quasi-Static Stowing with Gravity.</th>
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</thead>
<tbody>
<tr>
<td>Element</td>
</tr>
<tr>
<td>Beam (FE_Part)</td>
</tr>
</tbody>
</table>

Only a snapshot of ADAMS model is shown in Figure 5.

**ADAMS:** Simulation of straps will require large deformation of beam-like elements which fits the description of the FE_Part. Each strap is modeled with 20 elements and strap ends are connected to the cart above it with a pivot joint.

**LS-Dyna:** Shell elements are used to model the thin straps and the carts. Penalty method based revolute joints are used to connect straps to all three carts as shown in Figure 4. Contact is established between two straps in all cases and also between straps and carts during dynamic deployment cases.

**Sierra:** The Sierra model was implemented with shell elements for the straps and solid elements for the carts. The implicit and explicit solvers were both compared on this benchmark. This proved to be the most challenging benchmark for the implicit solver, due to the softness of the contact between the straps. As discussed in [5], a reduced augmented lagrangian factor enabled the implicit solver to converge. However, the speed advantage of the implicit solver was reduced to only about a factor of 2-4 with respect to the explicit solver.
E. Benchmark Problem 4 - Straps and Fabric

A highly flexible structure comprised of five straps and one mesh type fabric is constructed to study its behavior during stowing and deployment. The straps are connected to one another with riveting joints where they cross each other. The fabric is connected to the straps along their lengths at discrete points, in addition to intersection points. There is an additional catenary along the free edge of the fabric to provide needed hoop stiffness. The purpose of this study is to assess solvers capability for this type of problem and ability to maintain stability during stowing and deployment while having contacts between very thin dissimilar materials. Additionally, contact forces and joint connections between straps are assessed. These effects are studied in realistic condition considering gravity effect during stowing, with and without gravity effects during deployment. The case with gravity simulates ground testing and without gravity is equivalent to on-orbit deployment. Displacement is applied in X and Y directions, and gravity is applied in -Z direction. This case was solved in LS-Dyna only mainly due to lack of resource for Sierra and limitations of ADAMS.

1. Physical Data

   The structure is made of two equilateral triangles of length = 0.432 m (see Figure 6).
   Strap: similar to strap properties in benchmark 3.
   Fabric: thickness = 0.0254 mm, $E = 344.7$ MPa, $\nu = 0.3$, $\rho = 553.6$ kg/m$^3$.

2. Results and Discussion

   The results from LS-Dyna solver are summarized in Table 4. A snap shot of the stowed, deploying and deployed structure is shown in Figure 7.
Table 4. Fabric and Straps – Quasi-Static Stowing with Gravity.

<table>
<thead>
<tr>
<th></th>
<th>LS-Dyna Solver</th>
<th></th>
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</tr>
</thead>
<tbody>
<tr>
<td>(Shell and membrane elements)</td>
<td>X-axis</td>
<td>Y-axis</td>
<td></td>
</tr>
<tr>
<td>Maximum Stowing Force [N]</td>
<td>0.35616</td>
<td>0.29912</td>
<td></td>
</tr>
</tbody>
</table>

Figure 6. Fabric and Five Straps Model.

Figure 7. Fabrics and straps in stowed, deploying and deployed configurations, LS-Dyna Solver [3].
The companion papers provide details about the implementations and performance of each of the considered codes. While the distinctions between the codes prevent a fair comparison of performance, overarching conclusions and recommendations can be drawn.

First is the observation that the nonlinear FE codes are just as adequate for modeling large angle motion of flexible bodies as the examined multibody code [10,11]. When multibody codes were first developed almost 30 years ago, nonlinear FE codes were not yet practical for large rotation problems such as these. In the intervening years, this has changed. Features such as hyperelastic material models that can undergo large rotations, and implicit and explicit solvers that can directly handle contact have essentially enabled FE codes to solve the types of problems originally targeted by multibody codes. This issue is no longer a technical distinction.

It is interesting to note that while FE codes have taken on some of the capabilities of multibody codes, the opposite is also the case. ADAMS now allows a reduced order structural stiffness representation of a flexible component, and it also includes contacts. A remaining distinction between multibody and nonlinear FE codes appears to be the use of reduced order modeling versus full order modeling. That is, the multibody codes still depend upon a reduced order nonlinear model of the structural system using body frame degrees of freedom and component modes, while the nonlinear FE codes (by default) solve the full order model using nodal degrees of freedom. This distinction may continue to blur over time.

A compelling, remaining distinction, however, may be that the nonlinear FE codes have been able to exploit multi-processor parallel computing. This is the second major observation from our study. Since multibody codes were originally developed, there have been great strides in the development of parallel solvers for the nonlinear FE codes. LS-Dyna and Sierra both have parallel computing capability. LS-Dyna can demonstrate increased run time efficiency by using up to 192 CPU cores, and newer developments promise the use of larger scale computing. Sierra can exploit any number of processors, up to thousands, if necessary. For the benchmarks examined here, their capabilities largely overlapped, but the observed scalability of Sierra is most promising.

