DYNAMICS OF AN ELASTICALLY DEPLOYABLE SOLAR ARRAY: GROUND TEST MODEL VALIDATION

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This paper presents an analytical, computational and experimental study of the deployment dynamics of an elastically deployable solar array. The thin film array is folded in multiple stages with elastic hinge and deployed depth stiffening elements and then allowed to deploy under its own elastic strain energy. A computational model of the geometrically nonlinear deployment is assembled using reduced order models of the elastic hinge elements. Restoring torque models developed for each hinge line are validated through isolated testing of each of three deployment stages. Both linear and nonlinear models of the corresponding elastic mechanisms are updated from the results of these experiments. The ground tests use a simple low-stiffness suspension system and videometry to measure the angular displacement of the deploying panel to be measured. The angular velocity and acceleration are computed from this data for use in the model updating process. The testing is performed on an early version of the array, called the engineering model (EM), and on the flight unit. For the first stage, the root hinge, the classical and computational models adequately predicted experimental results. The second stage, or z-fold deployment, experimental results indicated a stiffness that was 2.3 times smaller than the predicted stiffness for the EM. Flight unit results more closely matched predictions. For the final stage of deployment, the tri-fold, the nonlinear elastic response of the shallow shell, or “tape” hinges, is predicted by analytical models of their post-buckled mechanics. For this stage, the observed long range stiffness is greater than the expected results based on thin plate theory and computational simulation. Further testing of flight unit components and updating model parameters will improve the understanding of the deployment dynamics of the array.

Nomenclature

b = Width
D = Flexural Rigidity
E = Young’s Modulus
I = Moment of Inertia
k = Hinge Stiffness
L = Length R = Radius of Curvature
T0 = Torque in the Deployed State
t = thickness
ε = Strain

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ν = Poisson’s Ratio
θ = Deployment Angle
DDS = Deployed Depth Stiffeners
EM = Engineering Model

I. Introduction

The development of microsatellite architectures over the past decade has lead to the need to package and deploy one meter-class components. While some packaging schemes for larger components are scalable to this class of deployable structure, corresponding constraints on mass, power, and cost have motivated a renewed interest in strain-energy mechanisms in place of classical motorized mechanical designs.

Modeling and verification of deployable structure dynamics has been addressed for many different types of deployment. Several methods of deployment have been analyzed for antennas, booms, solar arrays, and other scientific instruments. Recently, several studies regarding the deployment of inflatable structures have been conducted [1,2]. Other studies use more traditional methods of deployment [3-5]. These studies differ with respect to this work in that this study deals with an elastic deployment of a thin film array; however, techniques presented in the literature for deployment testing and dynamic analysis are still applicable. Meguro [4] notes the necessity for testing at all levels of integration—from individual component to the entire system. Knowledge of ground testing effects such as gravity and drag are integral to successful interpretation of experimental results. Fischer [5] addresses the use of suspension systems with deployable structures stating that if high accuracy is required and deployment torques are low, more complex suspension systems are needed to accurately predict the test specimen’s behavior in microgravity.

Several studies have addressed deployment characteristics post-flight [6,7]. For example, by recording various reactions occurring on the spacecraft (the pitch rate for example) during the deployment, certain characteristics can be inferred about the deployment and compared to analytical, computational, and ground test results [7]. Lomas points out that on-orbit deployment characteristics will not precisely match idealized models. Manufacturing tolerances, slight differences in packaging or preloading of devices can all play large factors in the on-orbit deployment dynamics. In addition to post-flight correlations of flight data with simulation data, a statistical analysis could be applied to the models to determine a range of performance and probability of successful deployment [8].

Tape spring mechanisms have been studied in a variety of prior concepts and applications. Models of the elastic post-buckling of both isotropic and composite shells integral to these mechanisms have been developed [9]. Prior research includes modeling of dynamic deployments under their nonlinear restoring force [10]. A somewhat smaller body of work has studied the mechanics of built-up kinematic systems of these mechanisms [11,12]. This prior work is used here to compute reduced-order analytical models of their discretized hinge mechanics in the deployable array of interest.

