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ACTUATOR DEVELOPMENT FOR THE INSTRUMENT POINTING SYSTEM (IPS)

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ABSTRACT

This paper gives a brief introduction to the mechanisms of the instrument pointing system (IPS). Particular emphasis is placed on the actuators which are necessary for operating the IPS. The actuators are described as follows:

- Two linear actuators that clamp the gimbals down during ascent and descent
- Two linear actuators that attach the payload to the IPS during the mission, and release it into the payload clamps
- One rotational actuator that opens and closes the payload clamps
- Three identical drive units that represent the three orthogonal-gimbal axes and are the prime movers for pointing

Design features, manufacturing problems, test performance, and results are presented in this paper.

HISTORY

In 1972, Dornier under the sponsorship of the European Space Agency, began investigating the feasibility of an instrument pointing system (IPS) to be flown on a Spacelab Pallet on board the Space Shuttle. The task of IPS would be to support solar, stellar, and Earth sensing payloads during launch and landing, and to point them on-orbit, with arc-sec accuracy.

The design phase started in 1976 and until 1980 the qualification programme was proceeding successfully. Because increased Shuttle mechanical environment and changed Shuttle and Spacelab requirements did not allow the original concept to continue, a rigorous redesign was initiated that resulted in the totally new construction shown in Figure 1. Only a few parts of the original concept survive in the new design, although the experience gained from the previous phase was very useful for the new start. See Figure 2.

Since that time the new IPS has almost completed its qualification programme and the activities for the July 1984 delivery are proceeding.

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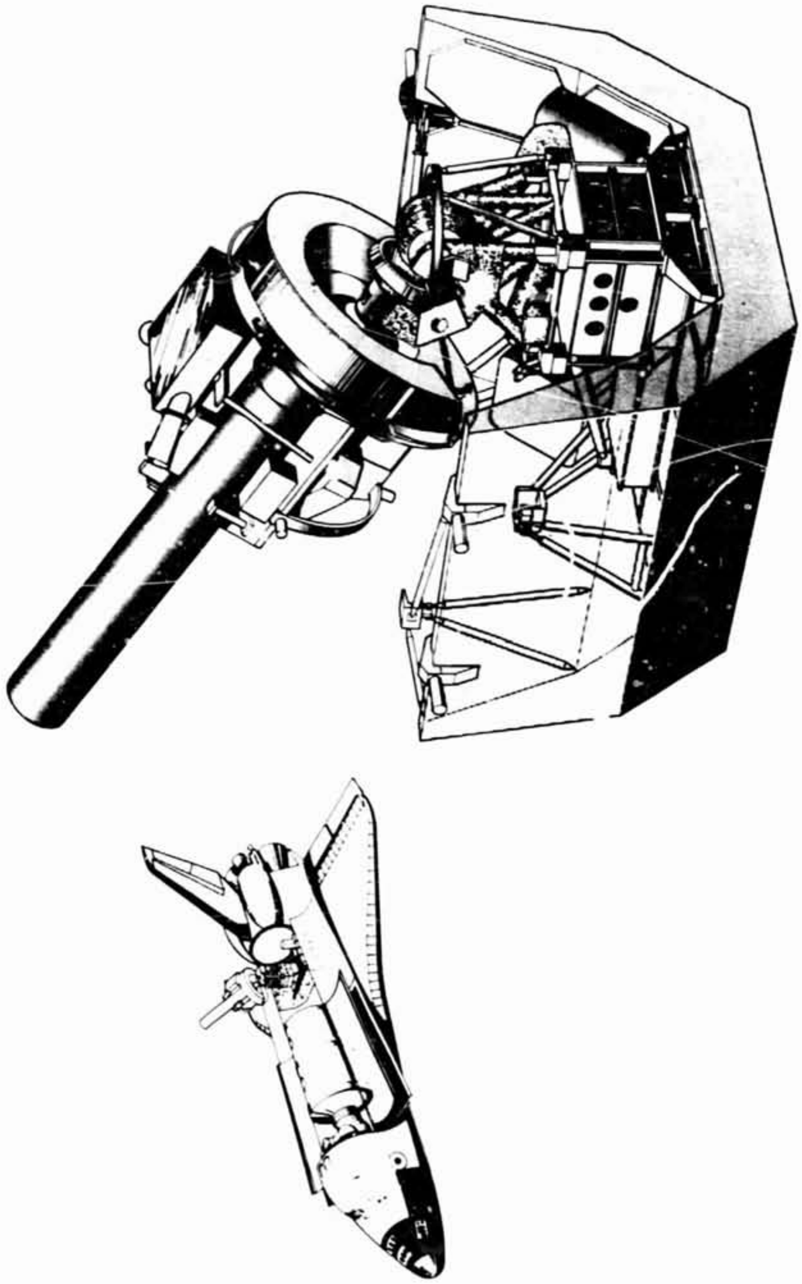

