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EVOLUTION FROM A HINGE ACTUATOR MECHANISM TO AN ANTENNA DEPLOYMENT MECHANISM FOR USE ON THE EUROPEAN LARGE COMMUNICATIONS SATELLITE (L-SAT/OLYMPUS)

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ABSTRACT

This paper describes the evolution of an antenna deployment mechanism (ADM) from a hinge actuator mechanism (HAM). The result is a mechanism capable of deploying large satellite appendages in a controlled manner.

The development testing of the HAM identified many improvements to the concept which were incorporated into the ADM. Both of the designs are described in detail and the improvements made to the ADM are highlighted.

INTRODUCTION

The hinge actuator mechanism (HAM) concept was developed by British Aerospace (BAe) in response to an ESTEC development programme.[1] The experience gained in developing the HAM has been applied in the design of the antenna deployment mechanism (ADM), which originally was for the OLYMPUS telecommunications satellite programme but has been further developed for the UNISAT and EUROSTAR satellite programmes.

The HAM was conceived as a device able to deploy large satellite appendages in a controlled manner with a low speed of deployment such that the disturbance impulse to the satellite was minimal. The energy supply to power the HAM was self-contained and required no external actuator to enable operation. The appendage to be deployed would be held down to the satellite by a pyrotechnic release assembly, release of the mechanism would only require power to fire the pyrotechnic and would therefore minimize power requirements for deployment.

The development of the ADM, which required a significant increase in the available deployment torque coupled with a decrease in rotational speed, minimizes any increase in size or mass.

Details of the ADM application within the OLYMPUS antenna deployment subsystem has been included to show a typical application of the ADM and to highlight the operational requirements to be satisfied by the mechanism.

MECHANISM DESCRIPTION

Hinge Actuator Mechanism

Essentially, the HAM is a shaft supported on rolling element bearings that is driven by a large diameter helical torsion spring. The shaft, which in application, would be coupled to a large inertia, has its rate of deployment controlled by an eddy current damper which is driven by a shaft at high speed via a high ratio gearbox (Figure 1).

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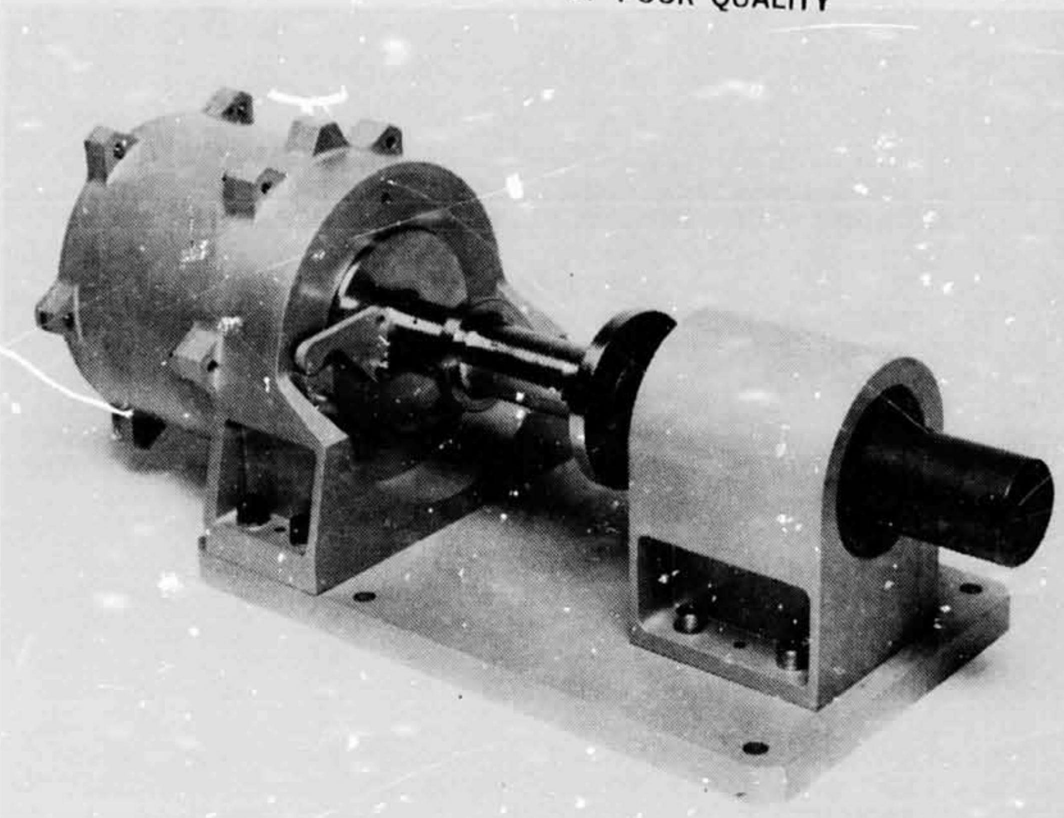