The deployable thin film array being studied in this program presents a novel approach to the simultaneous deployment of both the array panels and deployed-depth stiffening elements. The design employs a multiple-stage packaging and deployment scheme to substantially reduce all three dimensions of the deployed array. Both classical shallow shell, or “tape” hinges, and a deployed-depth stiffener are used to elastically fold and support the array. As this design is preparing for an on-orbit demonstration, verified models of its deployment dynamics are required. This paper will review progress in both the modeling and ground test model validation efforts for this development program.

While a fair amount of prior attention has been given to tape spring modeling, predicting the mechanics of the deployable stiffeners requires careful attention. Each stiffener is, itself, a kinematic assembly composed of relatively rigid carbon fiber reinforced polymer (CFRP) panels and compliant z-fold hinges. The unique kinematics and resulting elastic response of these mechanisms are of primary interest in this modeling effort. Reduced order equivalent hinge approximations are used to include the stiffener contributions to the assembled array’s deployment behavior. Future focused testing of these elements is planned to refine this initial model.

Deployment testing is performed in three stages, and requires the use of a gravity compensation system and an understanding of the level of drag on the array during deployment. A videometry system is used to measure the deployment angle as a function of time. Angular velocity and acceleration are computed from the experimental angular displacement data. A comparison is made between the observed and predicted stiffnesses, and the deployment characteristics are compared to the computational and classical models. The array will be deployed on-orbit, and although there is limited availability of verification of deployment, increased power levels will be an indicator of successful deployment. In addition, an investigation into the use of attitude sensors to monitor deployment is being conducted.
II. Array Description

The Fold Integrated Thin-film Stiffener (FITS) Solar Array, pictured in Figure 1, consists of six panels. The array is stowed by first folding along two hinge lines in the longitudinal direction (a “tri-fold”) and then along a single lateral hinge line (a “z-fold”). Finally, the stowed array is folded along a root hinge line to complete the stowing process. The deployment occurs in three stages: root, z-fold, and tri-fold. The root hinge deployment rotates the array away from the spacecraft. The second stage is a z-fold deployment. The final stage is a tri-fold deployment, which exposes the solar panels. The deployment stages are seen in Figure 2. The flight unit will consist of only one z-fold panel, although scaled up designs could contain many.

Figure 1. FITS Solar Array  a) from the front, and b) from the back

Figure 2: Array Deployment Stages
Figure 1 indicates key features of the array discussed here. The two pin-clevis root hinges are each driven by two torsional springs. Four z-fold hinges drive the z-fold deployment. A z-fold hinge consists of RTV silicone constrained with woven glass fiber cloth, together having a thickness of 1.8 mm on the EM. The thickness of the z-fold hinges on the flight unit is considerably smaller, 0.89 mm. The z-fold hinges are strained during stowage of the array, and the stored strain energy forces the deployment when the panel is released. The tri-fold deployment is driven by the use of four tape springs, two along each hinge line. These tape springs are set up for “same-sense” bending, or bending in the same direction as the radius of curvature of the tape spring. In addition to the tape springs, two triangular deployed depth stiffeners (DDS) are used to provide bending stiffness in the longitudinal direction along the length of the array.

III. Modeling

To characterize the deployment dynamics of the array, various modeling approaches are applied. Primarily, these approaches fall into two categories: classical and computational. The three stages of the deployment are modeled separately to simplify the non-linear nature of the problem, especially with regard to classical modeling. Computational modeling is exercised to capture the nonlinear geometric dynamics and contact mechanics of the deployment that are ignored in the classical models. In addition, coupling between each deployment stage is investigated using computational modeling approaches. The complexity of the computational model is managed by using a reduced order model that utilizes results from previous analyses and experiments to capture hinge behavior. The general model verification approach is to perform focused ground tests on each deployment stage to refine corresponding hinge torque models. The resulting verified model is then used to simulate the multi-stage deployment that is excessively difficult to perform on the ground.