Figure 1. Configuration of IPS

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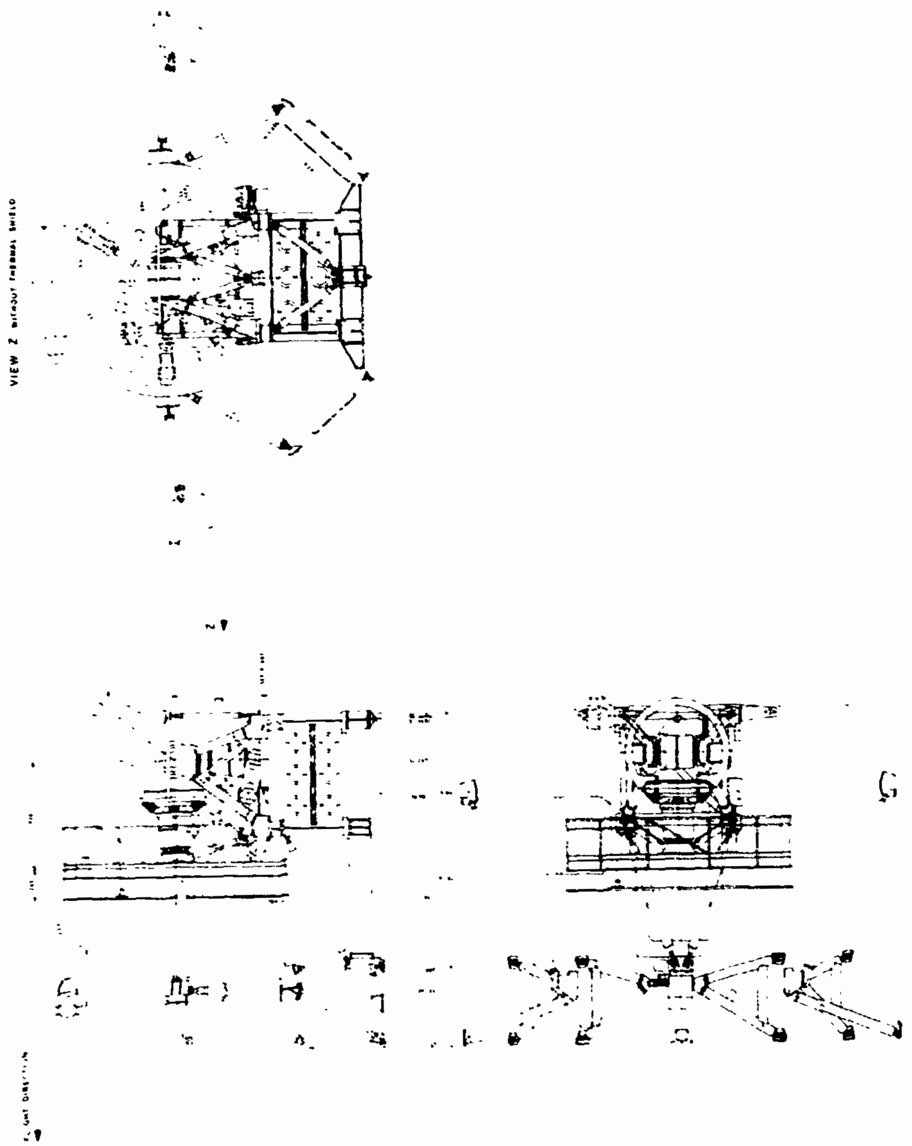

