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# DEPLOYMENT AND RELEASE MECHANISMS ON THE SWEDISH SATELLITE, VIKING

Stefan Eriksson, Saab-Space AB, Linköping, Sweden

## 1. ABSTRACT

Two mechanism types are presented, a rigid boom system and a "hold and release" mechanism for spherical sensors. Both mechanisms have been designed, developed and tested by Saab-Space AB, Linköping, Sweden for the VIKING project under a contract from the Swedish Space Corporation.

## 2. INTRODUCTION (Ref. Fig. 1)

VIKING is a Swedish scientific satellite to be launched together with the SPOT remote sensing satellite by the Ariane Launcher, presently scheduled for late 1984. Its aim is to study a number of physical parameters in the auroral zones of the magnetosphere. The payload consists of five main experiments. Participating scientist groups come from Sweden, Denmark, Norway, Canada, United States of America, West Germany, England and France. Two of the experiments require the spacecraft magnetic influence to be minimized at their sensors. These two sensors are a three axes magnetometer and a loop antenna, located at the tips of the two radial stiff booms.

Two other experiments use six spherical probes (Ø = 10 cm), including inner/outer tips, for electric field measurements. These probes are mounted on four wire booms which can each be deployed to a length of 40 m in the satellite's spin plane, and on two tubular-element booms which can be deployed 4 m each, parallel to the spin axis. Until deployment is called for, these probes are to be stowed by Hold and Release Mechanisms (HRM).

Below the requirements of the radial stiff booms and the HRMs are summarized. Their design, performances and test programs are described in detail.

## 3. STIFF RADIAL BOOMS

### 3.1 Summary of requirements

The design requirements can be summarized as follows:

Launch Mode (stowed)

| | | |
|---|---|---|
| Sine vibration, all three axes | 5 - 100 Hz | 8.0 g max |
| Random vibration, all three axes | 20 - 2000 Hz | 0.3 $g^2$/Hz max |



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Deployment Mode

| | |
|---|---|
| Spin rate | 8.5 to 14.5 rpm |
| Release | pyrotechnic device |
| Deployment shock (on boom tip) | max 50 g |
| Temperature | $-30^{o}$ to $+80^{o}C$ |

Deployed Mode

| | |
|---|---|
| Boom lateral deflection | within a cone angle of $0.1^{o}$ |
| Boom torsional deflection | $\pm 0.2^{o}$ torsional angle |

### 3.2 Detailed description (Ref. Fig. 2 and Fig. 3)

Boom Design

The rigid boom system consists of two identical booms symmetrically mounted. The deployable boom element is an aluminium tube O.D. 53 mm, thickness 1.5 mm and a length of 1385 mm. The limited space available in the launcher adapter and the fact that the boom has to be stowed alongside the payload deck, have set the boom length. Aluminium is also used for the tip sensor mounting flange and the hinge part, both welded to the tube. The other hinge part is designed as a separate unit, fastened to a tubular levelling beam which is mounted on the payload deck. A titanium shaft in two flanged bushings forms the hinge. The bushings have a reinforced P.T.F.E. liner material as lubricant. The hinge also includes the locking mechanism: two beryllium copper leaf springs latching independently on two conical titanium locking pins. These pins are designed with a slope, giving a smooth latching. Between the two spring blades a titanium bolt acts as a stop, taking the collision load at locking. The boom is stowed alongside the payload deck on two supports having a delrin sole. The support nearest to the hinge is elevated with reference to the outer support. The boom is furthermore stowed by means of a "hold down" wire tightening the boom on to the outer support. The boom is thus preloaded to protect the bushings in the hinge from heavy vibration. The preloading also aids in lifting the boom off the supports.

