Deployment and Latching Simulation of Large Reflectors

Sidharth Tiwary[1], B. Lakshmi Narayana[1], B.P. Nagaraj[1], G. Nagesh[2] and C.D. Sridhara[3]
[1] Engineer, SMG [2] Project Director, Chandrayaan-2, [3]Deputy Director, SPA
ISRO Satellite Centre , HAL Airport Road,
Bangalore-560017, India
Email: sidharth@isac.gov.in

***Abstract*— Large deployable reflectors are widely used in satellite communication. These are light weight and flexible. The reflectors are mounted with revolute joint hinges to deploy and cam and roller mechanism to latch. These reflectors are stowed during launch to accommodate within the launch envelope. In orbit, the reflector is deployed and latched at the end of deployment through cam and roller mechanism to the intended position. The reflector deployment is a mission critical activity hence it requires detailed modelling and analysis to ensure positive deployment and latching. This paper presents the dynamics of reflector from the stowed to deployed configuration and latching simulation which has been carried out in a single finite element software-ABAQUS. The reflector and reflector hinges are modelled as a flexible body during deployment. The contact between roller and cam is modelled as a surface to surface contact. The finite sliding approach is used to continuously update the contact connectivity during deployment. The time variation of loads induced in different components of hinge after latching is evaluated.**

Keywords—FEA; Deployment-dynamics; contact; latching;

I. INTRODUCTION

Large reflectors are kept stowed in the spacecraft during launch and deployed in orbit. Due to its large size it is held at couple of locations by a hold down mechanism during launch. The number of hold-down points is decided based on the structural/frequency requirements and the inertia forces due to launch and manoeuvres during orbit rising. The hold down mechanism on actuation releases the reflector. The reflector gets deployed and latched to the intended position in the orbit. Any malfunction of the mechanism will cripple the satellite mission and may also result in total loss of the satellite. Hence the deployment dynamics of the reflector has to be analysed to ensure positive release of the mechanism, proper deployment and final latching.

The latch up moment was estimated for the single solar panel by transient analysis and energy method [1]. In the *transient analysis*, the spacecraft appendage is considered rigid till the latch up. The velocity distribution of the satellite appendage is triangular and is obtained from the rigid body dynamics simulation. This latch up velocity is taken as initial condition for transient dynamic analysis to estimate the latch up forces and moments at the support point. The *energy method* is based on the assumption that the latch up kinetic energy is absorbed by the deployed solar panel in the form of strain energy. The loading distribution is assumed triangular. The magnitude of loading is the product of equivalent angular acceleration, mass per unit length of appendage and distance along the appendage. The strain energy of the deployed array is calculated for unit angular acceleration. This strain energy is scaled to latch up kinetic energy to obtain equivalent angular acceleration. The energy and transient analysis gives similar results for the single revolute joint systems. The transient analysis is used as it can be easily be implemented in the commercial finite element software. The deployment dynamics of the antenna used in satellite also latch at the end of deployment. These have one rotational degree of freedom during deployment unlike solar array which are multi degree of freedom connected by closed control loop. The latch up was modeled for the antenna deployment mechanism by using transient dynamic analysis by considering the flexibility of hinge mechanism [2]. The latch up was modeled by both energy and transient analysis for a large antenna with damper [3]. The deployment and latching of a new type of Hold down and Release Mechanism of a large antenna considering the flexibility of the deployable link is studied through transient dynamic analysis [4]. In this paper the deployment dynamics of reflector with hinge is studied assuming reflector and hinge as flexible during deployment. The latch up is modelled by considering the surface to surface contact conditions of moving elements of the hinge during and after deployment in ABAQUS[7] finite element software which was not considered in previous works. The paper is organised as follows. Section 2 presents the system description. Modelling for deployment and latching is presented in Section 3. The results and discussion are presented in Section 4. Finally the conclusions are presented.