Figure 2. Three-Sided View of IPS

PERFORMANCE DATA

The IPS is designed to include a wide range of payloads for solar, stellar, or Earth sensing missions. To cover all imaginable demands of potential users very severe requirements had to be observed and fulfilled by an adequate layout. The main features of the IPS are:

a. IPS weight--1200 kg

b. Payload data (max)

- Weight--3000 kg
- Diameter--3 m
- Length--3 m

c. Pointing precision--2 arc-sec normal to line of sight--15 arc-sec around line of sight

d. Usable viewing angle--$\pm 60°$ half-cone angle; $\pm 180°$ around line of sight

e. Lifetime--10 years or 50 missions

MECHANICAL CONFIGURATION

Three drive units form the orthogonal axes of the gimbal system. Structural members connect them mutually with the pallet and the payload.

The gimbal latch mechanism, driven by one of a pair of linear actuators, fixes the gimbal system rigidly to the mounting structure during ascent and descent. For the mission, the latch opens to allow the gimbals to be moved. Normal operation requires only one actuator for safe locking in the launch and landing configuration; the second actuator is a backup if the first should fail. The latching concept consists of two hooks attached to the movable gimbals and two cranks driven by the actuators, which catch and pull the hooks down to the structure.

During ascent and descent, the payload is separated from the gimbal system. The payload gimbal separation mechanism is used for attaching the payload to the gimbals. One of a second pair of linear actuators activates this mechanism. The actuator pulls the payload out of its clamps, which have been previously opened, and provides a rigid connection between the payload and the gimbals by means of a cable and pulley system. When the cables are released by the main actuator or in a failure case by the redundant actuator, leaf springs push the payload away from the gimbals back into the clamps.

The three clamps are V-shaped housings with a sliding keybolt that closes the opening, each capable of accommodating one of three trunnions bolted to the payload. The keybolts are operated simultaneously by flexible rotational shafts driven by one rotational actuator.

LINEAR ACTUATOR

Performance requriements are as follows:

- 18 Vdc, 1.9A-maximum
- 120-mm stroke
- 6000-N force
- -30° to +70° operating temperature
- 2800 cycles under ambient and orbit environment

Design Description

A design that used a dc brush motor driving a harmonic drive gear and a threaded shaft/nut assembly was chosen to meet the previous requirements (Figure 3). Titanium was selected for the structural parts owing to its good stress-to-weight ratio and the thermal requirements.

The frameless dc motor turns at approximately 200 rpm, depending on the applied load. The rotor is bolted to the wave generator of the harmonic drive gear, which reduces the speed by a ratio of 1:78. The output of the gear is transmitted by a hollow shaft to a rotating nut in which a nonrotating threaded shaft, with a pitch of 8 mm, can move longitudinally. The shaft is attached to a plunger sliding in a tubular section of the housing.

A prism fixed to the plunger penetrates a slot of the outer housing and prevents the plunger and subsequently the shaft from rotating. Also the slopes of the prism operate the actuator endswitches.

With the exception of the harmonic drive, all rotating parts are mounted on deep groove ball bearings of the same size. Races and balls are manufactured from AISI 440 C CEVM, and the cages from phenolic resign impregnated with Fomblin oil Z 25. This lubricant is also used for the harmonic drive and the nut/shaft assembly.

Fabrication Problems

Because of mass restrictions, it was necessary to machine the parts dowN to the absolute minimum, which led to very expensive, lightweight filigree parts.

Computer calculations taking temperature influences and applied loads into account led to extremely close tolerances for the ball bearing fits. Even under these conditions smooth running and proper performance had to be guaranteed.

Test Performance

For qualification testing, one qualification model actuator was built in addition to the four flight models. A normal test programme with vibration, thermal-vacuum, life-cycle and functional performance tests will be used to qualify the actuator.