Deployment

Deployment of the booms is initiated by ordinance cutting of the hold down wires. The cuttings are effectuated by electro explosive guillotines through which the hold down wires pass. Nominal spin rate for boom deployment is 11.5 rpm but the boom system has been tested in the required spin rate range from 8.5 rpm to 14.5 rpm. The centrifugal force generated by the satellite spin provides the booms with deployment energy. At nominal spin rate, deployment takes 2.0-2.2 seconds. When the boom has locked into the deployed position the titanium stop bolt takes the load and the boom is deflected. At the return bending the two leaf springs transfer the load via the two locking pins. Since no damping device is used and the deployment only uses a discrete amount of energy the boom will oscillate until the whole energy is dissipated. The oscillation has a frequency of ~7 Hz. During this oscillation the two leaf springs work their way down further on to the locking pins as the locking pin angle acting against the leaf spring is set below the friction angle for the mating surfaces.

## Wiring

The power and signal cables are routed inside the boom and tied down to the tube with brass wires at 15 cm spacings. Through the hinge a flat cable is fed with a free loop length of 14 cm in order to lower the wiring torque. The two boom cables contain:

| | | |
|---|---|---|
| magnetometer | 5 coaxial cables | RG 178 |
| | 1 twisted pair | 0.25 $mm^2$ |
| loop antenna boom | 1 coaxial cable | RG 178 |
| | 5 single cables | 0.25 $mm^2$ |

## Mathematical Model

A mathematical model for boom deployment was developed. It was used for design purposes to estimate the deployment loads. The model is simplified to two identical booms deployed symmetrically. Counter torques from friction, cabling and latching were considered. The available energy for boom deployment is the energy dissipated when the spinning satellite changes its moment of inertia during boom deployment. This energy is

$$T_{diss} = I_p \times \left(1 - \frac{I_p}{I_a}\right) \times \frac{\omega_p^2}{2} \qquad (1)$$

where $T_{diss}$ is dissipated energy

$I_p$ moment of inertia prior to boom deployment

$I_a$ moment of inertia after boom deployment

$\omega_p$ satellite spin prior to boom deployment

### 3.3 Boom Tests

Four types of tests were carried out to evaluate the boom system performance. Firstly, deployment tests have been conducted on a spinning table. Secondly, the hinge has been tested separately in a vacuum chamber. A third test has been conducted to determine the boom wiring performance with regard to torque loss changes due to storage and extreme temperatures. Finally, vibration tests have been performed.

#### Deployment tests

Deployment tests were performed with the booms mounted on a spinning table, with a mounting plate having the same moment of inertia as the satellite itself, and with the spin table drive decoupled just before deployment. Thus the boom deployment was adequately simulated and the interaction of the bodies at locking was not disturbed. To minimize spin rate decrease caused by friction, the spin table bearings were axially off-loaded by means of two support wires from the ceiling. The two booms were also supported at CG location to unload their weight in the g-field. All data acquisition equipment was placed on the rotating table to avoid slip ring arrangements. The main parameters recorded were: the bending moment at the boom root, the tip mounted experiment acceleration, the deployment angle, the latch spring movement and the spin rate. The tests were performed at three spin rates: 11.5 rpm which is the nominal spin rate, 8.5 rpm and 14.5 rpm. After each functional test an alignment check was performed.

#### Separate hinge test

During the development phase a detached hinge was tested in a vacuum chamber. Deployment was effectuated by means of a pendulum stroke outside the chamber. The counter torque from friction and latching was measured in the axis from the pendulum to the hinge inside the chamber. As the hinge was cooled and heated, the effects of different temperatures and temperature gradients over the bushings were studied.

#### Wiring test

A test rig for the study of the wiring through the hinge was manufactured. It consisted of an electric motor for simulated deployment with means of measuring deployment angle and counter torque with hinge equivalent wiring. The counter torque changes due to storage of the boom in stowed position and due to the extreme wiring temperatures anticipated were also studied.

Vibration test

For mounting the booms on the vibrator, a fixture shaped as a segment of the payload deck was used.This fixture was also used for mounting on the spin table. Thus the boom could be moved back to the spin table in stowed position for a functional test.

### 3.4 Test Results

During the functional tests the boom bending moment at the boom root was recorded. The results are presented in fig 4. The maximum recorded bending moment at 14.5 rpm deployment spin rate was 378 Nm. The designed ultimate load was set at 1100 Nm, which resulted in a safety factor of 2.9. The bending moment, 378 Nm, corresponds to an experimental acceleration of 24 g. At nominal spin rate deployment the acceleration was 18 g. A typical interaction of the booms after locking is shown in fig 5.

