Cryogenic Nano-Actuator for JWST

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Abstract

An extremely precise positioning mechanism has been developed for use in space for optical positioning of large mirrors. The design incorporates traditional mechanical components such as gears, bearings and flexures in a unique configuration covered by two patents. This linear actuator is capable of 10 nano-meter position resolution over a range of 20 mm and can operate under cryogenic conditions. The design, assembly, construction and testing of this mechanism are presented.

Introduction

The James Webb Space Telescope (JWST) is configured to be a large deployable spacecraft as shown in Figure 1. A key component of JWST is the Optical Telescope Element (OTE), which consists of all the components along the optical path including the Primary and Secondary mirrors. The Primary mirror is about 6.5 meters in diameter and is made up of 18 segments. These mirrors are folded up during launch and deployed in space. Once unfolded, the mirrors must be deployed away from the launch restraints and then adjusted very precisely.

Figure 2 shows how each primary mirror segment and the secondary mirror is supported and controlled by six linear actuators to obtain six degrees of positioning control. Each mirror can be positioned in tip, tilt, piston, horizontal & vertical decentering and clocking. Two actuators are assembled into a bipod assembly as shown in Figure 3. The final hexapod configuration is made up of three bipods. In addition, each primary mirror segment features a central actuator for adjusting the radius of curvature of the segment. The positioning and focusing of the primary and secondary mirrors require a total of 144 actuators.

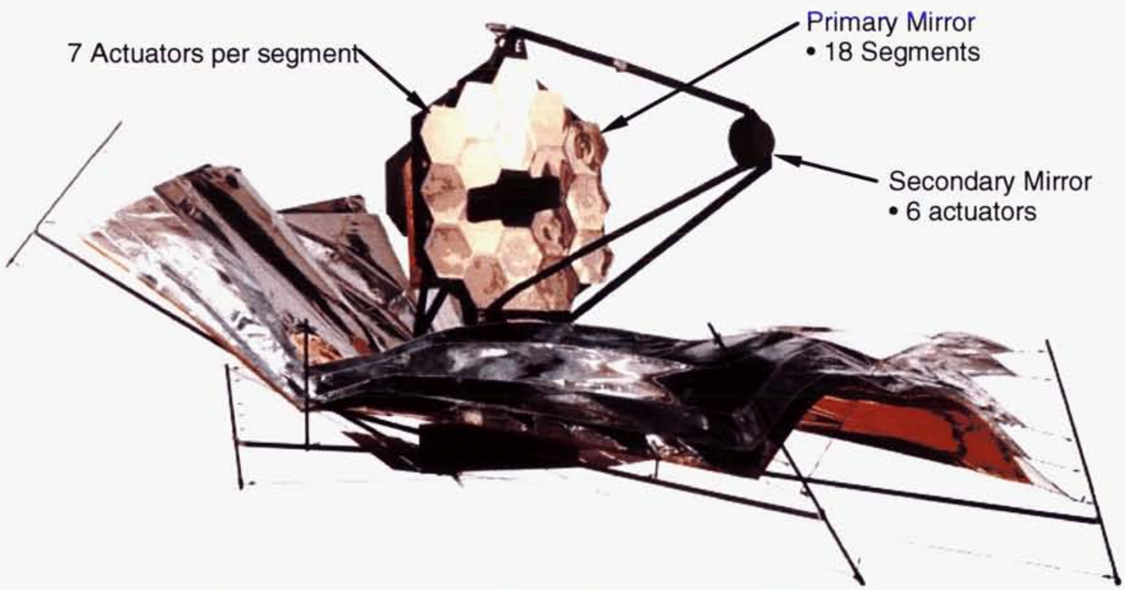

Figure 1. James Webb Space Telescope (JWST)

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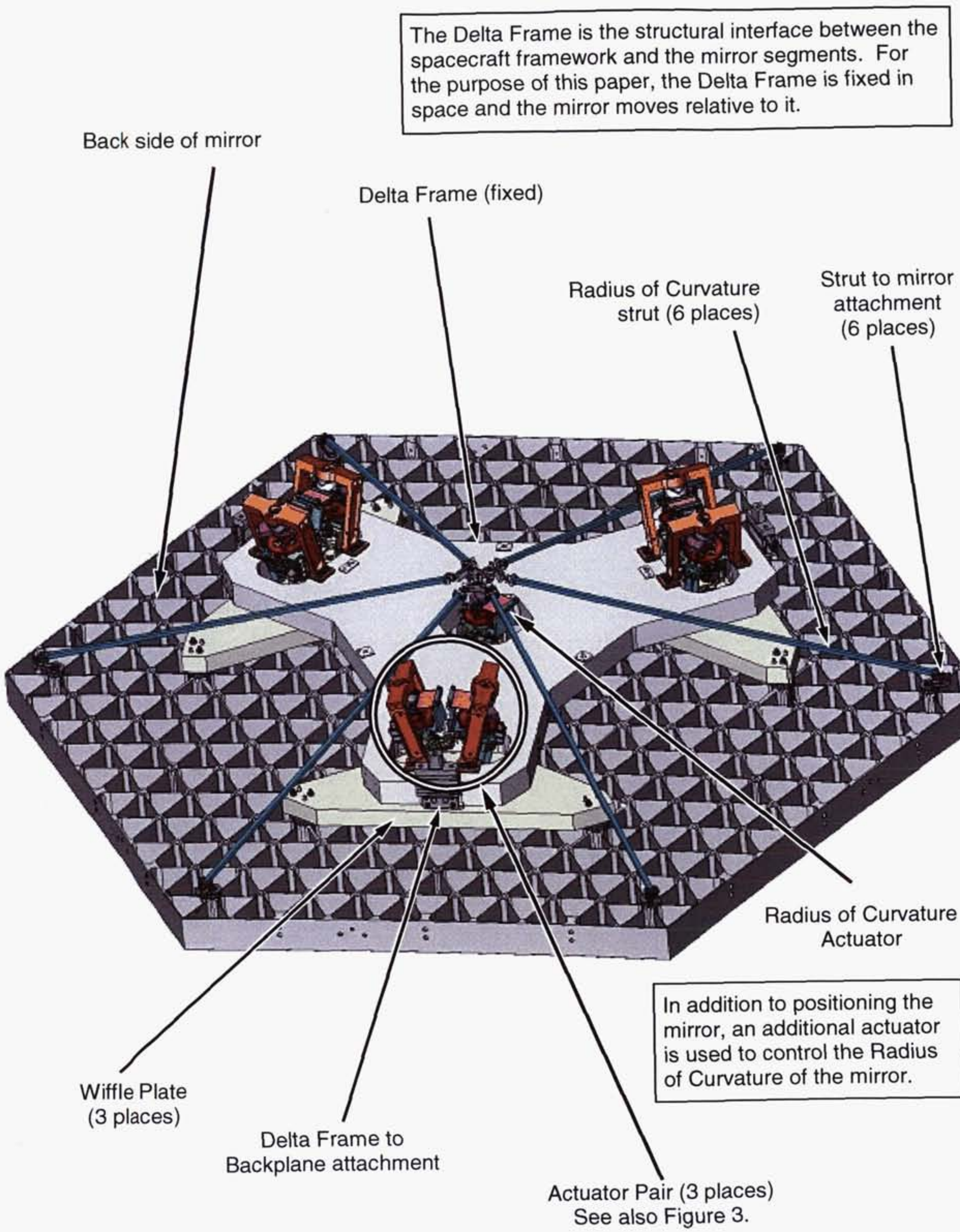