II. SYSTEM DESCRIPTIONS

The deployment mechanism for reflector consists of two hinge assemblies to mount the reflector and two hold down assemblies to clamp the reflector on east/west faces of the spacecraft as shown in Figure 1. The hinge assembly shown in Figure 2 consists of an inboard bracket connected to the spacecraft and the outboard bracket connected to the reflector. These two brackets are interlinked by a mono-ball spherical bearing mounted on a shaft supported by the inboard bracket. The spherical bearing which is dry lubricated takes care of misalignment between the two hinge assemblies. The two spherical joints along the axis form a revolute joint. The preloaded torsion springs mounted at the spherical joints provide the required torque to deploy the reflector. The inboard bracket will have a roller and a locking mechanism which moves over the cam surface of the outboard bracket and latches into a notch provided in the sector of the outboard bracket at the end of deployment. The two torsion springs mounted on the locking linkage shaft keeps the roller in contact with the sector. The reflector changes from mechanism to a structure after latching. The diameter of the reflector is 2.0 m with a focal length of 2.4 m. The reflector is tilted towards Y_s-axis by 3.76 deg. The reflector is required to deploy through 77.0 deg before latching. The reflector latches to its intended orientation after deployment as shown in Figure-1

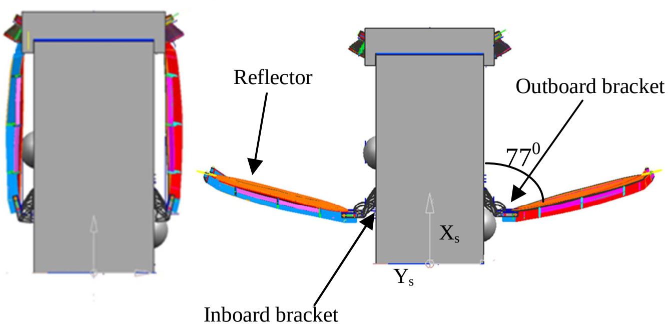

Figure 1: Stowed and deployed configuration of reflector

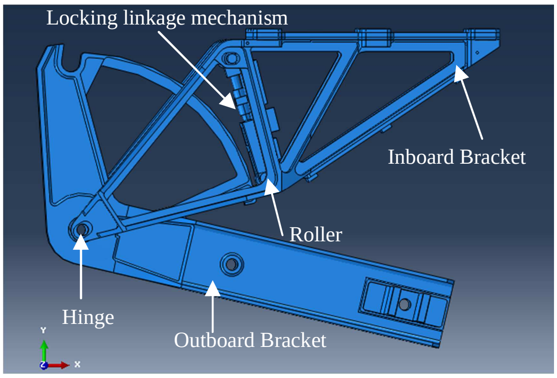

Figure 2 : Hinge Assembly

III. MODELLING

The finite element model of reflector along with hinge assembly is shown in Figure-3(a). The reflector is modelled by using S4R quad elements. These are 4 noded linear quad elements with reduced integration. This element allows transverse shear deformation and uses thick shell theory for the increased shell thickness and uses discrete Kirchhoff thin shell theory for the reduced thickness. The finite element model of the hinge assembly is shown in Figure 3(b). The outboard and inboard brackets are modelled using tetrahedral elements (C3D4) which is a four noded linear tetrahedral element.

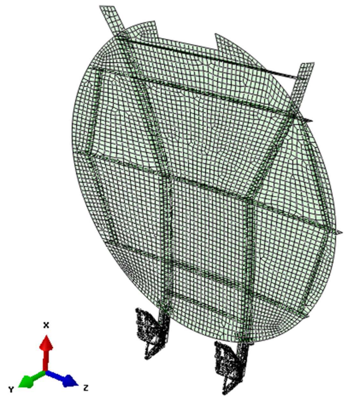

Figure 3(a) : Stowed configuration of reflector assembly

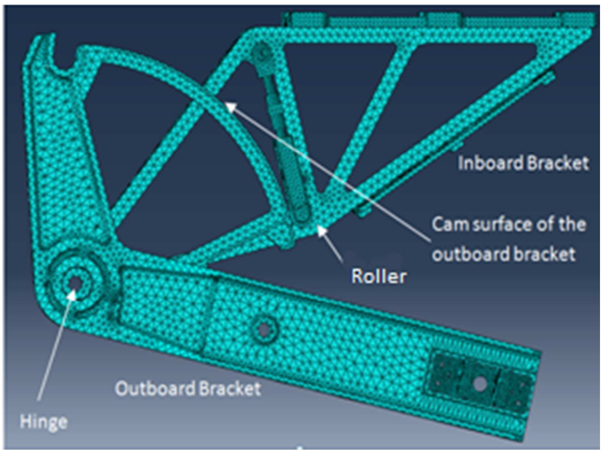

Figure 3(b) :Reflector hinge Assembly

Outboard bracket and in-board bracket hinge location is connected by two multi point constraints (MPC). The revolute joint is modeled connecting these two MPCs by another MPC and by releasing the Z axis rotation of the newly connected MPC. Similarly revolute joint is created by connecting the locking linkage to the in-board bracket. The rib of the reflector is connected to the outboard bracket through MPC's at two points. The inboard bracket which gets connected to the spacecraft deck at six bolt positions is given fixed constraint. The bolt location of inboard bracket is shown in Figure-4.