An alignment check was also performed after each test and the deflections were as follows:

| | |
|---|---|
| Boom lateral deflection | within a cone angle of $0.05^{\circ}$ |
| Boom torsional deflection | max $0.05^{\circ}$ |

From the hinge vacuum test and the wiring rig test an energy budget is deduced as follows. The values presented are the maximum obtained. (Ref fig 6)

| Action | Energy consumed (J) Pre-locking phase | Locking phase | Temp./Temp.gradient ($^{\circ}C$) |
|---|---|---|---|
| - Bearing/shaft friction | 0.60 | 0.05 | -30/10 |
| - Harness work | 0.05 | | -30 |
| - Latch work | | 0.40 | N/A |
| - Latch-locking pins friction | | 0.40 | N/A |
| Total Energy Consumed | | 1.5 J | |

The figures represent one boom; thus a total energy of 3.0 J is needed for the boom system. The equation (1) given in 3.2 yields the lowest spin rate limit for a feasible latching, which is 5.9 rpm. The safety factor on energy to the predicted lowest deployment spin rate, 8.5 rpm, is 2.1.

During vibration tests the experiment unit experienced an overall RMS level of 6 g. The magnetometer boom configuration resonance frequency at 35 Hz showed a peak value of 46 g at the magnetometer location.

### 3.5 Problems encountered

Since the vibration level at the magnetometer boom resonance frequency during the preliminary tests was quite high, a 1.5 mm rubber isolator was inserted at the mounting flange. This measure reduced the peak value by a factor of 1.7.

After the preliminary tests a larger safety factor on energy loss was requested. Measures were taken to minimize the energy dissipation during deployment and latching. Decreased width of the leaf springs and adjustment of the locking pins slope resulted in a smoother latching.

A change in the payload deck was found necessary after the deployment tests, because the boom vibrated after release and bounced against a HRM. For protection of the HRM a L-profile with a plastic material on the edge was introduced. The boom now bounces against the profile and the vibration is damped.

## 4. HOLD AND RELEASE MECHANISM

### 4.1 Summary of requirements

Stowed Condition

| | | |
|---|---|---|
| Sine vibration, all three axes | 5 - 100 Hz | 10.0 g max |
| Random vibration, all three axes | 20 - 2000 Hz | 18.1 g RMS |
| Resonance frequency | 100 Hz min | 0.3 $g^2$/Hz max |
| No sensor movement | | |

Release Mode

| | |
|---|---|
| Temperature | $-30^{\circ}$ to $+80^{\circ}C$ |
| Swing of sensor at release | within a cone angle of $10^{\circ}$ |
| Switch indication at completed armpair deployment | |

## 4.2 Detailed description (Ref. Fig. 7)

The Hold and Release Mechanism (HRM) is used to hold the sensor in stowed position until the wire boom and axial boom deployment is initiated. Ten seconds before boom deployment starts, sensor release is effectuated. The HRM consists of a mounting bracket and two moveable armpairs. The mounting bracket is designed as a flat beam reaching out over the edge of the payload deck. The axes of the two armpairs rotate in bearings similar to those used in the stiff boom hinges. Armpair deployment is effectuated by means of preloaded torsion springs. A wire is used for tightening the armpairs around the sensor rods. The wire is fastened in the armpairs and passing through an electro explosive guillotine in its housing. The holding force is set when the guillotine housing is pulled towards the bracket.

A strain gauge, which is used to measure the holding force, is glued to a beam on one of the armpairs. Each arm is set with 50 N against the sensor tip. The pads (made of delrin) holding the rods have cylindrical grooves for position of the sensor rods. On the sensor outer rod a cylindric pin ( $\varnothing$ = 1 mm) is used to prevent the sensor from radial and rotational movements. The pin is placed in a track in one of the outer pads. Two micro switches are mounted on top of the bracket for indication of full armpair deployment. The HRM for the two axial tubular element booms is not described in detail but is similar to the unit presented above.

