Refinement of a Low-Shock Separation System

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Abstract

This paper discusses the design of Marman Clamp-band Separation Systems, and several lessons learned by SNC over 12 years and multiple programs. Historically, there have been enough failures associated with Marman band designs that practical design guidance is publicly available. Utilizing a shock-dissipating release device has allowed us to sidestep many of the typical challenges associated with Marman systems and helped create a highly robust design baseline that differs significantly from the traditional system.

An overview of the purpose, components, and function of a clamp-band system will be presented. Common failure modes of this type of system will be discussed, and how these can be addressed. Early examples of successful SNC systems will be reviewed (NANOSAT, Orbital Express). Much of our early work on Clamp band systems and release devices was performed collaboratively with SAAB-Ericsson (now RUAG). The heritage design was refined recently to meet a challenging set of requirements for the EELV Secondary Payload Adaptor (ESPA Grande). By retaining the proven features of the design, we built an optimized sub-24 inch (61 cm) separation system with high stiffness, low-release shock, and a compact envelope.

Overview of Marman Clamp Separation Systems

A "Marman clamp" or "Marman ring" is a generic ring clamp used to join two cylinders butted together at the ends. "Marman Products" was the name of the company which first produced this type of clamp in the 1930s[1]. The Marman clamp was a sensible choice for spacecraft separation in the 1960s, as spacecraft stages are usually comprised of butted cylindrical structures of truss/beam construction covered with a skin. Separation of these structures normally entails severing the bolted connections. The advantage of the Marman clamp was to reduce the number of bolts to be severed for release, improving reliability.

The first Marman clamps were a flexible strap or band held around a series of circumferential V-wedges over the angled flanges of mated cylinders[2]. Load was retained by tensioning the strap such that the wedges could hold the ring flanges together with no gapping under the worst load case. A redundant release could be achieved by severing 1 or 2 (or more) bolts that hold the flexible band/strap in tension. This allows the V-wedges to move radially outward, freeing the mated flanges. Bolt cutters, pyrotechnic or frangible bolts are historically common choices for the release element. Release of these systems results in significant shock as the band strain energy (and pyrotechnic shock) is dissipated. Typically there are features to retract and "park" the band after release to prevent the band from interfering with release of the deployed payload. Marman Clamp separation systems (also known as "V-bands") have been used successfully on many missions. A generic system is shown in Figure 1.

Other features typical of these systems include indication of positive spacecraft separation with switches or continuity loops in electrical connectors, and kickoff springs sized and balanced to provide proper separation velocity and attitude.

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Design of a Separation System

Marman Clamp systems usually share the design objectives below. All three design objectives should be kept in mind and balanced against each other (along with weight and cost):

1. Create a stiff connection to the payload under worst case load conditions; retain until deployment
2. Release the payload upon command, with proper separation dynamics, minimal imparted shock, and no "re-contact" between separated payload and band or any other part of sep system (including debris from pyrotechnic release)
3. Create robust processes for installation, preloading, and testing of flight system, compatible with proper safety measures

Figure 2 shows the components of a typical system. The band is tensioned to the required preload, compressing the shoes (V-wedges) radially, generating axial preload, clamping the rings together. The clamped rings behave like a preloaded fastener head, creating predictable joint stiffness as long as gapping between the rings does not occur. Most systems have no trouble being stiff in axial compression, as the joint acts as a solid cylinder. Stiffness under axial tension is more difficult to achieve as the load path goes from flanges to the clamped toes of the joint to the mating flange.

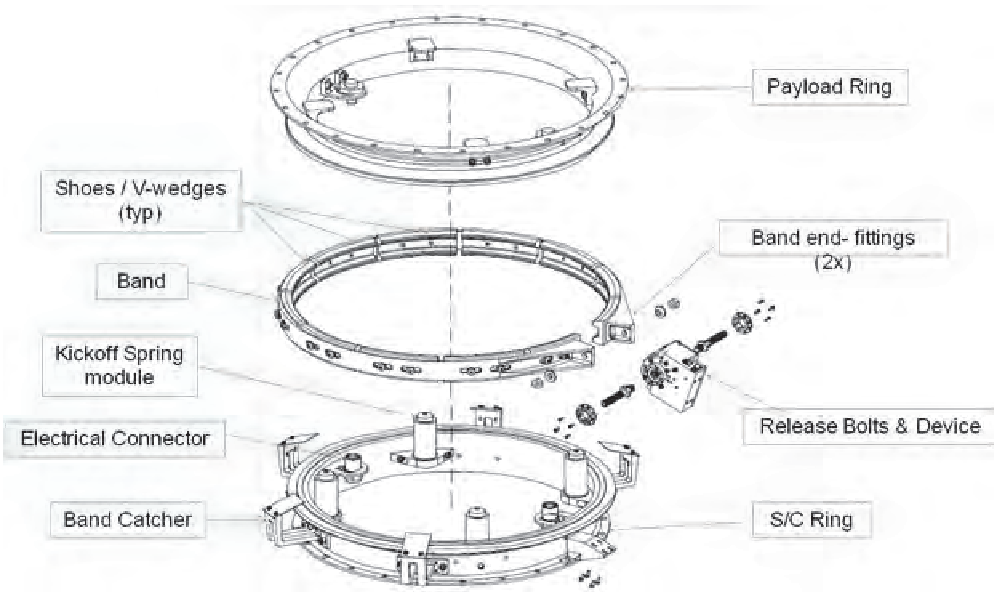


Figure 1: Generic Marman Clamp System

Table 1 shows a comparison of the traditional Marman system design approach to the alternative approach taken in our systems. These differences will be discussed in more detail to illustrate why our approach is an improvement.