Figure 1. HAM Assembly

The mechanism was designed to give satisfactory performance against the following design requirements:

a. Hinge load

- Inertia–Up to 100 kgm^2
- Mass–Support 12 kg at launch
- Acceleration–30 g caused by vibration at launch
- Resistive torque–Up to 0.5 Nm
- Radial load–1200 N
- Axial load–120 N

b. Range of rotation–90°

c. Speed of rotation–Between 0.2 deg.sec^{-1} and 0.5 deg.sec^{-1}

d. Accuracy of deployment–0.05° or better after latching

e. Mass of HAM–1.45 kg

The design, which is to be externally mounted on the satellite sidewall, is capable of successful operation in a thermal environment of -50° to +70°C. The main design features in the HAM are summarized for comparison with the ADM.

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Shaft and Main Bearings

The main shaft is supported by a deep groove ball bearing and linear bearing assembly. The deep groove bearing provides the axial location and the in-line linear bearing allows for thermal expansion. The bearings and shaft are sized to accommodate anticipated launch loads of 30 g acceleration acting on the 12 kg mass. Liquid lubricant was used for the main bearing for a low friction torque. The linear bearing was not lubricated but ran on a solid molybdenum disulphide film. Both bearing housings were mounted on a common base to reduce misalignment. In a flight case it was anticipated that any in-line bearings would be separated by about 1 meter and would support an antenna backing structure. As previously stated these bearings would be designed to support only radial loads, the axial movement being unrestrained to accommodate thermal expansions. The shaft in the HAM represents the backing structure, and was manufactured from steel. The bearing housings and base were manufactured from aluminium alloy for lightness.

Spring

The drive spring is a large diameter helical torsion type driving the output shaft directly. The spring is housed within the gearbox and damper subassemblies and attaches to the main shaft adjacent to the main bearing. The housing allows sufficient clearance for the increase in diameter of the spring during deployment and provides adequate support.

The spring has been sized to generate a torque of 1 Nm, providing a comfortable margin over the anticipated resistive torque limit of 0.5 Nm. The steel wire spring was solid lubricated using a molybdenum disulphide film to reduce friction between the coils and housing.

Gears

The output shaft of the HAM drives the damper assembly via a high ratio gearbox. The overall gearbox ratio of 225:1 is obtained using four passes, this is to minimize the size of the gearbox while maintaining adequate tooth strength. During deployment the damper will balance the excess spring torque, the gearbox must therefore be capable of transmitting 1 Nm applied at the drive shaft.

The gears are isolated from the deployable structure launch loads by mounting them out board of the main bearing. The gears and their associated bearings need only be capable of transmitting the torque and containing the gear separation forces during deployment. Because of the high overall ratio the friction torque must be minimized. Therefore, the gears are dry film lubricated using molybdenum disulphide and the bearings are oil lubricated with Fomblin Z25.

An escapement device was incorporated into the design to eliminate any excessive torque being applied to the gearbox when back-driven, thus the design allows simple integration and stowage of the appendage to be deployed.

The escapement device is a one way rolling element clutch that is oil lubricated using Fomblin Z25.

The gears are standard spur gears with the full tooth form and 20° pressure angle. Because of the method of lubrication the gears were manufactured from stainless steel and surface hardened to 60 Rockwell 'C'.

