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NUTATION DAMPER SYSTEM

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ABSTRACT

The Nutation Damper System is a three function mechanism designed for the Galileo Spacecraft, a spin stabilized deep-space probe to Jupiter. By damping the movement of a large deployable science boom acting as an outboard pendulum, the nutation damper rapidly stabilizes the spacecraft from dynamic irregularities.

The system includes the boom deployment device and the ultra-low friction boom hinge. This paper describes the mechanism, the degree to which friction, stiction and ~ st motion have been eliminated, and the unique test methods that allow its performance to be measured.

INTRODUCTION

The Galileo spacecraft is the first large, spin-stabilized space probe designed by Jet Propulsion Laboratory and is intended to be the most stable science platform ever sent into deep space. This stability is in part delivered by the active nutation damping of the craft by a large deployable science boom acting as an outboard pendulum. This pendulum action of the deployable science boom is provided by a specialized zero friction boom hinge, and the damping is produced by coupling the boom to the spacecraft bus with the Nutation Damper (essentially a flight-qualified fluid shock absorber). The damper, hinge and science boom deployment strut are the three elements that compose the Nutation Damper System.

Spacecraft Dynamics

Figures 1 and 2 help to illustrate the difference between spacecraft wobble and nutation. Wobble is due to the spacecraft being

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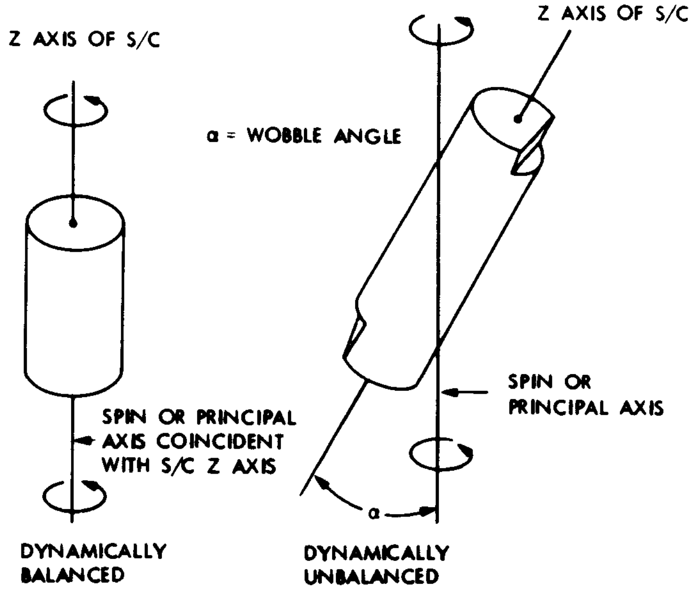

Figure 1. Spacecraft Wobble Only--Z Axis of Spacecraft Not Coincident With Principal Axis

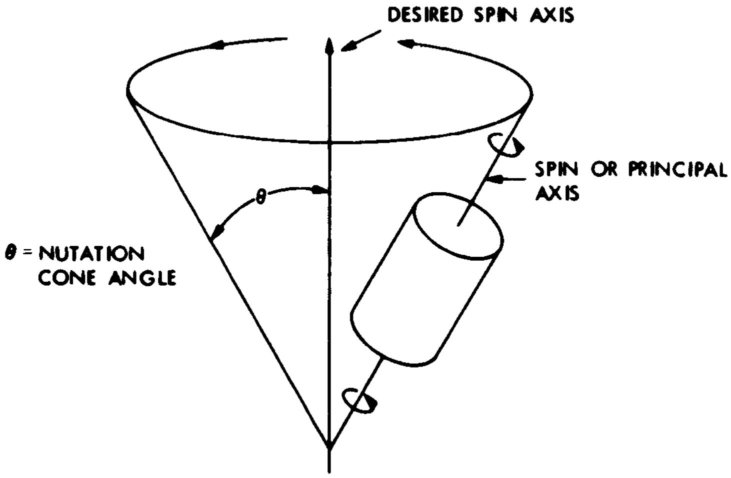

Figure 2. Spacecraft Nutation Only (No Wobble)

dynamically unbalanced, such that the spin (or principal axis) is not coincident with the Z axis of the spacecraft. On the other hand, nutation is the action of the spin axis of the spacecraft circulating in a "cone" about a desired spin axis. This action takes place when the spacecraft is disturbed by outside forces such as attitude correction maneuvers, fuel slosh, or if the main engine thrust vector does not pass directly along the principal axis. Figure 3 shows how the science boom will act as an outboard pendulum under the action of the nutating spacecraft if it is allowed to pivot at its base. By

