Verification of numerical solutions for the deployment of the highly nonlinear MARSIS antenna boom lenticular joints

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ABSTRACT

The MARSIS antenna booms are constructed using lenticular hinges between straight boom segments in a novel design which allows the booms to be extremely lightweight while retaining a high stiffness and well defined structural properties once they are deployed. Lenticular hinges are elegant in form but are complicated to model as they deploy dynamically and require highly specialized nonlinear techniques founded on carefully measured mechanical properties. Results from component level testing were incorporated into a highly specialized ADAMS model which employed an automated damping algorithm to account for the discontinuous boom lengths formed during the deployment. Additional models with more limited capabilities were also developed in both DADS and ABAQUS to verify the ADAMS model computations and to help better define the numerical behavior of the models at the component and system levels. A careful comparison is made between the ADAMS and DADS models in a series of progressive steps in order to verify their numerical results. Different trade studies considered in the model development are outlined to demonstrate a suitable level of model fidelity. Some model sensitivities to various parameters are explored using subscale and full system models. Finally, some full system DADS models are exercised to illustrate the limitations of traditional modeling techniques for variable geometry systems which were overcome in the ADAMS model.

Keywords: MARSIS, antenna, deployment, dynamic, modeling, lenticular, hinge, ADAMS, DADS, Mars

1. INTRODUCTION

MARSIS (Mars Advanced Radar for Subsurface and Ionosphere Sounding) is one of six instruments on ESA's Mars Express spacecraft, which launched on June 2, 2003, and entered Mars orbit on December 25, 2003. The MARSIS instrument is a long wavelength radar sounder that serves the dual purpose of measuring the Mars ionosphere and searching for evidence of subsurface water. It is designed to make measurements as high as 800 km above the surface when performing subsurface sounding and at altitudes up to 1200 km for ionospheric sensing. The transceiver operates in 1 MHz wide bands centered at 1.8, 3.0, 4.0, and 5.0 MHz which allows MARSIS to search for water at depths up to 5 km below the surface. This is the first time that orbiting radar has been used to probe the surface of Mars.

The antenna for the MARSIS instrument was developed and built by NGST Astro Aerospace in Carpinteria, CA. The structural portion of the antenna is comprised of three Foldable Flattenable Tubes (FFT)™ that are stowed in a cradle for launch and the trip to Mars. The stowed antenna is shown in Fig. 1 and a full description of the basic design can be found in Ref. 1. An illustration of all three booms in their deployed configuration is shown in Fig. 2. The two dipole booms are each 20 m long and are constructed using 1.5" diameter Kevlar and fiberglass tubes with transverse cutouts that form hinge sections as shown in Fig. 3. The 7 m monopole boom is made from a 0.79" diameter tube using the same materials. The actual conducting portion of the antennas is a pair of 22 gage wires in the interior of each boom. The FFT booms are extremely light and space efficient but their deployments are intended only for the microgravity vacuum of space which renders full ground based testing prohibitive. Any significant friction, gravity, or air drag, combined with its large motion during deployment, makes it impossible for credible terrestrial based deployment testing. As a result, the only recourse available to the MARSIS team was an extensive component test program and careful modeling of the deployment.

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The deployment of MARSIS was originally scheduled for April 20, 2004, but was delayed after it was discovered that the previous analysis (done in 2000 during the development phase) had underestimated the deployment dynamics[2,3]. The model used in this first analysis made use of the standard built-in damping formulation in ADAMS. However, in early 2004, during the design and analysis of another FFT antenna boom being built for SHARAD (SHAllow RADar), it was discovered that the ADAMS documentation incorrectly used the term ‘structural damping’ to describe what is actually a Rayleigh damping implementation in the software. As a result of this error, the original ADAMS model of the MARSIS antenna had an unrealistically high modal damping that varied from 188% to 1% as the dipole booms locked into place instead of the intended 1% damping throughout. The resulting dynamics predicted from this overdamped model were benign and well behaved. However, once the damping was corrected, the boom dynamics were shown to be much more energetic with many of the boom hinges experiencing multiple buckling events. The FFT was not designed for this type of buckling behavior and the ensuing uncertain nature of the corrected analysis prompted the delay of the deployment in order to study and better understand its characteristics. If these dynamics had been known during the antenna development then the design would have been changed to remove the hinge buckling phenomenon.