Damper

An eddy current damper is used which utilizes the damping effect of eddy currents induced within a conducting disc rotating in a magnetic field. The induced current, and the energy dissipated, is related to the speed of rotation and the magnetic field strength. The speed necessary to provide damping can be minimized by optimizing the field with respect to the magnetic geometry.

The magnetic circuit was created by 1[illegible] magnet pairs (Figure 2), set radially on the damper housing. Simple bar magnets with shaped pole pieces to concentrate the flux between the magnets were used to optimize the magnetic field. Commercially available samarium cobalt magnets and low carbon steel pole pieces were used.

Mass

The total mass of 1.45 kg could be significantly reduced by the adoption of lightweight materials, for example an aluminium shaft and Delrin, an acetal copolymer resin, could be considered for the housing and gear materials.

Performance of the HAM

A test programme designed to fully test the HAM against its design requirements resulted in the following evaluation of the HAM.

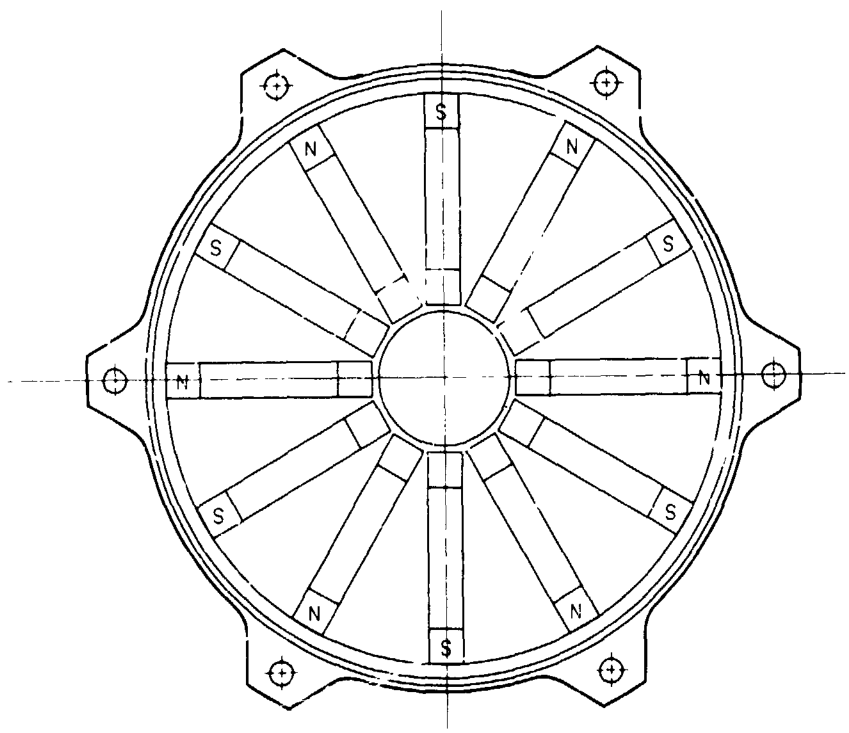

Figure 2. HAM Magnet Layout

The HAM satisfied all the functional tests and operating reliability with the exception of deployment speed. The damper design used pure aluminium disc and the actual manufactured unit used an aluminium alloy with a higher re[illegible]ivity. Under test a deployment speed of 1.3 deg.sec^{-1} was obtained. This speed compared with a calculated figure of 1.[illegible] deg.sec^{-1} (using the higher resisitivity). The accuracy of deployment was well within 0.05°, being 0.01° with a repeatability of deployment better than ±0.004°.

Conclusions

The HAM programme proved that it was feasible to produce a mechanism capable of accurately deploying an inertia, reliably and at a controlled speed. The error in the deployment speed caused by a change in disc material highlighted the sensitivity of the eddy current damper and magnetic field theory, and the difficulty of reliably predicting the magnetic field strength and the speeds at which the desired damping torques could be achieved.

DEVELOPMENT OF THE ANTENNA DEPLOYMENT MECHANISM

The OLYMPUS telecommunications satellite programme required a multipurpose mechanism capable of a variation in torque output and deployment angle but able to control the rate of deployment for [illegible] expected output torques. During development of the ADM, experience that was gained from the HAM programme was extensively applied to the ADM's design. The areas where the most significant improvements were made have been highlighted within this paper.