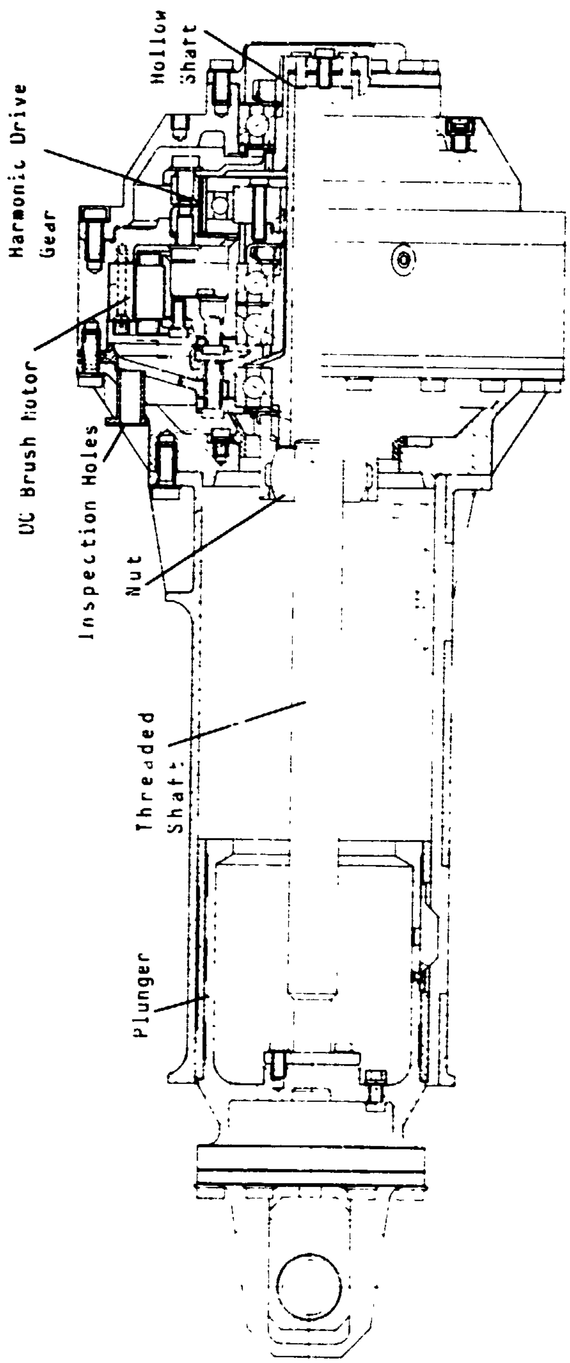

Figure 3. Linear Actuator

Problems

Originally Dornier intended to hermetically seal txe entire actuator; the gaps between the flanges ere equipped with O-rings, and the sliding plunger was connected with the housing by a metal bellows. This seal was difficult to maintain because:

- The atmosphere inside the actuator must contain a carefully controlled amount of moisture which is required to achieve good motor brush performance
- The rubber seals become hard at low temperatures and leak under temperature gradients between the flanges
- The welded bellows cracked during the vibration test

After this experience the motor manufacturer was requested to provide motor brushes capable of operating in vacuum. Their positive answer encouraged Dornier to abandon the sealed concept.

During the thermal-vacuum testing however, the brushes failed after the low-temperature test. To find a suitable brush material, an investigation was begun but it was not successful. Brushes were subsequently declared as life-limited items that have to be inspected regularly, and exchanged if necessary, and the mechanical design was modified to ease inspection and replacement.

Owing to the very short lifetime achievable by the brushes--little more than one 7-day mission--the search for more durable brushes is continuing with the aim of increasing the number of missions using the same brush set.

ROTATIONAL ACTUATOR

Performance requirements are as follows:

- 18 Vdc, 3.6-A maximum
- Two motors in one housing
- 60 revolutions cw and ccw at three outlets
- 2.4 Nm maximum at each of the three outlets simultaneously
- -25° to +90°C operating temperature
- 1400 cycles under ambient and orbit environment
- Manual operation must be possible

Design Description

The rotational actuator (Figure 4) consists of two major parts: the motor housing and the gear box.