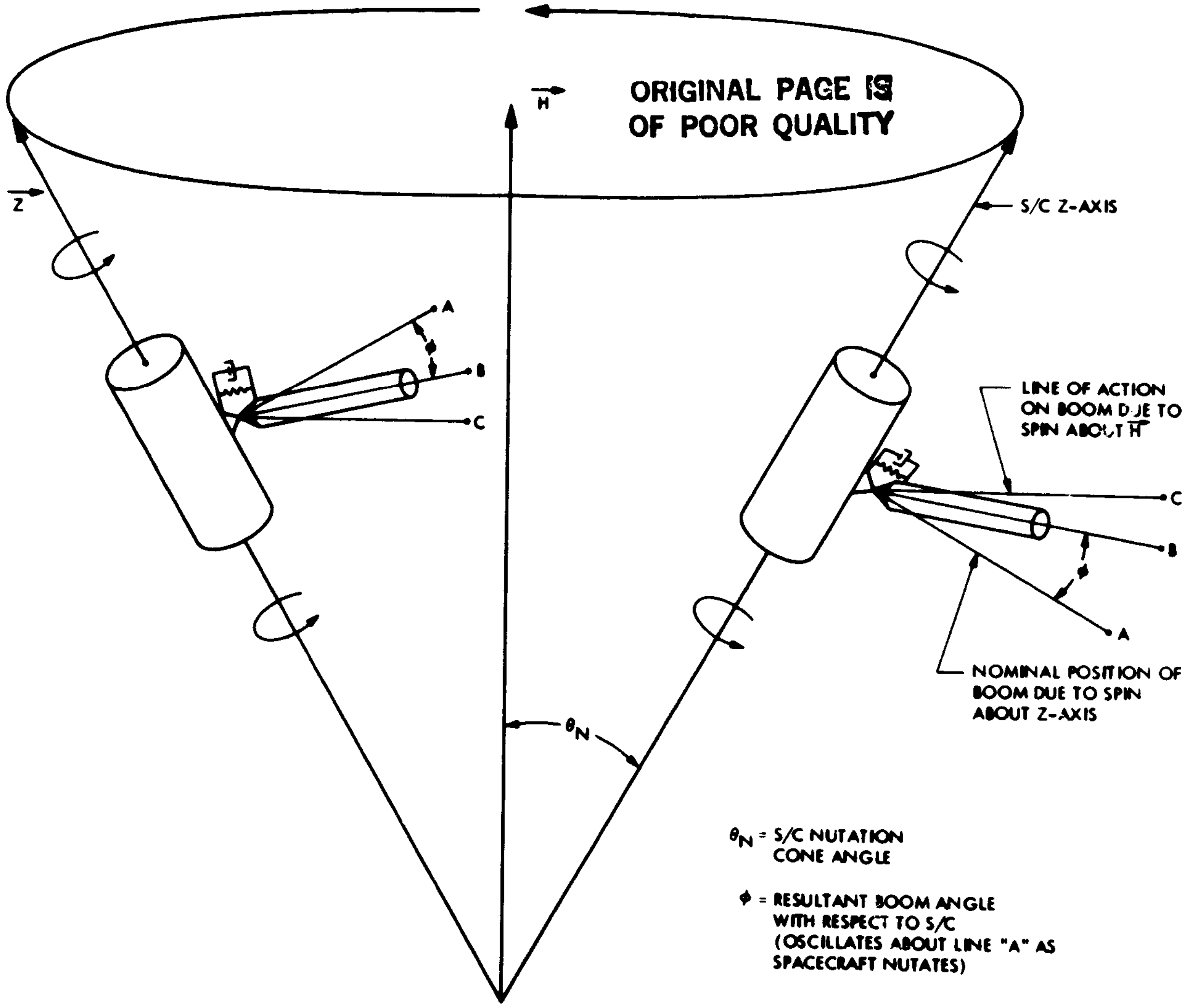

Figure 3. Boom Oscillation With Nutation

damping the motion of the boom, the time for the spacecraft to return to the desired spin axis is greatly reduced. This damping of spacecraft nutation provides an ultra-stable platform in inertial space for the science instruments, allowing science data to be gathered within minutes after spacecraft attitude changes.

What Was Required ?

The system engineers, when considering stability problems with the spacecraft, asked for a pendulum with 2 commandable hinge spring rates and excursions. Although a variety of spacecraft models were configured over the years, this pendulum (the science boom) eventually was required to have about its hinge a normal cruise spring rate of 335 N-m/rad (247 ft-lb/rad) with a maximum excursion of ± 6 degrees. A commandable stiffer spring rate of 2000 N-m/rad

(1475 ft-lb/rad) with a maximum excursion of ± 2 degrees was to be used during orbital insertion engine burns. The stiffer spring rate would compensate for boom sag during thrust by limiting the movement at the base of the boom. Again after numerous configuration changes, the damping constant of the boom eventually converged to 2455 N-m-sec/rad (1800 ft-lb-sec/rad). Because the mechanism was to be a silicone fluid filled damper and the damping-constant was not an extremely critical value, a tolerance of +100% and -50% was selected. The +100% tolerance allowed room for within-specification damping function over an environmental temperature range, and would provide a larger damping-constant target during development. The -50% tolerance provided within-specification redundancy to the mechanism.

The movement of the boom was required to have absolutely zero backlash. The excursions of the boom during 98% of the life of the spacecraft will fall below 4.4 mrad (1/4 degree) and thus full damping is required in that travel. Also, to insure theoretical pendulum action of the boom in this travel range near zero stiction (break-out force) is a must, and thus ultra-low friction is a design driver. The spacecraft requirement was for stiction torque of the boom (the minimum torque for movement of the boom) to be less than .039 N-m (.35 in-lb) about the hinge. Therefore, the major design effort was toward this goal.

Description of the Mechanism

Figure 4 is a sectional view of the deployment strut component of the mechanism. The strut consists of a large compressed deployment spring to pull the science boom up into position. Due to its size, this spring was fabricated out of titanium wire to save weight. The outboard end of the strut has a Dow Corning 510 silicone fluid filled chamber, and a boss on the deploy rod inside this chamber acts as a fluid orifice to provide damped movement. This damper limits the speed of the boom while allowing a large deployment force margin (see Figure 5). The deployment rod is guided by moly-impregnated Vespel bushings and sealed by O-rings on each end of the fluid chamber. An inboard and outboard felt wiper excludes any possible dirt contamination to the O-rings. Boom deployment is halted smoothly by an Elgiloy rebound spring, thus limiting the deceleration forces. Vespel latch fingers at the outermost end of the rod prevent any bounce-off. With 18,000 centistoke fluid, the maximum pressure in the chamber during deployment is 3.4 MPa (480 psig).

The silicone fluid in the deploy damper is volume and temperature compensated from - 60° C to + 65° C by the addition of a series of evacuated nickel bellows inside the deploy rod. This chamber is O-ring sealed and connected to the damper chamber by an orifice through the wall of the rod. The bellows act as an accummulator by compressing or expanding as the fluid is heated or cooled. This is accomplished by filling the deploy damper in a vacuum chamber at the hot temperature. In addition, a strip heater and temperature