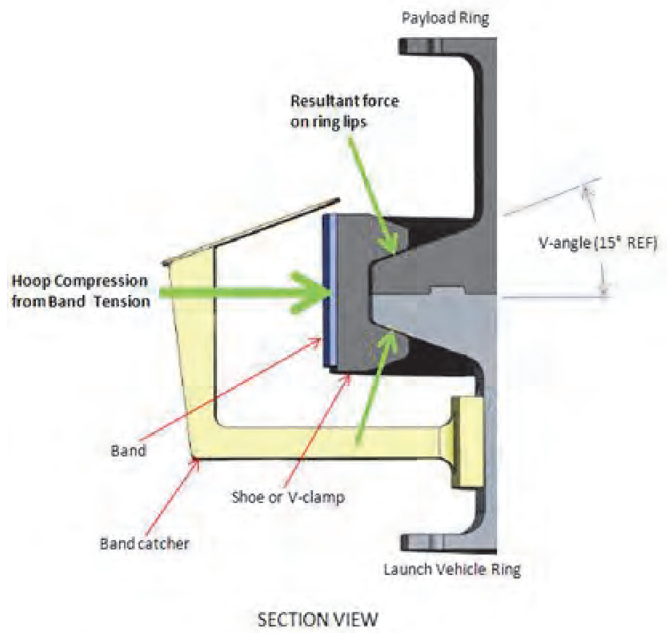


Figure 2: Generic Marman Clamp System Section

Table 1: Design Comparison: Traditional versus Improved Separation Systems

Feature	Traditional Marman Systems	Improved Separation System
Release Element	Pyrotechnic or frangible bolt separation	High-reliability, shock dissipating release device
Number of release points	2 release points minimum, 3 or more for larger rings	Single point release, redundant initiators
Band	Thin, flexible “strap” of steel (un-controlled shape)	Thicker, less flexible machined Alum band of controlled shape with free-state larger than installed state for release energy
Band Dynamics	Less predictable release dynamics; Dependent on band strain energy for release, Springs and/or tethers (extractors) to assist release and move band away from rings into catchers; Straps to contain ends of band after release	Predictable release dynamics due to controlled shape; Band “spring” energy sufficient for release, catchers “park” band away from rings after release; tethers and straps not necessary
Band End fittings	Trunnions w/ spherical ends connected to strap, or riveted “bathtub” fittings	“one-piece” band with integral machined ends (bathtub fittings); release bolts at each end have spherical washers
Band preload	Lower preload, lighter system	Higher preload, stiffer/stronger system
Interface Rings	Lighter weight on “flyaway” ring, shear lip, controlled gap, longer lips, angle >20 deg to reduce tension requirement	Matched stiffness, short lips for direct load path, minor inner “gapping” acceptable, smaller angle (15 deg)
Clamps/Shoes/V-Wedges/Retainers	Aluminum, Titanium, or steel	Aluminum w/ Titanium for highly loaded end shoes
Install and preload process	Set preload w/ bolt torque or strain gauges, tap band to equalize; pyrotechnic safety practices	Instrumented bolt indicates load, no tapping - single segment band w/ proper lubrication; reduced safety requirements
Preload Indication	Bolt torque, strain gauge on band or bolt(s)	Calibrated, Instrumented bolt

General Design considerations

Good design practices for any highly loaded system also apply to Clamp band systems:

Stress-Corrosion Cracking: These systems will be preloaded to the flight load, and remain loaded, possibly for months, until launch and deployment. Stress corrosion cracking has caused sudden failure in clamp band systems, so using materials with an A-rating for Stress-corrosion cracking is essential. Minimizing potential stress concentrations with controlled fillet radii, controlled thread roots, and rolled threads on loaded bolts is normal practice.

Minimizing Thermal Effects: Thermal strains can cause changes in band preload. Load drift should be avoided by using materials with similar coefficients of thermal expansion for rings and band, and shoes (to a lesser extent). Aluminum is a typical choice for rings. Shoes are typically titanium or CRES steel, and band is normally CRES steel. An all-aluminum design (with the exception of titanium in the more highly loaded shoes at the band ends) is preferred by SNC as will be discussed further.

Control of friction: This is critical at moving interfaces, especially between shoes and band, and shoes and rings in order to prevent lockup, equalize loading in band and prevent cold welding between rings and shoes. Surface finish (roughness) and solid-film / dry-film lubricants (DFL) MoS_2 ($\mu \sim 0.1$-0.15) and Dicronite® DL-5 ($\mu \sim 0.05$) are commonly seen in clamp band systems. With aluminum rings, chromate conversion coating followed by DFL provides $\mu \sim 0.1$. One tradeoff is that reduced friction between the rings, and between shoes and ring lips increases the required band load, but improves release performance and reliability.

Release Device

Traditional Marman clamp systems have used explosive bolt technology to load and release the band. The obvious disadvantages of these for this application are high release shock and inability to test the flight unit. While these devices have been used with success in clamp band systems, they present a number of other design issues to be solved which can complicate the design.

Multiple release points on the band have been recommended with pyro-bolt technology: 2 release points for systems up to 18 in (46 cm) and 3-4 for 60 in (152 cm) or greater[3]. This is primarily to add redundancy for release in the event of a pyro failure. Multiple release points also are recommended to allow more symmetric expansion of the band, to help it clear the rings. However, this approach increases chance for non-uniform loading of band/rings, and complicates release dynamics as multiple devices must operate simultaneously. More release points also increases the need for extraction features or added hardware to actively pull the band away from the rings such as springs and tethers. This adds parts and complication to the design, and increases system weight (more bolts, trunnions, tethers, springs). That weight may be better used to improve margins elsewhere in the system to improve stiffness (rings, band, etc.).

Inadvertent bending loads are common in Marman clamp systems and have been responsible for flight failures (overload, stress corrosion, or concentration in roots or creep of explosive bolts). Some technical literature suggests that despite good trunnion design, the failure mode may be inherent in the perpendicularity tolerancing of bolt head and nut threads and seats. Previous failures in these areas have been addressed with the addition of spherical seats or other misalignment features, and use of greases at the bolt attachments. [4]

Explosive bolts have other well-known drawbacks. Flight units cannot be tested. Margins are determined statistically, through lot testing. There are also safety concerns of unintended release. NASA design guidelines state that all debris from pyro-technic release should be contained – as in the case of exploding bolts, bolt cutters, etc.

During our early work with these systems, we developed a release device specifically for clamp band separation. The Clamp Band Opening Device (CBOD) uses patented FASSN (Fast-Acting Shockless Separation Nut) technology to dissipate stored strain energy resulting in extremely low shock to

Spacecraft and payloads. Use of a high-reliability release device allows for one release point. This not only addresses the above shortcomings of pyros, but drives the design of the entire system in a very favorable direction.

The CBOD (see Figure 3) consists of two bolts with opposite hand, high lead threads which engage the same central flywheel nut. The flywheel is fixed, and released, with a retractable pin. When the CBOD is stowed, it functions as a single long bolt, which engages the ends of the clamp band with a simple nut at each end. The band preload creates tension in the CBOD bolts, which exert a back-drive torque on the flywheel. Upon release of the pin, the flywheel is free to spin, releasing both bolts (and the band ends). CBOD slows the release event, dissipating band strain energy into rotational kinetic energy of the CBOD flywheel. This allows for higher band preloads without excessive release shock. Excess release shock can create unpredictable band behavior, potential rebound of the band, and possible damage to the vehicle or spacecraft.

