Analysis of a Novel Solar Panel System with ADAMS,

Modelling and Simulation of the *Curwin* System

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Abstract

Within Fokker Space a study was undertaken during the period 1998 to 1999 to investigate solar array designs for the constellation markets. In close co-operation with Boeing and MMS (now Astrium), several solar array designs options were investigated for the Teledesic and Celestri programs. One of the solar array concepts studied in more depth was the *Curwin* solar array. In the project, the preliminary performance characteristics are provided. A physical model at 50 % scale was made to correlate hardware with theory.

The *Curwin* concept is quite different from conventional concepts as it uses geometric stiffness to reach a higher deployed wing frequency instead of using a separate backbone structure. The fully deployed wings are slightly curved (like a measuring tape), the actual curving occurs after the wings have been deployed. Both deployed bending stiffness and torsional stiffness of the array are covered. Tension wires form a kind of shear web, closing the open C-structure and thus providing the on-orbit torsion stiffness.

It appeared that in-house software was not able to predict the deployment behaviour. A new deployment program was required, which could deal with the highly non-linear behaviour. This analyses activity was then initiated with ADAMS. The paper discusses the evaluation of the *Curwin* concept using ADAMS. Model components and model approach are discussed as well as some results obtained in the simulations performed.

Introduction

For constellations, it is mandatory to keep the total system cost down. Therefore, the launch costs are to be decreased, for instance by implementing multiple launch options. For this reason the tight stowage volume requirement for the solar array implies a considerable design driver. For future spacecraft concepts, this may be required as well, as the power demand is expected to increase whereas the available stowage volume may become critical.

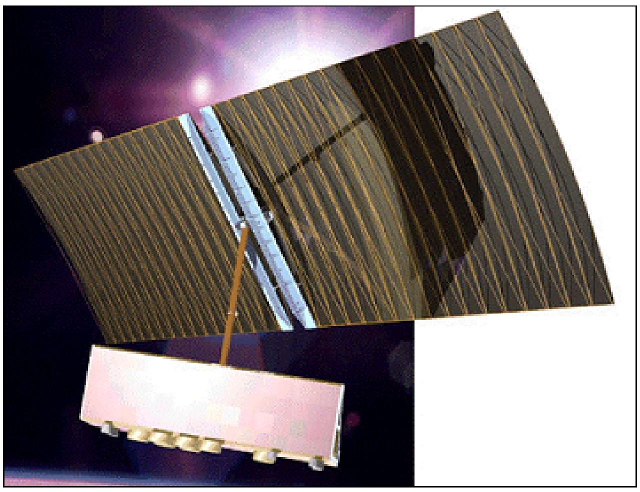

Figure 1: *Curwin* solar array in deployed configuration

The tight volume requirements in combination with the required on-orbit stiffness led to some very constraining features. The required on-orbit stiffness can be achieved in 3 ways:

1. Thick rigid panels
2. Separate mast that supports the panel structure or a flexible blanket
3. Geometric stiffness with semi rigid panels

In the design phase of the solar array, options 1 and 2 were discarded. It was decided to use a novel concept that allowed a very compact stowage method while still being inherent stiff in the fully deployed configuration.

By curving the thin panels in the deployed configuration (i.e. like a measuring tape) using special tension wires, the *Curwin* solar array concept was born. The solar array consists of two wings on top of the Primary

Deployment Mechanism boom. Each solar array wing is identical and has 14 sandwich panels of 4.3 m x 0.50 m to meet the power demand of 4.3 KW at End-of-Life (EOL).

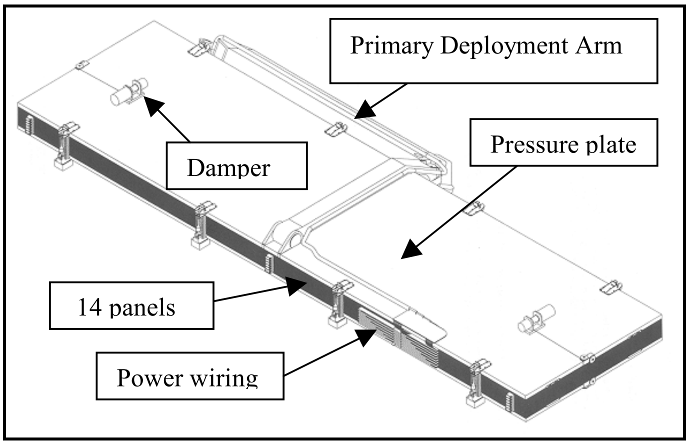

Figure 2: Solar array in stowed configuration

The sandwich panels are made of an aluminium honeycomb with CFRP carbon fibre face sheet on both sides. The panel sandwich thickness is now only 4 mm. The front sides are covered with solar cells. The rear side is used for electrical wiring and the second deployment synchronization system. In stowed position, the 14 panels are stacked with a spacing of about 3 mm each, leading to a total stack height of about 100 mm. Inter-panel hinges, cups and cones between the panels assure the spacing in stowed position at and guarantee that the stowed resonance frequency remains above 35 Hz.