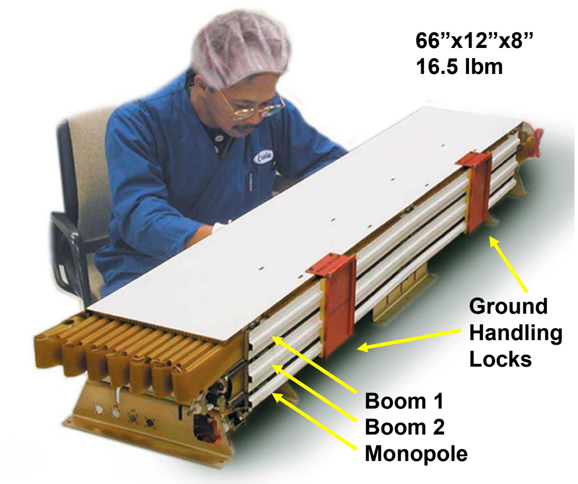

Fig. 1. The three MARSIS antenna FFT booms stowed in their cradle prior to launch. (Photo courtesy of NGST Astro Aerospace.)

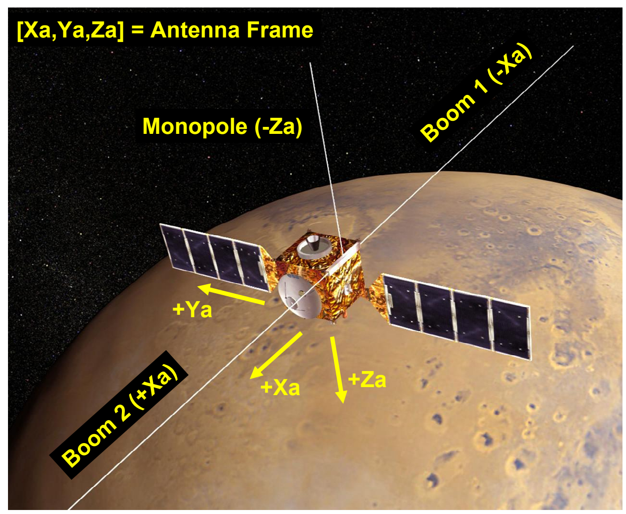

Fig. 2. The two dipole and single monopole MARSIS booms with the deployed antenna reference coordinate system.

Modeling even a single lenticular hinge is challenging due to its highly nonlinear behavior[4,5]. The hinge properties are a strong function of temperature and the combined dynamics of several hinges greatly increases the uncertainty in the dynamics. In particular, the buckling events encountered during the deployment present a special challenge due to the sharp changes in moment which makes the governing equations very stiff and difficult for numerical integrators to track. Unfortunately, the seemingly straightforward task of integrating the deployment model was found to be fragile. In order to build confidence that the ADAMS model was performing accurate calculations, a second model was commissioned using DADS software. A third model was also built using ABAQUS with the objective of querying the hinge buckling mechanism to help explain some of the test results. Results from this suite of models was ultimately used together to estimate the boom dynamics and any risk the booms might present to the spacecraft[2]. A failure modes analysis of the spacecraft was completed using these results and, based on its conclusions, the decision was made to deploy the MARSIS booms in May and June of 2005[3].

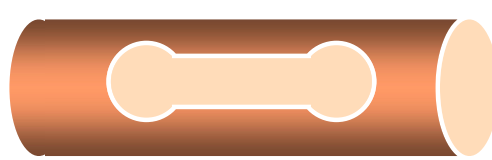

Fig. 3. Approximate geometry of the cut-out of the lenticular joint used in the MARSIS booms.

2. SINGLE PENDULUM ADAMS/DADS MODEL CORRELATION

Another commercially available code, Dynamic Analysis and Design Systems (DADS), was used to verify the on-orbit dynamic responses of the antenna boom deployments predicted by ADAMS. In general, ADAMS and DADS are multi-body dynamic simulation packages which can simulate motions of mechanical systems with nonlinear constraints and forcing functions. Nonlinear equations of motion of the system are derived using either Lagrange's equations or Newton-Euler formulations and solved numerically for motions of large displacements and large angular rotations. The dynamic analysis of an antenna boom deployment serves as a good example of the type of dynamic problems either ADAMS or DADS are well suited to solve.

