1. ABSTRACT

The solar array deployment system for the LMX line of buses deploys rigid Solar Array Wing Assemblies (SAWAs). Each SAWA has a set of Solar Array Deployment Mechanisms (SADM), which consists of two hinges, a strut, and two Hold Down Release Mechanisms (HDRMs) (Fig. 1).

Fig. 1. LMX Bus Solar Array Deployment

To qualify the SADM for flight, each mechanism component was qualified individually, then assembled to a qualification SAWA on Special Test Equipment (STE) and deployed in a thermal vacuum chamber at ambient, hot, and cold temperatures. These mechanisms were designed, built, and tested by the Power and Mechanisms part of the Power, Thermal, Structures & Mechanisms Product Center, which develops products for both internal and external customers. This paper will discuss the qualification effort for the LMX Solar Array deployment, including qualification hardware and STE. It will focus on unique challenges presented by each aspect of the qualification, and lessons learned from the hardware integration and the qualification testing.

2. SADM QUALIFICATION

The SADM is qualified to temperatures ranging from -59°C to +67°C. Once the electrical release command is sent, the paraffin-driven HDRMs release each SAWA within 4 minutes and 30 seconds. After both HDRMs release, the SAWA deploys out to 90 degrees in approximately 4 seconds in vacuum, producing less than 5204 N (1170 lb) of lock-up load through the strut into the spacecraft at the end of travel. The SADM qualification test flow is shown in Fig. 2.

Fig. 2. SADM Qualification Test Flow

2.1 HINGES

The primary source of deployment torque is a set of negator-spring-driven hinges. A matched set of one “left” hinge and one “right” hinge comprises a hinge
line (Fig. 3 and 4). Each hinge line is tested together to best simulate operational conditions.

![Image](image1.png)

**Fig. 3. Stowed Left Hinge**

![Image](image2.png)

**Fig. 4. Deployed Right Hinge With Damper**

The solar array hinge qualification test program began with a hinge torque test prior to the installation of the damper to verify the health of the negator springs and friction surfaces. After damper installation, ambient, hot, cold, and post-thermal ambient deployment testing in a dry environment established a functional performance baseline for the hinge set. The hinge set was then subjected to qualification random vibration levels and durations, and a post-vibration ambient deployment test verified in-family performance compared to the baseline. Lastly, the hinge set was repeatedly deployed during the hinge life test, such that the total number of deployments performed during qualification testing is 50 times, twice the operational life of 25 cycles.

2.2 **STRUT**

The SADM strut is the primary source of kick-off energy, a secondary source of deployment torque, and the only end-of-deployment stop. The lenticular strut resembles a large tape measure with a rigid bracket at one end. It interfaces to the spacecraft bus and the SAWA through spherical joints.

In its stowed configuration (Fig. 5), the strut is rigid with a bend in one location. A kickoff spring and two strut guides rigidly mounted to the solar array panel keep the strut aligned and pre-loaded through the pre-deployment launch loads. When fully deployed, the strut is straight and rigid. The bend in the lenticular strut is not controlled or predictable during the course of its deployment.

![Image](image3.png)

**Fig. 5. Stowed Solar Array Strut Assembly**

The strut, strut guides, and mounting bracketry weigh approximately 0.9 kg (2 lb). An initial torque of 277.9 N-m (2460 in-lb) is provided by the kickoff spring. The lenticular strut provides an average deployment torque of 0.51 N-m (4.5 in-lb), and an end of travel torque of 15.3 N-m (135 in-lb).

Strut qualification testing began with a stowed geometry check to size pre-load shims and verify back-buckling force. The strut torque test then characterized
the deployment torque contribution of the strut, and finally, the strut life test exercised the strut up to 50 times, twice the operational life of 25 cycles.

2.3 HDRMs

Two HDRMs are required to hold each SAWA in the stowed position. They each provide 3670 N (825 lb) of pre-load. At worst-case cold temperatures and voltages, each HDRM will release within 4 minutes and 30 seconds.

