ABSTRACT

In the time when the Crew Return Vehicle (CRV) was being considered as a follow on joint venture between ESA and NASA, Contraves Space AG was being considered as the sub-contractor for the Fin Folding Mechanism. Although due to the lack of funding in the USA, the CRV programme was stopped, ESTEC decided to continue the development of the Fin Folding Mechanism (FFM) for a “future Space Transportation Vehicle” (STV) for launch and re-entry. Contraves Space was subsequently awarded with a contract to develop a simplified mechanism to replace the mechanism that was designed for CRV.

The function of the FFM is to fold the Vehicle Fin inboard so that any future STV can be accommodated in today’s launchers. The FFM must be able to support the fin during launch, deploy the fin and then support the fin in the deployed position during re-entry.

Contraves Space reviewed the functionality of the old CRV design, with its’ different mechanisms for each of the functions, and a synchronisation system, and established a novel design where one mechanism can be used to perform all three functions.

The final design utilises a four bar link mechanism, driven over-centre in both of the end positions, driven by a high-torque drive unit comprising a Harmonic Drive, a conventional gearbox and a 3 Phase DC motor. To protect the high-torque drive system against stall, a clutch has been utilised based on spring and rotating ball technology.

These devices are used seldom in space, but as surviving stall is often a problem for mechanisms, the results from the test programme will be interesting with respect to the use of such devices in future space mechanisms.

The design of the mechanism consists of three almost identical nodes, one of which has been built and is undergoing “qualification” testing at Contraves Space. The mechanism will be subjected to the full range of tests including, functional tests, random vibration tests, thermal vacuum tests and a static load test.

This paper will detail the design of the mechanism and highlight the problems that had to be overcome in the design phase.

1. INTRODUCTION

There was a stage a few years ago when the Crew Return Vehicle (CRV) was for seen as the follow on vehicle from the X-38. The CRV was to be a joint European/US development but due to financial constraints in the USA, was cancelled in late 2002. Since this time, ESTEC have been working on future vehicles for space transportation and as part of this investigation, various programmes have been activated within Europe to provide supporting technologies to aid in the development of a “Future Space Vehicle”.

The Fin Folding Mechanism is one of these developments.

The main purpose of the Fin Folding Mechanism (FFM) is to allow for the accommodation of any future STV by folding the fin inwards during launch and deploying the fin in orbit.

The initial design produced by NASA [1] was considered to be relatively complicated and heavy so the aim of the activity described by this paper was to produce a “simplified low mass” mechanism, using the same Fin and Vehicle structural parameters that were used for the original X-38 V-201 design.

The design for the X-38 shown on figure 1, utilises two separate motors (with brake) to meet the redundancy requirements necessitating a specific gear (differential gearbox) to switch to the redundant motor as the same drive chain is used. In addition, the latching and deployment was achieved using two separate mechanisms and thus another two differential gearboxes were needed to separate these two functions.

The differential gearboxes add mass to the system and also introduce additional failure modes into the mechanism, which reduces its reliability. This results in a very complicated design that has a mass of some 39.37 kg per side.
Although the design parameters for the FFM were similar to those used for the CRV design, the aerodynamic loads on the fin were some 1.5 times larger and the launch loads approximately 3 times those for X-38. This has lead to problems during the design that will be discussed later.

2. OVERVIEW OF REQUIREMENTS

2.1 Functional Requirements

The FFM has to fulfill more than one function as shown on table 1. Here the deployment sequence is identified and whether or not Extra Vehicular Activity (EVA) and On-Ground manual resetting is required in the given configuration.