Results from both codes are generally identical for simulation of rigid body systems if the same forcing functions and similar integration error tolerances are used. Similar conclusions cannot be drawn for simulation of systems with flexible components as there is no unique modeling approach for component flexibility and damping effects. Component flexibility can be represented either by an assumed modes approach or by approaches that employ finite element methods. The former usually retains fewer degrees of freedom in modeling the flexibility but requires some judgment in selecting proper modes. Several FEM-based multi-body dynamic simulation approaches have been used to relieve the difficulty of the modal approach in modeling component flexibility. For a beam-type structure, such as the antenna boom discussed in this paper, the intuitive and easiest way of modeling flexibility would be the beam/lumped mass FEM-based approach as employed by the ADAMS model and shown in Fig. 4. In this approach, each link (straight section between hinges) is modeled as three lumped masses connected by two beam elements. A standard beam element stiffness matrix is used to compute internal elastic forces between the lumped masses based on the relative displacements and orientations between the masses. Proportional (or Rayleigh) damping is usually used to compute the damping force. The hinge opening torque vs. deployment angle is shown in Fig. 5. The hinge properties were measured in a series of tests performed at NGST Astro Aerospace and the resulting average profile was used throughout all simulations.

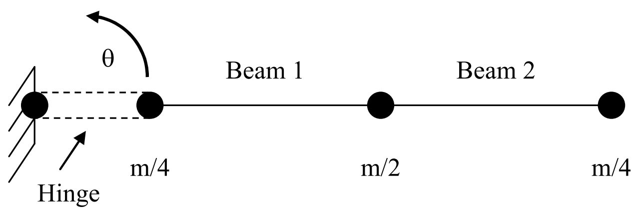

Fig. 4. Lumped mass beam construction used for the single pendulum model.

In this study, a DADS model was used as a second data point to verify the ADAMS model's results. Two approaches were used in modeling component flexibility in the DADS model. The first approach used the same FEM-based method as the ADAMS model used. The second method used a component modes approach to further verify the simulation results and to avoid any oversight in the final selection of a modeling approach. The first four cantilever modes of each link (the first and second bending modes in each direction) were used in the DADS assumed modes approach. A step-by-step procedure, starting from a simple single pendulum problem, moving to a double pendulum problem, and finally analyzing the complete system, was taken to verify the two models.

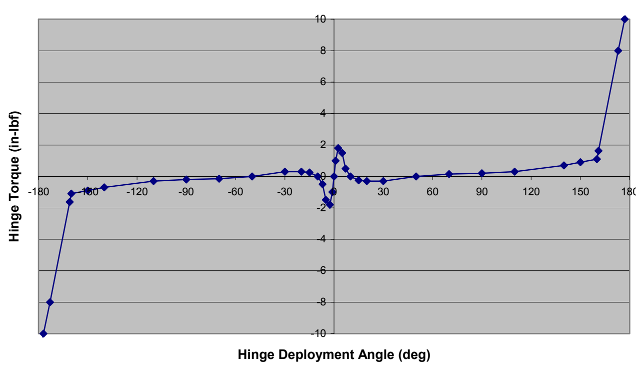

Fig. 5. Average measured hinge deployment torque profile at -70°C.

Several forcing functions that represent energy in the stowed configuration, joint stiffness, viscous damping force (post-buckling hysteresis), and structural damping, were modeled in the simulation. To ensure that these forces were modeled exactly the same way in both the ADAMS and DADS models, a single pendulum problem with only one link and one hinge was used to verify the modeling approach. The link was modeled as three lumped masses with two beam elements using an FEM-based approach as was planned for the full 13 links dipole boom model. Four simulations were tested, each with different forcing scenarios. Simulation results giving the time history of the pendulum angle from these models are shown in Figs. 6 and 7.

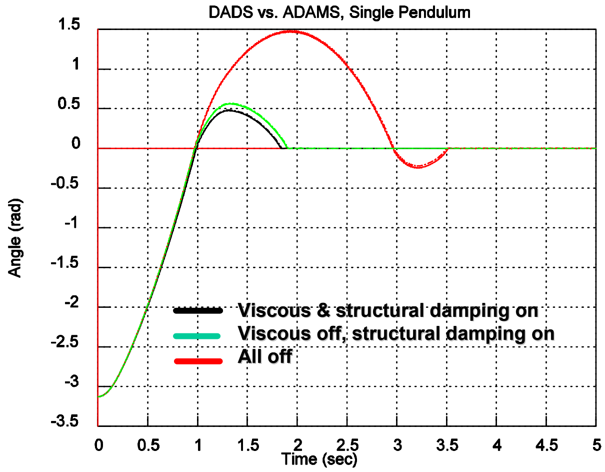

Fig. 6. Comparison of ADAMS and DADS single pendulum results.