Figure 2. Hexapod Mounting

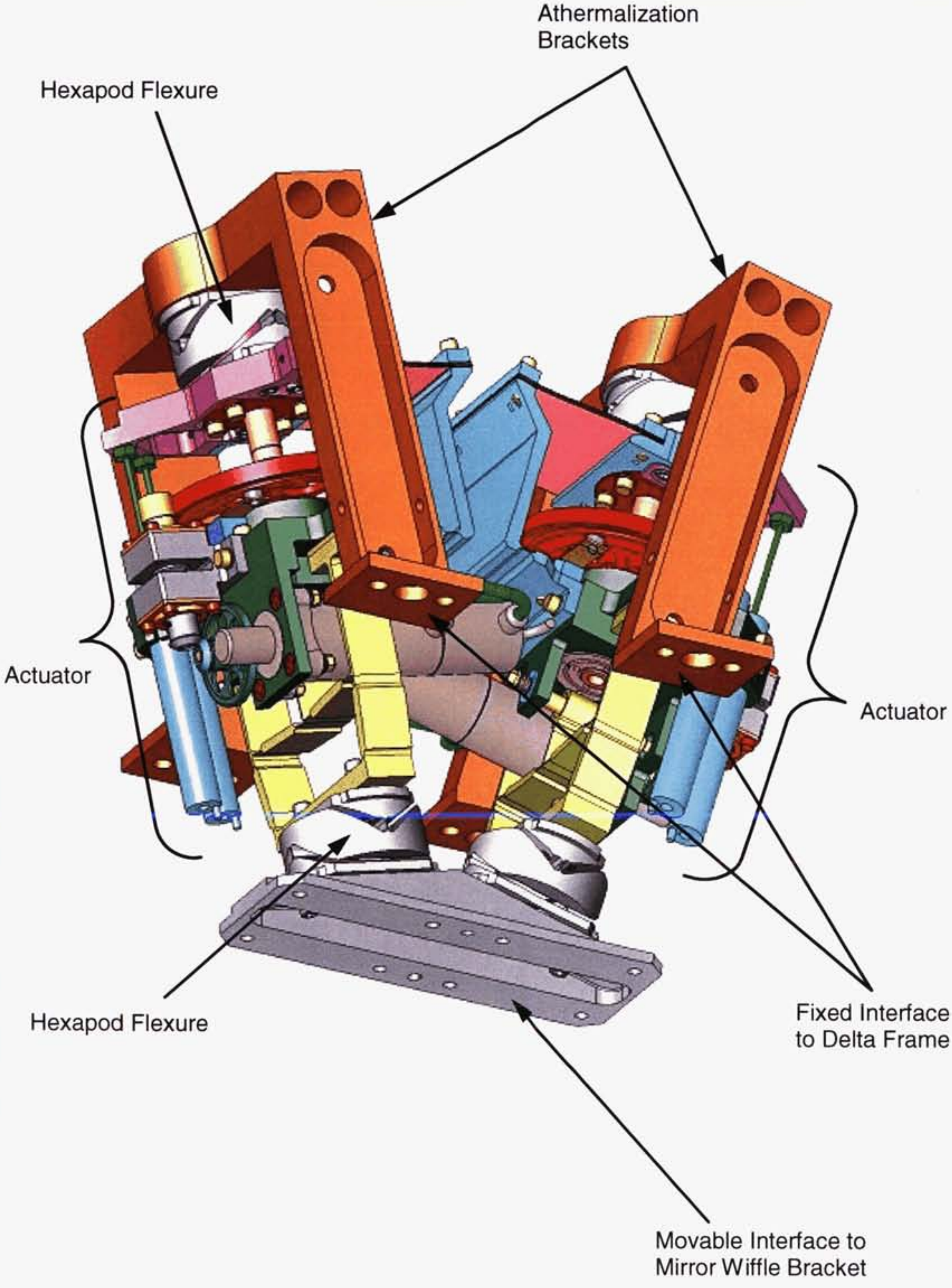


Figure 3. Actuator Pair

Actuator Requirements

The Actuator for JWST has two top level requirements:

I. *Accurately position the mirror segments.*

II Support the mirror segments during ground test and launch

These requirements apply to the actuators supporting the 18 segments of the primary mirror as well as to the actuators that support the secondary mirror.

<u>Positioning requirements</u>
The first requirement of positioning was used to generate the following derived requirements:

1. Move the mirror from the stowed position to the nominal deployed position.
2. Move the mirror from the nominal deployed position with 6 degrees of motion.
3. Position each segment to nanometer resolution.
4. Support the segments in a hexapod configuration.
5. Operate at cryogenic & ambient conditions
6. Operate over the life of the mission.

