MULTI-BODY DYNAMICS SOFTWARE TOOL: TWO CASE STUDIES

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ABSTRACT

ESA has a long experience in using multibody software in order to analyse complex dynamic systems. For this purpose, several software (such as DCAP, Adams, Samcef/MECANO, ...) are currently used. In this paper two mechanism-based problems are presented using multibody software.

In parallel, this study has been extremely useful in order to evaluate whether SIMPACK software, expensively used for terrestrial vehicle dynamics, can simulate aerospace mechanism systems.

Two case studies were selected: the first one is on simple boom deployment and the second one on a complex deployment system of the International X-ray Observatory (IXO) as designed in a Concurrent Design Facility (CDF) study at ESA. Throughout this paper, SIMPACK will be extensively compared with different multibody software available at ESA.

1. MULTIBODY DYNAMICS IN AEROSPACE APPLICATIONS

In recent years, space-system design has shown a clear trend towards increasing complex configurations. Typical examples are the use of several flexible components (antennas and solar arrays), the need for deployment and retrieval mechanisms, the demand for high precision pointing systems, and the increase in mission scenarios implying the assembly of large structures in space. This trend has also caused an evolution towards a multi-disciplinary design approach, particularly in the area of dynamics and control (Ref. [3]).

In order to study the performance of generic controlled dynamic systems, it is essential to have a dedicated tool, which allows the user to model, in a short time, the complex behaviour of the dynamic systems and their interactions with the control. In fact, some systems require a model with more than one body in order to take into account their different characteristics and their mutual dynamic interactions. This task is pretty complex and requires one to dedicate quite some time to understand the code and to validate the dynamic behaviour of the system.

A lot of research has gone into the development and improvement of multibody software, with the aim of reducing the time of modelling a system and the computation time required to run an analysis. Multibody software involve the derivation of the equations of motion for multibody systems, which are systems characterized by several bodies connected by hinges that permit relative motion across them. Robots, launchers and spacecraft including articulated appendages such as solar arrays are typical examples for such systems.

ESA has a long experience in using multibody software in order to analyse complex dynamic systems (Ref. [1]). For this purpose, several software (such as DCAP, Adams, Samcef/MECANO, ...) are currently used in different studies (Ref. [2], Ref. [6]).

Lately, based on past successful experiments (Ref. [4]), it was decided to take advantage of multibody software also in ESA Concurrent Design Facility (CDF) studies. Indeed, since CDF applies the concurrent engineering method to the design of future space missions (Ref. [7]), the use multibody software is an key tool for analysing performances at system level taking into account several subsystems.

2. CASE STUDIES SELECTION

Besides the main objective of this paper; to show how mechanisms can be analysed using multibody software, this study has been extremely useful in order to evaluate whether SIMPACK software, expensively used for terrestrial vehicle dynamics, can simulate aerospace systems. Based on the previous aerospace applications mainly on structural dynamics (Ref. [5]), SIMPACK s/w was used to analyse aerospace mechanisms. For this goal, two test cases have been selected.

The first case study is on a classic deployment of a magnetometer on top of a boom, which is deployed by a torsion spring and a rotary damper to control the deployment velocity (see Figure 1). When the boom

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reaches its final position, a latch mechanism is activated to keep the magnetometer in the open position. Despite the relative simplicity of the mechanisms, it requires that multibody software is able to handle changes of the boundary conditions for the flexible boom and, moreover, the simulation of the non linear latch element.

A more elaborate case study entails the International X-ray Observatory (IXO) deployment system (see Figure 9). The IXO study is part of a series of CDF studies to assess candidate missions for the Cosmic Vision program of ESA's Science Directorate (Ref. [8]).

3. BOOM DEPLOYMENT CASE

The system is composed of a 1 m long boom with a magnetometer as payload, which will be deployed by a passive rotational spring. A viscous damper to regulate the deployment speed is included in the system to decrease the end shock upon full 90 deg deployment. Once at 90 deg, the boom has to be held in its place with a latch.