A. Classical Modeling

1. Root Hinge

The linear elastic response of the root hinge, z-fold hinges, and the long-range linear response of the tape springs are analyzed using classical methods. The root hinge is analyzed as a torsional spring, with a stiffness of four times one spring’s specification, since the root hinge is deployed using two hinges with two springs each. The torque for the root hinge springs as a function of rotation angle is a linear relationship with a non-zero holding torque in the springs in the deployed state. The relationship between torque and rotation angle leads to the following governing differential equation:

$$\ddot{\theta} + \frac{k}{I} \dot{\theta} - \frac{\pi k + T_0}{I} = 0.$$ (1)

$\theta$ is measured as the angle between the two deploying panels. Thus, $\theta$ is zero degrees in the stowed state, and 180 degrees in the deployed state. Note that this model does not incorporate any frictional torque, assumes the z-fold remains completely stowed throughout deployment of the root hinge, and does not account for the hard stop at full deployment.

2. Z-Fold

Initially, a plate-bending model of the z-fold hinge is constructed using the geometric dimensions and elastic modulus of the silicone:

$$k = \frac{Db}{L} = \frac{Ebt^3}{12L(1-\nu^2)},$$ (2)

where $k$ is the effective torsional stiffness. This analysis, however, results in a stiffness that was eight times higher than the experimental stiffness found by testing of actual z-fold hinge samples. This was especially surprising, since the glass cloth is completely ignored in this analysis. Thus, the mean measured stiffness of 0.0738 Nm/rad is used in a linear elastic model of the hinge behavior. In the same approach as the root hinge analysis, a torsional spring model is used, resulting in Equation 1, where $T_0$ is zero and $I$ is the moment of inertia of the deploying z-panel.
3. Tri-Fold

Although tape spring behavior for the tri-fold is highly non-linear, the long range bending stiffness is predictable using flat plate bending. Using the geometric and material characteristics of the tape spring and equating the arc length to the width of the plate, Equation 2 is used to approximate the long range bending stiffness of an individual tape spring, using material properties of steel.

B. Computational Modeling

Computational modeling is applied here to capture the geometrically nonlinear dynamics and contact mechanics of the multi-stage deployment. The ABAQUS Finite Element (FE) software package is used for this analysis. Simulations are run to model the deployment of all experimental and classical configurations. The models are built focusing on the deployment behavior of the driving mechanisms, the root hinge, tape springs and z-fold hinges, as opposed to the flexible mechanics of the array panels. Therefore a simplified approach uses a reduced order model with discreet point hinges, which is simple to update with experimental data. The results of the simulations are then compared to the classical analyses and experimental results.

1. 2 DOF Deployment

The coupled deployment behavior of the root hinge and z-fold is initially modeled as a simple 2 DOF rigid body simulation. The simulation is performed to assess dynamic response accuracy for the minimal model with condensed hinge line mechanics. Rigid beams are used to model the length dimension of both panels, honeycomb and z-fold (Figure 3), to obtain the correct pivot points for the hinge lines. Point mass elements and rotary inertia elements are positioned at the midpoints of each beam in order to model the inertia and mass properties. Hinge connector elements are used to model the torsion behavior of both the root hinge and z-fold hinges. The elastic stiffness (moment vs. rotation) for each point hinge is taken from the experimental values obtained from the flight article. These values are incorporated into the model to obtain a solution that closely depicts the actual deployment behavior. In order to model the complete deployment, the restoring force for the root hinge is modeled as a nonlinear function that accounts for the physical hard stop implemented on flight hardware. This is accomplished by modifying the root hinge elastic stiffness curve with an abrupt positive slope change at zero degrees of deployment (the final deployed angle). It is important to simulate the dissipated energy after each rebound to accurately capture the complete deployment as this directly affects the dynamics of the z-fold panel. A trial and error method is used to find appropriate damping values to model the energy dissipation at hard stop rebounds.

2. Finite Element Model

The array is meshed using shell elements for the six panels and Deployed Depth Stiffeners (DDS) as illustrated in Figure 3. This model does not concentrate on the dynamics of the individual panels; therefore, a relatively coarse mesh of 100 and 36 elements was used for the array panels and DDS respectively. Additionally, the model assumes that each panel is made of homogenous and isotropic material. The CFRP orthotropic properties are converted to average laminate properties, which are then used in the FE isotropic material definition. Two different shell section definitions are defined for the model: one definition for the central panel (honeycomb), which interfaces with the root hinge, and a second for the five remaining panels and DDS. In order to allow the compound hinge motion at the hinge line intersections, coincident corners are not included in the mesh. This produces the diamond shape holes in the array panels and DDS sections. The mass budget is used to determine the density properties for both section definitions. Strain energy is built up in the driving hinge elements during stowage steps in the static environment. Deployment steps are performed in the dynamic implicit

Figure 3. Mesh Details
ABAQUS environment. Global rotational displacement, velocity and acceleration histories of deployment stages are extracted from the FE model and compared to the measured responses.