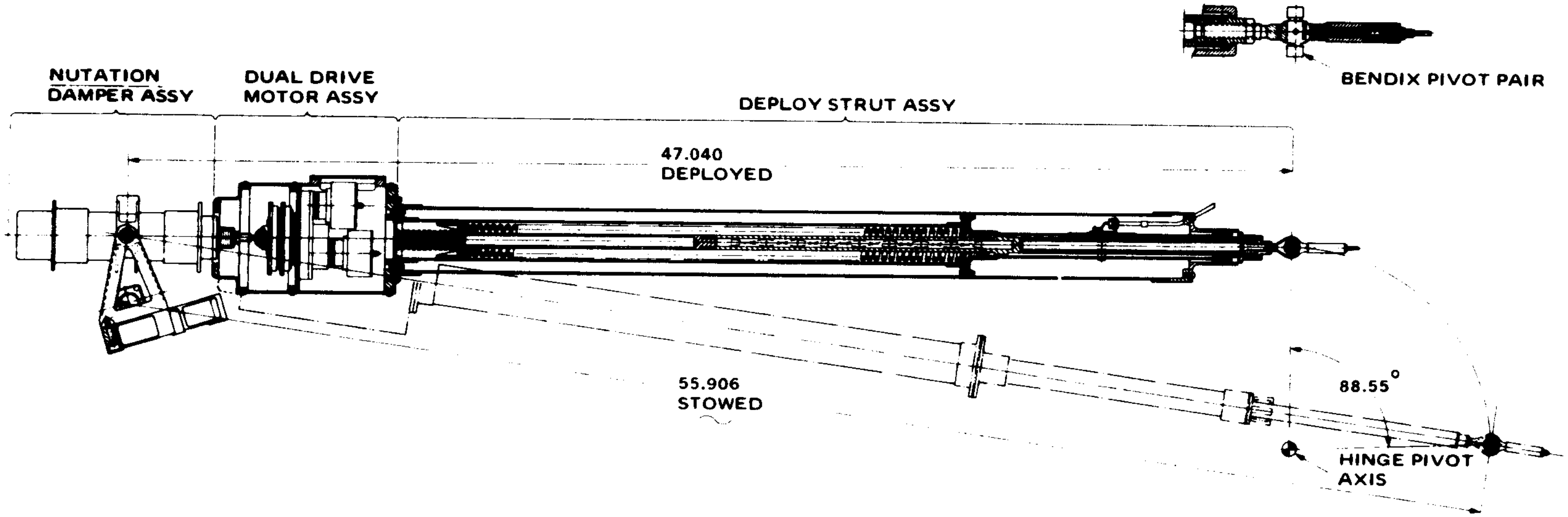

Figure 4. Section Drawing Deployment Strut

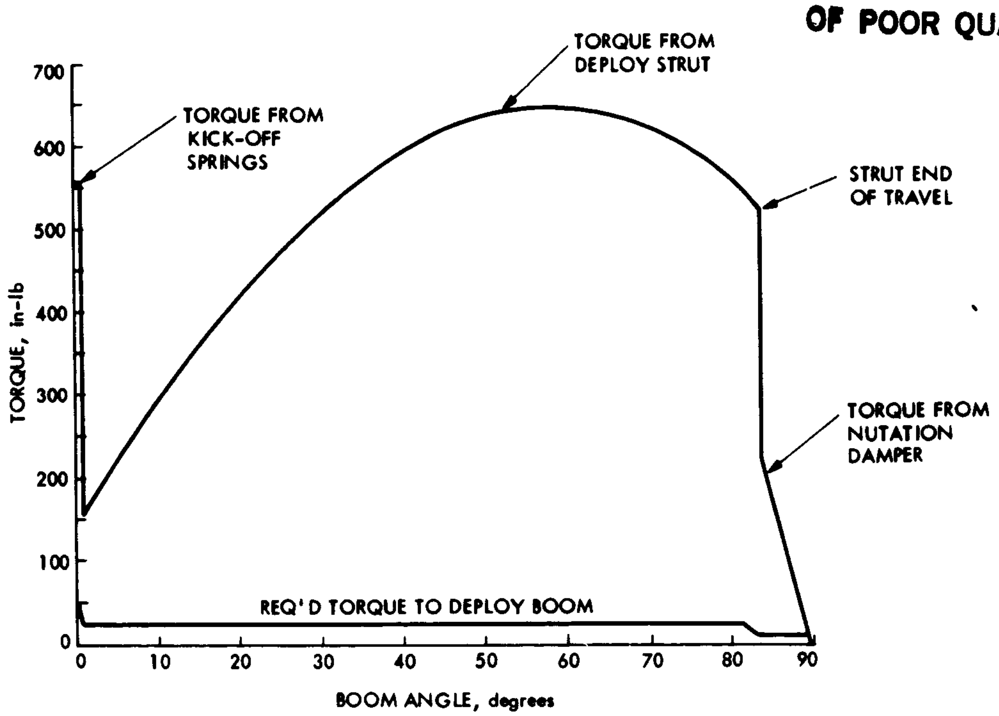

Figure 5. Deployment Torque vs. Boom Angle

transducers are attached to the outside of the damper tube to provide a controlled fluid temperature during deployment only.

Relatively late in the development of the mechanism, the launch loads analysis showed that the combination of the fiberglass spring tube and the titanium deploy damper and rod could deflect too much during launch. A boron-composite overtube was added that stiffens the entire strut. This overtube is allowed to slip fit on the outboard end to avoid any thermal expansion stresses.

Once the science boom has been deployed, the deploy strut acts only as a coupler from the nutation damper to the hinge. Figure 6 shows some prototype hardware on a spacecraft cable mock-up. The deploy strut (sans overtube) is seen between a large can containing a dual-drive motor assembly and the hinge. The nutation damper attaches the whole mechanism to the spacecraft bus. While the hinge pivots on spherical bearings to deploy the boom, it must allow oscillatory motion of ± 6 degrees with near zero friction, and absolutely no backlash. Any bearing here would have some unacceptable stiction torque. The solution was to attach the boom hinge to the spacecraft outrigger with Bendix flexural pivots, which would allow frictionless motion for the required amplitude.

Large 25 mm (one inch) diameter cantilever style pivots, custom fabricated by Bendix out of titanium, were structurally bonded into the bores of special spherical bearings. These bearings have a groove

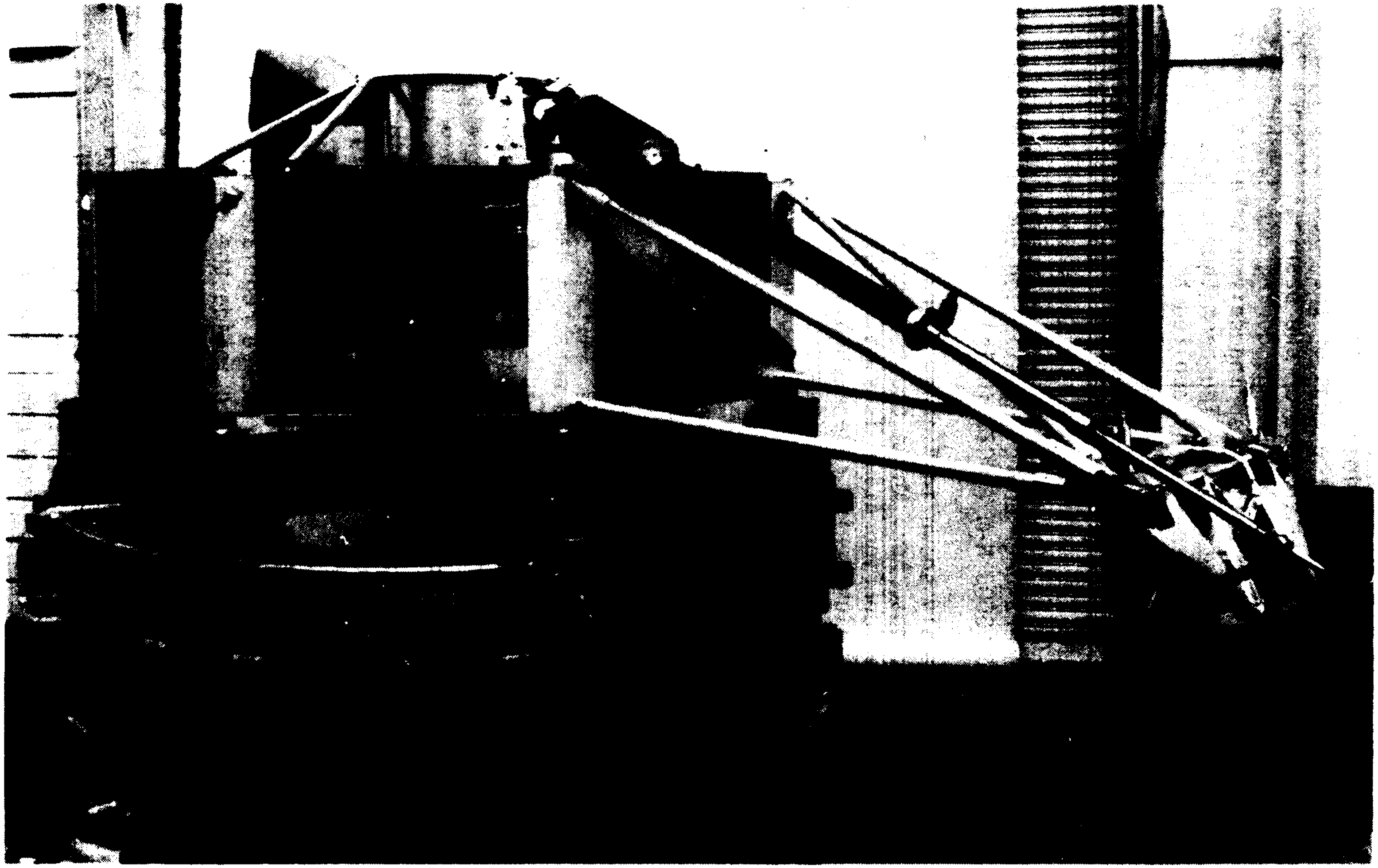

Figure 6. Nutation Damper System Prototype

ground into the ball and an access hole through the outer race to allow the installation of a spring loaded ball detent mechanism. Thus the boom hinge is supported by the ends of the flexural pivots and the outrigger supports the outer race of the bearings (Figure 7). The boom rotates about the spherical bearings during deployment, and when the boom is in position, a ball is spring driven into the groove of each spherical bearing to lock it up. The boom then pivots only on the flexures. This identical technique is applied at the male rod-end on the end of the deploy rod (Figure 4) using 8 mm (5/16 inch) diameter flexures. Since this method of pivoting the boom is completely frictionless, the only contributers to a possible boom stiction would then be the 200 conductor hinge cable assembly or the nutation damper itself. Tests on the hinge and cable assembly (discussed later) at temperature showed the stiction due to the cable assembly to be less than .0003 N-m (.0027 in-lb).