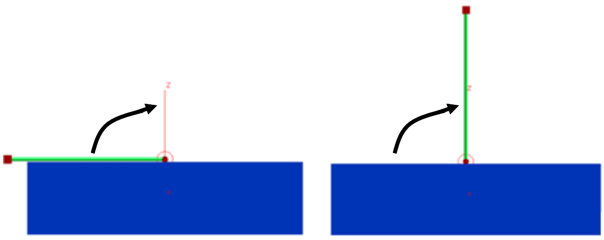

Figure 1: Boom deployment case. Left: Stowed configuration. Right: Deployed configuration

As already mentioned, this test-case has been run using some multibody software available in the ESA Structures and Mechanisms division, namely SIMPACK, DCAP, Adams. Even if not reported in this paper due to lack of time, a preliminary analysis has also been performed using Samcef/MECANO. In the frame of an ESA R&D project on a large deployment antennas (Ref. [9]), dedicate numerical algorithms, which consider energy and momentum conservation during long integration time also in presence of shocks, have been implemented. The preliminary results were quite promising.

3.1 Model Description

The boom deployment case study simulates the deployment of a magnetometer, mounted at the top of a flexible boom, from a spacecraft. For this analysis, the satellite is considered fixed to the inertial frame since we are not interested on the disturbances acting on the spacecraft. The system characteristics are reported in Table 1.

Using the classic multibody topology representation, the whole system is represented by two rigid bodies and one flexible one as reported in Figure 2. The only relative degree of freedom among the different bodies is the rotation around the x-axis between flexible boom and the spacecraft, on which a deployment spring and a deployment speed regulator are modelled as one spring-damper element.

Table 1: Boom deployment case: system characteristics

Flexible boom	Length = 1m Diameter = 200 mm Thickness = 1 mm Mass = 0.161 kg Aluminium hollow tube
Magnetometer	Mass = 0.3 kg
Deployment spring	Stiffness = 20 Nm/rad Pretension=62 Nm
Deployment speed regulator	Damping = 60 Nms/rad
A latch	Stiffness = 10^5 Nm/rad Damping = 7.08 Nms/rad

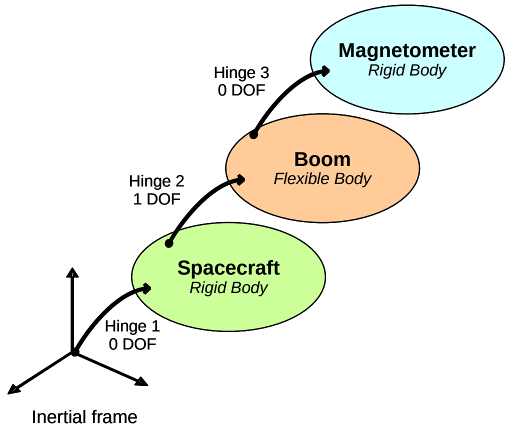

Figure 2: Multibody topology sketch for Boom deployment case

a. SIMPACK implementation

A latch element is not available in SIMPACK. However, SIMPACK provides the opportunity to model your own building blocks using the User Routines. A predefined structure in FORTRAN code provides the user the basis to build elements. The latch is defined in a way that makes sure bounce back cannot occur, in contrast to the available hard stop element in SIMPACK. Early models used the hard stop element, which resulted in non-linearity due to the bouncing back of the boom.

Based on SIMPACK manual, flexible elements can be modelled in various ways. The presented model contains a dynamic reduction of a Nastran model of the boom, using the Craigh-Bampton method. Nastran has been chosen, as it is commonly used for space applications. The interface between SIMPACK and Nastran has some peculiarities. The boundary conditions within Nastran have to be identical to the joint connecting the flexible element in SIMPACK, in this case a one degree of freedom rotation.

This way of working causes problems, since the mass, inertia and centre of gravity values of the boom in SIMPACK are not identical anymore to the original Nastran ones. It seems that SIMPACK does not always take the element into account on which the boundary condition has been imposed. To avoid this, there are no boundary conditions in the Nastran model of the boom. In other words, the Nastran model is considered completely free in space. In order to comply with the rule to have the same joint condition in SIMPACK, the joint has to have 6 degrees of freedom. This is possible if a dummy body is introduced. The dummy body is connected to the spacecraft with a rotational joint. The boom is connected to the dummy body with a six degree of freedom joint. In order to make sure that the boom is connected to the dummy body, both bodies have to be connected either with a spring element or with a constraint element. A constraint element yields the fastest simulations and has been selected in this case.

b. <u>DCAP and ADAMS implementation</u>

DCAP and Adams models are based on the same approach used in SIMPACK. However, the latching problem is directly solved using on dedicated elements and the Nastran interface has shown no particular problem.

c. <u>Analytical implementation</u>

An analytical closed form solution of the model has been obtained by integrating the second order differential equation of motion with the assumption of a rigid boom.