The ADM concept is fundamentally the same as that of the HAM (i.e., a spring driven shaft mounted in rolling element bearings and controlled by an eddy current damper). The design requirements for the ADM, which have been modified to satisfy the multipurpose roles, are summarized for comparison with those of the HAM as follows:

- Hinge capacity–Up to 27 Nm damping requirement
- Deployment angle–Up to 90°
- Deployment accuracy and repeatability–±0.01°
- Resistive torque–0.5 Nm
- Lifetime–5 years storage, 50 'ground' operations, 10 year operational life
- Drive torque–At least 5 times resistive torque
- [illegible]e shock to spacecraft–[illegible] 1 Nms
- [illegible]ads–2700 N in any direction

[illegible] design improvements in spring motor and damper design are highlighted in the following [illegible]graphs.

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Spring Motor

The driving torque capacity for the ADM was increased to 14.85 Nm. As it was not feasible to use a helical torsion spring an alternative spring motor was developed. The motor consists of six constant torque springs each with a torque capacity of 0.1 Nm. The springs are mounted on individual bobbins, and the end of each of the springs is fastened to a common output shaft. To operate the motor, the output shaft is back driven, causing the individual springs to be reverse wound from their bobbins to the common shaft. The shaft is coupled to a two pass reduction gearbox, the ratio of which is 24.5:1. This results in the required output torque at the main shaft. Operation of the motor occurs as the individual springs rewind onto their bobbins.

One advantage of the spring motor is that each of the springs unwinds onto individual bobbins, and because the springs are free to rotate on the bobbins there is no chance of motor seizure, resulting in high reliability. High reliability is achieved by provisions for six individual springs, compared to the single spring of the HAM. Variation in output torque is achieved by varying the number of springs.

The detail design of the spring motor is shown in Figure 3.

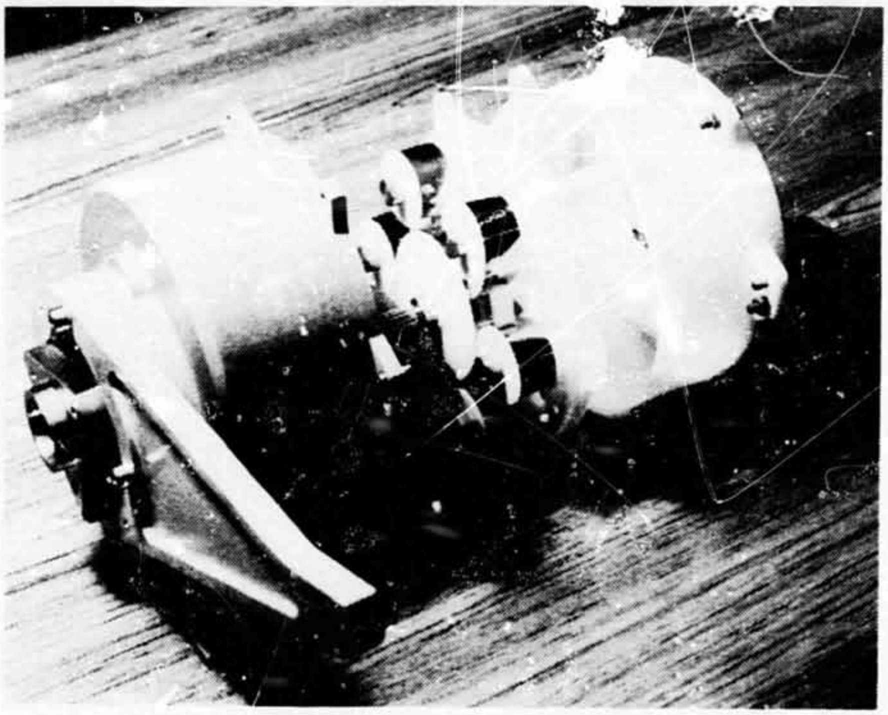

Figure 3. ADM Spring Motor Assembly

Damping

To control the mechanism with the increase in available motor torque, the damping capacity of the ADM was increased accordingly. The HAM programme indicated that the theoretical predictions of deployment speed using the eddy current damper theory were, because of their complexity, difficult to predict. Using the experience gained from the HAM the ADM damper configuration was redesigned. The next section describes the theory used in the predictions of damper performance, and the changes made between the HAM and the ADM.