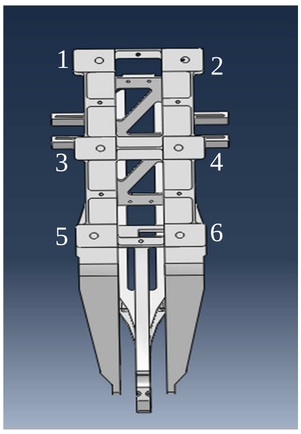

Figure 4 : Bolt locations of inboard bracket

A. *Modelling contact*

The rolling phenomenon of the roller over the cam surface of the outboard bracket is simulated using the contact modeling in the ABAQUS software. Surface to surface contact is defined between the cam surface and the roller. The roller is defined as slave surface while the cam surface is defined as the master surface. The master surface can penetrate into the slave surface hence the meshing of the roller is refined as compared to the cam surface. The surface to surface contact methodology enforces constraints in an average sense over a finite region, rather at discrete points as in the traditional node-to-surface approach. The relative motion of two contact surfaces is modeled by finite sliding approach. This approach allows arbitrary relative separation, sliding and rotation of contacting surfaces. The finite sliding approach ensures that the contact connectivity information is continuously updated. The contact area and contact pressures are calculated according to the deformed shape of the model. The contacting surfaces are adjusted to be precisely in contact at the beginning of the analysis. Friction coefficient of 0.1 is defined between the contacting surfaces. Linear penalty contact formulation is implemented to approximate the pressure over closure for the hard contact. In this method the contact force is proportional to the penetration distance. The penalty stiffness is taken as 10 times the element stiffness. This contact allows some degree of penetration between the meshes. This formulation mitigates the over-constraint issues and reduces the number of iterations required in the analysis due to the numerical softening associated with it. The equations of motion along with contact equations are numerically solved by implicit dynamic Hilber-Hughes method.

IV. RESULTS AND DISCUSSIONS

The physical parameters of the deployment mechanism are shown in Table-1. Dynamic implicit method is used to evaluate the joint angle and joint velocity of the reflector during deployment. The stowed and deployed configuration of the reflector is shown in Figure-5. As the roller moves over cam surface the contact interaction changes continuously and finite sliding approach is used to continuously update the contact connectivity information.

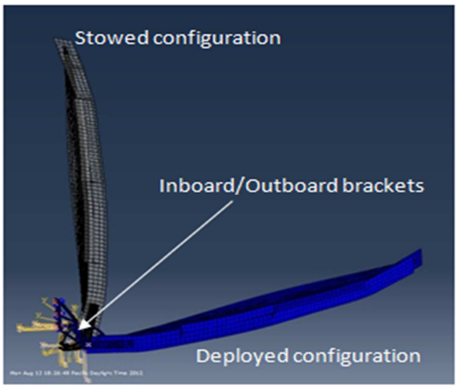

Figure 5 : Stowed and deployed state of the reflector

TABLE 1: PHYSICAL PARAMETERS OF HOLD DOWN MECHANISM

Parameters	Value
Total mass of the reflector with hinge assembly	17.8 Kg
Spring Stiffness of each spring	0.08731Nm/rad
Pre-rotation angle of torsion springs	270 deg
Deployment angle of the reflector	77 deg

The variation of angle of deployment and angular velocity of the reflector with time is in Figure-6(a) and Figure-6(b) respectively. The reflector takes 9.6 seconds to move from the stowed configuration to deployed configuration. The measured on-orbit deployment time is 9.5 sec and matches well with the computed data. The peak angular velocity just before latching is -14 deg/sec. It can be observed from Figure-6, 7 that the angular displacement and the angular velocity shows large oscillations due to the flexibility of the reflector and hinge assembly after latching and very small oscillations as the reflector hinges are very rigid during deployment. The large oscillations are due to energy transfer after latching. The tip deflection of the reflector during the deployment is very small as the reflector is very rigid and increases to $0.1\text{x}10^{-3}$ m after latching is shown in Figure-7.

