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SIRTF/IRS CRYOGENIC GRATING DRIVE MECHANISM (ARC SECOND POSITIONING AT 4 K)

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ABSTRACT

This paper describes the requirements, design, and test results of a grating drive mechanism for the Infrared Spectrograph (IRS) science instrument on the proposed superfluid helium-cooled Space Infrared Telescope Facility (SIRTF). The IRS grating drive mechanism, tested in fall 1989, satisfied all performance requirements in vacuum at 4 Kelvin. Measured mechanism performance included: 1.5 arc second (arc sec) root-mean-squared (rms) error positioning resolution, 2.2 arc sec rms position repeatability error, less than 10 milli-joules/degree dissipated power, and ±170 degree (deg) angular range of travel. Mechanisms that precisely position optical elements at very low cryogenic temperatures (at/below 4 Kelvin) are vital to the operating success of a number of proposed infrared scientific instruments, like those in SIRTF.

INTRODUCTION

The SIRTF telescope shown in Figure 1(a), is considered part of NASA's Great Observatories Program along with the Hubble Space Telescope (HST), the Gamma-Ray Observatory (GRO), and the Advanced X-Ray Astrophysics Facility (AXAF). SIRTF will be the first true infrared observatory in space, with >1000 times greater sensitivity and a broader wavelength coverage (1.8-700 μm) than the earlier, successful Infrared Astronomical Satellite (IRAS).[1] The performance gains will result from the use of extremely sensitive, cryogenically-cooled infrared detectors. The size, weight, power, and spatial constraints of a space-borne telescope necessitate positioning optical elements and related mechanisms in the same vicinity, near the 'cold' detectors, and demand proper function of these mechanisms at liquid helium (LHe) temperature; 4 Kelvin (4 K) and below.

Four mechanisms have been identified as necessary to obtain the desired scientific configuration of the IRS instrument shown in Figures 1(b) and 1(c), consistent with the SIRTF application: (1) filter wheel, (2) slit wheel, (3) grating wheel, (4) echelle.[2] The object of the IRS cryogenic mechanism development effort was to demonstrate, with a single mechanism, the technology to meet the most stringent performance requirements of all the proposed mechanisms. The IRS grating drive mechanism was therefore designed, built, and tested to meet the following requirements in a vacuum environment at 4 K: ±4 arc sec positioning resolution for 8 separate equally spaced ±3 deg ranges of grating motion, ±2 arc sec command position repeatability, <1 milli-watt average power dissipation, ±170 deg rotational travel, in a maximum package size of 6-in. diameter by 6-in. length.

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The IRS demonstration mechanism was designed to specifically address the tight positional accuracy and 4 K functional requirements. The design uses structural materials selected for their desirable properties at cryogenic temperatures, such as low relative brittleness and close matching of coefficients of thermal expansion (CTE) for dissimilar materials between room temperature (300 K) and LHe temperature (4 K). Critical mechanism drive, position sensing components, and required processes were identified, while extensive vendor searches were performed with the cryogenic environment in mind. Survival qualification tests followed for a D.C. brushless torque motor. An arc sec rotary transducer configuration was selected and developed for position sensing. Dry lubrication was specified for the mechanism ball bearings. These components and processes were all reviewed and evaluated for their capability to meet all system performance requirements for operation in vacuum at 4 K, before final selections were made for incorporation into the IRS mechanism design.

MECHANISM DESCRIPTION

Figure 2 shows an isometric cut-away view of the IRS grating drive mechanism. The unit is a simple drive mechanism that consists of a hollow rotating shaft which is simply supported by a single pair of angular contact ball bearings. An 8-faceted grating drum is used to mount test mirrors to the near end of the shaft. The mirrors were later used to verify the angular motion of shaft and drum. The bearings are mounted into the stationary housing and preloaded axially by means of a diaphragm. The physical envelope of the unit is approximately 6 inches in diameter by 6 inches in length.

The rotating assembly is driven by a brushless D.C. torque motor. The permanent magnet rotor is mounted integral to the mechanism shaft, and the stator with armature windings, is mounted into a stationary ring on the housing. The radial and axial spacing of the rotating components is accomplished by individual spacers that fit over the shaft and are shimmed at assembly to coincide with the respective stationary part (stator, housing, etc.) axial locations. The rotating assembly is axially loaded and retained by a jam nut screwed onto threads provided at the far end of the mechanism drive shaft.

The mechanism employs two position sensor configurations which indicate the coarse (±10 arc min) and fine position (±1 arc sec) of the rotating assembly. The coarse sensor is located at the far end of the unit near the torque motor and consists of a rotating, 'Nautilus shell'-shaped cam sensor wheel and a stationary sensor reading head coil. The fine position sensor assembly is located at the near end of the unit, adjacent to the grating drum, and is comprised of a rotating 8-toothed sensor wheel (corresponding to the 8 facets of the grating drum) and two stationary fine sensor reading cores. These sensors will be discussed in more detail later in this paper.

The stationary portion of the mechanism is comprised of the main bearing housing, diaphragm, and separate mounting rings for the torque motor stator and fine sensor assemblies. The radial location and axial spacing of the mounting rings and housing is handled with three separate, 3-piece sets of mounting flexures that physically link the stationary components together. The cold plate flexure set also serves as a tripod base that mounts the entire mechanism assembly to the dewar cold plate.

BEARINGS AND LUBRICATION

The IRS mechanism design approach focused on two main objectives: arc sec position accuracy at 4 K and ±170 deg of rotational travel. The large range of travel dictated the use of ball bearings to support the mechanism shaft. Operation at cryogenic temperatures necessitated dry lubrication in order to maintain bearing torques at uniform, acceptable levels during operation at 4 K. The ball bearings also required a compliant preload scheme to avoid brinelling during cool-down and required precise control of the preload, in order to meet the position accuracy requirements for operation at both 300 and 4 K.

The bearings used for this application were 440C stainless steel, precision (ABEC-7), angular contact ball bearings (bore diameters 0.875 and 0.750 in., respectively), with 440C stainless steel ball separators, manufactured by Miniature Precision Bearing (MPB) Corp. The bearing balls, separators, and races were lubricated with a sputtered, 800 Angstroms-thick coating of Molybdenum disulphide (MoS_2). This particular combination of bearing and lubrication was chosen for a number of reasons,[8,9] based on meeting the most severe position/design-driving requirements at 4 K. Angular contact ball bearings in particular were selected so that the internal radial clearance of the bearing could be eliminated with sufficient preload. The thin, hard- sputtered MoS_2 lubrication coating was selected to insure smooth rotational motion, minimize torque non-uniformities due to lubrication 'pile-up', and maintain absolute minimum contamination levels at 300 and 4 K.[9]

Bearing friction torque is nominally a function of the bearing geometry (pitch diameter), the bearing preload, and the ball lubricant. To achieve the required positional accuracy in a cryogenic environment, a single pair of reasonably sized ball bearings consistent with the mechanism package were chosen. The larger diameter bearing (0.875 in. bore) was positioned just beneath the grating drum in order to provide the most stable shaft restraint as close to the grating drum and critical mirror surfaces as possible. The other bearing (0.750 in. bore diameter) was mounted into an axially compliant diaphragm. The compliant diaphragm ensures that the preload level is closely maintained at both 300 and 4 K, without over-stressing the bearings throughout the temperature transition. Snubbers were designed and would be used to carry the axial loads on the diaphragm for a launch environment.

