The X-38 V-201 Fin Fold Actuation Mechanism

Christian Lupo*, Brandan Robertson* and George Gafka*

Abstract

The X-38 Vehicle 201 (V-201) is a space flight prototype lifting body vehicle that was designed to launch to orbit in the Space Shuttle orbiter payload bay. Although the project was cancelled in May 2003, many of the systems were nearly complete. This paper will describe the fin folding actuation mechanism flight subsystems and development units as well as lessons learned in the design, assembly, development testing, and qualification testing. The two vertical tail fins must be stowed (folded inboard) to allow the orbiter payload bay doors to close. The fin folding actuation mechanism is a remotely or extravehicular activity (EVA) actuated single fault tolerant system consisting of seven subsystems capable of repeatedly deploying or stowing the fins.

Introduction

The X-38 Project consisted of multiple unmanned drop test vehicles of various scales and one full-scale unmanned space flight proto-flight Vehicle 201. The vehicle shape, derived from the X-24, is a lifting body that lands via a parafoil and skids. The project's purpose was to perform the development work for an operational International Space Station Crew Return Vehicle. In order to launch V-201 in the shuttle payload bay, the two vertical fins must be folded inboard to allow the payload bay doors to close.

Figure 1. V-201 (Fins Deployed) on Mobile Transport Fixture after Static Testing

Fin Fold Actuation Mechanism Overview

The fin folding actuation mechanism is a complex system consisting of seven subsystems housed in a structure that narrows to 13.97 cm (5.5 in.) at the highest load point, the mid hinge. The system must react greater than 11,300 N•m (100,000 in•lbf) at the fold line from aerodynamic loading and seal the fold line for re-entry from orbit. The mechanism is required to be a remotely actuated single fault tolerant system operated via laptop in the orbiter cabin. The mechanism is also required to include an interface for unplanned EVA actuation of the fin. Motors are used because hydraulic power is not available on V-201. The upper fin structure also served as the housing for the electromechanical actuators for the rudder.

* NASA Johnson Space Center, Houston, TX

This added additional challenges to the design due to the necessity to accommodate large power cables across the fold line and around the fin drivetrain. The rudder drivetrain will not be discussed in this paper. The fins are stowed (folded and locked) for launch and deployed (unfolded and locked) for free flight. The system is capable of being re-stowed for return in the shuttle and then re-deployed if required.

The fins are folded via two four-bar linkages that toggle over-center in both the stowed and deployed positions. In the deployed position, three additional latches are engaged to transmit load from the upper fin to the lower fin. All of these operations are actuated with a common drivetrain. Each fin drivetrain consists of a Power Drive Unit (PDU), an EVA interface, two commercial off-the-shelf (COTS) torque limiters, two secondary gearboxes, and various connecting shafts.

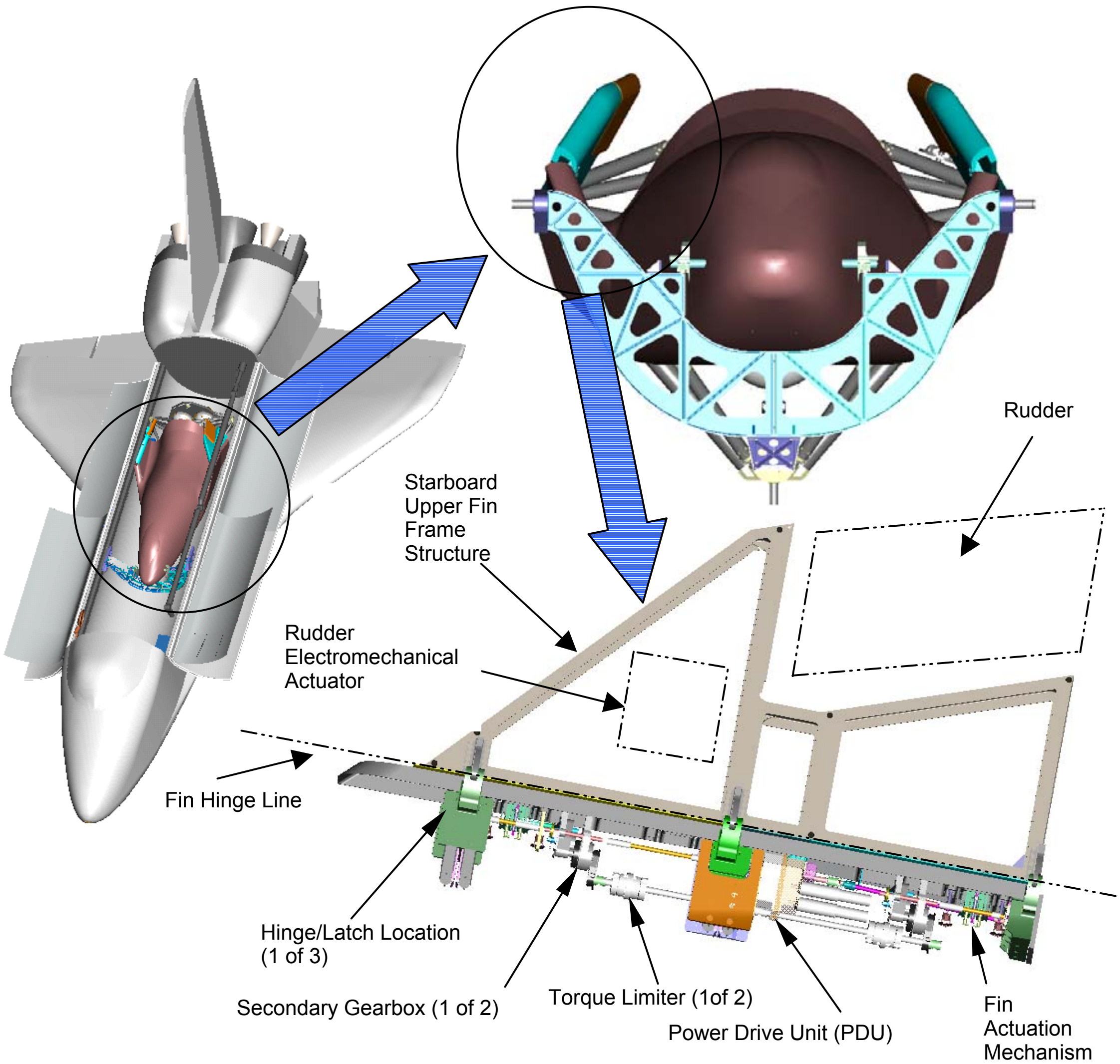

Figure 2. Overview of Fin Actuation

The PDU is the combination motor assembly/gearbox that provides torque to the fin folding and latching mechanisms. The PDU gearbox consists of planetary and translation gears that provide the mechanical logic to allow the fin to be actuated by a primary gearmotor, a secondary gearmotor, or an auxiliary manual/EVA input. A layout of the PDU is presented in Figure 7.

