Lessons Learned from the TIRS Instrument Mechanisms Development


Abstract

This paper describes the many lessons learned during the design and development of several mechanisms for the Landsat Data Continuity Mission (LDCM) Thermal Infrared Sensor (TIRS) instrument, built by an engineering team at NASA Goddard Space Flight Center (GSFC). Several mechanisms were developed for TIRS including an arc-second precise mirror positioning system, a launch lock for a 90-lbm (41-kg) cryo-cooler assembly, and a large deployable earth shield. These mechanisms were developed over a 2 year period, and several obstacles were encountered and subsequently solved prior to delivery.

Introduction

The LDCM satellite will launch into a low polar orbit in late 2012. LDCM will provide earth resources data continuity between the currently operational Landsat 5 and Landsat 7 missions and the Joint Polar Satellite System (JPSS) Missions. It will provide a high spatial resolution complement to the lower spatial resolution, higher temporal sampling JPSS data set. LDCM will be carrying TIRS, an actively cooled, nadir-looking, mid-infrared imager. TIRS on LDCM is a 100-meter spatial resolution push-broom imager whose two spectral channels, centered near 10.8 and 12 microns, split the spectral range of the Thematic Mapper (TM) and Enhanced Thematic Mapper (ETM+) instruments.

Figure 1. LDCM will launch in late 2012 carrying the TIRS Instrument. The LDCM spacecraft is shown in orbit in the left figure. The TIRS instrument is shown on the right with the large white earth shield panel deployed.

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The TIRS instrument has 3 mechanisms and a set of mechanism control electronics, all built at GSFC. All of the mechanisms were developed on very compressed schedules, resulting in increased development risks. Each mechanism had unique development problems which were successfully overcome.

**Scene Select Mechanism (SSM)**
TIRS requires multi-scene calibration every orbit, so a flat scene mirror is used to switch the instrument field of view between nadir, cold space, and a warm black body calibration target. The Scene Select Mechanism will rotate and hold the scene mirror in position within ± 9.7 µradians using closed-loop digital control. The location of the SSM within the TIRS instrument is shown in Figure 2.

![Figure 2. The SSM is located in the heart of the TIRS instrument, just above the cryogenic telescope. The quarter section view on the left shows the location of the SSM within the TIRS instrument structure. The figure on the right shows the SSM above the telescope. Baffles and secondary structures have been removed for clarity. Note the close proximity of the edge of Scene Mirror to the cold telescope, which radiatively drives the mirror temperature down.](image-url)
Earth Shield Deployment Mechanism (ESDM)
As TIRS is a cryogenic instrument, it must reject a large amount of heat to keep the focal plane array at the operational temperature of 43K. In order to keep the radiator to a reasonable size, a 2.5-m² deployable earth shield panel is used to block albedo. This earth shield is stowed at launch, and is rotated & locked into position by the ESDM. The ESDM is shown in Figure 3.

Cryo-Cooler Launch Lock (CCLL)
TIRS is actively cooled by a Stirling cycle cryo-cooler mounted beneath the instrument. This cooler is a source of jitter and is separated from the instrument by a passive vibration isolation system consisting of damping flexures. The flexures are too soft to survive the launch environment without a launch lock. The CCLL mechanism constrains the cryo-cooler supporting structure during launch and is released on orbit prior to instrument operations.

Scene Select Mechanism
The SSM is a single axis, precision mirror positioning mechanism, capable of 3 µradian stability. It can be driven in either direction for unlimited rotations. The rotating mirror is dynamically balanced over the spin axis, and does not require launch locking. Several configurations were traded before the SSM flight architecture was finalized. The mechanism is shown in Figure 5. The SSM met or exceeded the driving requirements defined in Table 1.
Table 1. SSM Driving Requirements

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>33 lbm (15 kg)</td>
</tr>
<tr>
<td>Power</td>
<td>6 W average</td>
</tr>
<tr>
<td>Mirror Size</td>
<td>8.15 x 5.87 in (207 x 149 mm) elliptical</td>
</tr>
<tr>
<td>Knowledge</td>
<td>± 9.7 µradians over 34 minutes</td>
</tr>
<tr>
<td>Stability</td>
<td>± 9.7 µradians over 2.5 seconds</td>
</tr>
<tr>
<td>Duty Cycle</td>
<td>100%</td>
</tr>
<tr>
<td>Thermal Operational</td>
<td>0 / +20°C stable to ±1°C</td>
</tr>
<tr>
<td>Thermal Survival</td>
<td>-50° / +40°C</td>
</tr>
<tr>
<td>Lifetime</td>
<td>3.25 years on orbit</td>
</tr>
<tr>
<td>Redundancy</td>
<td>A/B side block redundancy</td>
</tr>
<tr>
<td>Operational Cadence</td>
<td>Stare nadir for 30-40 minutes</td>
</tr>
<tr>
<td></td>
<td>Rotate 120° in &lt;2 minutes to space view</td>
</tr>
<tr>
<td></td>
<td>Stare for ~30 seconds,</td>
</tr>
<tr>
<td></td>
<td>Rotate 120° in &lt;2 minutes to blackbody view</td>
</tr>
<tr>
<td></td>
<td>Stare for ~30 seconds</td>
</tr>
<tr>
<td></td>
<td>Rotate to 120° in &lt;2 minutes to nadir view</td>
</tr>
</tbody>
</table>

Structure
The SSM will operate at a nominal 7-10°C; the scene mirror it rotates will be near 0°C. To minimize heater power, low thermal conductivity titanium 6Al4V was chosen for the primary structural material. The 8.15 x 5.87 in (207 x 149 mm) scene mirror is made of an aluminum 6061-T6511 extrusion with a gold optical surface coating; the back is bare. The mirror is secured to a titanium mount with three aluminum 7075-T6 flexures to minimize thermal deflection of the mirror surface. The flexures are pinned to the mount then float bonded in place with Stycast 2850 epoxy to the mirror to reduce assembly-induced moment loading of the mirror. The mirror mount is bolted and liquid pinned to the SSM rotor shaft. The shaft is suspended in the bearing housing with a pair of angular contact ball bearings. The bearing pair is mounted within the bearing housing, which serves as the main housing of the SSM. Three radial legs extend from the housing and provide the mounting interfaces to the instrument. The encoder code disk
and motor rotor are bolted and liquid pinned to the shaft. The motor stator mount houses the stator, and attaches to the non-mirror end of the SSM. The motor stator mount is a separate part from the bearing housing so that it can be shipped to the motor vendor for stator build up independent of the rest of the SSM. Similarly, the mirror mount is a separate part from the shaft to allow concurrent, independent assembly. The motor mount, mirror mount, and rotor shaft are finished in Tiodize type II for corrosion protection. The bearing housing was not Tiodized; it was left bare since it is almost entirely covered with strip heaters and other thermal control components which require non-anodized surfaces to minimize the generation of thermal hot spots.