Redundancy in CBOD is achieved with the use of a pin pulling device with dual NASA standard initiators (NSIs). This is the smallest and lightest part of the release device, and a very efficient means of meeting redundancy requirements. The CBOD is fully-resettable for multiple uses, with replacement of the NSIs only. The CBOD has controlled thread-form, rolled threads, and each is proof-tested. Margins on CBOD are verified by proof-load testing, and release performance of every flight unit is tested at component level and after integration. A Small CBOD version has $^{5}/_{16}$-in (8-mm) bolts and 3500-lb (15.6-kN) rating; the larger version has ½-inch (13-mm) bolts and a 13,500-lbf (60-kN) rating. The CBOD has been used successfully in over 450 Flight band releases.

Several pyrotechnic system failures have been associated with failure of the explosive bolt [5]. Higher design margins are possible with the CBOD compared to explosive bolts since the latter must balance margins of retention and release against each other. Mounting configuration of the CBOD includes spherical seats for the end of the bolts and ensures CBOD is loaded purely in tension. High design margins, and an 18,500-lb (82.3-kN) component proof load test (~1.4x nominal load) on the CBOD cover potential overload encountered during or after preloading.

A single release point band equipped with the CBOD allows for a new design approach in which the band release is not dependent on stored strain energy in the band, or on external springs & tethers. Instead, the band is fabricated to a free-state which is larger in diameter than the rings. This creates stored elastic energy in the stowed condition and positive motion of the band away from the rings upon release. This will be discussed in more detail, to illustrate how the device has driven the improved design.

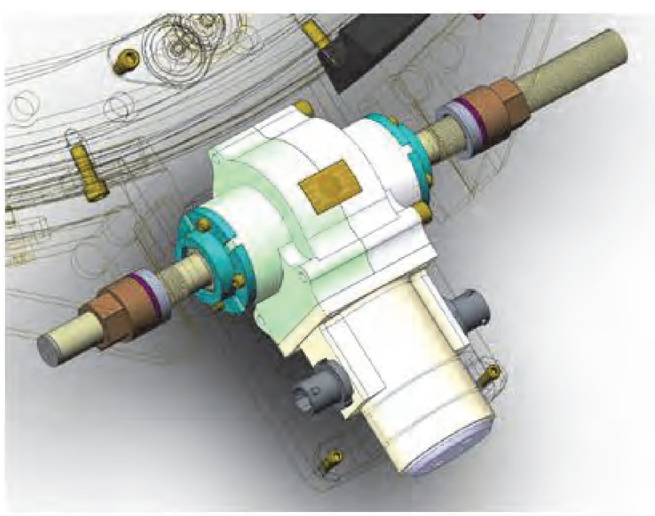


Figure 3: 13,500-lb (60-kN) Clamp-Band Opening Device (CBOD)

Band

The band (or "strap") is the most highly-loaded element and is the primary carrier of all loads in a well-designed system. All forces acting to separate the clamped joint including shear, bending moments and axial loads should be reacted by tension in the band -- for reliable performance of both release device and clamp band. This also ensures that loads are more easily quantified, preventing the possibility of overload. The band also contains most of the strain energy, which requires a ductile material with high elongation. Resistance to stress corrosion cracking is also a key requirement as this has been cited as a cause in previous clamp system failures [6].

The band must have fittings at the ends to engage the release device. These fittings are typically either trunnions or deeply gusseted "bathtub fittings". Because of the large strains in the system as load is applied, there is potential for misalignments, which can impart unintended loads into the tensioning bolts or release device. The end-fittings should be equipped with spherical seats to ensure the bolt sees only tension. It is also important to keep the tangential line of action of the bolts as close as possible to the band to minimize bending moments and radial load on rings caused by these moments. Attachment points need to have extra margin due to the potential to overstress these during installation as load is applied and increased.

Most traditional systems have used a thin steel band (thus the term "strap") to carry the tensile load (most systems are in the range of 3,000 to 10,000 lb (13 kN to 44 kN) depending on size)[7]. This seems like a good choice for the load retention design objective. It is easy to fabricate from band or sheet stock and secure end-fittings with rivets or screws. The disadvantage is the dynamic behavior of a thin-steel band (sized to carry the static loads) during release. Insufficient strain-energy in the band is a known failure mode. In one case, a design combined the shoes and band into one piece (in aluminum) and during a test, the band hung up on the rings and did not come free. The system was revised with separated shoes and steel band increasing strain energy, correcting the problem [8]. At the same time, excessive strain-energy in an overly flexible band (such as a thin steel "strap") can lead to a chaotic release dynamic and re-contact between the band and ring.

We have found that a one-piece Aluminum band, from 7075-T7351 (stress corrosion cracking resistant), with integral "bathtub fittings" is an ideal approach. First, it is a similar material to rings and shoes in our design, eliminating CTE effects and load drift. Second, it removes the reliance on high band strain energy for the proper release dynamic. We design for a known installation deflection (i.e., known force) by machining a one-piece aluminum band to a free-state dimension such that the installed diameter provides a known strain energy (see Figure 4). This type of band springs outward, free of the rings on its own, creating a robust release dynamic. After release, the band remains rigid enough to prevent re-contact with the rings. One drawback of this is that the deflection of installation adds to the total stress state in the band, but thickness of the band can usually be adjusted to achieve proper margins.

Manufacturing cost for the band is higher with a one-piece aluminum band, which is machined from plate-stock, compared to a steel strap. But this cost is recouped by fewer parts in the assembly, by eliminating separate end-fittings and the attachment method (screws or rivets). Springs or tethers to extract the band are also unnecessary with this approach, resulting in a simpler, more robust system with less analysis, less assembly effort, and less testing.

Figure 4: One-Piece Aluminum Band Machining

Rings

Since a stiff connection is a primary design objective, it makes sense that the stiffness of the rings (both spacecraft and launch vehicle sides) should be as high as possible, limited by weight and/or envelope. Deformation of one or both rings has been identified as a failure mode in clamp band systems[9]. The term "ring rolling" was coined for severe ring deformation under band and applied loads. Ring rolling is a collapse of the ring under hoop compression due to increased radial loads or unintended local loads. This can cause loss of band tension, bending loads at the release device or separation bolts, or in severe cases, the band coming free of the ring lips. Rings should also be of uniform stiffness along circumference, with no structures or attachments that alter stiffness (thus deflections) around circumference. This applies to both radial and torsional stiffness. Local force concentrations can result in deflections and create instability of the band.