<table>
<thead>
<tr>
<th>Sequence</th>
<th>Description</th>
<th>HW Configuration</th>
<th>Required Actuation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Fin is stowed</td>
<td>STOWED</td>
<td>Auto EVA On-ground manual</td>
</tr>
<tr>
<td>2</td>
<td>Release fin</td>
<td>TRANSIENT</td>
<td>YES YES YES</td>
</tr>
<tr>
<td>3</td>
<td>Deploy fin</td>
<td>TRANSIENT</td>
<td>YES YES YES</td>
</tr>
<tr>
<td>4</td>
<td>Seal and preload fin</td>
<td>TRANSIENT</td>
<td>YES YES YES</td>
</tr>
<tr>
<td>5</td>
<td>Latch fin</td>
<td>TRANSIENT</td>
<td>YES YES YES</td>
</tr>
<tr>
<td>6</td>
<td>Fin is deployed</td>
<td>DEPLOYED</td>
<td>- - -</td>
</tr>
<tr>
<td>7</td>
<td>Release fin</td>
<td>TRANSIENT</td>
<td>NO NO YES</td>
</tr>
<tr>
<td>8</td>
<td>Stow fin</td>
<td>TRANSIENT</td>
<td>YES** NO YES</td>
</tr>
<tr>
<td>9</td>
<td>Latch fin</td>
<td>TRANSIENT</td>
<td>NO NO YES</td>
</tr>
<tr>
<td>10</td>
<td>Fin is stowed</td>
<td>STOWED</td>
<td>- - -</td>
</tr>
</tbody>
</table>

**Partial Restowing Requirement

Table 1: Functional Requirements

2.2 Thermal Environment

The thermal environment for the FFM is shown on table 2. The large range for the temperature environment places demands on the design of the FFM such as:

- Differential Thermal Expansion
- Selection of lubrication
- Selection components
- Design Tolerances
- Thermal Stresses

In addition to these requirements, the mechanism must also survive the re-entry environment with the inherently high temperatures. Although it is possible to renovate or replace parts after the re-entry phase, the mechanism has to remain structurally sound which places additional demands on the design.

2.3 Aerodynamic Loads

The aerodynamic loads provided for the fin during re-entry are almost 1.5 times the value for the X-38. This is mainly to allow for growth of the fin in the course of the development of the FSV. The increase in the aerodynamic loads has lead to a dramatic increase in the pre-load necessary to prevent gapping between the fin and the root of the FSV.

Table 2: Temperature Environment for the FFM

2.3 Launch Loads

The launch loads provided were almost 3 times larger than those specified for the X-38. This has a dramatic effect on the pre-load in the mechanism. With the fin in the stowed position, the pre-load within the mechanism must be sufficient to avoid gapping at
the end stops. The launch loads have been used to determine the necessary pre-load to avoid gapping.

2.4 Pre-load Requirements

From the analysis of the loads induced in the mechanism during both launch and re-entry, the Pre-load requirements have been determined. These are:

Pre-Load (Launch) = 25 kN
Pre-load (Re-entry) = 17 kN

These pre-loads are extremely influential in the design of the FFM.

3. MECHANISM DESIGN

3.1 Overview

The X-38 design presented in [1] has been well thought through and fulfils all the requirements placed on it. In the fulfilment of the FFM design activity, it would have been quite easy to copy the design of the X-38 Fin Folding mechanism but because of its complicated combination of latches, deployment levers and gearboxes, it was decided that an alternative should be investigated.

At first glance, the design for the FFM seems similar to that used on the X-38, however the X-38 design utilises a synchronisation system with different mechanisms for each of the functions. The design of the FFM is greatly simplified utilising one mechanism that can be used to perform all three functions of latching/unlatching, deploying and re-latching. Additionally, with a simplified mechanism, there is a low probability of failure and less single point failures. Due to the single torque tube connecting the three nodes, synchronisation problems do not occur. The X-38 mechanism included a timing mechanism to activate the latching systems. The X-38 mechanism had many brackets to support the drive train which are no longer required in the FFM design.

This mechanism utilises a four bar link mechanism that is driven over-centre by a high-torque drive unit comprising of a Harmonic Drive, a conventional gearbox and a 3 Phase DC motor, at both end positions. To protect the high-torque drive system against stall, a clutch has been utilised based on spring and rotating ball technology.