The amount of preload necessary to maximize mechanism positional performance characteristics in the arc sec regime involves a trade-off between maximizing system response stiffness in all extraneous degrees of freedom, yet minimizing resistive, and most important, nonlinear or discontinuous resistive torques (stiction, Dahl effects, etc.) opposing pure rotation. As the axial preload on a bearing increases, the system response stiffness to undesired motions increases, but so do the unwanted resistive and non-uniform torques (both are functions of preload and lubrication). The approach adopted was to apply 5 pounds (lb) of preload to the bearings via a diaphragm deflection, which corresponds to approximately a 60,000 psi. Hertzian contact stress between the bearing balls and races, and yields an emperical torque minimum for a particular bearing.[3]

MATERIALS CHOICES

A selection of 'flight-like' materials and careful specification of dimensional tolerances were performed in the detailed design to closely match material CTEs and account for the differential thermal contraction of dissimilar materials from room temperature (300 K) to 4 K. The most critical mechanical component required to achieve the mechanism accuracy and travel requirements was undoubtedly the angular contact ball bearings. As previously mentioned, the bearing material chosen was 440C stainless steel, a common ball bearing material. Structural material selections for the mechanism piece part fabrication then, was based primarily on their respective CTE match with the 440C bearings. Figure 3 shows published values for the percent of thermal contraction between 300 and 4 K of some candidate structure materials considered.[4]

The thermal contraction of 6Al-4V titanium (6Al-4V Ti) most closely matches that of 440C while avoiding the extremely brittle properties of the 400-series stainless steels at low temperatures (see Figure 4), and was therefore chosen for the mechanism shaft, housing, diaphragm, and flexure/motor mounting ring materials. The decision was made to fabricate the bearing spacers and mounting plate flexures out of 416S stainless steel (ss). At the same time, this choice was accounted for by increasing slightly the bearing preload (by approximately 2 lb) at room temperature to offset the anticipated decrease in bearing preload at 4 K, due to the differential thermal contraction between the titanium housing and the 416S ss bearing spacers. Test results show that the bearing preload did decrease somewhat between 300 and 4 K as predicted, but the diaphragm was compliant while maintaining a preload on the bearings as designed.

There is however, an unavoidable disadvantage of using either 440C stainless steel or titanium seen in Figure 5, which shows the low thermal conductivity of both materials. Low thermal conductivity slows the overall mechanism cool-down, increases temperature gradients, and could possibly limit the mechanism moves or available scientific observation time, due to inefficient removal of local dissipative heat inputs during flight operation. Temperature gradients of more than 100 K between rotating and stationary assemblies were analytically estimated to occur, if uncontrolled during mechanism cool-down. This is due to the small contact area and poor thermal conduction across the interface between the 440C bearing balls and races. There was serious and immediate concern, since temperature gradients of this size were certain to cause brinelling of the outer bearing races. Flexible copper thermal strapping (multiple strands of 44 AWG 'Litz-wire') was therefore added to both rotating and stationary portions of the mechanism to speed cool-down, efficiently remove dissipative heat inputs, and minimize the undesirable temperature gradients.

ADDITIONAL MECHANICAL DESIGN FEATURES

Several features were incorporated into the IRS grating drive mechanism design, shown in Figure 6, to meet the tight system level position performance requirements and achieve these at 4 K. For instance, different bearing sizes were chosen so that the shaft diameters corresponding to these bearing bores could be machined from the same machine set up and the bearings could be assembled from the same end of the shaft. This allows tighter machining tolerances to be held and the most accurate bearing bore-to-bore locational press-fits to be achieved so that shaft fits and possible tilt within the bearings are minimized. The axial span of the bearings was also maximized in order to mitigate the cyclic tilt effect that runout in the 0.750 in. bore diameter bearing will have on the shaft and grating drum. Runout in the 0.875 in. bore diameter bearing would be seen more as a lateral, cyclic shaft wobble at the grating drum and was evaluated to be slight, less than 50 millionths of an inch, but unavoidable. This wobble motion due to bearing runout is dealt with and rejected by the specific configuration of the fine sensors that will be discussed later in the description of the fine sensors.

The construction of the mechanism also features the three separate sets of 3-piece flexures which were mentioned previously and which are used to locate the stationary mounting rings for the torque motor, fine sensors, and mechanism housing. Each flexure in a set is equally spaced around the circumference of the respective mounting rings (see Figures 7 and 8) to serve dual functions. The flexures provide accurate radial and axial location of the mounting rings, and they compensate for the differential thermal contraction between the dissimilar ring materials that they physically join. The individual flexures within each set were match-machined for identical registration of mounting surfaces, designed with sufficient stiffness to maintain mechanism torsion mode resonant frequencies outside the servo control bandwidth, and designed to carry lateral launch loads. Yet the flexures are flexible enough radially to handle the differential thermal contraction between rings with a minimum of induced stresses. Once accurately located during the initial assembly at 300 K, these flexures were match-drilled and pinned in order to negate any flexure base movement during cool-down and insure disassembly and subsequent assembly repeatability.

DRIVE COMPONENTS SELECTION AND DEVELOPMENT

The motor selection for the IRS demonstration mechanism focused on a commercial vendor search based in part on previously reported experience of motor testing at cryogenic temperatures.[6,7] The search centered primarily on materials and configuration options for motor magnets, epoxies, laminations, armature windings, encapsulants, and their behavior down to 4 K. A torque motor was chosen to drive the mechanism because it provides an 'infinitely precise' positioning drive without gear reduction or associated backlash; it met the low power dissipation requirements; and a configuration was available that had survived multiple excursions to 4 K without degradation.

The particular torque motor selected for the IRS grating mechanism is a 'pancake-type', 8-pole, 3-phase, D.C. brushless torque motor manufactured by Magnetic technologies, Inc. (P/N 2374H-050P-1750). The physical package of the motor unit is 0.5 in. thick by a 2.375 in. outside diameter. The stationary portion of the motor (stator) has three armature windings wound in a wye configuration around a stack of M-19 silicon steel laminations that are encapsulated with Epoxylite CF682A epoxy resin. Samarium-Cobalt (SmCo) permanent magnets are retained on the rotating portion of the unit (rotor) using 3M-EC-2214 adhesive. To date, the motor and spares have experienced approximately 15 thermal excursions to 4 K without a measured or visible degradation in performance or mechanical integrity.[5]

Accurate and precise angular position sensing at 4 K posed the single most significant component challenge to the operational success of the IRS mechanism design, as no previously published experience at cryogenic temperatures had been found. The rotational position sensing of the IRS drive mechanism was accomplished with an angular transducer technology developed specifically for this mechanism demonstration and more generally for precise angular shaft motions at cryogenic temperatures. The mechanism grating position is indicated by two different magnetically inductive, rotary transducer configurations. The coarse sensor and cam plate shown in Figure 7 provide absolute position feedback (±10 arc min) for large grating moves (between facets) in the ±170 deg range of mechanism travel.