Deployment mechanism

After the Primary Deployment, the Secondary Deployment will take place by relative rotation of the panels over five inter-panel hinges along the panel fold lines. This deployment is actuated by preloaded torsion springs placed along the inter-panel fold lines. Synchronization of the deployment is ensured by torsion springs (Synchrony tubes) connecting two adjacent hinge lines (i.e. connecting panel 1 to 3 and panel 2 to 4 etc.).

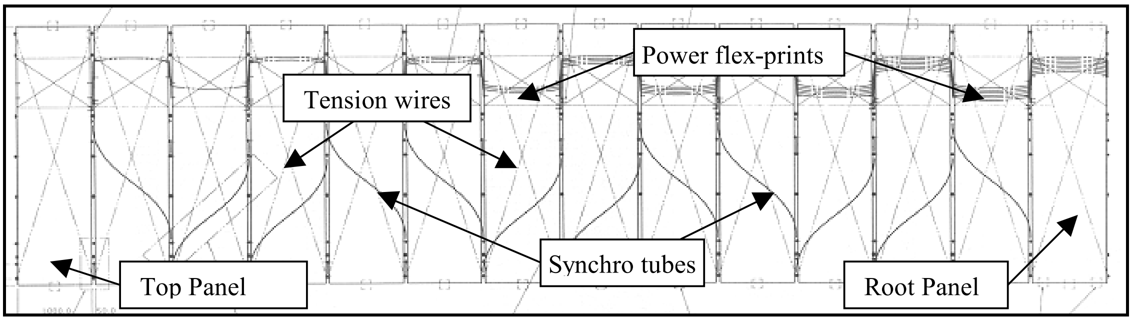

Figure 3: Components and secondary deployment mechanism of the *Curwin* system

Panel hinges have an end stop but no lock, as a result there is hardly a deployment shock. During deployment, the panel fold lines are stiff because the panels are in different planes. As soon as the panels have lined up, the panels can form a curved wing. The curvature of the panels is maintained by a number of spring loaded tension wires which run diagonally behind each panel to all four corners. Pre-tension in the wires decreases from 100 N per cable in stowed situation to about 60 N in the curved situation. As the panel hinges are pre-loaded via the curving mechanism, hinge backlash is inherently eliminated when the hinge is curved forming a stiff wing once fully deployed.

50 % scale physical tests

With a special deployment rig, manufactured to simulate zero gravity conditions, the following tests were performed on a 50 % scale physical model:

- Functional deployment and life testing
- Torque surplus measurements (indicative)
- Stiffness and backlash (qualitative & quantitative)

The following observations were made from the measurements:
- The tension wires were able to initiate the curving so no extra curving mechanism was required.
- During the deployment, some panels already became fully curved while others still remained flat. This premature curving was not expected and could not be predicted, as it was a highly non-linear kinematics behaviour. It seemed to help deployment by the energy released from the tension cables.
- Some form of damping would be required to control the additional energy coming from the curving panels and to better control the off-axis deployment trajectory of the solar array.

Dynamic Analysis using ADAMS

As was already stated in the previous section, a number of unexpected results were obtained during the measurements. At this point, Fokker Space decided to perform a non-linear dynamic analysis of the deployment of the *Curwin* system. It appeared rather quickly that the calculation method with an in-house developed tool (SMX deploy) could not be used. Thus, a feasibility project was initiated to evaluate the solar array deployment with the Multi Body program ADAMS.

The desired outcomes of the ADAMS simulations are:
- Verification of the measurements, due to the required extra masses and remaining friction effects caused by the 0-g ground support equipment (GSE), the developers were still not certain if and how the deployment of the system will work in a real outer space situation.
- The erratic behaviour noticed in the measurements indicated the need for extra damping. From experience with previous systems it was decided to use viscous or Eddy current damping by means of a cable mounted on a pulley. Using dynamic simulations, the required method and amount of damping application must be evaluated.
- The application of extra damping mentioned before must be tuned in combination with the other parameters in the deployment phase.
- Evaluation of use of a general purpose Multi Body simulation code. Employees from Fokker Space realized there is a growing need for non-linear kinematics analysis due to the fact that solar arrays become more complex and lighter, thus making reliable measurements under zero gravity conditions with GSE increasingly complex, expensive and error prone.

Modelling Approach

The ADAMS model of the *Curwin* system was defined in a macro oriented manner to allow for both detailed simulations as well as simulations with simplified models and a variable number of panels in the complete solar array. Furthermore, it was decided to define the flexibility in the panels both using a modal flexible and a discrete flexible method. For creation of the models, high-level ADAMS template command files were used. As all model equations are created from these template files, development and testing of component macros is strongly supported. All model components macros have a strict separation between data (using design variables) and model equations. Combined with a high level of model parameterisation, this resulted in a user friendly modelling environment with strong support for optimisation and data verification.

Modal flexible versus discrete flexible models

Main attention in the modelling process was put in defining the flexible panels constructing the solar array. In the defined macro library, all panels are defined as hyper structure entities. Using this approach, all connections to other model components are defined in a similar manner. Thus, parallel development of both two-dimensional, modal flexible and discrete flexible models was feasible with a minimal overhead.