Several features were introduced to facilitate handling, reduce risk, and ease testing. Lift points and shipping interface points were independent of the flight interface points. This protected the precision flight interface from possible damage and wear from the numerous shipping and handling operations prior to installation in the instrument. A quartz optical cube was bonded to the motor housing to provide a boresight reference, facilitating SSM alignment and shimming into the TIRS instrument. Mirror pointing and stability measurements were referenced to this fiducial.
Baffles
A fixed aluminum baffle is fitted over the mirror end of the SSM, attached to the bearing housing. An aluminum rotating baffle, attached to the shaft, protects the back of the scene mirror and the mirror mount from the cold space view and hot blackbody view thermal loads, and reduces stray light introduced into the telescope.

Actuator
The SSM is driven by a frameless, brushless, direct-current “zero cog” motor with redundant windings and damping coils. The 2-phase, 48-pole, moving magnet motor has an air gap of 0.4 mm, and a motor constant of 1.79 N-m/amp (253 in-oz/amp).

Bearings
The SSM uses a duplex pair of back-to-back mounted, 70-mm bore angular contact ball bearings with a 25-degree contact angle. The bearings are bonded in place to ensure stability at the µradian level. They are hard preloaded to 50 lbf (222 N), with the preload calculated to drop to 25 lbf (111 N) at the operational cold temperature. The bearings use tri-cresyl phosphate coated 440C balls in a phenolic cage, with 440C races. Pennzane lubricant is utilized; specifically an oil + grease slurry (50% each by weight) of Nye 2001 ultra-filtered synthetic oil and Nye Rheolube 2000 grease. The oil is also vacuum-impregnated into the retainer. Lubricant retention is provided by labyrinth seals sized for < 5% mass loss over the mechanism lifetime. Lubricant surface migration is prevented by the application of Nye-bar surface barrier coatings within the labyrinth seals. Since twice the estimated bearing lifetime (including margin) was only 100,000 cycles, a full flight fidelity life test was not performed. Instead, a life test using only a pair of bearing cartridges, with identical preload and thermal conditions, was executed successfully.

Figure 6. Optical encoder is shown in this section view mounted on the SSM with the motor removed.

Encoder
The SSM uses a 22-bit, pseudo-absolute optical encoder, which is an incremental unit that emulates an absolute encoder with a software counter and index pulses. The encoder has redundant read heads and drive electronics cards shown in Figure 7. The 125-mm code disk is fixed with liquid pinning to the rotating shaft. The read heads are fixed to the bearing housing. The encoder is shown in Figure 6. The
motor stator housing fits over the top of the read heads, encapsulating them; an integral contamination
shield protects the encoder from any potential particulates coming from the motor area. The encoder
drive electronics are housed in a dedicated enclosure, which is mounted to the cover of the motor stator
housing. The encoder design has some flight heritage as it is based on the pseudo-absolute architecture
from the National Polar-orbiting Operational Environmental Satellite System (NPOESS) Preparatory
Project (NPP) satellite Cross-track Infrared Sounder (CrIS) instrument.

![Figure 7. The optical encoder is shown mounted on the SSM with the motor removed.](image)

**Scene Select Mechanism Development Issues**

Conceptually, the SSM is a very straightforward mechanism: it is single-axis, direct-drive, without a
launch lock. However, a challenging stability requirement, a complex thermal gradient, a custom digital
servo-controller, and most of all a compressed schedule complicated the development. Many obstacles
were encountered and overcome; the most significant technical issues are described below. Schedule
and management issues also occurred but have been described previously.

The importance of functional breadboards

At the beginning of the development effort, the control engineers required hardware that could be used to
develop the arc-second-level servo controller be available much earlier than we could expect to receive
the actual motors and encoders. The hardware would need to have at least a 22-bit encoder and a similar
motor to be useful to the development.

A commercial 24-bit absolute encoder was procured from the same vendor that was to deliver the flight
unit. These units were available on a short delivery. The digital output from this unit was identical to that
which would be delivered for flight; i.e., from a telemetry standpoint, it was electrically identical. A spare 2-
phase, frameless brushless DC motor that was similar in performance to the flight motor was fortuitously
available. A commercially available back-to-back duplex (DB) bearing pair equivalent to the flight design
was quickly procured.
A simple aluminum bearing housing and spindle were fabricated, and the bearings installed with a spring preload. The frameless motor was installed at one end of the shaft, and the encoder code disk and read head installed and aligned at the other end. While this configuration did not exactly mimic the SSM (which has the motor and encoder on the same end of the shaft), this was adequate for initial controls development.

Figure 8. The SSM functional breadboard used a leftover frameless motor from another program mounted with commercial bearings and an industrial 24-bit encoder to demonstrate positional stability that exceeded the SSM requirements in a laboratory environment. Stability measurements were within 2.5 arcseconds with less than an arcsecond of noise.

This breadboard unit was available for use over 6 months before the encoders were delivered. This allowed the control electronics team to build a set of functional breadboard electronics much faster than if they would have waited for the flight hardware. An optical cube was bonded to the shaft, and an autocollimator was used to measure the stability and repeatability of the shaft position. This allowed 24-bit positioning stability and knowledge to be demonstrated in the laboratory. This was an important technical milestone which gave program management confidence that the SSM controller could be delivered on schedule. The unit, and some sample performance data is shown in Figure 8.