The fine sensor configuration shown in Figure 8 uses two sensor reading cores and an 8-toothed wheel to provide arc sec position resolution feedback over the 8 separate ±3 deg ranges of shaft rotation that correspond to the 8-facets available on the grating drum for mounting mirrors. The sensor cores are placed on opposite sides of the toothed wheel center to simultaneously detect the position of the shaft within each of the 8 grating drum facets. In this configuration, sensitivity to axial and radial motion, such as bearing runout, are rejected by the differential nature of the two sensors spaced on opposite sides of the wheel.

The IRS control system to drive the mechanism had to be capable of performing 3 major functions: commutating the torque motor, providing control for velocity feedback from the coarse sensor channel for large grating moves (between facets), and providing control for arc sec position feedback from the fine sensor channel for small grating moves (±3 deg within a facet). Figure 9 shows the block diagram of the overall servo control system implemented. The logic gate determines the selection of the individual control loop to be employed. The mechanism control is passed between three separate servo loops during the functional operation of selecting a location and then moving to that exact position within the appropriate facet. The three loops are the velocity loop (trajectory generator), the coarse position loop, and the fine position loop, as well as the torque motor commutation to carry these out. The trajectory generator is utilized in order to optimize dissipative power consumption for grating moves between facets, as a way to conserve dewar cryogen.

A more detailed discussion of the torque motor selection, sensor technology development, and control system design and implementation for this application is presented in reference 5. It should be mentioned here, however, that the fine position servo loop has an integral compensation feature which accounts and compensates for difficulties controlling the ball bearing suspended hardware in the arc sec positioning regime. The servo loop compensation adequately handles any potential control system difficulties experienced as a result of an increase in the torsional spring constant of the ball bearings for very small angular movements. This torsional stiffening or "Dahl" behavior of ball bearing systems (i.e., ball bearings acting like springs over very small angular excursions) has been quite well documented for room temperature applications,[8,11] and was not found here, to differ drastically at 4 K. The fine position servo loop, along with this associated compensation and arc sec position feedback supplied by the fine sensors, are the functional blocks in the architecture of the control system that afford the mechanism the capability of arc sec positioning resolution and repeatability performance within the required ±3 deg angular range of each grating facet.

APPARATUS AND TEST SET-UP

Figure 10 shows a schematic layout of the mechanism performance test set-up. All mechanism performance tests were completed in a vacuum environment at LHe temperature (4 K) inside a bench-top dewar manufactured by Precision Cryogenics, Inc. The dewar has a 40 hour hold time and a working volume adjacent to the aluminum cold plate that is approximately 8-in. in diameter by 8-in. in length. The mechanism housing support flexures were attached directly to the dewar cold plate via threaded fasteners and holes tapped into the cold plate (see Figure 2).

Polished mirror surfaces were attached to the rotating, grating drum surfaces and viewed from outside the dewar by means of a 1/4 wave flat, BK-7 glass window. The window was placed in the side of the dewar, insulated and 'stopped-down' with reflective material to minimize parasitic/radiative thermal load to the dewar cryogen. The mirror surfaces themselves were diamond turned blanks of 6061 aluminum, polished to 1/2 wave flat, and mounted to three of the grating drum facet surfaces by using simple 3-point mounts. In order to expedite testing at 4 K, only three grating facets were chosen to verify the mechanism positioning capability and performance accuracy. The facets employed were not uniquely selected in any predetermined manner other than by the desire to choose two adjacent facets, as well as a facet more than 90 deg (2 facets) away from these, in order to vary the test movement range between facets. All actual grating position verification tests were performed optically using a Wild-Heerbrugg T-2000 digital theodolite positioned external and adjacent to the dewar on an optical bench in the lab. An additional mirror was mounted on the mechanism housing in view of the window to serve as a stationary reference.

MEASURED PERFORMANCE [5]

The main objective of the mechanism performance tests was to determine the stepping precision and repeatability of the IRS grating drive mechanism at 4 K. All results reported here were performed in a vacuum at 4 K inside the dewar described. The unit achieved the cool-down from 300 to 4 K in approximately 12 hours, and the expected temperature gradients between the mechanism shaft and housing were significantly reduced as a result of the thermal strapping. Stepping resolution was determined by advancing the mechanism grating with a sequence of steps ranging from 2 to 200 arc sec and optically measuring angular deviation (error) from the commanded step size. In this manner, the rms error for small steps (2 $<\theta<$200 arc sec) within a single facet was measured to be 1.5 arc sec.

The performance capability of the mechanism to accurately repeat large angle steps was also measured. For 3 deg steps within a single facet, the rms error for repeatability was measured to be 2.2 arc sec. The same rms error was observed for larger angle steps from facet-to-facet, ranging in step size from 45 deg to 135 deg. Tests were also done to measure the time it took the mechanism to settle within 2 arc sec of the command step position for three separate angular displacements. The step and settling times for 330 arc sec, 7.9 deg, and 263 deg moves of the instrument grating were 130 msec, 180 msec, and 1.7 sec, respectively.

Since the spectrometer will be operated at 4 K, the power consumption of its cryogenic components directly impacts the SIRTF helium consumption, and hence the mission lifetime. Total power dissipation at 4 K for both coarse and fine position sensors was measured to be 60 microwatts. Under normal operational conditions the grating mechanism is moved intermittently, and its impact on the consumption is best expressed as the energy required to step and settle to a new angular position. Figure 11 shows this relationship for angular displacements up to about 2 deg. For larger angular displacements, considerable reduction in the slope of this curve is possible by employing the principles of the optimum velocity trajectory mentioned earlier.[5]

In the process of testing and optimizing the precision and repeatability of the IRS unit, significant effort was also directed towards characterizing the Dahl behavior of the mechanism ball bearings. Figure 12 shows a graph of the measured torsional spring constant of the mechanism for various step angle sizes at 300 and 4 K.[10] This data was generated by analyzing the open loop frequency response of the mechanism to position commands of the angular step sizes shown, computing the torsional stiffness (spring constant) of the mechanism, and characterizing the appropriate control compensation required to account for the changes, at both 300 and 4 K. The author believes that the difference in the overall magnitudes of the two curves (between 300 K and 4 K curves) may be attributed primarily to the reduction in bearing preload (approximately 2 lb) of the system between room temperature and 4 K that results from differential thermal contraction between the bearing spacer (416S ss) and housing (6Al-4V Ti).

