1995120848

N95- 27268

405425

22p.

# The Clementine Mechanisms

William Purdy* and Michael Hurley*


## Abstract

The Clementine spacecraft was developed under the "faster, better, cheaper" theme. The constraints of a low budget coupled with an unusually tight schedule forced many departures from the normal spacecraft development methods. This paper discusses technical lessons learned about several of the mechanisms on the Clementine spacecraft as well as managerial lessons learned for the entire mechanisms subsystem. A quick overview of the Clementine mission is included, the mission schedule and environment during the mechanisms releases and deployment are highlighted. This paper then describes the entire mechanisms subsystem. The design and test approach and key philosophies for a fast-track program are discussed during the description of the mechanisms subsystem.

The mechanism subsystem included a marman clamp separation system, a separation nut separation system, a solar panel deployment and pointing system, a high gain antenna feed deployment system, and two separate sensor cover systems. Each mechanism is briefly discussed. Additional technical discussion is given on the marman clamp design, the sensor cover designs, and the design and testing practices for systems driven by heated actuators (specifically paraffin actuators and frangibolts). All of the other mechanisms were of conventional designs and will receive less emphasis. Lessons learned are discussed throughout the paper as they applied to the systems being discussed. Since there is information on many different systems, this paper is organized so that information on a particular topic can be quickly referenced.


## Clementine Mission With Mechanism Activities Inserted

The Clementine satellite (Figure 1) was designed to map the lunar surface in many wavelengths and to do an asteroid flyby. The satellite was launched January 25, 1994 on a Titan IIG rocket from Vandenburg Air Force Base. The Clementine space vehicle was separated from the Titan approximately 45 minutes after launch via a pyro marman clamp release and four balanced kickoff springs. Clementine was in a Low Earth Orbit (LEO) for approximately 1 week. During the week in LEO, the star tracker covers were opened several times and the spacecraft was oriented based upon the star tracker readings. Also, all the satellite systems went through health checks to verify their proper operation. The satellite was then oriented and spun-up to approximately 60 rpm for the solid rocket burn that began the transfer orbits to the moon. The solid rocket burn took place on February 2, 1994, this burn put Clementine in a transfer orbit approximately 60-80% of the way to the lunar orbit.

After the solid rocket burn the spacecraft was spun-down and the solar panels were deployed shortly afterwards. Approximately 16 hours after the solid burn, the main sensor cover was opened and the interstage and solid rocket case were separated

---

* Naval Research Lab, Washington, D. C.

from the satellite (Figure 1) via a pyro release of eight separation nuts and four balanced kickoff springs. The high gain antenna feed was then deployed. A 490 N (110 lb) liquid propellant thruster was used to make the remaining burns for lunar transfer orbit changes and lunar insertion. Clementine spent 26 days transferring from LEO to a lunar orbit, the final lunar orbit was achieved on Feb 21, 1994. Note the staging of the solid rocket and the 490 N thruster significantly increased the allowable satellite weight that could attain a lunar orbit. The next seventy days were spent mapping the moon, 100% mapping of the surface was achieved. Clementine was scheduled to begin the transfer orbits for the asteroid Geographous in early May but the mission was ended due to a software failure.

Some significant miscellaneous information about Clementine. The total space vehicle weight was 16,020 N (3600 lb), with the following breakdown: 11,125 N (2500 lb) was the solid rocket, 445 N (100 lb) was the interstage, 2,225 N (500 lb) was propulsion fuel and oxidizer, and 2225 N (500 lb) was the dry satellite. The entire satellite was on a very tight weight budget as a 2225 N (500 lb) spacecraft had to perform a very complex mission. The five sensors on the satellite were a Ultra Violet/Visible Spectrum camera, a Long Wave Infrared camera, a Near Infrared camera, a LIDAR laser ranging system, and two star tracker cameras. The satellite design began in April, 1993, giving only 22 months for design, manufacturing, integration, test, and launch. The satellite was sponsored by Ballistic Missile Defense Organization, the sensors were supplied by Lawerence Livermore National Laboratory, and the satellite, integration, and mission operations were done by the Naval Research Laboratory.

## List Of The Clementine Mechanisms Systems

1. Marman clamp and kickoff spring system at Titan II/spacecraft interface (Figure 1).
2. Separation nuts and kickoff spring system at satellite/interstage interface (Figure 1).
3. Solar array release, deploy, and pointing system (Figure 2). This system used frangibolts for array release, springs for deployment, and a stepper motor for pointing.
4. High gain antenna feed release and deployment system (Figure 3).
5. Main sensor cover system (Figure 4). This system was driven by a paraffin actuator.
6. Two star tracker covers systems (Figure 4). These systems used paraffin actuators.

## Mechanisms System Approach And Key Philosophies

To begin any project all the necessary mechanism subsystems must be identified and their purpose in the mission completely understood. The following Clementine examples illustrate this: 1) Is the purpose of the sensor cover to keep out debris, light, or both?, 2) What data is the high gain antenna needed for and when? Next the major design requirements must be understood. **It is extremely important to understand the purpose of each mechanism and to be involved in the iterative process of defining the requirements** to assure that unnecessary and difficult requirements are not placed on the design since there is not enough time to meet these unnecessary requirements. For Clementine, high reliability was a requirement, redundancy was not - redundancy was determined on a system-by-system basis and was incorporated only where it significantly improved the design. Besides avoiding unnecessary requirements, being involved in this process of determining the design requirements improves your design as your understanding of the system is much more complete.

The long-lead items, such as actuators and motors, must be sized and ordered very early in the design process. On our 22-month schedule, mechanisms that had lead times up to 15 months, including contract preparation, had to be ordered just three months into the design process, at which time the spacecraft design was still very immature. **Ordering these long-lead components with high functional margins was critical to our success.** Having high margins on critical, long-lead items allows the mechanisms to meet evolving requirements without needing to change critical components, thus allowing the program to continue on schedule. Further, high margins on driving components gives a robust design that greatly improves reliability. Specifying high functional margins enabled us to get through the development process without any major redesigns, which was crucial to meeting our schedule. As a rule of thumb, all actuators and motors were designed with a 3:1 ratio of driving to known opposing forces. It is not acceptable for evolving requirements or the discovery of unanticipated loads during testing to stop the entire program.

Understanding which components to put high margins on is extremely important. Long lead time items and/or driving components such as actuators are a must. Since these components only make up a small part of the system, the weight penalty is very small, yet the increased reliability and level of tolerance for changes or problems later is greatly improved. For example, to use a solar array drive motor "one size larger than necessary," or in our case, a motor that gave a 3 to 1 driving torque margin, only increased the motor weight by 0.45 kg each. However, had we discovered that we needed a larger motor (i.e., if array size increased) the entire program would have missed the launch window as the motors took 15 months to design, build, and test. This does not mean that all the components in a subsystem have high margins, on a weight-critical program this is not possible. Weight savings is obtained primarily from reducing the margins on the structural components of the mechanism. These structural components can be verified by testing to the maximum required load and, if they fail, can be often be modified as quickly as 1-5 weeks if necessary. The reliability of these components was kept high by testing them to qualification-level loads.

**Early prototype testing proved to be extremely valuable.** As soon as a design concept firmed up, a prototype was made and tested or more appropriately played with. Early prototype testing, even with very crude prototypes, proved very valuable. This testing often uncovered problems easier to recognize in hardware than in 2 or 3 dimensional drawings. Additionally, many subtleties of the mechanisms quickly became apparent in prototypes. The prototypes always greatly improved our understanding of how the system really works. The problem catching and understanding gained from the cheap, crude prototypes greatly improved the final designs and prevented many costly and schedule impacting mistakes.