## 4.3 HRM tests

Release test

The release test was conducted in a vacuum chamber (1x $10^{-6}$ torr) with a heating/cooling arrangement. Three release tests, with a $10^{\circ}C$ temperature gradient over the bearings at $-30^{\circ}C$, $+25^{\circ}C$ and $+80^{\circ}C$, were performed on each unit. The effect of temperature was studied by comparing the deployment time for each release test. During the test the HRM was vertically mounted such that the sensor dummy used could be studied with regards to side forces. The parameters recorded in these tests were: release initiation (strain gauge voltage drop), armpair deployment angles (potentiometers), switch indication at fully deployed armpairs and the sensor dummy movement. The sensor dummy movement was recorded by the use of a small pick up coil mounted on the rod tip of the dummy. A larger sending coil induced a voltage variation proportional to the swing of the dummy.

Vibration test

The HRM was tested to the levels specified. During these tests a sensor mass dummy was attached. After vibration test, a release test was performed.

## 4.4 Test Results

No significant changes due to bearing temperatures of $-30^{\circ}C$ to $+80^{\circ}C$, or $10^{\circ}C$ temperature gradients over bearings, were found during the release tests. The deployment times for all tests stayed within the interval 50-60 ms. At release, the sensor swing stayed within a cone angle of $5^{\circ}$ for four units. Larger swings were recorded for the last unit due to an unintentional torsional preloading of the wire boom cable, caused by the test set up.

## 4.5 Problems encountered

During the preliminary test two major problems were dealt with. One, the resonance frequency of the mounting bracket was 60 Hz for the first HRM design. By adding a box shaped stiffener the resonance frequency was increased to 130 Hz. Two, a number of different designs of the pads holding the sensor from rotational and radial movements were tested in vibration. The design chosen is such that the wear in the delrin pads during vibration is low. This will minimize the sensor swing after release.

# 5. CONCLUSION

The two mechanisms described above have successfully passed their qualification and flight acceptance tests and meet the requirements. They are presently being integrated with the VIKING satellite.