The nutation damper assembly shown in Figures 8 and 9 is basically an ultra-low friction shock absorber. While the section drawing appears complex, in actuality the configuration is simple. The damper can be seen as three parts; the high spring rate mechanism through the midline of the device, and two redundant dampers. Each damper element is composed of a main frame (in the center) that slides by a pair of linear ball bushings on a guide shaft that is rigidly fastened to a forward and aft frame. The fluid chambers are created by attaching two welded metal bellows between the frames. Since the main frame is fixed to the spacecraft bus, and the forward/aft frame and guide shaft combination is connected to the science boom hinge by the deployment strut, it can be seen that as the boom oscillates up and down the fluid chambers will alternately be compressed or expanded. The boom displacement of $\pm 6^{0}$ equates to a nutation damper translation of $\pm$ 22mm (.85 inches.) The fluid orifice between the alternating high and low pressure chambers consists of an annular ring sandwiched between the duplexed linear ball bushings. Flats on the guide shaft provide a fluid passage to the orifice. This orifice and the viscosity of the fluid are what control the damping value of the mechanism. As configured with 50 centistoke silicone fluid, the maximum pressure in the bellows is less than 183 KPa (12 Psig).

The silicone fluid is volume and temperature compensated from - 60° C to + 65° C with the temperature compensation (T.C.) bellows fixed through the aft frame. This chamber is pressure regulated by a compression spring on the bellows guide rod inside the damper guide shaft. The fluid passage to the chamber is from the forward damper bellows through a hole into the fill plug cavity and through a .15 mm diameter (.006 inch) orifice into the guide shaft. This orifice is sized to prevent T.C. bellows pump-up from the oscillating pressure in the damper. In operation, the temperature of the silicone fluid is regulated from + 20 ° C to + 35° C with two redundant heater and temperature transducer assemblies, each containing four series/parallel resistive heaters and two temperature transducers. The damping-constant changes approximately 30% over this temperature