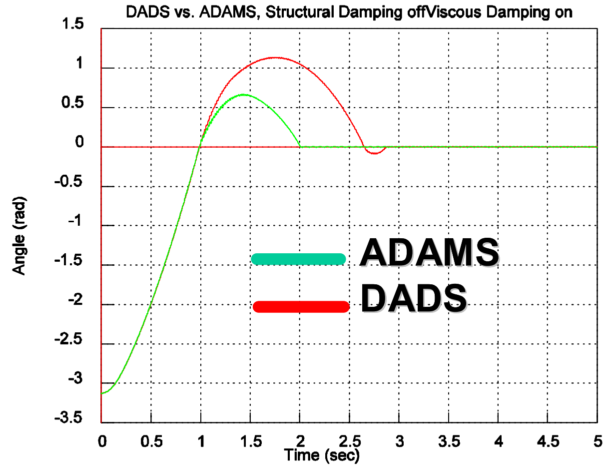

Fig. 7. Comparison of ADAMS and DADS single pendulum results.

Fig. 6 actually contains a total of 6 curves with three solid curves from DADS results and three dashed curves from ADAMS results. Since the ADAMS and DADS models obtained nearly identical results for the three cases in Fig. 6: 1) with both viscous and structural damping on, 2) with viscous damping off and structural damping on, and 3) with all damping forces off, only three curves are actually visible as the ADAMS and DADS results lie right on top of each other. Note that here the term "viscous damping" is used to refer to a secondary post-buckling torque that was included in the model to better match the test measured properties. For the case without any structural or joint damping, both models show that the joint buckles twice (the angle passes through zero) before locking into position, and that it only buckles once for the other two cases.

The only case that ADAMS and DADS didn't match was when the structural damping was turned off and the viscous damping was turned on as shown in Fig. 7. Further investigation of this particular case in DADS revealed that the simulation results were highly sensitive to the initial angle of the pendulum. This chaotic phenomenon is illustrated in Fig. 8 where two runs with slightly different initial conditions give very different results. This result also indicates that the modeling of joint viscous damping (joint hysteresis) would need to be reworked to obtain a more stable result.

In order to test the convergence of the model used for the beam itself, a number of single pendulum DADS models were constructed using two, four, and eight beam elements, as well as an assumed modes model. A comparison of the results obtained from these models is shown in Fig. 9. The plot shows that the beam models do indeed converge as expected. Similar results were also obtained for the modal approach. The relation between theses two approaches will be further explored in the next section.

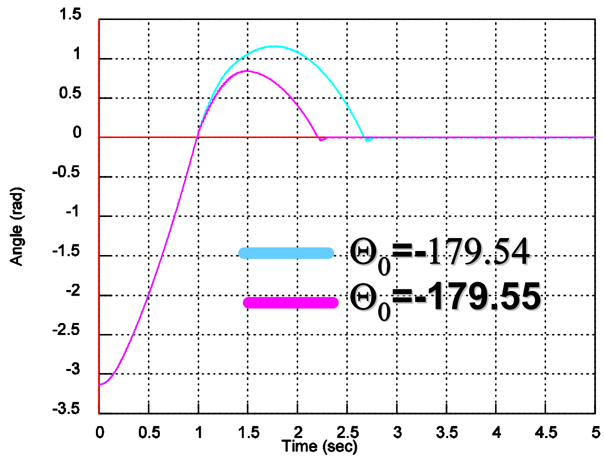

Fig. 8. Instability of single pendulum results modeled in DADS using slightly different initial hinge angles.

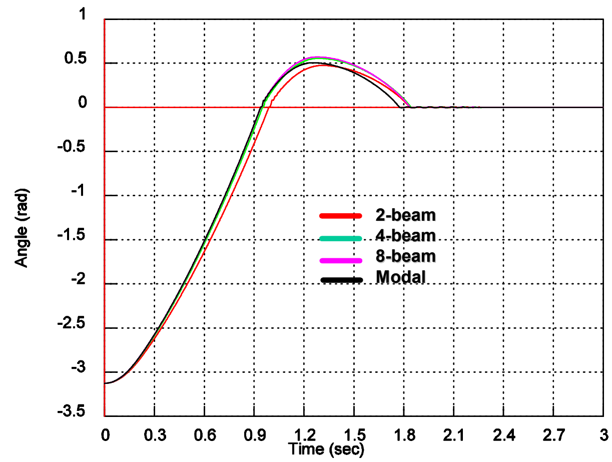

Fig. 9. Comparison of single pendulum results obtained in DADS using different levels of beam modeling fidelity.