The stowage and deployment characteristics of the array require simulations of torsion spring behavior. To represent the various elastic mechanisms, hinge connector elements that allow nonlinear moment-rotation relations about a single axis are used. Coincident nodes at panel boundaries are connected by hinge elements along the hinge lines for the Tri-fold, Z-fold, DDS/panel interface, and the DDS spine (see Figure 3). Results from multiple configurations modeling the folding mechanism showed that hinge elements were the most reliable option for simulating the required torsion spring behavior and idealizing the true “plate” bending behavior.

Root hinge simulations are modeled by using one discrete point hinge. The physical root hinge consists of a pin-clevis joint driven by multiple torsion springs. This hinge is connected to the central array panel (honeycomb) with two rigid beams. The elastic stiffness is implemented using a linear curve with a pre-load moment value in the deployed configuration.

Z-fold modeling is accomplished by placing hinge elements at the coincident panel nodes along the z-fold hinge line, as well as four hinge elements at the coincident nodes along the hinge line on the DDS (Figure 3). To simulate the restoring forces of z-fold hinges that drive the z-fold deployment, all hinge elements connecting the array panels are given negligible stiffness. This allows a “free” rotation behavior that does not interfere with the restoring force of the z-fold hinges. Each element modeling a z-fold hinge is given a linear moment-rotation curve, which approximates the plate bending mechanics of the actual mechanism.

The tri-fold deployment stage is modeled using a similar approach to the z-fold. Hinge connector elements are placed between the coincident array panel nodes along the tri-fold hinge lines (3). The driving force of the tri-fold deployment is provided by tape springs. To capture this restoring force, theoretical values of the nonlinear moment-rotation relationship are obtained from a separate ABAQUS FE model. The tape spring simulation uses the geometric characteristics of the actual carpenter tape springs used on the array. The nonlinear torsional behavior is shown in Figure 4. Due to the peak moments (“snap through”) experienced in either direction of rotational displacement, a “locking” behavior is induced due to the stable state at zero rotation angle. The snap through and locking behavior is modeled by applying a piecewise linear representation of the nonlinear stiffness curve to the four hinge elements. These hinge elements are located at the outer edge of the tri-fold hinge lines (Figure 3). In order to simulate the kinematics and dynamic behavior of the tri-fold deployment accurately, all hinge elements (excluding the tape spring points) are given a negligible elastic stiffness. Therefore, potential restoring torques of the Teflon tape used for the DDS construction (panel interface and DDS spine connection) is not accounted for, though this may prove to be a significant factor during model refinement.

3. Simulation Procedures

The stowage procedure is modeled in the static environment. The static procedure is chosen for efficiency and the inherent neglect of the kinetic energy gained during stowage steps. All stowage steps are achieved by using redundant boundary constraints (fixed in all DOF) to all non-moving nodes while forcing rotations at the corresponding hinge line hinge element nodes. Rotations begin at zero degrees and stowed rotations of 180 degrees are performed to model all the stowage steps.

Deployment simulations are modeled in the dynamic environment. A fixed boundary constraint is applied to one of the nodes on the root hinge/spacecraft interface element for all simulations except the z-fold only deployment. This particular deployment focuses on the z-fold hinge performance, therefore the tri-fold hinges and the central array panel are constrained. For all deployment simulations the boundary constraints of the corresponding hinge elements are instantaneously lifted. It must be noted that friction of all hinge elements is neglected for all deployment configurations.
Contact issues heavily influence full deployment and tri-fold simulations. The issues arise from the compact packaging of the array. The tri-fold panels overlap in their stowed state, and then the z-fold is performed. Due to the idealized point hinges used to model the z-fold hinge mechanism, the potential restoring force gained from the compression of the compound z-fold hinge folds (Figure 12) must be accommodated by the nonlinear elasticity of the connector hinge elements used to represent their mechanics. In the tri-fold deployment, the interaction due to contact between the tri-fold panels is significant to the dynamic behavior. During deployment both tape springs act in a similar manner, and the deployment of the inner fold aids the motion of the outer fold. This results in an asymmetrical deployment with snap through of the tape springs in each fold occurring at different times. Utilizing contact definitions on the shell element surfaces simulates this behavior. Only the penetration interference is modeled, while the friction of this contact is neglected.