3.2 Comparison of Results

In order to compare the different multibody software, a linear analysis at a 0º deployment angle and non-linear time-based analysis are run.

a. <u>Linear Analysis</u>

The linear analysis calls for a different model. Due to the way the connection of the boom to the joint is modelled in the non-linear model, the linear analysis gives results which are not consistent with the other software packages for this model. In order to have representative values of the eigen-frequencies of this case study, a model is used without the dummy body. The consequence of this is that slightly wrong values of the boom mass and boom inertia are used. The impact on the linear analysis is negligible.

The linear analysis did not yield satisfying results. The third mode and fifth mode differ severely from the DCAP and ADAMS results. Further investigation revealed that an important improvement was made by increasing the amount of flexible modes of the boom that are taken in to account. The original model incorporated 8 modes for all the multibody software. The increase to 46 modes delivers results which are consistent with ADAMS and DCAP. It should be noted that the high number of modes has a negative effect on the computation time. The need for a high number of modes to obtain more correct results is unexpected.

Comparison of the different values for the first five modes can be seen in Table 2 and Figure 3.

Table 2: Linear analysis: Frequencies [Hz] of the first 5 modes

ID	Analyical	Simpack 8 mod	Simpack 46 mod	Adams	Dcap
1	0	0	0	0	0
2	6.51	6.51	6.51	6.51	6.45
3		7.07	6.55	6.52	6.51
4		85.48	85.12	86.88	85.34
5		99.87	86.73	86.89	85.40

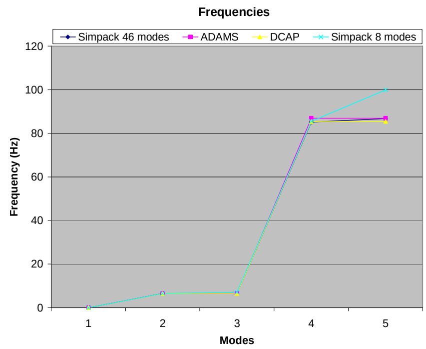

Figure 3: Linear analysis: Frequencies of the first 5 elastic modes

b. <u>Non-linear Analysis</u>

Following a "step-by-step" approach two different models have been created. One model has a rigid boom for comparison with the analytical solution and DCAP, whereas the second model has a flexible boom. The latter will be compared with DCAP and ADAMS.

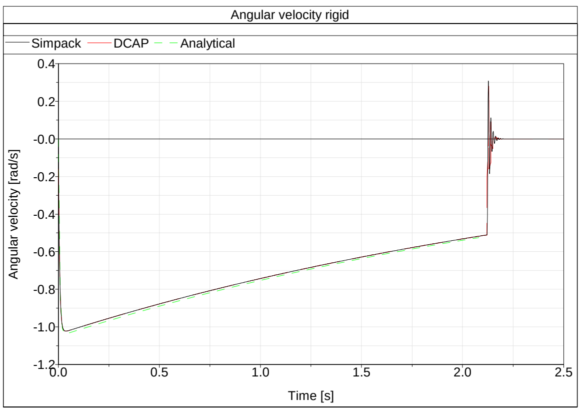

Figure 4: Angular velocity of the rigid boom model

The angular velocity of the rigid boom model, Figure 4, is consistent with the analytical solution and the DCAP model. The peak velocity upon latching is comparable.

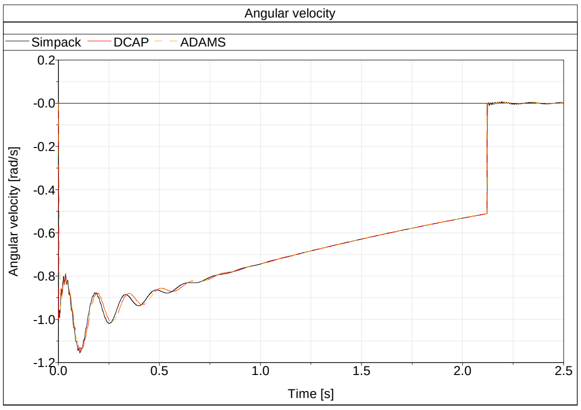

Figure 5: Angular velocity of the flexible boom model

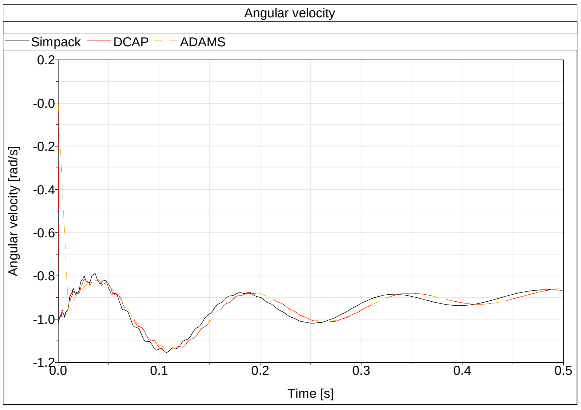

Figure 6: Angular velocity of the flexible boom model (zoom)

