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Design of Wire Boom System for a Satellite

Konstruktion av trådbomsystem för satellit

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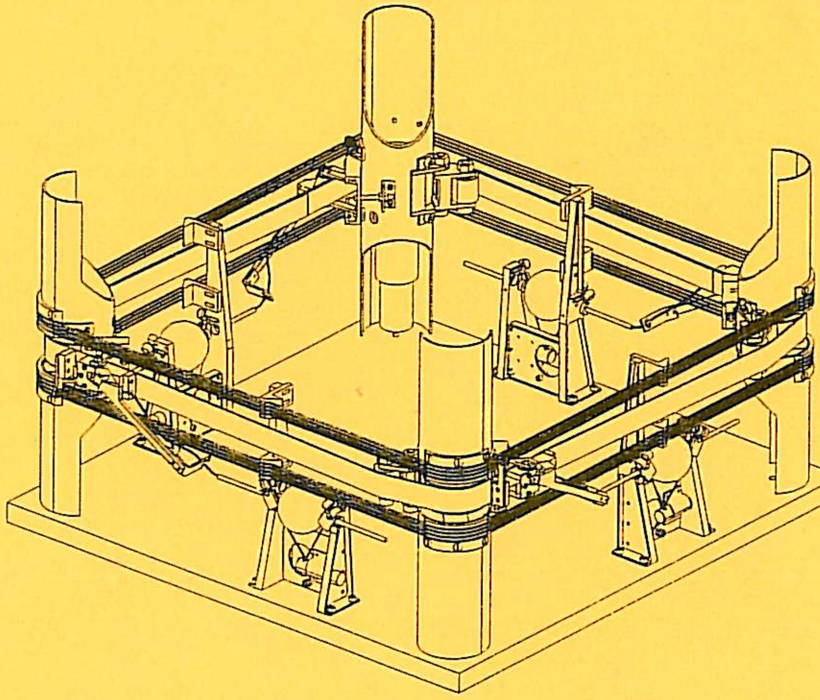

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Design of Wire Boom System for a Satellite

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Abstract

The report deals with the design of, selection of materials for and the tests of a wire boom mechanism for the Swedish micro satellite Astrid II. The mechanism has the function to deploy four wire booms, each three meters long. These wire booms are equipped with probes for electric field measurements. In deployed state the wire booms are held straight by the centrifugal force of the rotating satellite, acting on the probes at the end of the wires.

The wire boom system is based on a completely new concept where the boom wires are wound around the satellite. The main component of the deployment mechanism is a cog belt that runs round the satellite body, parallel to the wound wires. When the booms are about to be deployed the belt is driven by an electric stepper motor and its motion is governing the wire deployment. To verify this new and unproved design solution several functional tests, some of which still remain to be done, are required. Some preparatory tests have also been carried out before the design work started.

Konstruktion av trådbomsystem för satellit

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Sammanfattning

Rapporten behandlar konstruktion, materialval och provning av en trådbomsmekanism till den svenska mikrosatelliten Astrid II. Denna mekanism har till uppgift att mata ut fyra stycken tre meter långa trådbommar med prober för mätning av elektriskt fält. Satelliten roterar och därmed drar centrifugalkraften ut kablarna under utmatningen och håller de sedan sträckta i utfällt läge.

Trådbomsmekanismen bygger på ett helt nytt koncept där bomkablarna under uppskjutningen är upplindade runt satelliten. Huvudkomponenten i utfällningsmekanismen är en kuggrem som löper runt satelliten parallellt med de upplindade kablarna. Denna kuggrem drivs av en stegmotor och dess rörelse styr utmatningen av bomkablarna. Den nya oprövade tekniken har krävt många funktionsprover för att verifiera konstruktionen, några återstår fortfarande att göra. Ett antal förberedande tester har även gjorts före konstruktionsarbetet.

