A Low Power Cryogenic Shutter Mechanism for Use in Infrared Imagers

D. Scott Schwinger* and Claef F. Hakun*

Abstract

This paper discusses the requirements, design, operation, and testing of the shutter mechanism for the Infrared Array Camera (IRAC). The shutter moves a mirror panel into or out of the incoming light path transitioning IRAC between data acquisition and calibration modes. The mechanism features a torsion flexure suspension system, two low-power rotary actuators, a balanced shaft, and a variable reluctance position sensor. Each of these items is discussed along with problems encountered during development and the implemented solutions.

Introduction

IRAC, developed by Goddard Space Flight Center, under agreements with The Smithsonian Astronomical Observatory and the Jet Propulsion Laboratory, is 1 of 3 scientific instruments on board the Space Infrared Telescope Facility (SIRTF) to be launched in December of 2001. SIRTF is the fourth "large observatory" performing imagery, photometry, and spectroscopy of astronomical bodies over the spectral range of 3.6 to 160 μm. The IRAC instrument takes images in 4 bands centered on the following wavelengths: 3.6, 4.5, 5.8, and 8.0 μm. The major objectives of IRAC are to produce the following science data sets:[1]

- Deep confusion-limited broadband (25%) surveys for high-redshift normal galaxies
- Large area shallow surveys for brown dwarfs
- Imaging surveys of star clusters to search for brown dwarfs
- Surveys of nearby stars for brown dwarfs and superplanets
- Photometric observations of selected ultraluminous galaxies and active galactic nuclei
- Imaging surveys to identify protoplanetary disks and young stellar objects

Due to its use of state-of-the-art large format infrared detectors, IRAC will be more sensitive and produce more compelling, higher resolution images than previously possible with cryogenic imagers.[2]

A top view of the IRAC instrument is shown in Figure 1. A pickoff mirror reflects light from the focal plane of the spacecraft telescope off of two mirrored surfaces at slightly different angles. The result is a transmission of two separate light paths into the top and bottom compartments of the IRAC structure. Each compartment has a full set of refractive optics and two near infrared detectors positioned almost symmetrically about the mid-plane of the structure.

Following the light from the pickoff mirror and into the instrument, it first travels through the aperture of the shutter. The shutter is the focus of this paper and will be addressed extensively later. After the shutter, the light travels through a doublet lens to a beam splitter. Here the beam is split into two separate wavelengths. The longer wavelengths are transmitted and the shorter wavelengths reflected. After the beam is split, each subsequent beam passes through a filter and a lyot stop and onto a detector array. There are 4 of these arrays, one for each of the stated wavelengths, each 256 x 256 pixels and covering a 5.12 x 5.12 arcmin field of view. Two detectors are Arsenic-doped Silicon (Si:As) IBC arrays and the other two are Indium Antimonide (InSb) arrays.

Along with the optics and the detectors, there are two on board calibration systems: a transmission calibrator and four flood calibrators. Each of the four flood calibrators is attached to a detector mounts and projects light directly onto the arrays. The intent is to flood the array with a known infrared source and confirm an expected level of response across the entire array. The transmission calibrator is designed to send a known source through all of the optics and onto the detector arrays. The transmission

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* Code 544, NASA – Goddard Space Flight Center, Greenbelt, MD

system consists of a calibration sphere that produces a uniform source of known intensity. When the shutter is in the closed position, the light from the transmission calibrator is relayed through the optical train to the detector arrays. With the calibrator system off and the shutter closed, a dark level calibration of the detectors can be obtained.

IRAC, along with the other instruments aboard SIRTF, is a cryogenic instrument thermally coupled to a superfluid Helium dewar. Therefore, the base temperature of the instrument is expected to be 1.4 Kelvin. The mission lifetime requirement is set at 2.5 years with a goal of 5. At that time, the cryogen will be expended and the instrument chamber is expected to rise to 30 Kelvin. Two of the detectors, the InSb arrays, are still expected to work in the post-cryogen state further extending the useful life of the instrument. The shutter mechanism is expected to continue to work, as well.

The Shutter Mechanism

The requirements, design, operation, and testing of the IRAC shutter mechanism are described. The shutter has been developed over a three-year period. During that time, two ETUs were built and tested. One of these units remains in the IRAC Instrument Demonstration Module (IDM) which was built for engineering testing and proof of concept. In addition, three flight units have been designed and built and are currently in various stages of test. Two of the assembled flight units have been tested and characterized at liquid helium temperatures and one of these units has been delivered to the IRAC flight cold assembly. The test data taken includes actuator characterization, position sensor characterization, repeatability of mirror position, optical metrology, and excess actuator torque.

Functional Description

The IRAC shutter (Figure 2), approximately 0.15 m (5.9 in) in length, is designed to move a 7.98E-02 m by 3.71E-02 m (3.14 in x 1.46 in) mirror to two positions identified as open and closed. The motion is rotational and sweeps a total arc of 38 degrees (Figure 3). When the shutter is opened, the mirror is stowed out of the incoming light path. Therefore, the instrument is able to receive light from the focal plane of the spacecraft telescope and take images of the outside sky. When the shutter is closed, the outside light is to be attenuated by 1.0E06 providing a dark environment in the instrument for infrared detector calibration. To achieve this, the mirror is translated directly into the light path completely filling the aperture and interfacing with a baffle in the IRAC housing. The result is the light path being completely blocked with very little leakage around the mirror. The inside portion of the mirror, itself, provides a reflective surface off of which the transmission calibration source is reflected. The source then passes directly through the optical path of the instrument and onto the infrared detectors.

Requirements[3, 4]

Spacecraft:

- Stiffness must assure fundamental frequency > 50 Hz
- Must survive launch loads generated by a Delta II H launch vehicle
- Uncompensated momentum shall not exceed 200E-06 N-m-s
- Shall operate at 1.4 Kelvin for mission lifetime and 28 Kelvin for extended mission

Mechanism:

- Mass not to exceed 1.6 kg
- Shall translate mirror to two positions: Open and Closed, with 0.5 deg repeatability
- Time to close shall not exceed 10 seconds
- Shall be failsafe open
- Lifetime operations shall be 20,000 cycles
- No microphonic transmission beyond 0.5 second transient
- Maximum power dissipation shall not exceed 0.5 J/100 s (5 mW average)
- Attenuate incoming photon radiation by ≥ 1E06
- Redundancy:
  - There shall be no electrical single point failures
  - Minimize mechanical single point failures

Electromechanical design and operation

A cross-section of the IRAC shutter is shown in Figure 4 and an exploded view is included as Figure 5. To facilitate the description of the component parts of the shutter mechanism, they will not necessarily be discussed in order of their assembly with the unit. The description will start with the suspension system of the mechanism, move to the actuation system, continue on to the structure and stress limiting components, and conclude with the sensory components.