From the PDU, the torque is sent forward and aft with a torque tube to the torque limiters. The torque limiters are ball-detent slip clutches that are set to disengage when the torque in the torque tubes reaches a level that could potentially damage downstream components.

After the torque limiters, the torque is input into a secondary gearbox. The purpose of the secondary gearbox is threefold: 1) to shift the torque axis into the proper position for the mechanism shafts to pick up; 2) to further increase the torque provided to the fold mechanisms and latches; and 3) to split the single torque input from the torque tube into two separate and mutually exclusive torque outputs, one for the fold mechanism and one for the latches. Control of the relative motion of the two outputs is performed by a mechanism that is a separate subassembly contained within the secondary gearbox called the timing mechanism. From the secondary gearbox outputs, shafts deliver the torques to the two fold linkages and the three latch linkages. The forward secondary gearbox delivers torque to the forward linkage, forward latch, and mid latch. The aft secondary gearbox delivers torque to the aft linkage and aft latch.

Figure 3. V-201 Port Fin Stowed

Cams on the shafts engage limit switch assembly pairs. There are five limit switch assembly pairs; one for each of the two fin actuation linkages and one for each of the three latches. Signals from these are used by the fin fold software in the V-201 computer to shut the motor(s) down and provide feedback to the crew on a laptop in the orbiter. For the stowed position, the motor is shut down when a signal from each of the two limit switch assembly pairs indicates both linkages are locked overcenter in the stowed position. For the deployed position, the motor is shut off when a signal from at least one switch of each of the five limit switch assembly pairs indicates both linkages and all three latches are fully engaged.

The linkages and latches lock the fin in the deployed position as well as compress the fin fold line environmental seals. The environmental seal is based on the shuttle landing gear door seal. An outboard thermal barrier consists of three layers of a alumina-boria-silica continuous filament braided tubular sleeving over a knitted Inconel X750 wire spring tube filled with silica mat or batting. The thermal barrier is capable of withstanding the high temperature of entry. However, it is porous. An internal Teflon coated, Dacron stiffened silicon rubber pressure seal prevents the pressure difference from the interior of the fin to the exterior of the fin from ingesting the plasma through the thermal barrier. This pressure seal cannot take the high temperature; therefore, the thermal barrier and pressure seal are both required for a complete environmental seal. The fin is covered with tiles on the outboard surface and leading edge. The inboard surface is covered with a blanket except for tile hinge fairings around the protruding hinge lugs.

A counterbalance mechanism was designed to be installed internal to the upper fin to make 1-G ground fin actuation more representative of on-orbit use. This mechanism is removed for flight. The mechanism will not be discusses in this paper.

Fin Actuation Mechanism Linkages

Fin Actuation Mechanism Linkage Description

The fins are folded via two four-bar linkages that toggle over-center in both the stowed (folded inboard) and deployed (locked in outboard for atmospheric flight) positions. The upper fin is driven by a crank on the lower fin that is attached to the upper fin via a turnbuckle coupler. The turnbuckle consists of a center sleeve with a right-hand thread on one end and a left-hand thread on the other. Two M81935/1 rod ends with spherical bearings are threaded into the sleeve. Hence, when the turnbuckle linkage is installed with the rod ends pinned they cannot rotate; rotating the sleeve will extend or shorten the turnbuckle. The upper rod end is keyed to the sleeve with a NAS1193 locking device engaged in the castellated end of the sleeve and a jam nut locked with safety wire. The standard rod ends are cadmium plated, which is generally avoided in space vacuum environments. However, the JSC Materials and Processes Branch gave approval to fly these rod ends provided the exposed cadmium plated surfaces were painted with Super Koropon Epoxy Primer and the exposed cadmium plated threads were sealed with RTV-142.

The upper fin stow angle is set by two drag links, one forward and one aft of the four-bar linkage. The drag links are also turnbuckles. The crank and coupler linkage are housed in a sub-assy module to minimize assembly on the vehicle.

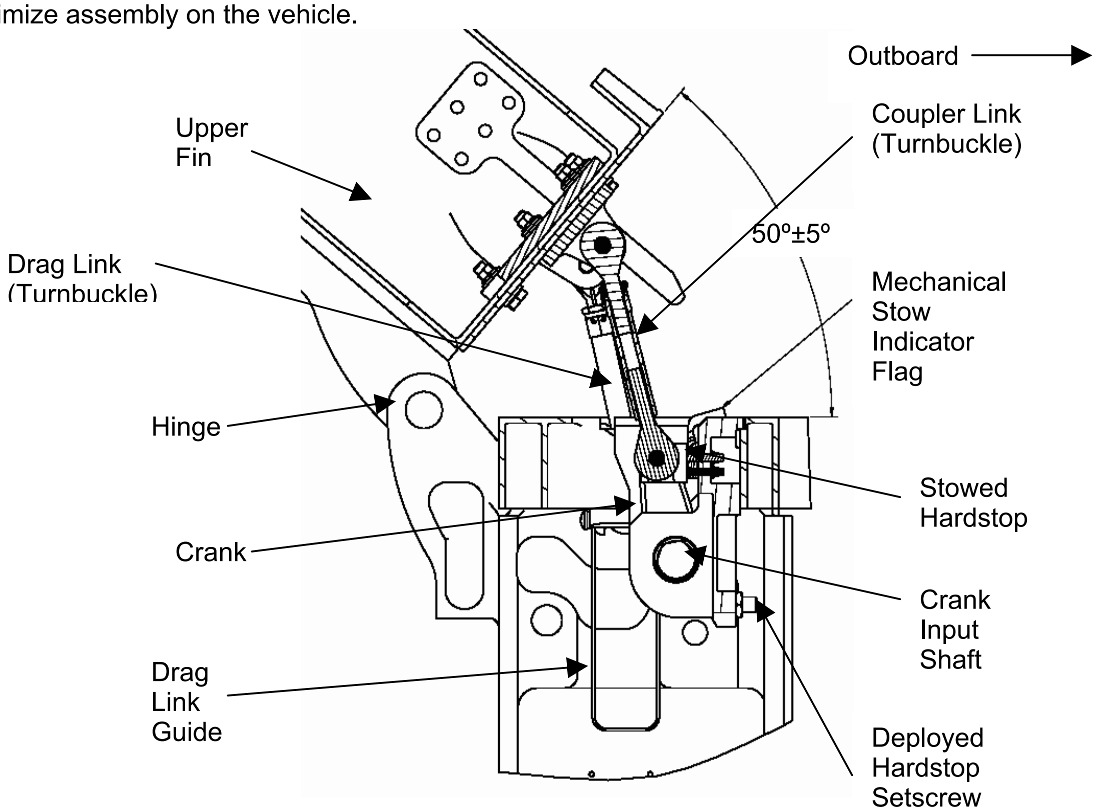

Figure 4. Fin Actuation Mechanism Linkage - Stowed

As the crank drives the fin to the stowed position (Figure 4), the coupler link must compress to rotate overcenter through the on-center position until the crank hits a hardstop. The link is over-center when the axis of the pin connecting the coupler and crank is rotated past an imaginary line drawn between the upper coupler pin axis and the Crank input shaft axis. A cam on the input shaft engages the limit switch assembly pivot arm which trips the limit switch. The hardstop is designed to stop the crank motion while the coupler link is still compressed. This compressed link preloads the coupler link in the overcenter position. Inertial and vibro-acoustic forces act on the fin during the shuttle launch. The direction of force may be outboard or inboard. Outboard acting forces only result in attempting to drive the linkage further overcenter. Inboard acting forces must overcome the compression preload in the coupler link and deform the drag links enough to get the linkage to an unstable on-center position. The compression preload can be increased by either lengthening the compression link or decreasing the fold angle by shortening the drag links. Reference SKK51356551 for rigging procedure.