Preface

This diploma thesis work (20 credit units) has been carried out at the Alfvén Laboratory of the Royal Institute of Technology under supervision of the Department of Machine Design/Engineering Design, Royal Institute of Technology.

I would like to thank my tutors Prof. Jan-Gunnar Persson at Machine Design/Engineering Design and chief engineer Lars Bylander at the Alfvén Laboratory for the support they have given me. I would also like to acknowledge everyone at the Alfvén Laboratory that has helped me during my work, especially Sverker Christenson, Ola Carlström and Göran Olsson in the Space Group.

Stockholm, February 1996

Hans Hellman

1 CONTENTS

2 INTRODUCTION AND BACKGROUND

2.1 The Astrid II Satellite

The Astrid II project was initiated at the Alfvén Laboratory in 1993. At that time the micro satellite concept was a quite new idea and the research team at the Alfvén Laboratory laid down the general outlines for a new experiment for monitoring the electric and magnetic field of the aurora. The project was named EMMA, an abbreviation of Electric and Magnetic field Monitoring of the Aurora. The Astrid II platform shall be ready for launch the first of March 1997 and is also carrying two more experiments, a particle detector and a photometer. The Astrid II satellite will be launched as a piggyback together with a Russian satellite. This means it will be launched with the same rocket as a large satellite, implying a relative low price.

The EMMA-experiment, that the wire booms are a part of, will perform accurate and sensitive measurements of the electric field and the magnetic field in the upper ionosphere. Of particular interest is the electrodynamics of the aurora and its fine structure. The four wire booms are performing the electric field measurements and they are located in the spin plane of the satellite. A great advantage with the integration of electric and magnetic field measurements in the same experiment is that the measurements can be perfectly time correlated.

The Astrid II satellite platform is a so called micro satellite and the main body of the satellite has the approximate dimensions 0.45x0.45x0.3 m and approximate mass 30 kg. The satellite is spin-stabilised and the spin axis is sun-pointing, which means that the solar panels always are perpendicular to the incident sunlight. The orbit is preferably almost polar (80° inclination) and the altitude is about 1000 km, this orbit crosses the most interesting part of the Polar cap for this experiment. The satellite is communicating, by using S-band telemetry, with the ground station at Swedish Space Corporation, Stockholm. The ground station is communicating with the research group at KTH through a Internet connection.

2.2 The Electric Field Wire Booms

The method of electric field measurement in the space plasma that the EMMA experiment is designed for is well established and has been applied for several Swedish research satellites. For the Astrid II micro satellite the limited available volume and weight however require a completely new mechanical design solution of the wire boom system.

A wire boom is an electric cable with some kind of measuring instrument in its end, in this case a spherical probe for electrical field measurements. The measurement of the potential in the plasma is done by measuring the potential of spherical probes located in the plasma. The reference potential in these measurements is the potential of the satellites surface. The spherical probe has typically a diameter between 30-80 mm and it is mounted on a coaxial cable. The coaxial cable has two functions, as a signal wire and as a boom. The length of the cable varies from one case to another between 3 and 50 meters for different satellites and experiments. On a spinning satellite the coaxial cable will be straighten by the centrifugal force on the probe and this will keep the booms straight in radial direction and in an almost fixed position. During the launch of the satellite the wire booms have to be packed within the satellite and when the satellite is in orbit and is rotating, the wire booms have to be deployed in some way. This is done by a boom mechanism.

On previous Swedish research satellites a boom mechanism designed and manufactured by Weitzmann Consulting, Inc. has been used. Weitzmann Consulting, Inc. is a company in San Francisco, California. In this mechanism the deployment of the booms is controlled by one deployment unit per wire boom. Each wire is wound on a separate spool and the deployment is controlled by an electric motor. This mechanism is too heavy and oversized for a micro satellite like Astrid II. It is designed to be able to carry a 50 meters long cable and each deployment unit weighs about 2.3 kg. For this application we need a cable length of about three or four meters and the available mass is slightly over one kg for the whole system with four booms. So the mass has to be reduced by nearly a factor of eight as compared to the Weitzmann mechanism.