Two dc brush motors are mounted on a single common shaft inside the motor housing. The motors and shaft bearings are the same as those used in the linear actuators.

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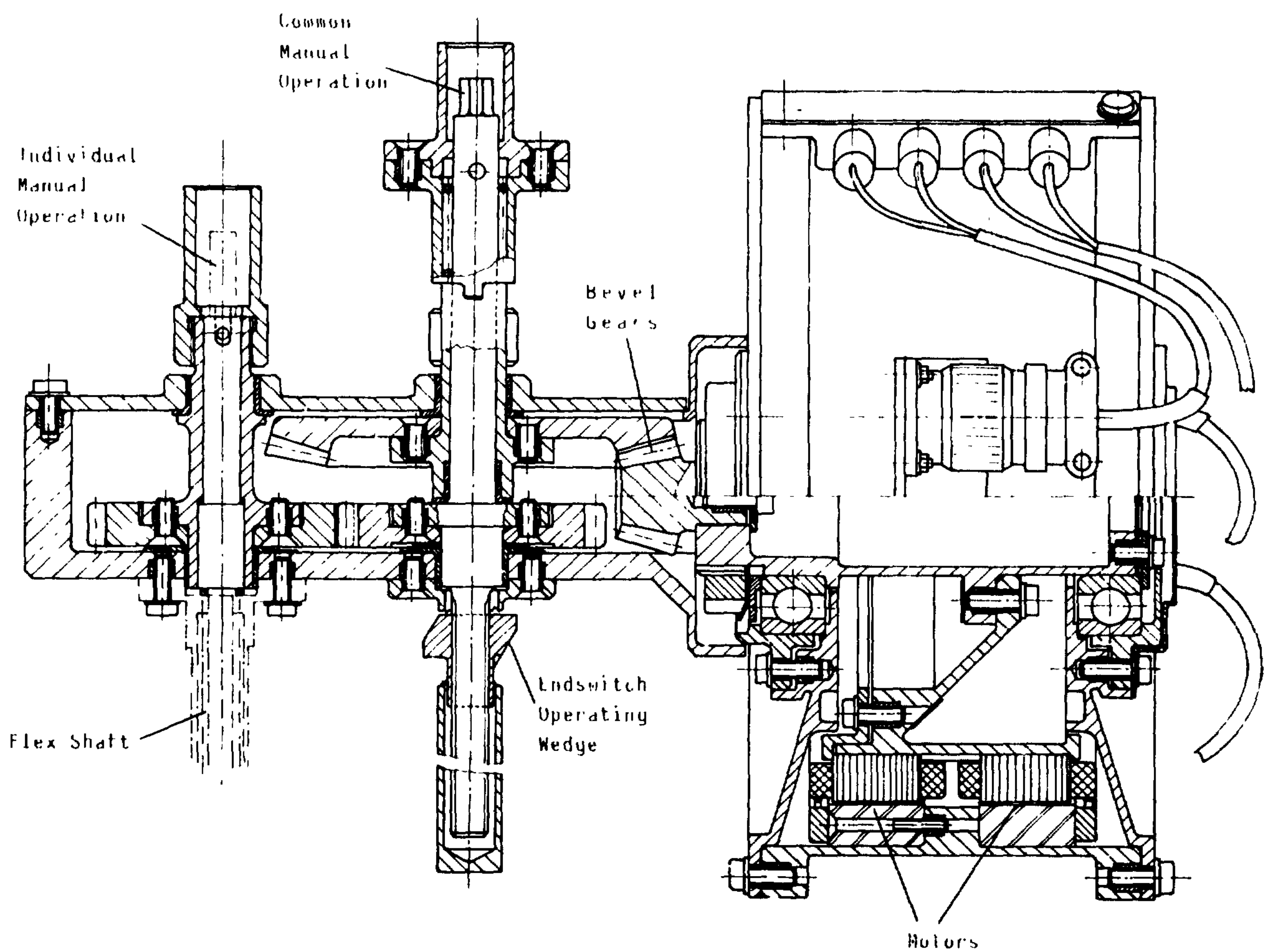

Figure 4. Rotational Actuator

A bevel drive gear located at the end of the shaft is in contact with a bevel crown wheel, both located inside the gear box. This first gear stage reduces the motor speed by four. To distribute the motion to the three payload clamp flex shafts a spur gear with a central wheel and three adjacent wheels is used.