Lesson Learned: Create a functional breadboard using components that can be procured quickly as early in the program as possible. These components do not need to be expensive. It will facilitate breadboard-level controls development, demonstrate performance in a laboratory setting, and potentially identify technical issues early.

Critical assembly procedures should be practiced to reduce risk
In order to meet the arc-second-level stability requirement, critical bearing shaft and bearing housing surfaces had to be ground to tight tolerances and the bearings epoxied in place. Then the critical encoder code disk mounting surface on the shaft and the read head mounting surfaces on the bearing housing are ground true to the as-assembled spindle axis. These operations occurred at the encoder vendor which
allowed the encoder to be “built up” directly onto the flight hardware. This would save many weeks of development time; the vendor did not have to build a GSE spindle, and calibration testing could occur on the flight spindle. It also saved the time of de-integrating the encoder system from the GSE spindle and re-integrating (and re-calibrating) onto the flight spindle.

This process meant that the bearings would be epoxied in place at the encoder vendor. Since this would clearly be a risky process (epoxy could get into the bearings), a practice bearing shaft and mechanism housing were fabricated from aluminum. These breadboard parts, with a flight equivalent, commercial grade DB bearing pair, were used to validate the bonding procedure. A dry build, where the bearings are installed without epoxy, was done to fit check all of the components prior to the wet (with epoxy) build. NASA engineers were on site to supervise the dry and wet build operation, ensure that no epoxy had migrated onto the bearing faces, and verify that proper preload was applied. The process was successful, and prevented a major mishap during the flight procedure.

Many months later, during the flight bearing installation procedure, it was noticed by vendor technicians that the flight bearings did not look correct during the build; the wrong face of the bearing was visible. Further inspection revealed that the back-to-back preload marking on the outer race of the bearings had been erroneously reversed by the bearing manufacturer. If the bearings were installed according to the preload markings, as specified by the procedure, the bearings would have been installed face to face, as opposed to back to back. This error was identified because the procedure was practiced (with non-flight, correctly marked bearings), and the assembly technician and supervising engineer knew which side of the bearings should be visible in a correct installation. If this had not been noticed, it is possible that the error would have been allowed to occur on the flight and flight spare unit as all of the flight bearings were incorrectly marked, not just one pair. The error would not have been discovered until very late in the program during environmental testing, when the SSM flight and flight spare would not meet stability requirements.

The only corrective action would have been to disassemble the SSM and to assemble a new bearing housing. This would have taken several months and would have been devastating to the program. Another option would have been to fly the unit with degraded performance, resulting in lower data quality. This was avoided thanks to a sharp-eyed engineer noticing that the dry build installation did not appear correct, even though, according to the markings on the bearings (and per the installation procedure), they were installed correctly. A review of the documentation provided to the bearing vendor showed that the bearings were incorrectly marked by the vendor. The markings were not inspected upon delivery as they were not removed from their sealed bags until installation to minimize potential contamination.

Lesson Learned: Practice critical procedures with expendable hardware. The same personnel that practice must be the same that assemble the flight hardware. Supervisory engineers must be expert in all aspects of the operation so subtle errors, which may be overlooked even by experienced technicians and quality assurance inspectors, can be avoided.

The importance of preload analysis
The SSM has the scene mirror at one end of the rotating shaft and the motor and encoder at the other. The scene mirror shares a view of the warm Earth and a cryogenic telescope. The mirror edge is within 0.5 in (12.5 mm) of a 150K infrared transmission lens and cold shroud. This drives the mirror to be cold, about 0°C, and the optic mount at 2°C. At the other end of the mechanism, the SSM motor is dissipating power and tends to be warmer, up to 21°C. In addition, the bearing housing has heaters which cause a hot region between the bearings. The overall result is a combined standing set of axial (between the bearings) and radial (across the races) gradients. However, the axial and radial gradients were different for the motor end and mirror end bearings.

A calculation of what preload was to be applied at room temperature, such that the system would cool down to the operational preload of 25 lbf (111 N), was undertaken. An initial, simplified calculation estimated that the operational thermal conditions would cause little change in the preload applied at
ambient conditions; the axial and radial gradients would cancel each other out. However, this analysis did not consider the differing conditions between each bearing, and indeed it was unknown how to create this complex gradient case in current bearing analysis codes such as COBRA and Bearings10C.

A bearing specialist was consulted and a bearing analysis code was modified to consider the dual gradient case. The results of the new analysis showed that the preload would decrease with temperature, and the ambient assembly preload should be 50 lbf (222 N). This result was counter intuitive; in a hard mounted, DB configured bearing pair, preload usually increases as the shaft cools relative to the housing. This analysis showed just the opposite was occurring. The gradients combined to decrease the preload as the temperatures of the mechanism decreased.

Due to the new analysis results, the ambient assembly preload was doubled to 50 lbf (222 N) at assembly. While preload was not directly measured after assembly, performance of the life test bearing cartridges and environmental test performance of the flight SSM indicate that the operational preloads were adequate.

Without the combined gradient analysis, we would have preloaded the bearings to 25 lbf (111 N) at ambient; which would have yielded a low operational preload (close to zero!) likely resulting in our not meeting the stability requirement and potential ball sliding leading to lubricant degradation.

**Lesson Learned:** Do not underestimate preload analysis for precision positioning applications if your bearings are subject to complex thermal conditions. Many resources exist within the space mechanisms community that can support complex preload analysis, and they should be utilized.

**Lesson Learned:** For precision pointing mechanisms in harsh, poorly defined or complex thermal environments, consider a compliant bearing support technique, such as a preloaded, diaphragm-mounted outrigger bearing to maximize flexibility to the unknown thermal conditions without sacrificing precision. Diaphragms are usually larger in diameter than a typical hard mounted or compression spring preloaded configuration and it is better to allocate the additional volume required during the conceptual design phase than attempting to acquire it later in the program.