SUSPENSION SYSTEM

The shaft assembly (Figure 6) provides the suspension system for the mirror panel. The shaft, itself, is fabricated of Aluminum 6061-T651. The central hub, the most robust portion of the shaft, is designed to hold the mirror and counterweight. The mirror panel is Aluminum and has two diamond-turned surfaces coated with gold. An arm extends from this panel and mates to a flat surface machined into the shaft's hub. Opposite the mirror, a 0.1058 kg Tantalum counterweight is attached to the shaft. This counterweight minimizes the CG offset of the shaft which minimizes moments under 1G and vibration.

The shaft also supports the two rotors of the actuators and two A286 Steel bushings. A square hole in the center of each rotor slides axially over the shaft and mates with a matching square profile. As a result, each rotor is fixed in rotation such that relative motion between the parts is not possible. The rotors are fixed axially by a nut on a threaded portion of the shaft at either end. This threaded portion also supports steel bushings that provide bearing surfaces for rotation and act as the radial centering device for the shaft. The bushings fit into holes in the end caps of the mechanism such that there is only 3.81E-05 m (0.0015 in) clearance radially. The mating surfaces in the end caps are plated with Teflon-impregnated anodize to minimize friction.

Finally, the shaft is machined to allow a Beryllium Copper torsion flexure to run down the central axis. This component provides the axial stiffness required to locate the shaft assembly and provides the restoring torque to reopen the shutter when power is terminated. The flexure is chemically etched out of 6.60E-04 m (0.026 in) thick BeCu 25AT sheet and is heat-treated at 315.6° C (600° F) for 3 hours to obtain the following strength and fatigue properties at 4 K: $\sigma_{ult}/\sigma_{yield}/\sigma_{fatigue}$=220/190/110 ksi. It has two active portions, 6.60E-04 m (0.026 in) square and 5.21E-02 m (2.05 in) long, and three flanges for attachment to other components. The middle flange is attached to the central hub of the shaft. Two Aluminum end pieces are bolted to the end flanges such that the two ends of the flexure can be fixed in rotation to the end caps of the mechanism. As the shaft rotates with the ends fixed, restoring torque is generated in both sides of the flexure to fight this motion. This restoring torque reopens the shutter when the power is off.

ACTUATORS

The shutter contains two actuators built around each of the rotors on the shaft (Figure 7). These actuators are electromagnetic variable reluctance rotary devices capable of providing 45 degrees of angular motion, though the shutter only requires 38 degrees of rotation. They are cylindrical in design and consist of a rotor, two stators, an electromagnetic coil, and a magnetic "closeout" cylinder. These devices contain no permanent magnets. Hiperco 50A, heat treated to allow high magnetic flux density with little hysteresis[5], provides a closed path for magnetic fields generated by an electromagnetic coil. All components of the actuator except the bobbin and the coil, itself, are machined of this material. The active portions of the stators are radially contained within the bobbin. The coil is 99.99% pure 38 AWG, HAPTZ insulated copper wire precision wound on a black anodized Aluminum 6061-T651 bobbin to 11,000+ turns. Passing current through the coil generates a magnetic field that is focussed by the stators and flows through the rotor. The coil was sized by analysis to provide an N·I value greater than 600 with 60 mA. To complete the magnetic circuit, a closeout cylinder covers the coil and connects the two stators. The moving portion of the actuator, the rotor, is positioned between the two stators and is rigidly affixed to a shaft that runs down the central axis of the device. This shaft maintains the radial position of the rotors within the actuators and transfers rotary motion to the mirror.

Rotary motion is produced by the geometry of the device. The footprint of the rotor is best described as a bow tie. There are raised portion of the stators that match the profile of the rotor exactly, though the footprint of the base of the stators is circular to mate with the closeout. The stators face each other when the actuator is fully assembled. There is just enough separation of the stator faces to allow the rotor to pass between them with 3.30E-04 m (0.013 in) clearance on both sides. The rotational position of the rotor in the open state is such that its footprint is out of phase with those of the facing stators except for a approximately 30 degrees of overlap at one side. In this position, the rotor is at a state of high reluctance in the magnetic circuit. When current is applied, the rotor wants to achieve a state of minimum reluctance. Thus, the rotor is drawn between the stator faces. This motion produces a torque and rotates the shaft and mirror to the closed position.

A final component to the actuator design is paramount to achieving the 5 mW requirement. That component is a magnetic latch that extends vertically from each of the stator faces. These "tabs" are positioned and sized such that the rotor contacts the tab on each of the two stators when it is fully rotated to the closed position. With the rotor against these tabs, the magnetic flux can travel directly from stator to stator through the contacting rotor and on to the magnetic closeout. The rotor effectively completes the magnetic circuit, reducing all air gaps to nearly zero, producing a large drop in magnetic reluctance. This drop in reluctance means that the shutter can be held closed, fighting the restoring torque of the torsion flexure, with a much lower magnetic field. Since the field is electromagnetic in origin, the reduction in field means a reduction in current and less power dissipation in the coil. The result is a shutter that requires 55 mA to close and less than 1 mA to hold closed.

Of further benefit to achieving low power dissipation is the fact this mechanism operates at liquid Helium temperatures. The result is a reduction of resistance of the electromagnetic coils from 2700 ohms at room temperature to 20 ohms at 4.2 Kelvin, a factor of 135 change. This change has been verified during liquid Helium testing. Since the shutter closes in approximately 0.5 second, the control electronics sends a high current pulse, then quickly drops the magnitude to a hold current value. By including a position sensor (described later) to resolve the position of the shutter mirror, the shutter electronics immediately know when the shutter is closed and time spent at the higher pull-in current is minimized. Time averaged power dissipation can then be calculated using Equation 1. Assuming a typical shutter closure duration of 500 seconds and including margin on the current levels, the inputs to the equation are as follows:

Pull-in Current	-	$I_{pull} = 60$ mA
Hold Current	-	$I_{hold} = 3$ mA
Pull-in Current Duration	-	$t_{pull} = 0.5$ s
Hold Current Duration	-	$t_{hold} = 499.5$ s
Total resistance -	-	$R_{tot} = 20\ \Omega$

$$\boxed{P_{ave} = \frac{I_{pull}^2 \cdot R_{tot} \cdot t_{pull} + I_{hold}^2 \cdot R_{tot} \cdot t_{hold}}{t_{pull} + t_{hold}}}$$

Equation 1 – Time Averaged Power

The result is a shutter mechanism that dissipates 252 μW, a factor of 20 less than the requirement.