The wheel shafts are mounted on dry lubricated journal bearings.

In an emergency, the flex shafts can be manually operated either individually or by the common central shaft, using a special torque limited detachable crank tool.

At the two extreme positions of the keybolt in the payload clamps, the motor is switched off by two sets of endswitches (not shown in Figure 4). The switches are operated by wedges sliding along a thr ded spindle whch extends from the central gear shaft.

Test Performance

One qualification model actuator will undergo the qualification programme of vibration, thermal-vacuum, life cycle, and functional performance tests. A set of flexible shafts will be tested in parallel.

To simulate the mechanical actuating forces, either a payload clamp or load simulators using a thread/nut assembly and springs may be used. The simulator is designed to perform under ambient and vacuum conditions.

Problems

Because the same motors were used as in the linear actuator, this assembly had similar motor brush problems. Operation of the payload clamps requires fewer motor revolutions, so a higher number of missions between brush exchanges can be anticipated.

GIMBAL DRIVE UNIT

Performance requirements are as follows:

- 18 Vdc, 9 A-maximum
- 15 Nm torque maximum
- 2 arc-sec pointing accuracy
- -10° to +80°C operating temperature
- 25.2 kN axial load; 43.5 kN and 13 kNm lateral
- 30000 cycles over $\pm 180°$ and 500,000 cycles over $\pm 5°$ under orbit environment
- Infinite number of revolutions of the bare drive, equipped with cable feed through $\pm 60°$ to $\pm 193°$.
- 200 signal and 290 power lines across each drive unit

Design Description

The main active elements of the IPS gimbal system are the drive units. Three drive units form the three axes and provide

- The capability to carry the loads of ascent, descent, and ground operations
- Sufficient angular freedom
- Low friction torques over the whole travel
- Generation of torque to move the gimbals
- Position indication
- Passage for approximately 500 electric leads

When the IPS was redesigned, the drive units were also significantly reconfigured (Figure 5). Whereas in principal most of the functions remained the same, the drive units were simplified and some of the functions removed and separately allocated.

The main differences are:

Old	New
a. Load by pass for: • Unloading the bearings • Active emergency braking • Self aligning to zero • Locking for ascent	a. No load by pass, instead: • Sufficiently dimensioned bearings • External passive end stop • External guiding slots • External gimbal latch mechanism
b. Cable follow-up with spirally wound special flat-band cables	b. Cable feedthrough with normal cables running axially through the hollow shaft
c. Numerous parts, complicated function, hermetic sealing necessary	c. Fewer parts, simple function capable of operating under ambient and vacuum conditions
d. Small bearings requiring auxiliary bearing for ground operation and an offloading device	d. Bigger bearings with no extra measures required
e. Bearings with expensive titanium carbide coated tungsten balls	e. Normal stainless-steel bearings
f. "Soft" shaft with changing diameters and cutouts, set together from several pieces	f. One solid stiff shaft