**The reliability of the mechanisms relied on 1) good engineering understanding of the system requirements and environment, and 2) on**

**rigorous qualification and acceptance testing.** Design analysis included hand calculations, Roark and Young stress calculations, and TK Solver math models of the various systems. Finite element modeling was neither necessary nor useful except for the analysis of the marman clamp rings. A rigorous testing program was a major reason for the success of the mechanisms. The goal of a thorough testing program is to uncover all of the problems, and potential problems, on the ground where they can be solved. We performed our qualification testing in an informal manner although we tested the mechanisms very rigorously, often testing them to their functional limits. Our acceptance tests were taken to protoflight levels and included lengthy burn-in testing. The acceptance testing uncovered many minor and a few major flaws which we repaired before delivering the mechanisms for spacecraft integration. This rigorous testing provided us with a very high confidence in the reliability of our mechanisms upon delivery for spacecraft integration.

Furthermore, thorough testing gives a deep understudying of how the system works and what its subtleties are. This understanding can be invaluable during flight operations. For example, during a one-time lunar pass involving rapid temperature changes, the main sensor cover switch opened for ten minutes and then closed. Fortunately, during thermal testing we had found that the cover would bow causing the microswitch to open during periods when one side of the door was heated rapidly. This warping was due to the temperature difference between the outside and inside door surfaces that the poorly conducting door would temporarily support. Having seen this behavior in testing allowed us to quickly explain this unusual telemetry and kept us from disrupting the mission.

## A Note On Primary Satellite Structure For Weight Critical Programs

It is important to realize on weight-critical programs that the loads requirements passed down from the launch vehicle are typically the results of a **3 sigma** situation, in other words results are inflated so that the actual launch loads will be less than predicted loads **99.7%** of the time. Further, combinations of loads that do not even occur at the same time are often used as design load case. The mechanical system manager of the satellite should truly understand the assumptions and method used to derive the design load cases to assure that an impossible situation is not becoming a driving design requirement. See the marman clamp section for more information.

## Marman Clamp Separation System

The marman clamp separation system is shown in Figure 5. This system was used between the Titan II and the Clementine space vehicle (Figure 1). The clamp release was done via two explosive bolts, 180 degrees apart for redundancy. Balanced kick-off springs were used to separate the two vehicles and tension springs were used to restrain the clamp after release. The clamp design was very weight efficient totaling only 36 N (8 lb) and supporting a maximum line load of 32,280 $^{N}/_{m}$ (184 $^{lb}/_{in}$) over a 1.067 m (42 in) diameter, which corresponds to a launch vehicle combined load case of +3g tensile, ±2.2g lateral. The clamp was tested to 125% of this load case. The clamp preload was 16,020 N (3600 lb). The highlightable features of this clamp include the lightweight design, the joint design, and finally the trunnion design. In developing this design, several other marman clamp designs were researched, the best design features were taken from these clamps and implemented on the Clementine clamp. All of the following comparisons are made against the designs that were researched.

**The number one way to reduce weight is to choose a reasonable preload.** First, the maximum line load around the ring is calculated based on the worst launch vehicle load case, which is conservative (see the next paragraph). Clamp preloads are sized to prevent gapping of the joint when the maximum predicted line load is applied all around the joint, a case the clamp never experiences. Additionally the required preload to prevent this gapping is typically greatly padded. This padded preload number is then increased again to allow for preloading errors. The above is a perfect example of how margin is added on top of margin producing tremendously conservative design loads. The preload that is determined as necessary to prevent gapping is the driver for sizing all of the clamp components and often the rings also. So for weight savings, it is very important to choose a reasonable preload. However, unless extremely weight critical, leave a ±25% tolerance on your preload to avoid the need for a very tedious installation procedure.

**This next paragraph may sound like an exaggeration to make a point, but it is an actual example of how design loads can get out of control if not well understood and monitored.** The original load case that the marman clamp was designed to was a +3g axial tension load superimposed on a ±3.5 g lateral load. Each of these loads independently (+3g axial and ±3.5 g lateral) is a 3 sigma (0.3% probability of happening) launch vehicle load case that in reality do not even occur at the same time. In addition, a finite element model was made of the interstage and the recommendation made that a stress concentration factor of 1.5 be applied to the maximum line load as well as an 8% model uncertainty factor. Finally, the clamp was going to be tested to 125% of the design load. Had we actually used these loads, the marman clamp would have been designed to handle loads 4 times the maximum expected loads (99.7% probable loads) and 8 times the actual predicted flight loads. The clamp was not designed to these loads. The clamp was designed to a +3g tension combined with a ±2.2g lateral load case and tested to 125% of this load case.

Once reasonable design loads and a preload have been selected, the marman clamp can be significantly lightweighted by reducing the high margins often put on the strap and the shoes, each of which account for approximately 40% of the clamp weight. Specifically, the strap margins are typically in the 3-6 range for ultimate strength; I found 2.5 to work fine (Note: all my margins are defined as Material Strength / Stress). Also, try to hold the same strap margin all the way around including rivet sections, around the trunnions, etc. It does no good to have one section of the strap stronger than the rest. The strap material was 301 corrosion resistant 1/2 hard steel. The marman clamp shoes were designed with a margin on yield of 3.0 which is lower than the typical margins of 4-8. Since the shoe load is much higher for the shoes located under the trunnions, significant weight can be saved by designing and optimizing an aluminum 7075-T7 shoe at the non-trunnion locations and using titanium shoes under the trunnions (Figure 5). Our surface preparation involved hard anodizing the aluminum shoes and rings, Tiodizing the titanium shoes, and dry lubing all the shoes with Tiolube 460. We found no galling and always had clean separation between the shoes and the rings. Finally, **do NOT reduce the margin on your separation bolt below 2 times the nominal preload** and keep it even larger if you anticipate the satellite weight may increase significantly. Reducing the margin on the bolt saves little weight and is very high risk in that it is a long lead item that is very difficult to change. Also, there is a significant increase in bolt tension as load is applied to the joint, this increase in bolt load is directly related to the joint design which is discussed in the next section. Bolt load should only increase 10-15% with a good joint design. The Clementine separation bolt only had a margin of 1.5 over the nominal preload. While the design did work, the low margin caused much suffering, requiring tremendous effort in developing a careful preload procedure and two static load tests.

As with any preloaded joint, the joint design greatly affects how the clamp carries applied loads. NRL has traditionally used a joint that is gapped between the outer surfaces of the rings so that the rings rest on the inner surfaces (Figure #6A). **Locating the gap between the outer rings gives a much stiffer, linear joint which reduces the applied load that the clamp must carry as well as increasing the satellite's natural frequency on the launch vehicle.** The benefit comes from providing a load path that does not have to take a circuitous route around the outer rings. While the analogy is not perfect, comparing a marman clamp joint in a pure tension load case to a bolted joint in a pure tension load case gives tremendous insight (Figure #6B). The well-designed bolted joint has a low bolt to joint stiffness ratio, around 1 to 5, so that as tension is applied and the bolt is stretched, the joint relieves by approximately 80% of the applied load so that the bolt only has to carry 20% of the applied load. The same is true for the marman clamp joint; the stiffer the joint and the better the load path, the more the joint relieves when it is in tension and therefore clamp carries less of the applied load. Looking at the cross-sections and load paths shown in Figures 6A and 6C, it can easily be seen that the joint with the gap between the outside ring surfaces (6A) provides a much better load path and is much stiffer.