For the same reason, rings on each side should be as close as possible to *each other* in stiffness. An accepted guideline for these systems is that the spacecraft ring must be at least 70% of the stiffness of the adapter ring[10]. This ensures rings share load and do not deform excessively. To maintain the stiffness of the entire system the load path between the bolt circles on each ring and the clamped lips on each ring should be as short and as direct a load path as possible. A proven approach to achieving this has been to maintain a small gap between the ring lips so the ring-to-ring contact point is closer to the bolt-circles, rather than having the load path extend out to the clamped lips. We have achieved good stiffness results by simply keeping the toe of the ring short, and not intentionally gapping anywhere. Keeping the toe short also reduces stress in the v-wedges.

In clamp band systems, gapping between the rings is considered unacceptable and a sign that the load limit has been exceeded. During early development work, we found that a small gap could be formed on the ID of the rings (0.005 - 0.010 in / 0.13 - 0.25 mm) due to deflection of the rings under band load and applied axial load. These gaps did not impact the stiffness of the joint or the performance of the system (did not cause non-linear load vs. deflection. Also, it did not cause shock or dynamic responses typical of gapped systems). We concluded that small local gaps at the ring interface (not complete gapping) are due to elastic deformations of rings and clamp and should not be considered a failure criterion. The reason to prevent gapping is to maintain stiffness of the joint, and if stiffness measurements show proper performance with local gapping present then this is an acceptable condition. There is support for this position in a previous technical paper on the subject[11].

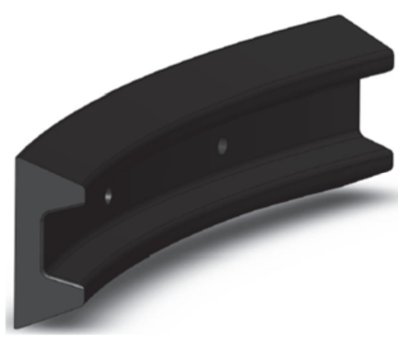
Historically, most systems have used a half-angle on the flanges in the 15° – 30° range[12] (see Figure 2). Larger angles increase the band load required to prevent gapping, but smaller increase risk of lockup between ring lips and v-wedges for reliable release. The ring lips (and mating surface on the v-wedges) are a critical release interface and require a dry-film lubricant. This provides a friction coefficient of about 0.1, and prevents lockup between wedge and lips. It also prevents cold-welding or galling with the v-wedges, and ensures proper sliding to allow even loading of the band as tension is applied. We also recommend use of the same lubricant at ring interface mating surfaces (unless electrical grounding requirements through this interface preclude it).

The rings of most systems contain an angled step or lip on the mating surface of the rings normally referred to as a “shear lip”. It can appear that this step is intended to react payload shear loads. In our view, good systems do not react shear loads with a lip on the rings since this can lead to large local axial loads in the ring, causing deflection and possible ring rolling. Despite the name implying shear load capability of this feature, the shear lip is only a locating feature to help mate the rings during integration, which is helpful with heavy payloads. We call this feature a “centering ramp”, since our approach is to react shear loads with band tension.

Other approaches for reacting transverse loads between the rings have been used successfully. Cup/cones at the ring surfaces have been used successfully, but increase the risk of unintended tip-off. Another successful design for reacting shear loads is shear pins inserted axially through each v-wedge, mating with axial slots in the ring lips all around the outer diameter. But shear pins must precisely fit mating rings and shoes, and can make for expensive, fussy fits. Generally cones, pins, or splines to take shear loads may save ring weight but add tolerancing and machining difficulty, and can interfere with clean spacecraft separation. They may also make integration and mating of rings much more difficult, especially with large spacecraft. There is support for increasing band preload to take shear, rather than having separate shear features[13]. It may be that shear features are added in order to keep band tension low to show higher margins or reliability, and to save overall weight.

Shoes / Clamps

The shoes (also called “clamps”, “v-wedges” or “retaining wedges”) transmit hoop compression from the band into axial compression between the rings. They must also resist the full axial tension acting to separate the payload from the spacecraft. Ideally the shoes would cover the entire circumference of the rings, but the release device is normally in-line with the band and requires a small area with no shoe. The shoes at each end of the band carry higher radial load due to the moment arm created by the offset between the release device and the band lines of force. This offset should be minimized but is difficult to eliminate. Excessive offset, and poor support of the ends of the band can lead to band end deflections and bending moments dumped into the release device. This could lead to premature release, or failure to properly release the band. Since the end-shoes carry higher loads, they should be made of a higher strength material such as titanium, compared to the remaining shoes which we make from aluminum 7075 (for similar thermal expansion with rings and band). Relatively short, angled faces minimize engagement of the shoes on the ring lips. This reduces stresses in the shoes, and improves release performance.



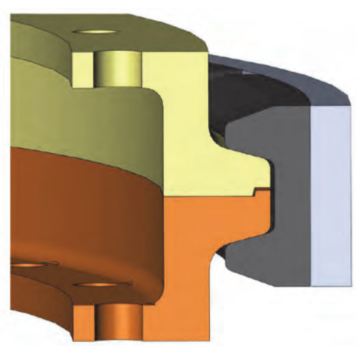


Figure 5: Shoe example (left) and Shoe fit on mated rings (right)

Proper mating between the shoe and the angled ramp of the rings is critical. The half-angle on the shoes matches that on the rings: (15 degrees typical & recommended). The shoes must have freedom to self-align at this interface, so all of the shoes’ mounting points have degrees of freedom. Shoulder bolts in that back of each shoe, riding in slots in the band, allow the shoes to move along band during tensioning, helping the band to load evenly as tension is applied. It also helps prevent potential bending overload of the shoulder bolts. Since the shoes move relative to the band and ring during loading it is important that the shoe position is monitored or actively controlled (with spacers, for example) as band load is incrementally increased.