<u>Load Requirements</u>
The second requirement of support was used to generate the following derived requirements:

1. Support the mirror segments during launch.
2. Support the mirror segments during ground optical testing.
3. Support the mirror segments during ground transportation.
4. Hold the mirror segments in place with power off.
5. Support the segments in a hexapod configuration.
6. Operate at cryogenic & ambient conditions

The derived requirements are summarized in Table 1.

Table 1. Requirements Summary

Property	Requirement	Capability	Compliance
Fine range	>7 micrometers	10 micrometers	yes
Fine step size	<10 nanometer	7.7 nanometer	yes
Fine repeatability	<3 nanometers	2 nanometers	yes
Coarse range	>20 mm	21 mm	yes
Coarse step size	<1.0 microns	0.058 microns	yes
Axial load	1890 N (425 lb)	2650 N (595 lb)	yes
Axial stiffness	24,500 N/mm (140,000 lb/in)	25,200 N/mm (144,000 lb/in)	yes
Axial holding	380 N (85 lb)	890 N (200 lb)	yes
Position feedback	20 microns	12 microns	yes
Cryogenic operation	30 K	20 K	yes
Nominal length	138.8 mm (5.5 in)	138.8 mm (5.5 in)	yes
Mass	700 gram (1.55 lb)	665 gram (1.47 lb)	yes

Early Development

The success of the actuator is primarily due to two important inventions: the fine stage flexure and the coarse drive coupling. It is important to understand the operation of these two elements before going on to the overall actuator description. The actuator is comprised of numerous individual design elements that all work together in order to satisfy the demanding requirements of cryogenic nanometer-level positioning. However, these two inventions enable the fine adjustment capability over the long range of motion required for the actuator.

Fine Stage Flexure

The development of the actuator began at Ball Aerospace in 1997 with the invention of the fine stage flexure. A need was established for a motion reduction device that could convert a relatively coarse input motion into a well controlled, optical-level, fine motion. It was well known that flexures work well for motion control because they enable motion without backlash or hysteresis. However, simple flexures were not able to achieve the large ratio required for this application.

The solution, partially shown in Figure 4, was the "Motion Reducing Flexure Structure", developed by Ball Aerospace and covered by United States Patent number 5,969,892 dated October 19, 1999. This compound flexure operates in two stages. As the middle of the cross bar is moved up and down, the sides are moved out and in, thereby causing a small but controlled change to the overall height. Motion reduction of up to 100:1 can be achieved using this design.

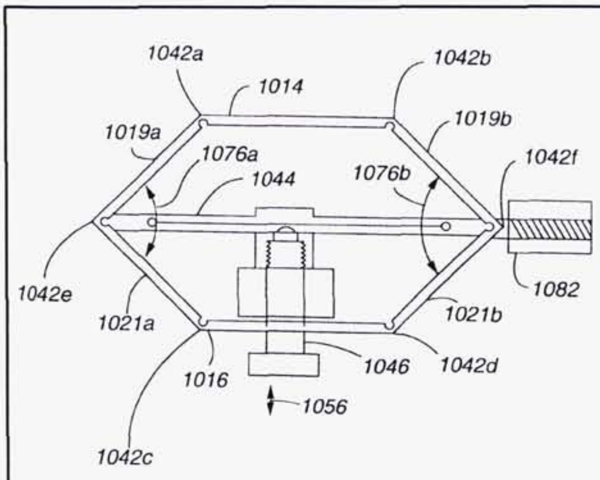


Figure 4. Flexure Patent Description

Coarse Drive Coupling

The next development was the use of a single motor to operate both coarse and fine motion. Although the fine stage flexure provides the nanometer-range positioning accuracy needed for aligning the mirror segments, it does not accommodate the large range needed for moving the mirrors from the stowed position to the deployed position. For this reason, a coarse motion feature was needed for the actuator. It was desired to have one motor drive both the coarse motion and the fine motion.

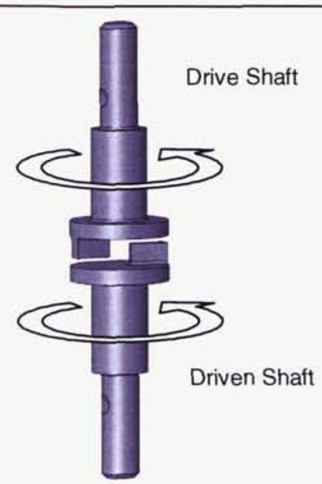


Figure 5. Coupler

To move both the fine stage and coarse stage with a single motor required the use of a coupling that could switch between coarse and fine modes or at least disengage the coarse motion. This was achieved by the invention of a tumbler type coupling, which connects the fine drive to the coarse drive as shown in Figure 5. This coupling consists of two rotating disks, each with a protruding pin. This results in a deadband or backlash of approximately 90% of the rotational input. When the motor is reversed, the drive pin backs away from the driven pin so that the coarse motion is decoupled from the drive train. In this deadband zone only the fine stage is engaged, which enables precise mirror positioning. At the end of this travel, the coarse shaft is again engaged to enable coarse motion. This mechanism was developed Ball Aerospace and is covered by United States Patent 6,478,434 dated November 12, 2002.

Actuator Chronology

The complete development of the actuator is beyond the scope of this paper but several important milestones should be noted

1997 Fine stage flexure developed[1]: Large axial input results in small axial output due to compound flexure configuration.

1999 First Actuator built[2]: 10-mm motor, dry film lubrication on all gears and bearings, tested at cryogenic temperatures (30 K), demonstrated fine resolution of less than 10 nanometers.

2000 AMSD Actuator built[3]: ¾-inch motor, more robust, twin counter-rotating coarse drive screws, three monopods per mirror, first mirror phasing demonstrated.

2002 IR&D Actuator: ¾-inch motor, more modular to reduce fabrication & assembly costs, large coarse drive screw to accommodate launch loads. Never tested.

2003 Test Bed Telescope Actuator[4]: 10-mm motor, low cost, 150 units built, hexapod configuration, smaller size, flight-quality positioning, ambient conditions, new fine stage flexure design.

2004 Flight Actuator: ¾-inch motor, ball screw for improved axial load capability, dry lube on all bearings & gears, new fine stage flexure.