**Heritage does not mean perfect**
The architecture of the 22-bit encoder used by the SSM was chosen based on heritage from a previous mission, with some modifications as many heritage board-level, radiation-hardened electrical components were no longer manufactured. Upon delivery of the encoder unit, already built into the spindle assembly at the vendor, encoder function was verified, the motor was installed and the rest of the mechanism built up. The encoder position signal was used for closed loop position control, and motor commutation.

Initially the SSM function was demonstrated successfully in laboratory conditions. Four months before delivery, just before environmental testing of the SSM was to begin, the position error detector in the Mechanism Control Electronics (MCE) began to trigger. The error detector is a watchdog that triggers the motor to power off and resets the controller whenever the difference between the 24-bit commanded position and the 24-bit encoder position exceeds a user-defined threshold for more than 4 consecutive 200-millisecond samples. The threshold value is set to ~2 milliradians; this is eight times larger than typical startup transients. Note that the MCE is based on a 24-bit absolute position sensor, but the encoder that was flown was 22-bit. This is because the initial MCE development was done using a commercial 24-bit absolute encoder. The flight 22-bit encoder position output was right padded (addition of 2 extra bits of zero value) to 24-bit to minimize changes between the breadboard and flight MCE.

The error detector was triggering because the most significant bits of the encoder position output would reset to zero values, but not when the encoder was actually resetting. This was the error. The reset occurred randomly, on timescales from hours to days, on both the A side and the redundant B side of the encoder system. At first, electromagnetic interference (EMI) due to inadequate grounding or non-flight harness was thought to be the culprit, due to the sporadic nature of the occurrence. Grounding schemes
and harness fidelity were improved, and the error event frequency decreased, but still occurred. The encoder vendor attempted to replicate the error at their facility with spare units that were not yet delivered, but could not. This indicated that the cause was due to some condition with the flight configuration, and was related to electrical noise.

A detailed noise review of the MCE and proprietary vendor encoder electronics was undertaken, and the problem was traced to a specific point in the encoder readout circuit. Of the 22 binary bits of position information returned by readout circuit, the most-significant bits were generated by a counter circuit. The least significant bits were generated by an interpolation circuit. The reset was traced to the counter circuit. The design of this circuit had heritage. However, analysis showed that it was potentially susceptible to noise. The noise generated by the MCE, even though of low level, was adequate to occasionally reset the counter bits to zero, corrupting the encoder readout. Unfortunately, being four months before delivery, there was no time to rework the MCE or the counter circuit. Random resets of the SSM would have been unacceptable to instrument operations and would greatly reduce science efficiency. A work-around had to be developed.

The error would occur as the upper most significant bits (MSB) of the counter were reset; the lower interpolated bits were unaffected. A solution was developed which assumed that the mechanism most significant bits were not changing; i.e., the mechanism was stable to 1.3 arc-minutes so the upper bits of the error signal would be zero. The corrupted bits could be ignored, and precision position maintained using only the least significant interpolated bits. The reset could occur randomly in this mode and not affect the positioning of the fine positioning mirror. However, there is no way to determine if a reset event actually occurs while in this mode. This mode was referred to as the fine pointing mode.

The only drawback to the fine pointing mode is the possibility of violating the underlying assumption that the actual error will not exceed the range of the lower bits. If the real error grows too large, this assumption is violated, and it is possible that the controller could drive to the wrong position. To ensure the limit would not be exceeded, an operational measure was put into place. The mirror should never be commanded to move from position to position while in fine pointing mode.

The fine-pointing mode effectively mitigated the noise-induced, MSB random reset error and allowed operations of the instrument to be unaffected. It should also be noted that the reset error never occurred again once the SSM and MCE (with flight harness) were integrated into the instrument, suggesting noise in the test setup as the culprit.

Lesson Learned: Heritage in a system should not be used as an excuse to reduce analysis. A signal integrity analysis, which would have identified the counter circuit noise vulnerability during the design phase, was not done in order to save time, and justified because of the heritage of the system.

Never skip the bake out
At the conclusion of thermal vacuum testing of the TIRS instrument, a small smudge was discovered on the scene mirror, at the cold end of the SSM. Further examination showed not only the smudge, but a fine layer of contamination, looking much like a light fogging of the entire mirror surface, was discovered. The smudge and contamination are shown in Figure 9. Further analysis showed that this was a film of Pennzane, which had outgassed from the bearings and onto the cold optical surface of the scene mirror, and the cryogenic lens surface adjacent to it.

The bakeout step was omitted from development at the vendor in order to save schedule. The unit would have had to leave the vendor facility for a vacuum bake out, and this would take at least 2 weeks assuming immediate availability at the external bake out facility, or a NASA facility. A thermal exposure test was done (not in vacuum) at the encoder vendor of 50°C and the risk was assumed by the project.

There appeared to be no lubricant outgassing issues during SSM component thermal vacuum testing. However, the contamination could have been building up during component level testing, and continued
to build up to a visibly detectable level during the instrument test. It should be noted that the film did not
degrade the science performance of the instrument or the performance of the SSM during the thermal
vacuum testing. The film collected on the scene mirror and the first surface of the cold telescope lens; the
two coldest surfaces closest to, and below the bearings. There was no detectable Pennzane
contamination above the bearings near the warmer encoder code disk and encoder read head optics. The
contamination was left in place as it did not affect the instrument and was not worth the cleaning risk.

Figure 9. After instrument level thermal vacuum, a smudge was detected on the scene mirror and
a haze of pennzane was discovered on the mirror surface. The figure shows the surface of the
scene mirror as viewed from the exterior of the instrument.

Lesson Learned: While the contamination did not affect the instrument or mechanisms performance, a
bake out would have prevented it. If the instrument operated in different wavelengths or a different
lubricant was used, science performance could have been adversely affected. The bake out should have
been done after spindle assembly but before installation of the optical encoder and scene mirror to
prevent contamination of the main optical surface, code disk or read head optics. It was again confirmed
that contamination builds up on the coldest surfaces close to the outgassing points, and even the best
labyrinth seal, properly coated with a barrier film, will not prevent the escape of outgassed lubricant. Note
that even with a bake out, lubricant outgassing will never be completely eliminated, only significantly
reduced.