IV. Experiment Configuration

A videometry system is used to record rotational displacements for the panel(s) during the deployment. Testing is performed for each stage of the deployment separately. A suspension system allows for off-loaded testing. The engineering model (EM) is connected to an aluminum plate using a clamping mechanism. As shown in Figure 5 this plate can be mounted onto the test stand in two configurations, vertical and horizontal, for the tri-fold and z-fold deployments respectively. A slightly different mounting mechanism is required for the flight unit due to compatibility with the test stand. Two spotlights are connected to the test stand to provide adequate lighting for image processing. A CCD camera is mounted approximately four meters above the test specimen to ensure the field-of-view includes the entire EM and to minimize perspective distortion. Black cloth is used to cover certain parts of the test stand to avoid unnecessary reflections that could interfere with the image processing. Black electrical tape is applied to parts of the mounting hardware that could not be covered with the black cloth. As shown for the EM tri-fold in Figure 6, two targets are placed on each panel (two panels for z-fold, three panels for tri-fold), ensuring that the targets do not interfere with each other in the stowed state or during deployment. Tests are also conducted for the root and z-fold combined deployment for the flight unit. The camera field-of-view did not capture the entire deployment for the combined root/z-fold deployment, so four targets are placed on each panel such that at least two targets are in the field-of-view at any given time.

![Figure 5. Array Mounting to Test Stand for the a) Tri-Fold deployment, and b) the Z-Fold deployment](image)

![Figure 6. Tri-Fold target placement for the EM, shown in the stowed state](image)
A suspension system is constructed by attaching Kevlar thread to each free panel. These suspension lines are attached approximately 4 meters above the panel hinge lines. A compliant spring (spring constant of 4.73 kg/m) is used near the top of the line to apply the appropriate amount of tension in the line for gravity off-load. The lines are attached directly above the hinge line of the panel, producing no torque that aided or hindered the deployment. Since the lines are mounted high above the test specimen, any slight misalignment will induce very little torque on the hinge. An analysis was performed to determine the effects of misalignment of the suspension lines on the torques induced about the hinge lines. Figure 7 displays the results of this analysis for the z-fold deployment. Misalignments of 2.8 centimeters produce torques that are significantly lower than the expected torque produced by the hinges.

V. Experimental Results and Analysis

Results are presented here for tests performed on the EM and the flight unit. Staged testing allows for gravitational unloading cases of each deployment stage: the root, z-fold, and tri-fold.

A. Root Hinge Results and Analysis

Two solar arrays will be deployed on-orbit. One array’s initial deployment is with a hinge assembly consisting of two hinges each with two standard torsional springs. The second array is deployed with an Elastic Memory Composite hinge developed by Composite Technology Development, Inc. (CTD). Modeling of the EMC hinge deployment dynamics was beyond the scope of this study, thus the elastically deployed array is analyzed for the purposes of these tests. The results of a simplified computational model of the root hinge deployment are shown in Figure 8, and Figure 9 displays the results of the classical analysis and experimental testing for the root hinge. The computational model, classical model, and experimental data correspond well for the deployment up until the array encounters the hard stop. This was expected, since both the computational model and the classical analysis ignored the hard stop.

The root hinge deployment test is only performed on the flight unit, since the EM does not have the root hinge mechanism. During the root deployment test both the tri-fold and z-fold panels are restrained from deploying. The angular displacement is measured from the target positions. The solution the Equation 1 takes on the form:

\[
\theta(t)=a_1 + a_2 \sin \left( a_3 t + \frac{\pi}{2} \right).
\]

A fit was performed using a generalized least-squares method to obtain a functional form for the angular displacement as a function of time. From this function, the angular velocity and acceleration were computed. Figure 8 displays the angular displacement data as a function of time for the duration of the deployment. A theta value of 180 degrees corresponds to the hard stop for this deployment stage.