The HDRM is comprised of a latch body, which mounts to the spacecraft bus, and a panel bracket, which mounts to the solar array panel. The latch body has two paraffin-actuated cams, which trap part of the panel bracket. The panel bracket is released when either cam is actuated.

The HDRM was qualified on a previous program.

3. HINGE TEST CONFIGURATION

After successful completion of hinge (with damper installed) deployment tests at ambient, hot, and cold temperatures, the hinge assembly was installed onto the hinge vibration test fixture. In flight configuration, the inboard side of the solar array hinge is attached to the spacecraft and the outboard side is attached to the solar array, which flexes when subjected to acoustic launch loads. To simulate the flight configuration and loading of the hinge assembly, the inboard hinge clevises were rigidly constrained, and the outboard hinge clevises were mounted to a rigid plate that was allowed to float the amount the solar array is predicted to flex (Fig. 6). Accelerometers were mounted in 11 locations to control the input spectrum and record the responses.

The accelerometer mounted to the end of the eddy current damper showed a peak of 157 g²/Hz at 390 Hz. Qualification vibration testing on the damper alone produced a peak of 179 g²/Hz at 1560 Hz. Comparing the two data points, the peak from the hinge assembly testing was similar in magnitude to the peak of the damper testing, but the frequency was shifted lower by 1170 Hz.

Post-vibration deployment testing at ambient temperatures showed an increase in deployment time of 26% (4.0 sec) from pre-vibration to post-vibration. During a subsequent ambient deployment performed for troubleshooting, the difference in pre-vibe to post-vibe deployment time increased to 35% (5.4 sec).

The loads on the hinge eddy current damper during hinge qualification vibration testing exceeded the preload of a bearing internal to the damper. The damper was mounted to four posts on the hinge, which are more compliant than the rigid interface used for the damper qualification vibration test. This difference in interface stiffness caused the predicted peak load to shift into a natural frequency of the damper, overcoming the pre-load of the gear train.

The damper failure during hinge vibration testing led to a re-design of the hinge, this time without a damper. This change in design required the restart of the hinge qualification program.

The lesson learned is to fully understand how components are qualified and how that data should be used at the next higher assembly. A more complete analysis of the hinge and damper would have predicted the appropriate vibration test levels for a rigid interface test that are equivalent to the specified vibration levels for the more compliant interface.

An alternative to specifying different vibration test levels for the damper component would be to provide a mass and geometry simulator of the relevant hinge parts to the damper vendor for testing. This would make the damper test environment more representative of the flight environment. On the down side, the damper test fixturing would increase in cost, and the damper qualification testing would be dependent on the completion of the hinge design, which would affect the damper qualification schedule.

4. HINGE SINGLE POINT FAILURES

To increase reliability, single point failures were minimized in the design. In the original hinge design, which had flown on two previous spacecraft, the left hinge rotated about a pin, which rotated within a self-aligning bearing. However, the right hinge lacked
redundant rotating surfaces, so the inboard hinge clevis was re-worked to allow for slip-fit bushings in place of press-fit bushings.

The LMX solar array deployment system was based on a successful commercial satellite that flew 10 years ago. The LMX design tried to minimize risk by keeping as much of the heritage design as possible, while meeting new system requirements. The hinge geometry for the hinge set and spherical bearing in the left hinge were derived from this original design. A more recent hinge, also with flight history, provided the heritage for a wobble shaft that allowed two degrees of rotational freedom and supplied a damper interface for the right hinge.

During a methodical review of single point failures, the program using the more recent hinge discovered that the hinge half with the wobble shaft and damper interface lacked redundant rotating surfaces. Since the right half of the LMX solar array hinge set was modeled after that particular hinge, a re-design of the LMX right hinge was also required.

This re-design occurred near the end of the hinge qualification program. This minor design change in the inboard hinge clevis required repeating the entire hinge qualification program. Fortunately, this re-design coincided with the re-design of the hinge to remove the damper, which also necessitated the restart of the hinge qualification program.