STRUCTURAL AND STRESS LIMITING COMPONENTS

The housing is a continuous piece of Aluminum 6061-T651. Pockets are machined into each end along the main axis to house the actuators. A gap in the structure, between the two actuator pockets, allows the rotation of the mirror attached at the central hub of the shaft. End caps on either end of the shutter housing contain the actuators within the pockets and provide support structure for several components on either end of the flexure. The actuators are preloaded against these end caps by a wave spring in the bottom of each actuator pocket. This preload maintains the position of the actuator components during launch and provides axial compliance necessary for CTE mismatched parts when going cold.

A hole in the center of the end cap provides a bearing surface for the steel bushings on the ends of the shaft. The end cap is plated with a Teflon-impregnated anodize to provide some lubricity during actuation. Precision concentricity is maintained in the position of these holes in the end caps, the interface of the end caps to the housing pockets, and the relationship of the centers of each of the pockets to ensure that the axis of rotation of the shaft is centered.

The square cross-sectioned end piece attached to the flexure in the shaft is designed to protrude through the end cap where it engages a component with a square hole, the Spring Preload Ring. This component is designed to be rotated until the proper torsional preload is imparted to the flexure. This preload is necessary to return the mirror to the open position when power to the actuators is terminated. In addition, this preload helps to ensure that the mirror does not flap uncontrollably during launch. The preload rings are set, clamped in place during preliminary assembly, and pinned to the end cap during final assembly.

To anchor the flexure axially to the end caps, a steel nut is run onto threads in the flexure end pieces. The threads extend through the square hole in the preload ring allowing the nut to effectively anchor the flexure to the end cap/preload ring stack. The nut on one side is run onto the threads completely to the preload ring while the nut on the other side is run down onto a stack of wave springs. In addition to affixing the flexure to the end caps, these nuts allow axial adjustment to ensure that the rotors attached to the shaft are in the proper location for optimal performance of the actuators. The springs under one of the nuts provide axial preload for the flexure and maintain a desired preload as the structural components shrink going cold. Since the CTE of the Beryllium Copper flexure is 25% less than that of the Aluminum of the structure from room temperature to 4 K[6], the preload and the rotor position would be lost without the springs. This preload is necessary to maintain the rotor position during operation and launch.

A hard stop further limits the loads transferred to the flexure during launch. This component is Aluminum 7075 plated with Teflon-impregnated Anodize for lubricity. It provides a rotary stop for the mirror in the open position. In addition, it has two uprights that straddle the mirror arm throughout its entire range of travel. These uprights are positioned such that there is a 2.03E-04 m (0.008 in) gap between each upright and the mirror arm. This gap is sufficient to allow freedom of movement for the shaft but to restrict axial motion in the event that launch loads excite the suspended mass of the shaft components. Restricting the axial motion is necessary to prevent stress in the flexure from violating material strength limits and ensure that the rotors do not impact against the stators in the actuators.

Finally, a retaining device is mated to the end cap to capture the stack of rings at the flexure interface. The gap between the top of the stainless steel nut and the underside of the retainer is less than 1.27E-04 m (0.005 in). This gap ensures that in the event of a breakage in one half of the flexure with the mirror in the closed position, the axial motion of the shaft in the opposite direction will be less than the gap between the mirror arm and the hard stop uprights. Therefore the restoring torque of the remaining half of the flexure will be enough to open the shutter without friction on the hard stop.

SENSORY COMPONENTS

There are three sensory components on the shutter: 2 Cernox resistance thermometers and a variable reluctance position sensor. The Cernox thermometers are calibrated to operate from room-temperature (300 K) down to pumped liquid Helium temperature (1.4 K). One of these sensors is located on each of the end caps. They will be used to monitor temperature on orbit and to verify power dissipation during component level testing.

A variable reluctance rotary position sensor was developed to monitor the position of the mirror. The sensor is an electromagnet in the shape of a "C." There are two electromagnetic coils wired in series, one on either side of the opening. Each coil has greater than 300 turns of 99.99% pure 38 AWG, HAPTZ insulated copper wire. The bobbins for the coils are part of the magnetic circuit and are made of Hiperco 50A heat-treated to optimize performance. To resolve position, a magnetically conductive ramp is passed through the gap in the sensor. As this ramp passes between the poles, the change in thickness of the material changes the air gap of the circuit. This change in gap changes the reluctance of the circuit. The reluctance change is measured by passing an AC signal through the coils and measuring the change in inductance of the circuit. The ramp is curved, bonded to the counterweight, and sized to be at a minimum thickness when the shutter is open and at a maximum thickness when it is closed. The raw response of the sensor is approximately 1.5 V across the full range. This response is then scaled to 1.25 to 3.75 V from open to closed position, respectively. Since the ramp is continuous, it can provide continuous position data for all intermediate positions of the shutter mirror, as well.

Under normal operation, the position sensor will provide feedback to the electronics to minimize the power dissipation in the actuator coils. The sensor position will be continuously checked after the actuator is sent the 60 mA pull-in current level. Once the sensor reads a threshold indicating the shutter is closed, the electronics will autonomously drop the current to the hold level of 3 mA. The delay from the time the sensor reaches the threshold to the drop in current is 1.0E-03 sec. When the shutter is given the signal to open, the sensor verifies the open position and returns this signal in a data packet prior to completely powering down the shutter mechanism.

UNIQUE OPERATION

There are additional unique characteristics of this mechanism. As stated above, the shutter is designed to fail open if there is a flexure breakage. However, the shutter is also designed never to close again if this failure should occur. The internal forces of the actuators (discussed in the Problems section) are large and the rotor is in an inherently unstable position. Therefore, without a continuous flexure under tension, the rotor is pulled away from the failure side and large frictional loads are generated either between the rotor and the stator in the actuator, or between the mirror arm and the hard stop upright. This fiction is enough to prevent the actuator from closing.

Finally, the choice in flexure suspension system combined with the bearing surfaces as designed cause the internal forces to approach zero when the shutter is powered off and the mirror is returning to the open position. The flexure allows us to avoid conventional ball-bearing designs which are difficult for cryogenic usage and always have some level of internal friction. In addition, the fatigue properties of BeCu increase dramatically at liquid Helium temperatures. We have performed a life test on an ETU and have driven the mechanism to greater than 450,000 cycles, 22.5 times the rated life.