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Eddy Current Damping Theory

Equation 1 has been derived from the theory covered in "Permanent Magnets."[2] This equation predicts the damping torque that may be expected from a disc rotating within a magnetic field.

$$Td = \frac{2 B^2 r^2 abt\, \omega_d N}{5\rho} \tag{1}$$

A magnet pole is assumed to generate a rectangular magnetic field. See Figure 4.

Where,

B	=	Magnetic field strength	(Tesla)
a	=	Field or magnetic width	(m)
b	=	Field or magnetic length	(m)
r	=	Mean radius of magnets around disc of rotation	(m)
N	=	Total number of magnets around disc housing	(m)
t	=	Thickness of the conducting disc	(m)
ρ	=	Resistivity of conducting disc	(Ωm)
ω_d	=	Angular velocity of conducting disc	(rad/sec)
Td	=	Damping torque generated by the disc	(Nm)

The magnetic field strength in the gap between the magnets is estimated from equation 2.

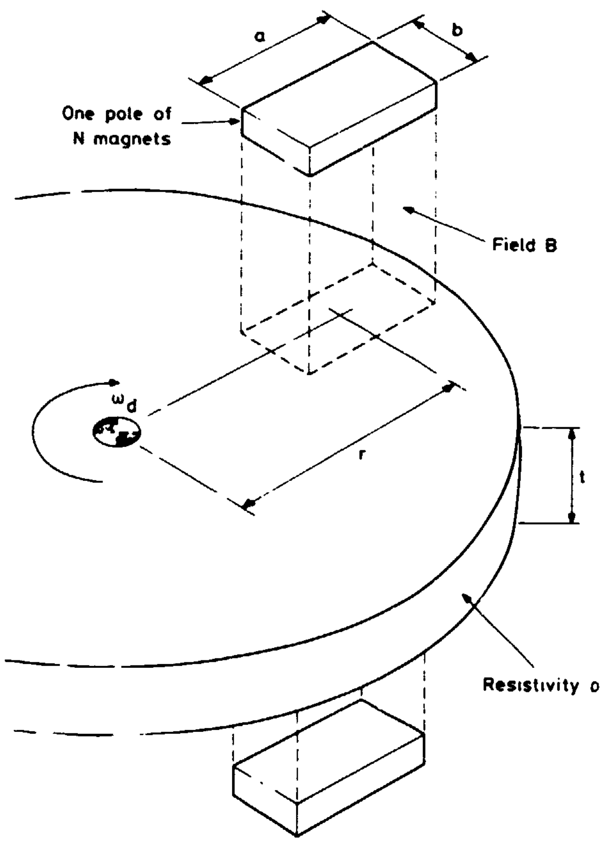

Figure 4. Damper Theory Figure

$$B = \frac{Br}{K_1 \left(\frac{Ag}{Am}\right) + K_2 \left(\frac{Lg}{Lm}\right)} \qquad (2)$$

Where,

B	=	Flux density in gap	(Tesla)
Br	=	Remanance of magnet material	(Tesla)
K_1	=	Leakage factor	
K_2	=	Magneto motive force factor	
Ag	=	Cross-sectional area of gap	(m^2)
Am	=	Cross-sectional area of magnet	(m^2)
Lg	=	Length of gap	(m)
Lm	=	Effective length of the magnet	(m)

In equation 1, where ωd and Td represent the speed and torque in the damper, the output speed and torque may be represented by the expressions,

$$\omega d = \omega . G \quad \text{and} \quad Td = \frac{T}{G}$$

where G = gear ratio between the damper disc and output shaft.

By substitution into equation 1 and by rearranging equation 3 is obtained.

$$\omega = \frac{\rho T}{G^2 . B^2 . a. b. t. Nr^2} \cdot \frac{5}{2} \qquad (3)$$

In calculating the flux density in the gap, (equation 2), two factors need to be quantified, K_1 and K_2.