We use a dry-film lubricant (DFL) on all shoe surfaces. This mitigates any potential for cold-welding and ensures a low and relatively controlled friction value between the shoe, ring lips, and band. In addition, a thin layer (0.003 in / 8 µm) of Teflon tape may be used on the back of each shoe to further reduce static friction with the band. Some technical literature warns against the use of Teflon between shoes and band, for the possibility of decay of preload due to cold-flow of the Teflon[14]. We have not experienced preload loss using this thin Teflon layer, and any small change in film thickness would have a negligible impact on band tension. The Teflon tape does require inspection after each load and release cycle, and can easily be refurbished when necessary.

The number of shoes is not critical as long as there are enough to keep the shoe length down. The longer each shoe, the greater the radial distance the shoe must move to clear the ring lips. Shorter shoes also reduce potential binding between the shoes and ring lips from small rolling deflections in the rings under load. Early in our clamp-band work we selected 12 shoes based on other similar systems, and have used this number successfully in our 17-inch (43-cm) and 24 (61-cm) systems.

Band Release and Band-catching Features

Clamp band systems normally have features which receive the band after release and “park” it where it cannot interfere with the separating rings. “Recontact” or “rebound” of the band into the rings can interfere with successful separation. Some systems actively retract the band with tethers and/or springs to move the band into catchers. Catchers normally consist of a radial stop for the band at some safe diameter away from the ring lips. Some systems have used energy absorbing features in the catchers such as a crushable element to reduce system shock.

The release behavior of a highly loaded, flexible, thin strap of steel could be hard to predict or control. Since the strain energy in our systems is mostly dissipated by the CBOD, shock absorption is not required at the back of our catchers to prevent rebound, simplifying their design. The stored elastic energy of the deflected band causes the band to spring free from the ring lips, so no tethers or retraction systems are required. An angled piece of Elgiloy spring at the top of the catcher deflects the band away from the deployed ring and into a rest position within the catcher (see Figure 2).

We use high-speed video extensively to study the band release dynamic in a new design. By observing band behavior with high-speed video, we can make changes or adjustments to ensure the band expands symmetrically from the rings, and cleanly separates. Catcher locations can be tuned to ensure the band is received in the correct places to limit band motion and potential rebound. During a flight build, virtually every band release from confidence testing through acceptance is filmed and reviewed for proper release dynamic and any anomalies which might otherwise go unseen.

Separation Dynamics

Kickoff springs between rings are used in clampband systems to provide the proper separation velocity for the payload. Normal separation velocities are in the 30.5 cm/sec (1 ft/sec) range and there is typically some maximum allowable tipoff rate specified. Electrical Connectors that cross the separation plane, and separation switches, require energy to separate and can impart forces to the rings during deployment. To minimize tipoff, the net forces driving ring separation must be balanced around the circumference. Placing matched pairs of switches and connectors 180° is normal practice in clampband design. Kickoff spring forces are balanced by including features for tuning spring force within the kickoff module. This could be accomplished by testing and matching opposed sets of springs, but adjustability can be included with little added effort, expense, and weight.

High-speed video is used to observe the separation dynamic and measure velocity and tipoff rate. High-speed video during development is key to understanding and tuning release dynamics of system. This includes the design and location of catchers and springs, and removing features which interfere with band motion away from rings and into catchers. The challenge in the setup is creating a mass simulator and off-loader that is representative of the flight release condition. The Nanosat clampband system was successfully tested in a zero-g flight to verify proper separation dynamics (which is not as expensive as one would think).

Installation and Preloading

The ground installation procedure is critical to set the band properly for flight. Specific tooling and processes for installation and preloading are an important part of the design effort and should be considered earlier rather than later as this can drive aspects of the design. A controlled, safe, and repeatable process is necessary and must be used in qualification and acceptance testing, as well as during integration. Development of ground support tooling and reset and preload procedures are critical to mission success. If possible, it helps to understand how the payload is to be integrated. How will kickoff springs be compressed and rings mated? How is the access to fasteners, and to the release device? Is there sufficient room to use any support tooling for the tensioning process?

Safety concerns must also be addressed. Traditional safety measures were taken to avoid an unintended pyrotechnic release. Regardless of the type of release element used, redundancy is required during loading of the band. In most systems we have seen, the tensioning tool serves as a backup to prevent catastrophic release due to a failure during load application. Shearing shoe fasteners during preloading is a potential failure mode. Shoes must be properly spaced before loading, and monitored as band load is increased, to accommodate band stretch. A shoe spacing tool can be used to ensure shoes and fasteners are positioned for maximum travel and even loading of the rings. Position of the release device is similarly adjusted and controlled as load is increased.

Ensuring the band loads evenly is critical for safety. Tapping around the band with a small mallet is the traditional recommended method, but is definitely not desirable as a practice. During development testing a reliable process for ensuring uniform band stress during loading must be developed by instrumenting the band with strain gauges in several locations (~5). We've found in our systems that with controlled friction between the band and the shoes, tapping is not required for uniform band loading.

Accurate application of band preload is difficult in systems which do not include direct, calibrated load sensing devices. In many cases this has led to over-design of Marman band systems to cover variations in preload application of up to ±25%. The CBOD uses an instrumented bolt to accurately set a known preload to the tensioning band, which can be verified prior to launch. The instrumented bolt eliminates inaccuracy of torque measurements or sensitivity to friction at fastener interfaces to indicate proper preload. It also eliminates need for strain gauges on band, and affords better accuracy via factory calibration (strain gauging band during development testing is still required to ensure band loads evenly). In the case of the sub-24-inch (61-cm) system, specific hydraulic tooling was developed for ground support. Features were incorporated into the band ends to mate with the hydraulic loading tool (see Figure 6).