These devices are used seldom in space, but since surviving stall is a critical issue for this mechanism such a device is necessary to ensure the survival of the mechanism.

3.2 Overall Design

With the exception of the motor drive unit attached to one of the three nodes, all of the nodes are similar. For this reason, the performance of the FFM mechanism will be demonstrated on one node only. Before the individual nodes are described, the complete mechanism will be described in the following paragraphs.
The Fin Folding mechanism shown on figure 2 consists of 8 elements:

- Aft Link Mechanism
- Aft Gearbox
- Mid Link Mechanism
- Mid Gearbox
- Forward Link Mechanism
- Forward Gearbox
- Drive Unit
- Connecting Torque Tubes/EVA

The flight version of the Fin Folding mechanism will have different link mechanisms, two identical units at the Aft and Mid Hinge locations and a mirrored unit at the Forward Hinge location. Initially, it was intended to build the complete mechanism shown here, however as all the mechanisms are almost identical, the Breadboard under development here will consist of just one node. Other differences between this Breadboard unit and the Flight Unit include:

- The omission of the thermal seal. The spring effect of the seal on the closed fin is not significant when compared to the force required to pre-load the Fin.
- A clutch mechanism can be built into the flight mechanism to allow the Drive Unit to be decoupled from the Torque Tube during the EVA activity if necessary.

3.3 Mechanism detail Design

The breadboard model of the FFM consists of three main elements as shown on figure 3.

- The link mechanism
- The Gearbox
- The Drive Unit

The construction of the three sub-assemblies is described in the following.

**Link Mechanism**

The link mechanism has two functions. The first is to provide pre-load in the Fin-Body joint during the launch and re-entry phases.

As the link mechanism consists of an over-centre driven crankshaft, the performance of the “locking” function removes the need for a positive lock of the pin during both the launch and re-entry phase which reduces the complexity of the overall mechanism. The second function is to perform the deployment and stowing action on the fin.

As shown on figure 2, the Fin will be attached to the space vehicle using three hinges that provide the pivot for the link mechanism and enable the mechanism to rotate the Fin by the specified 50° from the stowed position, into the deployed position.

The Link mechanism is utilised in all three positions, Aft Hinge, Mid Hinge and Forward Hinge.

As shown in figure 4, the mechanism is realised using the elbow lever and tension bars to pre-load the system in the stowed position. The deployment is achieved in a similar fashion to that used for the original X-38 deployment mechanism. The approach of driving the elbow lever over-centre is used to latch the fin in the deployed position.

In the stowed position, the pre-load is applied by the tension bars, which react at one end against a hard stop that inhibits further movement of the system, and the link bar, which applies a pre-load in the tension bars when fully extended. A Shim is used to adjust the pre-load in the stowed position.

In the deployed position, the crankshaft rotates, pulling the fin against hard stops at the Fin/Body interface. The pre-load is induced when the link bar is fully extended.
Again a shim is used to adjust the pre-load in the deployed position.

The 4 bar link is initially in the stowed position with the Tension Bars pre-loaded to inhibit opening during launch. To deploy, the whole system needs to be back driven and the crankshaft driven over-centre which requires an applied load that is larger than the applied pre-load.

![Figure 4: Link Mechanism](image)

As the crankshaft passes over-centre, it needs to rotate further to deploy the Fin. When the crankshaft rotates over the centre position, the load in the tension bars increases and subsequently decreases rapidly once the crankshaft has passed the centre position. The loads induced when driving over the centre position are higher than the loads developed in the mechanism due to the launch loads which inhibits the mechanism from going over the centre position and hence causing an inadvertent deployment of the Fin during launch.

Once it is driven passed the centre position, the crankshaft is rotated a further 180° by the drive system, pulling the link bar at the same time, which reacts with the fin by rotating it about the hinge pivot points and hence deploying the Fin.

When the Fin approaches the deployed position, the Fin contacts the hard stop at the Fin/Body interface. The crankshaft continues to rotate pulling the Link Bar over the centre point. As with the deployment from the stowed position, the load at this point is at a maximum that reduces slightly as the crankshaft passes over the centre point.

