Six degrees of freedom, sub-micrometer positioning system for secondary mirrors

Ryan C. Sneed*, Michael F. Cash, Trevor S. Chambers, Paul C. Janzen

CSA Engineering, 2565 Leghorn St., Mountain View, CA, USA 94043-1613

ABSTRACT

Secondary mirrors for large ground-based telescopes often require positioning systems with payload capacities around 1000 kg, relative accuracies within a few micrometers, and resonant frequencies above 15 Hz. A suitable six-legged parallel manipulator, or hexapod, has been developed for sub-micron level positioning of large optical payloads in six degrees of freedom. This 1000 kg class hexapod has tip/tilt rotational ranges of ±1800 arcsec, relative accuracies within 1%, and resolutions of better than ±0.2 arcsec, along with a piston translational range of ±30 mm, relative accuracy within 1%, and resolution of better than ±1 µm. The center of rotation of the system may be placed at an arbitrary location within the overall range limitations. The axial stiffness of each of the six actuators tested greater than 100 N/µm. The actuators use high precision roller screws and employ two degree of freedom universal end-joints. The preload on the joints eliminates backlash due to transitions from tension to compression and maintains friction moment of <10 Nm. An additional rotational degree of freedom is allowed in the body of the actuator to achieve the proper kinematic constraints for the motion platform. The actuators have power-off hold capability to protect against power loss and reduce heat dissipation. Overall heat dissipation has been measured and techniques have been studied to reduce its impact. The paper describes the actuator design and hexapod performance in support of planned use in ground test and validation of the James Webb Space Telescope.

1. INTRODUCTION

Large ground-based telescopes require precise alignment of secondary and tertiary optics with approximate scale of 1-3 m and 500-8000 kg. Similar requirements appear in applications for space telescopes and terrestrial primary mirror segment alignment or instrument positioning. A six-legged parallel robotic manipulator, or hexapod, can be an effective multi-axis positioning solution for these types of payloads.[1-6] Such a hexapod has been designed, built, and tested and is capable of sub-micrometer level positioning in six degrees of freedom.

Previous research[7] has compiled typical requirements for these precise positioning applications. A summary is presented in Table 1 with a focus on secondary mirrors for large ground-based telescopes.

Table 1: Summary of requirements for positioning of telescope optics with a focus on secondary mirrors for large ground-based telescopes

Payload	500-8000 kg
Range of motion	±10 to ±50 mm in translation ±2 deg (7200 arcsec) in rotation
Velocity	100-500 µm/s
Repeatability	±1 µm
Resolution	<1 µm or 0.1 arcsec
Relative accuracy (constant temperature)	1%
Resonant frequencies (with load)	15-20 Hz
Power dissipation	1-5 W
Attachment diameters	1-2 m
Operating temperature	-10 to 30 deg C
Operating pressure	Reduced atmospheric pressure at altitude (2000m)
Earthquake loading	0.7 g (horizontal and vertical directions)

While the payload mass range is significant, the other primary actuation criteria of range of motion, velocity, resolution, and accuracy are fairly consistent across various applications. This made it feasible to develop a hexapod strut (actuator) that will be appropriate for many similar precise positioning applications with limited or no modifications. The paper will begin with a discussion of the strut design and how the different elements determine the strut's ability to meet the performance requirements. A special focus will be placed on the end-joints since their properties were crucial to meeting the overall specifications. The environmental parameters will also be analyzed along with their impact on design and performance. Test results will be presented from a hexapod recently built for ground testing of the James Webb Space Telescope (JWST). The paper wraps up with conclusions and future applications.

2. STRUT DESIGN

Each of the six hexapod struts consists of a prismatic actuator element and a rotational joint on each end. The type of actuator used, including its sensing element, is crucial for meeting the performance requirements listed in Table 1, and the end-joints also contribute to the resonant frequencies via compliance, and to related specifications including friction and backlash.

2.1 Actuation

The actuation of each leg for this type of hexapod is usually accomplished with a power screw and a rotary motor. A large mechanical advantage, precise positioning, and self-locking are all possible with a power screw. In this case a recirculating roller screw was selected primarily because small leads of 1mm are possible. This allows for high accuracy (6 µm per ISO G1 specification) and improved resolution. Input torque is reduced due to the large mechanical advantage, and this, in turn, reduces heat dissipation and further enhances resolution. Recirculating roller screws also have many contact points resulting in high load ratings and extended life.

Backlash is generally considered unacceptable in this type of precision positioning application because load direction reverses from compression to tension (or vice versa depending on orientation) in at least one strut when the telescope zenith angle surpasses roughly 15 degrees.[7] A split-nut preload arrangement is used to eliminate all backlash at the screw/nut interface. Wipers may be used on both sides of the nut to prevent contaminants from reaching the interior of the nut.