Factor K_1, the leakage factor, is calculated by relating the useful flux existing between two opposing poles through the disc and that lost between poles parallel to the plane of rotation of the disc. The values of K_1 were quantified during the design stage and are unique to the design of the HAM and ADM.

Factor K_2, the magneto motive force factor, determines the ability of the circuit to produce a current. This factor, however, is difficult to quantify accurately and has been taken in both cases to be 1.4.

Summary of Damper Calculation for the HAM

The following values have been used in the determination of the flux density from equation 2.

Br	=	0.9 Tesla
K_1	=	3.32
K_2	=	1.4
Ag	=	25×10^{-6} m^2
Am	=	25×10^{-6} m^2
Lg	=	4×10^{-3} m
Lm	=	20×10^{-3} m

These result in a calculated flux density of 0.250 Tesla.

During deployment, once a constant speed is obtained, the system is in equilibrium and the damping torque is equal to the excess spring torque. The HAM torsion spring sizing was based on the average available torque.

T	=	driving torque to be damped		
T_s	=	average supply torque	=	0.9015 Nm
T_r	=	resistive torque	=	0.50 Nm
T_d	=	bearing frictional torque	=	0.0005 Nm

$$\therefore \quad T = T_s - T_r - T_d$$

Driving torque to be damped = 0.40 Nm

The radial distribution of the magnets led to two magnetic circuits with a mean effective radii of r_1 and r_2

r_1 = 0.002 m
r_2 = 0.038 m

Also,

ρ = $5.7 \times 10^{-8}\ \Omega m$
G = 225
t = 0.002 m
a = 0.005 m
b = 0.005 m
N = 12

Substitution into equation 3 for both circuits and summation of the results reveals the expected deployment speed.

ω = 0.0207 rad/s

or, $1.19\ \text{deg.sec}^{-1}$.

This indicates that the mechanism should take 75 seconds to deploy 90°. From testing it was discovered that the mechanism took 68 seconds to deploy.

In comparison the following developments were carried out for the ADM and the results summarized in the following section.

Summary of the ADM Developments

The magnet arrangement was changed from the radial arrangement as shown in Figure 2 to a circumferential layout in Figure 5. This resulted in more magnetic pole pairs at an increased mean effective radius. Additional benefits of this change are that the reduction in the number of magnets has reduced the mass and that the leakage factor has been reduced, resulting in an increase in the flux in the gap and consequently the overall efficiency of the damper.

The gear ratio between the damper and output shaft has been increased from 225 to 1378:1. This increase was achieved by incorporating an additional two pass gearbox between the spring motor drive shaft and the damper shaft with a ratio of 56.25:1. Therefore, the damper may be expected to produce the results that follow.