Fine Stage Improvement

Figure 6 shows the simplified fine stage flexure. The shape is generally that of the capital letter "A". Like the original fine-stage flexure, movement of the cross-beam deflects the side beams, which, in turn change the overall height of the flexure. The cross-beam is attached to an eccentric cam shaft, which is driven by a gearmotor. As the cam shaft rotates, the cross-beam is driven up and down resulting in a sinusoidal displacement pattern. The sinusoid is distorted due to the compound nature of the geometry. The fine range of motion can be easily adjusted by changing out the pair of shims under the feet of the fine stage flexure. A change in shim thickness of about 0.1 mm (.004 in) results in a change of fine motion of about 1 micron.

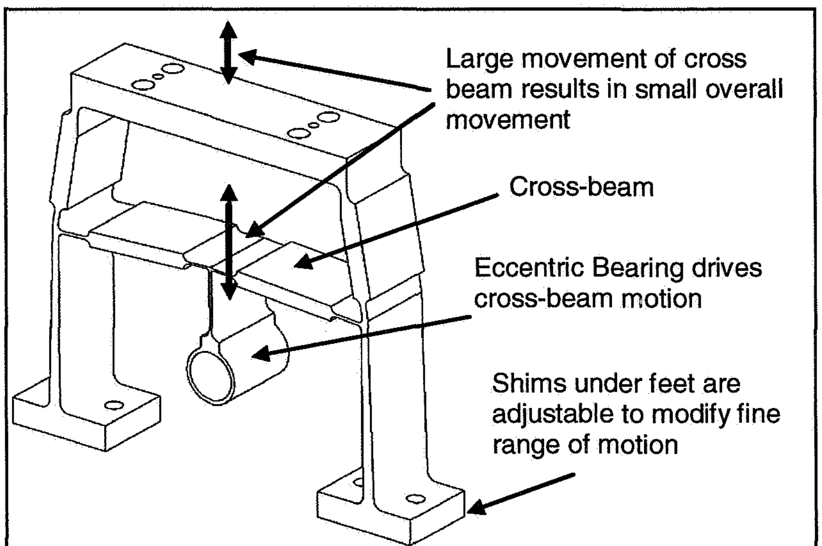


Figure 6. Simplified Compound Flexure

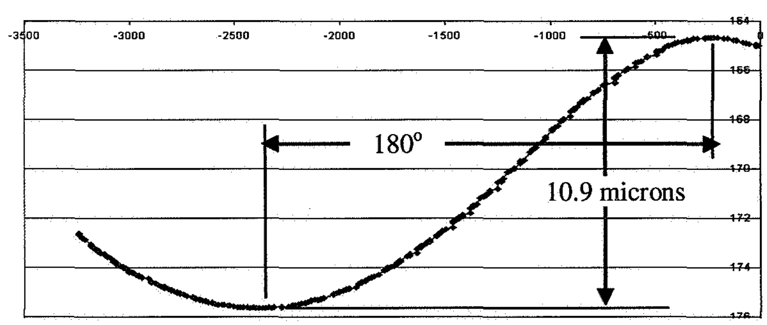


Figure 7. Fine Stage performance

Figure 7 shows an actual collection of data from the first fine range of motion test. The graph shows absolute height vs. motor steps and the distorted sine wave is clearly visible. Note that the fine range here is = 10.9 microns (0.0109 mm). Subsequent adjustment of the fine stage flexure resulted in a fine range of 10.5 microns.

Actuator Mechanical Description

The best way to describe the functioning of this mechanism is to follow the drive train. The major components in the actuator are shown in Figure 8 and in the diagram in Figure 9. All major components attach directly to the main housing.

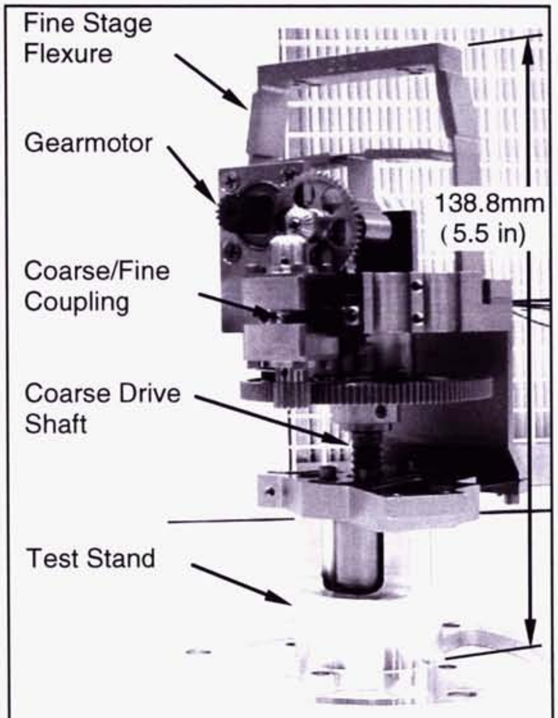


Figure 8. Actuator on Test Stand

Gearmotor

The drive train of the Actuator begins with a stepper motor. This type of motor is often used for positioning mechanisms because the rotation is divided into discrete steps. The motor for this application has 24 steps per revolution or 15 degrees per step. The stepper motor is attached to a 60:1 gear head and a resolver to form an integral gearmotor assembly. All bearings and gears are coated with a Ball proprietary dry film lube to enable operation at 30K.

First Pass to Cam Shaft

The output from the gearmotor uses a simple 3:1 spur-gear pass to drive the cam shaft. The cam shaft incorporates an eccentric bearing in the middle that drives the fine stage flexure shown in Figure 8. The shaft is supported by two simple bearings that are preloaded so that the shaft is always pulled away from the gear end. The cantilevered mounting of the shaft enables the use of 1:1 right angle bevel gears to change the drive axis from horizontal to vertical.

Second Pass to Coupling

The bevel gear pass connects to the coarse drive shaft through the coarse drive coupling as shown in Figure 5 described previously. Both sides of the coupling are supported by preloaded bearing pairs. Except for the gears, both sides of the coupling are identical to take advantage of commonality in fabrication. The drive side has a bevel gear and the driven side has a spur gear pinion attached.