Another consideration for nearly all telescope applications is that the struts must be self-locking (not back drivable). Power-off hold capability is necessary to protect the payload in the event of a power outage. A power screw will be self-locking (external torque required to lower the load) if $f > \tan \lambda$, where f is the coefficient of thread friction and λ is the lead angle of the thread.[8] With the coefficient of friction predicted by the manufacturer to be a minimum of 0.01 and a lead angle of 0.46 degrees, the recirculating roller screw selected for this application is self-locking.

Figure 1: Actuation for hexapod struts including harmonic drive / servomotor (partially concealed within gold alodine housing) and recirculating roller screw

The rotational motion for the roller screw is generated by a brushless DC servomotor combined with a harmonic drive gearing system. The 50:1 gear reduction ratio in the harmonic drive allows for high torques (500 N-m max) in a compact package size. The motor design eliminates backlash and can handle axial loads up to 39.2 kN. An absolute encoder can maintain position data during power failures and provides positioning accuracy of 40 arc-sec, repeatability of less than 6 arc-sec, and resolution of 3.2 arc-sec at the output flange. The maximum speed is 70 rpm although this may need to be reduced based on duty cycle and torque requirements. Even higher gear reduction ratios (100:1 and 160:1) are available to meet more demanding torque or resolution specifications where actuation speed can be sacrificed. For example, with the 160:1 reduction ratio the maximum torque and resolution can be improved to 820 N-m and 1.0 arc-sec, respectively, while the maximum speed will be limited to 22 rpm.

2.2 End-joints

To properly constrain a hexapod's motion, 36 kinematic degrees of freedom (DOF) are necessary[9]. Each leg has one active DOF provided by the linear actuators, and the remaining 30 passive DOF are usually incorporated into one 3-DOF and one 2-DOF end-joint on each hexapod leg. In this case an additional DOF is allowed at the rotation to translation interface of each strut by not using an anti-rotation device on the roller screw. This method therefore only requires 2-DOF joints at each end of the strut. Note that allowing rotation of the roller screw relative to the motor does result in some hexapod motion due to the associated translation of the screw. Fortunately, these errors caused by motion that is not sensed by the motor's encoder are small for the limited rotations necessary ($\pm 3^{o}$) and can be compensated for in software, if desired.

Ideally, the end-joints would have zero backlash, add no friction to the system, show no signs of wear, require no maintenance, and be infinitely stiff in the constrained degrees of freedom. In addition the ideal joint should have a center of rotation that is known exactly. Along with backlash, friction is a major enemy for end-joints. Unfortunately, eliminating backlash often requires the application of a heavy preload which results in large normal forces and high friction. Actuators can become unresponsive to small commands as a result of friction in hexapod joints and this, in turn, degrades resolution. Stick/slip type motion can result in impulse loading to the payload and deterioration of tracking performance. End-joint friction can also create moment loads in the strut. These moments then transfer into the optical support structure in the case of telescope positioning. The moments also increase wear on the roller screw which is designed for axial loading. End-joint friction is often thought to have an important benefit in reducing settling times. While this is true for large displacements, research has shown this advantage to be absent for small commands.[7, 10]

No practical joint exhibits all of the desired properties, but different physical realizations of the constraints have particular advantages and disadvantages. Fluid interface bearings are one such option. Hydrostatic and hydrodynamic bearings are typically not acceptable for optical applications where fluid contamination would be possible. Air bearings may be an acceptable option and can provide extremely low friction values. The disadvantages are complexity and cost. Flexure elements have appeal in that they are free of backlash and friction. However, their allowable strokes are generally small and their stiffness low. Flexures can also create significant moments in the strut, it may difficult to locate the exact center of rotation, and their failure modes are often catastrophic. Sliding contact bearings require velocity differentials at the contact surface(s) between two separate components. They can carry high loads with small package sizes, but the heavy preloads needed to eliminate backlash create high friction. Rolling contact bearings rely on point or line contact and trade lower friction for higher contact stresses. Comparable load ratings require larger bearings than with sliding contacts. However, rolling contact bearings can be preloaded to eliminate backlash without large increases in friction. For this reason and their high stiffness, rolling element bearings are considered the preferred choice for large telescope applications where the increased package size is not a major concern.

Rolling contact bearings include two major types: ball element and roller element. Ball elements have much lower load capabilities and stiffness and, thus, roller elements are favored. Tapered roller bearings are a convenient choice for a 2-DOF universal joint because a radial preload can be establish from preloading in the axial direction.