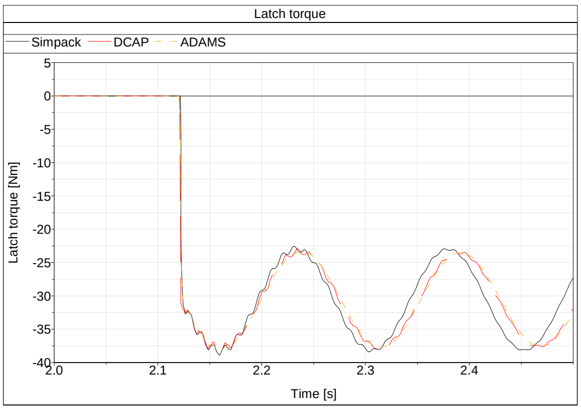

Figure 7: Latch torque for the flexible model

The angular velocity of the flexible boom model, Figure 5 and Figure 6, shows consistency between DCAP and ADAMS, but they lag behind with respect to SIMPACK, which can be seen more clearly in fig. 6. Although the overall behaviour of the SIMPACK model matches fairly well, the frequency content seems to be flawed, despite of a high number of modes that have been taken into account in the flexible boom.

The results for the latch torque, Figure 7, confirm the trends. The overall performance is good, even if there are some differences in the frequency content.

4. IXO DEPLOYMENT CASE

International X-ray Observatory (IXO) was originally intended to be a formation flying mission with two spacecraft flying at a 35 m distance, known as the XEUS (X-Ray Evolving Universe Spectrometer) mission. The new IXO design foresees to have a single spacecraft with a focal length limited to 20 m (see Figure 8). In order to reach this length a new deployment mechanism, based on articulating booms, has been designed.