The residual pre-load in the end position is dependent on temperature and at all times must be greater than the loads that will be induced during re-entry causing the mechanism to remain in this position without additional features. In a similar manner to the stowed position, the link bar cannot rotate in the reverse direction as the load in the link bar increases and remains above the flight loads.

Micro-switches are included in the design to protect the mechanism against running into the end stops in both the stowed and deployed positions. The micro-switches are connected to the drive controller. When all three are activated in the deployed position the drive electronics will turn off the mechanism electronics. In a similar manner, when the mechanism is stowed, with the activation of all three switches, the controller will be switch off.

The micro-switches are operated by an auxiliary actuator, which is activated by the lever attached to the crankshaft. This presses against the micro-switches to operate them in both the stowed and deployed positions. Over travel of the lever is limited to 1° as the crankshaft will contact the end stop after this angle is reached. This means that the micro-switch does not sustain damage when there is a system problem causing the mechanism to try to rotate further. To protect the rest of the mechanism, a clutch (described later) has been built into the system.

**Gearbox**

As shown on figure 2, the gearboxes are also located at the Aft, Mid and Forward Hinges. The three gearboxes are identical to each other.

Each gearbox, shown on figure 5, consists of a Harmonic Drive Unit, a Safety Coupling and a set of three Spur Gears to transfer the torque tube rotation to the gearbox drive axis.

The gearbox connects the crankshaft input to the 4 Bar Link mechanism on the output side of the gearbox, and to the output Shaft from the Drive Unit.

As shown on figure 6, the Harmonic Drive is attached in the gearbox housing and is supported by an input and output bearing set.

The Safety Coupling is supported by the Harmonic Drive input bearing and a second single sleeve bearing made of steel, and is attached to the input shaft. The input bearing is mounted within the housing and supports the output shaft from the spur gear stage. The shaft axis is supported by a journal bearing in the back cover of the gearbox.
The function of the Safety Coupling is to limit the maximum torque that can be applied at the input of the Harmonic Drive as a torque increase from the drive chain could cause damage to the Harmonic Drive if the 4 Bar Link mechanism is blocked in any way.

Figure 5: Gearbox Assembly

The Spur Gear set is located between the Harmonic Drive and the Safety Coupling and transmits the torque from the Input Shaft through three spur gears to the Harmonic Drive axis.

Figure 6: Drive Unit

In order to ensure that the gearbox operates with sufficient margin, it is necessary to heat the gearbox above 0°C. It is not intended to allow operation at temperatures of the gearbox below 0°C.

The reason for this is that the wet lubrication system selected for the gearbox exhibits higher friction torque characteristics at low temperatures which has the affect of requiring a higher motor torque to satisfy the safety margins.

If the drive chain between the harmonic drive and the link mechanism is blocked for any reason, it is possible for the harmonic drive to develop a torque in excess of 1000Nm which would lead to damage within the system and hence to a catastrophic failure of the mechanism.

To avoid this, the Safety Coupling has been included. This is an important part of the mechanism and is used to restrict the input torque to the harmonic drive to inhibit damage by providing a clutch mechanism when the input torque exceeds a defined level.

It is often the case that space mechanisms have to withstand the affects of “stall” and such devices are not freely available in the space mechanisms market. The “qualification” of the Safety Coupling will provide additional options to mechanism designers in the future.

The Harmonic Drive

The Harmonic Drive a transmission ratio of 120:1 and a maximum allowable torque at the output of 686 Nm. The harmonic drive was selected over a planetary gear system because it offered the following advantages to the design:

- Excellent Positioning Accuracy and Repeatability
- High Torque Capacity
- Zero Backlash.
- High Single-Stage Reduction Ratio
- High Efficiency
- High Torsional Stiffness
- Back Driving
- Central hollow shaft

The Safety Coupling

The function of the Safety Coupling is to limit the maximum torque moment that can be applied to the Harmonic Drive input to 5.2Nm ± 10%. If a failure occurs in the 4 Bar Link mechanism, the input torque in the Harmonic Drive would increase to about 70Nm which would cause damage in the Harmonic Drive. To avoid this a Safety Coupling has been build at the Harmonic Drive input to stop this happening.