Figure 2: Custom 2-DOF universal joint uses tapered roller bearings and closely approximates an ideal end-joint for telescope positioning applications

A survey of off-the-shelf universal joints (using rolling contact bearings or otherwise) with the required properties did not reveal any suitable options. As a result, custom units were designed and built as shown in Figure 2. Two tapered roller bearings are used in parallel for each rotational axis of the joint. The joints can handle static loads up to 88 kN since each bearing carries half of the strut load and is rated for 44 kN. Dynamic load capacity for the joint is reduced to 72 kN, but hexapod speeds are slow enough to consider conditions as static. The rotational range is ±11degrees in both axes. Shoulder bolts are used to preload all bearings to 10 kN providing a radial stiffness of 1620 N/µm for each bearing. Stiffness can be increased even more by using a higher preload (1720 N/µm with 15 kN), but the reduction in expected L-10 bearing life (from 2,082,000 hours to 556,000 hours) outweighs the benefit. Based on the multiple bearing arrangement (two pairs of parallel bearings in series) and finite element analysis estimates for the stiffness of the other components, the overall stiffness of the joint is predicted to be 490 N/µm with an L-10 joint life expectancy of approximately 700,000 hours or 80 years. The friction in the joints is very low with a moment less than 2 N-m when preloaded with no external load. Large external loads are not expected to increase the moment substantially.

These custom universal joints closely approximate the ideal characteristics described above. They have no backlash, low friction, high stiffness, and very long life expectancy at the low speeds of the hexapod. The center of rotation is repeatable and well known within the machining tolerances of the joint. Periodic maintenance may be required for grease redistribution since the bearings do not make complete revolutions.

2.3 Complete strut

The complete strut including the harmonic drive, servomotor, roller screw, and end-joints has a mass of 59.2 kg and an axial load limit of 39.2 kN. A good rule of thumb is the maximum payload for a hexapod can be approximately four times greater than the allowable strut load. In this case, the potential payload mass could be up to 16,000 kg though application of safety margins and usable life considerations would reduce this value in practice. The maximum stroke is 160mm which can be adjusted for different applications, but the range of the absolute encoder does constrain the stroke to 164mm with a 1mm screw pitch. Software limits, limit switches, and hard end stops are all used to prevent the strut from over-travel. The maximum speed of the strut is 1.17 mm/s, but the speed must be reduced to 0.5 mm/s for continuous operation. The maximum speed and stroke can be increased by using a larger roller screw pitch, but accuracy and resolution will suffer accordingly.

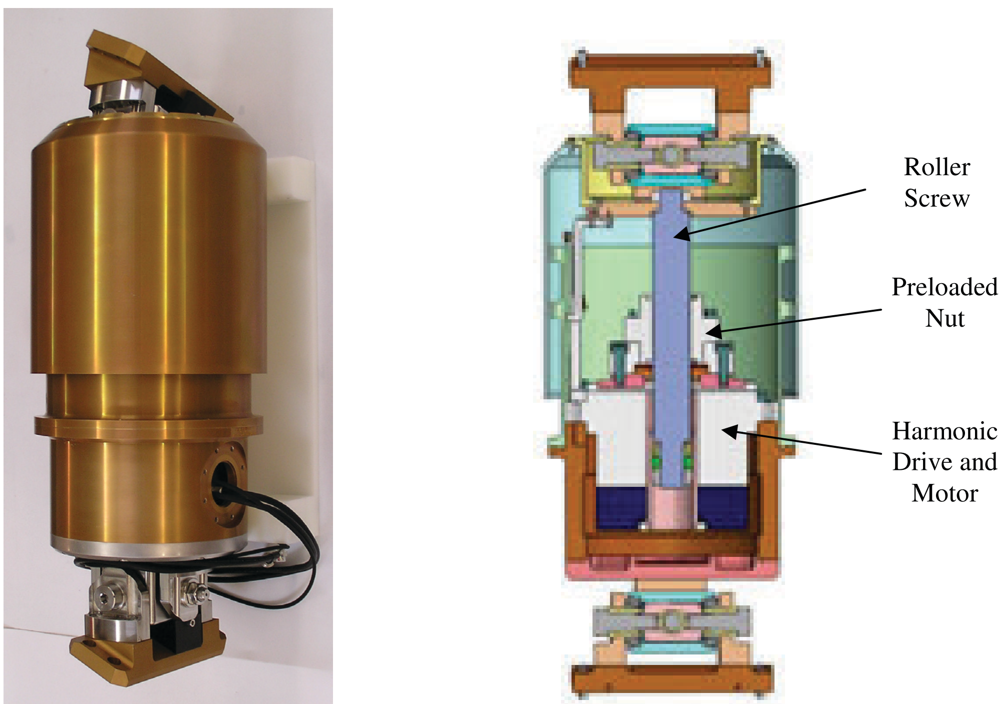

Figure 3: Photograph of hexapod strut (left) and cross-sectional view (right)