CONCLUSIONS

The IRS demonstration mechanism met all system performance requirements at 4 K and successfully demonstrated this grating drive positioning technology at cryogenic temperatures. In addition, the behavior of the ball bearings used in this mechanism was characterized by measuring the torsional flexure constant at both 300 and 4 K for various ranges of motion precision in the arc sec regime. The test data revealed only a slight change in the mechanism stiffness and position resolution performance between 300 and 4 K. The reduction in preload was, in all probability, significant enough to account for the decrease in position resolution by reducing mechanical hardware stiffness and frequency response to the point of introducing nonlinearities that were not as tightly controllable. It is believed that the performance of the mechanism at 4 K could be made to match that measured at 300 K by using a bearing spacer fabricated out of the identical material as the housing (6Al-4V Ti), thus insuring that the bearing preload remains unchanged over the temperature excursion.

In addition to changing the material for the bearing spacer, future work and improvements to the mechanism design and configuration include: overall reduction in the mechanism package size and weight, further test and characterization of Dahl friction effects for varying values of bearing preload in order to optimize mechanism performance with this parameter, 4 K vibration survival and subsequent performance data acquisition to simulate the environmental condition to which the mechanism will be subjected for flight, as well as extended mechanism life-test data acquisition. Present plans for reduction in package size entail: eliminating the flexure mounting ring and fastening the cold plate base flexures directly to the mechanism housing, moving the axial location of the torque motor inboard of the bearings to mount it within the mechanism housing, and refining the position sensor technology as a means of further reducing the necessary mechanism envelope diameter. These modifications are expected to reduce the mechanism to a 5-in. diameter by 5-in. length maximum package size.

The schedule for further characterization of the mechanism performance under varying bearing preloads and vibration qualification testing is uncertain at the time of this publication. However, the current mechanism hardware is to be life-tested and incorporated into a cryogenic, infrared, spectrographic instrument built by Cornell University, which is scheduled for operation with the ground-based telescope facility at Palomar Observatory in California, sometime late in 1991.

The IRS mechanism development program success has established baselines for mechanism design at cryogenic temperatures in a number of key areas: motor selection and design, angular transducer design and development, ball bearing behavior, and lubrication use. The IRS grating drive mechanism test results have demonstrated the technical and functional viability of all these technologies together in a cryogenic environment (4 K) that successfully meets required optical grating accuracies approaching 1 arc sec for the Infrared Spectrograph (IRS) instrument for SIRTF.

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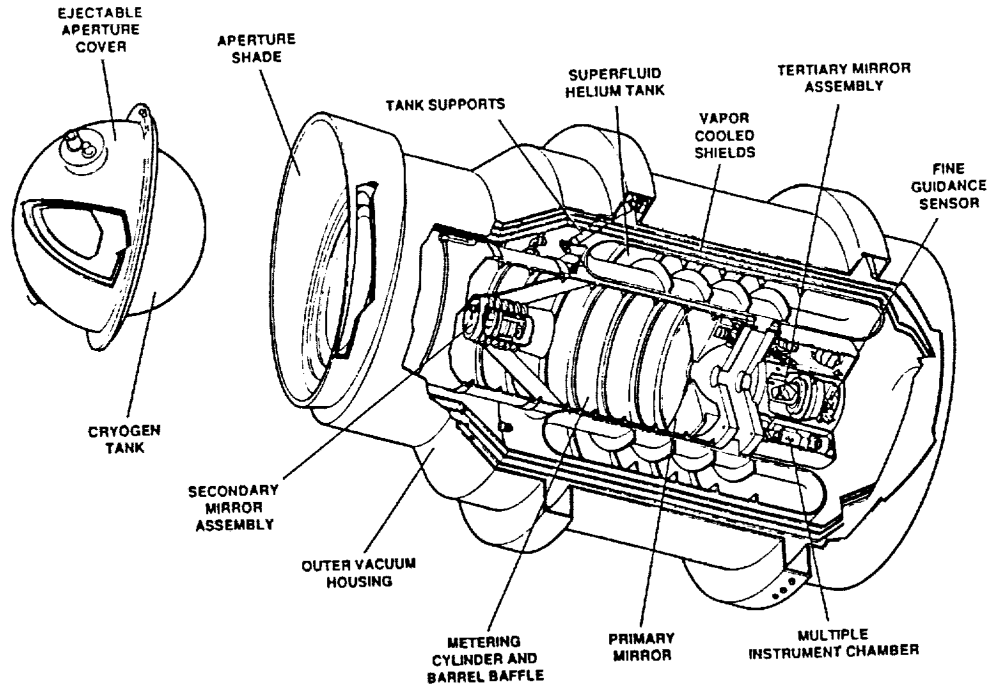

Figure 1(a). Cut-away view of SIRTF telescope assembly[1].

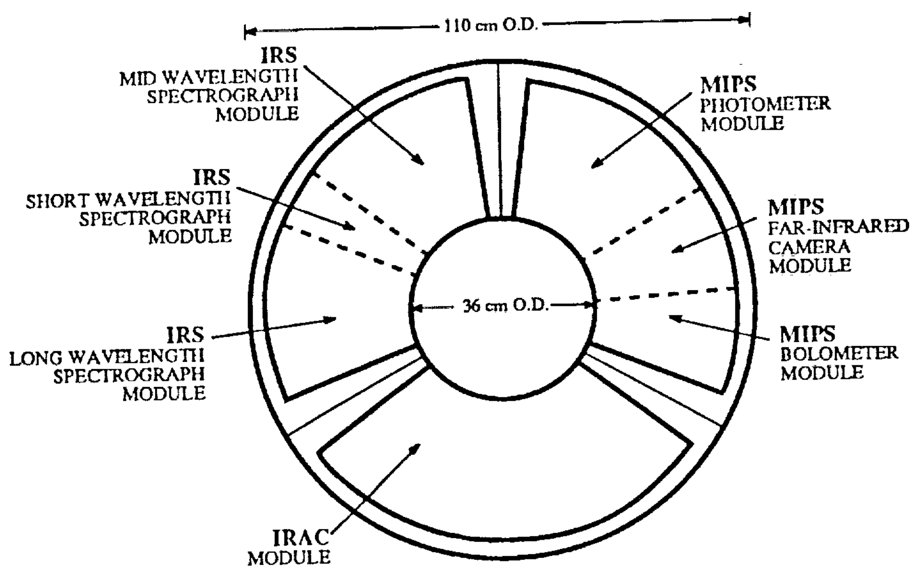

Figure 1(b). SIRTF multiple instrument chamber (MIC)[2].