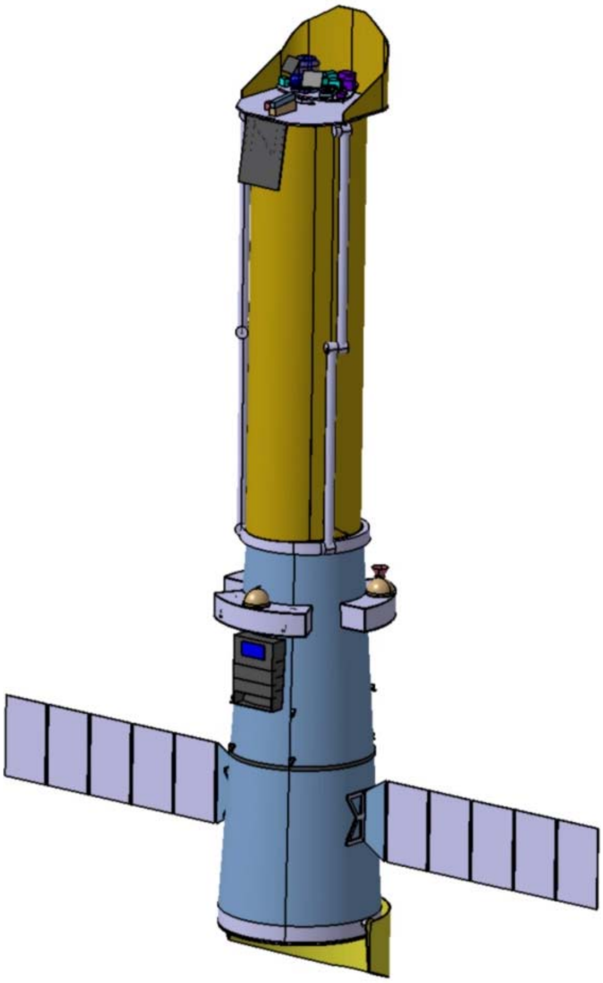

Figure 8: The International X-ray Observatory in deployed configuration

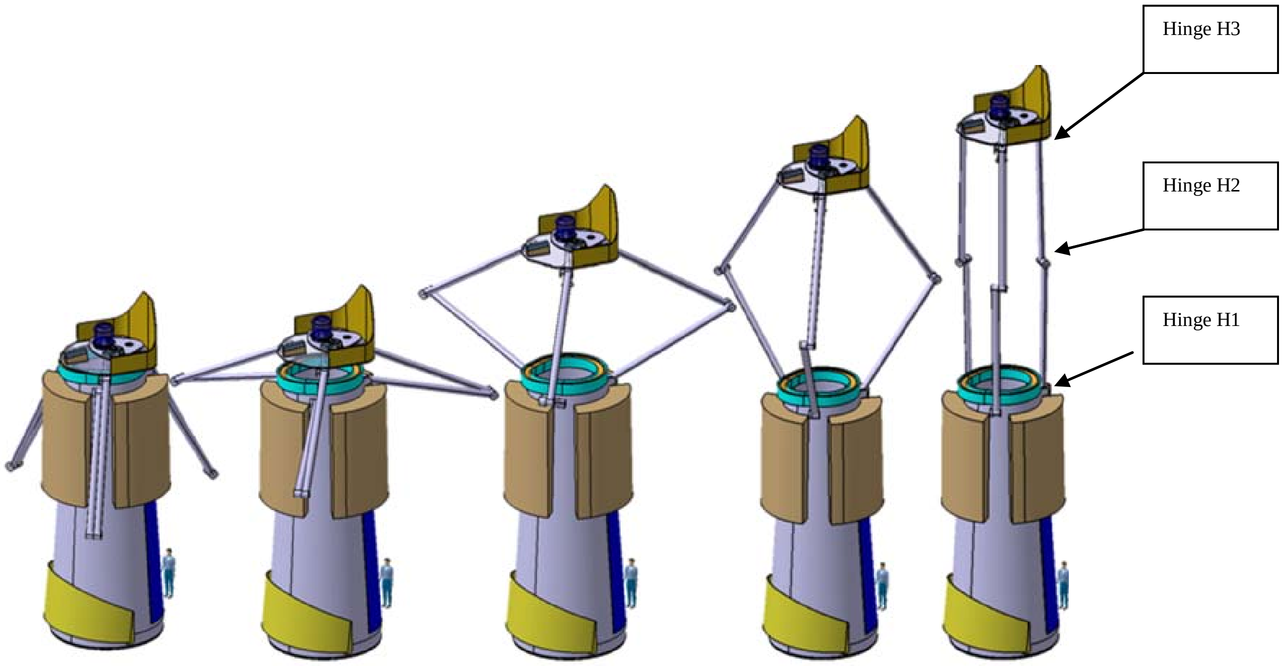

Figure 9: The IXO deployment sequence

Due to the complexity of this system, multibody software is the only viable solution to investigate such system, since a standard analytical solution is not available as in the deployment boom case.

In the frame of the CDF study, ADAMS has been used in order to verify the functionality of the deployment system, taking several disturbances (such as assembly errors, friction, actuator malfunctions, thermo-elastically disturbances and dynamic perturbations), which can influence the performance of the deployment system. Referring to ADAMS results as a reference, the same system has been implemented in SIMPACK. As the work is still ongoing, the comparison with ADAMS will be limited to the hinge velocities and the deployment torque for a frictionless deployment.

4.1 Deployment mechanism description

Figure 9 shows the deployment principle of the extension mechanism. The main principle is to use simple booms and motorised hinges, avoiding sophisticated telescopic or extendable booms. For each arm two booms and three hinges are foreseen as a baseline. Hinge activation results in boom rotation and thus the required translation of the instrument module.

The booms configuration is based on a tubular geometry manufactured from ultra high modulus CFRP material. Baseline diameter ranges 250 mm, for a 2.0 mm wall thickness.

In order to provide the required actuation torques, one active hinge (as a minimum) is needed for each arm. Standard ball bearings to provide rotational motion, stepper motor with reduction gear and pinion / crown wheel coupling can be envisaged for this application. One motorised hinge per arm, namely the one located at the middle arm point, H2 from Figure 9, is analysed for the baseline configuration. In principle also a location of the motorised hinge at arm lower end, namely H1, interfacing the service module, can be envisaged without incurring into deployment kinematics singularities. The ADAMS simulation performed in the frame of the CDF study however showed that H1 is more suited to overcome high parasitic moments in the three hinges.