3. DOUBLE PENDULUM ADAMS/DADS MODEL CORRELATION

In order to expand the scope of the model verification, a second series of models was constructed using a double pendulum configuration as shown in Fig. 10. Proportional damping of 3.5% at the two-link level (first modal frequency=3.9425Hz) was used throughout the case. Results of the double pendulum problem, with both links modeled using the beam/lumped mass approach, are shown in Figs. 11 and 12 where Fig. 11 is time history of the joint angle between links (hinge 2 in Fig. 10) and Fig. 12 is the time history of the joint angle at the root of the double pendulum (Hinge 1 in Fig. 10). Almost identical results were obtained from both the ADAMS and DADS double pendulum models.

In order to make a comparison between the FEM-based approach and the assumed modes approach, a different technique in modeling the hinge joint was implemented in the new DADS modal model. Instead of connecting the two flexible links at the joint, the new approach added two massless bodies rigidly attached at each end of the flexible link and formed the hinge at these massless bodies. The approach effectively filters out high frequency content in the hinge angle and velocity measurement in the model. Results of an 8-beam FEM-based model and assumed modes approach for the double pendulum problem are shown in Figs. 13 and 14. Note that throughout this paper the DADS model results are presented using an old numbering scheme where the innermost root hinge is hinge-13 and the outermost tip hinge is hinge-1. The ADAMS model uses the opposite convention with the root labeled hinge-1 and the tip labeled hinge-13.

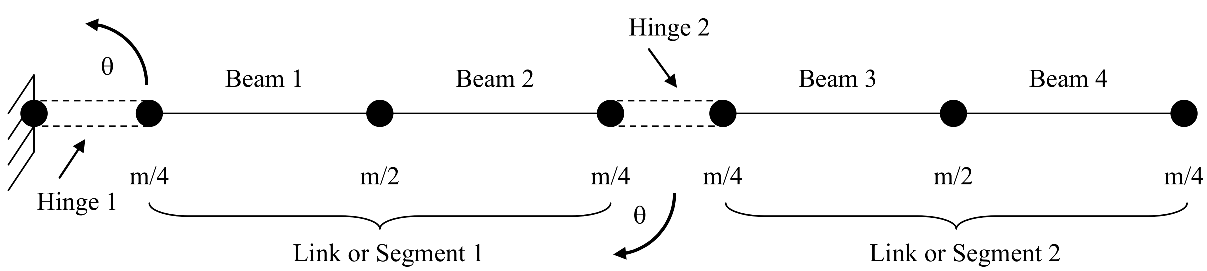

Fig. 10. Lumped mass beam construction used for the double pendulum model.

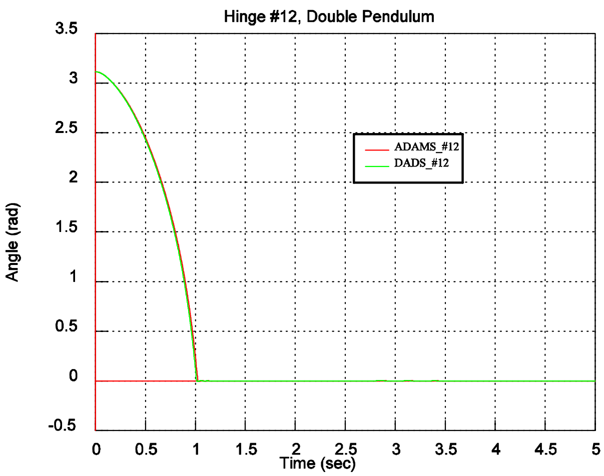

Fig. 11. Comparison of the double pendulum ADAMS and DADS model results for the outer (#2) hinge.

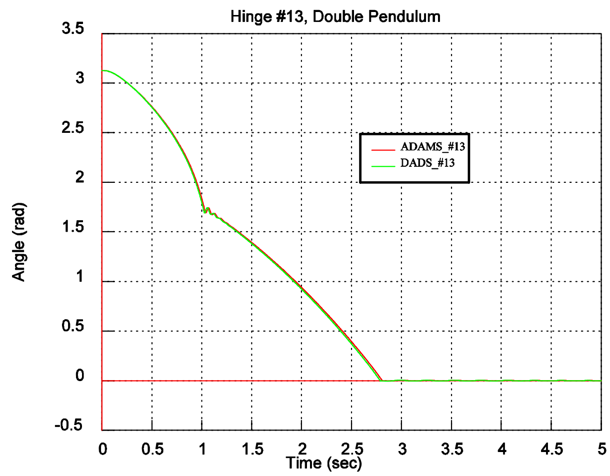

Fig. 12. Comparison of the double pendulum ADAMS and DADS model results for the inner (#1 or root) hinge.