Lesson Learned: Visible and high-resolution photographic inspection of critical mechanisms optical
surfaces must occur before, during (if possible) and after critical development events if possible. Proper
photographic documentation did occur at the instrument level but at the mechanisms component level
was of inadequate resolution to be useful for surface contamination detection.
Earth Shield Deployment Mechanism

The ESDM consists of a pair of spring-driven hinges and a release/restraint mechanism which allow a large earth shield panel to rotate 90 degrees from the stowed to deployed position. The large, 2.5-m$^2$ earth shield is a composite panel with carbon fiber facesheets and aluminum honeycomb core; it is shown in Figure 10. The overall architecture has all active elements housed on the instrument side of the hinge line; no harness is passed through the hinges. Triangular mylar “wing” sheets, used to close out the ends of the earth shield panel, were folded between the earth shield panel and the radiator panel. These wings were pulled along and unfolded as the earth shield rotated to its full deployed position. The major requirement was the instrument/spacecraft uncompensated momentum requirement; no more than 158 N-m of uncompensated torque and 25 Nms of momentum could be generated by this deployment. The driving requirements for the ESDM are summarized in Table 2.

Table 2. ESDM Driving Requirements

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Earth Shield Dimensions</td>
<td>78 x 50 x 0.625 inch (1.98m x 1.27m x 15.8 mm)</td>
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<tr>
<td>Earth Shield Mass</td>
<td>24.9 lbm (11.3 kg)</td>
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<tr>
<td>Deployment Angle</td>
<td>90 degrees</td>
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<tr>
<td>Deployment Time</td>
<td>≤ 60 seconds</td>
</tr>
<tr>
<td>Uncompensated Momentum</td>
<td>≤ 25 Nms</td>
</tr>
<tr>
<td>Uncompensated Torque</td>
<td>≤ 158 Nm</td>
</tr>
</tbody>
</table>

![Figure 10. The Earth Shield shown in the stowed configuration during TIRS instrument integration.](image-url)
Damped Actuator
The two ESDM latching hinges were based upon the deployable solar array panel hinges used on the Lunar Reconnaissance Orbiter (LRO) spacecraft. They were powered by stowed energy, using torsion springs. The torsion springs (one in each hinge on the hinge line) were sized to provide positive deployment torque margin even if one of them failed. Each spring hinge contained a locking pawl which engaged when the deployment angle reached 90 degrees. This prevented any type of backdriving, and positively locked the panel in the deployed position. In order to accommodate thermal expansion and contraction of the panel, one of the hinges had axially floating bearings that were compliant to small linear deflections along the hinge line axis.

Figure 11. The Earth Shield hinge system. Two spring-powered hinges drive the deployment. One hinge mounts a rotary damper to significantly reduce the angular velocity while maintaining high torque margin. The other hinge mounts a potentiometer to provide position feedback. The strongback beam allows the hinges to maintain relative alignment when the Earth Shield is removed.

The torque generated by the drive springs had to meet the torque margin requirement specified in the GSFC Gold Rules; it also had to meet the one failed spring success criteria. This resulted in the springs being strong enough to deploy the earth shield with enough velocity to exceed the uncompensated momentum requirement. In order to not exceed this requirement, a viscous fluid damper was used to reduce the deployment velocity, without significant reduction of the deployment torque margin. The heritage LRO viscous damper was utilized.

Due to the large size of the earth shield panel, it is removed from the instrument before shipment to the off-site spacecraft integration facility. The panel would be re-attached after the instrument was integrated to the spacecraft. To ease detach/re-attachment hinge alignment issues, and to facilitate testing, an intermediate stiff beam structure, or “strongback”, was introduced between the hinges and the earth shield panel. The hinges attached the strongback to the instrument, and the earth shield rode on the strongback. The earth shield could be easily removed from and reattached to the strongback without disturbing the hinge alignment. This configuration allowed the ESDM to be tested without disturbing the hinge alignment, and without the flight earth shield (using an equivalent inertia simulator).
Sensor
A potentiometer provided coarse position. The same model of potentiometer that was successfully flown on the LRO solar arrays was used.

Release/Restraint Device
A single-point, command-releasable restraint device was used to prevent the spring-powered hinges from deploying the earth shield. Two compliant snubbers were used to share launch loads with the hinges and the release restraint device.

The device used was an Ejector Release Mechanism (ERM). The model flown is rated for 1000 lbf (4.4 kN) of holding force. It utilizes a shape-memory alloy which, when heated, releases a threaded cap which is held in place with a ball lockup type of lock. It releases with very low shock. It is manually resettable, and no hardware change out is required between testing and flight; the same hardware that is tested is flown. The ERM-1000 is shown in Figure 12.

![Figure 12. The ERM-1000 release restraint device shown on the left with the release cap captive, locked to the ERM. The figure on the right shows the ERM during component thermal testing with the release cap ejected after actuation.](image)

The stock version of the ERM was modified slightly to work with the TIRS structure and electronics. The mounting flange, usually at the base of the device on the opposite end from the released cap, was moved to the same end as the released cap. This mechanical modification had heritage from other programs and was deemed low risk. Electrically, a 10-ohm in-line resistor was added to the heater circuit to better match our source current.

The ERM device is mounted to the instrument structure, and protrudes through the instrument radiator. The releasable cap attaches to the earth shield though a spring-loaded, bolt-catcher type compliant retraction interface, called the Coupler Retraction Mechanism, shown in Figure 13. This ensures that the cap clears the ERM completely upon release, and that the cap does not protrude beyond the interior surface of the earth shield, as this would slightly degrade the radiative performance of the shield. The mechanical contact between the structure and the earth shield at the lock down/release point is a matched radius cup and ball. This interface constrains all translations but is compliant to all rotations. To avoid the possibility of cold welding at this interface, dissimilar materials were used. The radiator side ball is aluminum 6061T6 with a type III hard anodized finish. The cone is titanium 6Al4V with a Tiodize® type II (Teflon® impregnated) finish. The ball and cup both have a large clearance hole through them that the ERM releasable cap and bolt catcher extend through. The cup and cone effectively isolate the ERM from earth shield moment and shear loads at the release point, allowing it to be loaded in pure tension.
A kick-off load was introduced into the stowed earth shield by bending it slightly. This was done by increasing the deflection at the restraint point, effectively pulling in at the center of the panel, reacting it against the snubbers and hinges. Due to this preloading, the panel was “bowed-in” when stowed. This bowing provided a kick-off load when the panel sprang back upon release, and obviated the need for dedicated kick-off springs.