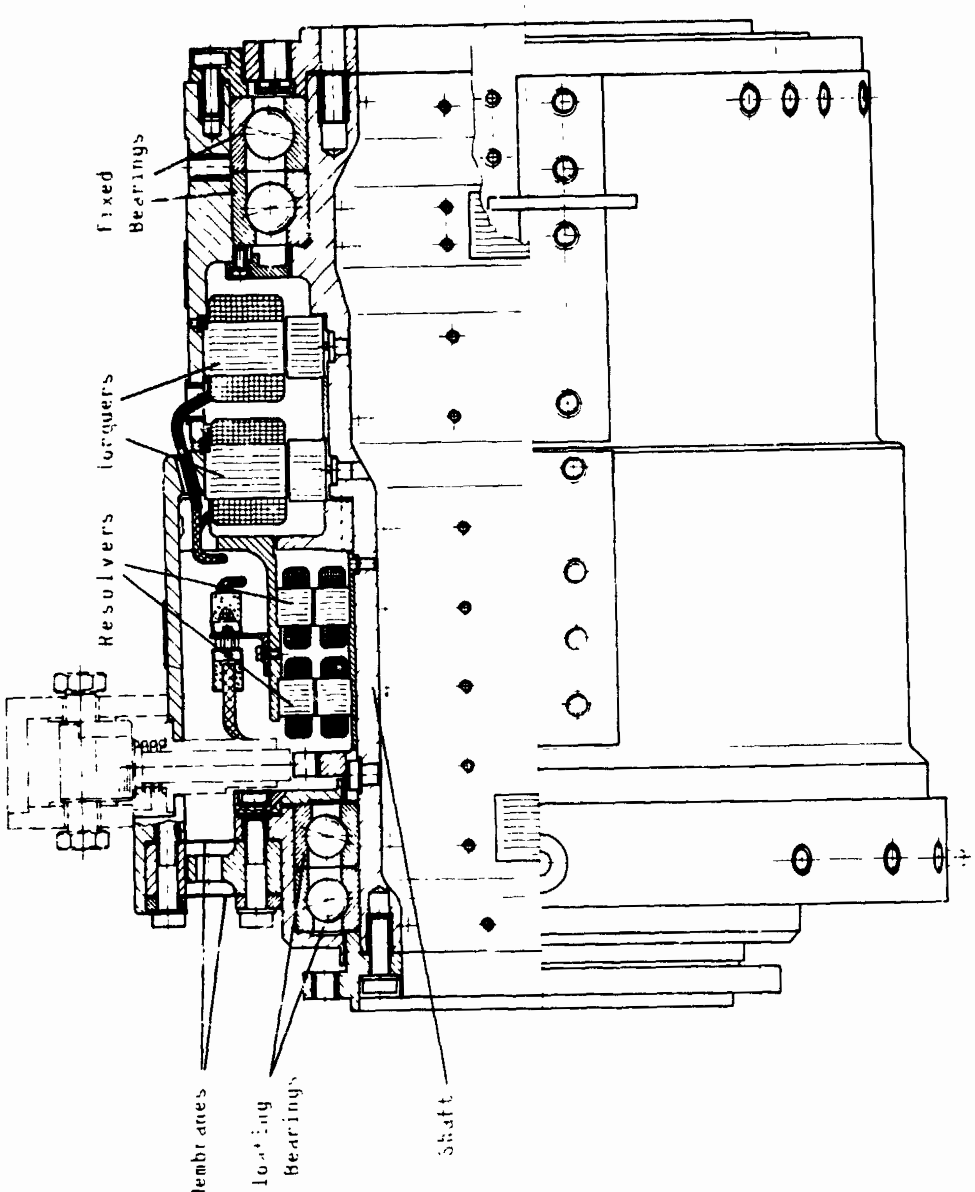

Figure 5. Drive Unit

The new drive unit features two pairs of angular contact ball bearings arranged in a face-to-face configuration. Races and balls are manufactured from AISI 400 C - CEVM, with ABEC 7 (ISO 4) tolerances, the cages from phenolic resin with 2-percent porosity. The bearings are lubricated with Fomblin oil Z 25.

The larger diameter pair are rigidly fixed carrying both axial and radial loads, and the smaller pair are mounted on membranes and float axially.

The frameless brushless dc torque motors, the only items taken over from the previous design, use a samarium cobalt permanent magnet rotor and a sine and cosine winding in the stator.

Two resolvers, matching the torquers in number of polepairs and windings, provide the torquer position and speed-control signals. Two singlespeed resolvers, integrated within the previously mentioned resolvers, determine the angle between shaft and housing.