As the crank drives the fin to the deployed position, the drag links retract into the lower fin along guides and the upper fin contacts hardstops on the lower fin. Due to space constraints, the deployed position is not shown; however, components described are visible in Figure 4. The crank continues to drive the coupler link over-center. Since the upper fin can no longer move, the crank must stretch the coupler link to rotate through the on-center position until the crank hits a hardstop in the crank assembly module. As the crank approaches the hardstop, a cam on the input shaft engages and trips the limit switch. The hardstop is a setscrew that is set to stop the crank motion while the coupler link is still tensioned. A jam nut with lockwire retains the setscrew position. This coupler link preloaded in tension provides a clamping force to lock the fin in the overcenter position. Aerodynamic forces on the upper fin only results in attempting to drive the linkage further overcenter. The tension preload can be increased by shortening the coupler link or raising the hardstop on the lower fin. Reference SKK51356551 for rigging procedure.

If the test flight is aborted, V-201 will be returned in the shuttle with the fins locked in the stowed position. If the limit switches fail, the crank assembly module contains a mechanical indicator flag to provide positive indication that the fins are locked stowed (Figure 4). It was decided not to include a mechanical flag for the deployed position because the assumption was made that at least one of each of the limit switch pairs for each latch would be working or the test flight would be aborted. A mechanical indicator for the deployed position would have required penetrating the TPS and a complicated linkage.

Fin Actuation Mechanism Linkage Prototypes and Testing

The concept for this linkage is similar to the Shuttle's Remote Manipulator System Manipulator Positioning Mechanism (MPM) pedestal linkage. Despite this proven flight history, the concept was not accepted initially. In the stowed position for launch, the two over-center linkages are the only locking feature that restrains the fins from striking the shuttle door radiators. In order to further explain this concept, a working wooden mockup was made to demonstrate the kinematics of the crank/linkage/hinge four-bar mechanism. A full scale prototype of the fin actuation mechanism linkage was then designed and built. The prototype uses two rectangular box structures to represent the interface for a portion of the upper and lower fin structure. The box structures are hinged together and actuated by a single four-bar linkage. The linkage included an instrumented coupler link. The prototype demonstrated that the desired tension and compression preloads could be achieved with the linkage. It allowed for correlation between input torque and preload. This information was required in order to design the drivetrain. The prototype also had mockups of the TPS to provide clear visualization of the outboard tiles and thermal barriers, the inboard blankets, as well as location of the pressure seal. Later in the project, the loads changed such that additional preload was required for the stowed position. A new sleeve with an increased outer diameter was designed, instrumented, and assembled in the prototype to verify the desired preload/input torque combination could be achieved. Every linkage used in the prototype, qualification, and flight units was instrumented and calibrated and proof tested in tension and compression. The instrumentation consisted of strain gages wired into a full bridge designed to amplify the signal due to the strain.

Fin Actuation Mechanism Linkage Lessons Learned

Lessons learned from the early prototype influenced the final design. The prototype did not include a crank assembly module. However, due to the difficulty of assembling the hardstops, bushings, crank, and coupler unit on the prototype, all these items were combined into a single sub-assembly for the flight design. Further, it was noted that the hardstops for the crank in the stowed position were not adjusted on the prototype from the original nominal position. Therefore, these were replaced with a fixed dimension striker plate and pivoting head on the crank for the flight design. The prototype used fixed gussets with setscrews to set the stow angle. This resulted in large holes in the upper structure. The gussets were placed in bending which placed the lower fin rib web in bending. The setscrew had limited adjustment capability. These drawbacks led to the drag link concept using the same turnbuckle linkages. This concept minimized the hole size in the structure. The turnbuckle also had a much greater adjustment capability. The drag links could be instrumented just as the coupler link. This aided in the rigging of the mechanism to ensure each preload reaction was split between the drag links equally.

Although this drag link design was more robust than the gusset design, it did lead to other challenges for the flight design. Since the drag links retracted into the fin as it was deployed and extended as it was stowed, the links had to be guided. The drag link fitting occasionally jammed on the sheet metal track. This problem was still being resolved when the project was cancelled.

It was noted in the prototype that if the locking device tab was oriented incorrectly it would plastically deform during fin actuation due to interference with surrounding structure. This caution was incorporated into the rigging procedure for the flight mechanism.

The TPS mockups led to redesign of the lower tile hinge fairings so that they were mounted on carrier panels and could be removed to facilitate fin removal.

The coupler link calibration worked best when the strain gage bridge was placed at the center of the turnbuckle sleeve over the hollow section where neither the upper or lower rod end thread was engaged. This resulted in smooth linear plots with no jump between tension and compression. However, in the second prototype set, the strain gages were located near the bottom of the sleeve. This was done to improve the wire routing. However, this resulted in a jog between tension and compression. Further, it was noted that to perform stiffness calculations based on the linkage calibration, it was necessary to add displacement instrumentation between the two pin connections of the rod ends. Simply using the head displacement of the tension/compression machine did not provide accurate results. Reference “X-38 V-201 Fin Linkage Calibration Report” for the complete calibration report on all the linkages.

Fin Deployed Position Latch Mechanism

<u>Latch Mechanism Description</u>

The fins are latched in the deployed position via three latch mechanisms installed in the three hinge fittings: forward, mid, and aft. Many naval aircraft use two hinges with two hydraulically fired pins. The pins are long pins with a tapered nose. This long stroke allows the taper to self-align the lugs with the pin as it is fired. However, V-201 did not have hydraulic power as an option. In order to meet the single fault tolerance requirement, it was decided to drive all systems with a single PDU. See the PDU section for further description of single fault tolerance. This restriction limited the stroke and force that was available to drive a pin. Initial concepts all focused on using three pins. Because the fin environmental seal required significant force to compress, the pin concept was abandoned in favor of a latch that could compress the seal. This eliminated the pin and lug alignment issue.