The lesson learned is to understand the features and design philosophy of a heritage design, along with the requirements, when using it as a model for a new design. In the risk-averse aerospace industry, when a new product is designed, utilizing as much of a heritage design as possible minimizes risk. Many designs are “just like” a heritage design with a long list of exceptions that follow. The heritage hinge design should have been assessed more carefully and this minor change should have been incorporated in the original design of the LMX hinges.

A second lesson learned is to keep abreast of other programs and findings in the industry. Lessons are constantly being learned, and being aware of other people’s lessons may prevent you from having to learn a costly lesson yourself somewhere down the line.

5. MARGIN

A power harness that crosses the hinge joint, delivering power from the solar array power to bus, is the primary source of resistive torque accounted for in the torque margin calculation. For torque margin calculations, we initially used a conservative estimate of harness resistive torque of 3.68 N-m (32.6 in-lb), which was based on test data from a previous vehicle. Taking into account friction and other vehicle dynamic effects, the hinges were designed to provide 15.987 N-m (141.50 in-lb) of torque per hinge line to ensure a 2:1 (100%) torque margin. This high level of deployment torque necessitated the use of a damper to mitigate end-of-deployment lock-up loads. With the damper failure, a refined harness torque test was performed, resulting in 0.68 N-m (6.0 in-lb) of measured resistive torque at cold temperature. The final hinge configuration of 4.796 N-m (42.45 in-lb) of torque per hinge line lowered the lock-up load to the point where we could eliminate the damper from the hinge design.

The initial conservative prediction of 3.68 N-m (32.6 in-lb) for harness torque was calculated by scaling up harness test data on a previous vehicle. The tested harness was a 17-wire (20 AWG) bundle with copper wrap and Kapton® tape that was tested to –62 °C, and had a measured resistive torque of 1.1 N-m (9.5 in-lb). The LMX harness is a 38-wire (20 AWG) bundle with a silicone sleeve to protect against atomic oxygen, and a predicted low temperature of –105 °C.

Preliminary testing of the LMX harness at -105°C resulted in approximately 2.3 N-m (20 in-lb) of resistive torque. This value was suspect due to a large thermal gradient in the thermal chamber so the original scaled value of 3.68 N-m (32.6 in-lb) was used for the torque margin calculation. A configuration error in the thermal model was corrected and the predicted harness low temperature was updated. During the hinge redesign effort, harness torque testing was performed to an updated thermal prediction of -86°C. Fans were used in the thermal chamber to minimize thermal gradient. This harness torque test produced a measured torque of 0.68 N-m (6.0 in-lb), which is 18% of the original predict.

To ensure a 2:1 torque margin with the original 3.68 N-m (32.6 in-lb) harness resistive torque prediction, 1.0 N-m (9 in-lb) damper resistive torque, 20% allowable hinge friction, and 85% of nominal negator spring torque, a hinge torque of 15.6 N-m (138 in-lb) was required. Each hinge negator spring leaf provides 1.64 N-m (14.5 in-lb) of deployment torque, so the hinge torque can only be adjusted in 1.599 N-m (14.15 in-lb) increments. This resulted in the original hinge torque of 15.987 N-m (141.50 in-lb) per hinge line.

With a maximum harness resistive torque of 0.68 N-m (6.0 in-lb), no damper resistive torque, 20% allowable hinge friction, and 85% of nominal negator spring torque, only 2.15 N-m (19.0 in-lb) of deployment torque was required for the hinge line. To facilitate hinge testing so that the hinge deployment torque
dominated any potential STE effects, the final design of the hinge used two leaves on the left hinge and one leaf on the right hinge for a total of 4.796 N-m (42.45 in-lb) of torque for the entire hinge line.

The lesson learned is to look at the big picture. Fully understand the relationships between different aspects of a design and look forward to assess the ramifications of a decision. Conservatism, by definition, errs on the side of more margin, which is less risky. However, when all requirements are considered, what is conservative for one may not be for another, and could potentially cause a higher risk situation.