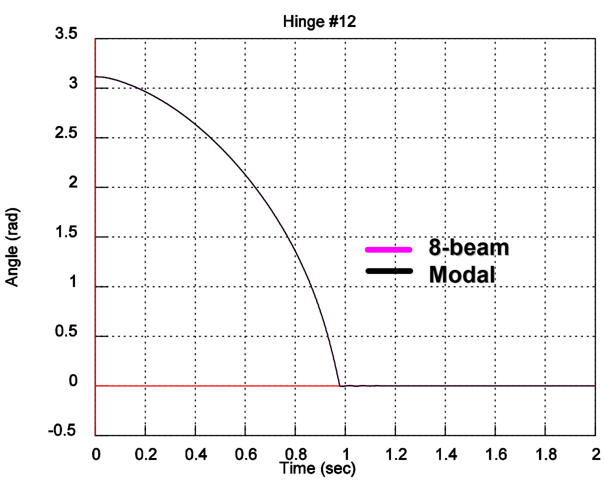

Fig. 13. Comparison of the double pendulum DADS model results using modal and 8-element beam representations for the modified rigid-body outer (#2) hinge.

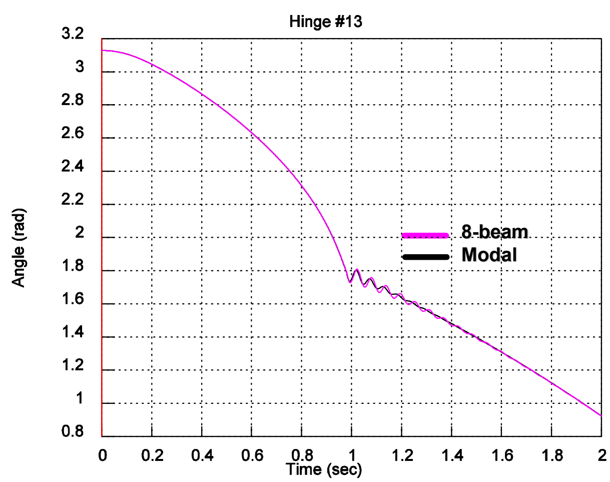

Fig. 14. Comparison of the double pendulum DADS model results using modal and 8-element beam representations for the modified rigid-body inner (#2) hinge.

Another purpose of this example was to demonstrate the fundamental difference between modal and finite-element approaches in assigning proper damping to the system. Because the damping ratio is a system level concept and depends on the system configuration, it becomes very tedious and tricky to assign proper damping to a component in order to have a desired system damping in the modal approach. The correct procedure requires solving the eigenvalue problem at each intermediate configuration (as each hinge locks or unlocks) and back-calculating the component damping using the desired system level damping ratio.

To understand the differences between the beam/mass and modal approaches, several different double pendulum models were studied in DADS by increasing the number of modes per link for the modal approach or by increasing the number of elements per link for the finite-element based approach, to gradually improve the model's accuracy. The models were numerically linearized and solved for their frequency content and final damping ratio at the fully deployed configuration. All joints are assumed to be locked into position in the calculation. To assign proper damping for the modal model, the modal damping ratio was calculated based on a system damping of 3.5%, which tends to over-damp the component modes. For the beam model the proportional damping matrix was calculated on a link level, which tended to underestimate the damping ratio at the system level.

Results from modal approach are shown in rows 2 to 5 of Table 1. These results show that, in order to have a system damping of 3.5% for the first mode (column 4), a damping ratio of 13.4% (column 2) has to be used for the component if only one mode per link was used. Component damping has to be increased even further to maintain the desired first mode damping ratio if more modes were used to represent the flexibility. Table 1 also shows that damping for the second mode (column 6), which was *not* prescribed, decreases as more modes are used in the links. As expected, frequencies of both modes (columns 3 and 5) approach the true frequency obtained from finite element method (row 6) as the number of modes per link is increased.

Results from the finite-element based approach are shown in rows 7 to 9 of Table 1. Column 2 shows the prescribed damping used at the link level (3.5%) and column 4 shows the damping ratio the system actually has. The result shows a similar conclusion as the modal approach in using component level damping for calculation of system level damping. The accuracy of frequencies of the first modes (columns 3 and 5) are also improved when more elements are used for each link. Columns 3 and 5, in Table 1, also show an interesting point that the frequency approaches the "true" (correct) model (FEM) frequency from the high and low sides and bound the true value. This indicates that the model built using the modal approach tends to behave stiffer than the real system and the model built using finite-element based methods tends to behave softer than the real system as expected.