It was decided to use three hinge/latch pairs instead of two for several reasons. The fin fold interface is approximately 2.13 m (7 ft.) long. The vehicle has three main frames that provide a load path all the way through the vehicle. Structurally, the main aerodynamic load from the rudder transferred down the mid spar of the upper fin frame. The cleanest load path was to have a hinge/latch pair at this mid location with a frame lined up with the spar. A single hinge/latch pair forward or aft would not be sufficient to react the remaining loads. Finally, three latches in conjunction with the two actuation linkages are necessary to compress the environmental seal and minimize gapping at the fold line under aerodynamic loading.

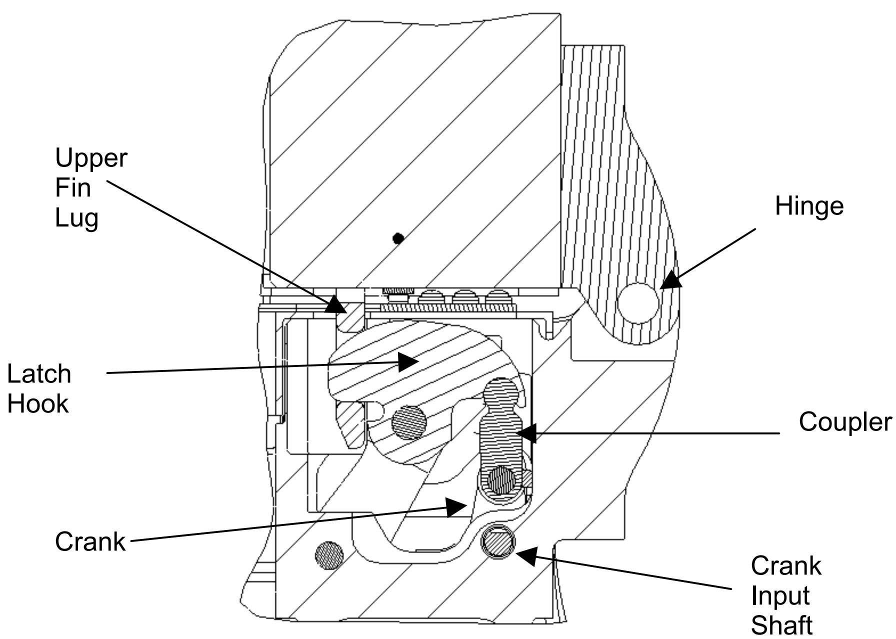

Figure 5. Fin Latch Mechanism

Each latch hook is driven by a four-bar linkage that toggles over-center to prevent the latch from disengaging. The linkage consists of a crank attached to hook via a coupler link. The entire latch mechanism is housed internal to an integrally machined hinge fitting. Due to limited available volume and access for assembly, the coupler-to-hook connection is a unique design that does not require an additional pin (Figure 5). The crank drives the hook to engage the lug attached to the upper fin, compress the seal until the upper fin engages a shimmed hardstop, and the coupler link is overcenter to lock the latch hook. As the crank approaches the hardstop, a cam on the input shaft trips the limit switch.

<u>Latch Mechanism Testing</u>
For all joints, the X-38 project required a positive margin of safety on the design load with a 1.5 factor of safety and a 1.15 fitting factor applied (or 172.5% of the design load). The mid hinge/latch analysis could not show positive margins with this fitting factor applied. The lead analyst agreed to waive the fitting factor if the joint was tested to destruction. The test assembly shown in Figure 6 was tested to destruction. It failed at the splice joint below the hinge line at a load equal to 171% of the design load. This closely matched the finite element predicted failure load and location. However, this margin must be reduced due to thermal considerations which results in an 8% decrease in strength for the aluminum hinge fitting. This reduces the margin of safety to 5% above the safety factor of 1.50. Reference “X38 V-201 Fin Mid Hinge/Latch Qualification Test Plan” and “X38 V-201 Fin Mid Hinge/Latch Qualification Test Report”.

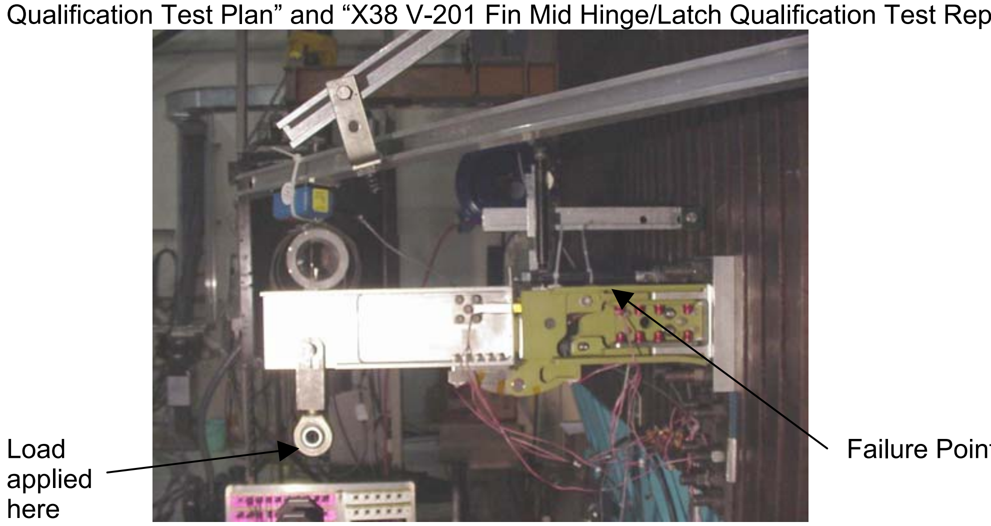

Figure 6. Fin Hinge/Latch Qualification Test Unit

<u>Latch Mechanism Lessons Learned and Observations</u>
- *Avoid shims.* Requires significant time to adjust small increments.
- *If you need shims, use laminated shims and manufacture extra shims.* Laminated shims allow small increment adjustment 0.05 to 0.08 mm (0.002 to 0.003 in.). However, it is easy to remove too many laminations when pulling the shims; therefore, it is important to have extra shims.
- *Negotiate interfaces carefully.* The failure point was a stress concentration that was originally away from the load path of the mechanism. However, as the mechanism evolved, the load path shifted closer to the stress concentration leading to a lower load capability.
- *Solid Film lubricants are often applied too thick.* It is a good idea to include a burnishing procedure in installation and/or rigging specifications. Often it is as simple of sliding the shaft into a mating bushing on the bench top to remove excess lubricant. This has been seen with Everlube and Tiolube.

Fin Actuation Mechanism Drivetrain

<u>Power Drive Unit Description</u>
The PDU gearbox consists of two planetary gear trains in series providing mechanical logic that allows multiple inputs. Translational gearing after the planetary gearing output is used to transfer the torque to the required output shaft axis of rotation. The first planetary train has one gearmotor attached to the sun gear and a second gearmotor attached to the ring gear allowing either gearmotor the ability to independently drive the first planetary train of the gearbox. The carriage output of the first planetary train

then drives the sun gear of the second planetary train and a manual EVA interface drives the ring gear of the second planetary train. This allows either gearmotor or the manual EVA interface to independently drive the overall gearbox. Each of the gearmotors has an electromechanical power-off brake (power to the gearmotor releases the brake) and the EVA Interface is locked when not in use. For overall fin actuation, the PDU provides full two-fault tolerance against motor failures. The CAD view in Figure 7 displays the geartrain.