Performance Testing Results
The ESDM underwent several deployment tests, with the velocity through most of the range varying between 2 and 5 degrees per second. The ESDM was deployed under ambient conditions, and in vacuum at room temperature and at -5°C. The thermal vacuum test is shown in Figure 14. Following acoustic and vibration tests, it was deployed again. All tests were successful and the results are shown in Figure 15. As is expected, the deployment time increased at cold temperature as the damper viscosity
increased. The analytical predictions were too fast by about a factor of 2 due to incorrect assumptions about the drag caused by the unfolding wings, which turned out to be much higher than predicted.

Figure 14. The Earth Shield and ESDM shown deploying within the thermal vacuum chamber after the successful cold vacuum deployment test. The drag torque from the wing close out sheets proved to be variable and made estimation difficult. Note the reflection of the chamber door and photographer’s camera flash on the inner surface of the deployed earth shield in the right figure.

Figure 15. ESDM deployment analysis and test data. The ambient analytical prediction is the blue dots on the left, the ambient test data is the green, blue, and red traces, and the purple trace is the cold vacuum deployment data. Some of the non-linear behavior of the damper is shown in the purple (cold vacuum) test trace near $t = 3$ seconds. Also, the variable drag from the wings unfolding is shown on the blue (post environmental test) trace near $t = 8$ seconds.
Earth Shield Deployment Mechanism Development Issues

Use of a heritage design allowed the ESDM to change very little between the system requirements review and the critical design review, which was extremely beneficial due to the compressed schedule. The ESDM did not have any major developmental issues, but several notable minor ones.

A 4 factor of safety is there for a reason
The foldable mylar wing close out sheets made analysis difficult. Estimates made of the drag torque produced by the wings during the design phase were too low and assumed it was constant; i.e., not a function of deployment angle. The flight wings ended up much heavier than predicted. The wings drag torque varied with deployment angle, and did not vary consistently, but hysteretically. However, the high torque margin allowed the design to absorb the increased torque without modification.

Lesson Learned: High torque margins are extremely desireable in deployable mechanisms. The high factor of safety (usually = 4) used for non-conservative forces/torques in the design of the mechanism allowed for unpredicted behaviors to be compensated for.

Not everything is in the included documentation
The ERM device would self-release if manually reset while the unit was still cold. This occurred immediately after the successful cold vacuum test, when the unit was being reset. Fearing a defective ERM, we immediately contacted the vendor, and prepared to do an in-depth failure examination. The vendor stated that it is perfectly normal for an ERM to inadvertently release if reset at cold temperatures, and that there should be nothing wrong with the ERM. This was not mentioned in the documentation.

Lesson Learned: Communication with vendors is vital at all levels, down to the smallest component. Timely contact with the ERM vendor saved the team the effort of pursuing a failure review, and identifying alternative release/restraint devices.

Test harnesses may interfere with deployments
The ESDM performed as designed and experienced only a single failure during development testing. This failure was not due to the ESDM itself, but to test instrumentation. During the cold temperature deployment test, an accelerometer popped off of the strongback; harness running from the accelerometer to the chamber wall snagged on the strongback and prevented it from deploying the full 90 degrees. The ESDM deployed nominally after reattachment of the accelerometer, and the offending harness was rerouted so as to not interfere with the deployment even if the accelerometer popped off a second time.

Lesson Learned: Assume test components such as accelerometers and thermal sensors will fall off during environmental testing. Contingency harness control measures should be implemented such that when the sensors unintentionally fall off, they, along with their associated harness, will not interfere with instrument or mechanism operations. Examples of these measures could be harness safety lines and/or additional harness tie down points. Adequate test harness length should be available to accommodate these features if this is considered early enough in the test program.

Cryo-Cooler Launch Lock Mechanism

Even with internal compensation, reciprocating elements within the TIRS cryo-cooler were known to generate jitter. Throughout the TIRS instrument development, this jitter was specified to be below a threshold where it would interfere with instrument operations. A few months after the critical design review, the as-built TIRS cryo-cooler was shown to have significantly higher jitter than was specified. This jitter needed to be suppressed. Passive vibration isolators were implemented between the cryo-cooler and the instrument structure, but these isolators did not have adequate stiffness to support launch loads. A launch lock would be needed. Luckily, a single extra pyro channel remained unused in the instrument control electronics, and could be utilized. It was available as a pyro-activated aperture door mechanism was removed from the design after the preliminary design review, but the electronics to run it were left in
place. The CCLL would use this channel. Several flight spare ERM release/restraint devices were on hand as ESDM spares. These would be used as the schedule precluded obtaining other release devices, and the available control electronics were designed to actuate this particular device. Spare ESDM potentiometers were also available, and one was used as as a sensor to verify the state (locked, unlocked, or somewhere in between) of the launch lock. The driving requirements for the CCLL are listed in Table 3.

Table 3. CCLL Driving Requirements

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Keel Dimensions</td>
<td>17.24 x 19.48 x 8.50 inch (438 x 495 x 216 mm)</td>
</tr>
<tr>
<td>Keel Mass</td>
<td>90 lbm (40.9 kg)</td>
</tr>
<tr>
<td>Allowable Keel Deflection</td>
<td>&lt; 0.015 inch (0.4 mm)</td>
</tr>
<tr>
<td>Stowed Frequency</td>
<td>≥ 60 Hz</td>
</tr>
<tr>
<td>Cold Operational Temperature</td>
<td>-40°C</td>
</tr>
<tr>
<td>Uncompensated Momentum</td>
<td>≤ 25 Nms</td>
</tr>
<tr>
<td>Uncompensated Torque</td>
<td>≤ 158 Nm</td>
</tr>
</tbody>
</table>

The cryo-cooler is mounted to an aluminum Keel plate which houses the compressors, expander, and cold heads. This Keel is suspended beneath the instrument -X panel, near the focal plane array (FPA). The first stage is coupled to the telescope 170K cold shields, and the second stage cold head is coupled to the 43K FPA. Both stages utilize flexible conductive link heat straps to provide a thermal conduction path that would not transmit jitter loads from the cryo-cooler to the FPA or shields. Circleflex™ damping flexures were used to provide passive vibration isolation between the keel and the instrument structure. Use of these flexures reduced the cryo-cooler frequency to levels which would damage the cryo-cooler at launch. The Keel Assembly is shown in Figure 16.