Third Pass to Coarse Drive

The spur gear pinion drives a large ring gear resulting in an 8:1 ratio. The coarse drive shaft is an 8-mm ball screw with a 2-mm pitch that is mounted as a cantilever with two preloaded bearings at one end to attach the shaft to the main housing.

Other Components

The remaining actuator components are discussed later in the paper and include:

- Friction brake
- Torsional stabilizer
- LVDT position sensor
- Dry film lubrication

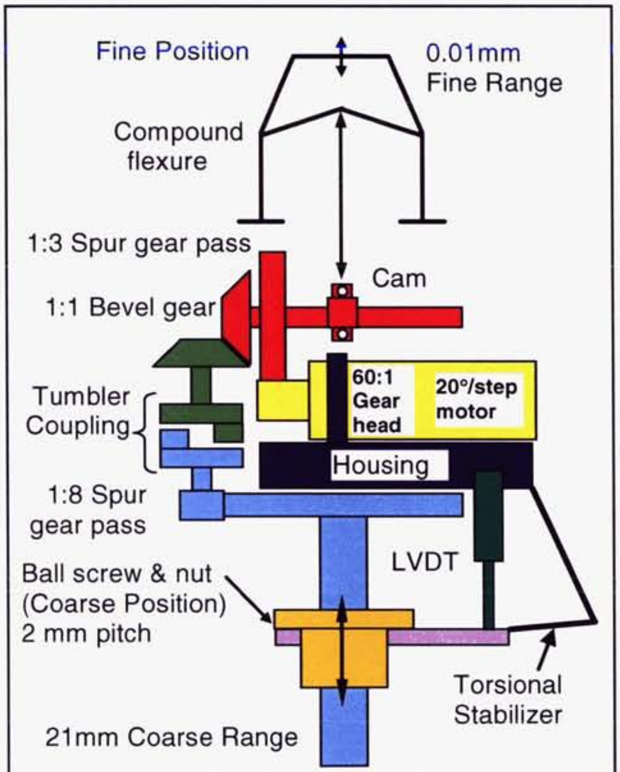


Figure 9. Schematic Diagram

Design Discussion

Figure 10 is a cut away view of the actuator showing the preloaded bearing pairs and gear passes. Standard recommended fits and tolerances were used. These are all standard mechanical configurations derived from catalog information.

<u>Dry Lube</u>
Operation at 30K prevents the use of liquid lubrication. Instead, dry film lubrication was applied to the moving surfaces of all bearings and gears. This Ball proprietary process as been used successfully on other cryogenic programs such as SIRTF and Hubble. Moving components can therefore be operated at ambient as well as cryogenic temperatures.

<u>Gearmotor Life</u>
The life requirement for the gearmotor was estimated using expected operational cycles for the life of the unit. A total of 1.7 million motor revolutions were estimated. The dry film lubrication in the gearmotor has been analyzed to last at least 3 million cycles.

Factor = spec/total = 1.75

Margin = spec/total -1 = 0.75

The gearmotor has been noted as a life limited item and motor revolutions must be recorded.

<u>Torque Margin</u>
The motor is a stepper type that rotates a precise amount for each step command. Torque margin calculations were generated to compare the output torque of the motor to all of the resistance torques including friction, inertia, coulomb drag and operational loads. Appropriate safety factors were placed on the various loads including cryogenic operation. The motor shows a positive margin under all conditions.

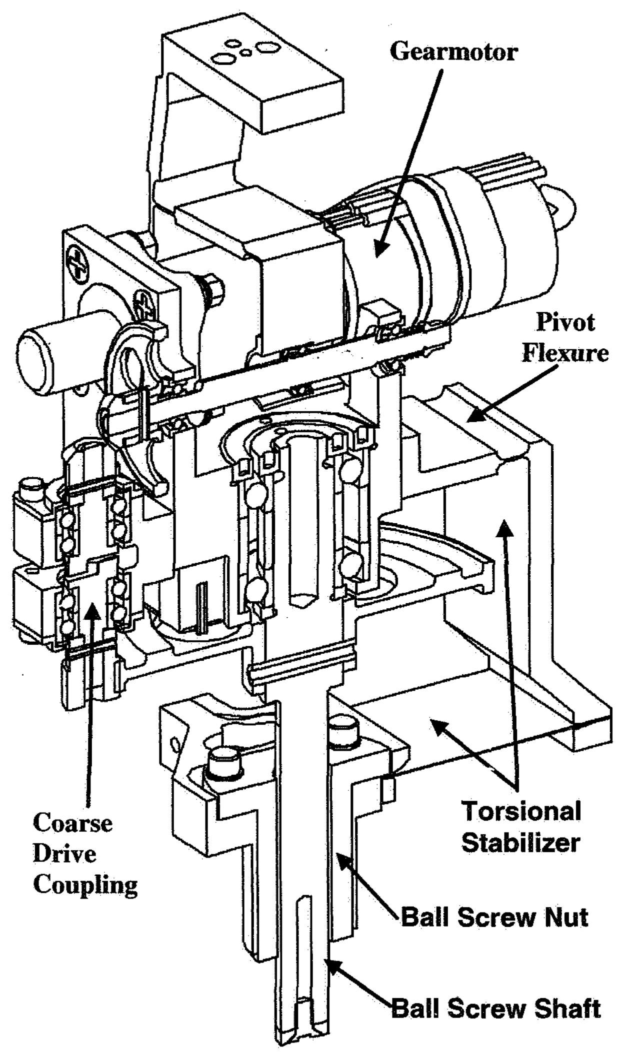


Figure 10. Section View

<u>Friction Brake</u>
A small friction brake was added to the actuator to prevent the coarse drive screw from back-driving operation. The axial force on the ball nut applies a torsional component to the coarse drive shaft. The coarse drive gear was chosen for the brake location because it is the largest torsional element on the coarse drive shaft thereby requiring the least force to constrain. The brake uses a double cantilevered beam to support and apply force to two Vespel buttons. The buttons slide along a raised surface on the coarse drive gear.