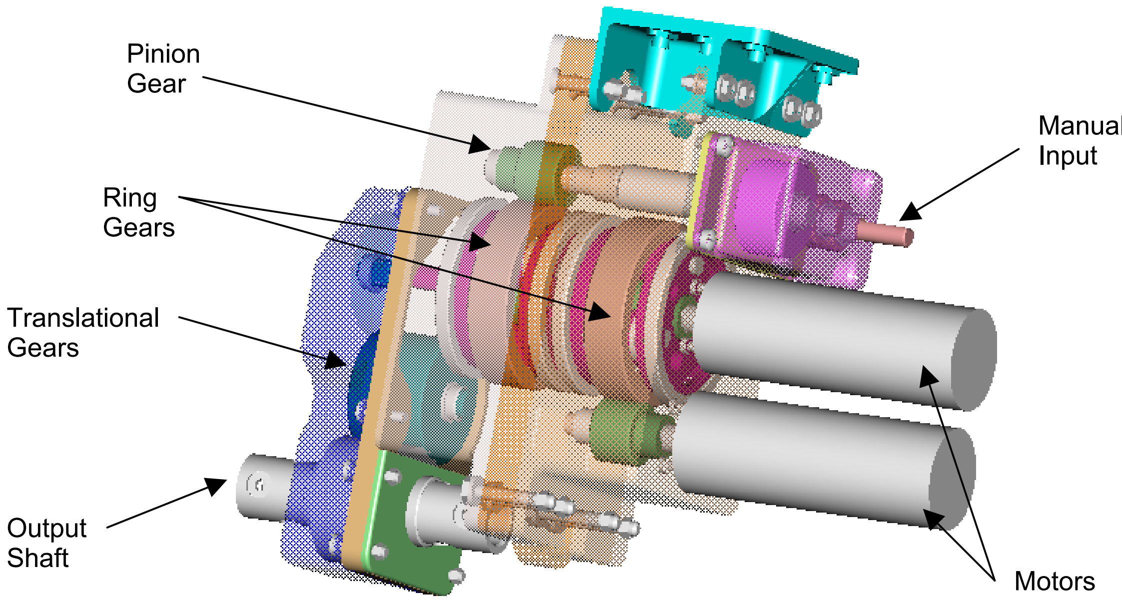

Figure 7. PDU

Each gearmotor assembly is equipped with a power-off brake that locks the motor shaft when not in use. This power-off brake is required due to planetary gear kinematics. Not locking the motor shaft of the un-powered motor would result in the powered motor backdriving the un-powered motor as opposed to supplying torque through the gearbox for fin actuation. The manual EVA interface is also locked while not in use for this same reason (see EVA Interface section). Originally, COTS DC brush motors were selected for use, but several problems were encountered. Problems included improper motor/brake assembly, unknown and unidentifiable materials usage, no secondary locking features for fasteners, non-vibration rated electrical interfaces, and inefficient volume usage. This eventually drove the team to specify custom flight-certified gearmotors for the final flight assembly. These flight motors were never purchased due to program cancellation. Each gear in the PDU is coated with a Dicronite TiS_2 dry film lubricant, and due to the low rotational speeds their dual-supported shafts are housed in aluminum bronze bushings lubricated with Braycote 602EF, with the exception of the planetary carriages which were supported by thin-section ball bearings. Because high rotational accuracy of the PDU output shaft is not required, a small backlash was designed in and the center distance for all gearing was fixed without provision for adjustment.

PDU Testing

The PDU underwent many informal functional tests after assembly, during which time it was discovered that five of the six COTS motors had been sent from the vendor incorrectly assembled, causing them to seize after about a minute of running. Seizure was due to motor operation with a partially engaged brake that would heat and swell until torque/current cut-off. A work-around was implemented allowing the PDU, torque limiter, and secondary gearbox to undergo thermal-vacuum testing. The PDU worked as designed during and after all phases of testing. Later, a fin-level random-vibration test was conducted and the PDU passed this test with no problems as well.

PDU Observations

- Using Geometric Dimensioning and Tolerancing significantly enhances ease of assembly and reduces tolerance stackups.
- Work with gear manufacturers to establish pre-coating dimensions and tolerances to allow for removal of recast layer or tooling marks, case hardening, and/or dry film application.
- Work with materials and processes experts to select bushing, shaft materials and lubricants.

PDU Lessons Learned

- *Avoid using primer for corrosion protection on any of the mechanism housings.* The PDU housings use Super Koropon, which was found to flake and generate debris during assembly. Subsequent housing designs used a hard anodize finish.
- *Be sure to include alignment aids for features requiring high positional accuracy or that influence geartrain performance.* The original design assumed the bolt holes' positional accuracy for the housings were sufficient to provide alignment aids for the gear train. However, it was found that the hole sizes required to allow assembly of multiple fasteners allowed the housings to shift enough to significantly reduce gear train efficiency.
- *Design appropriate access for those design elements that may require removal.* Many components, including the gearmotors, could not be removed without disassembling most of the gearbox. This became very inconvenient when circumstances arose which required replacement of the motors.
- *Use caution when deciding to use COTS products*. In this application, fixing shortcomings and certifying COTS motors for flight was significantly more expensive than originally considered. However, COTS motors were useful for initial ground testing of the geartrain.
- *For lower level components in a drivetrain, be sure to understand assembly interdependencies at the next higher level.* Originally, the drivetrain torque tube passed through a bushing in the housing of the PDU. Upon assembly, it was determined that this bound the torque tube and the bushing was subsequently removed.

Torque Limiter Description

In order to protect components downstream of the PDU against overload caused by failures, jams, or hardstopping, torque limiters were inserted into the drivetrain between the PDU and the each secondary gearbox (See torque limiter in Figure 10). Sizing the downstream components to handle the contingency loads, given such large gear ratios, proved not to be feasible from packaging and mass standpoints.

Based on the lesson learned during PDU development, COTS candidates were carefully evaluated before determining if this was an acceptable way to proceed. Detailed discussions with vendors concerning materials, production methods, and feature modifications were held prior to the decision. In the end, a COTS vendor was selected, and a flight-like but undocumented set of torque-limiters was procured for testing. The product was COTS except for the use of Braycote 602EF in place of the standard lubricant.

The torque-limiter is an adjustable ball-detent slip clutch set to slip at 40.3 N•m (357 in•lbf) and which automatically re-engages after 360 degrees of rotation or reversing the rotation so that the synchronization of the two sets of mechanisms is not lost if one clutch slips and the other does not. The clutches are fixed to the torque tubes through a friction-clamp interface.