Based on the roller screw pitch, harmonic drive reduction ratio, and encoder properties, the theoretical strut resolution could be as low as 2.5 nm. However, harmonic drive windup and hysteresis effectively reduce the resolution to 26 nm. The relative accuracy of the strut is less than 0.5% of the commanded move assuming constant temperature with screw lead precision being the dominant source of errors. The strut stiffness can be estimated as the combination of springs in series representing each of the component stiffness values: two end-joints (490 N/µm each), two joint end plates (1678 and 1350 N/µm), roller screw (1441 N/µm), roller screw nut (1395 N/µm), harmonic drive (1400 N/µm), and motor housing (6000 N/µm). These component stiffness values were determined from finite element analysis or provided by the manufacturers. The net stiffness prediction for the strut is 130 N/µm. In the space test application, assuming a rigid body payload mass of 650 kg plus another 85 kg for the payload attachment platform, this strut stiffness results in hexapod resonant frequencies of 38.6, 38.6, 116.5, 116.8, 123.8, and 133.4 Hz.

3. ENVIRONMENTAL SPECIFICATIONS

The environmental specifications will vary for each optical positioning application, but the strut design can accommodate a wide range of environmental conditions. The following sections discuss several important environmental parameters and their potential impact on strut performance and compatibility.

3.1 Operating temperature

The struts were tested within a smaller operating temperature range (13-23^{o} C) than may be required for other applications. The major concern is on the low end of the potential temperature range (-10 to 30^{o} C) because the harmonic drive/motor combination is rated for 0 to 40^{o} C. Additional testing and/or discussions with the manufacturer would be necessary if the operating temperatures drop below freezing. Alternatively, heating elements could be included to maintain the strut temperature within the allowable range, or a thermal stabilization fluid may be circulated to the struts. Moderately elevated temperatures (~30^{o} C) are not viewed as a major problem but may reduce the allowable duty cycles. The greases used on the roller screw and end-joint bearings have allowable ranges in excess of the expected conditions, but viscosity changes could increase friction or wear.

A wide temperature range will also have an effect on positioning accuracy due to thermal expansion. If the associated errors are unacceptable, strut length compensation could be performed through the control software. The tolerancing on several close fitting parts will also need to be reassessed in light of coefficient of thermal expansion mismatches.

3.2 Operating pressure

Reduced atmospheric pressure at high altitude locations will not cause significant problems with strut functionality, but duty cycles for the motor may need to be reduced to account for less heat dissipation in the thinner air. The strut was developed and tested for a vacuum chamber application where the exterior of the strut was exposed to 10^{-3} Torr and the interior was left open to atmospheric pressure. In non-vacuum applications the pressure seals still act as effective barriers against contamination whether that means potential contamination of the strut interior from outside elements or vice versa.

3.3 Earthquake loading

Earthquakes and the resulting loads (horizontal and vertical directions) placed on the hexapod are realistic occurrences. The load magnitudes are not the major concern as they can be designed for and are typically lower than the loads associated with shipping on a truck (up to 3g vertical, 1.75g horizontal). However, in addition to the stresses imparted to various components, high enough vibration levels will eventually cause a power screw to move. An electromagnetic brake may be used to provide supplemental protection against back driving in this situation. The brake engages when de-energized since a power outage may accompany a large earthquake.

3.4 Power dissipation

Power dissipation is an important concern for certain optical applications. The amount of heat that can be dumped into the environment may be limited to a few Watts. For a typical operating case shown in Figure 4, the average power consumption of the strut is 75 W. Only a small fraction of this electrical input power is converted into mechanical output power, and the majority will be dissipated as heat to the environment. Some of the power loss comes from friction in the roller screw, but the majority results from losses in the motor and harmonic drive. In addition energizing the electromagnetic brake to disengage during operation requires 6W. This results in an estimated power dissipation around 500W for the entire hexapod.