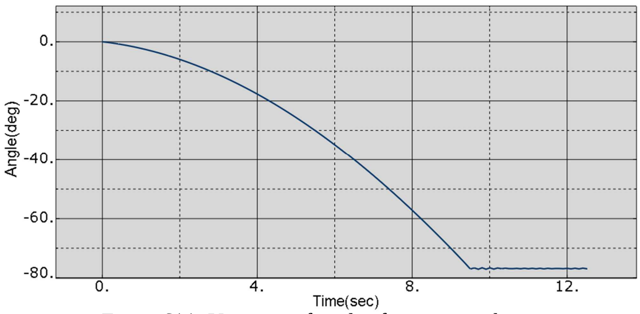

Figure 6(a): Variation of angle of opening with time

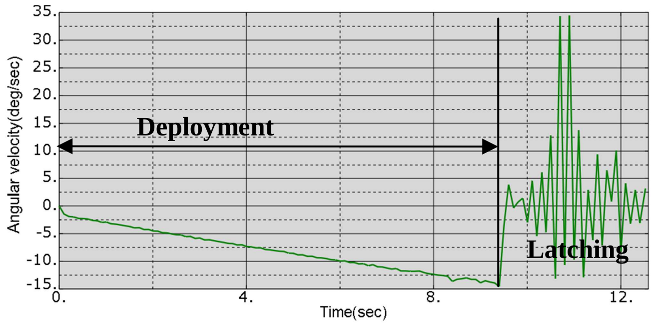

Figure 6(b): Variation of angular velocity with time

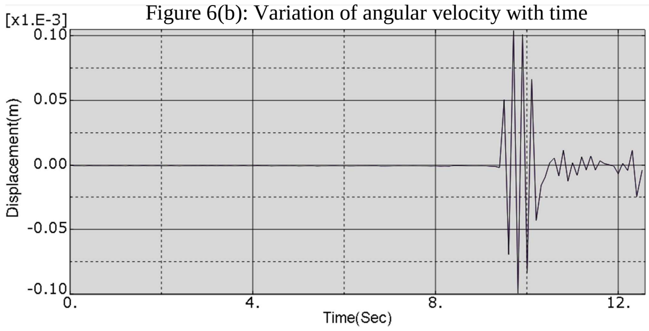

Figure 7 : Variation of tip displacement of reflector with time

Reaction force and moment during deployment and latching is monitored by connecting an MPC to six bolt hole locations as shown in Figure-4. As can be observed from Figure-8(a) and 8(b) that the reaction loads are negligible during the deployment and after latching the reaction load increases due to the transfer of kinetic energy of antenna to strain energy of antenna and hinge. The peak reaction force 100 N and peak reaction moment 178 Nm (Figure-8). This data is used in designing the inboard and outboard brackets and the antenna interfaces.

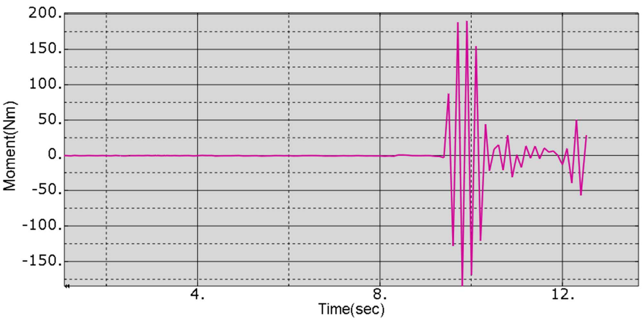

Figure 8(a): Latch up moment observed during deployment.