ADAMS flexible bodies were used for the modal flexibility solar panel models. In ADAMS, flexible bodies are defined by importing modal data as calculated by an external Finite Elements (FEM) program. A special data file (the mnf file) is used to transfer frequency and amplitude data for a selected number of vibration modes from a FEM code to ADAMS. Craig Bampton modes, required for defining constraint connections to flexible bodies are transferred from the FEM code by defining master nodes. For each master node, constraint mode information is stored in the mnf-file. ANSYS was used to generate FEM models of quarter solar panels (see Fig. 4).

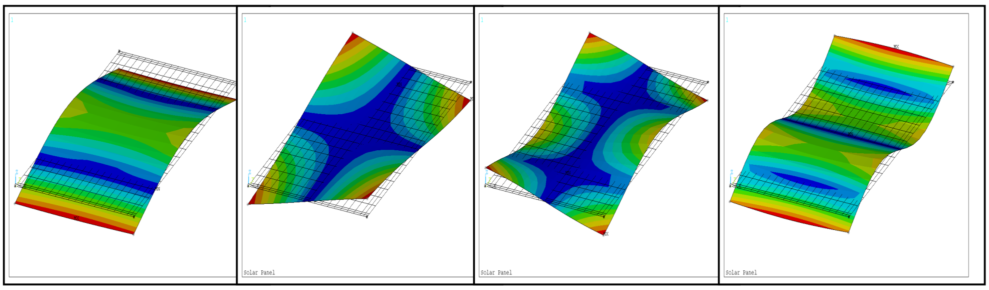

Figure 4: First 4 modes of a quarter panel FEM model (44.8, 54.5, 118 and 123 Hz)

Each flexible panel is modelled as a sub-structure of 4 identical flexible bodies. Each flexible body has 6 master nodes, 3 on the top edge and 3 on the lower edge. On the master nodes, dummy parts are defined for definition of connections to the different solar array components listed in the next paragraph. The sub-structuring was required to describe the non-linear geometric stiffening effect on the panels due to the curving deformation.

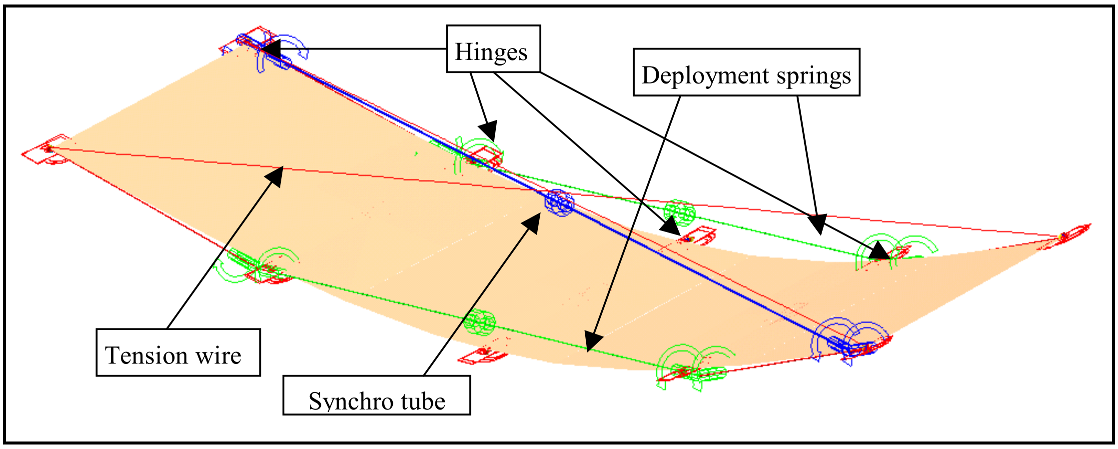

Figure 5: Four flexible parts in a solar panel model using modal flexibility

Figure 5 shows four connected flexible bodies in one solar panel. Diagonal lines represent two tension wires (thin) and a synchronization tube (thick). Along the joint line with 5 hinges, pre-loaded deployment springs are defined that generate the driving torques for the solar array deployment.

Besides using the modal approach, complete solar panel array models were also generated using a discrete flexible approach. The ADAMS discrete flexible beam macro was extended to create a parameterised discrete flexible plate macro. From the simulations, it was found that a discrete model with 15 rigid parts (3 X 5) per panel describes the non-linear curving behaviour similar to the modal flexible model. In both modelling methods, the remaining force in the tension wires with curved panels agrees with the requested rest force of 60 [N].

Models have been made of a single free moving panel to verify both the discrete flexible and the modal flexible approach. Vibration modes are calculated of the flat and curved panel to verify changes due to the geometric non-linearity. The results of the modal flexible panel are shown in Fig. 6. Due to the curving, the frequency of the first bending mode (half sine) has increased from 2.8 Hz to 3.3 Hz. This increase in structural stiffness is the main reason for applying panel curvature. The second and third vibration mode are also affected by the curving of the panel. The results of the discrete flexible model (see Fig. 7) show a good correlation to the results of the modal flexible method.