Although the selected safety coupling has not been demonstrated in the space environment and has not been qualified for the space environment, the supplier was able to provide a version that is compatible the space environment. Material and lubrication changes were defined to bring the design up to a “space standard”.

The coupling is able to function in the range –65°C and +70°C and can survive the temperature range of –85°C to +200°C.

The accuracy of the safety torque moment is ±5% of the value set. When the coupling operates, a residual moment of 0.5Nm is witnessed in the coupling that is sufficient to rotate the mechanism in the free rotation state but not to take it over centre.
On future models of the FFM, a proximity sensor will be utilised to indicate if the safety coupling has operated. The sensor will also be linked to the control electronics and when it indicates that one of the safety couplings has operated, the drive system will shut down.

The Spur Gear Stage

The Spur Gear Stage shown on figure 7, consists of three gears and is used to translate the rotation of the motor drive axis to the drive axis of the harmonic drive. The transmission ratio for the gearbox is 1.56:1. The gears are each supported by a journal bearing made from A286 Stainless Steel. Care was taken to match the material CTE’s to avoid problems due to thermal expansion within the mechanism.

The three gears are mounted within a casing, made in aluminium, that is attached to the main gearbox.

Figure 7: Spur Gear Stage

Motor Drive Unit

The drive unit selected for the Breadboard model of the FFM is of a space standard but is not capable of operating at the lower temperatures defined in the requirements and requires heating when testing at lower temperatures.

The motor drive was selected on the basis of cost only to ensure that the project remained within the financial constraints. On the flight unit, a motor drive unit that can survive at the lower and higher temperatures must be selected. A motor drive has been defined but was too expensive to purchase for this development activity.

As shown on figure 5, the Motor Drive Unit consists of the following elements:

- a drive unit (combined gearbox and motor)
- a primary stage gearbox to transmit the output from the drive unit to the drive shaft
- an EVA shaft

These elements are described in the following paragraphs.

Combined Gearbox/Motor

The Gearbox/Motor selected for the Flight Model will be taken from CDA Intercorp (Gear ratio 120:1) but as mentioned earlier, for the Breadboard, a motor drive from Faulhaber (gear ratio 111:1) will be used.

Both motor drives are similar in performance so there will be very little change in the system design for the Flight Model.

The motor gearbox from Faulhaber can deliver up to 16Nm continuously but needs to be heated for operation at low temperatures because of the Hall Sensor regulation used for the position and speed control.

The drive output speed for the FFM application is 6.6 rpm for a 30 minutes deployment and 9.9 rpm for a 20 minute deployment.

Primary Stage Gearbox

On the output from the drive unit, there is a pinion gear that drives the primary stage gearbox that rotates the Drive Shaft connecting to all three of the nodes as shown on figure 5.

The primary stage gearbox shown on figure 8 is connected to the output from gearbox/motor combination and transmits the gearbox output torque to the Drive Shaft. Its relationship with the rest of the mechanism is shown on figure 9.

The transmission ratio of the gearbox is 2.09:1.

Each of the gears is supported by a journal bearing made from A286 Stainless Steel, again taking care to match the material CTE’s to avoid problems due to thermal expansion within the mechanism.

De-coupling Mechanism

As the flight model must be capable of being driven by the manual EVA operation, to avoid unnecessary
energy being expelled by the astronaut in back-driving the motor drive unit, it may be necessary to de-couple the EVA shaft from the drive unit.

The de-coupling mechanism shown schematically on figure 10 may be incorporated in the Flight Model of this mechanism.