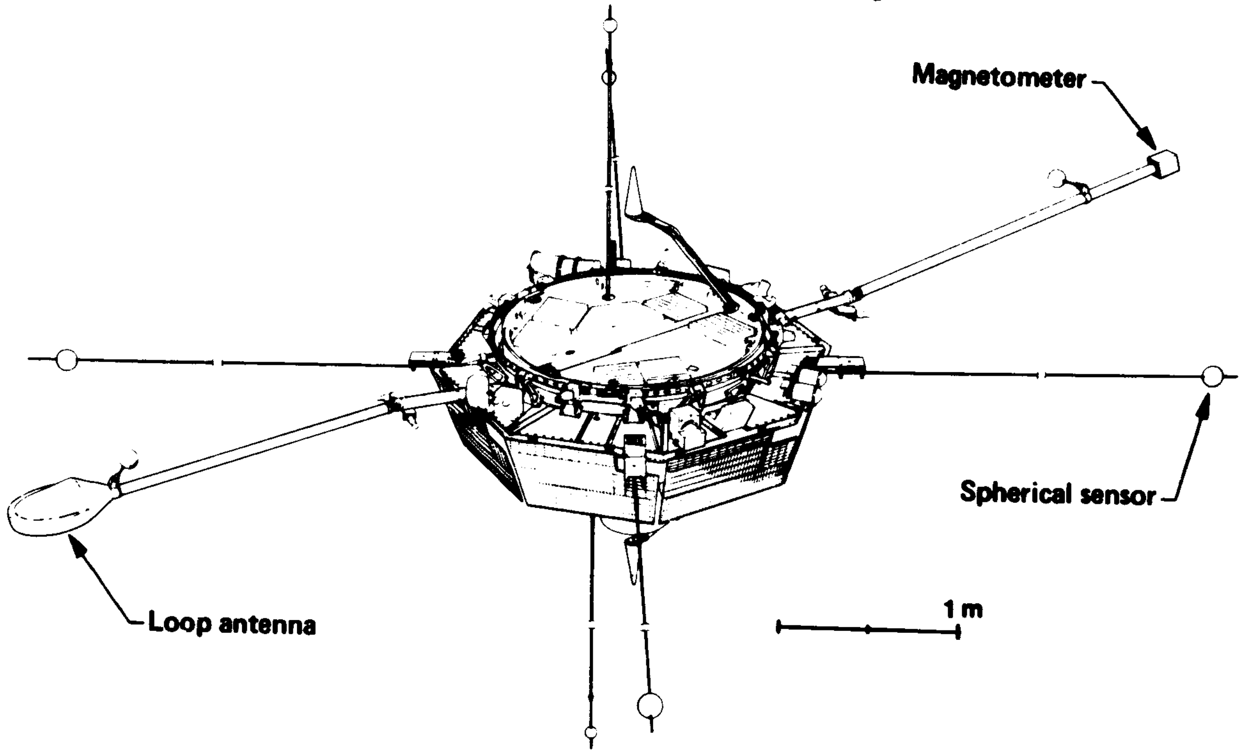


Fig. 1 Viking flight configuration

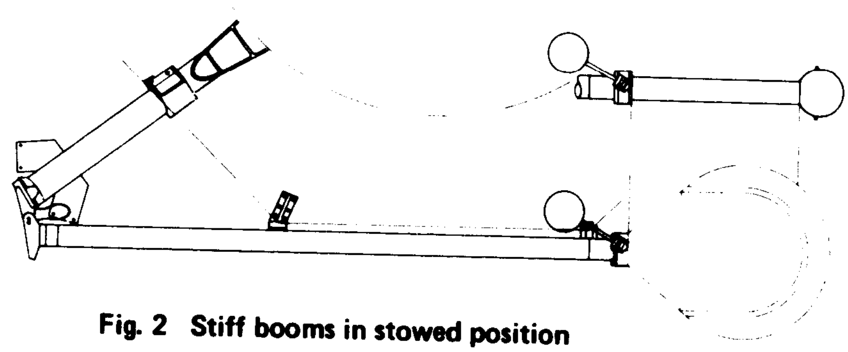

Fig. 2 Stiff booms in stowed position

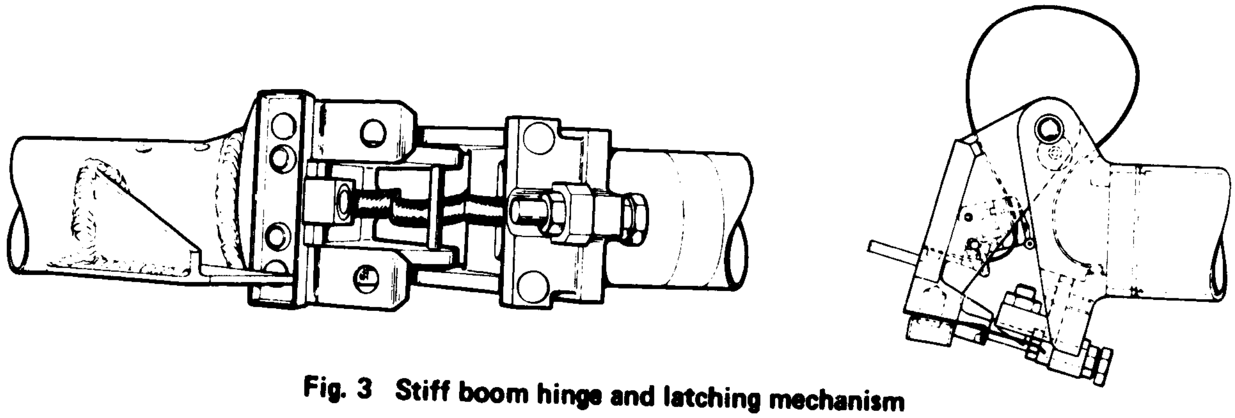

Fig. 3 Stiff boom hinge and latching mechanism

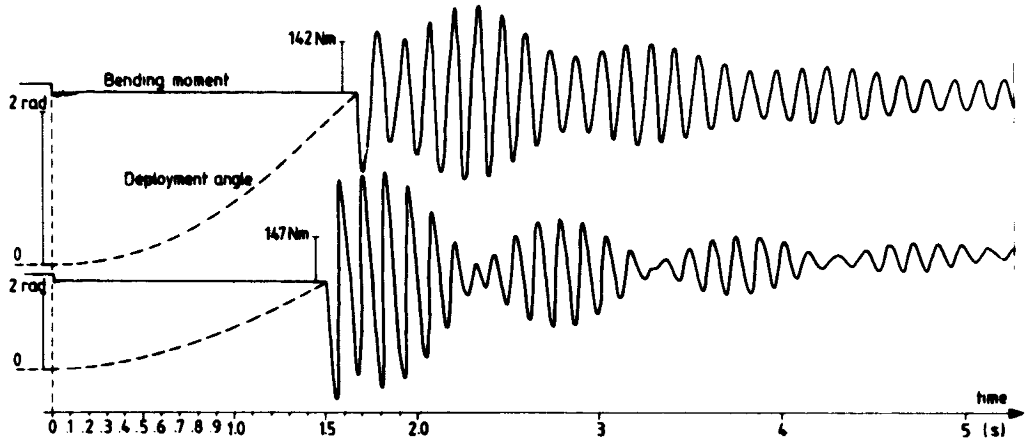