The fit was performed for the data preceding the first hard stop, from the beginning of deployment till about 0.9 seconds, since the hard stop was not modeled. Figure 9 provides a comparison between the model predictions, experimental results and fits. The computational and classical models closely agreed, so only the classical model provided in Figure 9. The slight difference in deployment time can be attributed to small frictional torques in the hinge, the drag on the array, inaccuracies in the torsional spring constant, or a combination of these effects. Future models will include these effects drag. Also, computational models will be developed to model hard stop during the root deployment. The experimental velocity reaches a slightly lower peak value than the model predicted. Table 1 shows a numerical comparison of the expected results with the experimental results. The experimental results match up relatively well with the classical model predictions, although the springs are slightly stiffer experimentally.
Table 1. Root Hinge Results Comparison

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<th>Experimental</th>
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B. Z-Fold Results and Analysis

Video was taken of the z-fold deployment for both the EM and the flight unit. Video was taken for the duration of the deployment, including the oscillations that followed the initial deployment. These oscillations occur for about eight seconds after the panel reaches 180 degrees (the deployed state). The targets allow for measurement of the angular rotation as a function of time. As with the root deployment, a fit to the displacement data was performed to obtain a functional form of the angular displacement as a function of time. This function is differentiated to obtain angular velocity and acceleration as functions of time. Figure 10 provides the measured angular displacement as a function of time for the duration of the deployment. Measured angular data, function forms of the angular displacement and acceleration, and model predictions are compared in Figure 11.

To obtain an experimental stiffness for the z-deployment, the angular acceleration is plotted as a function of angular displacement (Figure 12). The slope of this line is the stiffness divided by the moment of inertia and was compared to stiffness of the z-fold hinges, which was determined in an earlier

![Figure 8. Root Hinge Experimental Angular Displacement as a Function of Time.](image8.png)

![Figure 9. Root Hinge Deployment Comparison to Model Predictions](image9.png)

![Figure 10. Z-Fold Angular Displacement as a Function of Time for the Flight Unit.](image10.png)
experiment on sample z-fold hinges at MicroSat Systems, Inc. [13]. Due to differing z-fold hinge designs between the EM and the flight unit, separate classical models were required. The flight unit hinges had a decreased thickness. The experimental results for the thicker hinges were scaled to obtain a better estimate for the flight unit hinge designs. Future testing of z-fold hinge specimens will provide better model parameters.

Table 2 provides a comparison of the computational model, classical model, and experimental results. The stiffness of the hinge is taken from the aforementioned independent testing of the z-fold hinge. The deployment of the EM takes about three times longer than the classical model predicted. In addition, the initial angular acceleration and maximum angular velocity are significantly lower than predicted for the EM. The flight unit results were closer to predictions than the EM. The flight unit did, however, show significantly larger initial angular acceleration and stiffness. The experimental results for the EM and the flight unit showed that the panel actually decelerates after about 100 degrees of deployment.

The decelerating effect seen is not predicted in the models. It was initially postulated that drag could be the cause of the deceleration. Drag effects on the panel are computed estimating the panel as a flat plate. The maximum angular velocity, converted to a linear velocity, is taken to obtain a worst-case scenario for the drag force acting on the panel. The resulting drag on the panel is less than 5 percent of the z-fold deployment torque. Therefore, drag
could not contribute the amount of resisting torque necessary to slow the panel down. However, future models will incorporate drag effects to better approximate the panel deployment. Another theory involved the suspension system providing a resisting torque that had not been properly modeled. To test this, the z-fold deployment is conducted without the suspension system. The deceleration of the panel is still seen. The deployment did, however, take slightly longer and reach a lower maximum angular displacement during the oscillations. This deceleration has yet to be explained and is currently being studied.