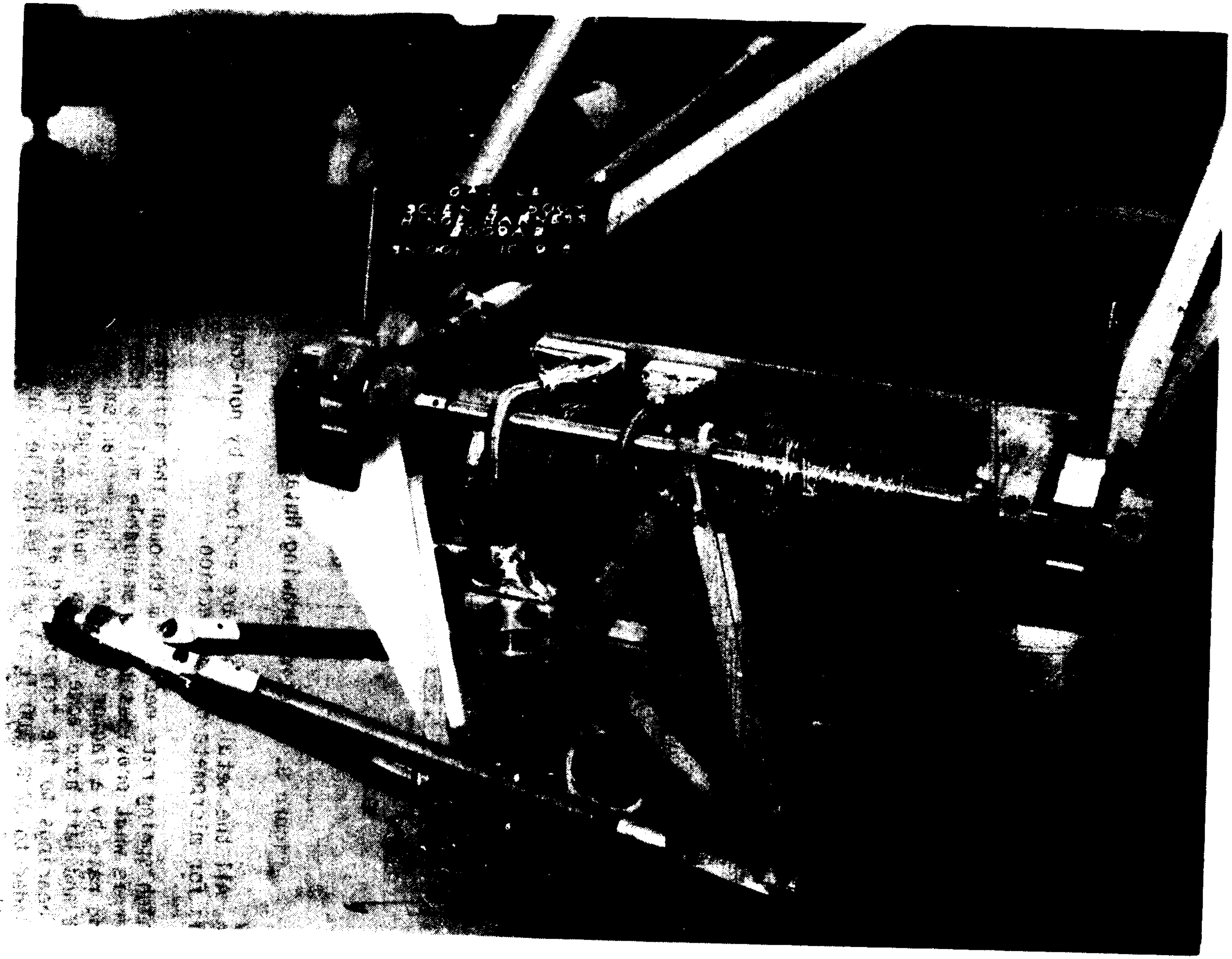

Figure 7. Science Boom Hinge

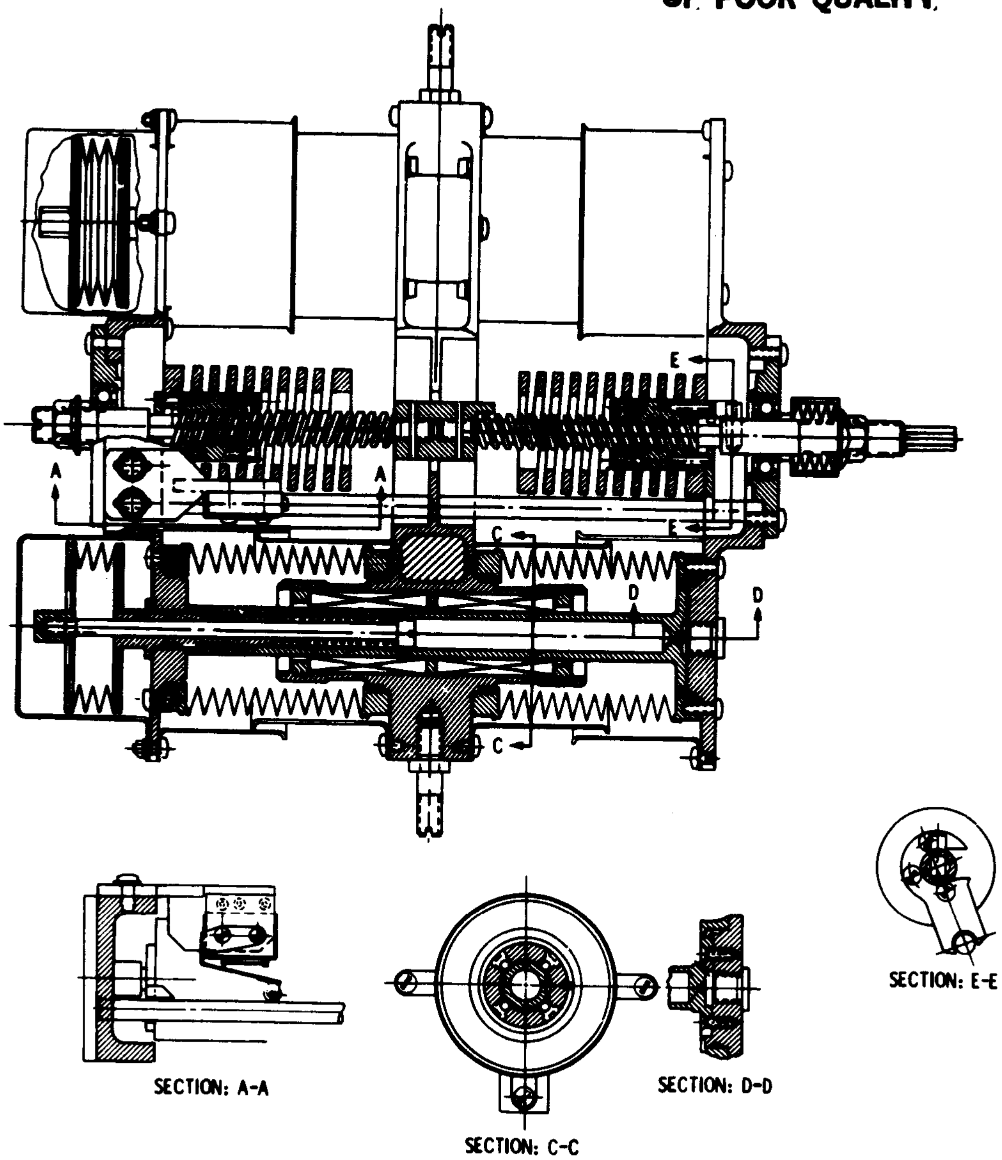

Figure 8. Section Drawing Nutation Damper Assembly

range. All the metal bellows are enclosed by non-contacting aluminum shells for micrometeroid protection.

The high spring rate mechanism through the midline of the nutation damper is what provides the commandable multiplication of the hinge spring rate by a factor of seven. The mechanism consists simply of right and left hand acme screws coupled together and fixed through ball bearings to the forward and aft frames. The bearings are preloaded to 22 N (100 lbs.) with bellville spring washers to insure that there is no backlash. Acme nuts carry machined helical titanium springs that are driven against the main frame. The stroke of the nuts are controlled by non-jamming stops, and the nuts are kept from

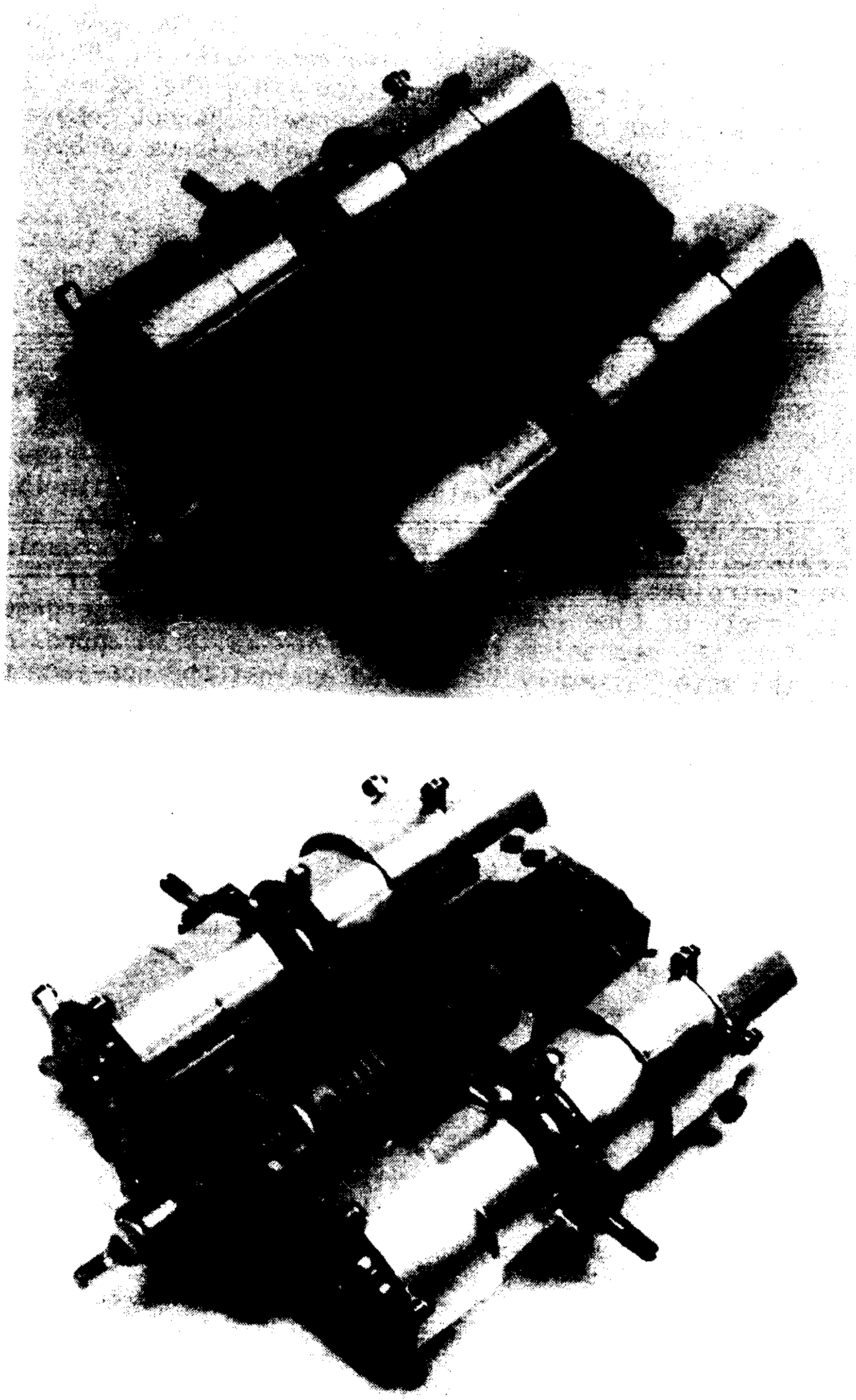

Figure 9. (a) Nutation Damper--Cruise Mode
(b) Nutation Damper--High Spring Rate Mode Engaged

rotating by engagement to an antirotation rod. Both the acme screws and the antirotation rod pass through clearance holes in the main frame. Any contact would cause boom friction since the screws and rod are oscillating with the boom. The steel screw/brass nut combination is lubricated by silicone fluid-impregnated felt wipers on both sides of each nut.