Fig. 5 Boom interaction after locking

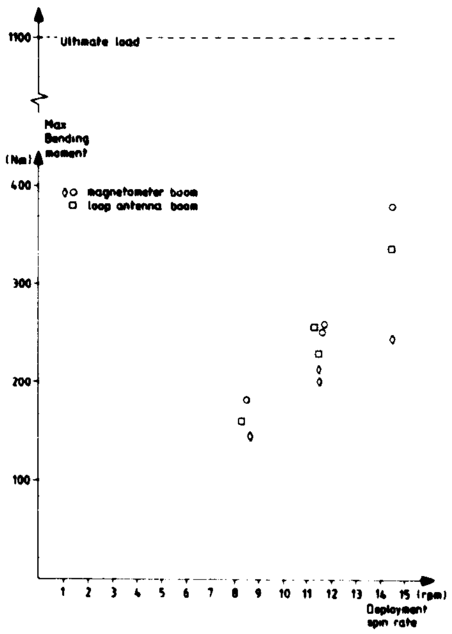


Fig. 4 Max boom bending moment at locking

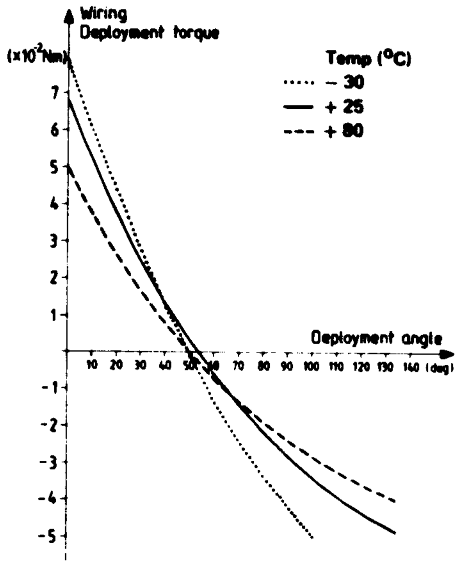


Fig. 6 Boom wiring torque (Temp. 25°C)

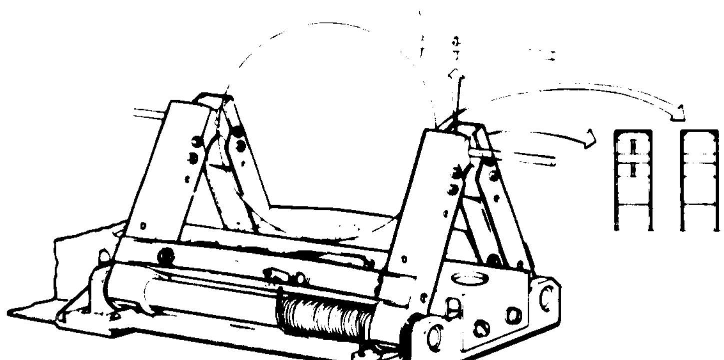

Fig. 7 The HRM with the pads holding the sensor in place