The flight unit has similar results to the EM, however, the z-fold hinge design for the flight unit was changed significantly. The thickness of the EM z-fold hinges is 1.8 mm, and the thickness of the flight unit hinges is 0.89 mm, two times lower. After extensive testing of the EM through hundreds of deployments, examination of the z-fold hinges on the EM revealed cracks in the hinge as shown in Figure 13. These cracks occur most noticeably in the outside hinge. The maximum bending strain of the z-fold hinge can be expressed as

\[ \varepsilon = \frac{t}{2R}, \]  

where R is the minimum radius of curvature of the hinge and t is the hinge thickness. Using this relationship, a maximum strain of 72.5% is computed, although there is significant uncertainty in the measurement of the radius of curvature. The typical range for RTV silicone elongation at break is 100–825%. A value for the specific silicone used for the z-fold hinges was not available. The maximum bending strain is below the elongation at break; however, since the array has been deployed many times, fatigue is the main issue of concern regarding the cracks in the hinges. To address the fatigue concerns, the thickness of the hinge for the flight unit was reduced. Decreasing the hinge thickness will decrease the strain on the hinge during the stowed state such that fatigue is no longer a concern. With cracks present in the hinge the stiffness will be greatly reduced, leading to the lower stiffness values for the EM experimental data than the model predicted. The model was scaled to obtain a model for the thinner hinges used on the flight unit.

One other item of note is the interaction between the two z-fold hinges that are stacked on top of one another in the stowed state. When the array is stowed, one hinge bends inside of the other and the inside hinge becomes kinked (this can be seen in Figure 13a). This could induce some non-linear effects during the deployment. Concern was raised that this effect could cause the actual stiffness of the hinges together to differ from a single hinge. Currently, it was assumed that four hinges would produce four times the stiffness of one hinge. Testing could be conducted on two z-fold hinges stacked together, resulting in more accurate data for the classical and computational models of the deployment. Since it appears deployment for the z-fold occurs using the stacked hinges, this analysis is not considered time critical and has yet to be performed.

C. Tri-Fold Results and Analysis

Video is taken of the tri-fold deployment for several trials for the EM and flight unit. The EM array is stowed with the right panel under the left (when facing the test set up), and the flight unit was stowed with the left panel under the right. Initial analysis indicated that snap through occurs at about 155 degrees for both the left and right panels for the EM, but the panels do not lie at 180 degrees in the deployed state (the neutral position is about 172 degrees). The flight unit results demonstrated snap through more clearly than the EM. For the first panel to deploy, snap though occurred at 170 degrees, and the second panel at 166 degrees for the flight unit.
As with the z-fold deployment, the angular rotation of each panel is measured, a fit is performed, and the angular velocity and acceleration are computed by differentiating the fitted function. The function is fit for angles smaller than 150 degrees, since the tape spring long-range linear behavior is only valid up to this point. This fit takes on the same functional form as Equation 3.

Figure 14 displays the results for a single trial for the flight unit. Some data is missing due to obstruction of the targets. Snap through can be seen at about 175 degrees for the outside panel and 152 degrees for the inside panel. The right panel takes a significantly longer time to deploy, in part because the outer panel obstructs it from deploying. In addition, the outer panel is accelerated during the first portion of the deployment due to the inner panel pushing on it.

Figure 15 provides a comparison of the measured data and the fitted function for the angular displacement and acceleration. The model data was substantially different from the results and was omitted from this figure. Table 3 provides numerical comparison to the model. The angular data fit for the outer panel is not a very close approximation. The fit assumes totally idea motion. During the first 0.5 seconds of the deployment, the outer panel has a higher acceleration than the fit shows. For this time, the inner panel is pushing on the outer panel. The model does not take into account this interaction. It is hypothesized that the persistent deviation from the fit is due to these contact issues. After the inner panel is free from interaction with the outer panel, the inner panel begins the main portion of its deployment (this occurs at a time of 0.6 seconds). The inner panel fit more closely matches the data for the duration of the deployment, with the exception of a small reaction to the outer panel “snap-though” that occurs at about 1.2 seconds.