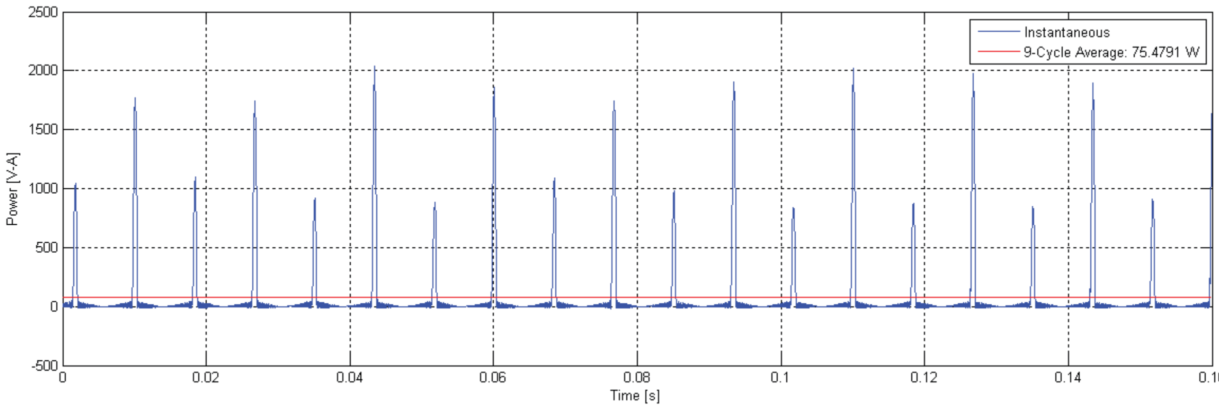

Figure 4: Power consumption of a hexapod strut extending at 0.25 mm/s under a 2220 N (500 lbf) compressive load

For applications where this level of heat dissipation would be unacceptable, modifications to the operation of the hexapod could result in significant reductions. Limiting the actuation speed can reduce the average power dissipation to a minimum level of 19W per strut with no motion. Lighter payloads would also result in less dissipated power, but the improvements will be modest compared to those achieved with speed reduction. Some applications may be able to limit operational duty cycles to 1% thereby slashing the heat dissipation to roughly 5 W. The brake would be de-energized and power to the struts cut during the non-operational periods. If none of these options are compatible with the application, air cooling of the strut interior is possible given the seal design of the strut. The exhaust air could be routed away from the hexapod to avoid potentially undesirable air currents.

4. TEST RESULTS

While the strut performance properties are crucial for meeting requirements of a given application, the overall hexapod properties are of primary interest to the end user. The following section presents test results for the range, speed, resolution, and accuracy of a hexapod developed for use in ground testing of the James Webb Space Telescope. Strut stiffness is also presented below because the resonant frequencies of the hexapod are a function of the strut stiffness and the payload mass. Life cycle testing performed on a single strut is summarized as well.

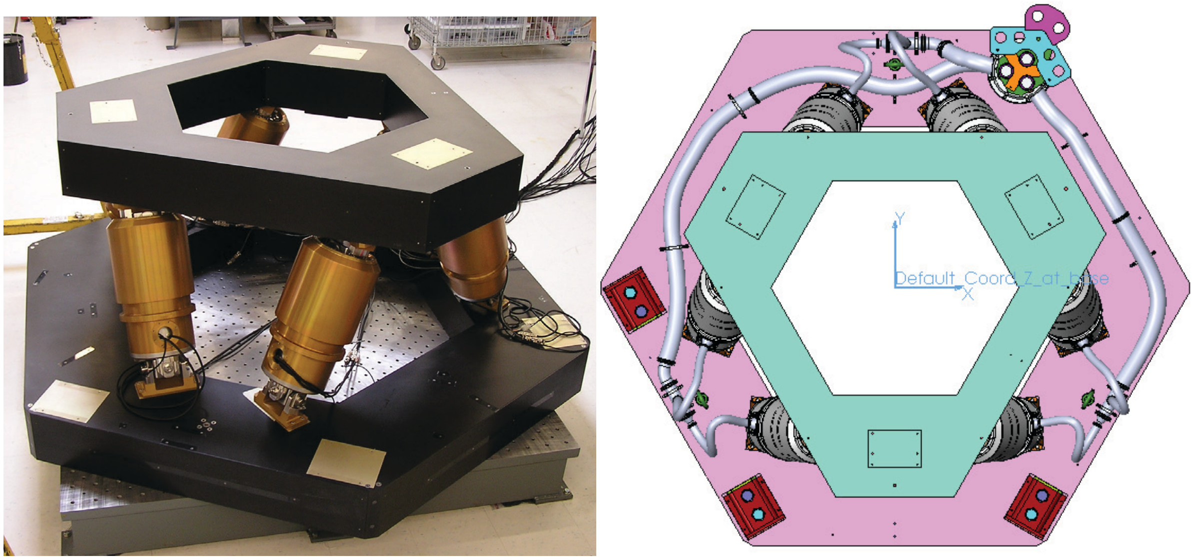

Figure 5: Photo of hexapod for JWST program (left) and solid model image showing coordinate system (right)

4.1 Hexapod level performance specifications

The achieved performance of the hexapod as verified by testing is listed in Table 2. All hexapod level tests were performed with a 925 kg payload in the arrangement depicted in Figure 5.