Torquers and resolvers are mutually aligned by locator pins in shaft and housing, respectively. The cable feedthrough is not shown in Figure 5. Almost 500 signal and power leads run axially through the hollow shaft, with power and signal leads separated by a concentric metal tube. This arrangement has to allow a rotation of up to $\pm 193°$ without a significant increase of torque.

Fabrication Problems

For weight and thermal reasons, the housing, the shaft, and the flanges of the drive units have been made from titanium. Highly accurate machines (partially with air bearings) were used to turn the bearing fits. The tolerances of these were calculated by computer in close cooperation with the bearing manufacturer. Tolerances of better than 6 μm in diameter and 3 μm in roundness were necessary.

In one case a kind of memory behaviour of the titanium was observed caused by an unfavourable clamping of the workpiece. To recover the out-of-tolerance part the distorted area was chemically nickel plated and remachined.

Test Performance

Besides the electrical and vibration tests, special attention was directed to the torque behavior of the drives. The old drive unit concept was designed for the lowest possible torque, which resulted in very small bearings with their attendant disadvantages, as previously described. New investigations into the control loop, however, revealed that the total system performance was not affected by higher torques. To determine these torques with and without a cable feedthrough under different environments, special test equipment was designed.

The drive unit is mounted on a rotating vertical axis turntable, driven by an electric motor through a 6000:1 speed reducer. The speed and position are monitored by a tachogenerator and a potentiometer, respectively.

The drive unit shaft is attached to a piezoelectric torque transducer. The whole assembly is mounted in a tubular structure that can be placed, completely, into a thermal-vacuum chamber. By rotating the turntable, the net resistance torque of the drive unit, with or without cable feedthrough, is indicated on the torque transducer. To cover the requirements for both pointing and slewing modes of operation, test runs were made for both small and large angles of rotation. The measurements revealed a torque of approximately 0.5 Nm for the bare drive unit without the cable feedthrough and approximately 4 Nm with the cables installed at $\pm 193°$. Under cold temperature influence, a slight torque increase of about 5 percent was observed.

A life cycle test was also performed in this test rig. The motor and the torque transducer were decoupled and the internal dc torquers were activated by a control and measuring unit consisting of a normal programmable desk computer with a subsequent power stage. The shaft position was monitored by the internal resolvers. During this test, $\pm 180°$ cycles equalling 30,000 revolutions and 500,000 cycles of $\pm 5°$ at extreme temperatures were performed and revealed no change in torque.

A fatigue and a quasistatic load test was conducted that used a vibrator on which the housing was fixed and a mass dummy of 160 kg attached to the shaft by a 420-mm long stiff beam. Accelerations of up to 20 g for a duration equivalent to 200 missions were applied, which yielded an axial force of 25 kN and a bending moment of 13 kNm.

Final resolver zero and stability checks and torque tests did not show deviations from the former results.

After these qualifications tests, the cable feedthrough was checked for cable breakage and insulator damaged with positive results. Thereafter, the cables were stripped and inspected, but no damage or abrasion was found.

The drive unit is fully qualified and without any reservations and showed its ability to meet all requirements and to operate successfully in the IPS.

CONCLUSIONS

The active mechanisms for the IPS are partially qualified: some tests are still running.

The mechanical performance of the linear and rotational actuators is proven; as their capability for meeting the requirements has been demonstrated. The remaining brush problem is under investigation. It is not likely a brush can be found that survives the entire required lifetime, but the aim, however, is to provide brushes for at least a limited number of missions.

The drive unit was fully qualified in an extensive test programme. In addition, the three flight units have been successfully acceptance tested and are integrated in the IPS.

No severe problems can be reported from the other IPS subsystems. The system integration is currently under way. After the final integration of IPS, some system tests are planned. At present, no complications are anticipated so scheduled delivery of the first IPS to ESA is planned for July 1984. A second IPS will be delivered to NASA half a year later.

We hope that the IPS will provide scientists with a precise tool on which to mount their instruments and to obtain accurate observations and measurements, enabling them to increase our knowledge of the world and the universe.