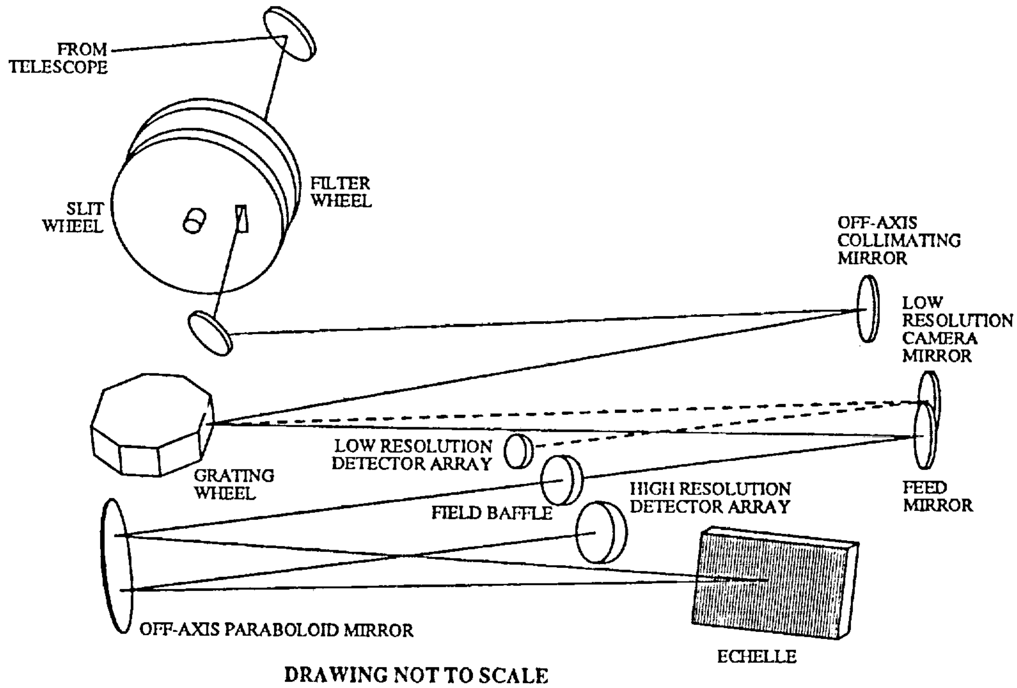

Figure 1(c). IRS mid wavelength module baseline optical system schematic[2].

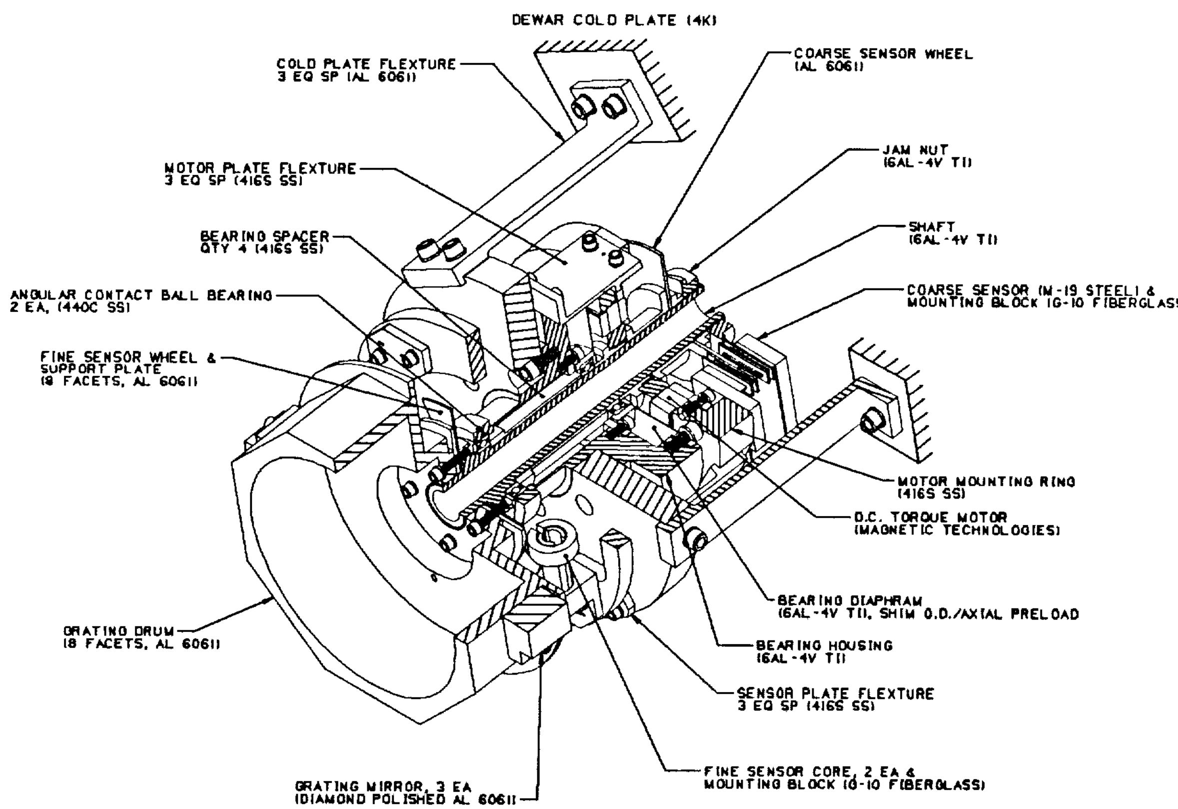

Figure 2. Isometric cut-away view of the IRS grating drive mechanism.

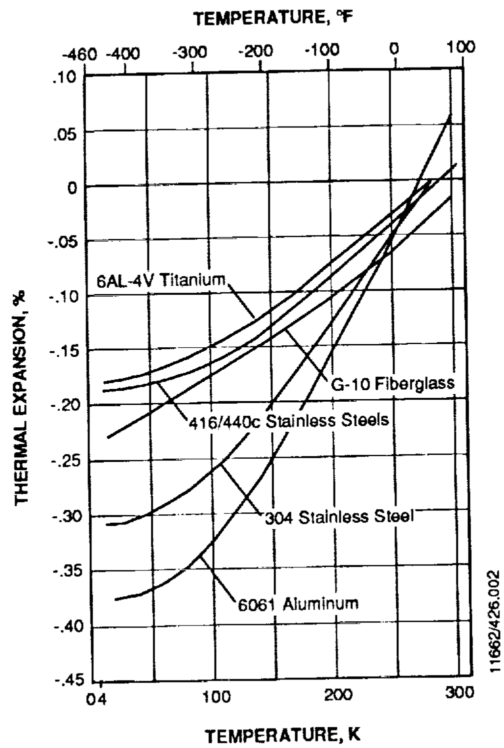

Figure 3. Published values of thermal expansion of candiate structure materials vs. temperature[4].