4.2 Description of the model

The logic has been to first deploy the telescope according to an ideal configuration (frictionless, no errors from sensors, no harness through the booms…) then to individually apply the foreseen major sources of perturbation induced by the realistic features of the hardware.

The following features need to be introduced:

- Stiffness in the hinge elements
- Friction in the hinge elements (including cross-axis friction components)
- Parasitic moment due to harness routed through hinges
- Lateral (orthogonal to telescope axis) forces due to inaccuracies during AIT phases on ground
- Axial resistance force due to shroud deployment
- Asynchronous arms extension motion
- Disturbances induced by different boom lengths

The SIMPACK model as described hereafter includes the stiffness in the hinges. Friction has not yet been introduced, as it doesn't have a big influence in the one load case under investigation, namely the ideal deployment of IXO without any loads.

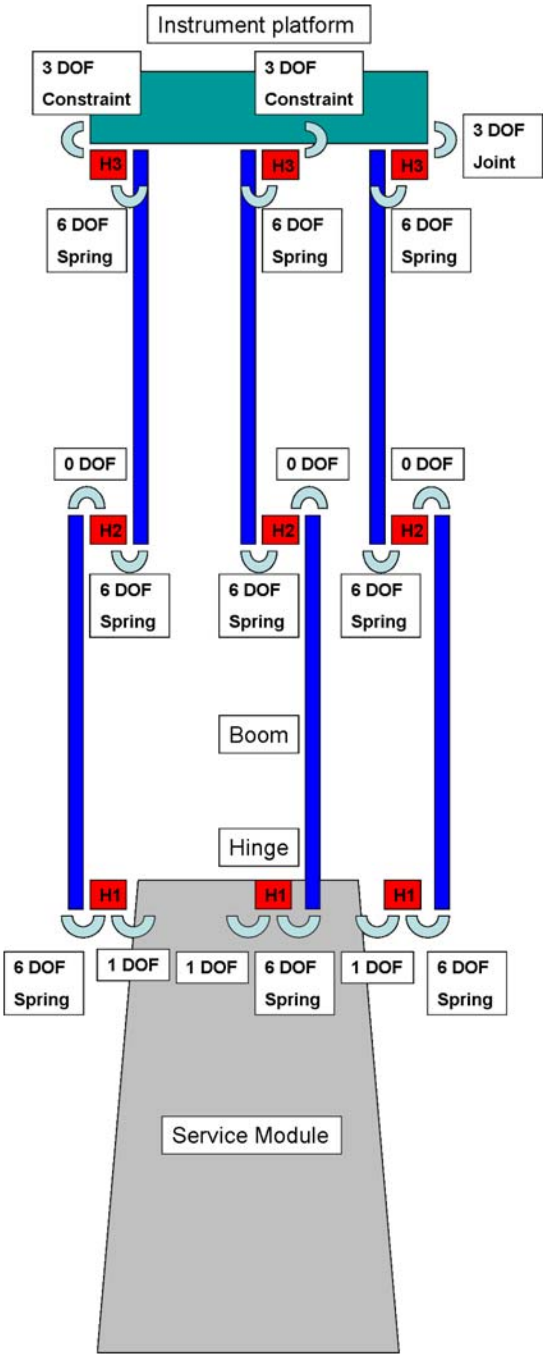

Figure 10: IXO schematic model

A schematic overview of the IXO model can be seen in Figure 10. It shows all the different elements incorporated in the model and the way they are connected to each other.

The lower hinges H1 are connected to the service module with 1 degree of freedom (DOF), one rotational joint. As the hinges themselves have a considerable mass in comparison to the booms, the hinges are modelled as a body. As the hinges are already present, they can fulfil the task of dummy body for the attachment of the flexible bodies. No extra dummy bodies are necessary. The flexible booms are connected to the hinge with a 6 DOF joint. The connection between the boom and the hinges will be made with a spring, not with a constraint as in the first case study. The stiffness of the spring making the connection has the values of the hinge stiffness.

The connection between the lower boom and hinge H2 is made with a 0 DOF connection. This does not mean that all movement is locked, but that the hinge motion follows a prescribed motion profile (rotation around the x-axis versus time). The prescribed hinge rotation, velocity and acceleration are shown in Figure 11.