![Figure 16. The Keel Assembly, as modified with CCLL interfaces is shown. The 3 posts were added to provide accessible interfaces for a launch lock mechanism mounted on the opposite (interior) side of the -X instrument panel. The triangular frame was added later to increase overall stiffness.](image-url)
The only available volume available for the launch lock was within the instrument, opposite the externally mounted keel. Posts would have to extend from the keel through clearance holes (that would have to be cut) in the -X instrument panel. The lock hardware would restrain these legs.

To properly launch lock the cryocooler, it must be constrained in all 6 degrees of freedom. Only a single ERM-type release device could be used; hence the system could only be constrained at a single point. It was also desirable for the ERM to be mounted on the instrument exterior to facilitate easy access for manual resetting. Additionally, the cryo-cooler could never displace by more than 0.015" (0.38 mm) due to close clearances between the flexible conductive links and the FPA cold shield. Calculations showed that the flexures would displace the keel by no more than 0.010" (0.25 mm) when gravity was removed, or as the gravity vector changed when the instrument was rotated. The engaged launch lock carried the launch loads of the cryo-cooler keel, and when disengaged, allowed the keel to float on the passive isolator flexures.

Figure 17. The CCLL Assembly, as shown with the instrument -Y and -Z panels removed. Note that the Rotor Arm protrudes through the -Y panel, and the ERM and Linkage Guide Housing are mounted on the exterior side of the -Y panel. A spare ESDM potentiometer is used to verify rotor arm position and indicate proper deployment. It is mounted to the -X panel underneath the rotation axis of the rotor arm, and is tied to the rotor arm using a compliant coupling similar to the one used on the ESDM.
Figure 18. The overall architecture of the CCLL is shown above. The solid colored components show the launch lock in the unlocked position, with the ERM Cap released from the ERM body. The wireframe shows the position of the launch lock in the locked position, with the ERM Release Cap locked to the ERM body. The Rotor Arm rotates ~9.5 degrees between the locked and unlocked positions. Commercial grade tie rod end hardware was used in all of the linkages.
Pin Pusher Actuators
Several concepts were traded, with a 3-point constraint architecture selected. This used 3 drive pins on the instrument going into three holes on the cryo-cooler keel structure posts. The dimensional clearance between the pins and the holes is only 0.002 inch (0.05 mm). Each point constrained 2 translations and 2 rotations, providing a 2-2-2 degree of freedom kinematic lock. To release the launch lock, the pins were pushed through the holes by spring pistons. To facilitate resetting, the pins were not completely removed from the holes, and the holes were given a toroidal profile. The drive pins were stepped down so that a thicker diameter matched the toroidal hole diameter, and a thinner diameter provided adequate free clearance. The pins were pushed the distance required to remove the larger diameter pin length from the toroidal bushing hole engagement diameter to a seating hole in the anchor fitting. Each restraint point had a pin pushing, compression-type spring, as shown in Figure 19. Each drive pin was tied to a central rotor disk via a drag link. As the pins are pushed out of the engagement bushings, the rotor disk rotated approximately 9.5 degrees, as shown in Figure 18. If the rotor disk is not allowed to rotate, the pins are held in place and the springs are compressed. If the disk is free to rotate the springs are free to expand and push out the drive pins. The disk also couples the motion of all of the pin pullers together; if any single spring fails, the rotor disk rotation, powered by the surviving springs, will pull the pin with the failed spring.

Figure 19. The spring-powered, pin pusher actuator shown locked (stowed) in the upper figure, and unlocked in the lower figure. The ERM restraint device prevents the compressed drive spring from pushing the the green Drive Pin out of the Toroidal Bushing. When the ERM releases the central rotor, the drive spring is free to expand and pushes the Drive Pin out of the Toroidal Bushing and into a guide hole in the anchor fitting. To reset the actuator, the the Drive Pin is manually pulled back through the Toroidal Bushing (via the Reset Linkage and the Rotor Arm) and connected to the reset ERM.

The titanium Anchor Fittings and Toroidal Bushings had a Tiodize® type II (Teflon® impregnated) finish. The Drive Pins are made of aluminum bronze, CDA 63020 per AMS 4590B. A light film of Braycote
Micronic 602EF grease was applied to the Drive Pin, and to the sliding surfaces in the anchor housings and the toroidal bushings.

**Release/Restraint Device**  
The ERM restraint device prevents the rotor disk from rotating. When fired, the ERM releases the disk, the pins retract, and the cryo-cooler unlocks. The ERM is not coupled directly to the rotor disk. An extension arm runs from the rotor disk through a clearance slot in the instrument -Y panel. A tie rod runs from the extension arm to the ERM mount. This allows the ERM to be mounted on the instrument exterior, facilitating reset access. A potentiometer is used to indicate rotor position, providing a positive indication of the locked or unlocked state. The ESDM potentiometer was used, as spares were immediately available. Telemetry lines to support this sensor were also available in the control electronics.

**Cryo-Cooler Launch Lock Mechanism Development Issues**

To reduce risk and expedite schedule, a functional breadboard unit was fabricated quickly to validate the overall architecture, and the flight unit and an engineering test unit were developed in parallel. The developmental intention was to qualify the design in parallel with the flight fabrication. There was no schedule available to environmentally qualify a unit and then fabricate the flight unit serially. It had to be done in parallel. The ETU unit was built onto a flight-like composite panel for environmental testing. The flight unit was built up directly onto the flight structure and was qualified at the instrument level. The ETU and flight units were identical.