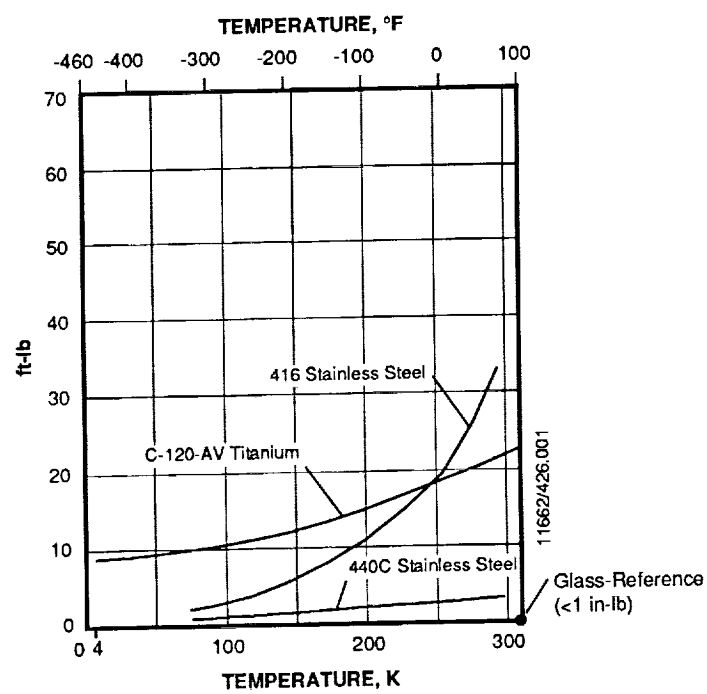

Figure 4. Published values of impact energy of engineering materials vs. temperature[4].

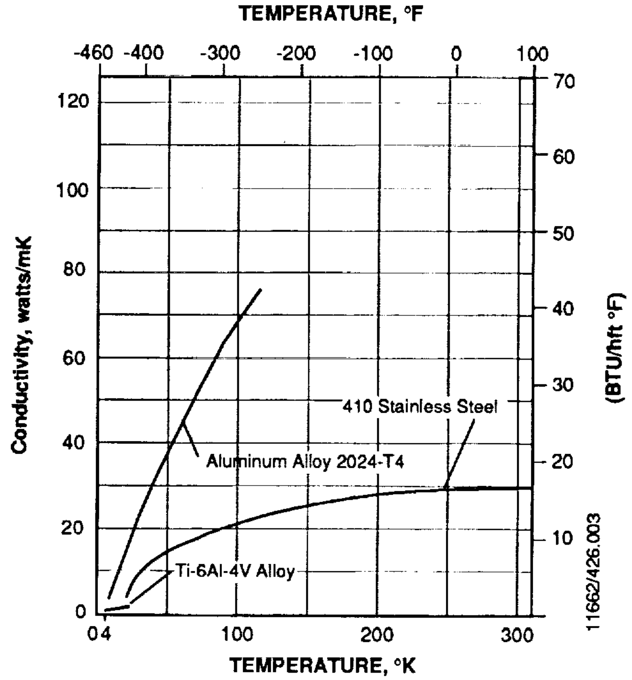

Figure 5. Published values of thermal conductivity of engineering materials vs. temperature[4].

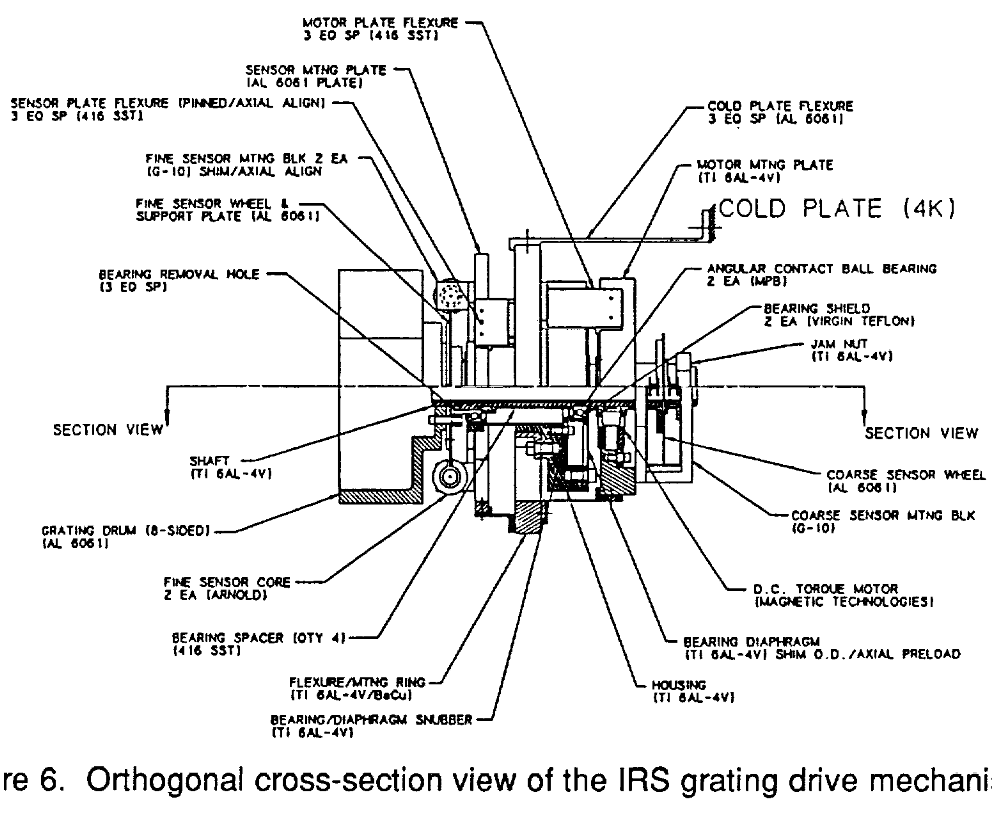

Figure 6. Orthogonal cross-section view of the IRS grating drive mechanism.