In fact, the first design iteration had a gap of only 0.254 mm (0.010 inch) between the outside surfaces, and this gap closed at approximately 60% of full preload as the lower ring rolled. The resulting joint contacted at both the outer and inner surfaces, but the inner surfaces were only preloaded to approximately 60% instead of 100% of the full preload. During static loads testing, this joint proved to be non-linear causing the clamp to carry a large percentage of the applied load which it was not designed for. The problem was corrected by increasing the gap size between the outer ring surfaces so that these surfaces did not contact when preloaded, thus preloaded the inner surfaces to the full preload. **The improved joint stiffness is easily seen in the results of the static loads tests. Figure 7 shows the increase in the marman clamp strap tension as a function of applied tensile load across the joint.** The graph shows the that the joint that was properly gapped and preloaded is linear and stiff, minimizing the percentage of applied load that the marman clamp needs to carry. On the other hand, the joint with the gap that closed at 60% of the preload is nonlinear and forces the marman clamp to carry a much larger percentage of the applied load. As could be anticipated, the biggest increase in slope (tension in strap / applied tensile load) occurs at approximately 60% (20,000 lb on Figure 7) of the applied load that the preload was designed to support.

Traditionally, the marman clamp and separation bolt designs do not account for any increase in the marman clamp loads once preloaded, however, extremely high design margins are put on all the components. This maybe explained by the fact that marman clamp joints are usually gapped between the inner ring surfaces (Figure 6C), making them highly nonlinear and therefore very difficult to predict and design. **Designing a stiffer joint greatly improves the joint behavior and allows one to confidently reduce the extremely high design margins.**

**The increased natural frequency of the satellite that results from the improved joint design (Figure 6A) should not be understated.** In fact, the stiffer joint was originally designed because a soft marman clamp joint caused the satellite's first mode on the launch vehicle to be below the launch vehicle requirements. The program that made this design improvement was in no way weight critical, the change was driven completely by the system's natural frequency.

Trunnions have received a bad reputation by some as there was once a problem with trunnions putting the separation bolt in bending and causing it to fail. **Trunnions make for a very compact and clean design and do not put the bolt in bending if properly designed.** The correct and incorrect trunnion designs are shown in Figure 8. The free-body diagrams show that the bolt load must go into the trunnion on the bolt side of the trunnion's center to give a self-correcting system. If the bolt load goes into the trunnion on the far side of the trunnion's center, the system is unstable in the sense that any misalignment will continue to worsen. Increasing misalignment in the unstable design will only be prevented by increased bolt bending that eventually either reaches equilibrium or breaks the bolt.

**Finally, some thoughts on when and what of the above to implement on a marman clamp design.** The proper joint design that has the gap between the outer ring surfaces (Figure 6A) should always be done, it is a far better design. Also, if trunnions are used, the load should be put into the trunnions on the bolt side of the trunnions' centers (Figure 8) so that bolt bending is prevented. The decisions that are much more program specific include what to use for design loads, whether to trim margins, what preload to select, etc. If your program is not weight critical, it is not worth fighting launch vehicle design load requirements. Launch vehicle design loads are a major interface issue, so simply designing to the worst load case makes things go much smoother, however, understanding the level of conservatism in the load cases is still important. Also, designing the clamp to accept a large tolerance on the preload, ±25% or more, will greatly simplify your installation procedure. A good installation procedure must always be developed, however, if the preload must be obtained very precisely (±10%). Developing this procedure becomes very time consuming and tedious. While a large error tolerance on the preload is desirable, a high nominal preload may not be desirable. Often high nominal preloads, in terms of a preload that gives high margins on gapping, are mistakenly thought to be conservative. The whole reason for designing preloads to prevent gapping is to maintain the joint stiffness, however, high clamp preloads often cause serious ring rolling problems which greatly reduce joint stiffness. Preloads should be sized to prevent gapping for whatever the design load case is, but excessively padding the nominal preload should be avoided. Finally, the design of the rings must be done in conjunction with the marman clamp design to assure both a good joint design and that the rings are stiff enough to prevent ring rolling problems.

## Separation Nut Separation System

The separation system between the Clementine satellite and the interstage (Figure 1) was done with eight separation nuts and four kickoff springs. A 9.5 mm (3/8 in) separation nut was located between each satellite longeron and the interstage ring. The preload of each nut was 15,575 N (3500 lb). The eight separation nuts provided far more preload than necessary, but provided a good load path and were in stock at NRL. The separation velocity between the vehicle was 0.458 $^{m}/_{s}$ (1.5 $^{ft}/_{s}$) and was achieved via balanced compression springs. Each kickoff spring was in its own canister and was balanced in terms of energy and preload prior to being installed on the spacecraft.

## Solar Array Deployment and Pointing Mechanisms

The Clementine spacecraft had two solar array wings, each about 1.3 m by 1.3 m (4 ft by 4 ft) and weighting about 60 N (14 lb) per wing (Figure 2). Each of these array wings had four folding hinges and two release joints. The arrays were pointed at the sun by stepper motors with harmonic drives. The hinges were conventional designs using vespel bushings, stainless steel hinge pins and latch pins. The release joints used the Frangibolt non-explosive release mechanisms. Clementine was the first flight for the Frangibolt, which is produced by TiNi Alloy Inc. in San Leandro, Ca.

While the solar array mechanisms were mostly conventional, we learned several lessons in developing them under the tight schedule. We built one array wing without solar cells to be devoted entirely to mechanisms testing which served us very well in meeting our development schedule. **We performed a large amount of testing to ensure that impact loads were acceptable so that we could keep our functional margins as high as possible.** We found that the impact loads were not excessive because we built a fair amount of compliance into the arm between the array and stepper motor and because of structural damping during impact.

Isogrid structural composite was the original array substrate early on in the program for a variety of reasons. The isogrid uses a flat panel with triangular reinforcing ribs on the backside and can provide stud mounting locations at each node between triangles. All structural attachments must be made at these nodes which severely limits the design flexibility of the hinges and attachments as they can only have triangular bolt patterns of a given size. It was surprising how much of our design was affected by the isogrid pattern. For example, the triangular shape and size of the frangibolt mounting plate caused us to change the shape of the sensor cover door, which caused us to change the layout of the main sensor bench. Further, all isogrid attachment points must be identified before the isogrid is laid-up, thus making the isogrid very difficult to adapt to almost any design change. Honeycomb panels with aluminum core and graphite facesheets were finally used because in addition to the above problems the isogrid was not capable of being manufactured to the flatness required for a solar panel.

**The first usage of the frangibolt went quite smoothly since it was developed on the ARTS program (Presented in 28$^{th}$ AMS).** Integrating the frangibolt into a spacecraft forced us to learn how to use it properly. Our first issue was that the arrays were used to generate power while they remained stowed for the week the spacecraft was in earth orbit. With the sun on the solar panels they stabilized at 100°C. The frangibolt actuators had to be kept below 70°C to keep them from actuating and releasing the array prematurely. This thermal control was accomplished mounting the actuator on the spacecraft side of the interface using an aluminum plate to heat sink the actuator to the relatively cool spacecraft. A poorly conducting titanium interface plate on the solar panel was used to block the heat flow from the hot panel. This scheme kept the actuator at 44°C while the solar panel was 100°C and the spacecraft was 30°C.

The major lesson we learned with the frangibolts is that they should not be driven with a widely fluctuating bus voltage. We learned this lesson for all heat actuated mechanisms and it is discussed later in the paper. The last thing we learned about the frangibolts was the importance of following a good installation procedure. It must be a point of discipline to ensure that the actuator has been compressed before it is installed and that a notched bolt and the proper hardened nuts and washers are used.

### High Gain Antenna Deployment System

The Clementine high gain antenna system is shown in Figure 3. This antenna system was very simple in design. The driver for deploying the antenna feed was two pairs of carpenter springs. The antenna feed was held in the stowed position via a preloaded cradle. A paraffin actuator was used to release the feed by driving a structure that pulled a pin. Once the pin was pulled kickoff springs under the cradle gave the system a large kickoff torque in a region where the carpenter springs are relatively weak.