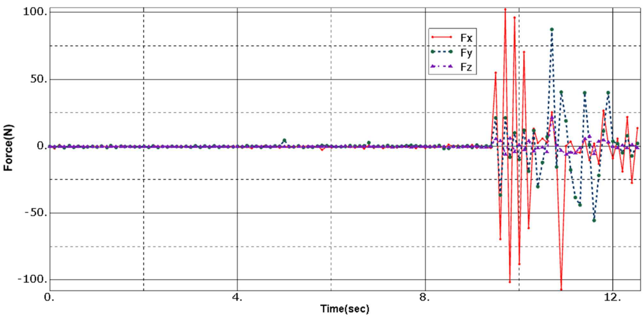

Figure 8(b): Reaction forces at all the six hold-downs of the inboard bracket base

Hitherto NASTRAN was used for similar analysis. As ABAQUS was used for the first time a comparative study was carried out. The Latch up moment evaluated through transient dynamic approach by using NASTRAN finite element software by giving latch up velocity as the initial condition [4]. It is observed that the results obtained from ABAQUS software is marginally higher than that predicted by NASTRAN as shown in Table-2. Numerical algorithm built into the individual software and the type of element formulation used could be the reason for the variation in these values.

TABLE 2: LATCH UP MOMENT COMPARISON AS OBSERVED FROM DIFFERENT SOFTWARE.

Location	Nastran (Nm)	Abaqus (Nm)
Top Hinge	170.0	178
Bottom Hinge	173.0	179

The latching induces stresses in brackets and large loads on the hinge joint. The deployment mechanism needs to be designed for these latch-up loads. Figure-9, 10, 11 represents the maximum von Mises stresses in the outboard bracket, inboard bracket and the reflector respectively during deployment and latching. The peak von Mises stress in the out-board and in the in-board bracket is 6.946×10^7 N/m^2 and 5.431×10^7 N/m^2 respectively and occurs at 9.7 sec. These stresses are less than the allowable yield stress of 2.4×10^8 N/m^2 of Aluminium-6061 material. The peak von Mises stress in the roller and locking linkage are 2.85×10^7 N/m^2 and 2.15×10^8 N/m^2 at 9.8 sec as shown in Figure-12a and Figure-12b respectively. The material used for roller and locking linkage is Titanium and the maximum stress on the locking linkage is less than the allowable yield stress of 8×10^8 N/m^2 for Titanium.