While harness resistive torque is a large contributor to deployment torque margin, its contributions to mitigating lock-up load are very small. Using a high resistive harness torque is conservative for a torque margin calculation, which decreases risk, but drove the hinges to a high spring torque. A high spring torque drove the lock-up load to exceed acceptable values, which drove the addition of a damper to the hinge design. Including the damper in the hinge design drove hinge-level deployment testing, which resulted in schedule, financial, and emotional cost when the damper was damaged during hinge vibration testing. Looking at conservatism from a torque margin standpoint alone resulted in increased risk rather than risk mitigation.

Investing time and money into a quality test setup and refined analysis is worthwhile. Had the preliminary harness torque test been performed with a smaller thermal gradient and an accurate thermal model configuration, the hinge design would have arrived at the final damper-free configuration much earlier in the program.

6. LENTICULAR STRUT TESTING

Designing simple, low-cost, short-lead-time STE for torque testing a lenticular strut presented a challenge. The spherical joints at each end of the strut in conjunction with the unpredictable bend location made it particularly difficult. The average strut torque excluding kick-off is 1.13 N-m (10.0 in-lb). Complicated systems would introduce effects that could mask the true strut performance. A shepherd’s hook, was designed, which offloaded the strut at its approximate center of gravity. It was manually moved to follow that spot throughout the strut deployment. The offload cord was centered through a 0.254-cm (0.100 inch) diameter gauge hole, which minimized misalignment and STE torque effects.

A torque test had been performed during strut development testing. The strut STE was primarily composed of 3.9 cm x 7.6 cm (1.5 in x 3.0 in) and 3.9 cm x 3.9 cm (1.5 in x 1.5 in) 80/20 aluminum slotted extrusion and was mounted to a level optical bench (Fig. 7). The strut itself was offloaded by a technician holding a hard flat surface beneath the strut. A nut hanging from the deployable arm of the STE traveled over pieces of tape, and indicated the current deployment angle.

![Fig. 7. Strut Installed in Torque Test STE](image)

The qualification strut torque test planned to use the same 80/20 slotted extrusion STE and the same test procedure. During the strut torque test readiness review, the mechanisms analyst levied new requirements of a maximum STE contribution of 10% of the average deployment torque, torque measurement accuracy to 1% of the average deployment torque, angular positioning of the strut accuracy to 0.25 degree, and measurement of the kick-off spring. The new torque requirements provided the required level of accuracy for the strut lock-up load calculations and ensured that the strut deployment torque was not masked by STE effects. The angular measurements at every 0.25 degree were required to characterize the strut torque at the beginning of deployment and at the end of deployment, when the strut approaches lock-up, because the strut torque changes rapidly with angle.

80/20 slotted extrusions are known as the “industrial erector set”. While it is extremely versatile and many
different test setups have successfully utilized it, 80/20 is not designed for accuracy. The 80/20 hinges used in the strut torque test STE were variable friction hinges, adjustable with the turn of a screw.

To achieve a more accurate test setup, a number of modifications were made. The STE deployable arm was weighted with the bus mounting bracket to achieve the appropriate friction characteristics in the hinges during deployment, and the torque in the stowing and deploying directions was measured. This data generated a hysteresis curve indicating the overall STE torque contribution and friction throughout the deployment.

To accurately determine the deployment angle, a high-accuracy potentiometer was mounted in line with the deployable arm hinge. To calibrate the potentiometer, the deployable arm was once again weighted to simulate the test condition. Then an optical cube was precisely mounted to the deployable arm of the STE such that an autocollimator would register a cube face at fully stowed and register the cube’s orthogonal face at 90 degrees deployed. The potentiometer voltage at fully stowed and at exactly 90 degrees generated the calibration curve that determined the angular position to 0.25 degree.