The carpenter springs were made out of SAE 1095 strip steel, hardened and tempered. These springs were simply 2.54 cm x 0.0114 cm (1.0 in x 0.0045 in) tape measures purchased directly from the factory with no paint or markings; the cost was $25 per hundred feet. During the year prior to flight, several materials for carpenter springs were looked at, but the more we played with our tape measure prototype, the more obvious it became that they were fine for the job. The only disadvantage to the SAE 1095 is that it is prone to corrosion. On Clementine, this problem was minimal as the total time from flight assembly of the antenna feed to launch was 4 months. The springs were, however, lubricated with Braycote 601 to help resist corrosion and frequently inspected. On future programs, we would use Elgiloy, a corrosion resistant alloy with very high yield strength that is not prone to corrosion. Also, Elgiloy is manufactured in strip form and can be specified to have rounded edges which is important as the edges are under very high stress.

The antenna system was tested in a similar way as the sensor cover systems as all these systems were driven by paraffin actuators. This testing is discussed in the next section. One on-orbit lesson learned, which goes for all paraffins, is fight to get a temperature telemetry point for each paraffin actuator. **This temperature telemetry is invaluable for predicting the behavior of the paraffins in different and sometimes unexpected on orbit environments.**

## Sensor Cover Systems

The Clementine satellite had a main sensor cover used over the primary suite of sensors and two star tracker covers. These covers are shown in Figure 4. The purpose of the sensor covers was to protect the optics from debris and solar radiation. Both types of covers employed a labyrinth seal to keep debris out of the optics areas. The covers were driven open with paraffin actuators, a latching mechanism was used to hold the covers open, and torsion springs drove the covers closed when the latch was released. The main sensor cover was designed and manufactured at Starsys Research in Bolder, Co. The main sensor cover was presented at the 28$^{th}$ AMS.

Often, neither the performance of a particular type of seal nor the sealing requirements are well quantified. In order to solve this problem, Starsys Research developed prototype seals and tested them by placing protoflight covers in a chamber, engulfing it in swirling flour using compressed air, and then quantifying the amount and size of the particles that got by the seal. Flour was chosen as the debris simulator because it has a range of particles from 0.51 mm (0.020 in) to less than 0.0254 mm (0.001 in) diameter. **This flour testing proved a very effective, as well as a cheap way, to evaluate various seal designs**. The protoflight labyrinth seal performed far better than expected. The good sealing performance of the labyrinth combined with the zero breakaway forces to open the cover and good design flexibility made the labyrinth seal the clear choice for our application. The flour testing was so effective that it was also used to evaluate the performance of the star tracker labyrinth seal. Figure 9 shows the before and after appearance of a prototype star tracker flour test.

**An often difficult requirement to obtain is the cleanliness requirement of the seal**. The sensor people often know they want their optics "clean" but do not have a good quantification of this. **The flour testing was invaluable for agreeing on a required seal cleanliness or performance level**. Since the flour test was a significant over-test due to the use of swirling, compressed air coupled with a large amount of flour, and since the seals only let 1 or 2 particles of 0.0254 mm (0.001 in) or less through, the seals were considered more than adequate and the entire issue was put to rest.

**Before testing any of the paraffin-driven systems, a baseline characterization of the system's performance was done.** This characterization involved characterizing the system's position and actuator temperature as a function of time. A performance characterization run was then done after every major test, such as random vibration and life cycling, and compared against the baseline performance. **This method of comparing the before and after performance was excellent for spotting and troubleshooting any cover**

**system degradation as well as for verifying that the cover was operating properly and ultimately ready for flight.** Figure 10 shows the baseline characterization curve for the main sensor cover. Note that for position and temperature to be repeatable functions of time, a consistent voltage and a consistent starting temperature for the paraffin actuator must be used. The starting temperature for the actuator should be well above the ambient temperature, we used 40°C, and is best achieved by heating the paraffin to a temperature above the desired starting temperature, turning the actuator off and letting it cool, and finally turning the actuator on when the desired starting temperature is reached. The baseline curves were all averages of three characterization runs. Finally, it is important to realize the baseline curves are specific to each individual paraffin actuator.

One important lessen we learned was that voltages above 34 volts should be avoided when using paraffin actuators. During the testing of the sensor cover one of the actuators had a heater circuit open due to overheating while being operated at 36 volts. This problem was exposed on the main sensor cover and may or may not be seen on other paraffin actuators because it is a heat transfer problem that resulted from a somewhat unique combination of factors. These factors included the main sensor cover paraffin actuator being designed for 150% higher power than typical paraffin actuators to increase stroke and maintain speed, the external load being relatively high, and finally being operated at 36 volts. Starsys does not recommend that the paraffin actuators be used at 36 volts, their testing as well as NRL's showed all such overheating problems could be eliminated by limiting the voltage to 34 volts.

**Heat-Actuated Mechanisms In General**

Clementine was NRL's first extensive use of heat-actuated mechanisms, specifically Frangibolts and paraffin actuators. As such, we learned much about integrating them into a spacecraft. Upon learning these lessons we were very happy with their characteristics and performance.

A voltage range of 24 to 36 volts is a brutal range for heat-driven devices leading to an extremely difficult heater design. The power is a function of the voltage squared which means that the heater must withstand two and a quarter times the power at 36 volts than it must withstand at 24 volts. Additionally, the heater must withstand applied mechanical stresses in both the Frangibolt and paraffin actuators. We feel that the spacecraft system design would be better served by reducing the heater supply voltage fluctuation to about ±2 to ±4 volts depending on the particular application. Note that limiting the operating voltage range greatly reduces the performance scatter of the devices making them much easier to characterize and predict on orbit.

Other lessons learned included discovering the need to protect the heat-driven mechanisms from accidentally being turned on during spacecraft integration and testing, specifically during software testing and debugging. Two flight star tracker paraffin actuators were destroyed because software accidentally turned them on for 8.5 hours, driving the covers against "remove before flight" hardware. Also we found that you have to be careful not to provide heat sinks to the actuators. For example, we once put a thermistor mounted on a small aluminum block on a paraffin actuator to measure its temperature. The aluminum block acted as a heat sink and prevented the paraffin actuator from reaching its full stroke. Finally, routing power through redundant microswitches that opened when the paraffin actuators reached the end of their travel and when the Frangibolt released the solar panels, provided good fault protection with a low impact to reliability.

## Conclusions

1. Understanding the purpose of each mechanism in the mission and being involved in defining the requirements is critical to assuring that unnecessary and difficult requirements are not placed on the design.

2. Ordering long lead components with high functional margins is critical on a fast track program. Further, high margins on driving components greatly improves a system's reliability and ability to adapt to changing requirements while costing little in terms of added weight.

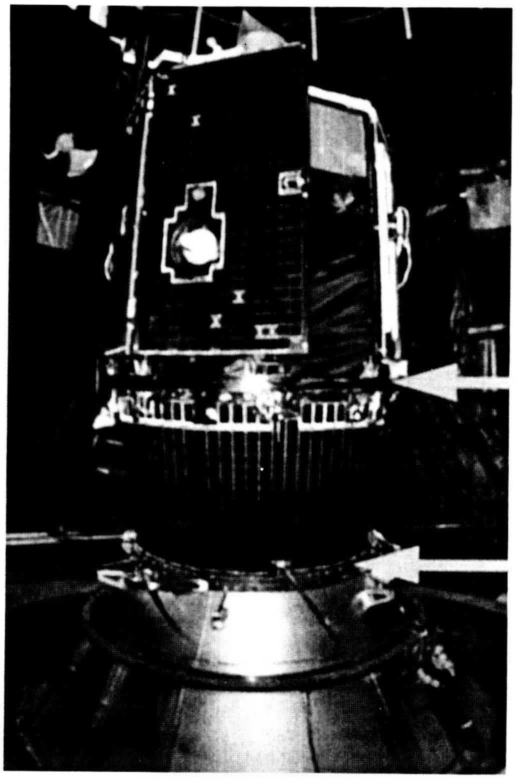
3. Early prototyping and rigorous developmental and acceptance testing was one key to the success of the Clementine mechanisms.