Torque Limiter Testing

Upon receipt, several torque tests were run on the torque limiters to verify the correct slip setting before using the torque limiters with the flight hardware. The torque limiters operated with no problems during and after exposure to all environments. For flight, a specification control drawing would need to be released documenting the modified COTS hardware.

Torque Limiter Lessons Learned

If you're careful, COTS can work well! Significant cost savings were achieved using COTS clutches which only cost a few hundred dollars total versus a custom design effort to develop a torque-limiter. Make sure that material vacuum compatibility, if required, is completely worked out prior to purchase.

Secondary Gearbox Description

The secondary gearbox serves as a mechanism for increasing torque, shifting the torque axis and splitting the torque output into two axes. The torque tube travels inputs to a gear train that shifts the torque to a planetary train similar to that used in the PDU. Because the locking and folding shafts must operate at mutually exclusive times a timing mechanism, described later in this paper, was designed to control the motion of the two outputs. The secondary gearbox is pictured in Figures 9 and 10.

Due to severe packaging limitations, most of the gearing in the secondary gearbox was herringbone helical gearing to allow maximum torque capability with the smallest possible face width. The gears were coated with Vitrolube, a ceramic solid film lubricant capable of withstanding the higher contact loads. The gear teeth were further coated with a light film of Braycote 602EF. These gears also were mounted in aluminum-bronze bushings with fixed center distances because there was simply no room for bearings that could withstand the magnitude of load required to react the tooth loading. The housings were hard-anodized with the bushing holes being masked and chemical conversion coated.

<u>Secondary Gearbox Testing</u>
Once fully assembled, the secondary gearbox underwent benchtop testing in order to measure the efficiency of the gearbox, necessary for torque margin determination and proper rigging of the mechanisms. The efficiency, as expected due to the necessity of using bushings under high loads, was poor. The gearbox also underwent thermal-vacuum testing with the torque limiter and PDU, and went through fin random vibration testing. In all cases, the secondary gearbox operated with no problems.

<u>Secondary Gearbox Lessons Learned</u>
- *Use hex-head fasteners whenever possible.* Originally many of the fasteners, of which there was a fairly large number for sealing purposes, used offset cruciform recess heads which can be very difficult to install to proper torque values and even more difficult to remove. Every fastener that could be swapped was replaced with a hex-head version and assembly became much easier.
- *If you can't use bearings, you'll pay the price in efficiency.*

<u>Timing Mechanism Description</u>
The original design concept for the timing mechanism was very simple and consisted of two discs similar to Geneva mechanisms mounted on the two output shafts (See Figure 8). A simple test article was developed to explore the concept. It became immediately apparent that though simple, the high torques involved (282.4 N•m (2500 in•lbf) in one shaft and 50.8 N•m (450 in•lbf) in the other) produced very high sliding friction forces on the discs which used all of the torque from the gearbox and allowed no motion.

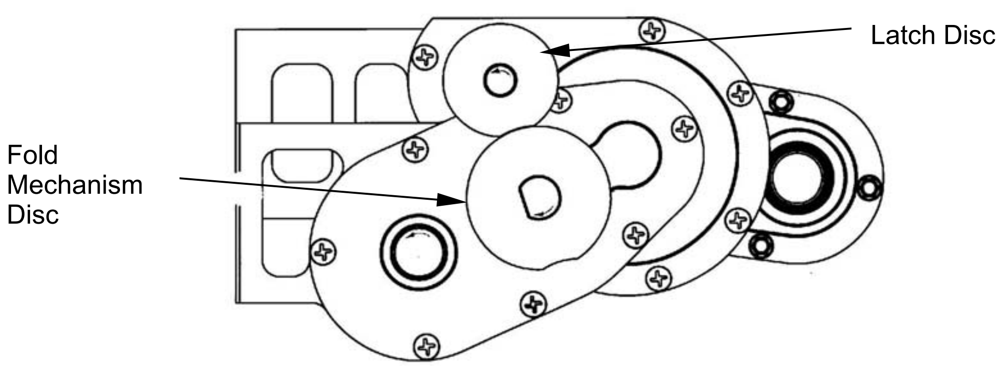

Figure 8. Geneva Mechanism Timing Discs

A second concept was developed to reduce the friction in the system. This was accomplished using a cam/roller/linkage concept that emphasized rolling friction rather than sliding friction. While the new design reduced the friction in the system, it also greatly increased the complexity and size of the mechanism. This larger design was moved from the output shafts to an open space between the gearbox housings (Fig. 9).

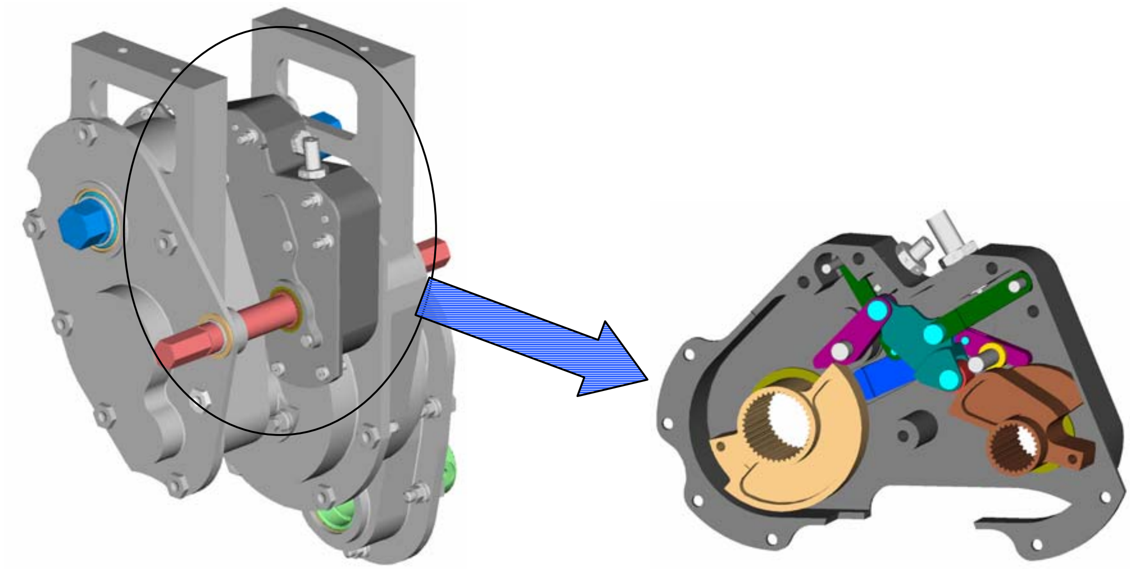

Figure 9. Cam/Roller/Linkage Timing Mechanism

The design was completed using solid models which were used for preliminary kinematic analysis. A detailed dynamic analysis was then performed. The strength analysis was performed with Mathcad based on the dynamic analysis results. An initial unit was then manufactured so that it could undergo operational proof testing. The proof test was planned to run to 100% of design limit load in the forward and reverse direction three times each, but during the initial cycle, the unit failed in the reverse direction. Upon opening the test unit, it was discovered that the pin upon which the roller rotated had sheared and the lug in which the pin rested experienced an ultimate bearing failure. In addition, other pins and holes in the linkage had yielded to various degrees.