The bushings on the shaft are designed to have 3.81E-05 m (0.0015 in) radial clearance with the hole in the end caps. Careful alignment and tolerancing is performed to center the bushing within this hole as precisely as possible and limit the contact between the two parts. However, due to operation under 1 G and physical limitations in the precision of the machining, these parts do touch. As the shutter is actuated, the internal loads of the actuators pull the shaft off axis and these bearing surfaces rub. Under this loaded condition, there are frictional losses at these interfaces due to the off axis normal load in the bearing interface. However, with power off, this load goes to zero as does the friction force. Therefore, the frictional force fighting the opening of the mirror is minimized. This effect will be realized to an even greater extent in the zero-G environment of space.

FUNCTIONAL TESTING

The shutter has been successfully tested cold many times. These tests included cold functional, position sensor repeatability, life operation for the ETU, infant mortality for the flight, and electronics board check-out. The performance during each of these tests was repeatable and consistent. In addition, life tests were completed with no apparent degradation in performance.

One additional test was in a dewar with a window to take optical metrology on the shutter mirror. Using a theodolite, we were able to measure the mirror angle relative to a starting position. This was our first opportunity to characterize the position sensor cold. The window prevented us from being able to achieve temperatures colder than 28 K, but the operation of the shutter at temperatures below 10 K had been previously verified. In addition, the shutter is actuated with a current drive system that compensates for any variance in actuator coil resistance.

The performance of both the actuator and the sensor proved to be repeatable. Mirror angles were measured at several different current inputs. At each of these current levels, the voltage output from the sensor electronics was measured, as well. The results are included here as Figures 8 and 9. The response of the position sensor is second order with regard to the mirror angle (Figure 8). Figure 9 indicates that the actuator needs approximately 42 mA before overcoming the torsional preload of the flexure. Above 42 mA, the mirror will rotate according to the plot until the current reaches 53 mA. At this time, the attractive force of the actuator tabs overwhelms the restoring force of the flexure and causes the mirror to quickly rotate to the closed position. The speed of this motion makes angle measurement difficult in this regime.

Several current input levels were repeated to verify the repeatability of the mirror position with current. The open and closed positions of the mirror were repeatable to less than 30 arcsec. However, we found that if current is stepped up to a particular level, the angle achieved by the mirror is slightly less than if the current level is input as a step function from zero. The difference in angle ranges from 0.05 to 2.2 degrees depending on location within the range of motion. There were not enough data points taken to achieve a statistical sampling. However, it seems apparent that the internal off-axis forces in the actuators are being transferred to the bushings on the shaft in the form of static friction in these intermediate positions. Since the coefficient of kinetic friction is always lower than the coefficient of static friction, the force fighting the rotation of the bushings will be lower for a moving bushing than for a static bushing under the same load. In addition, the rotational momentum of the shaft accelerating to a greater angular velocity under a step impulse rather than a slow increase may overcome more of this friction and complete a greater angle prior to stopping. Data is currently being reviewed to quantify frictional forces.

Testing has also quantified excess torque as a function of current and mirror angle (Figures 10 and 11). The test was performed at room temperature and measured the minimum excess torque in the motor to be 1.415E-02 N·m (2.004 in·oz) with a 60 mA input. Finally, the failure behavior of the shutter was tested on the second flight spare. The ability for the shutter to open itself if a flexure breakage occurred was verified; however, this ability is highly dependent on the gap between the flexure nut and the retainer on the end cap. If the gap is too large, the mirror arm will bear on the hard stop upright and the subsequent friction between the two parts will prevent the shutter from opening. If there is no contact between these parts, half of the flexure is still adequate to open the shutter mirror. With no axial force on the flexure (i.e. a broken flexure), the shutter is designed not to close. This behavior was also confirmed by this test. Finally, the shutter has survived a qualification level vibration test and a post-vibe life test is pending.

PROBLEMS AND SOLUTIONS

During the development of this mechanism, several problems were encountered that required solutions. This section identifies the major problems and the rationale behind the solutions.

Suspension System

During the initial concept of this mechanism, we wrestled with how we could achieve 38 degrees of rotation most efficiently and repeatably at cryogenic temperatures. We wanted to avoid ball bearings because of the difficulties involved with sizing the bearings warm to achieve the proper fit cold and with lubrication at liquid Helium temperatures. In addition, our motion was not going to be continuous and we were concerned about degradation of performance over time due to displacement of bearing lubricant. A flexure seemed logical, but what should it look like?

We initially thought of flex pivots but found that the most rotation we could get out of one was on the order of 15 degrees. This would mean that we would have to gang 3 of these pivots together to achieve our 38 degrees. Our concern with doing this was losing our center of rotation during operation and maintaining structural rigidity to survive launch loads.

We then thought of using a shaft pointed on each end that engaged a jewelled cup. We thought this configuration would be a relatively low-friction interface that would maintain its center and still allow full rotation. A clock spring would provide our restoring force. This concept seemed sound but we became concerned about the wear of the jewelled cup after launch and throughout the life of the mission. In addition, the cups would have to be axially preloaded to maintain contact as the mechanism went cold.

This preload would also have to be sufficient to maintain the position of the cup during launch. The design quickly became more complicated than we anticipated and was abandoned.

We finally returned to the flexure idea and decided to investigate a torsion flexure design. The concept is elegant in that the restoring torque and the axial position of the mirror can be obtained in one component. In addition, there is some heritage for this torsion flexure design in the Scanning Mirror for Infrared Sensors[7] developed by Lockheed Missiles and Space Company. Packaging the torsion flexure required a hollow shaft and sizing the flexure for stress/strain and restoring torque was simply a matter of a parametric trade study. The axial preload of the torsion flexure was established based on predictions for launch loads and axial forces expected within the actuator. Launch loads were predicted using conventional finite element analysis. Internal actuator forces were determined using a boundary element magnetic analysis program called Amperes. The axial load was finalized by investigating combined torsional and axial stress in the BeCu flexure and confirming that sufficient margin could be obtained.

Internal Actuator Forces

Axial preload in the flexure is needed for several reasons, not the least of which is to maintain rotor position under internal forces within the actuator. The initial design had a nominal gap of 4.52E-04 (0.007 in) between the rotor and each of the stators. The rotor is in an unstable position between the stators and is pulled in the direction of the portion of the magnetic circuit that is closest. Therefore, there are both axial and radial forces on the shaft within the actuator. These forces grow by 1.751E05 N/m (1000 lb/in) of misalignment. We knew that the position of the rotors between the stators would have to be precise but we did not fully appreciate the problem until we first tested the prototype.