4. Marman clamp joint designs can be significantly improved by locating the gap in the rings properly (Figure 6A). The proper gap location stiffens the joint which reduces the percentage of applied load that the marman clamp must carry and increase the satellite's natural frequency on the launch vehicle.

5. Paraffin actuators proved to be excellent drivers for sensor covers and frangibolts worked well for solar panel deployment. Both paraffins and frangibolts have the advantages of being lightweight, compact, capable of repeatable use (that is the flight component can be tested and then flown), and neither have any safety issues.

6. Heat-actuated mechanisms should be powered by supply voltages varying by only ±2 to ±4 volts depending upon the particular application.

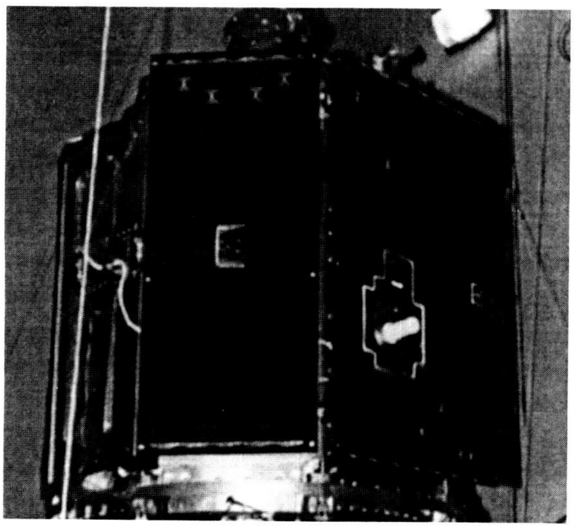
7. The labyrinth seals proved to be a good technique for moderate cleanliness requirements. They provided excellent mechanical reliability and very good cleanliness.



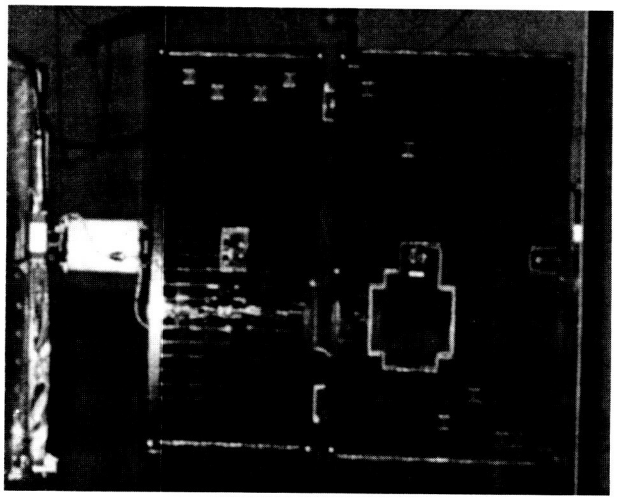
Satellite / Interstage
Separation Plane
(8 Separation Nuts)

Space Vehicle / Titan II
Separation Plane
(Marman Clamp)

Figure 1: Clementine Satellite On Titan II



Stowed Solar Panels (1 of 2 Wings Shown)

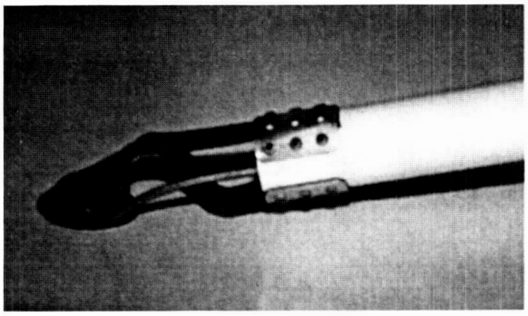


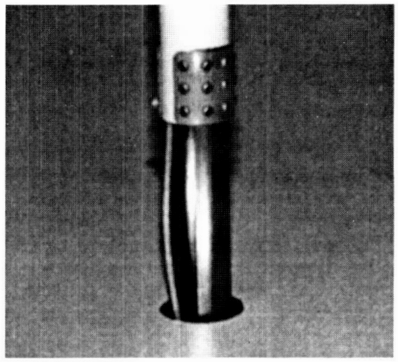
Deployed Solar Panels (1 of 2 Wings Shown)

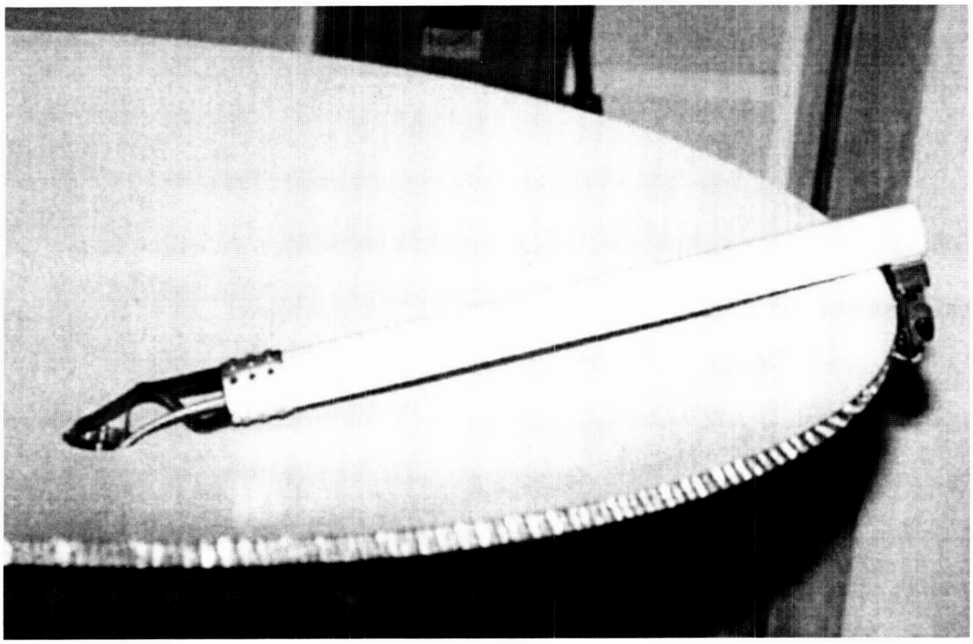
Figure 2: Stowed & Deployed Solar Panels