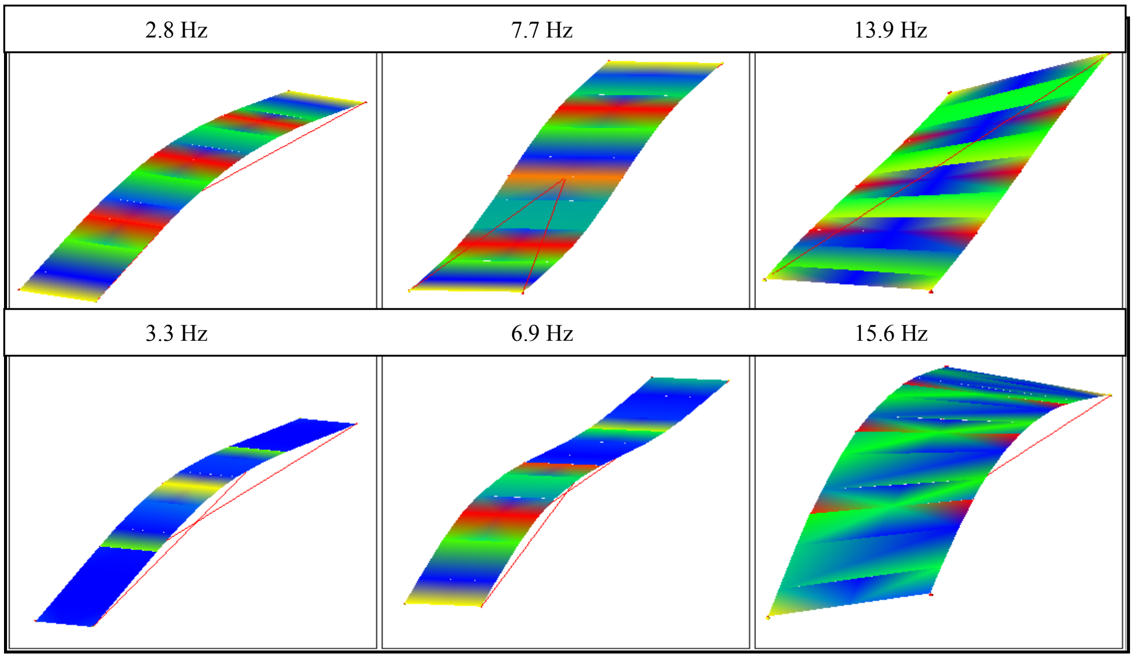

Figure 6: Vibration modes of a flat (top) and curved (bottom) solar panel using modal flexibility

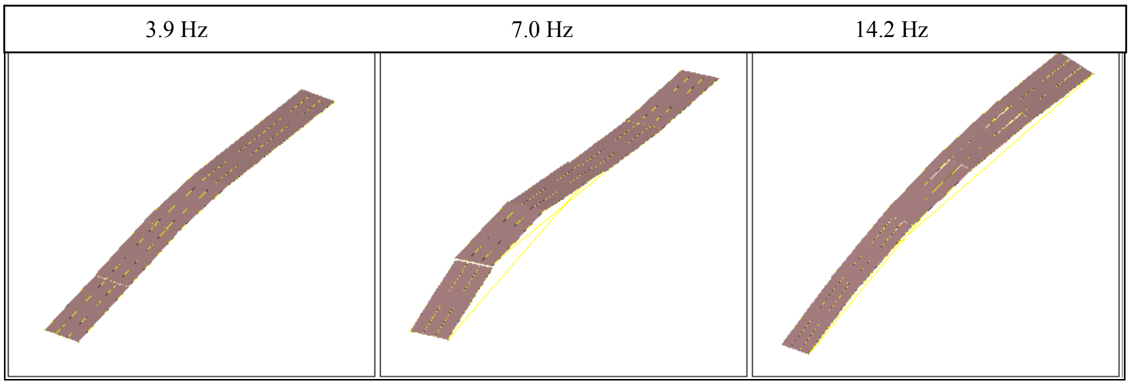

Figure 7: Vibration modes of a curved solar panel using discrete flexible elements

The complete modal flexible model contains 56 flexible bodies as the array has 14 panels. The rapid changes in model geometry due to the curving of the panels, combined with the large amount of flexible bodies in the model, caused problems to obtain stable simulation results. The new corrector algorithm, recently available in the ADAMS Gear [3] implementation, was required to obtain stable runs within feasible time limits. On average, a simulation run of 30 seconds array deployment time (in real time) with the modal flexible model takes 2 to 4 hours to simulate on a Pentium 3 PC. No special integration method was required to run the discrete flexible models. For these models, average run times varied between 1.5 and 5 hours depending on the defined parameter settings.

Because of these large simulation times, model parameter variations are initially performed with a 82 DOF discrete model. This model requires 2 minutes to complete a 30 seconds run, it describes the 2-dimensional behaviour of the system as curving of the panels is not yet included.