Initially, radial and axial motions of the shaft under these forces were accommodated by a bushing inserted into the clearance hole in the base of the actuator cavities in the shutter housing. The inside of the bushing was a toroid to limit friction radially. Axially, it had a bearing surface that the shaft hub could contact if the axial forces during operation or launch became too great. This bushing was Aluminum 6061-T651 and plated with Teflon-impregnated anodize to limit friction. The problems we discovered with this design were two fold.

First, there was very little "wheel base" as these bushings were both near the hub of the shaft. We could not limit the off-axis forces enough to prevent the shaft from tilting and the rotors from clamping against the stators. The second problem was that, to limit the axial motion, the axial clearance between the bushing bearing surface and the shaft hub had to be so small that assembly became difficult. In fact, under the initial testing we discovered that the axial forces were great enough to cause the shaft hub to clamp against the bushing seizing the rotation.

We first tried to correct the problem by increasing the axial tension in the flexure. Ultimately, we were able to make the actuator work, but not before we encountered another problem. The initial design of the shutter housing did not have the structural member between the two actuator cavities. Therefore, as we continued to increase tension in the flexure, the housing actually deformed and the gaps between the bushings and the shaft hub closed. We needed to increase tension on the flexure to maintain position of the rotors and in the process, we were deforming the housing and causing alignment problems. To correct this problem, we added a structural member that later became part of the housing design. We were then able to increase the flexure tension enough that the actuator worked.

To reduce the radial motion, we changed our bearing concept from a radial bushing capturing the shaft near the center of mass to extending the shaft and adding a precision bushing that would ride on the inner diameter of the end caps. Ultimately, this added "wheel base" and attention to concentricity sufficiently limited the tilt of the shaft. While examining the axial forces and motion of the shaft, we determined that the tolerances used to determine the actuator gap were insufficient. Therefore, we precisely measured each of the components that impacted the axial position of the stators and rotors. A rigorous CTE analysis of the shutter design to simulate the changes to the components at 2 K was developed[8]. The spreadsheets have been used for each of the shutter units developed to date and can quickly be modified for specific part dimensions.

As a result, we were able to precisely locate the rotors between the stators. With this knowledge, we were able to move away from the philosophy of limiting the axial motion due to misalignment forces and toward the philosophy of limiting the axial forces by precisely locating the rotors. In addition, we modified the actuator design to double the gaps to 3.81E-04 m (0.015 in) between the rotors and the stators. This increased gap also reduces the sensitivity of the mechanism to the forces generated by the magnetic circuit.

Consequently, we were able to lower the tension on the flexure to 111.2 N (25 lb) at room temperature. Since we improved the bearing design, began precisely measuring components, and reevaluated the cold motion, we have been 100% successful with our assemblies and cold tests.

Launch and Operational Loads

The initial design of the flexure was a parametric trade study of torsion, stress, and geometry. We also decided to limit the rotational loads the shutter would experience with an unbalanced cantilevered mirror in a vibration environment. Therefore, we counterbalanced the mirror with a mass of Tantalum. The result was a shaft that weighed 2.22N (0.5 lb). However, the design was completed before our launch load environment was fully defined. A coupled load analysis identified that the shaft could experience loads of 100 Gs. This was not anticipated and we became concerned about the vibration load the flexure would experience due to the increased mass of the balanced shaft. The FEA on the shutter confirmed that the flexure would yield under these loads.

We looked into increasing the size of the flexure, but due to schedule restrictions we decided not to change the flexure design. Next, we looked at removing the counterweight to reduce the axial loads on the flexure. However, without the counterweight, the mirror would rotate wildly in the vibration environment. Although the axial loads would be less, the combined loading of the rotational stress with the axial was still enough to yield the flexure. The value of the counterweight was worth the added mass. We had to limit the loads another way.

We implemented a launch-load stop to capture the mirror arm throughout its full range of motion. It had to capture the mirror arm near its base due to high loads. The design resembles a goal post in football and was added as a component of the hard stop. We chose Aluminum 7075 as the material due to the high loads. Finally, the part was plated with Teflon-impregnated anodize to minimize wear and friction. The gaps between the uprights and the mirror arm were precision controlled to be 2.03E-04 m (0.008 in). This gap would sufficiently limit the axial motion of the shaft while not allowing the rotors to impact the stators in the actuators. The component has worked fine to date and has survived cold vibration testing and supports the fails safe open requirement.

Position Sensor

The development of the position sensor proved more difficult than expected. The initial design was a simple "C" shaped magnetic circuit with one large coil between the two poles (Figure 12). As stated in the description, the ramp was simply to pass through the gap between the two poles that were slightly tapered to focus the flux and fit within the shutter design. The change in the reluctance of the circuit as the ramp passed between the poles would be measured electronically. Theoretically, the design was sound. However, when we tested the prototype, we only got a response of 30 mV full range. In fact, it was difficult to measure any response at all over the noise in the electronic circuit.

Upon conducting a rigorous analysis using the Amperes modeling software, it became clear that there were large losses and fringe effects within the magnetic circuit due to the geometry. The fringe effects in the circuit act as magnetic shorts and reduce the effectiveness of the sensor. Since we only had one coil with a magnetic pole attached at either end, we generated relatively large surface areas of opposite potential facing each other. As a result, the magnetic flux simply jumped the gap between the poles all along them rather than only jumping the gap between the pole faces. Since there was less flux focused to the active area of the sensor, the response went down appreciably.

The redesign of the sensor (Figure 12) was subtle but produced much better response. Effectively, the only change was to split the coil from one large coil between the poles to 2 smaller coils moved up to the faces of the poles. The coils were then wired in series to ensure that a North and a South surface faced each other across the gap. The result was a vast improvement in response. The final design of the circuit reads the change in reluctance and is converted to a voltage response scaled from 1.25 to 3.75 V from open to closed position, respectively.

CONCLUSIONS

The IRAC shutter mechanism provides precise, repeatable, limited angular motion ideal for infrared sensing cryogenic instruments. The unique design of the actuator enables the shutter to dissipate very low power during operation. The torsion wire suspension system is elegant in its simplicity and enables the shutter to be designed to fail open and remain open through subsequent attempts to close. As designed, the shutter will survive and operate throughout all stages of the IRAC mission and remain operable in the post-cryogen environment of the spacecraft. Finally, the inclusion of a continuous variable reluctance sensor enables the shutter control to be optimized and ensures the knowledge of the mirror during a trouble shooting operation. The shutter meets or exceeds all functional and science requirements and is the result of the dedication of a talented development team.