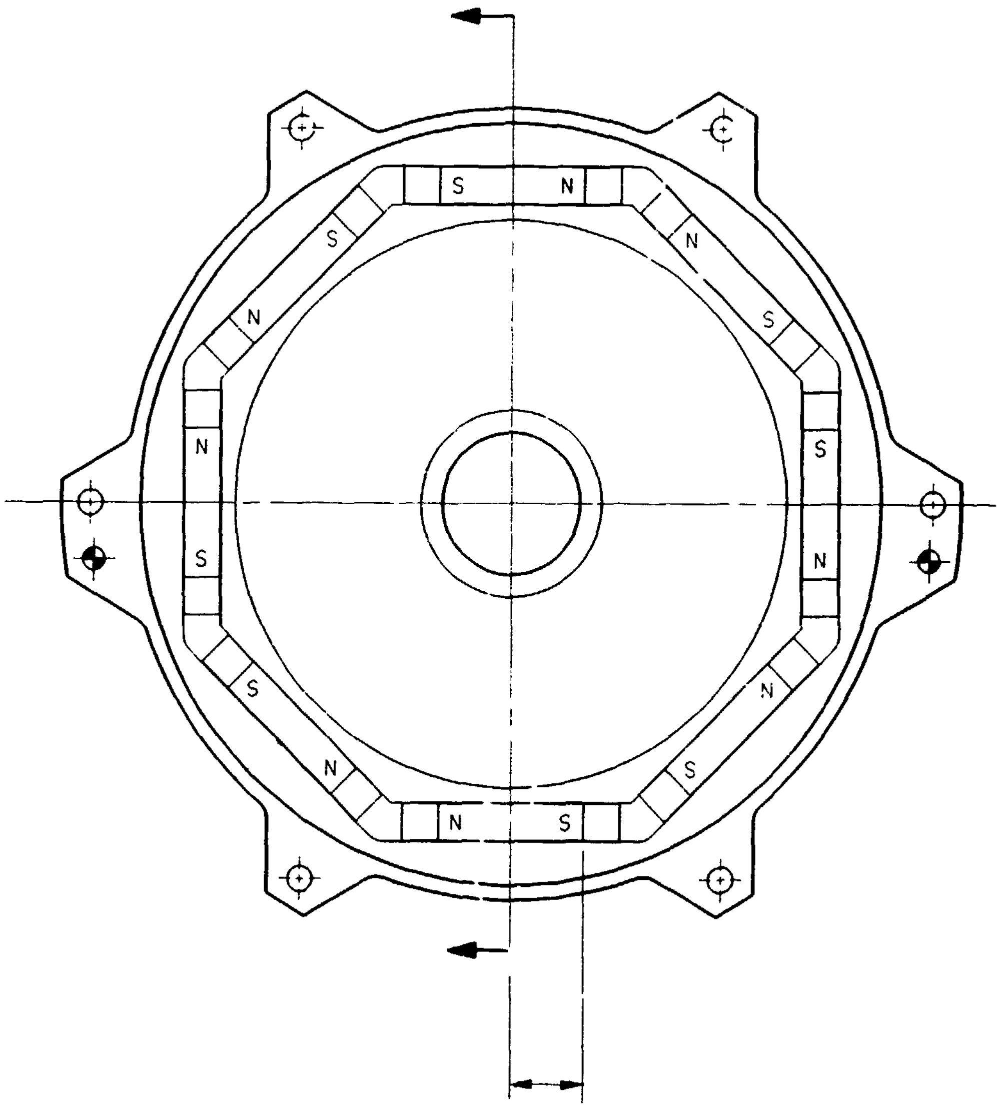

Figure 5. ADM Magnet Configuration

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Summary of ADM Damper Calculations

Flux in gap B

For application in equation 2.

$$
\begin{aligned}
Br &= 0.85T \\
K_1 &= 1.24 \\
K_2 &= 1.4 \\
Ag &= 25 \times 10^{-6}\ m^2 \\
Am &= 25 \times 10^{-6}\ m^2 \\
Lg &= 4 \times 10^{-3}\ m \\
Lm &= 20 \times 10^{-3}\ m
\end{aligned}
$$

$\therefore \quad B = 0.5592$ Tesla.

This may now be used to predict the deployment speed ω.

Driving torque T.

T_s	=	supply torque	=	14.85 Nm
T_r	=	resistive torque	=	0.50 Nm
T_f	=	frictional torque	=	0.011356 Nm

$\therefore \quad T = 14.33$ Nm

Also,

$$
\begin{aligned}
\rho &= 5.7 \times 10^{-8}\ \Omega m \\
G &= 1378 \\
B &= 0.5592\ \text{Tesla} \\
r &= 0.045\ m \\
t &= 0.002\ m \\
a &= 0.005\ m \\
b &= 0.005\ m \\
N &= 8
\end{aligned}
$$

Resulting on substitution into equation 3 a deployment speed.

$\omega = 0.0021$ rad/sec

or 0.12 deg/sec

This indicates a deployment time for a rotation of 90° of 750 seconds.

At present, test results are not available for the mechanism deploying a resistive torque of 0.5 Nm, so the conclusions made are based on the theoretical calculations and the experience gained in eddy current damping.

Conclusions of the ADM Damper Development

The damping capacity of the ADM has been significantly improved by increasing the overall damper gear ratio from 225 to 1378 and by increasing the mean radius while at the same time reducing the number of magnet pairs. The rate of deployment is down by a factor of four and the damping capability has been increased by about 14 times.

The most gains have been achieved theoretically in this area of development and it is expected that these gains will be supported experimentally.