As illustrated in Figure 16, the predicted stiffness was much lower than experimental results for both the inner and outer panels. The differences between predicted models and experimental results are partially due to inaccuracies in the model of the individual tape springs, since all the characteristics of the tape springs and their attachment to the array were not modeled. More detailed models of the tape springs could be built in ABAQUS to provide more accurate results. There could also be a contribution from the DDS to the deployment torque. Testing of the DDS assembly will provide insight into the magnitude of their contribution. The kinematics of the DDS mechanism most likely result in relatively small, but nonlinear, restoring torques for this deployment stage.
The contact issues that are not dealt with in the model of the tri-fold deployment can also be observed in the difference between the deployment stiffness of the outer and inner panels. Considering this effect, the outer panel to deploy should have slightly higher initial initial acceleration, due to the inner panel pushing on the outer panel. This does not account for the total difference in initial acceleration seen experimentally for the EM or the flight unit, or for the larger stiffness calculated for the outer panel to deploy. There is a permanent deformation, or “kink”, in the bottom tape spring on the right panel of the EM. This tape spring could be contributing a smaller stiffness, which would cause the inner panel on the EM to have both a smaller initial angular acceleration and slope. However, a large discrepancy between the inner and outer panels was still seen in the flight unit, which did not have a kinked tape spring. Both the flight unit and the EM have discrepancies between the deployment stiffness of the outer and inner panels. For both the outer panel has a higher stiffness; however, both panels on the flight unit had higher stiffness than the EM, which suggests that fatigue had played a role in degrading the stiffness of the tape springs or their attachment points, or there are manufacturing differences that gave the flight unit significantly higher stiffness.

As seen in Table 3, both the EM and the flight unit tape springs have significantly higher stiffness than the model predicted. The difference between the ABAQUS model of the tape spring and the flat plate analysis is also surprising. The large differences seen could be due to inaccuracies in the ABAQUS model of the tape spring. These inaccuracies include modeling the attachment method to the array and contact issues between the attachment mechanism and the tape spring. Also, the ABAQUS model assumes the tape spring has a constant radius of curvature. It is obvious, even from visual inspection, that the tape spring does not have a constant radius of curvature. The model can be developed further to address these inaccuracies.

<table>
<thead>
<tr>
<th>Table 3. Tri-Fold Results Comparison</th>
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<tbody>
<tr>
<td><strong>Theoretical</strong></td>
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<td></td>
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<tr>
<td>Snap Through Angle (deg)</td>
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<tr>
<td>Maximum Angular Velocity (deg/s)</td>
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<tr>
<td>Initial Angular Acceleration (deg/s²) using linear fit</td>
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<td>Slope of acc vs theta (1/s²) (theta in deg)</td>
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</table>

VI. Conclusion

The root hinge deployment was well predicted with the classical and computational models. The stiffness measured for the z-fold of the Engineering Model (EM) was much smaller than expected, which is likely due to cracks in the hinge material. For the flight unit, the z-fold hinge thickness was decreased by a factor of two in order to decrease the strain encountered in the hinge such that fatigue is not a concern. The experimental z-fold stiffness of the flight unit was slightly larger than the scaled model predicted. Testing of thinner z-fold hinges will provide better stiffness estimates to apply to the models. There could also be effects that are not modeled from having two z-fold hinges stacked on top of one another, causing kinks to form on the inside hinge in the stowed state. Tests of a hinge assembly could aid in assessing this effect.

The tri-fold stiffness was higher than predicted for the EM and the flight unit. More precise modeling of the tape spring mechanisms and determination of the stiffness contribution from the Deployed Depth Stiffeners will aid in creating a more accurate model of the tri-fold deployment. In addition, better moment of inertia estimates will also improve the classical and computational models for all three stages of deployment.

In order to better characterize the deployment, particularly for regions near snap through for the tape springs, a higher speed camera is necessary. A wider field-of-view will simplify the analysis significantly. This could easily be

* For some trials two snap through events were seen in the data for the right panel. This could be due to the kinked tape spring snap though occurring at a different location than the other tape spring. The location of the second snap though was not consistent and in some cases was not noticeable at all.
achieved through the implementation of a lens with zoom capabilities. In addition, interference of mounting hardware and the test stand with the camera angle need to be eliminated to provide more complete results, particularly for the tri-fold and combined deployments.

Future work will focus on developing scalable design analyses for the FITS architecture. A more detailed understanding of the z-fold deployment using the z-fold hinges is necessary, as well as the mechanics of the Deployed Depth Stiffeners. The computational analyses presented are from simplified deployment simulations as well as simplified mesh models. Simulations of the coupled root hinge and z-fold deployment, tri-fold deployment including contact, and more refined mesh models are currently in process. Coupling of each stage will need to be considered to fully understand the complicated non-linear behavior of the array deployment. Further testing of flight hardware and updating of the computational model will aid in the development of scalable design parameters.

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References


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