Upon closer examination of the failure, it was found that the load applied to the linkage was not consistent with the magnitude or direction predicted in the dynamic analysis. An investigation of the dynamic model uncovered a small error in the application of the motion driver in the reverse direction that resulted in an incorrect load condition. Correction of the error produced new load vectors consistent in both magnitude and direction with that seen by the failed linkage.

The mechanism was modified in an attempt to correct the deficiencies in the original configuration by using a stronger material for the pins and links. Analysis indicated that this would result in very low, but positive margins in the pins so the test was repeated with the new pins. No failure occurred but subsequent inspection showed slight yielding in the pins was still occurring. The design was more broadly modified to increase the pin diameters slightly to increase their load capability. This was a design challenge given the tight confines but a solution was found.

The new design was then retested to the same loads. No failures were experienced and the mechanism operated as designed. A post-test inspection of the internal mechanism revealed no problems.

Timing Mechanism Lessons Learned

- *The simplest concept doesn't always work.* Mechanisms that work kinematically in the CAD system may not work when under the loads and friction of the real world.
- *Be careful with your analysis*. Be sure to review all assumptions
- *Always test!* Analysis may not be sufficient, and testing may be the fastest way to iterate a design.

Drivetrain Testing

The test unit shown in Figure 9 was thermal/vacuum tested. The sub-assemblies were later used in the fin engineering unit and have continued to operate reliably without incident. Reference "X-38 V-201 Fin Drivetrain Thermal/Vacuum Test Report" for description of thermal/vacuum testing.

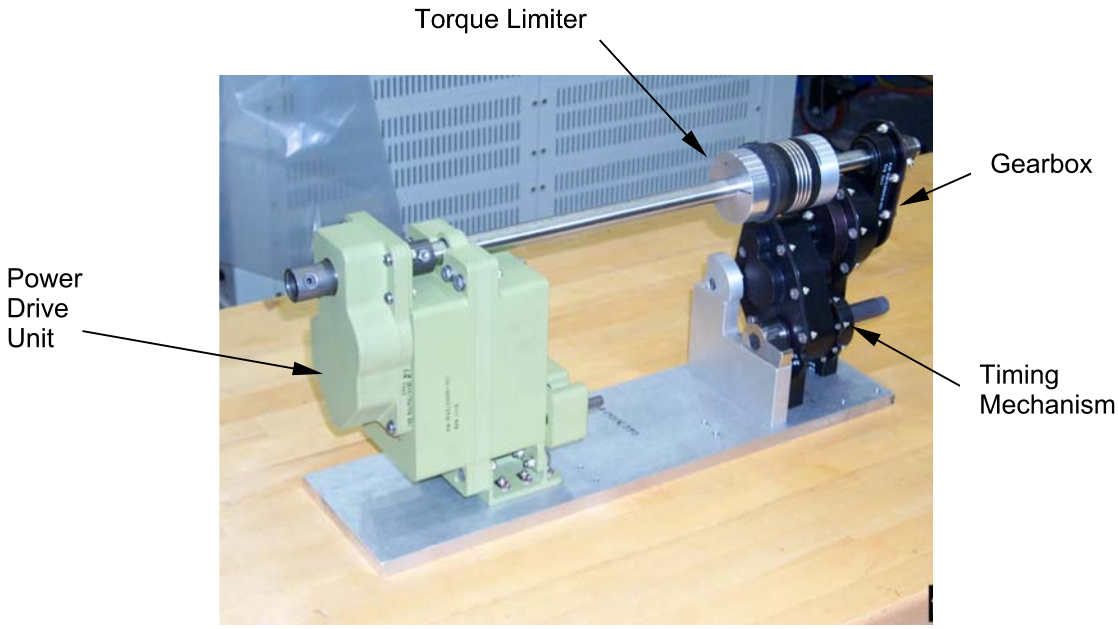

Figure 10. Drivetrain Test Unit

Limit Switch Pair Assembly

<u>Limit Switch Pair Assembly Description</u>
In order to indicate successful stow or deploy, five limit switch pair assemblies are used. First, the assembly positions the two limit switches side-by-side so that a single cam actuates both switches. The pair is required to meet the single fault tolerance requirement for remote actuation. Second, the assembly is designed to prevent damage to the switches. The pivot arm is spring loaded to engage the switches which are off-loaded as the cam engages it. In this way, over-travel of the cam could not damage the switches. See Figure 11.

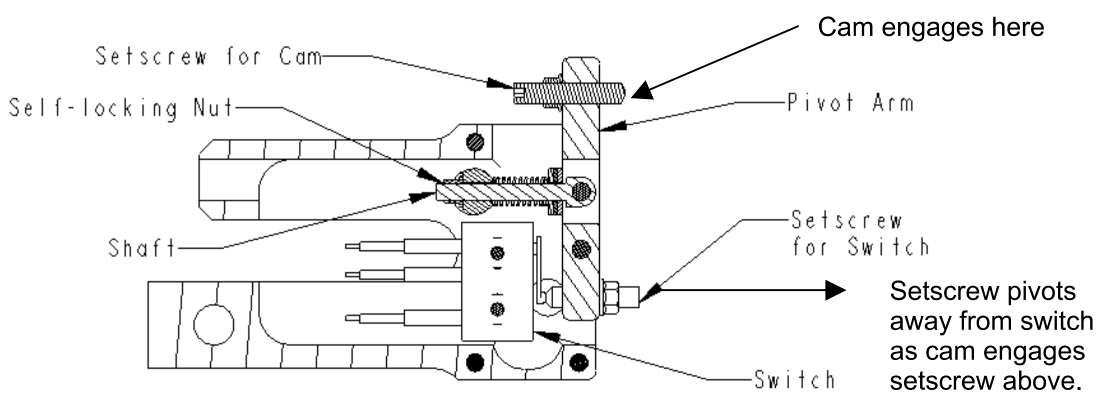

Figure 11. Limit Switch Pair Assembly

<u>Limit Switch Pair Assembly Lessons Learned</u>
Attempting to meet requirements for two locking features can result in other problems.

- The pivot arm has small # 6 setscrews (3.5 mm) made of 304 stainless which contact the limit switch arm in order to trip both limit switches simultaneously. The setscrews are locked in place using a self-locking insert with a self-locking jam nut. However, the 1.5 mm hex in the setscrew would occasionally strip while attempting to overcome running torque. Later versions of the assembly replaced the setscrew with socket head cap screws which had a larger hex and could be driven through the running torque.