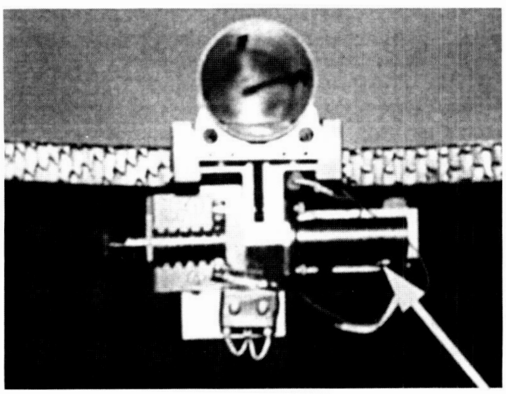
Carpenter Springs In Deployed Position

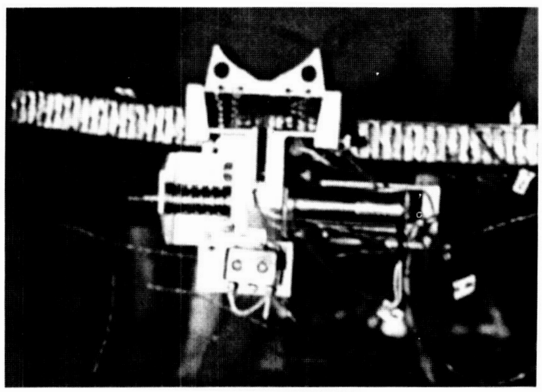
Carpenter Springs In Stowed Position











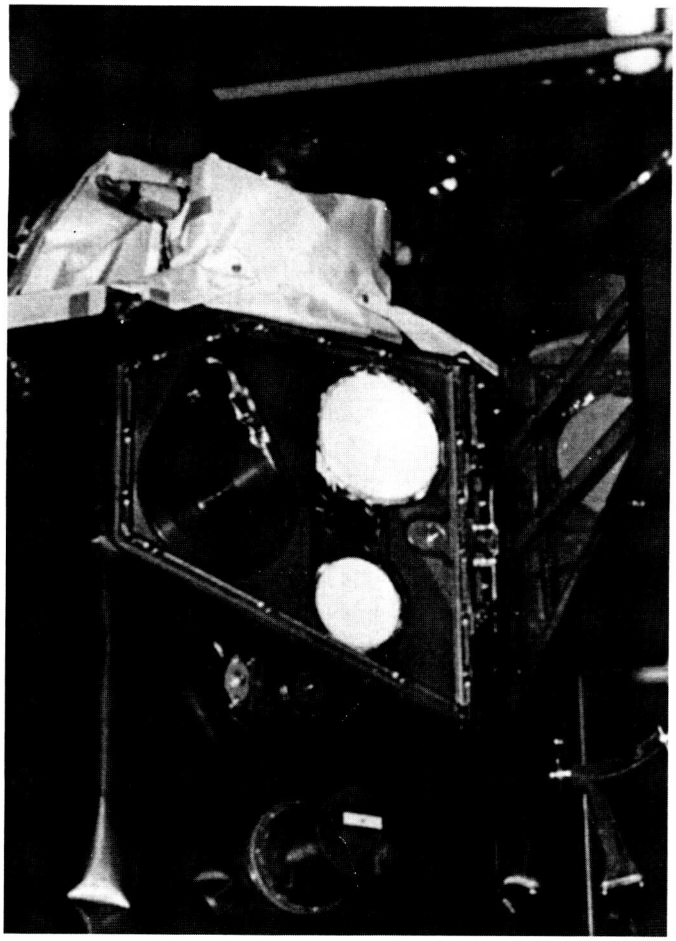
Feed & Cradle In Stowed Position

Feed & Cradle In Deployed Position

Paraffin Actuator

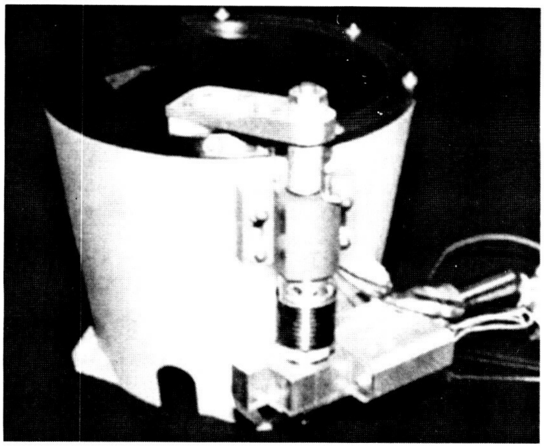
## Figure 3: Clementine High Gain Antenna System

## Sensors Included LWIR, NIR, LIDAR, & UV/Vis



Main Sensor Cover On Satellite

Both Systems Used Paraffin Actuators With A Latch To Hold The Covers Open & Springs To Drive Them Closed