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Chiachung Lee – mechanical analysis
Tom Hanyok, Bryan Grammer – mechanical design, drafting
Carlos Lugo, Carlos Bernabe, Gary Brown – electronics design, test

Other Contributors
Willie Barber, Willie Blanco, Derick Fuller, Paul Haney, Steve Hendricks, Dan McHugh, Dave Pfenning, Jim Towers, Steve Wood

REFERENCES

1. IRAC Project. “Infrared Array Camera (IRAC) Critical Design Review Package.” Goddard Space Flight Center, July, 1998.
2. Fazio, Giovanni G. “SIRTF Infrared Array Camera, IRAC Science Requirements.” Smithsonian Astrophysical Observatory (January 1998), p.1.
3. Miles, John W. and Dan W Trites. “Space Infrared telescope Facility, Observatory Performance and Interface Control Document.” Lockheed Martin, November, 1998.
4. Fazio, Giovanni G. “SIRTF Infrared Array Camera, Instrument Performance Requirements Document.” Smithsonian Astrophysical Observatory, March 22, 1999.
5. Carpenter Steel Division. “Alloy Data: Hiperco Alloy 50A.” Carpenter Technology Corporation
6. Reed, Richard P. and Alan F. Clark, Ed. *Materials at Low Temperatures.* American Society for Metals, Metals Park, Ohio © 1983, pp. 94-100.
7. Anderson, Richard H. and Sidney B. Bernstein. “Scanning Mirror for Infrared Sensors.” Lockheed Missiles and Space Co., Sunnyvale, CA., 7th Aerospace Mechanisms Symposium, Houston, Texas, 1972
8. Touloukian, Y. S., R. K. Kirby, R. E. Taylor, and P. D. Desai. *Thermophysical Properties of Matter, Vol. 12: Thermal Expansion, Metallic Elements and Alloys.* Plenum Publishing Corporation, © 1975.