<u>Torsional Stabilizer</u>
In order for the actuator to apply pure axial motion, the nut on the ball screw must be constrained to the main housing in such a way as to resist torsional loads but to allow axial movement. The torsional stabilizer provides this constraint by incorporating a thin flexible shear panel that resists the torque applied to the shaft while allowing the ball nut to translate. As the nut moves up and down the shaft, the flexure forms an "S" shape. A pivot flexure that is opposite the panel flexure minimizes the radial loading on the ball nut from the stabilizer.

Gear tooth loading
The gear teeth were analyzed using a short cantilever beam formula from Roark. The three gear passes were analyzed using the 5 g load case. The stress levels for the three passes are as follows:

First Pass = 17.0 MPa (2.47 ksi)
Second Pass = 17.4 MPa (2.52 ksi)
Third Pass = 11.4 MPa (1.66 ksi)

All gears are made of titanium, which has a yield strength of 869 MPa (126 ksi). The minimum margin of safety is therefore 49.

Axial load requirements
Large axial loads are imparted into the actuator from two sources:

1. Launch Load:
   The defined acoustic load for one mirror is 6480 N
   The angle of the actuator is 15 degrees
   The load is shared between all 6 actuators
   6480 N / 6 / cos15 = 1118 N (251 lb) x 1.4 = 1565 N (352 lb).

2. Ground Test Load:
   The maximum axial force on a single actuator during ground test is 378 N (85 lb).
   A factor of 5 is put on this value to protect against damage during ground test operations.
   378 X 5 = 1890 N (425 lb).

Since the ground test load is greater than the launch load, it was used for design calculations.

Length
The length of the actuator was calculated based on a nominal deployed actuator length of 138.8 mm. The bipod assembly is required to retract 12.5 mm to the stowed position and extend 5.0 mm to the maximum deployed position. The actuator is also required to translate the bipod at the same piston, which requires additional length.

Launch Restraint
The original concept was to pull the mirrors down upon some mechanical hard stop until the motor stalls. There are two reasons why this concept was discarded: 1. The large force exerted by the actuator would require a huge increase in mass in order to withstand the stall forces. 2. When a stepper motor stalls, step count is lost.

Another concept was to incorporate some extra feature in order to stop the motor when a certain preload was reached. Several techniques were investigated and discarded including: load cell (adds extra wires, mass, electronics); slip or magnetic clutch (lose step count, adds mass); visual indicator (impractical for all but the outermost actuators). In all cases, the extra feature adds mass and complexity. Also, the hard stop must be designed to be loaded to some factor above launch loads multiplied by some safety factor.

The current launch restraint design is such that motion is restrained in the plane of the mirror and allowed in the direction perpendicular to the mirror. The actuators, therefore, constrain the motion of the mirror in the perpendicular direction. The actuators must withstand axial loads in both tension and compression.

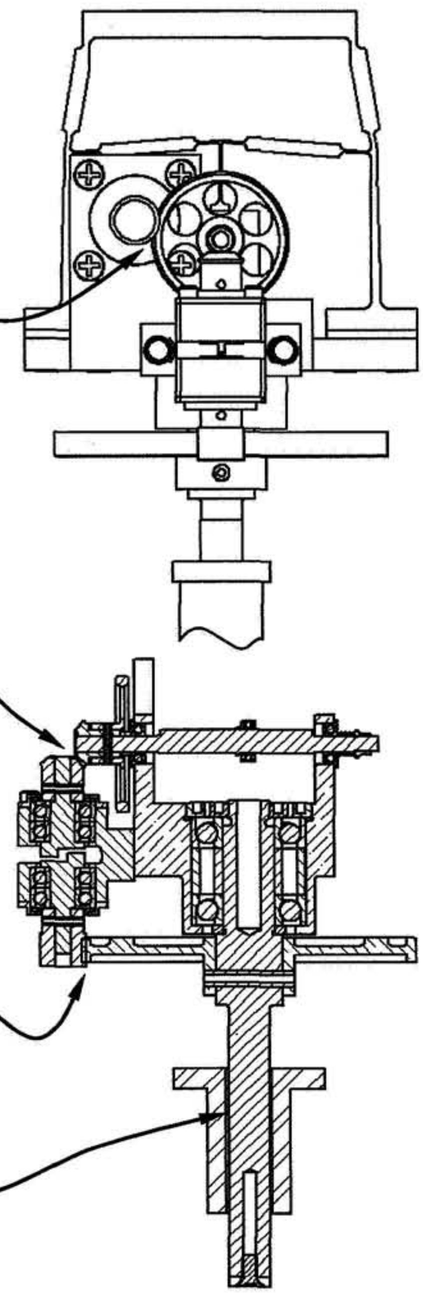
Power-train Summary
The stepper motor is integrated with a 60:1 planetary gearhead to form what is called the gearmotor. The output of the gearmotor drives a 3:1 spur gear in the first gear pass. The drive torque then branches to the fine stage and/or the coarse stage. The fine stage has a 1:1 bevel gear for the second gear pass to change the axis of rotation by 90 degrees to drive the fine stage flexure. The coarse stage has a tumbler type coupling that has 324 degrees of backlash before driving an 8:1 spur gear for the third gear pass to the coarse drive shaft. The coarse drive shaft is an 8-mm-diameter ball screw with a 2-mm pitch.

Power Train Capabilities

Because of the numerous gear passes, it was convenient to summarize the load at each step of the power train. Starting with the gearhead output torque, Table 2 lists the load at each step. The input torque is converted to a tangential force by dividing by the radius of the pitch diameter of the gear. The Pitch Angle is also factored in as well as the resulting load on the bearings. The tangential force is then multiplied by the radius of the driven gear to obtain the torque on the driven shaft. The torque on the shaft is then used as the input torque of the next pass. The calculations shown are for the 1 g operational load.