Table 2: Hexapod level performance specifications verified by testing

	Range (+/-)	**Maximum Speed**	**Resolution**	**Relative Accuracy Error**
Translations:	**[mm]**	**[mm/s]**	**[mm]**	**%**
X	148	0.925	< 0.001	< 1%
Y	155	0.975	< 0.001	< 1%
Z	84	0.500	< 0.001	< 1%
Rotations:	**[arcsec]**	**[arcsec/s]**	**[arcsec]**	**%**
Θx	28,200	185	< 0.2	< 1%
Θy	26,950	165	< 0.2	< 1%
Θz	47,000	300	< 0.2	< 1%

The x and y-axes represent horizontal translations and the z-axis represents piston or vertical motion as shown in Figure 5. The rotational specifications assume the center of rotation is located at the top center of the payload attachment platform. The ranges shown in Table 2 represent the maximum travel in each axis with no displacement in any of the other axes, i.e. the range for single axis motion. Simultaneous travel in multiple axes will reduce the allowable ranges, as will shifts in the center of rotation location. The maximum hexapod speeds allow for continuous operation with relatively light payloads (< 1000 kg). The speeds could be increased if the operational duty cycles are low, but must be decreased if the loads are heavy.

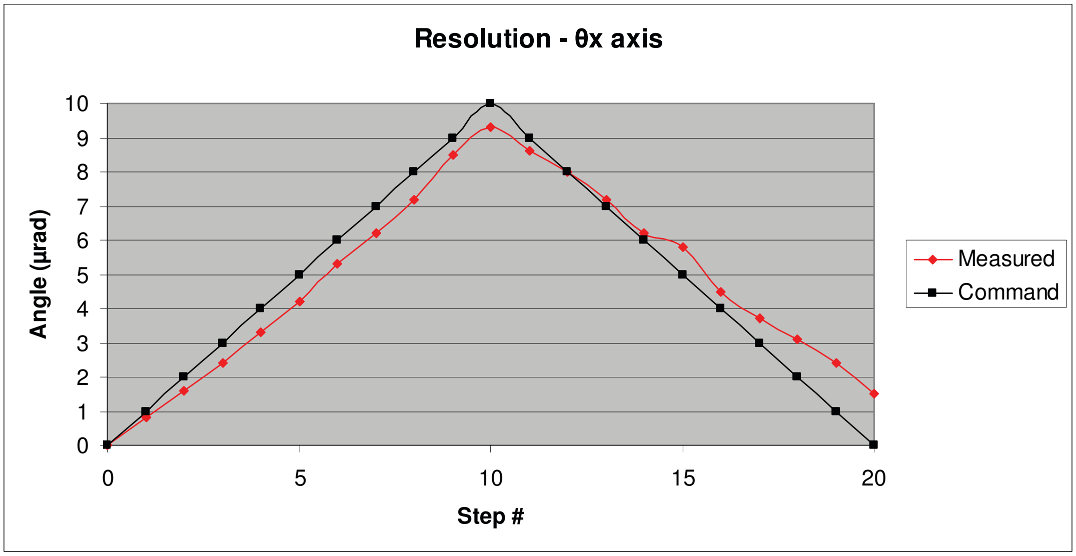

Figure 6: θx-axis resolution testing results for 1µrad step size

The hexapod is capable of making incremental moves of 0.2 arcsec (1 µrad) in the rotational degrees of freedom, as shown in Figure 6, and 0.1 µm in the translational degrees of freedom. The minimum resolutions of the system are predicted to be even smaller, but noise and drift in the external metrology equipment present challenges for validation. The system also maintains relative accuracies within 1% in all six axes. Accuracy testing results in Figure 7 show that incremental and cumulative errors are approximately 0.5% for 100 µm moves in the y-axis.