This was the problem that the Astrid II project group was faced to during spring -94. A competition for the students at the Royal Institute of Technology was arranged to generate ideas on how to solve this problem. I made a contribution in this competition and the jury was satisfied with my solution of the problem so I got the opportunity to work on with my idea. My work with this wire mechanism started in summer 94, when I developed the ideas further and constructed a demonstration model of the mechanism. This work is documented in the paper "Konstruktion av satellitbommar till Astrid II". The demonstration model showed that the concept was realizable and the Alfvén laboratory decided to go for this concept. At this time my degree project started and it was arranged as a cooperation between the department of Machine Design/Engineering Design, KTH and the Alfvén laboratory. My tutors have been Prof. Jan-Gunnar Persson at Machine Design/Engineering Design and Lars Bylander at the Alfvén laboratory.

During the winter 94-95 some preparatory tests were done. The tests were simple simulations of a cable running through two eyes in the same way as the boom cable runs through two eyes during the deployment. As this vital part of the deployment procedure seemed to work a detailed design work was initiated. This was in the beginning of the summer 95. The design has been done in the 3D solid modeler I-DEAS from SDRC and the design work went on until October 95 when a final prototype was completed. This final prototype has gone through basic mechanical tests of the mechanism and deployment tests in room temperature and in extreme cold.

3 SUMMARY OF DOCUMENTATION OF THE BOOM DEPLOYMENT MECHANISM DESIGN

This thesis report covers:

- The first conceptional ideas and alternatives. Discussion of advantages and drawbacks of the different solutions.
- Preparatory screening tests for a rough verification of the mechanism.
- Calculations, as stress calculations and other mechanical calculations.
- Discussion and description of details in the design.
- Discussion and documentation of materials selection.
- Mechanism design data.
- Verifying tests.
- A materials selection list.

In addition to this thesis report, the following documentation has been prepared:

- All components in the mechanism are documented in design drawings. These drawings are collected in the document "The Astrid II Wire Boom Mechanism, Drawings".
- Details from the early design work are collected in the paper "Konstruktion av satellitbommar till Astrid II.

4 Requirement Specification for the New Electric Field Wire Boom Mechanism

This is the list of requirements, as specified by the Alfvén Laboratory:

- The satellite is intended to carry four wire booms.
- The wires have to be deployed continuously to avoid generating instabilities of the satellite.
- The length of a wire boom should be at least three meters.
- The maximum mass of the complete wire boom mechanism must not exceed one kg (approx.).
- The available volume in the satellite is extremely limited.
- The development and manufacturing costs must be low, as the entire satellite project is a low budget project.
- Design solutions should be as simple and uncomplicated as possible, to give maximum reliability.

5 FURTHER STUDIES AND ADDITIONAL DESIGN WORK TO BE CARRIED OUT

Here is a list of further studies and additional design work that have to be carried out as the design work with the wire boom mechanism has not been completely finished in this degree project:

- No analysis of the oscillation in the wire booms has been done. Such oscillation is initiated when the booms are deployed and by spin/despin maneuvers of the satellite. This oscillation has to be damped and this damping is done by the internal damping of the cables and by the nutation dampers that are placed in the platform. It has to be determined if this damping is sufficient to counteract the oscillation. The analysis has to be done to make sure that nothing unexpected happens during and after the deployment. A purposed procedure to do this has been enclosed in appendix C.

- The surfaces of the plastic details in the mechanism that are exposed to the space environment have to be coated with a electrically conductive material. This has not yet been investigated, neither the coating material nor the coating process.

- The driving system, motor and gear, has not yet been settled. Some basic mechanical analyses of motor and gear have been done, these are enclosed in appendix B. The drive package on the prototype is a DC-motor with a gear head supplied as one unit from Minimotor SA in Switzerland. Both the motor and the gear head have too high magnetic moment to be used in the satellite. The gear-head may be made magnetically clean enough by changing the steel housing to a housing made of stainless steel. The motor has to bee changed, presumably to a stepper motor. A suitable stepper motor is the P110 from Escap, Switzerland.