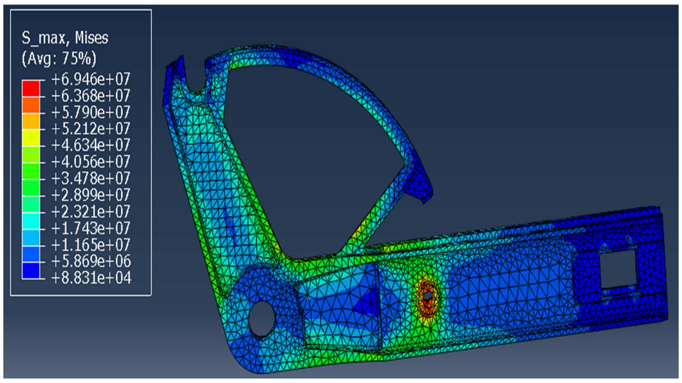

Figure 9 : Max stress(Pa) on outboard bracket during the analysis

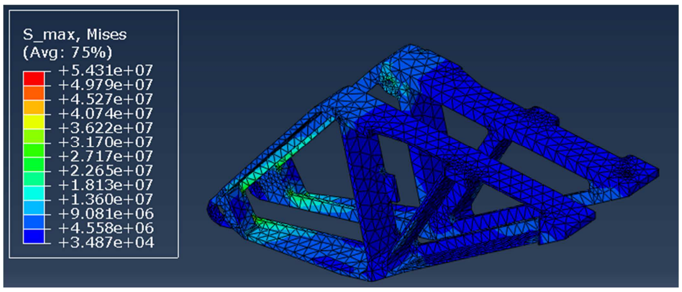

Figure 10: Max stress (Pa) on inboard bracket during the analysis

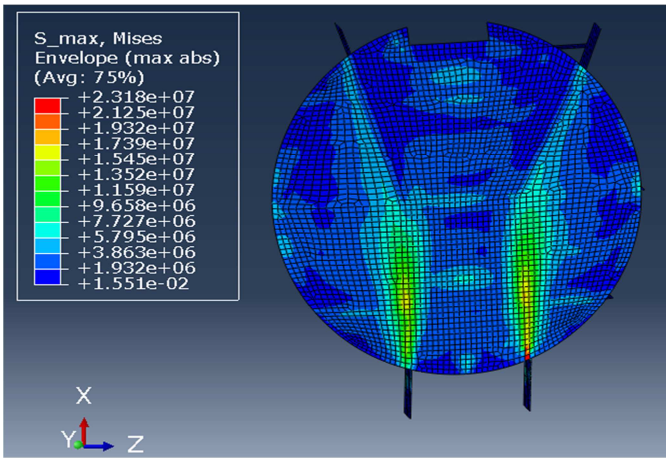

Figure 11: Maximum stress (Pa) on the reflector

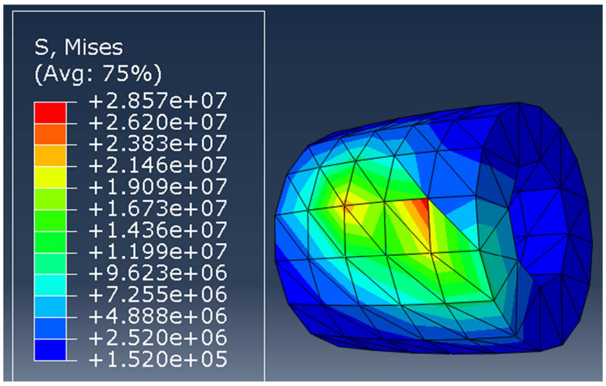

Figure 12(a): Maximum stress (Pa) at the roller

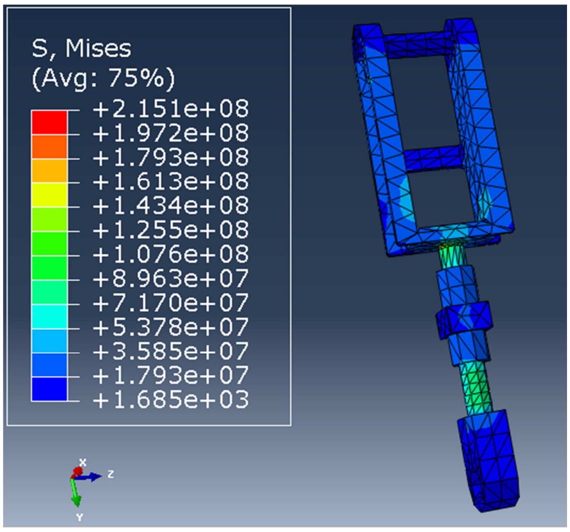

Figure 12(b): Maximum stress (Pa) at the locking linkage

A. *Contact Results*

The modelling of contact is useful in getting various contact parameters such as contact forces, normal pressure due to contact, contact status and contact area. At the instant of latching the roller rolls into the groove of the outboard bracket and consistently keeps on changing its contact position in the groove due to vibrations in the structure as can be seen in Figure-13.

The beginning of contact as the roller moves into the grove is shown at time 9.402 sec and the end of contact for the final position is shown at time 9.902 sec.

Maximum contact pressure is observed during latching of the roller into the groove of the outboard bracket with maximum pressure of 8.925×10^7 N/m^2 as shown in Figure-14 and is less than the yield stress of Aluminum.

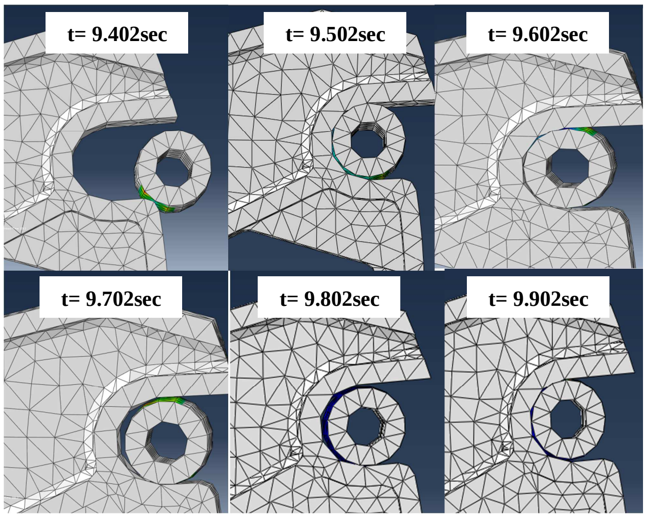

Figure-13: The oscillating contact of the roller after latching.