One Of Two Star Tracker Covers

**Figure 4: Clementine Sensor Cover Systems**

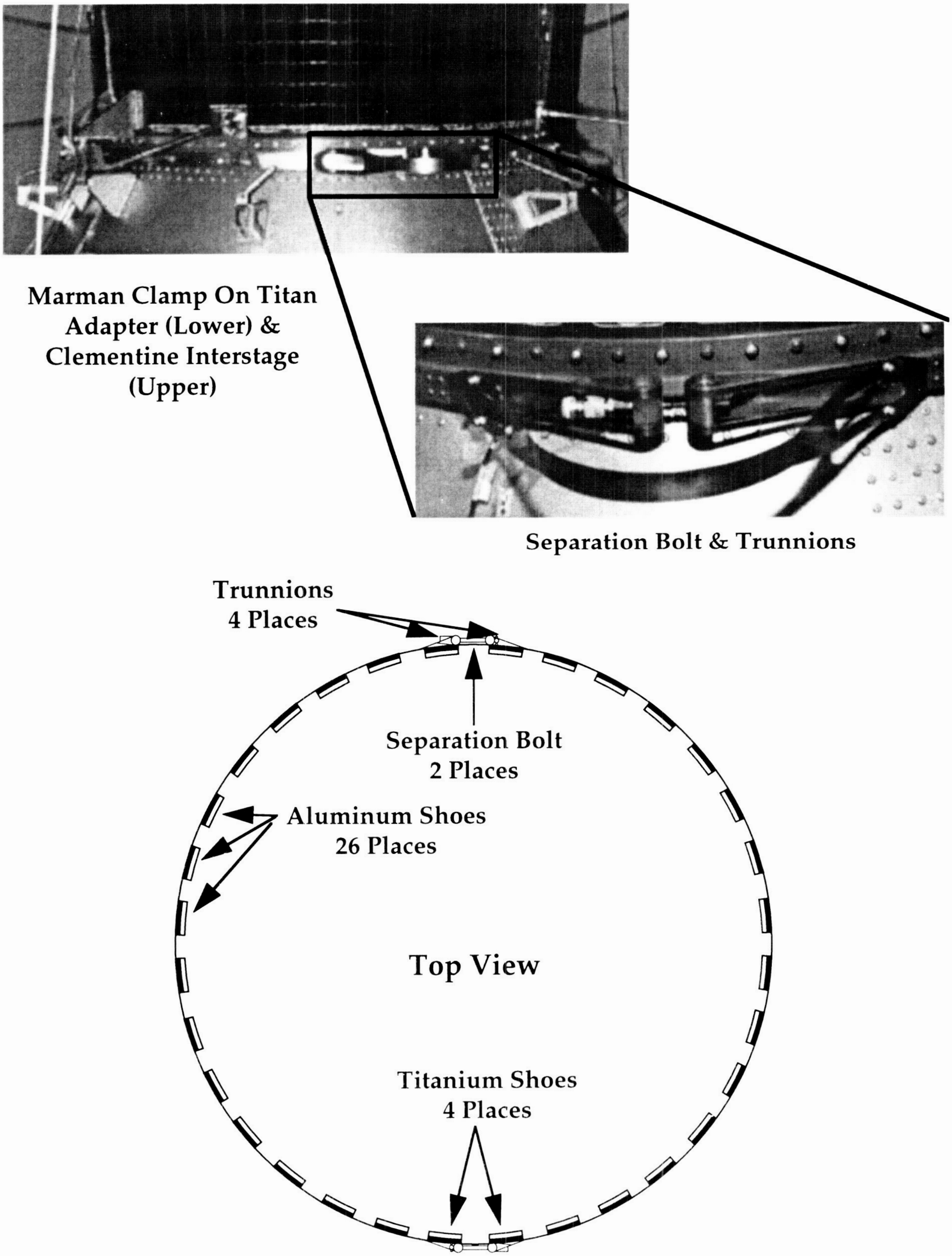



Figure 5: Marman Clamp System

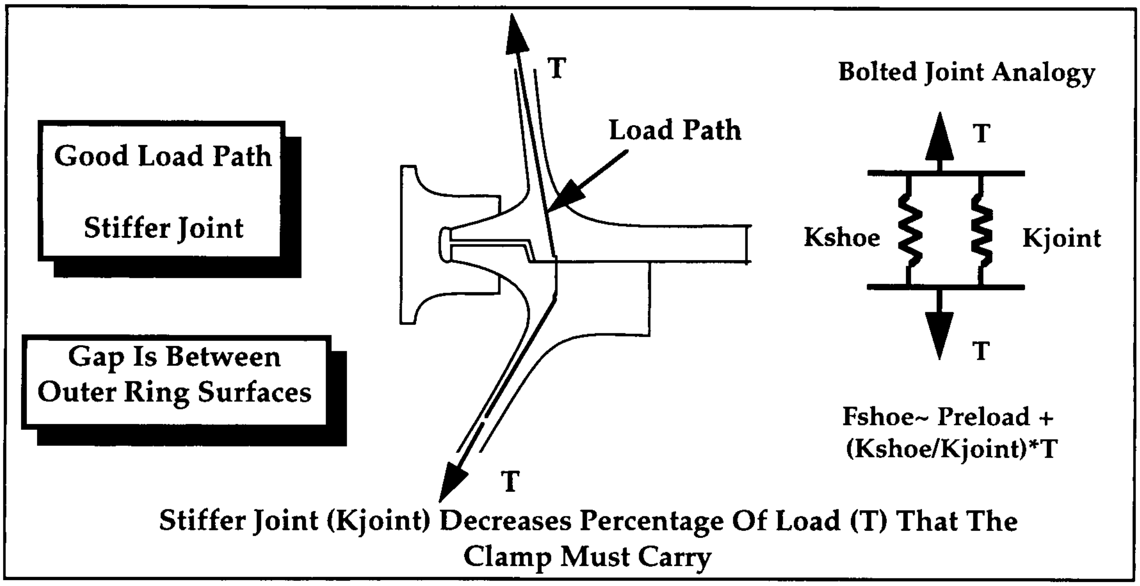



Figure 6 A: Improved Marman Clamp Joint Design

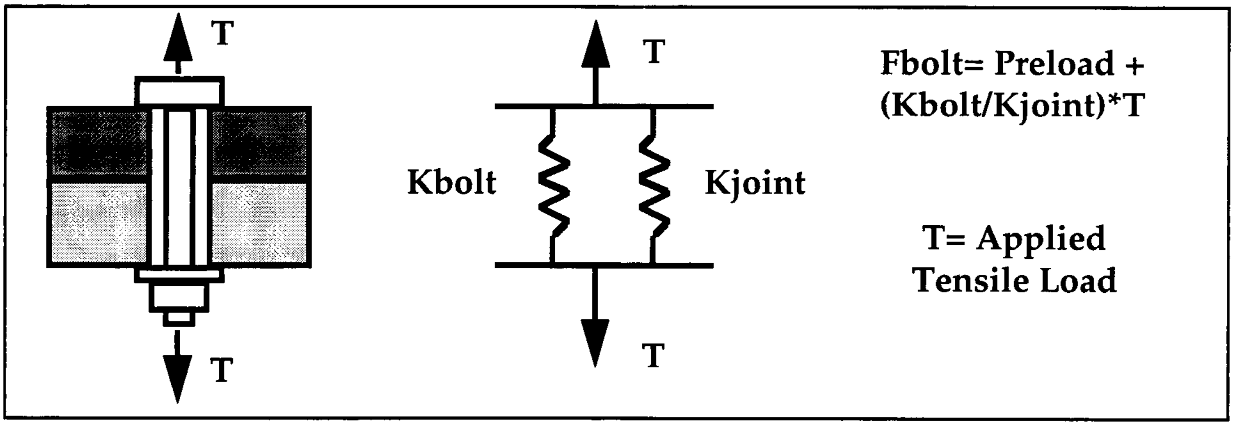



Figure 6 B: Bolted Joint In Tension

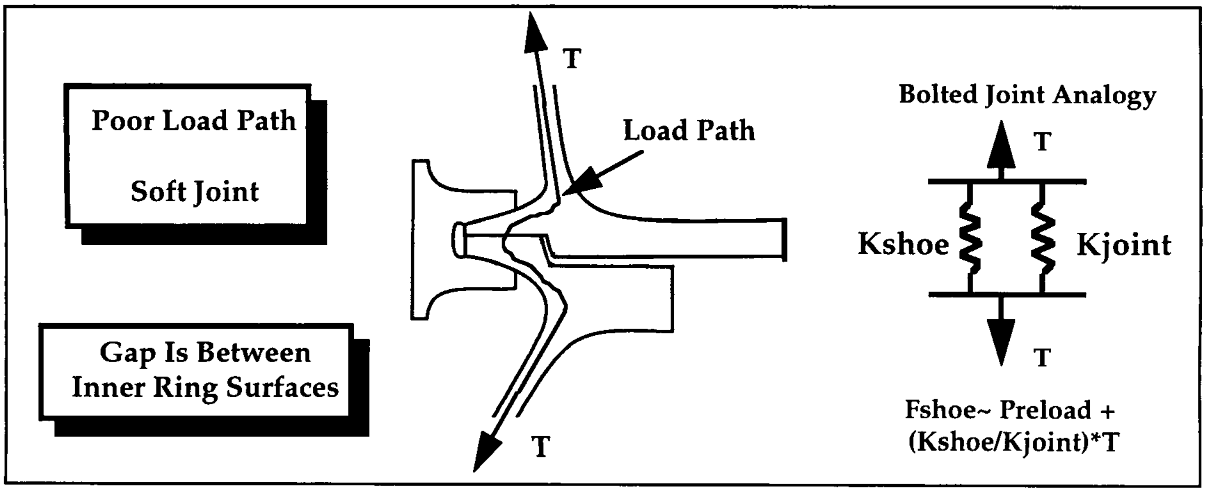



Figure 6 C: Traditional Marman Clamp Joint Design

Comparision Of Open Gap Vs Closed Gap In The Rings Affecting Strap Loads