Table 1. Component and system level modal damping for different modeling approaches.

Model	Zeta, link level	Frequency, 1st system mode (Hz)	Zeta, 1st system mode	Frequency, 2nd system mode (Hz)	Zeta, 2nd system mode
One mode/link	0.134	4.34	0.035	34.1	0.28
Two modes/link	0.152	4.14	0.035	29.6	0.20
Three modes/link	0.157	4.09	0.035	28.5	0.19
Four modes/link	0.16	4.06	0.035	28.0	0.18
FEM		4.00		25.89	
8 beams/link	0.035	4.04	0.009	26.3	0.061
4 beams/link	0.035	4.02	0.009	25.8	0.060
2 beams/link	0.035	3.94	0.009	24.1	0.056

4. FULL SYSTEM DADS SIMULATION RESULTS

Two DADS models were built for the deployment dynamic simulation of the full 13-link antenna boom. One is based on the modal approach and one is based on beam/mass approach. The FEM-based DADS deployment dynamic model of the 13-link boom was generated with each link modeled as three lumped masses connected with two beam elements. A schematic of the stowed dynamic model is shown in Fig. 15.

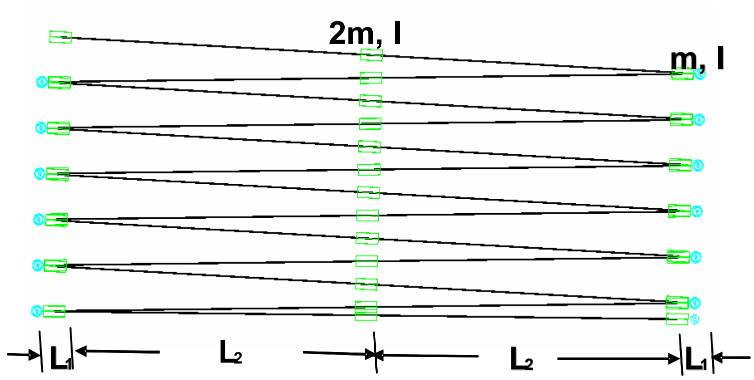

Fig. 15. DADS model of the full 13-segment MARSIS dipole boom in its stowed state immediately prior to release.

Two scenarios of the full system deployment were simulated: scenario-1 was with no stowed energy/push force (due to the lateral tube compression in the cradle) and scenario-2 used the nominal stowed energy/push force. Three cases were tested for scenario-1. In case-1, a modal model was used with light system damping ratio assumed (100% at the link level, or 0.7% at the system level). In case-2, the same modal model was used with a nominal system damping ratio (475% at the link level, or 3.5% at the system level) assumed. In case-3, a beam/mass model was used with a light system damping ratio (3.5% at the link level, or ~0.00% at the system level). Fig. 16 shows the joint angle of the link connected to the vehicle (the root hinge) for both case-1 and case-2. The hinges are modeled as starting at a 180° angle and are fully deployed at a 0° angle. This result indicates that damping was not a dominant factor in the dynamic response of the system deployment. Fig. 17 shows a similar result for case-3. When the simulation results are animated they indicate that all three cases deployed successfully in scenario-1.

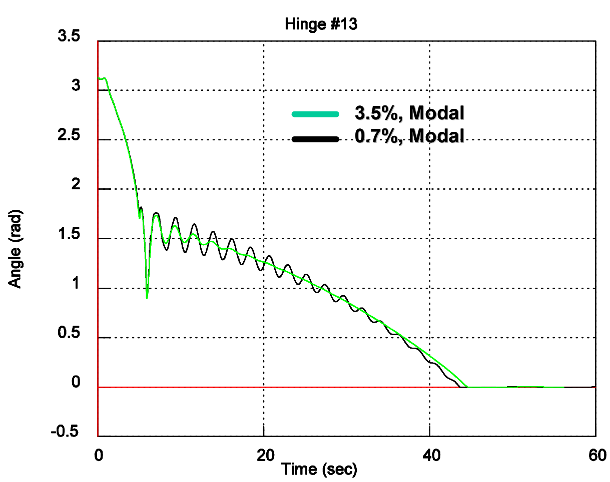

Fig. 16. Comparison of the root hinge angle for full boom deployment results of both light and heavily damped modal models in DADS