The acme screws are driven by a dual-drive motor assembly (described in Packard, 1982) mounted in the can outboard of the nutation damper. Briefly, the dual-drive motor assembly is a two-motor completely redundant geared driver with no internal single point failures and is an extremely reliable method to actuate spacecraft mechanisms. By using a "sloppy hex" input on the forward acme screw, a flex coupling was deemed unnecessary. The total torque required to engage the high spring rate mode is .56 N-m (5 in-lb) and the total torque required to hold the springs engaged is a minimum of .19 N-m (1.72 in-lb). The dual-drive actuator provides a torque output of 5.7 N-m (50 in-lbs) and a restraining torque of 3.4 N-m (30 in-lb). The motor running time is not controlled by limit switches. Instead, the motor is run for a fixed length of time (110 seconds) to engage and disengage the mechanism. Since the mechanism is operated nominally in approximately 25 seconds, the motor assembly is stalled against the non-jamming stops for the additional time. A telemetry switch indicates high spring rate mode engaged.

Measurement of Performance

Characterising the performance of the devices presented here posed a particular problem. Over twenty various aspects of design and performance were qualified during development, with the majority of the tests listed below:

- Helium Leak
- NDA Dynamic Stiction
- Temp/Vol Compensation
- Dynamic Spring Rate
- Life
- Damping (Static)
- High Temp No-Leak
- Deploy Time vs Temp
- Spring Tube Strength
- Bonding Shear
- T. C. Bellows Pumping
- ACME Screw Efficiency
- Cable Wrap Dynamic Stiction
- Running Friction (Deployment)
- O-Ring/Bearing Breakaway
- Deployment Damping
- Latch Breakaway
- Detent Breakaway
- Low Temp Vac No-Leak
- Vibration (Sine and Random)

Most of these tests were fairly conventional methods of proof testing performance such as vertical hinge-line deployment tests or O-ring breakaway friction tests, and will not be covered in this paper. However, the ultra-precise demands of the system required the development of new test methods. Of particular interest is the dynamic testing of nutation damper and boom hinge stiction, dynamic spring rate versus static spring rate tests, and dynamic versus "static" damping tests.

Dynamic Stiction

The nutation damper had been optimized for the lowest friction possible by holding machining tolerances extremely tight and performing multiple stress-relieving of the titanium hardware during fabrication to control precise parallelism of the moving parts. Also, honing the main frame bores for the linear bearings to .01mm oversize was found to minimize the running friction while still supporting the guide shafts with minimum slop. The performance of the system had to be measured as it was going to be operated; that is, dynamically at very small deflections. Though breakaway friction was measured by adding or subtracting weight to the vertically mounted device until it moved, the very act of how the weight was added varied the results. The end result of the program was to develop a dynamic method that was repeatable. What was done was to drive the nutation damper with a calibrated torque motor of well defined current versus torque characteristics. By fabricating an electrical drive source that provided a sinusoidal decaying or ascending current input to the motor, and plotting that current trace side-by-side the sinusoidal damper displacement trace, the dynamic stiction level would be known when the displacement trace ceased to follow the current trace. In other words, that motor would be providing torque at a level lower than is required to move the device. This threshold would be the dynamic stiction level.

The test set-up consisted of an I-beam pendulum pivoted by a Bendix cantilever flexure and driven by an Aeroflex frictionless torque motor at the pivot point (Fig. 10.) This motor is a limited rotation bearingless assembly that normally is held in position (cantilevered) by the hardware it is assembled to. Unfortunately, alignment difficulties required the motor to be assembled into a housing to support the armature on miniature instrument bearings. Dynamic stiction tests on the complete test set-up without the nutation damper proved that the apparatus' stiction was immeasurable. The damper was fastened to the beam about 21cm from the pivot and a target for a non-contacting linear transducer was mounted at the end of the beam about 65cm from the pivot. By changes in an R-F field, the transducer measured the movement of the target to a resolution of .002mm (.0001 inch) and thus measured the damper oscillations to a resolution of .0007mm (.00003 inch.) As the chart recorder trace in Fig. 11 shows, the stiction threshold for the damper on the down sweep is fairly evident as .017 N-m (.15 in-lb.) Optimization of the mechanism is shown by the fact that the stiction level is nearly identical on the upsweep also.

When the same test method was used to measure the stiction level of the hinge cable wrap assembly at room and cold temperature (less than -100^{o}C), unexpected problems were encountered. Normally, testing at this cold a temperature is performed in a vacuum chamber in an environmental lab. However, because the hinge and cable assembly had nearly zero internal damping of its own it was susceptible to