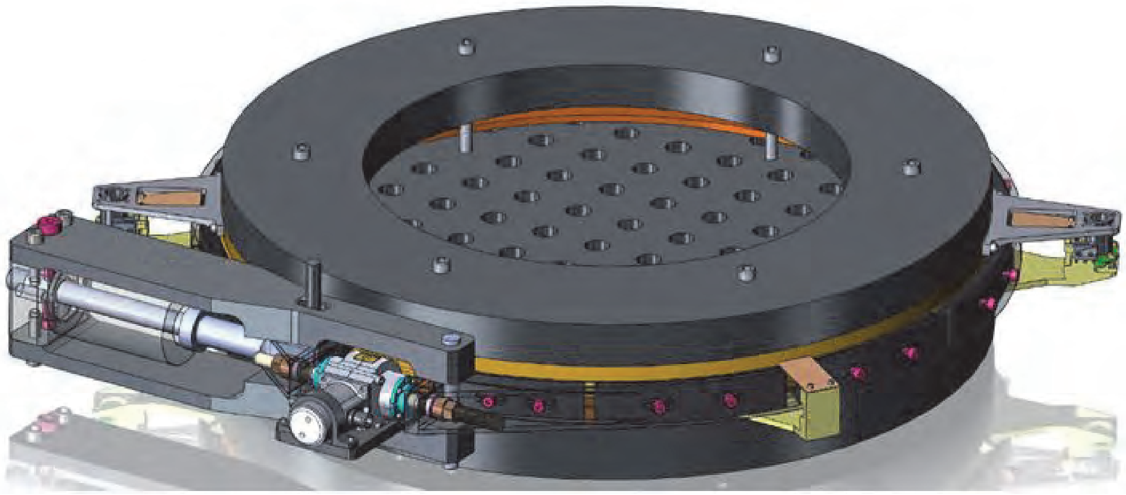


Figure 6: Hydraulic Loading Tool in-place, rings mated to fixture plates

Preliminary Sizing, Analysis, and Test

Marman systems are prone to stack-up of design margins resulting in over-designed (and overweight) systems. Analysis uncertainty, probabilistic load cases, margin-on-margin can drive design loads excessively high. Choose realistic load cases with reasonable margins to size band, shoes and rings as these have a big impact on system weight. A reasonable choice is to design and test to 1.25x worst case load predictions. Maintaining high margins on the release element (>2.00) has been recommended, and allows for any increase in loads, with a relatively low impact on system weight[15].

With design loads defined, a band tension can be determined such that complete axial gapping of the clamped joint will not occur at the maximum load case (an exception is small localized gapping due to elastic deformations). Initial band load is determined using a standard set of equations used for Marman systems, derived from a force balance about the clamped section[16]:

Axial Line Load: $W_{axial} = F_{axial} / \pi D$

Moment Line Load: $W_{moment} = 4M / \pi D^2$

Total Line load: $W_{total} = W_{axial} + W_{moment}$

Band Preload estimate: $P_{band} = W_{total} D (\tan\beta - \mu)/(1 + \mu \tan\beta) \sim W_{total} D \tan\beta$ *(neglecting friction)*

D = ring diameter
β = ring ramp half-angle
μ = coefficient of friction

A margin on gapping of 1.25 is normally applied to this result. The above loads allow for initial sizing and modeling of band, shoes rings, and other components. Finite Element Modeling is then used to analyze for stiffness and model system gapping behavior under load.

Not surprisingly, all agree on the need for extensive development testing to validate any separation system design. Testing is performed to confirm analysis results for system stiffness and gapping loads, and to validate or refine the flight band tension callout. Tests verify uniform loading of the band and proper stowing using the stow and load procedure steps. High-speed video is used to verify proper release performance of the band and system. These development tests are all performed prior to a comprehensive qualification test program.

Examples of Early Low-Shock Separation System Designs

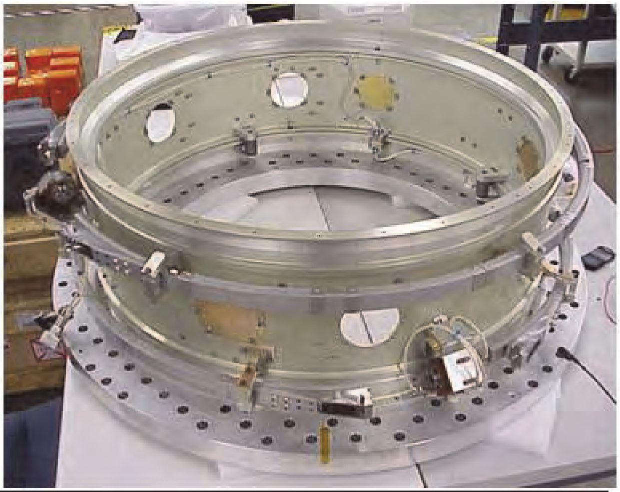


Figure 8: Orbital Express Separation System using 2 RUAG Bands and SNC mini-CBOD

Orbital Express Separation System

- Utilized a 37” (94-cm) Soft Separation System High Performance Clamp Band by RUAG with Clamp Band Opening Device (CBOD) by SNC
- Primary structure between NEXTSat and ASTRO during launch and initial on orbit activities
- Separated NEXTSat and ASTRO on flight and release intermediate structure so that subsequent mates and demates using the OE docking system can occur
- One Flight unit delivered
- Line Load: 32 kN/m (184 lbf/in)
- Band Preload: 14.7 kN (3300 lbf)
- Band Yield Stress Margin*: 1.54
- Launched in 2007

** with SFy = 1.1, SFu = 1.4*

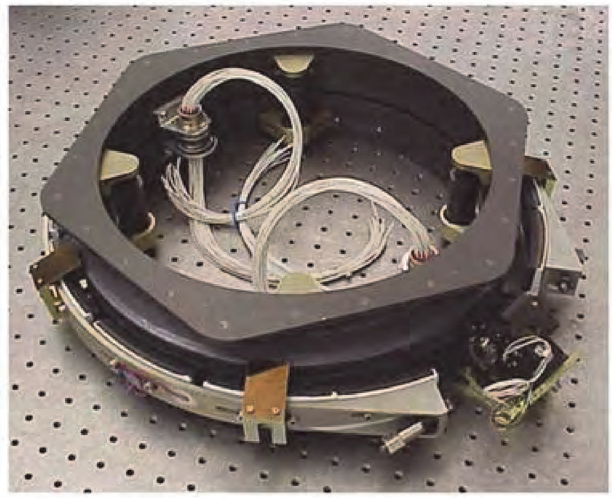


Figure 7: SNC NANOSAT 43-cm (17-inch) Separation System with Mini-CBOD

Nanosat Separation System

- Developed for AFRL, collaborative effort with RUAG
- Four flight units delivered
- 17” (43-cm) interface diameter
- Clamp Band Opening Device (14-kN / 3250-lbf preload Mini-CBOD) imparts extremely low shock to Spacecraft and payloads
- Protoflight test program included Band Proof load test, Thermal cycling, Random Vibration, Static structural testing, and Kick-off/Tip-off verification. Shock response characterized.
- Line Load: 42 kN/m (240 lbf/in)
- Band Preload: 14.7 kN (3300 lbf)
- Band Yield Stress Margin*: 0.42
- Launched in 2004, Delta IV Heavy Lift