![Figure 20. In the left figure, the flight Cryo-Cooler is shown in development at the vendor, mounted within the flight Keel structure. Vibration testing of the ETU Keel Assembly (with Cryo-Cooler mass simulators) to verify post stiffness is shown in the center figure. The CCLL functional breadboard unit is shown in the right figure.](image)

It became evident during the initial development and analysis of the mechanism that 60-Hz stiffness was going to be a very difficult requirement to meet, with initial concepts predicted to be ~42 Hz. Several design changes were implemented, including optimizing the Drive Pin/Toroidal Bushing clearance and the addition of the Triangular Frame, which raised the system stiffness to meet and finally exceed the requirement. The CCLL first mode frequency was measured at 65 Hz.

Upon actuation, the CCLL produces a peak torque of 99 N-m (877 lb-in), and an angular momentum of 12.4 Nms (110 in-lb-s), within the limits specified in Table 3.

The CCLL ETU successfully passed qualification testing in September 2011 and the flight unit has been installed into the TIRS instrument. Many issues were encountered and ultimately solved in this unique mechanism.
Use of commercial grade mechanical components for space flight
Due to the compressed schedule, procuring flight quality tie rod ends and/or spherical bearings was impossible. The only option was to fly commercial industrial components. Traceability paperwork to verify material type and quality was generally not available, or was not sufficiently trustworthy for mission assurance. To reduce risk, many extra components were procured, and used not as spares, but for testing. Components were proof tested to measure yield and ultimate tensile strength, and chemically analyzed to verify material type, finish, and surface hardness. Parts that were selected from the lot for flight were static-load tested to qualification load levels before use. The commercial components used in the CCLL successfully passed mechanism component level and instrument level testing.

Lesson Learned: Generally available, commercial industrial hardware may be qualified for spaceflight use if a comprehensive set of tests to verify base material type, coatings, finish, yield and ultimate strength are carried out. The hardware should be purchased in lots and many identical units tested to determine unit-to-unit statistical differences and aid in flight part selection from the lot.

Unpredicted behavior during vibration testing
During vibration testing it was found that the frequency of the system would increase as full level input was approached. This occurred in the lateral directions only. The shifts in frequency for the Z lateral axes are shown in the Figure 22. The shift that was seen during testing in the Y-Axis was not as significant as the one seen for the Z-Axis. The Y-Axis shift was 10 Hz from -18 dB to Full Level. The shift for the Z-Axis was 31 Hz from -18 dB to Full Level. The predicted frequency was higher than the 65 Hz measured during testing, but that was for an idealized system, and did not take into account a number of factors that could be attributed to the difference.
Figure 22. The frequency of the stowed CCLL was shown to increase as the test level was increased to full level. Note how the first mode frequency peaks shift the right (increase) as the test level increases from -18 dB to full level. This behavior was not predicted, and occurred in the 2 lateral (ZY mechanism plane) axis. It did not occur in the axial (X, thrust) axis.

The shift was a result of the gap clearance around the Drive Pin to Toroidal Bushing interface. The gap was necessary to ensure that the mechanism would operate reliably at cold temperatures and was misalignment tolerant. As more energy was input into the system the gap would close, and the system would become stiffer. This characteristic made predicting the behavior of the system difficult, but established that smaller clearances made the system stiffer.

Lesson Learned: Mechanisms which utilize sliding interfaces are difficult to model accurately, and will generally be less stiff than predicted. The measured system frequency will increase as test loads increase to full levels. This is due to sliding clearances being reduced in proportion to input level during vibration tests where the lines of action of the sliding device are close to the test axis. This was not seen when the test axis was perpendicular to the mechanism lines of action.

Conclusions

The SSM, ESDM, and CCLL span a broad range of mechanisms types; from microradian precision mirror positioning to coarse deployment of the large earth shield. Due to the compressed schedule, it was not possible to fully develop engineering test unit mechanisms before the flight mechanisms. A protoflight approach had to be utilized to save time, and even then some prudent development processes, such as the encoder electronics signal integrity analysis, and the SSM bake out, were sacrificed. Higher levels of risk had to be accepted to meet the delivery schedule. We were able to mitigate the consequences resulting from the analytical and contamination control omissions, and the heritage justification to forego critical analysis was proven false. The building of a simple breadboard SSM mechanism, which verified
the performance of the architecture to instrument management, had as much "political" value as technical value.

Whereas heritage arguments hurt the SSM development, the ESDM benefitted strongly from heritage. It required little more than cosmetic changes from the heritage LRO design; its development proceeded comparatively painlessly compared to the SSM, yet some lessons were learned, and classic lessons reinforced. The relative simplicity of the mechanism allowed for adequate analysis to be undertaken, and many developmental performance tests to be done. The vendor-supplied components, such as bearings, potentiometers, and dampers, were known early and hence procured early, with adequate time for characterization testing and bake out. A dedicated breadboard was (correctly) not needed, as the TIRS flight design was a derated version of the original LRO design. This was the classic, proper heritage application case; the new mechanism was as close to "build-to-print" identical to the heritage mechanism as was reasonably possible.

The CCLL was just the opposite of the ESDM; there were no heritage mechanisms that would meet its requirements. The 6-month development schedule, and a fixed set of available interfaces, drove the team to a non-traditional approach that was difficult to analyze, but relatively easy to test. Here the early breadboard was absolutely essential to verify function, measure forces, and test critical sliding clearances. Methods were developed to lot-qualify commercial grade mechanical hardware (with little or dubious traceability paperwork) for flight use (albeit at higher risk), as flight-quality components were impossible to procure in the available time.

The TIRS mechanisms development was undertaken knowing that a higher level of risk had to be assumed in order to minimize the development time. Many weeks were lost and the instrument schedule re-arranged several times to accommodate delayed mechanism deliveries due to the consequences of this decision. Ultimately, the difficulties were overcome, the mechanisms were integrated into the instrument, and have functioned flawlessly throughout instrument environmental testing.

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