The mechanism utilises a push-pull rod that is activated by EVA. This reacts against a spring on the main drive axis and de-couples a clutch mechanism moving the drive gear out of the primary stage gearbox. The push-pull rod is locked in position enabling the EVA to drive the shaft directly without back driving the motor drive.

Figure 8: Primary Stage Gearbox

Figure 9: Gearbox Arrangement for FFM

Figure 10: De-coupling Mechanism

**Lubrication**

The lubrication of the FFM presented a problem during the design phase. With the wide temperature range of operation from -65°C to +70°C and with the re-entry temperature of 204°C, the impact on the selection of lubrication was critical. Firstly, the lubrication must survive within this temperature range without change in its physical characteristics, and secondly, the changes in the friction torque characteristics over the temperature range must be sustainable by the mechanism.

The main problem with wet lubrication is the performance at the temperature extremes. The lubrication has to operate at temperatures of -65°C and at +70°C, and additionally it has to survive temperatures of above 200°C. The problem with dry lubrication is that the materials exhibit higher coefficients of friction than wet lubricants at room temperature and are expensive to apply. Experience with dry lubricants with items such as the harmonic drive is also limited.

As shown on table 3, there are in fact very few wet lubricants that are suitable. The two critical temperatures for the operation are -65°C and +204°C. In fact from those looked at, only two are suitable, namely, Braycote 601EF grease and Fomblin Z25 oil.

<table>
<thead>
<tr>
<th>Product</th>
<th>Type</th>
<th>Max Temp</th>
<th>Pour Point</th>
<th>Viscosity Index (20°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fomblin Z25</td>
<td>Oil</td>
<td>+250°C</td>
<td>-75°C</td>
<td>260 cSt</td>
</tr>
<tr>
<td>Braycote 601EF</td>
<td>Grease</td>
<td>+204°C</td>
<td>-73°C</td>
<td>250 cSt</td>
</tr>
<tr>
<td>Nye Rheolube 2000</td>
<td>Oil</td>
<td>+125°C</td>
<td>-48°C</td>
<td>Not available</td>
</tr>
<tr>
<td>MAPLUB PF 100-a</td>
<td>Grease</td>
<td>+130°C (+250°C)</td>
<td>-60°C</td>
<td>Not available</td>
</tr>
<tr>
<td>MAPLUB SH 050-a</td>
<td>Grease</td>
<td>+100°C (+200°C)</td>
<td>-40°C</td>
<td>Not available</td>
</tr>
</tbody>
</table>

Table 3: Typical Wet Lubricants [2]
The major problem with operation at the low temperature is that the viscosity increases dramatically thereby increasing the resistance experienced. Since it is possible to heat the subsystems to temperatures above 0°C, these two lubricants will be used for the Harmonic Drive, the Safety Coupling, the Spur Gears and the Ball Bearings.

Due to the number of cycles for the journal bearings being very low, and since wet lubrication at low temperatures exhibits a high friction torque, it was decided that dry lubrication would offer the best solution.

As shown on Table 4, the characteristics of various dry lubricants have been considered for the journal bearing lubrication. Due to its suitability in both the vacuum and the air environment [3] & [4], the NPI 1220C process has been selected for the lubrication system for the journal bearings.

**Mass**

The main purpose of the development activity was to produce a “simplified low-mass” mechanism capable of replacing the existing X-38 fin folding mechanism. The FFM is currently not compliant with the mass requirement of 27.6kg which was based on assumptions of mass reductions possible on the X-38 mass and its loads.

Although some of the mechanism functions were removed from the FFM specification (e.g. automated stowage) with the idea that mass could be saved, as shown on Table 5, mass savings were not possible. Although not required, automated stowage is also still possible with the new FFM design.