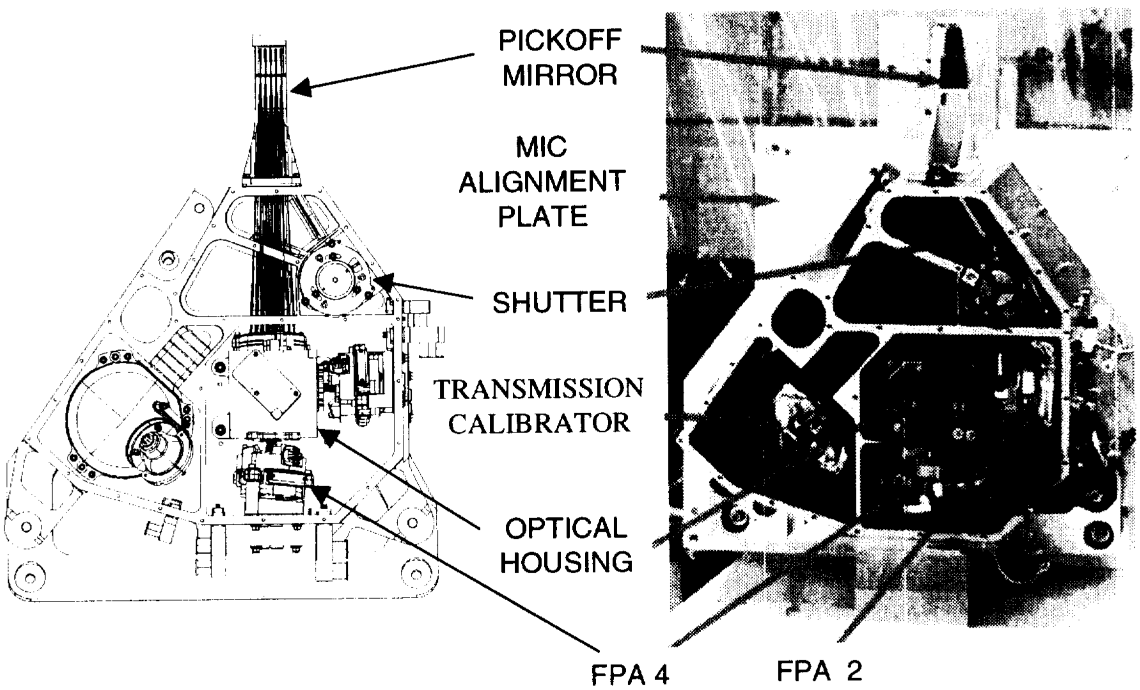

Figure 1 – IRAC Assembly

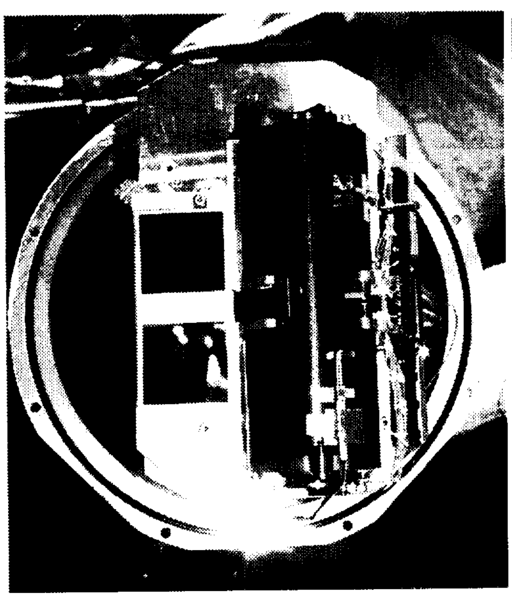

Figure 2 – IRAC Flight Shutter Unit

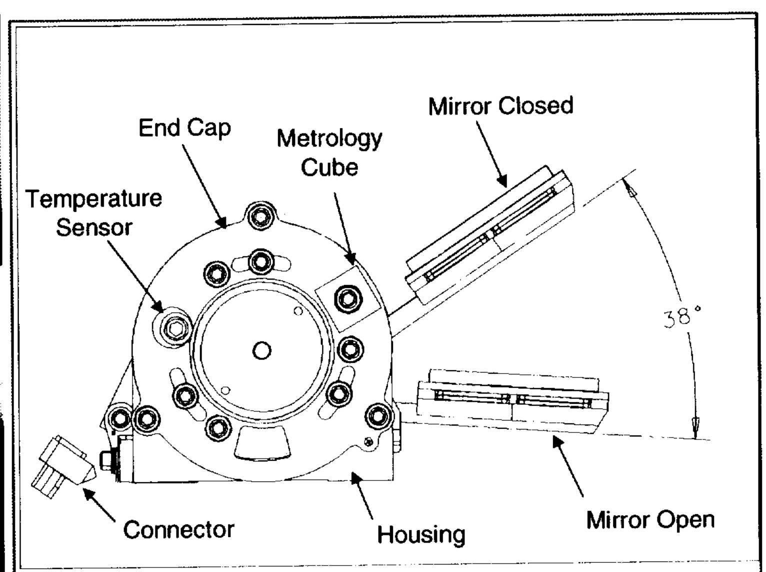

Figure 3 – End View Showing Mirror Angle

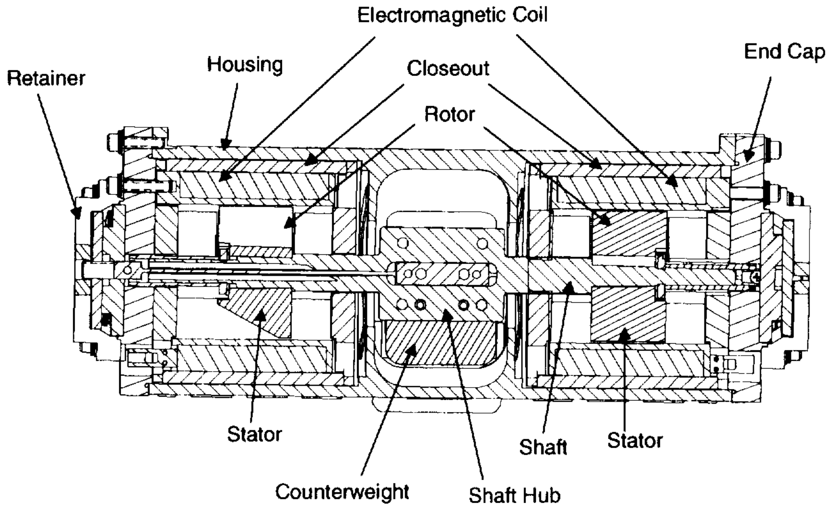

Figure 4 – Shutter Cross Section

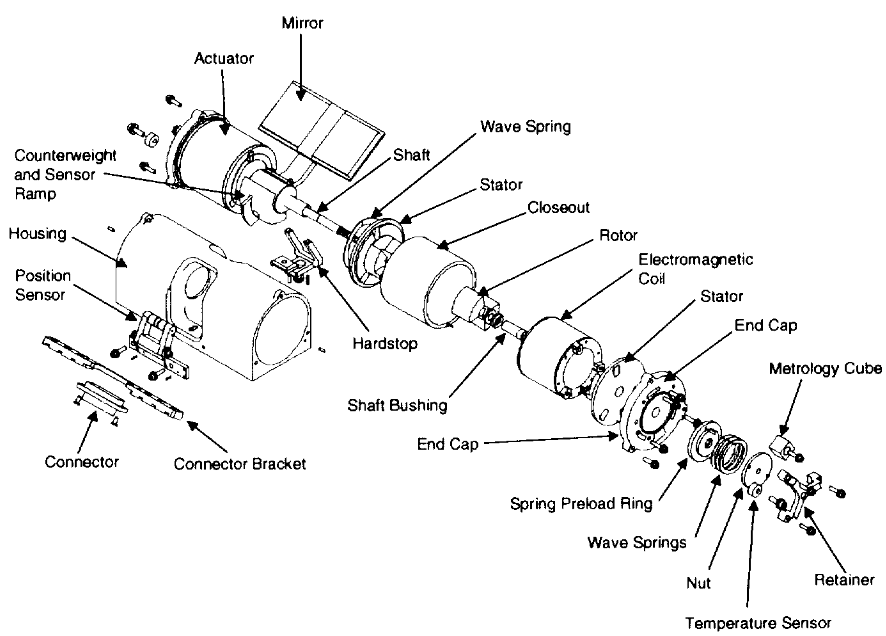

Figure 5 – Shutter Exploded View

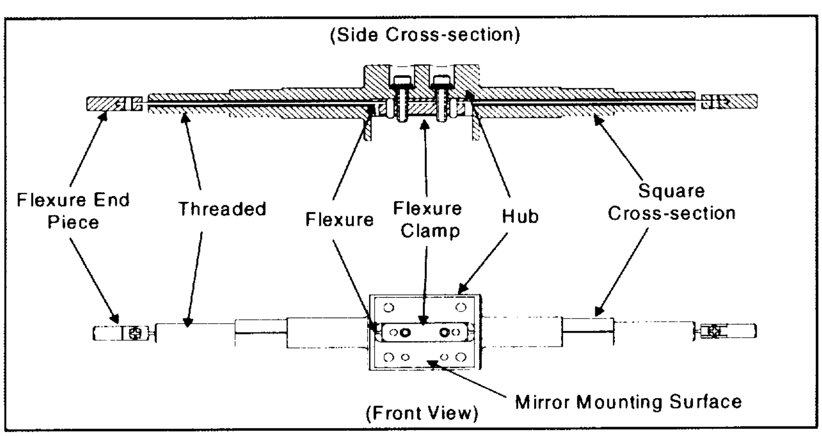

Figure 6 – Shaft Subassembly

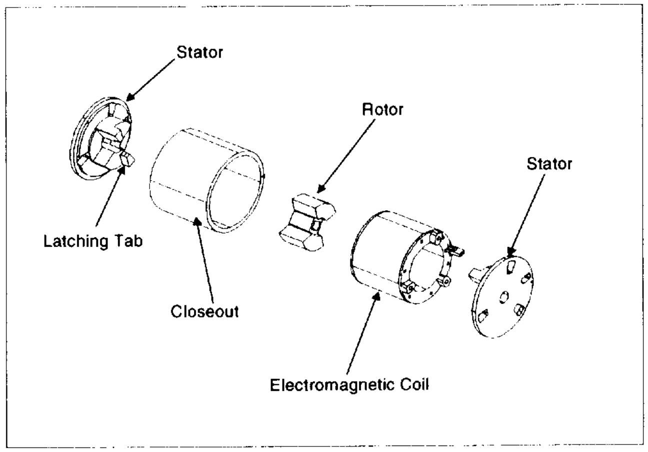