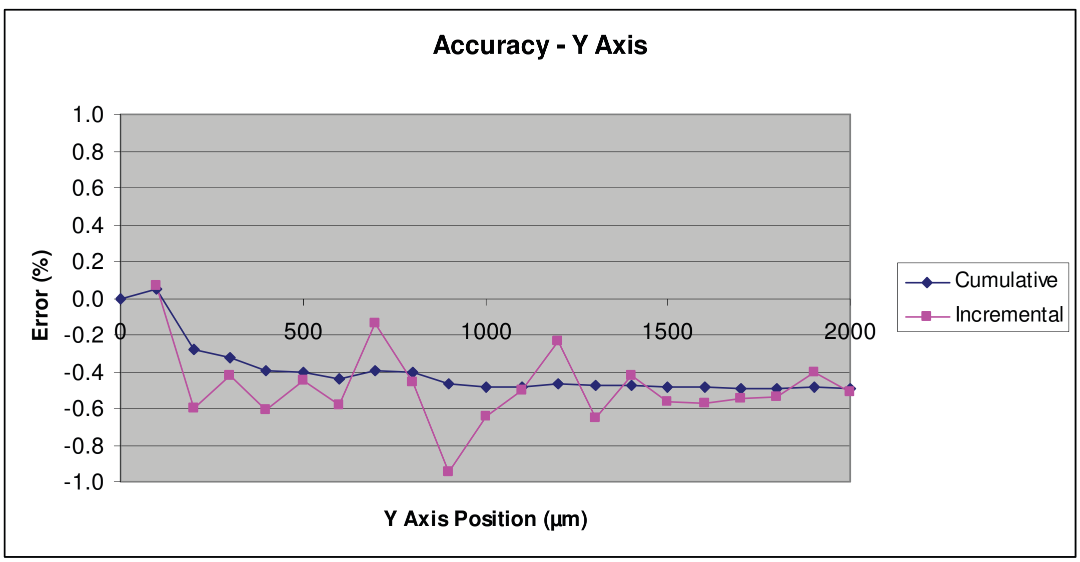

Figure 7: Accuracy testing results in the y-axis showing cumulative and incremental errors for 100 µm moves

The uni-directional repeatability is better than 0.5 arcsec in the rotational axes and better than 2 µm in the translational axes. The bi-directional repeatability measured less than 3 arcsec in the rotational axes and less than 15 µm in the translational axes. Cross-talk, or non-commanded movement in one axis resulting from commanded motion in another axis, varies significantly by axis. Some combinations of axes show virtually no cross-talk while piston translations (z-axis) typically result in the largest off-axis motions. For 1 mm and 300 arcsec commanded moves, the cross-axis motions are generally limited to 5 µm and 1 arcsec.

4.2 Strut stiffness

The strut stiffness is predicted to be 130 N/µm based on estimates for each of the individual components. Testing showed these estimates to be slightly conservative as the average strut stiffness was 156.8 N/µm. For a payload mass of 650 kg (assuming rigid body) and a payload attachment pattern mass of 85 kg, the lowest suspension frequencies are expected to be 42.4, 42.4, 127.9, 128.3, 135.9, and 146.5 Hz.