Model components and settings

All connection components were defined as macros with an identical interface to other model components. Each component macro is connected to rigid parts on the solar panel models. For the modal flexible model, dummy parts are created at the contact points.
Some details of model components relevant to the system behaviour are described:

- **Hinges**: At five locations the panels are connected using bushings elements. For the hinge stops, bi-stop functions are applied with an opening angle of 180 degrees (from stowed to deployed situation).
- **Tension wires**: Each tension wire is modelled as two (pull-only) forces and a centre part. A contact force is defined between the centre part and the panel to prevent the wire from penetrating the solar panels.
- **Damper wires**: Different options were evaluated to introduce damping during the deployment. In one design, two dampers are placed at the root of the solar array. The damper wires, distributing this force, are mounted between the centres of the short edges of each panel. The wires are modelled as point-point forces sharing the same wire force. The main effect, expected from this cable, is the counter-action of bending of the array. Components of the cable force perpendicular to the array deployment direction must compensate lateral deflection of panels in the array. This effect has been incorporated.
- **Synchronization tubes**: Torsion tubes are defined between two neighbouring hinge lines. To account for the spatial deformation, the tubes are modelled as a series of parts, hooke- and cylindrical joints and torsion springs.
- **Deployment springs**: The deployment spring model is identical to the synchronization tube. The preload in the deployment springs drives the deployment motion. In the initial design, the preload decreases linear from 0.30 [Nm] at the array root hinge line to 0.11 [Nm] at the top hinge line. The torsion stiffness of the springs is equal to 0.01 [Nm/rad] giving practically a constant torque throughout the full deployment. By modification of the preload torque in the first hinge and the bi-stops in the hinges, the solar array can be used for both 180 degrees and 90 degrees root-hinge deployment.

Deployment Simulation Results of the Complete Panel Array

Dynamic deployment behaviour of the array was simulated for both the 90 ^{o} and 180 ^{o} deployment angle of the root-hinge. In the latter situation, extra centrifugal forces due to the rotation of the complete array package affect system behaviour. Both cases were analysed for predicting the deployment trajectory to investigate any dangerous situations resulting from the deployment and the curving of the panels. It is critical that the panels will not collide with other parts of the satellite appendages. This means that the deployment must behave in a smooth manner and all panels must deviate as little as possible from their shortest path from stowed to deployed position. Simulation results are shown in Figs. 8 to 11. For each simulation, six snapshots indicate the course of the array during the deployment.

The following conclusions were drawn:

- The start of the 90 ^{O} root-hinge deployment with the 2-dim. model is quite smooth. After the initial stretching of the array (at 20 seconds) the panels do not come to a rest position because the kinetic energy in the panels creates large lateral oscillations.
- The 90 ^{O} root-hinge deployment simulation with the 3-dim. model (not shown) indicates that the curving of the panels has a large influence on the deployment (as expected). Panels that have curved completely will remain in a deployed angle with respect to their neighbours. This results in a stiffening effect of parts of the solar array. This effect was indeed already observed in the physical tests.
- The 180 ^{O} root-hinge simulation case indicates that no curving occurs in the first 30 seconds of deployment. Fig. 9 shows that the direction of the initial stretching is at an angle of some 60 ^{O} with the deployed direction, which is shown as a straight horizontal line. Therefore, angular velocities are introduced in the system resulting in large lateral deviations and possibility of collision between panels in the array.

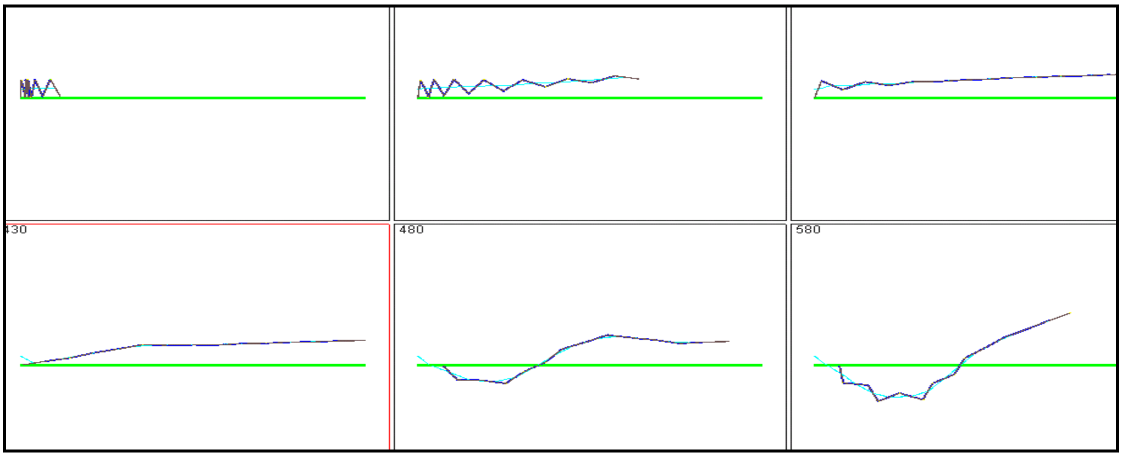

Figure 8: 90 degrees root-hinge deployment of the solar array using the 2-dim. model