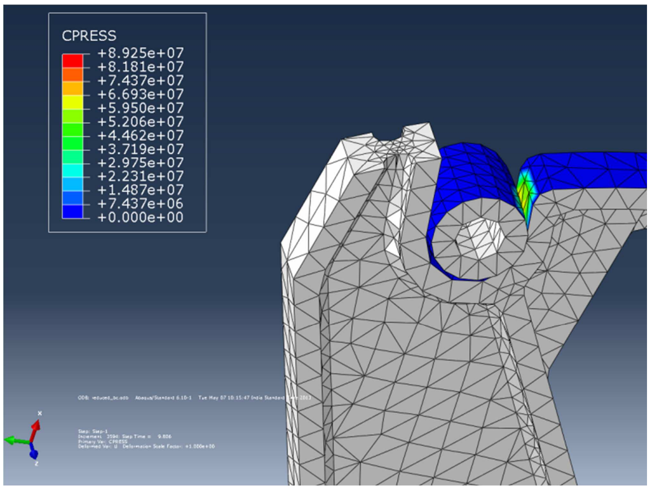

Figure-14: Maximum contact pressure between the roller and the groove of the outboard bracket

The contact forces due to the normal pressure on the roller are shown in Figure-15. Maximum contact force of 1.25×10^3 N is observed in y-axis. The other components are 0.4×10^3 N in x-axis and 50 N in z-axis.

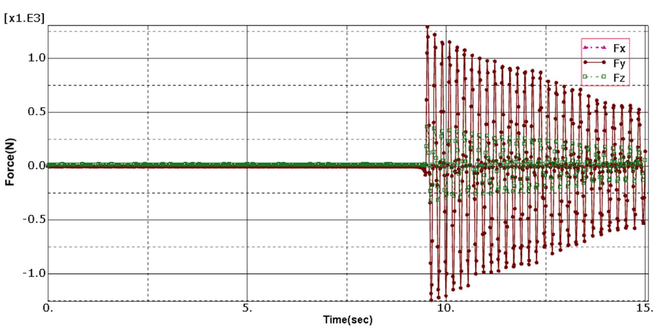

Figure 15: Contact forces on the roller surface due to normal stress

V. CONCLUSIONS

The deployment dynamics and latching of reflector has been simulated in ABAQUS finite element software. The antenna along with hinge assembly was assumed flexible during deployment and latching. The latching phenomenon was simulated through contact modelling of ABAQUS. The latch-up loads are evaluated and the stresses in various components of the reflector assembly are much less than the yield stress of the material. It is observed that the peak deflection of antenna after latching is very small. The deployment time predicted matches very well with measured orbit data. The results of the analysis are also validated by conservation of energy principle. A close match with the earlier methodology has been observed where the deployment dynamics and the latch up shock estimation are done in two separate steps using ADAMS and NASTRAN software. The study was useful in providing the design inputs for the mechanism.

ACNOWLEDGEMENT

The author would like to thank Deputy Director, MSA and Director, ISAC for their support and encouragement. The authors wish to thank all SMG colleagues for providing guidance and support.

REFERENCES

[1] B. S. Nataraju, R. Chinnasamy, T. S. Krishnamurthy, and D. H. Bonde, "Modelling of Deployment Mechanisms for Latchup Shock", ESA Journal Vol 13, Pp393-400, 1989. J. Clerk Maxwell,

[2] Sanjay Jaiswal, H. N. Suresha kumar, N. Viswanatha, and B. S. Nataraju "Latchup shock analysis of reflector deployment considering flexibility of mechanism" 5th Aerospace and Related Mechanisms Symposium, Bangalore, pp. 105-110, 2005

[3] B. Lakshmi Narayana, H. S. Vijayalakshmi, B. P. Nagaraj, G. Nagesh and C. D. Sridhara "Deployment dynamics and latch up modelling of large antenna" 7th Aerospace and Related Mechanisms Symposium, VSSC, Thiruvanathapuram, 2010.

[4] B. Lakshmi Narayana, B. P. Nagaraj, G. Nagesh and C. D. Sridhara "Deployment Dynamics and Latch-Up Shock Estimation of Large Antenna Hold Down Mechanism" 15th National Conference on Machines and Mechanisms, IIT Madras, pp. 240-247, 2011.

[5] ADAMS User's Guide (Version 9.0), MSC. Software Corporation, USA

[6] MSC/NASTRAN Reference Manual, 2005, MSC. Software Corporation, USA.

[7] ABAQUS 6.10 Documentation, 2010, Dassault Systems Simulia Corporation, Providence, RI, USA