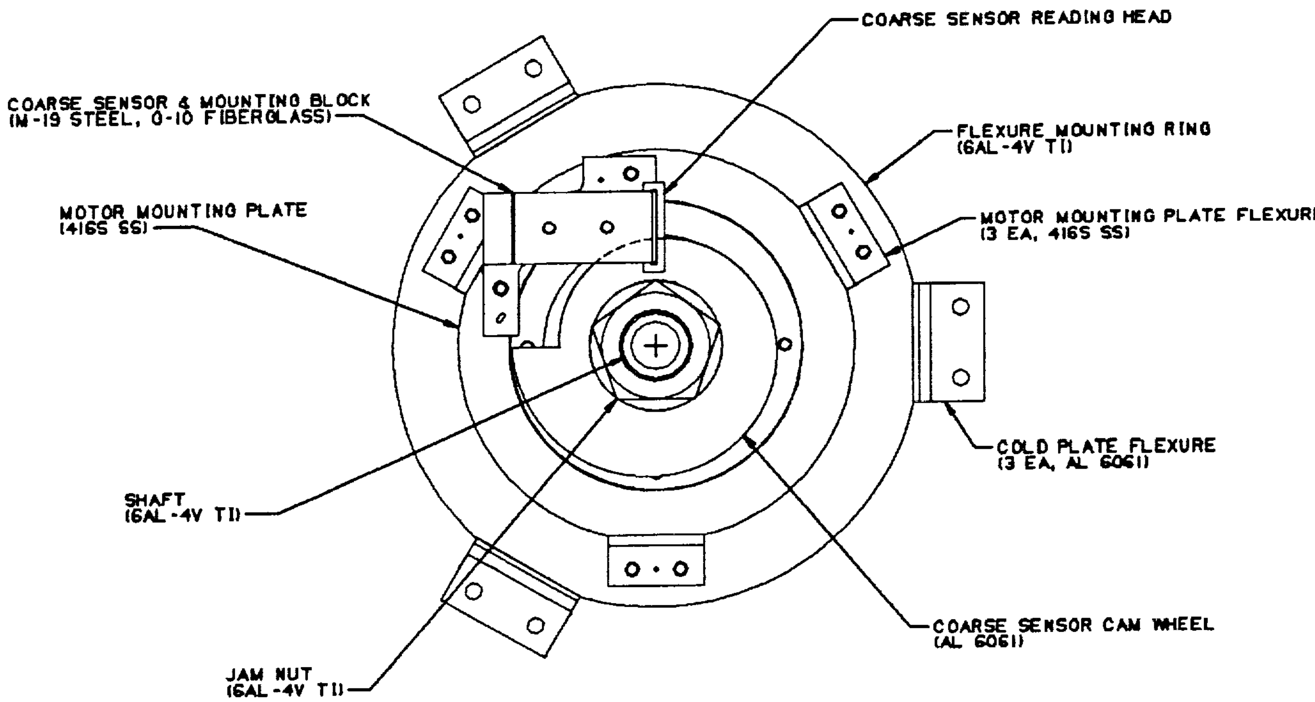

Figure 7. Bottom-end view of the IRS mechanism coarse sensor configuration .

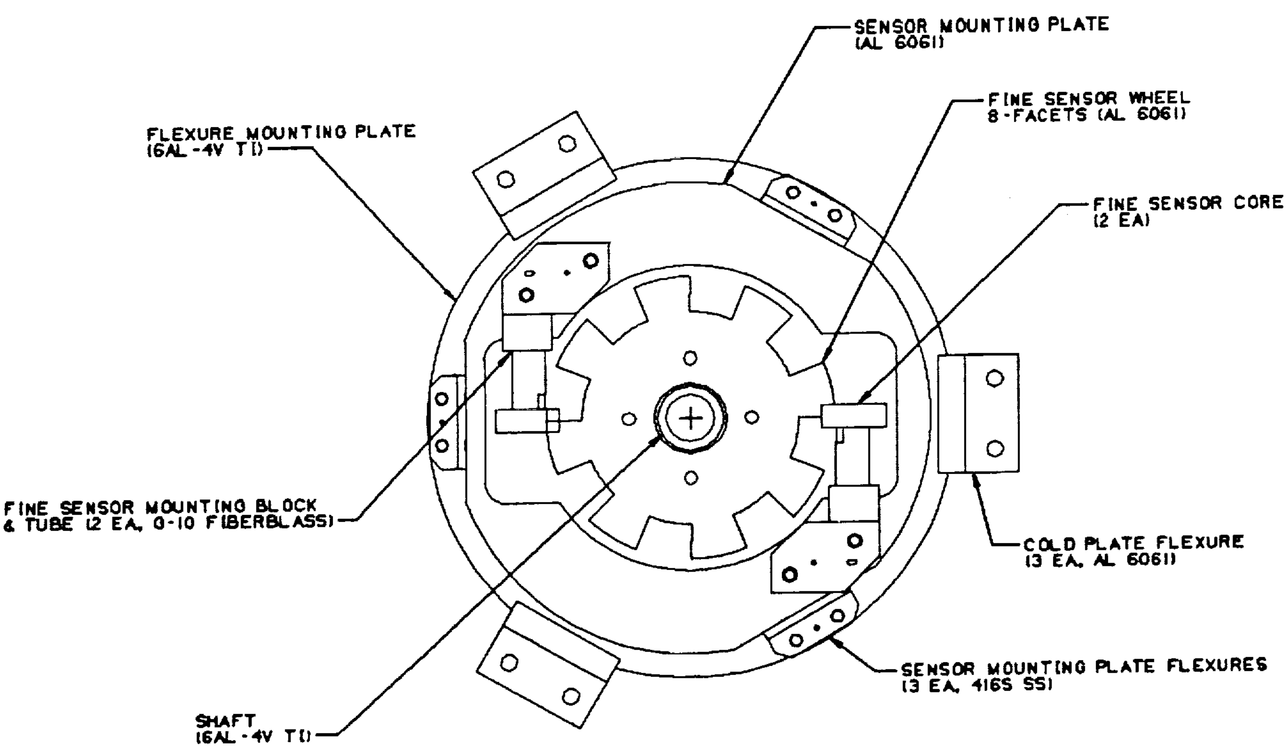

Figure 8. Top-end view of the IRS mechanism fine position configuration (grating durm not shown).