Development of Sub-24” (61 cm) Separation System

For ESPA and ESPA-Grande applications, the secondary payload is cantilevered relative to the launch axis and space is at a premium along this line (see Figure 9). The required stack height (payload interface to ESPA ring interface) was 2.0” (5 cm), or roughly half that of the NANOSAT system. Stiffness and load requirements were increased compared to our heritage systems. Load requirements for ESPA are based on a 660-pound (300-kg) payload, with c.g. located 20 inches (51 cm) axially from the interface with the Separation System with inertial loads of 8.5G applied in two directions simultaneously[17]. This presented a challenging set of requirements for the sub-24” system (interface bolt circle diameter is just under 24.00” (60.96 cm)). With a desire to retain as many aspects of successful previous designs as possible, the design was revised to meet or exceed the specified requirements for ESPA Grande. The first step to a qualified design was to complete the design and analysis predictions, and fabricate an engineering model for testing.

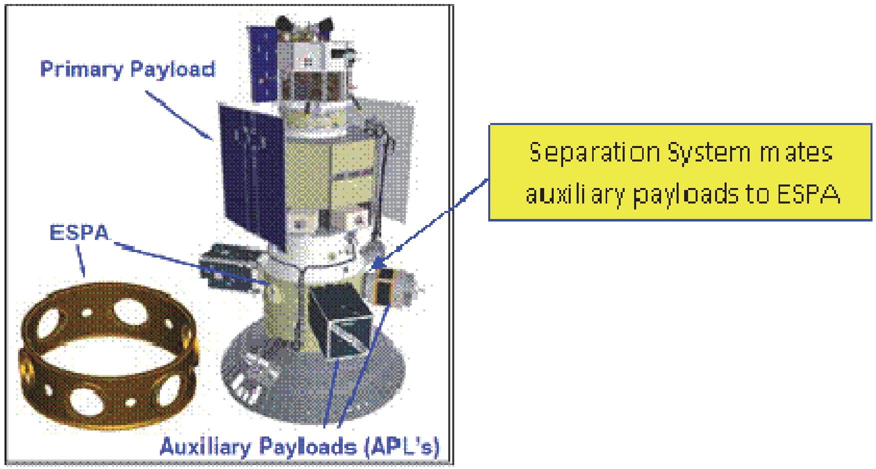


Figure 9: ESPA Separation System

To fit the required envelope, and to maximize stiffness, a major revision was made to the heritage ring profile (see Figure 10). The stack height of the system was reduced to within the required 2.0" (5 cm). Mounting flange of both rings were moved inboard of the tensioning band to improve load path and increase stiffness. This makes mounting fasteners less accessible, but is easily addressed by either accessing fasteners through holes in the ESPA ring, or installing rings prior to mating the payload to the adaptor ring. Preliminary stiffness analysis showed that the scaled, reduced height ring profile did not meet our stiffness goals, so the section was again revised. We thickened sections and shortened toes on each ring to improve stiffness and load path. These revisions resulted in roughly 3-fold improvement in stiffness, as seen in Figure 11.

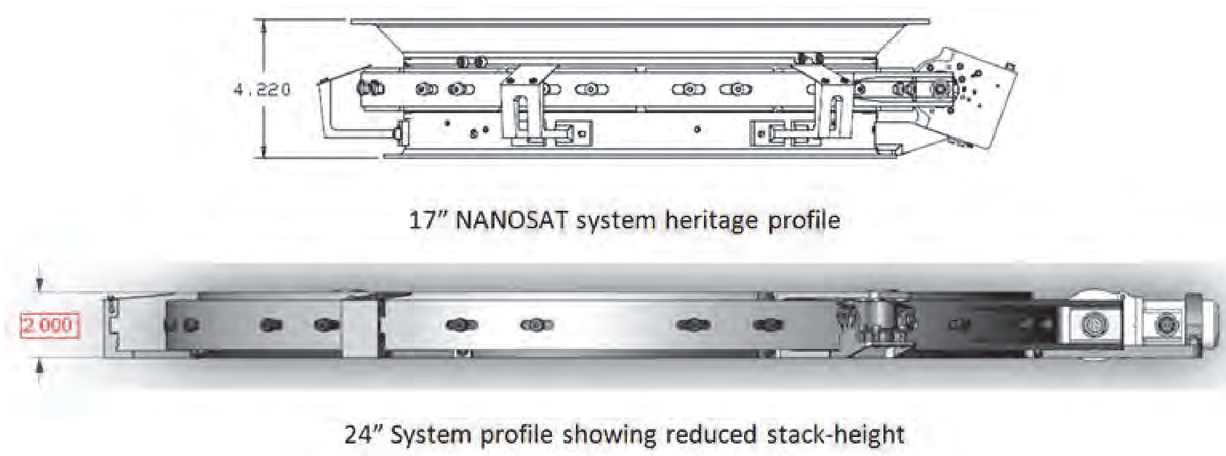


Figure 10: Profile Revision: Heritage design to sub-24" system

Catchers, switch mounts, and kickoff spring modules were also reduced in height. Fitting of kickoff spring modules into the smaller envelope was a challenge. The 660-lb (300-kg) payload mass requirement and 1 ft/sec (30 cm/sec) separation velocity (V_{sep}) goal required a larger amount of spring energy in a smaller volume than our prior designs. Our goal was to keep the kickoff spring module confined to the radial space between the rings, as a baseline. For our engineering system, we designed around a commercially available spring, and added 2 kickoff modules (total of 6) versus the heritage 4. Payloads up to $^{2}/_{3}$ the maximum load will achieve the V_{sep} target. Additional kickoff energy for future programs would be added by increasing the number of modules, utilizing volume inside the ESPA ring, or both.