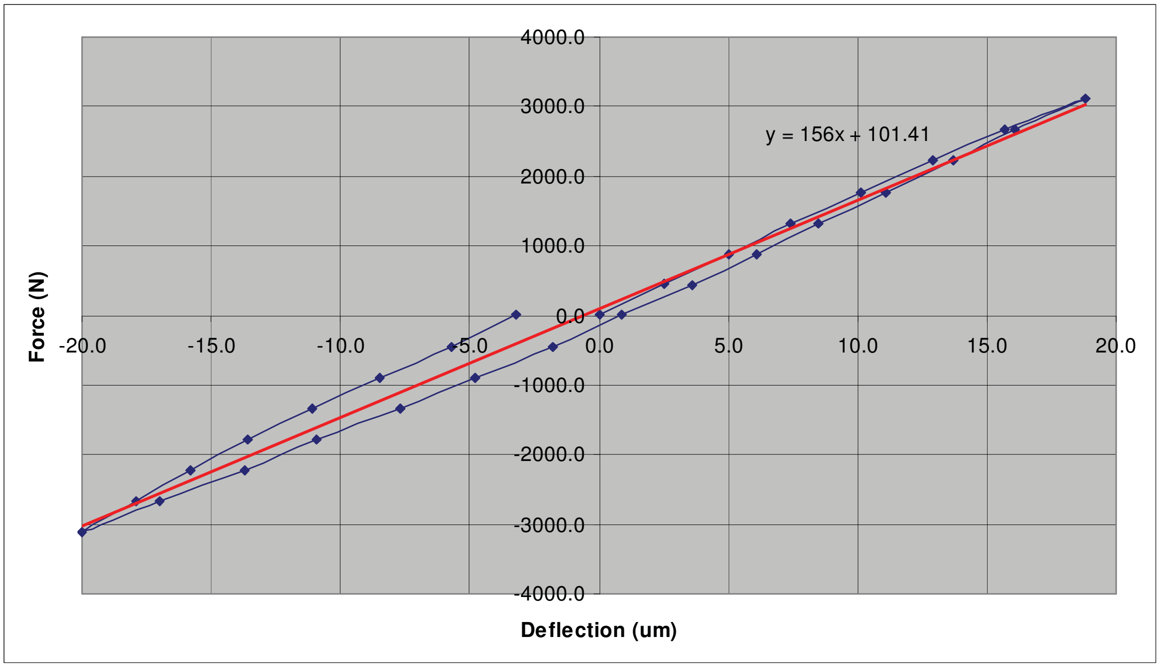

Figure 8: Measured force vs. deflection for a representative hexapod strut. Strut stiffness was approximately 156 N/µm

4.3 Strut life cycle testing

Life cycle testing was performed to verify the strut is capable of high reliability over a long time period. A strut was cycled continuously from full extension to full retraction. Approximately 300 cycles were completed at a speed of 0.5 mm/s under a 4500 N load, and roughly 1300 cycles were completed at a speed of 0.25 mm/s under a 2225 N load. This amounted to over 500 hours of continuous run time. Post-life cycle performance testing showed no major degradation in accuracy, resolution, or stiffness compared to pre-life test results.

5. CONCLUSIONS AND FUTURE POSSIBILITIES

A high accuracy, high resolution strut has been developed that can be incorporated into hexapods for precise six DOF positioning. The strut uses an appropriate combination of brushless DC servomotor, harmonic drive with a 50:1 reduction ratio, and a high precision recirculating roller screw for actuation. Custom universal joints using tapered roller bearings closely approximate an ideal joint for these applications with no backlash, low friction, high stiffness, and long life. The strut design also permits flexibility for different application requirements where trade-offs must be made between range, speed, resolution, accuracy, and operational duty cycles.

The paper has also emphasized that the design has proceeded from concept design and prototype development through pilot verification testing. A full-scale hexapod system has been built and tested and will soon be delivered as ground test equipment for the JWST. Common environmental specifications have been considered and largely addressed. Test data was presented demonstrating the hexapod has the performance necessary for other sub-micrometer positioning applications such as secondary and tertiary optics for large ground based telescopes. Secondary mirror positioning for the Thirty Meter Telescope and Cornell Caltech Atacama Telescope are two future possibilities that appear well matched to the hexapod's capabilities.[4,11, 12]

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REFERENCES

[1] Schmidt-Kaler, T., "The hexapod telescope, a new way to very large telescopes," Progress in Telescope and Instrumentation Technologies, 117 (1992).
[2] Pernechele, C., Bortoletto, F. and Reif, K, "Hexapod control for an active secondary mirror: general concept and test results," Appl. Opt. 37, 6816-6821 (1998).
[3] Chini, R., "The hexapod telescope, a never-ending story," Reviews in Modern Astronomy 13: New Astrophysical Horizons, 257-268 (2000).
[4] Anderson, E., Cash, M., Janzen, P., Pettit, G., and Smith, C., "Precision, range, bandwidth and other tradeoffs in hexapods with application to large ground based telescopes," Proc. SPIE 6273, 62731F (2006).
[5] Carrington, C. K., Davis, B.G., "The application of linear actuators for telescope pointing control," Proc. AAS/AIAA Astrodynamics Conference, 581-600 (1991).
[6] Pitz, E., Rohloff, R., Hippler, S., Wagner, K., Marth, H., "Five-axis secondary system for UKIRT", Proc. SPIE 2199, 516-522 (1994).
[7] Cash, M., Bruch, D., Jahn, B., Keas, P., "Practical considerations of joint friction and backlash in large ground-based telescope secondary optic positioning systems," Proc. SPIE 7018, 70181T.1-70181T.12 (2008).
[8] Shigley, J. and Mischke, C., [Mechanical Engineering Design], McGraw-Hill, New York, 450-454 (2001).
[9] Anderson, E., Cash, M. Hall, J., Pettit, G., "Hexapods for precision motion and vibration control," Proc. ASPE Control of Precision Systems (2004).
[10] Jahn, B., "Simulation, design, and testing of a high-performance multi-axis hexapod for vibration isolation," Proc. ASME IDETC/CIE (2007).
[11] Nelson, J. and Sanders, G., "Thirty Meter Telescope (TMT) status report," Proc. SPIE 6267, 626728 (2006).
[12] Sebring, T., Giovanelli, R., and Radford, S., "The Cornell Caltech Atacama Telescope (CCAT): a 25-meter aperture telescope above 5000 meters altitude," Proc. SPIE 6267, 62672C (2006).