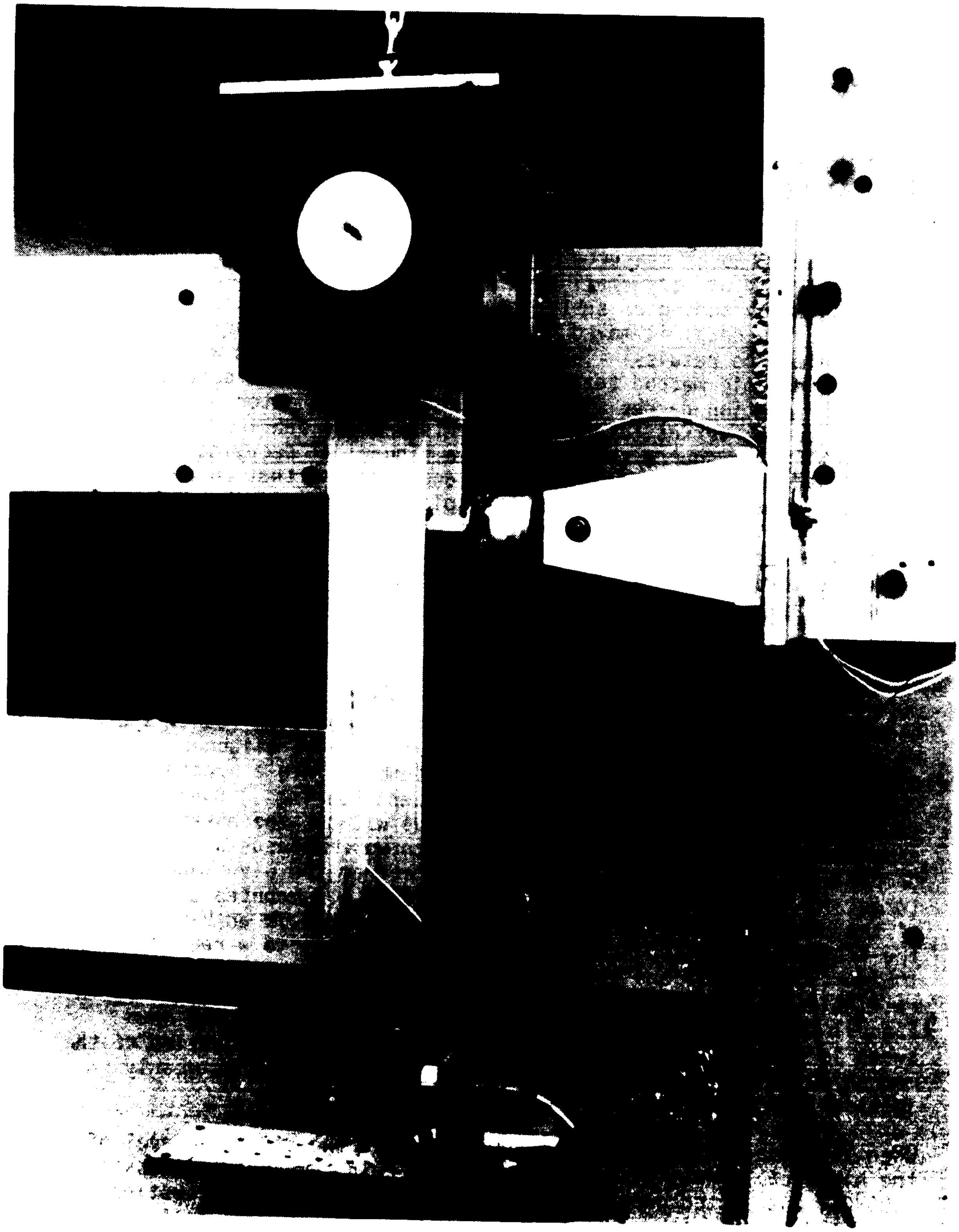

Figure 10. Stiction Torque Test Set-up--Nutation Damper

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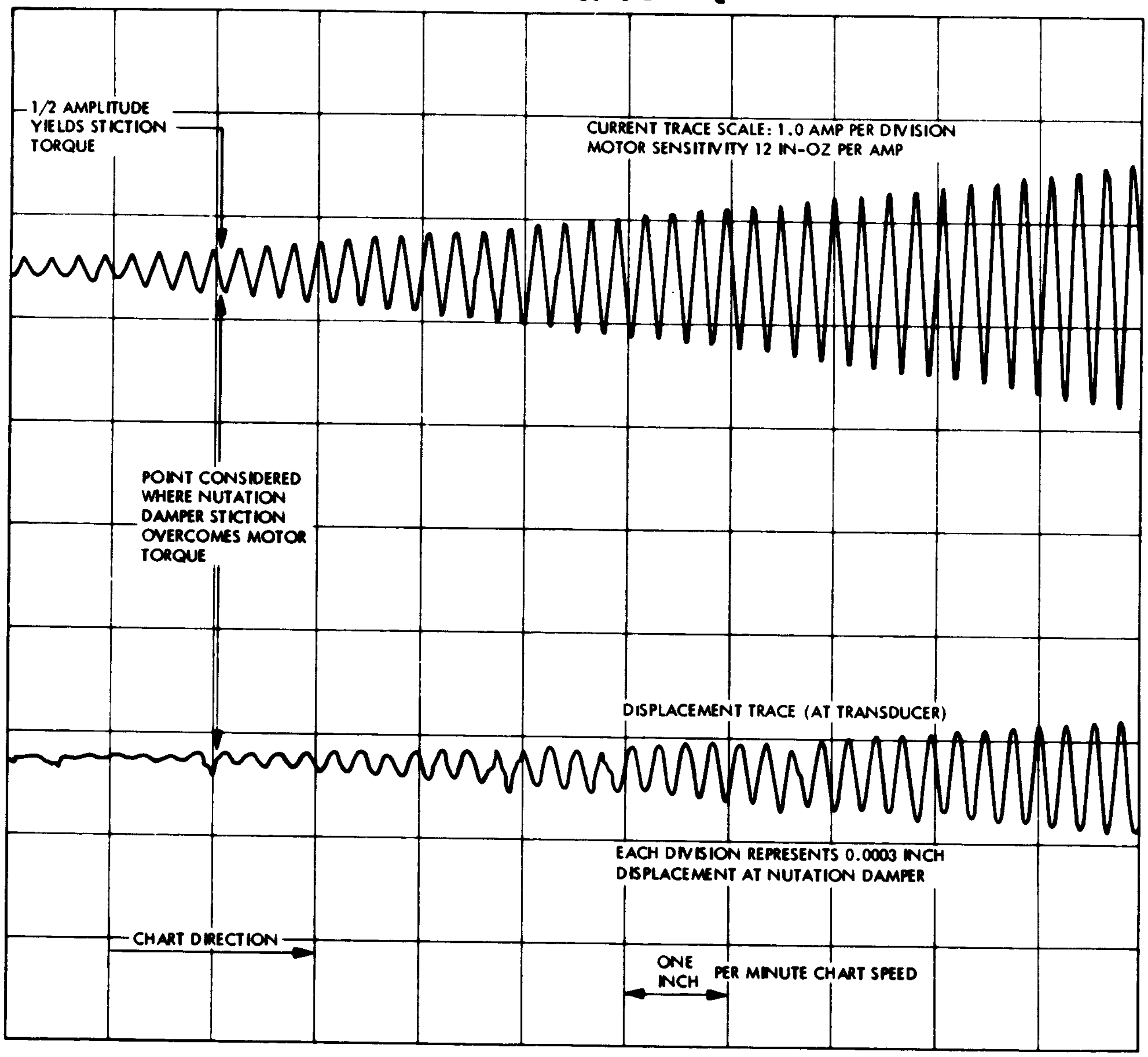

Figure 11. Sample Data--Dynamic Stiction Test

oscillations due to other sources. First it was seen that the vacuum and liquid nitrogen pumps attached to the chamber upset the test. When a run was performed with all pumps in the vicinity switched off, it was discovered that the entire building vibrated at a higher level than the torque that was being measured. By moving the testing back to the building where the nutation damper tests were performed, a sucessful test set-up was developed.