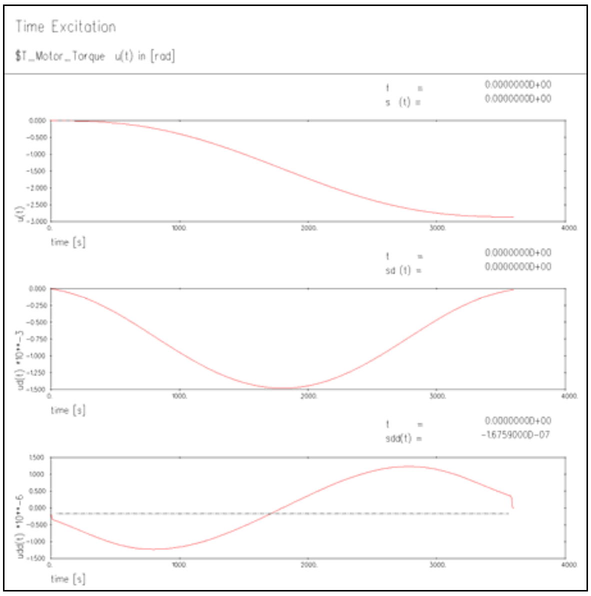

Figure 11: Hinge input. Top: Rotation. Middle: Angular velocity. Bottom: Angular acceleration.

The connection between H2 and the upper boom is identical to H1 and the lower boom. The H3 hinges are also connected to the upper booms with a 6 DOF joint and a spring representing the hinge stiffness. H3 of arm one is connected to the instrument platform with a 3 DOF spherical joint. As the number of joints in SIMPACK is limited to the number of bodies, a closed loop system like IXO has to be modelled using constraints. The connection between H3 of arm two and arm three are therefore made with 3 DOF spherical constraints. The connections between the arms and the instrument platform are made with 3 DOF, whereas in reality these joints will only have one degree of freedom. This results in an over constraining of the system, which works in reality due to the flexibility of the booms. Modelling this successfully has proved to be difficult; therefore this paper presents the SIMPACK results with the assumption of spherical connections. The ADAMS results do have a 1 DOF joint.

4.3 Comparison of results

Figure 12 shows cinematic results. The results of SIMPACK and ADAMS values for the angular velocities of the three different hinges of one arm are presented. The curves are not 100% coincident, but these first results are promising in confirming the validity of the SIMPACK model.

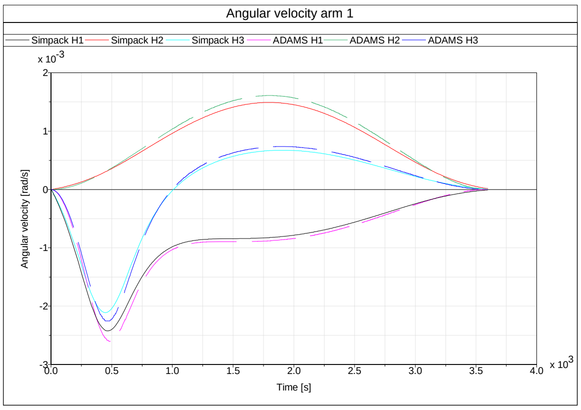

Figure 12: Angular velocity of the hinges for arm 1

The results for the motor torque proved more difficult to obtain, since early results showed very noisy curves. The solution was found by increasing the maximum integration step for the solver. The resulting maximum integration step is 15 s. The requested absolute and relative accuracy is $2.5 \cdot 10^{-7}$.

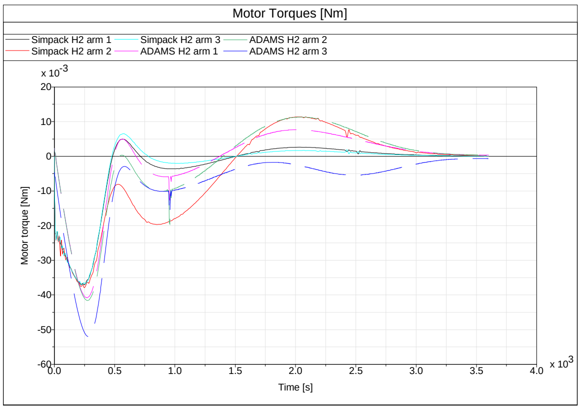

Figure 13: Motor torque for the three middle hinges

The SIMPACK results for the motor torques, torques in H2 of the three different arms, show a similar trend with respect to the ADAMS results (fig. 12). The general shape is similar; however, the curves are not close to being coincident.

The differences between the SIMPACK and ADAMS results could be a result of the previously highlighted difference at the level of the connection with the instrument platform.