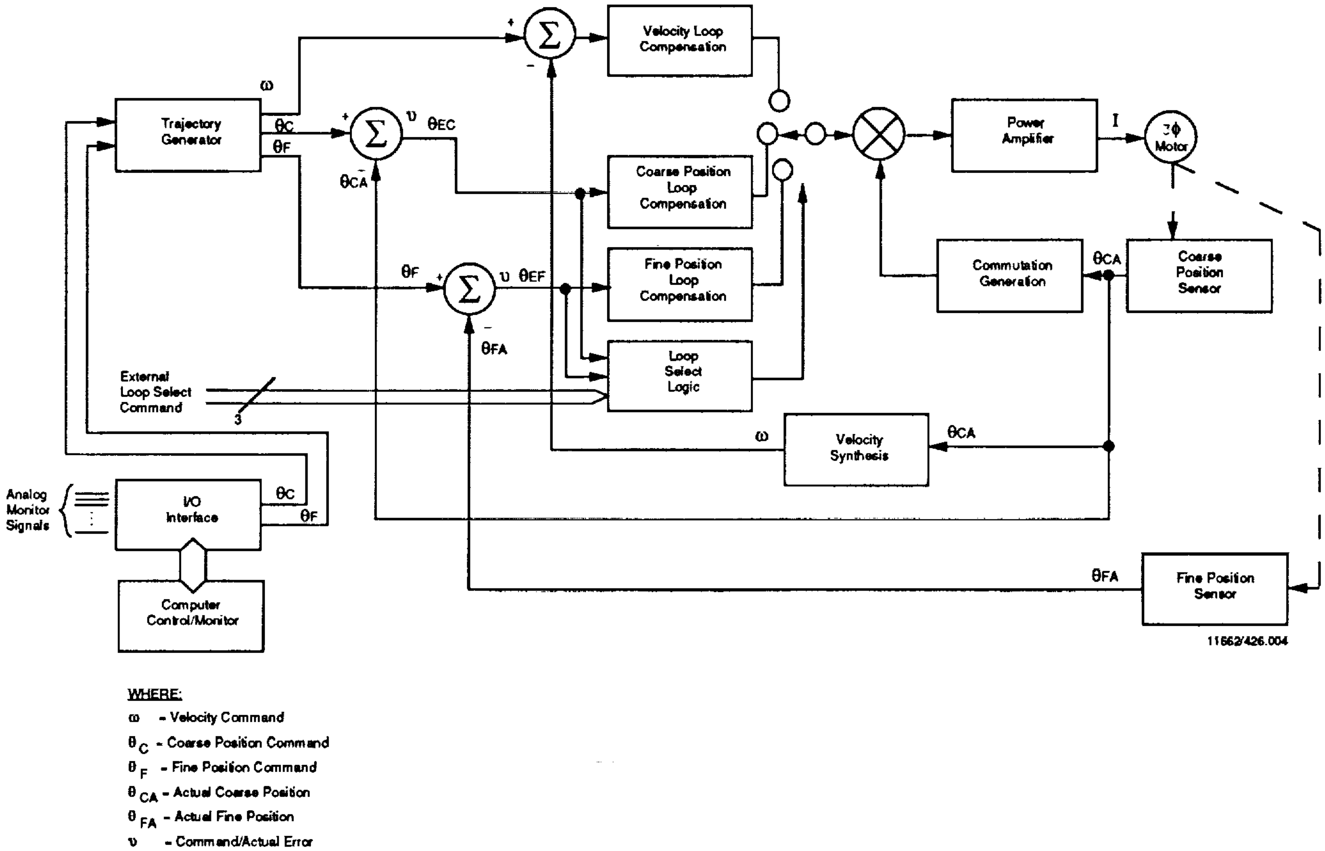

Figure 9. IRS grating mechanism servo system block diagram[10].

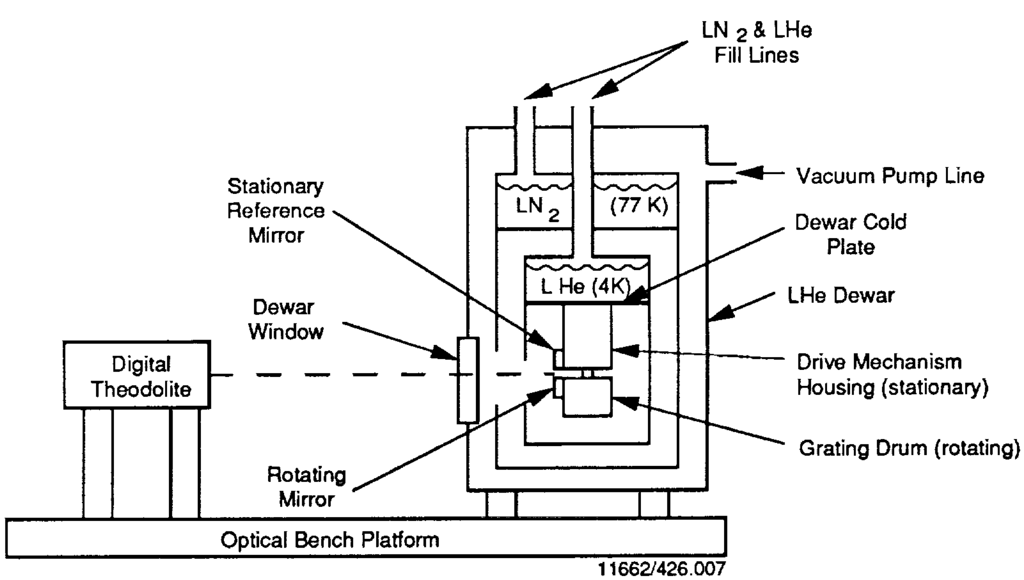

Figure 10. Schematic layout of the IRS grating drive mechanism cryogenic performance test setup[5].

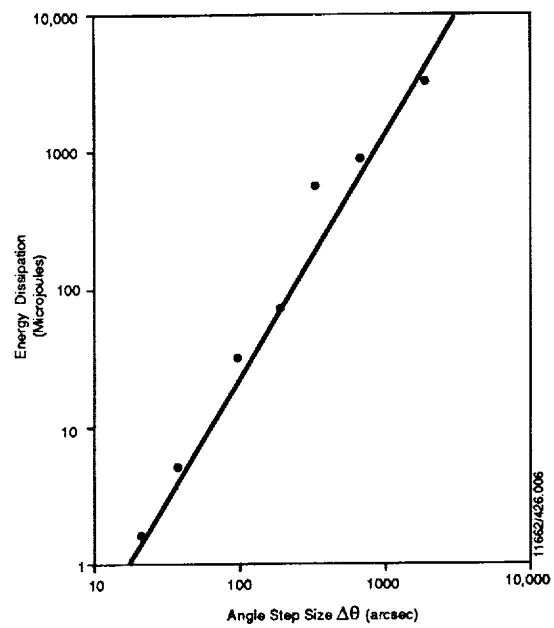

Figure 11. Energy per commanded grating step size at 4 kelvin[5].

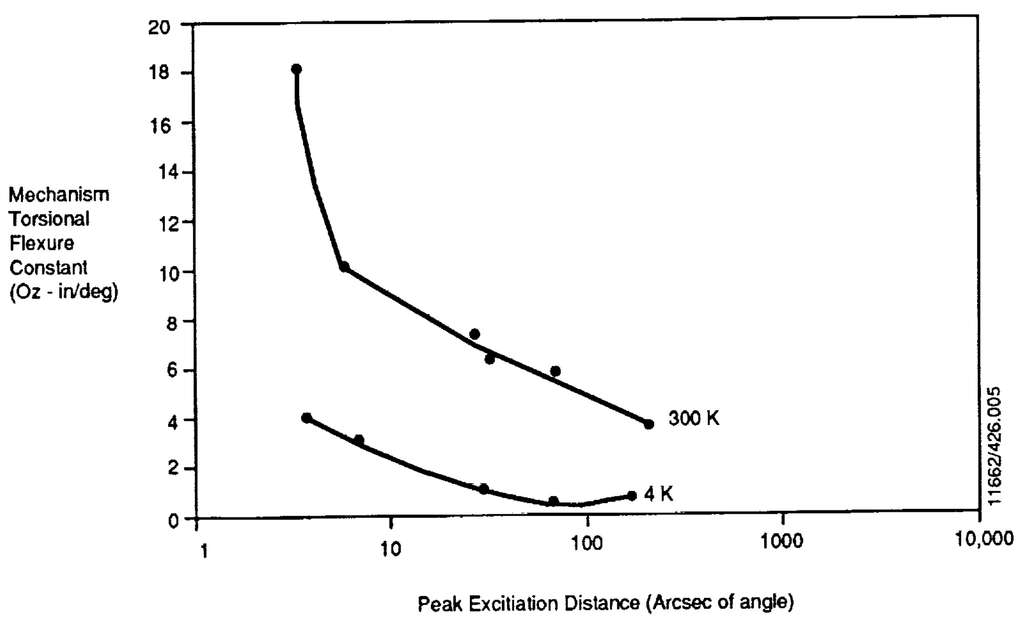

Figure 12. IRS grating drive mechanism torsional constant vs. sweep distance[10] (Random plant test – 6.9 V facet about fine transfer function center).