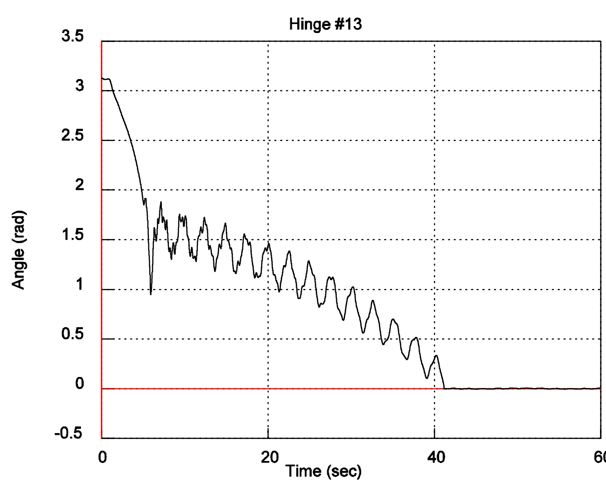

Fig. 17. Full boom deployment results using light damping and a beam/mass approach in DADS

For scenario-2, a total of two cases were tested. In case-1, the modal model was used with a light system damping ratio (100% at the link level, 0.7% at the system level) assumed and a nominal initial push-off energy. In case-2, the same modal model was used with a nominal system damping ratio (475% at the link level, 3.5% at the system level) assumed. Some similar sub-cases were also run. Simulation results in Fig. 18 show the deployment velocity of the outermost tip link for both light and heavy damping. The results show a high velocity rebound from the first link that could potentially impact the spacecraft for both cases. These results seem to indicate that structural damping is not the dominant factor that governs the boom deployment. In fact, the hinge buckling strength was found to play a much more important role in the overall dynamics[2,3].

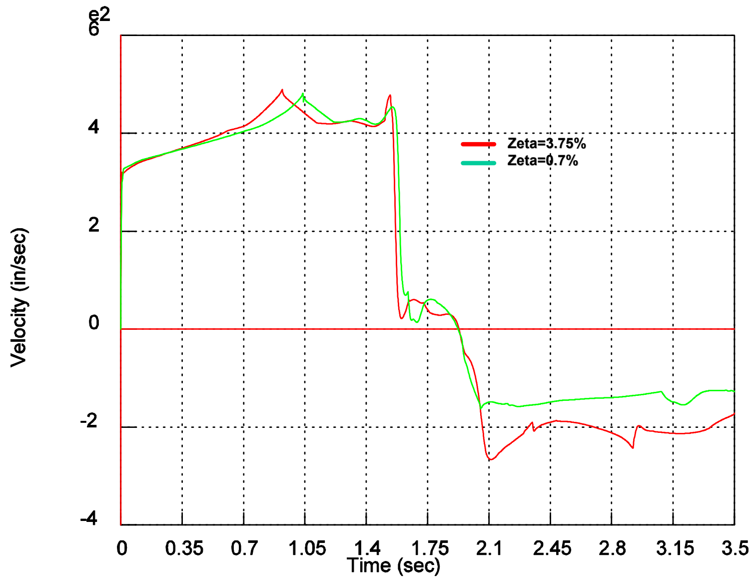

Fig. 18. Comparison of deployment velocity of the first link with light and heavy damping

5. CONCLUDING REMARKS

The modeling of lenticular (carpenter tape) deployments is a challenging task that requires patience and careful attention to detail. The dynamics of this type of deployment are chaotic at best and can be sensitive to tiny changes in parameters. One of the most significant modeling hurdles is the constantly changing geometry associated with a lenticular boom itself. The most straightforward and successful approach used in this study was classic proportional damping. It is possible to use component modal damping but this is significantly more complicated and does not eliminate the need to continually adjust to the current local boom section lengths in order to achieve correct damping levels. However, while damping does play a role, it is even more critical that the hinge stiffness and strength properties be rigorously measured and accurately modeled. Lenticular joints are replete with complicated mechanisms and when they are joined together in a serial fashion the interaction between them greatly amplifies the modeling difficulties.

Ground testing of the full system is not feasible so component level testing must be used to validate the modeling approach and to extrapolate to the full system behavior. The analysis presented here demonstrated the highly sensitive numerical challenges in simulating the sharp nonlinear transients associated with lenticular hinges. In particular, careful attention must be paid to the integrators ability to reproduce stable results for simplified problems as the boom models become cumbersome and chaotic when joined in series.

While it may seem to be a trivial matter, it is also important to view the deployment dynamics in real “wall-clock” time. Slow motion animation is quite useful for examining fine details and nuances of the model but the only way to gain an appreciation for the scope and level of the dynamics is to view them at full speed. Failure to do so can potentially lead one to conclude that a deployment is more benign than it may be in actuality.

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