But there remain challenges for all three codes. Overall, the greatest simulation challenge observed was the presence of contact between flexible bodies. All three codes could handle the flexible contact, but required varying degrees of simulation tuning to be convergent in some cases. For example, the Sierra implicit parallel solver was found to function up to two orders of magnitude faster than the corresponding explicit solver for a given number of processors. However, this advantage was found to degrade in the presence of high speed contacts (such as in the pendulum benchmark) and for soft contact (such as in the two-strap benchmark). In those cases, the explicit solver may be necessary for discrete windows of the simulations, and a hybrid approach is suggested in [3] to accomplish this.

Based on these few benchmark problems it is our conclusion that the accuracy of the results depends heavily on the user’s understanding of the software and the physics related to the problem. There are many options available to use in these software/tools in order to get convergent but not necessarily accurate results. The question of understanding the physics become vitally important to separate erroneous results from accurate and acceptable ones. It is dangerous to use these tools without an in-depth understanding of their capabilities and shortcomings, in addition to physical behavior of the case at hand. As these tools become more available and user friendly with more features, end users without much experience start using them and may end up with erroneous results. The hope is that these tools become robust enough to a point where user errors are minimized.

This study at JPL will continue to explore additional capabilities for very large systems, such as massively parallel simulation formulation, optimization of the code options, improvement of the contact problem, and physical modeling of failure cases with anomalies. Finally, the model uncertainties and its application to deployment analysis will be considered. We expect these kinds of simulations to primarily use the nonlinear FE codes, with the multibody codes as an important baseline comparison.

Appendix: Closed Form Solution for Pendulum Benchmark

The intension of deriving a closed form solution is to establish a metric for evaluating the simulation results and not to derive an exact solution. For this purpose, the maximum rotation angle of the pendulum after it contacts the barrier is selected and it is estimated using energy balance equation:

\[ U(\theta) = PE(\theta) - W_f(\theta) \]  

(1)

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The strain energy of the pendulum as a result of sliding on the barrier is determined using a beam theory:

\[
U(\theta) = \frac{1}{2} \left[ \int_{0}^{L(\theta)} EI \left( \frac{d^2 y(x, \theta)}{dx^2} \right)^2 \, dx + \int_{0}^{L(\theta)} EI \left( \frac{d^2 z(x, \theta)}{dx^2} \right)^2 \, dx \right]
\]  \hspace{1cm} (2)

Where \( y(x, \theta) \) and \( z(x, \theta) \) are deflections of the pendulum along the \( y \) and \( z \) axes after it contacts the barrier. After ignoring the higher vibrational modes of the pendulum the deflections are:

\[
y(x, \theta) = \frac{\mu \cdot P(\theta) \cdot x^2}{6 \cdot EI} \cdot (3 \cdot L(\theta) - x) \quad z(x, \theta) = \frac{P(\theta) \cdot x^2}{6 \cdot EI} \cdot (3 \cdot L(\theta) - x)
\]  \hspace{1cm} (3)

Where \( L(\theta) \) is the time dependent length of the pendulum from support to contact with the barrier:

\[
L(\theta) = \frac{L_0}{\cos(\theta - \theta_1)}
\]  \hspace{1cm} (4)

Also, the contact force \( P(\theta) \) is determined using deformation of the pendulum along the \( z \) axis, \( z_c \), due to its contact with the barrier:

\[
P(\theta) = \frac{3 \cdot EI \cdot z_c(\theta)}{L(\theta)}
\]  \hspace{1cm} (5)

where:

\[
z_c(\theta) = L(\theta) \cdot \sin(\theta - \theta_1) \cdot \tan(\alpha)
\]

The potential energy of the pendulum due to its rotation \( \theta \) is:

\[
PE(\theta) = \left( M_t + 0.5 \cdot M_b \right) \cdot g \cdot L_p \cdot \left( \sin(\theta_0) - \sin(\theta) \right)
\]  \hspace{1cm} (6)

The work done on the pendulum due to the resisting friction force while in contact with the barrier is:

\[
W_f(\theta) = \int_{\theta_1}^{\theta} \mu \cdot P(\theta) \cdot L(\theta) \, d\theta
\]  \hspace{1cm} (7)

The maximum angle \( \theta \) is determined from Eq 1 by first substituting Eq 5 into Eq’s 3 and 7, then substituting Eq’s 3 and 4 into Eq 2, and finally substituting Eq’s 2, 6, 7 into Eq 1.

Also, the upper bound of initial peak impact force is estimated using the energy equation:
The maximum rotational angle of the pendulum after initial contact with the barrier is 25.3 degrees and the initial impact force is 62.4 N. These numbers agree well with numerical results presented in Table 1. As a secondary check, the analysis was repeated with \( \mu = 0 \) and the maximum rotational angle increased to 40.5 degrees. This case was analyzed in LS-Dyna, which predicted a maximum angle of 40.9 degrees. Note, the secondary effects of barrier elastic deformation as well as local contact deformations were ignored for these derivations.

\[
F_{\text{cont}} = \sqrt{6(1 + \mu^2)} \cdot \frac{EI}{L(\dot{\theta})^3} \cdot \dot{PE}(\dot{\theta}) \cdot \tan(\alpha)
\]  

(8)

Acknowledgments

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References


2Development of the Large Aperture Reflector/Boom Assembly for the SMAP Spacecraft, Dr. Mehran Mobrem, Edward Keay, Geoff Marks, Dr. Eric Slimko; ESA/ESTEC, NOORDWIJK, THE NETHERLANDS 2-3 OCTOBER 2012


6MSC/ADAMS help documentation, MSC Software Corporation, 2016