Figure 7 – Actuator Exploded View

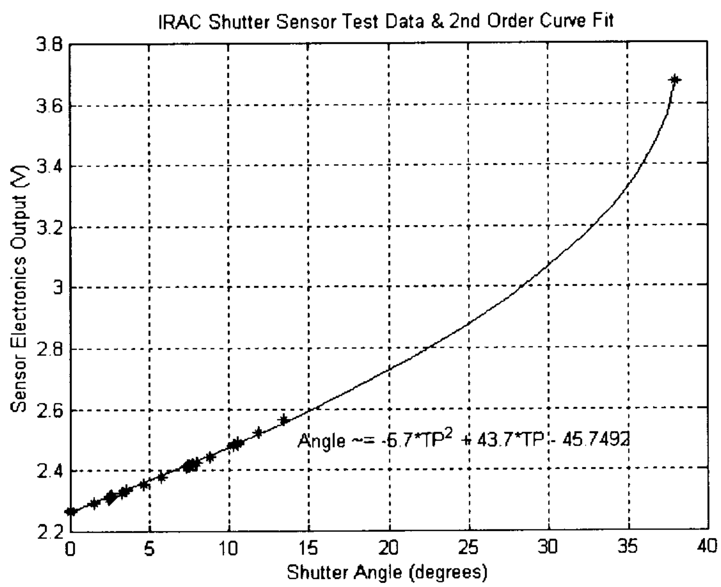

Figure 8 – IRAC Sensor Response vs. Mirror Angle

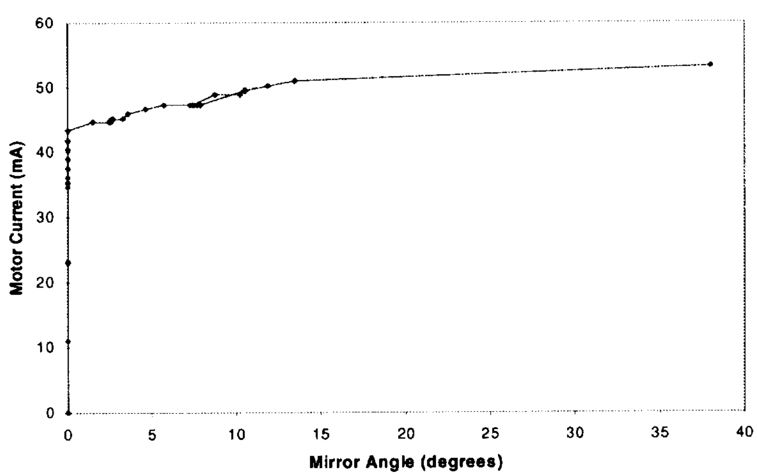

Figure 9 – Current vs. Measured Mirror Angle

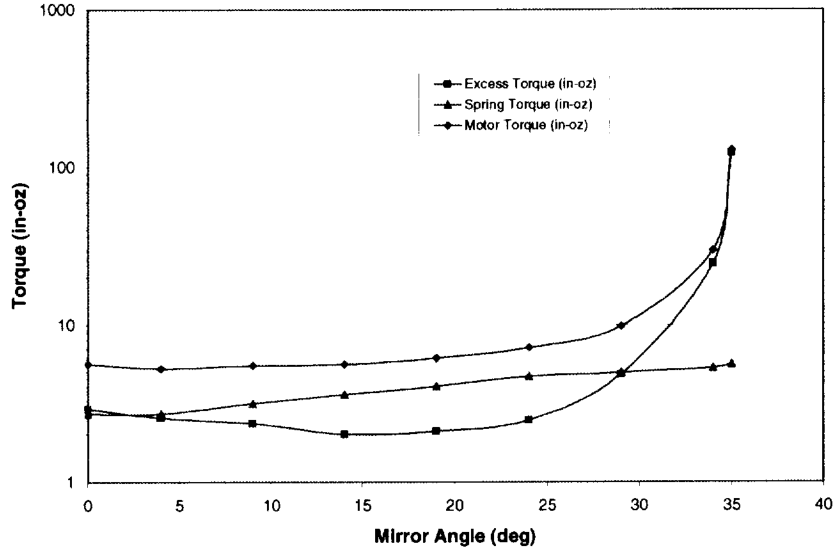

Figure10 – Spring, Motor, and Excess Motor Torques at 60 mA

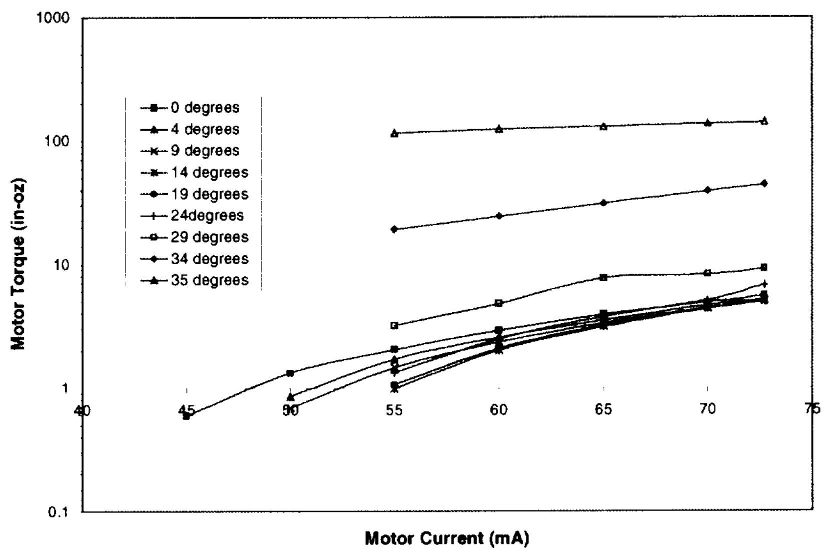

Figure 11 – Excess Motor Torque vs. Current for Various Mirror Angles

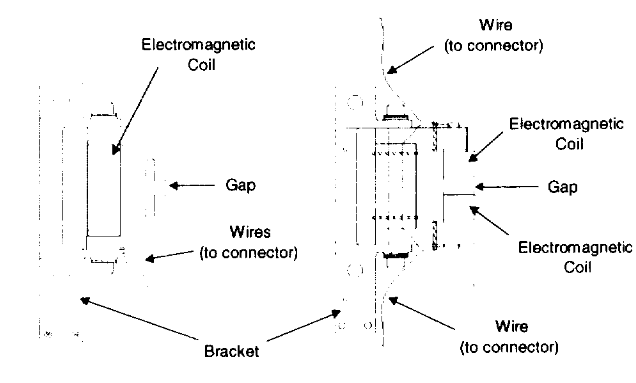

Figure 12 – Position Sensor Designs