Table 2. JWST Power Train Calculations

First Pass Calculations		
Gearhead Input Torque	0.200 in-lb	0.02 N-m
Shaft Radius	0.166 in	4.22 mm
Tangential Tooth Load	1.205 lb	5.36 N
Camshaft Gear Radius	0.500 in	12.70 mm
Efficiency	0.950	0.950
Torque to Cam Shaft	0.572 in-lb	0.065 N-m
Second Pass Calculations		
Cam Shaft Input Torque	0.572 in-lb	0.065 N-m
Bevel Gear Radius	0.166 in	4.22 mm
Tangential Tooth Load	3.448 lb	15.33 N
Bevel Gear Radius	0.166 in	4.22 mm
Efficiency	0.950	0.95
Torque to Coupling	0.544 in-lb	0.061 N-m
Third Pass Calculations		
Coupling Input Torque	0.544 in-lb	0.061 N-m
Pinion Radius	0.156 in	3.97 mm
Tangential Tooth Load	3.481 lb	15.48 N
Coarse Gear Radius	1.250 in	31.75 mm
Efficiency	0.950	0.95
Torque to Lead Screw	4.133 in-lb	0.467 N-m
Lead Screw Calculations		
Lead Screw Input Torque	4.133 in-lb	0.467 N-m
Brake Drag	3.000 in-lb	0.34 N-m
Lead Screw Radius	0.164 in	4.17 mm
Ball Screw Ramp	0.077 in	1.95 mm
Axial Force	90.09 lb	400.7 N



Actuator Testing

Testing Options

Early in the program it became apparent that the testing of certain properties of the actuator could occur at several different stages of assembly as shown in Table 3. Because some of the components are life limited and because of the relatively large quantity of units involved, it was impractical to test every property at every level. Also, it was desirable to limit cryogenic testing because it is expensive and time consuming.

Table 3. Actuator Testing Options

Assembly Stage: Motor only, Motor & gearhead ambient, Motor & gearhead cold, Actuator assembly, ambient, Actuator assembly, cold, Bipod assembly, ambient, Bipod assembly, cold, Mirror hexapod assembly, ambient, Mirror hexapod assembly, cold.
Property to be tested: Motor torque, Fine range of motion, Fine step size, Fine accuracy, Coarse range of motion, Coarse step size, Coarse accuracy, Axial stiffness, Axial backdriving capability, LVDT output accuracy

Test Plan

After much discussion and trades, a testing plan was established to identify those properties that would be tested, and at what stage. It was desirable to test certain properties early in the process to determine any potential problems. The final test plan is shown in Table 4.

Table 4. Actuator Testing Plan

Assembly stage	Property to be tested
Motor only	Motor torque
Motor & gearhead ambient	Motor torque, gearmotor torque
Motor & gearhead cold	Motor torque
Actuator assembly, ambient	Motor torque, fine range of motion, coarse range of motion, coarse accuracy, Axial stiffness, Axial backdriving capability
Actuator assembly, cold	Motor torque, fine range of motion, fine step size, fine accuracy, coarse range of motion, coarse step size, coarse accuracy, LVDT output accuracy
Bipod assembly, ambient	Motor torque, coarse range of motion
Bipod assembly, cold	None
Mirror hexapod assembly, ambient	Motor torque,
Mirror hexapod assembly, cold	TBD

Actuator Test Results

To date, there have been four major test activities: ambient motor tests, cryogenic motor tests, ambient actuator tests and cryogenic actuator tests.

Ambient Testing

Figure 11 shows the combination test stand at the actuator fabricator. The fixture is used to measure axial stiffness, fine range of motion and coarse accuracy under ambient conditions. To measure the coarse accuracy, the length of the actuator was measured from stowed to deployed at every cam shaft revolution. The theoretical position was then subtracted from the measured position and the results are presented in Fig. 12. Note each point represents 0.25-mm axial movement. The accuracy requirement is 2% for this level of move, which equals 0.005 mm (5 microns). The test was performed in both the deploy (CW) and retract direction (CCW).

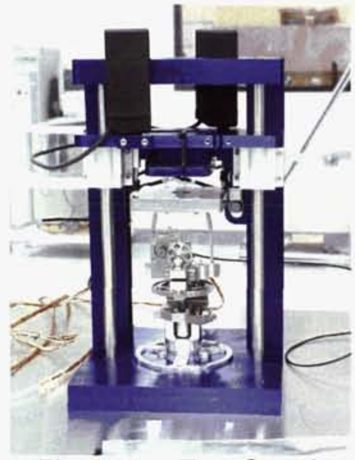


Figure 11. Test Stand

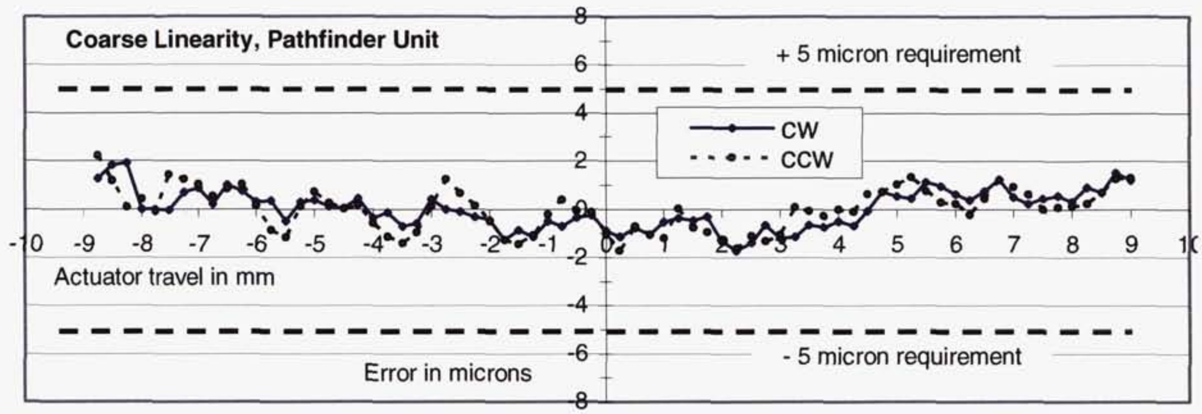


Figure 12. Coarse Linearity

Cryogenic testing

After the actuator was delivered to Ball, more accurate tests were performed to verify cryogenic operation and fine motion characteristics. Nanometer-level measurement testing requires interferometric instrumentation, vibration compensation and special software. Up to seven actuators can be tested at once using the special setup shown in Figure 13. One of the fine motion tests is single step repeatability. The results of this test are shown in Figure 14. The motor was rotated one step clockwise then one step counter-clockwise several times. The results show an average step size of about 7 nanometers and a repeatability of 1 nanometer.