MATERIAL SELECTION

During the design of the ADM, extensive use of lightweight materials for low-stressed components and housings has been incorporated. Delrin, an Acetal copolymer, was used for housings and the damper gears where considerable mass savings were achieved.

Mass

A mass estimate for the ADM includes the additional in-line bearing assembly and results in an overall mass of 2.3 kg. With this slight increase in mass, performance is improved substantially over the HAM.

Conclusion

The ADM is a versatile mechanism capable of deploying large deployable structures on satellites accurately and in a controlled manner such that the disturbance impulse to the satellite is kept to a minimum. A significant increase in drive torque and damping capacity was achieved in the development of the ADM without a significant increase in mass compared to the HAM.

APPLICATION OF THE ADM

The ADM has been used in the OLYMPUS antenna deployment subsystem (ADS). This subsystem supports during launch and deploys after launch the 1.2 m reflectors and antenna pointing mechanisms (APM) that are to be used on the east and west sidewalls of the spacecraft.

The ADS is comprised of three main equipments: the ADM, the deployable arm (ARM), and the pyrotechnic release assembly (PRA) as shown in Figure 6.[3] The PRA is used to hold the ADS to the satellite sidewall during launch, and via the pyrotechnics release the ADS after launch. The ARM forms the backing structure onto which the APM and reflector are mounted. The ADS is required during operation to form a thermally stable platform from which the APM/ and reflector may operate without degradation for the satellite lifetime of 10 years.

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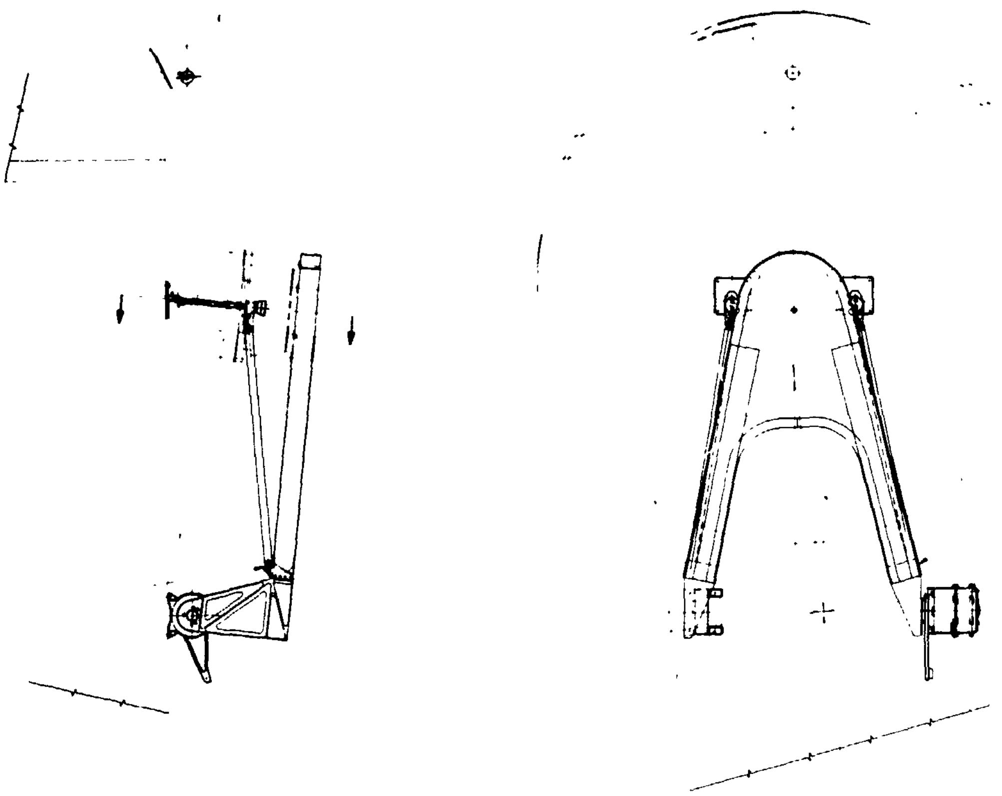

Figure 6. ADS Assembly

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