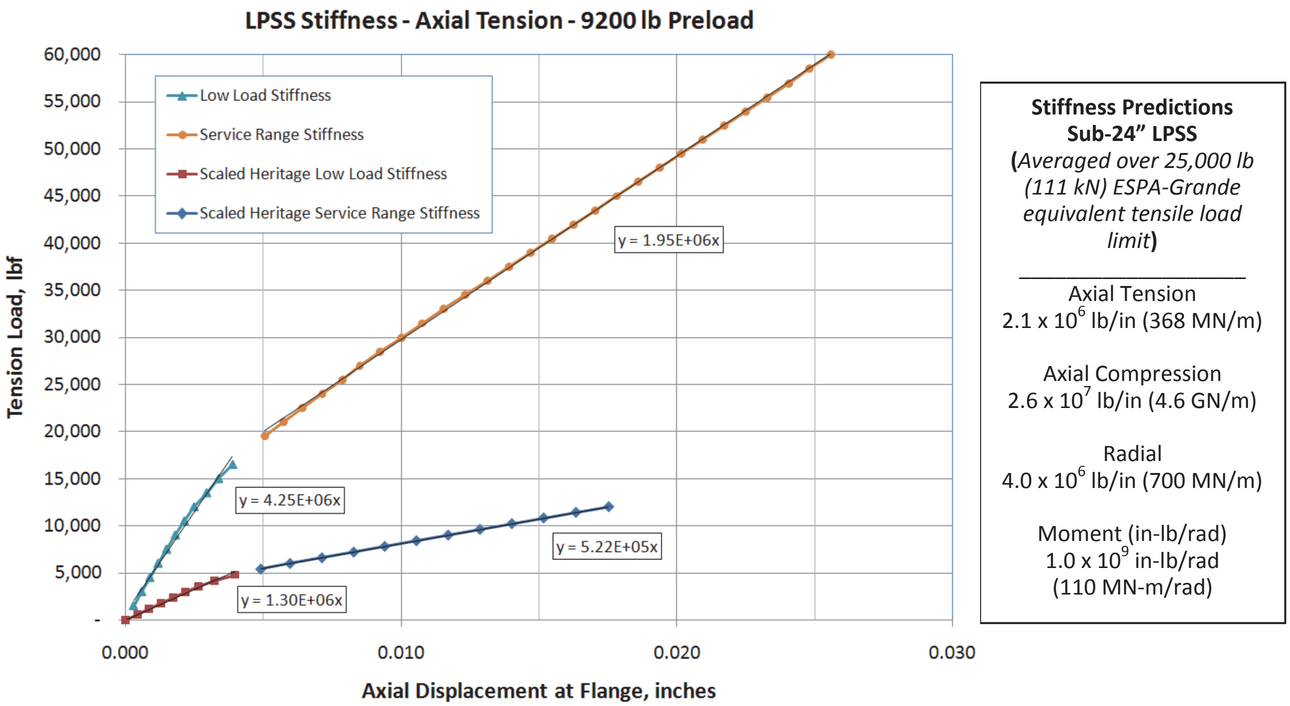


Figure 11: Results of ring profile revision on sub-24" predicted system stiffness

To maximize the load capability of the system (beyond the ESPA Grande requirements), the high-load CBOD (13,500 lb / 60 kN) was incorporated. Band thickness was increased to handle the larger CBOD loads, and it was found that the band stress associated with deforming the band from its free-state to the installed diameter was higher than expected, severely limiting the gains of a thicker band. The thickness of the aluminum was adjusted to maintain similar installed force with previous designs. To improve margin on gapping load, the ring half-angle was reduced to 15° from the 20° baseline.

The final band design (Aluminum 7075-T7351) uses a nominal 9200-lb (41-kN) band preload (10,000-lb (44-kN) proof load), with a yield stress margin of 0.05 and an ultimate stress margin of 0.22 (with SFy = 1.1 and SFu = 1.4, and a preload uncertainty factor of 1.1). The predicted stiffness and load capability for this size system is excellent even though only 75% of the CBOD load capability is used. The 9200-lb (41-kN) nominal band load in the sub-24 LPSS allows the CBOD to show >100% margin.

Analysis predicted local gapping of .0015" - .0035" (38 µm – 89 µm) at joint heel at the nominal 9,200-lb (41-kN) band tension, depending on the stiffness (or constraint) of the interfacing structure on each ring. Our FEM showed that this gap was an artifact of the hoop compression stresses of higher band tension on our ring geometry, with negligible effect on system stiffness. The load and location at which "true" gapping occurs were also predicted by FEM (see Figure 12). At an axial load of 67,500 lbf (300 kN) (about 2.7 times ESPA Grande equivalent axial loads, not shown on chart), a gap forms at the heel, such that only a small contact point at the toe remains, changing the system stiffness to ~ 1.27E+06 lbf/in (222 MN/m).

Summary / Conclusion

Using an understanding of the design features of a robust clampband system, we were able to scale a heritage design, reduce profile, and increase stiffness and load capability. Design and analysis of the sub-24 Low-Profile Separation System are complete, and engineering hardware has been assembled (see Figure 12). At the time of this writing, the sub-24" system is poised for development testing in early 2012 to confirm the analysis results and determine the true load capability.

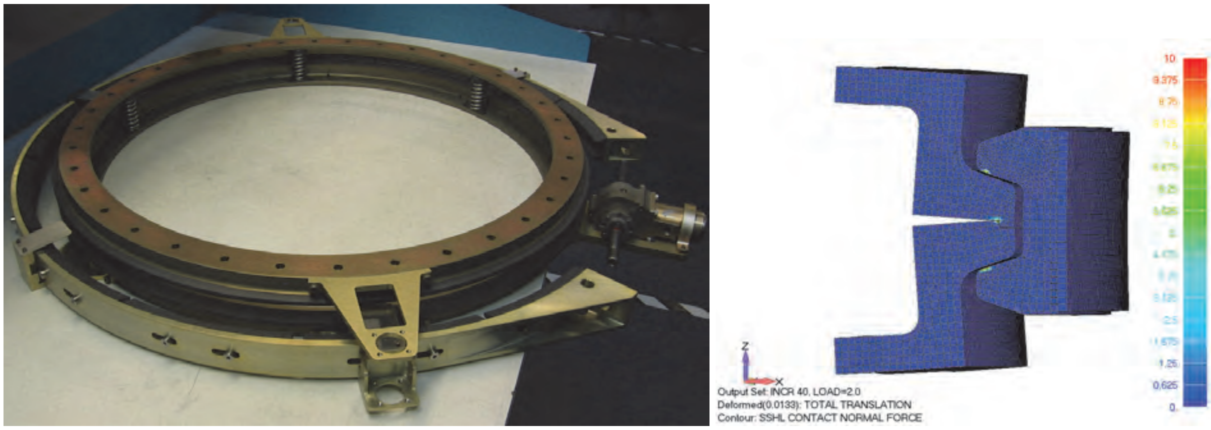


Figure 12: Engineering Development Unit of sub-24" (LPSS) and FEM

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