Figure 13. Cryogenic Testing at Ball

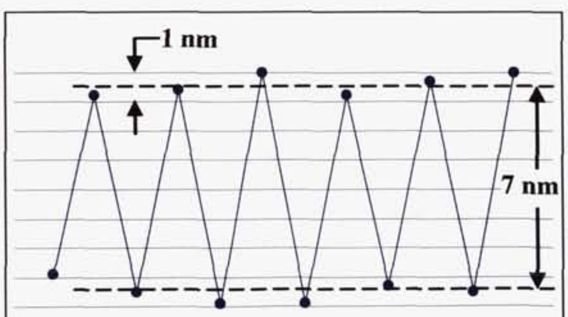


Figure 14. Single Step Repeatability

Lessons Learned

Mass production

One of the main lessons learned was that for mass production, it is worth some up-front time and expense to develop tooling and fixtures in order to save time later on. In order to more efficiently fabricate the 140 actuators, several mass production techniques were incorporated. For example, the application of dry film lubrication usually requires masking to prevent the dry lube from being applied to sensitive areas. This masking has traditionally been applied one unit at a time using tape or a painted on compound. For this program, however, special masking tools and holding fixtures were designed and built ahead of time to streamline this operation. Another aspect of mass production that we learned was that is well worthwhile to spend more at the machining stage to save time during the assembly stage. For example, match drilling for alignment pins is always problematic in titanium in a clean room. For this program the alignment holes were put in ahead of time while the parts were being machined. At assembly the parts went together very easily with no match drilling required as shown in Figure 15.

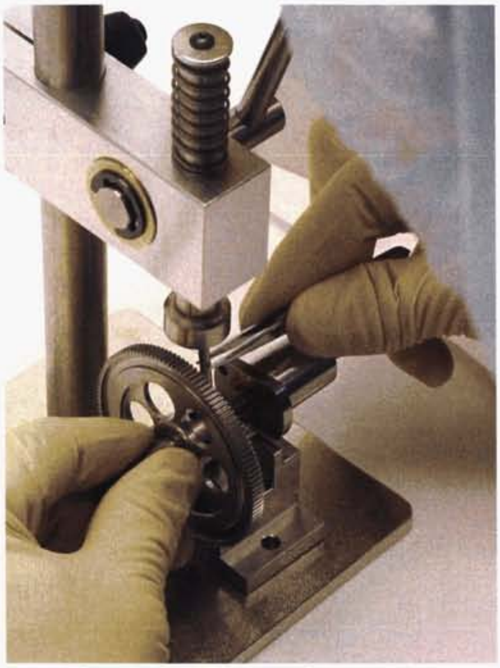


Figure 15. Production Tooling

Recast from EDM

Electro Discharge Machining or EDM was used to machine many of the critical areas on the actuator because it results in very good dimensional accuracy and repeatability. However, the EDM process produces a recast layer on the surface of the part. For a flexure, this recast layer can reduce the fatigue life of the part by up to 10 times. Multiple passes in the EDM process at Ball have resulted in a thin recast layer of about 0.0033 mm (0.00013 in), which is easily removed with light chemical etching.

For one particular flexure, however, standard etching did not remove the recast layer. Additional etching to remove the recast layer made the flexure area too thin. A review of the drawing and discussions with the vendor revealed a misunderstanding of the process. A specific EDM process sequence has since been defined, which results in a thin recast layer and acceptable flexure dimensions after recast removal. Inspection techniques have also been developed to simplify the verification of recast removal.

Thermal strapping

Titanium a poor thermal conductor and the gearmotor generates heat during operation, which affects the length of the actuator. In order to test the actuators in a timely manner, heat straps were attached to several points on the actuator. Again with mass production as a goal, special clamps were made so that the thermal straps could be easily attached and removed.

Pathfinder for the EDU

It is usually good practice to have an Engineering Development Unit (EDU) as a precursor to building a flight unit. JWST took advantage of this "lesson learned" that has been cited by many authors. The EDU for JWST consisted of 18 units to be used as follows: 7 actuators for the Primary Mirror Assembly, 6 actuators for the Secondary Mirror assembly, 2 actuators for a life test and 3 actuators for spares. With so many units to build, it was decided to turn one of the spare actuators into a "pathfinder" unit, which is essentially a first article development unit for the EDU. One actuator was taken from the initial build and designated as the pathfinder. This unit was the first to be assembled and the first to be tested. Any new testing or fixture was proven out on the pathfinder. To maintain schedule and to have processes verified in advance, certain shortcuts were allowed on this unit. For example, although the components were cleaned to a flight level, the pathfinder parts were not subject to an external particle count.

Conclusion

The design of the Cryogenic Nano-Actuator for JWST is now complete. The EDU units have been assembled and tested. The actuators will next be assembled into bipods, which will then be integrated into the primary or secondary mirror hexapod configuration. Long lead component procurement for the flight actuators is underway. Flight actuator fabrication is scheduled to begin in the last quarter of 2006.

The design of the actuator was extremely challenging but test results show that optical level positioning can be reliably achieved using simple mechanical components. The unique combination of the patented fine stage flexure with the coarse coupling proved to be quite effective in achieving fine accuracy over a long range of travel. Mass production techniques greatly simplified the design and assembly of the actuator. Reliable operation was achieved by the use of robust components and supporting analysis.

Acknowledgements

This paper presents the design of the Cryo Actuator for JWST. The design of the actuator is the result of important contributions from numerous people and organizations. I would like to acknowledge the contributions from those at Ball Aerospace including Robert Slusher, Scott Streetman, Lana Klingemann (nee Kingsbury), William Schade, Mike Matthes, Bruce Hardy and all the technicians, machinists, engineers and administrators. I would also like to acknowledge the outstanding work of outside vendors & suppliers including CDA Intercorp, All American Gear, New Hampshire Ball Bearing, Barden Bearing, ATK-Able, Next Intent, Schaevitz and Beaver Aerospace. The photographs in Figures 8, 11 & 15 are courtesy of Able. Finally, I would like to thank our customer, Northrop Grumman Space Technology (formerly TRW) and NASA Goddard.

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