- The pivot arm sub-assembly spring is retained on shaft with a 2.8 mm (0.112 in.) threaded end. A self-locking nut is preloaded against the shoulder of the shaft in the current design. In the original design, the nut was not preloaded. The secondary locking feature to retain the nut was a cotter pin through the threaded portion of the shaft. During the design, the shaft with the 1.5 mm hole (0.062 in.) was viewed at enlarged scale. However, the CAD model did not include the minor diameter of the thread. The hole left so little material that the lower end of the shaft snapped when the nut was installed.

EVA Interface Assembly

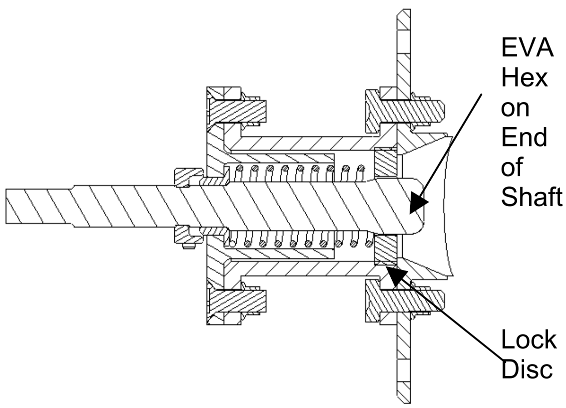

Figure 12. EVA Interface Assembly

<u>EVA Interface Description</u>
The EVA Interface is a standard 11.1 mm (7/16 in.) EVA-compatible double-height hex head. The hex is locked from rotation with a disc that has a 12-point interface to the hex and a splined outer diameter. The disc is spring loaded in the locked position. The disc is disengaged by engaging a deep well socket for the Pistol Grip Tool (PGT).

<u>EVA Interface Lessons Learned</u>
It is critical to always fully follow procedures regardless of how familiar the operation has become. The procedure calls for actuating the interface using an extended deep well socket. During initial functional testing, the same socket was always checked out from X-38 tool control. However, during static testing the vehicle was moved to another location with a different set of tools. The static test required functional testing before and after the test. At some point during the testing, an extended socket was used. However, the socket was not deep well and only partially disengaged the locking disc. When the interface was rotated, the hex ground into the disc creating metallic chips. The interface assembly was disassembled and cleaned. A new disc was installed. A placard referencing the procedure was installed to prevent the event from occurring again.

Fin/Rudder System Level Testing

<u>Fin/Rudder Static Testing</u>
The fins were statically tested while installed on the vehicle in both the deployed and stowed positions. In the deployed position, the fins were tested to 100% of limit load. The Shuttle program requires proto-flight hardware be tested to 120% of limit load. However, the X-38 project decided to lower this requirement because 1) the mid hinge/latch pair only had 0.05 margin of safety, 2) the deployed fins only affected X-38 mission success, not shuttle safety, and 3) 100% of limit load provided sufficient data for model correlation. Reference “X-38 V-201 Integrated Structural Test Plan – Aerosurfaces Tests” and “X-38 V201 Fin Deployed Test Report”.

Conversely, in the stowed position there was greater uncertainty in the load, but also larger margin over the design load. Therefore the fins were tested until the coupler links which are the lowest capability part in the system neared the rated load for the turnbuckle. The fins were tested up to 218% of design load for the stowed position. This provides additional test proven capability in the event that later analysis predicts higher-than-expected fin loads during launch. Reference “X 38 V 201 Integrated Static Structural Test Plan – Shuttle Launch & Landing Tests” and “X-38 V-201 Static Test Report – Shuttle Launch & Landing Tests”.

<u>Fin/Rudder Vibration Testing</u>

The starboard fin/rudder was removed from the vehicle after static testing and installed on an engineering unit that had a representative lower fin structure housing the fin actuation mechanism. This unit was successfully vibration-tested in the X, Y, and Z axes. Reference “X-38 V-201 Fin Engineering Development Unit (EDU) Vibration Test Plan”.

Functional Testing

The port and starboard fins have been actuated several dozen times. Before and after each static test and vibration test, the fin mechanism was actuated. The majority of the functional tests have been performed using the EVA interface either with a socket wrench or the PGT. The starboard fin was actuated multiple times using one of the COTS motors.

Conclusions

X-38 Status

The X-38 project was officially shut down in May 2003. NASA Headquarters decided that NASA needed to pursue a multi-use vehicle. The two B-52 drop test vehicles, V-131R and V-132, and the space flight prototype vehicle V-201 have been mothballed and are museum pieces at JSC. V-201 is approximately 80% complete.

Remaining work for Fin before Flight

Due to the cancellation of the X-38 project, the mechanism flight performance may never be known. However, the testing completed thus far indicates the mechanism would function in the thermal/vacuum environment after withstanding the vibro-acoustic launch loads and could also lock the fin deployed against the aerodynamic flight loads. Remaining testing would include a vehicle level acoustic test with the fins stowed and a vehicle level thermal/vacuum test where the fins would be functionally tested. The mechanism is currently in a ground test configuration with nonfunctioning motors installed. The fin can be actuated via the EVA interface. The 1-G counterbalance is also installed in the upper fin. The primary outstanding issue is the procurement of flight motors. A specification control drawing for the motors has been released. There are also several minor issues that have been documented in Discrepancy Reports that would have to be resolved in order to fly. The keys to success on this project included building prototypes early and performing sub-assembly level testing leading up to system level testing.

References

1. Robertson, Brandan “X-38 V-201 Fin Linkage Calibration Report” JSC-49740 – JSC Scientific and Technical Information Center
2. Lupo, Christian P. “Procedure, Fin Folding Drivetrain” SKK51356551 – NASA/JSC Engineering Drawing Control Center
3. Lupo, Christian P. “X38 V-201 Fin Mid Hinge/Latch Qualification Test Plan” JSC-28409 – JSC Scientific and Technical Information Center
4. Smith, James P. “X38 V-201 Fin Hinge/Latch Qualification Test Report” JSC-29294 – JSC Scientific and Technical Information Center
5. Robertson, Brandan “X-38 V-201 Fin Drivetrain Thermal/Vacuum Test Report” JSC 29911 – JSC Scientific and Technical Information Center
6. Edelstein, Karen S. “X 38 V 201 Integrated Static Structural Test Plan – Shuttle Launch & Landing Tests” JSC 29182 – JSC Scientific and Technical Information Center
7. Edelstein, Karen S. “X-38 V-201 Integrated Structural Test Plan – Aerosurfaces Tests” JSC 29184 – JSC Scientific and Technical Information Center
8. Lupo, Christian, “X-38 V201 Fin Deployed Test Report” (Draft available upon request)
9. Lupo, Christian, “X-38 V-201 Static Test Report – Shuttle Launch & Landing Tests” (Draft available upon request)
10. Edelstein, Karen S. “X-38 V-201 Fin Engineering Development Unit (EDU) Vibration Test Plan” (Available upon request).