To avoid stowing or deploying torque contribution to the strut torque test, a shepherd’s hook was designed. The shepherd’s hook stood on the optical bench beneath the strut, and reached over the strut to offload it at the approximate stowed center of gravity (Fig. 8). The offload cord was secured to the strut with flashbreaker tape. The offload cord was supported 1.111 m (43.75 in) above a 0.254-cm (0.100 in) diameter gauge hole 1.9 cm (0.75 in) above the strut. This aligned the strut offload cord to the gravity vector, and ensured a STE offload deploying or stowing torque contribution of less than 0.02 N-m (0.20 in-lb). During deployment and at each torque measurement angle, the shepherd’s hook was manually positioned over the strut such that the offload cord was centered in the gauge hole.

Despite being constructed out of 80/20, accuracy was achieved by carefully leveling the extrusions at each step and checking the level on the optical bench after each torquing operation.

The strut torque test pass/fail requirements had only been addressed from a torque margin standpoint in the development test. Since the strut is not a large deployment torque contributor like the hinges, a “conservative” low value of 0.3 N-m (3 in-lb) (from a previous program) had been used for the torque margin calculation. With the updated harness torque information and the hinge redesign to eliminate the damper, lock-up load became the limiting factor for the strut so a maximum average strut deployment torque of 1.6 N-m (14 in-lb) was established. The LMX struts produce twice as much torque as the heritage struts, despite similar design and manufacturing.

The lesson learned is to talk to your analyst early so he understands and agrees with the accuracy of your test and so you understand the requirements. When performing analysis, know which direction conservatism alters the parameters. Also select STE components with test accuracy in mind.

7. INTEGRATION

The thermal vacuum test that integrated all of the qualified SADM components required a test fixture that held the hinge line vertical in order to minimize the STE influence on the deployment test dynamics. A deployable boom equipped with a linear bearing offloaded the panel at the overall center of gravity of the panel and the deployable parts of the mechanisms. The STE and qualification hardware were instrumented with a load cell to determine the actual offloaded weight throughout deployment, inclinometers for STE
alignment changes during testing, and a strain gauge to measure lock-up load.

To minimize human error, a conscious effort was made to avoid manually maneuvering the SAWA during the integration of the panel and mechanisms to the STE. As a result of using the solar array handling fixture in a non-standard manner, two HDRM mounting bolts (one per HDRM) sheared off into the SAWA inserts. Stress calculations showed that the remaining three screws per HDRM were sufficient to continue with the test, and the screws were removed post-testing.

In the thermal vacuum testing configuration, the SADM STE was mounted to large I-beams (Fig. 9) thermally isolated from the universal test fixture (UTF), a large truss structure that rolls into the thermal vacuum chamber on specially designed rails. The top of the UTF is approximately 1.5 m (5.0 ft) high off the ground.

To integrate the SAWA to the STE on the top of the UTF, we hoisted the SAWA in its handling frame (panel perpendicular to ground) onto the UTF, used technicians to support and stabilize the SAWA, and tried to manually remove the SAWA from its handling fixture. This appeared to be the safest way of installing the SAWA onto the STE, minimizing risk to the hardware by minimizing human interaction.

During removal of the SAWA from the handling fixture, some of the captive fasteners had a higher running torque than expected, so we re-torqued all the fasteners and hoisted the SAWA and handling fixture back onto the ground. Despite an experienced technician carefully and slowly unthreading the captive fasteners from the SAWA by hand, two of the screws snapped off into the panel, one in each HDRM location. In handling the SAWA with the panel perpendicular to the ground, the handling frame captive fasteners were loaded in shear. The stainless steel screws galled in the stainless steel inserts. The shop had always handled the panel in a horizontal configuration, never loading the fasteners in shear, so galling had never been encountered.

Stress analysis showed that three screws per HDRM bracket provided sufficient clamping force, and that it was acceptable to continue on with the thermal vacuum testing.

In the end, we successfully integrated by manually removing the SAWA from its handling frame per standard shop practices, and handing the SAWA up to personnel staged at the edge and on top of the UTF.