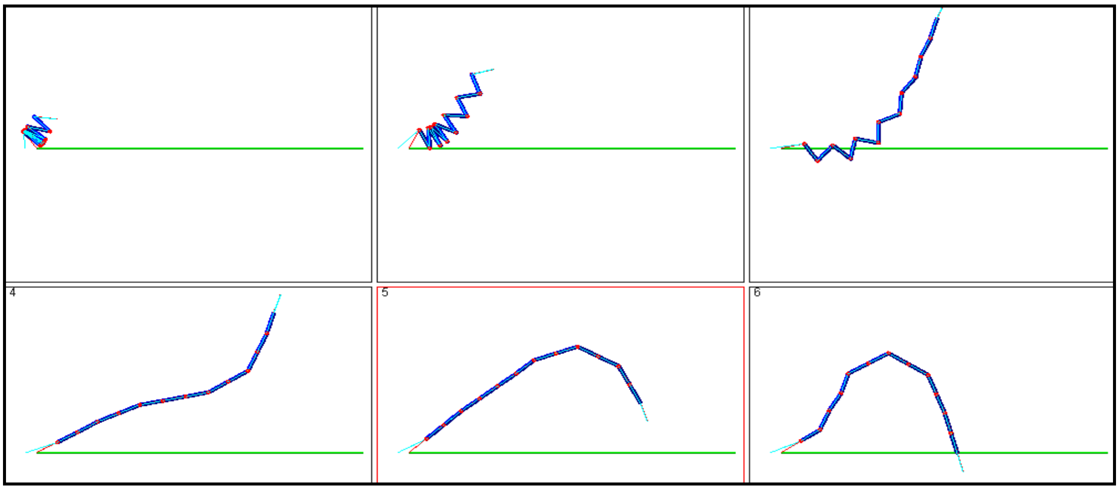

Figure 9: 180 degrees root-hinge deployment of the solar array using the 2-dim. model

- Fig. 10 shows the effect of extra viscous dampers at each hinge line for the 180 ^{o} case. Viscous damping torques are proportional to the hinge line angular velocity. Therefore, deployment of the array top part is delayed relative to the root part panels.

- Fig. 11 shows the 180 ^{o} root-hinge deployment using the modal flexible model. Initially, the flexible model behaves identical to the 2-dim. model. Similar to the observations in the 50 % scale measurements, curving of part of the panels already starts during the deployment. In the simulations, this causes a stiffening of the end part of the array influencing the trajectory. Due to the large angular velocities of the panels, the curving creates large forces and moments in the hinges. The tension between panels 4 and 5 becomes too large so the curved situation is reversed for some panels. As a result, large oscillations and force peaks occur in the system. It can be concluded from these analyses that a motion damper was mandatory.

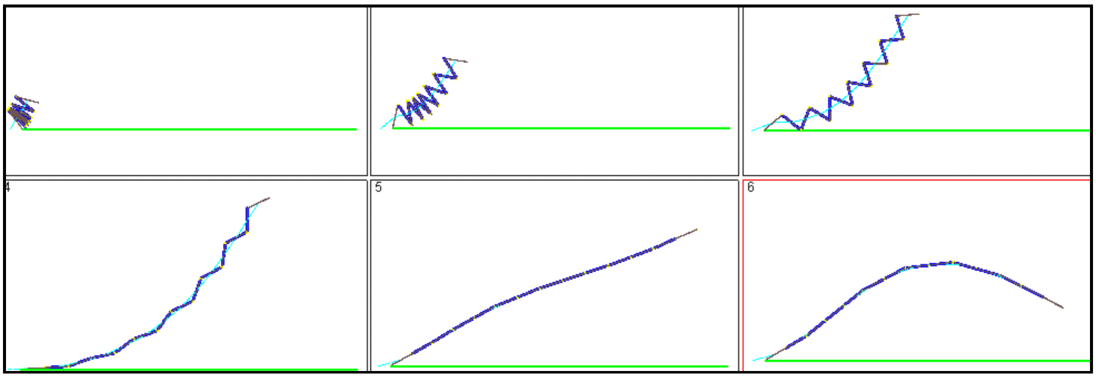

Figure 10: 180 degrees root-hinge deployment of the 2-dim. model with viscous damping