A different possible explanation of the discrepancies can be found in the input for the actuation hinges. Minor differences between the angular acceleration in the SIMPACK and the ADAMS model might be the reason of an overall error. This suspicion is due to the fact that the discrepancy between SIMPACK and ADAMS is at its largest in the very beginning of the simulation, which can clearly be seen in Figure 13.

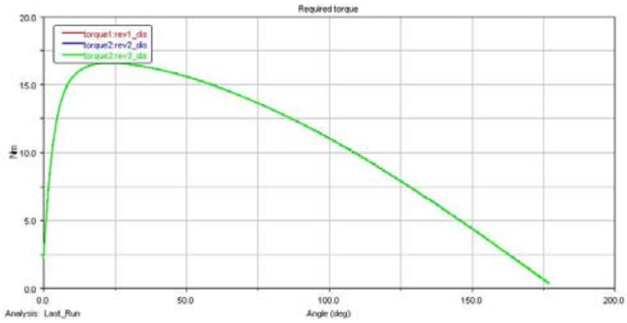

Figure 14: ADAMS actuation torque under a 10 N axial load [1]

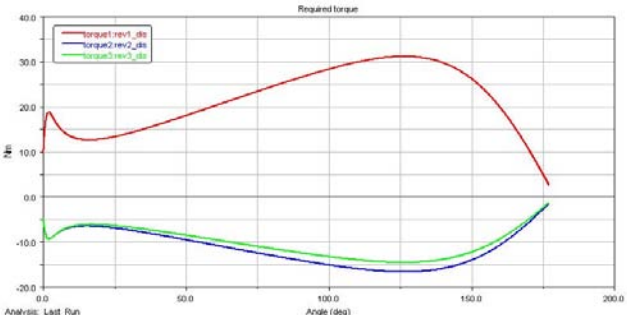

Figure 15: ADAMS actuation torque under a 10 N lateral load [1]

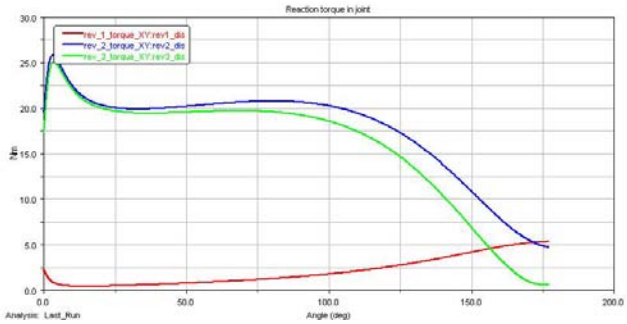

Figure 16: ADAMS hinge cross axis moments due to 10N lateral load [1]

Figure 14, Figure 15 and Figure 16 show the ADAMS results for two load cases. The actuation torques prove to be much larger with these disturbances with respect to the previously compared results for a deployment without perturbations. This gives rise to the suspicion that the existing differences between SIMPACK and ADAMS will become negligible once more demanding load cases are addressed.

5. CONCLUSION

In this paper two case studies in the field of aerospace mechanisms have been analyzed using multibody software. In particular, for the IXO problem, multibody software is the only solution to address such a complex system.

About using SIMPACK software for aerospace mechanism applications, this paper highlights a potential problem: the interface between SIMPACK and Nastran.

For the simple boom deployment case, different models have been implemented in order to obtain reasonable results for the linear and non-linear analysis. The linear analysis for the boom deployment case shows that the SIMPACK results similar to the other multibody software results. However, a higher number of modes have to be taken in account. This increases calculation time. The non-linear analysis delivers good overall results. However, the frequency content of the results is flawed. This seems to be an overall trend for SIMPACK, as the results obtained in the linear analysis already needed an abnormal number of modes for the flexible boom to obtain decent results.

For the IXO case, the first preliminary analysis of the SIMPACK model provides promising results. This confirms the relevance of SIMPACK as a software tool for this application. However, some improvements of SIMPACK model should still be performed. Mainly, a fully representative model, the DOF of the connections with the instrument platform have to be reduced to comply with ADAMS, and friction characteristics should be introduced. Moreover, when the SIMPACK and NASTRAN interface is clarified, a linear analysis should be performed.

Future activities are foreseen on a more detailed investigation on SIMPACK and Nastran interface taking advantage of SIMPACK support provided by INTEC. In particular, problems occur when simple elastic models are taken into account as it was the case of those case studies. For more complex structures, no main problem was experienced by ESA (Ref. [5]).

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