<table>
<thead>
<tr>
<th>Material</th>
<th>Friction Coeff. in Vacuum</th>
<th>Friction Coeff. in Air</th>
<th>Sliding Friction in (Vacuum /Air)</th>
<th>Temp Range</th>
<th>Drawbacks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sputtered MoS2</td>
<td>v.low</td>
<td>Moderate</td>
<td>Good/Moderate</td>
<td>4 – 700K</td>
<td>Reduced life in air &amp; high friction</td>
</tr>
<tr>
<td>Ionised Lead</td>
<td>moderate</td>
<td>Moderate</td>
<td>Poor/Poor</td>
<td>4 – 550K</td>
<td>Limited testing in air</td>
</tr>
<tr>
<td>Ionised Silver</td>
<td>moderate</td>
<td>Moderate</td>
<td>Poor/Poor</td>
<td>Up to 1100K</td>
<td>Rapid wear in ATOX</td>
</tr>
<tr>
<td>Fibreslip (from AMPEP)</td>
<td>moderate</td>
<td>Moderate</td>
<td>Good/Good</td>
<td>228 - 523K</td>
<td>Large increase on friction at low temperature</td>
</tr>
<tr>
<td>NPI 1220 C Vitro-lube</td>
<td>low</td>
<td>low</td>
<td>Good/Good</td>
<td>Up to 755K</td>
<td></td>
</tr>
</tbody>
</table>

Table 4: Characteristics of Dry Lubrication [2]

The main reasons that the mass target has not been achieved in this development can be seen in the following comparison between the X-38 and the FFM.

- The loads during launch are approximately three times higher than the loads encountered by the fin folding mechanism of the NASA X-38.
- The aerodynamic loads are 1.5 times those specified for the X-38.

Nevertheless mass saving was achieved on parts which are not directly affected by the higher loads (e.g. drive unit) as shown on Table 5.

<table>
<thead>
<tr>
<th>Designation</th>
<th>SLM-FFM [kg]</th>
<th>X-38 [kg]</th>
<th>Mass Saving</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gearbox</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Housing/gears/bearings/shafts etc.</td>
<td>14.882</td>
<td>8.437</td>
<td>44.5%</td>
</tr>
<tr>
<td>Torque Limiter</td>
<td>1.200</td>
<td>2.994</td>
<td></td>
</tr>
<tr>
<td>Nuts and Bolts</td>
<td>0.600</td>
<td>1.040</td>
<td></td>
</tr>
<tr>
<td>Subtotal</td>
<td>16.682</td>
<td>12.471</td>
<td>-25.5%</td>
</tr>
<tr>
<td>Link Mechanism</td>
<td>19.314</td>
<td>13.744</td>
<td>-26.9%</td>
</tr>
<tr>
<td>Total</td>
<td>39.656</td>
<td>39.372</td>
<td>-0.7%</td>
</tr>
</tbody>
</table>

Table 5: Mass Comparison between FFM and X-38

4. **SUMMARY**

It was initially intended when the abstract was submitted that it would be possible to present test results in this paper. Unfortunately, the development programme was subjected to a slippage of a few months meaning that this was no longer possible.

Although to-date, little testing has been performed, the analysis results show that the mechanism should have no problem fulfilling the specified requirements and that during the test phase that is on going this will be demonstrated.

To help in the understanding of the design the pictures shown on figures 11 and 12 are provided.

5. **CONCLUSION**

Although the mass goal for the FFM was not reached, the FFM is at least 30% more mass efficient than the X-38 design when the load capabilities of the two mechanisms are considered.
The design of the FFM has some advantages when compared against the design of the X-38 which also justify not achieving the mass target. These are as follows:

- FFM design is greatly simplified.
- The failure cases and single point failures (such as timing mechanism, PDU power-off brake etc.) in the X-38 design have been eliminated.
- The FFM mechanism is capable of carrying at least 50% higher loads than the X-38 mechanism.
- The FFM mechanism is insensitive to synchronisation errors (no timing mechanism necessary).
- FFM is capable of performing all the functions of the X-38 design (e.g. re-stowing automated or manually).
- No additional brackets are needed to support the drive train as the FFM design is very compact. The mass resulting from these components were not considered in the X-38 mass budget and thus this will save additional mass on the FFM under global consideration.

6. REFERENCES