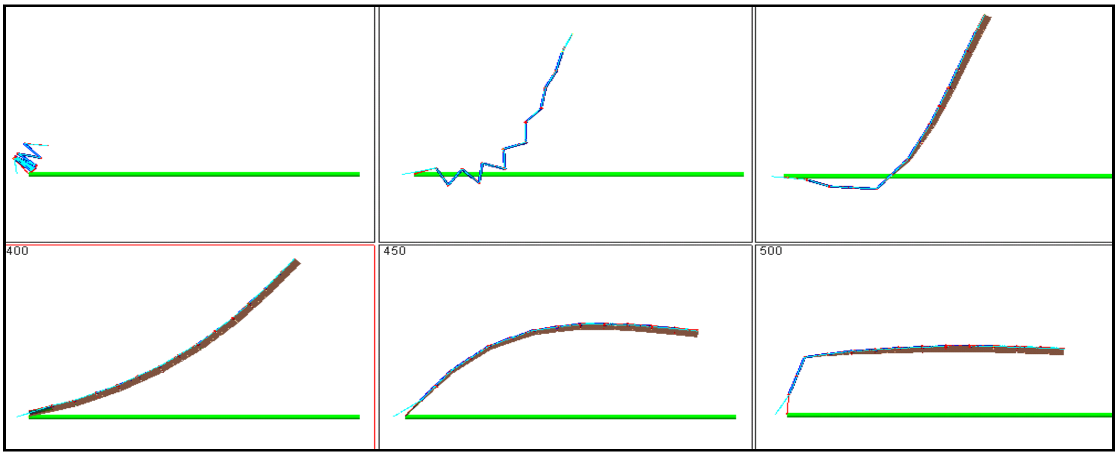

Figure 11: 180 degrees root-hinge deployment using a modal flexible model

Optimisation of the array deployment

By parameter optimisations, possible improvements in the deployment motion were investigated. From the preliminary simulations, it was concluded that the root panels of the array must finish deployment prior to the top part panels. This was motivated by the following observations:

- In cases where deployment of the top panels precedes, the curving of the panels introduces a shock wave starting at the top and moving towards the root. This shock wave counteracts the motion of the array towards the final deployed position and stimulates lateral oscillations.
- Stiffening of the already curved panels changes the bending properties of the complete array. If this occurs in the end part of the array, the root panels will encounter large acceleration, hence generating force and moment peaks due to the rotations of the array (see Fig. 11).
- In reality, it is unlikely that root panels that have already curved will jump back to an un-curved state. This means that remaining un-curved and un-deployed panels will be found at the top of the array. The deviation of panels from the final deployed situation will decrease reducing risk of collisions.

Possible parameter sets are discussed. Parameter changes are combined to optimise the solar array for meeting the desired kinematics behaviour.

- **Deployment Torques:** By increasing the preload torque in the root hinge, this hinge will precede the other hinges. The benefit is that the complete array will move more in the full-deployed direction.
- **Synchronization Tubes:** By adding stiffness in the synchronization tubes, the difference in opening angles between the different panels is reduced. As a result, the deployment is more equally divided

over the length of the array. For a 180 ^{o} root-hinge deployment, the rotational inertia of the deploying array will increase, requiring an increase of the root deployment torque.

- **Damping wires:** From the simulations, another effect was found of the damping wires. The damping forces appear to have a slowing effect on the deployment of the array top panels. As a result, deployment of the root panels precedes that of the top panels. This effect can be used to influence the location where curving of the panels will start.
- **Viscous Dampers:** Figure 10 shows that viscous dampers also stimulate preceding the deployment motion of the root panels relative to the top panels. The advantages of using these dampers, and possibly using different dampers over the hinges, must be weighed against the increase of mass and the technological difficulties in creating predictable viscous dampers for extreme temperature variations. For these reasons, damping wires along the small panel edges were preferred in the current design as it was based on known technology within Fokker Space. Further research may be required to identify the feasibility of application of individual hinge line rotational viscous dampers.

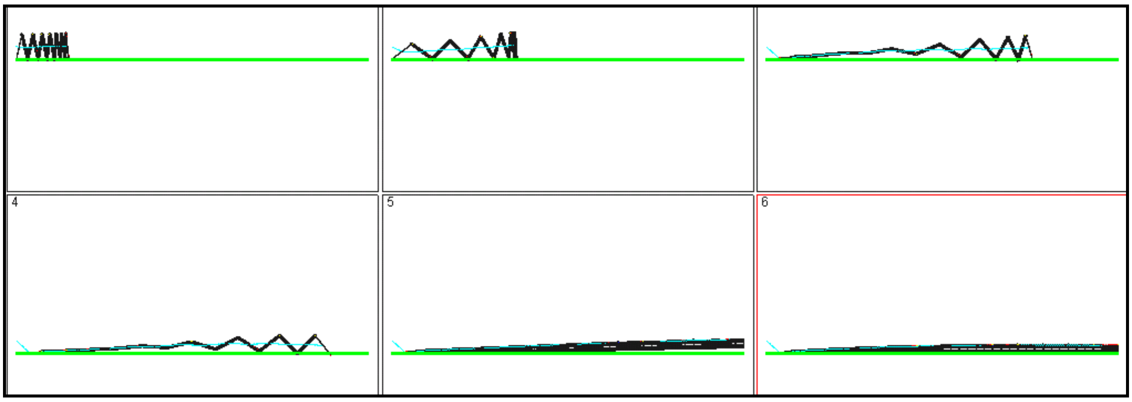

Figure 12: 90 degrees root-hinge deployment of the optimised flexible model

Results of optimised parameter settings are listed in Figs. 12 and 13. The settings used are:

- **90 ^{o} root-hinge deployment**: For this case, an optimised design was obtained by application of damping wires with a viscous damping coefficient of 2.0 [Ns/m]. Furthermore, the preload deployment torque in the root hinge was decreased from 0.3 [Nm] to 0.255 [Nm]. Fig. 12 shows simulation results for the discrete flexible model. The simulated overall wing deployment time for these settings was 60 seconds as the damper slows down the deployment process. The advantage of no lateral vibrations is evident.
- **180 ^{o} root-hinge deployment**: For this case, the root hinge preload deployment torque was increased to 0.678 [Nm]. Combined with a damping constant of 1.0 [Ns/m] in the damping wires and an increase of the synchronization tube stiffness to 1.60 [Nm/rad] this results in the deployment as shown in Fig. 13.