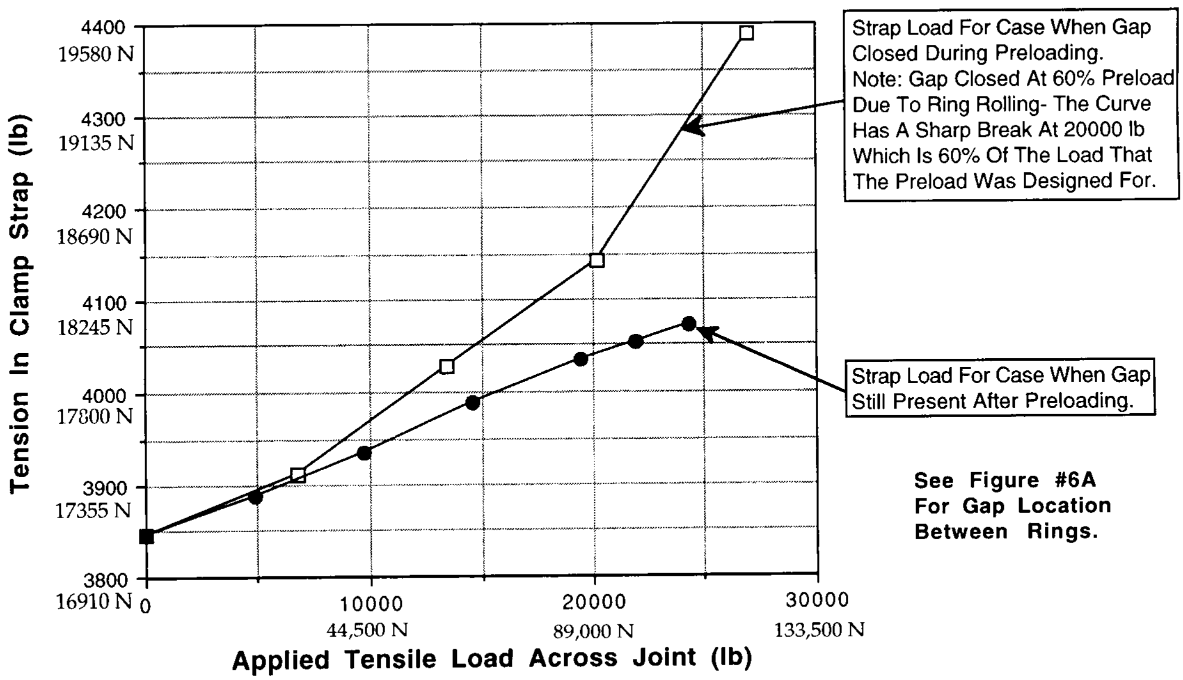



**Figure 7: Marman Clamp Strap Tension Vs Applied Tensile Load For An Open Gap and Closed Gap Joint**

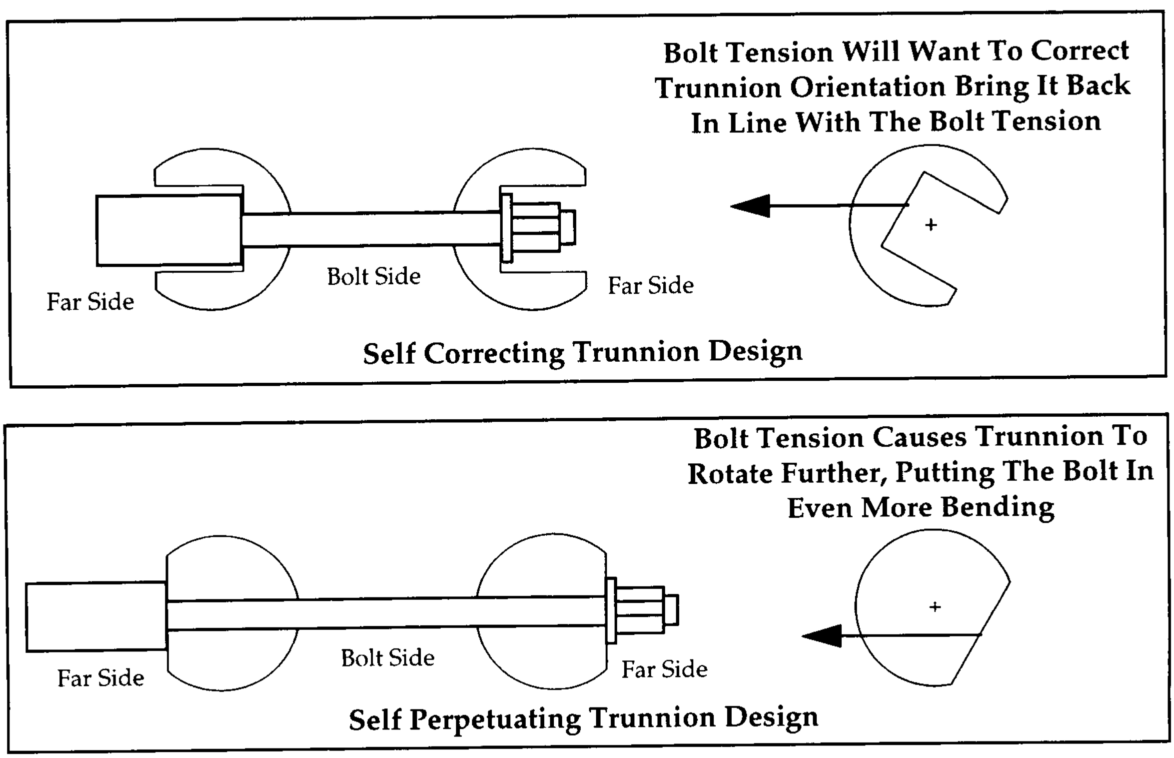
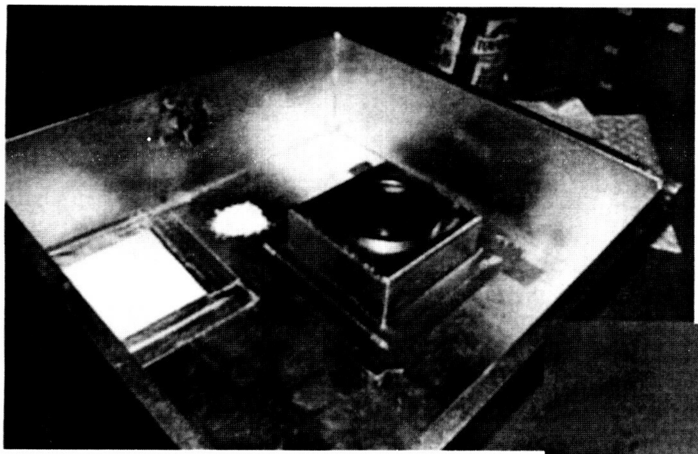



**Figure 8: Trunnion Design & Free Body Diagrams**

Prototype Seal After Testing



Setup Before Testing

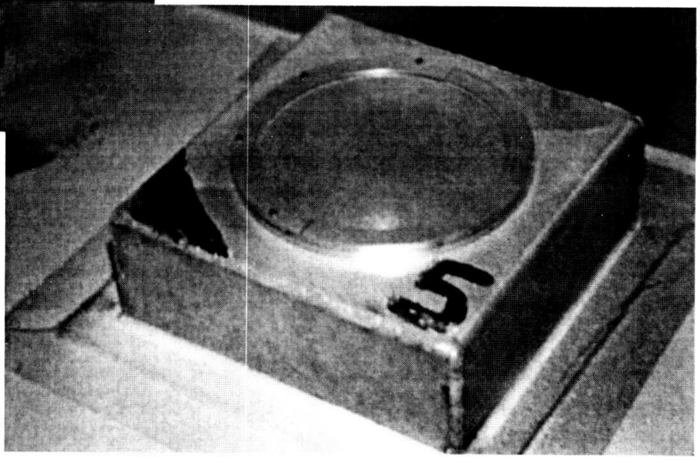


Figure 9: Prototype Star Tracker Cover Flour Testing

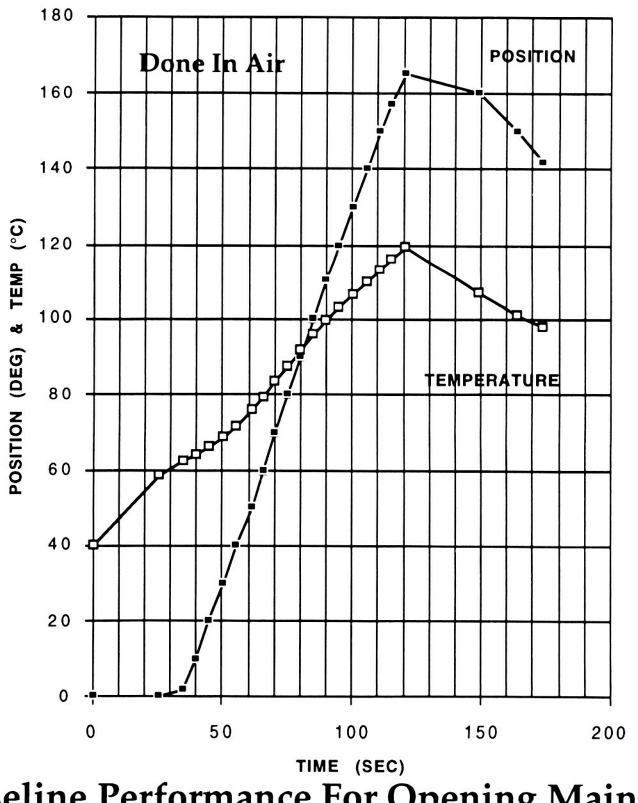



Figure 10: Baseline Performance For Opening Main Sensor Cover