The hinge hardware was suspended by fiberglass thermal isolators in an insulated aluminum box that had clearance openings for the isolators, a fiberglass motor drive shaft, thermocouple wires and liquid and gaseous nitrogen lines (Fig. 12). A frame holding the

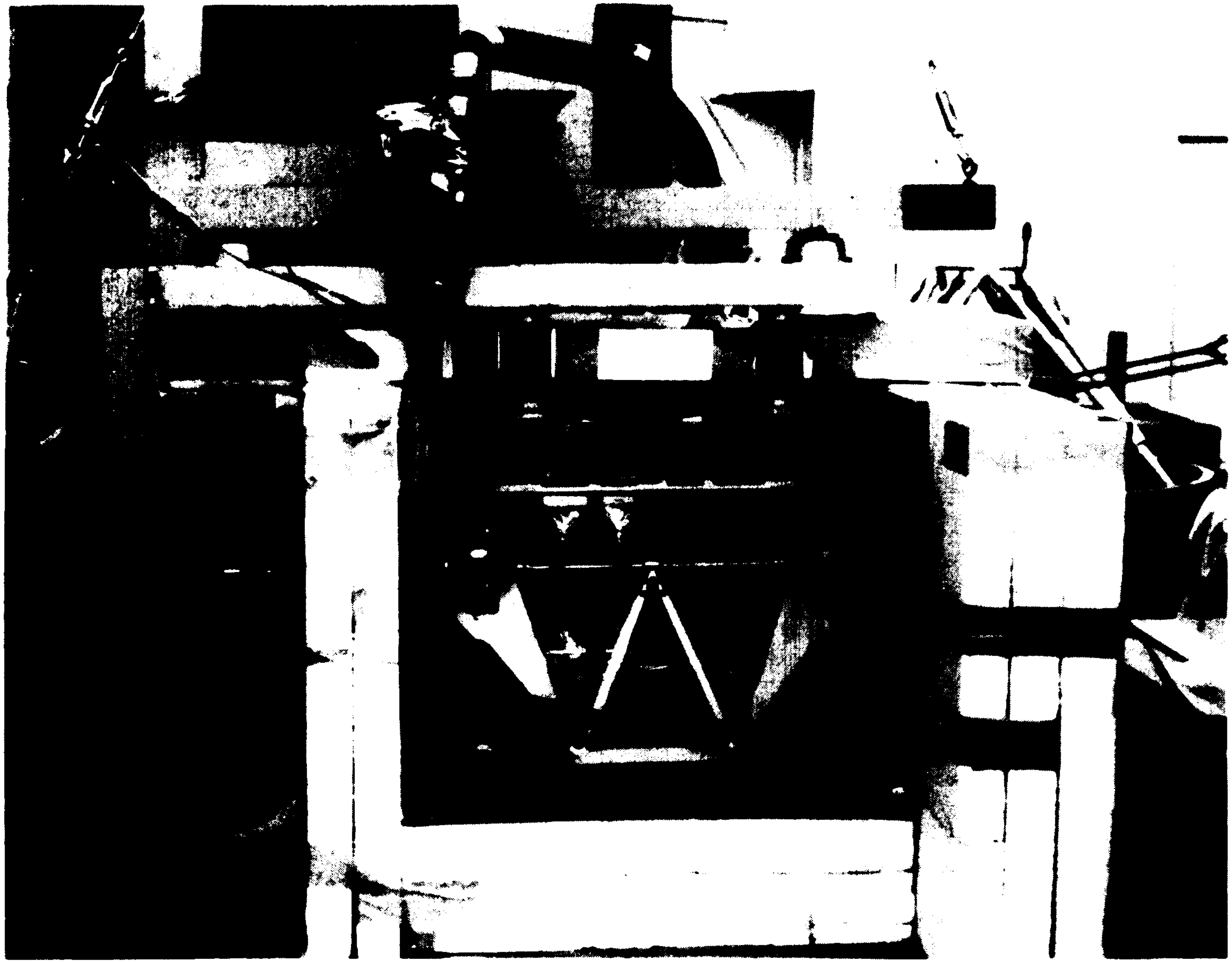

Figure 12. Stiction Torque Test Set-up--Science Boom Hinge

torque motor, linear transducer and the thermally isolated hinge assembly was mounted to the building superstructure through rubber vibration isolators. Two copper cold plates mounted on the inside walls of the box were liquid nitrogen fed in series, and the box was purged with cold gaseous nitrogen off the top of the LN_2 tank. After 10 hours of cooling, temperatures as low as -135°C were achieved in the structure with absolutely no icing problems. This was no easy feat, as previous to this set-up tests were rarely performed at less than -40°C in the atmosphere.

The test runs were performed after hours with all air conditioning equipment in the building shut down. The sensitivity of the test was such that it could be seen on the chart recorder when a truck drove by the building. Driving the hinge as a pendulum, the maximum resolution of the hinge oscillation was 2.7 arc seconds. The stiction of the hinge cable assembly was never actually measured. The displacement trace for the hinge would follow the motor current trace until both were lost in dither, the background noise or the limit of the mechanical and electrical set-up. Therefore, the stiction level had to be at most less than the dither level. This level was between .0007 N-m and .0002 N-m (.0062 and .0018 in-lbs.), depending on how much the chart recorder input was amplified.

Dynamic Spring Rate

Just as the dynamic stiction was measured on the nutation damper, the dynamic spring rate at small deflections could be measured also. By scaling the chart recorder to resolve the high current motor input at the start of the down sweep tests, the displacement versus force (calculated from motor torque input to the pendulum) of the damper/pendulum flexure combination would be shown. This would be an important piece of data as static spring rate testing is a rather inexact procedure when small deflections are used. By mounting the damper vertically and measuring the displacement of the frames as weight was added or subtracted, the spring rate was readily calculated. But as the magnitude of the weight was reduced for smaller and smaller deflections, the resulting displacement varied tremendously. Though in the 6mm to 25mm range the average spring rate or the damper was 2.8 N/cm (32 lbs/in), when deflections of less than .89mm (.035 inch) were used the calculated spring rate would vary from .18 N/cm to 6.73 N/cm (2 to 76 lb/in.) This was a haunting problem even when the weight used was liquid siphoned to a cup on the damper in as vibrationless a method as possible. By the dynamic method the spring rate of the damper is expected to be much more consistent. By measuring the dynamic spring rate of the pendulum and then the pendulum/nutation damper test set-up, the dynamic spring rate of the nutation damper will be resolved. As the damper has to be empty of silicone fluid for this test, it will be performed the next time the damper is disassembled.

Dynamic Damping Value

At the time of this writing, the dynamic damping test has not been performed. The damping value of the nutation damper was initially set by "static" damping tests; that is, displacing the center frame of the vertically held damper and recording the displacement versus time to return to the initial position. By mathematically curve-fitting the damping constant can be calculated. However, this method of measuring the damping value is in no way similar to the function of the device. It is possible that the actual dynamic damping characteristics would be different in the ± .89mm (.035 inch) oscillation at .01 hertz predicted service of the device. Therefore, tests are being prepared in which the damper will be driven by a motor at these values and the damping-constant will be calculated. By measuring the velocity of the damper in relation to the force input to the damper frame, the damping-constant is calculable. Since, by definition, damping is the force that is in-phase but opposite in direction to the velocity of the object, this data can be recorded concurrently on magnetic tape and manipulated by a computer to directly output the damping-constant.

Conclusions

Though the spacecraft requirements were severe in terms of the required precision of the Nutation Damper System, the design of the mechanism was initially promised as a "best effort." It was not known what performance level could be obtained. Some development was trial-and-error, but the design changed little throughout the program. The noteworthy achievement was being able to reliably gauge the performance of a mechanism by innovative test methods, allowing ultra-low friction levels to be resolved and separated from damping characteristics. These test methods showed that the Nutation Damper System will surpass the performance level required.

The device is currently scheduled to complete qualification level vibration and low temperature vacuum soak tests in early 1983 and will be delivered to perform in the spacecraft system testing in mid-1983. The Galileo spacecraft is scheduled to be launched to Jupiter in May 1986.

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