Conclusions

The following conclusions are drawn from the ADAMS simulations performed on the *Curwin* solar panels:

- Simulating non-linear behaviour of large models containing flexible elements is very feasible using a multi body code such as ADAMS. In this aspect, a competitive position can be claimed for the multi body approach when comparing to established FEM programs.
- For large models with repeating components such as solar panels, the application of a macro approach is advised. Especially when it allows creation of simple models, to do exploration simulations and component development tests, much time can be saved in simulation research.
- Using virtual prototyping, the critical sensitive parameters to be investigated during physical tests can be defined in advance.

- Use of discrete flexible models in parallel to use of modal flexible models for application in systems with simple geometrical shapes can increase efficiency. The overhead of extra work to be done is compensated by the extra possibilities in parameter checking and method validation.

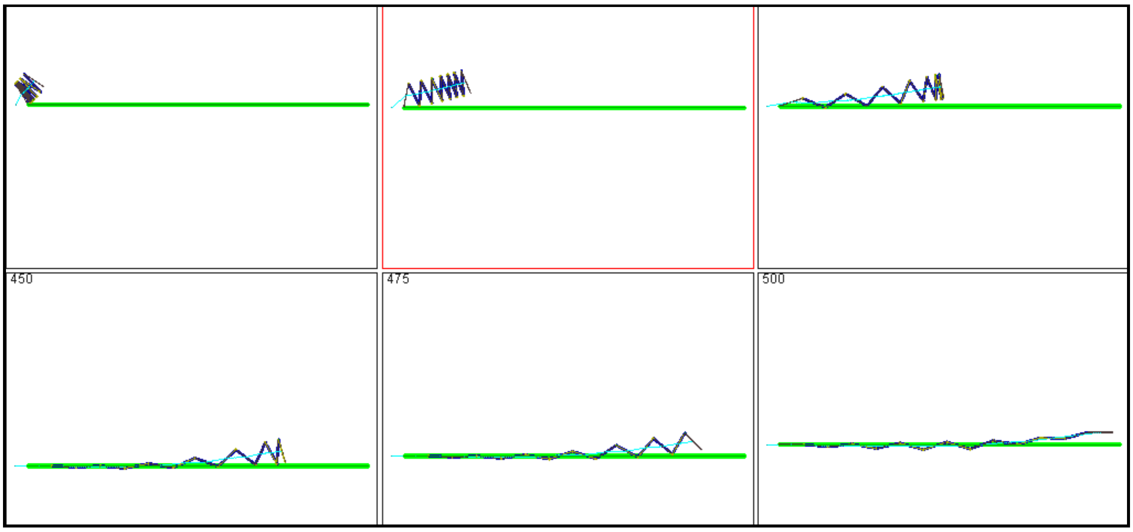

Figure 13: Optimised design for the 180 degrees root-hinge deployment, 2-dim. model shown

- The ADAMS integrators undergo continuous improvements. Simulation of rapid deformation changes in large models containing many flexible bodies became feasible (2 hours simulation for a 30 seconds run) using the new corrector scheme in the Gear integration algorithm. The new GSTIFF integration method [3] does not apply corrector error control to selected variables in the system. The effect of this change is to make the corrector convergence criterion less stringent. This integrator is considerably less likely to fail than the existing GSTIFF. The disadvantage, slightly less accurate results, is easily remedied using a tighter error tolerance.
- Physical tests were useful in evaluating the feasibility of the *Curwin* design. In the *Curwin* design, the spatial movements of the panels prove to be crucial for the operation of the system. Increasingly, the use of virtual prototypes becomes necessary to support the design of new space applications. This need becomes even more evident when one realizes that the design of future solar panels is moving in the direction of three dimensions. For such designs, zero gravity measurements could become practically impossible, making the use of dynamic simulations a necessary part of the product design.

References

[1] *Curwin Solar Panel System*: Patent No: US 6,091,016 and Europe 97204099.2

[2] *Breadboard test results of ½ scale Curwin model*: Phase B1 study note CONSA-FS-SER-094

[3] *The Simultaneous Solution of Differential Algebraic Systems*, Gear, C.W. (1971a). IEEE Transactions on Circuit Theory, CT-18, No.1, 89-95.

[4] *The Curwin design: A novel solar array concept for Constellations*, Henk J. Cruijssen, Gerard J.A.N. Kester, Fokker Space, Proceedings of the 35th Aerospace Mechanisms Symposium, Ames research Centre, May 9-11, 2001

[5] *The Curwin design, a novel solar array concept for Constellations*, Henk J. Cruijssen, Gerard J.A.N. Kester, Fokker Space, 28'th IEEE Photovoltaic specialists conference, 15-22 Sept. 2000, Anchorage, USA