The lesson learned is to be aware of current manufacturing practices and know why those have been established as standard practices. There’s a fine line between mitigating risk and introducing more risk. By trying to minimize the risk of human error, we introduced more risk by handling the SAWA in a way that differed from standard shop practices.
8. THERMAL VACUUM TESTING

During the thermal vacuum test, we encountered difficulties achieving our target temperatures due to limitations in the test setup. The thermal vacuum deployment STE was modified from a previous program and not designed with our thermal environment in mind. In an effort to bring the qualification hardware to their target temperatures, thermal distortion of the aluminum frame STE induced moments into the HDRMs that exceeded their qualification loads.

The deployment STE was equipped with heater tape for temperature control in three different zones. The thermal plan to achieve SADM target cold temperatures held the deployment STE at one homogenous temperature, and used heat lamps to fine tune the temperatures of the mechanisms. There was a large target thermal gradient (18.5 °C) between the hinges and the HDRMs, which proved to be unachievable with the entire fixture held at the same temperature. The thermal engineers proposed controlling the deployment STE temperature zones independently to different temperatures.

When consulted about the proposed thermal control change, the stress analysts were concerned about the amount of thermal stress that could potentially be induced into the mechanisms by the fixture. The coarse stress model of the deployment STE predicted loads with all zones of the deployment STE held to the same temperature. To assess the modified thermal plan, the model resolution was increased and the model was adjusted to accommodate different temperatures in each deployment STE zones. When the original thermal load case was run in the updated model, it was discovered that the HDRMs were in a slightly overstressed condition. Different thermal cases were subsequently analyzed to find one that achieved the target temperatures without overstressing the hardware.

The lesson learned is to try to match the coefficient of thermal expansion of the deployment STE to the spacecraft for thermal vacuum testing. The SADM STE is an aluminum fixture that was used for the solar array deployment of the heritage commercial spacecraft that the LMX SAWA was based on. The STE was modified to fit the scaled-up geometry of the LMX SAWA. In addition to having a larger footprint, the LMX target temperatures were more extreme, which contributed to the larger thermal gradients that overstressed the HDRM.

To accurately measure the thermal distortion induced inclination changes in the deployment STE throughout the thermal vacuum test, commercial off-the-shelf single-axis digital inclinometers were to be mounted to the main post of the STE. During inclinometer integration, the inclinometer manufacturer documentation was discovered to be incorrect. The hardware and documentation disagreed on the number of connections and the color coding of the leads. After a bit of trial and error and lost schedule, only one of the original three inclinometers was fully functional. To supplement that one axis, a manual clinometer was used to measure the angle in four different locations on the STE every time we repressurized and restowed. This verified that the SADM deployment STE did not permanently shift during the thermal vacuum test.

The lesson learned is that commercial off-the-shelf products can be very handy and save on development time, but sufficient lead time and personnel effort to understand the products need to be budgeted so they are ready to use before a test. The digital inclinometers seemed like they’d be simple to use. Insufficient time was devoted to understanding how they worked before integration onto the deployment STE, which resulted in schedule slip and only one functional inclinometer.

Thermocouples and calorimeters were used to measure temperatures during the thermal vacuum test. To achieve target temperatures, an array of heat lamps was positioned directly in front of the harness, which runs between the left and right hinge. The heat lamp output required to bring the harness up to temperature was acceptable for the harness calorimeter, but exceeded the melting point of the calorimeter adhesive, causing the calorimeter to fall into the right hinge, which caused us to abort the first hot test.

The lesson learned is to be familiar with all aspects of the test setup. Verify that achieving one test objective doesn’t negatively affect a different part.

9. CONCLUSION

The SADM design is redundant, robust, and simple. The HDRM, hinges, and struts are all one failure tolerant. The modular design is easily adaptable to a number of different performance requirements: the power of the solar array varies with the size of the panel or the type of solar cell, the final deployment angle varies with the length of the strut, and the rate of deployment and lock-up load vary with the number of negator leaves installed onto the hinges, and the addition of a damper.

Despite the qualification hardware, STE, integration, and testing challenges encountered throughout the solar array deployment qualification process, the LMX solar array deployment qualification was successful and all flight unit acceptance testing is complete.