- The position of the belt is important to know because it indicates how much of the wire booms that have been deployed. This position sensor can be a rotating potentiometer on one of the pulley wheel axis or a pulse gauge on a pulley wheel or on the belt. This arrangement has not been settled yet.

- The satellite is exposed to vibrations during the launch. The mechanism has to be vibrated in an oscillator with a spectrum that simulates the oscillations during the launch to verify its structural rigidity. This test has not yet been done but is briefly outlined in chapter 8.

- The position of the probes in relation to the satellite body, the attitude of the probes, is very important to know when the E-field data are analysed. This attitude must be specified in terms of nominal radial distance and nominal pointing angle with a deviations. The deviation of the pointing angle depends on the straightness of the wires and the uncertainty of the point of attachment to the satellite body. This determination has not been done yet.

6 The Concept, Alternative Solutions

My idea was to utilize the structure of the satellite platform as much as possible in the boom mechanism. In other words to give structural parts a function in the boom deployment system. I decided to wind up the boom cables on the satellite and use the satellite body as a winding spool. This idea had already been discussed in the Astrid II project group before the design competition was announced but the problem was how to deploy the cable in a controlled way. My solution of that problem was to use a toothed belt that runs around the satellite parallel to the boom cables. Each boom cable runs trough two eyes that are attached to the belt. If the belt is driven in the opposite direction compared to how the cables were wound up the wires will run through the eyes and cable will be released. This mechanism relies on that the centrifugal force on a probe is big enough to pull the cable through the eyes.

This was the idea that I used in the competition. The purposed mechanism design still looks the same but it has been reconsidered against other ideas such as:

- One rotating spool per boom is a quite simple and the most straight forward solution. To redesign the Weitzmann boom mechanism with smaller deployment units should give a conventional mechanism, with the advantage of being flexible. These mechanisms are flexible in that way that they are quite independent of the structural design of the satellite. Even though the boom cable is short, this kind of deployment units will probably still be quite large because the boom cable has to be winded on a spool with quite a big radius not to get damaged. This solution should therefore give a bulky mechanisms with the drawbacks of one electric motor per wire boom.

- -Another proposal from the competition was to release the boom cable in shorter lengths and let the satellite stabilize itself between these releases. This idea has the advantage that it provides a relatively simple mechanism but the major drawback is that the satellite becomes very unstable when the cable is released and that the probes probably are oscillating for a long while after the release. Another drawback with this specific solution is the storage of the cable, the cable is wound on thin pins that are folded when the cables are released. These pins give the cable a narrow radius and when the cable is released these radii probably stay and the cable keeps it's zigzag shape.

- A third proposal from the competition was to use stiff booms that were deployed with a linkage similar to many scissors connected to each other. This design eliminates the risks for oscillation in the booms. The booms are however to thick and would disturb the E-field measurements.

6.1 Advantages and Drawbacks with the New Wire Deployment Concept

The decision to use a mechanism with the wires wound on the structure are based on careful considerations. The following list summarizes the advantages of the new wire boom system:

- The deployment is continuos and very easily controlled
- The satellite platform itself is used as a spool to wind up the boom cables on, in order to reduce weight.
- Only one electric motor is used for the deployment of all four wire booms, this reduces weight.
- The boom cables and the mechanisms are located on the outside of the satellite and therefore intrude very little on the interior available space of the satellite.

The drawbacks with the new system are:

- The deployment of the cables is completely dependent on the centrifugal force. This means that the spin rate of the satellite must be relatively high and the probes have to be quite heavy. Heavy probes and high spin rate are however advantageous for more reasons, to help to straighten the cables and to make the booms more stable.
- The deployment mechanism is strongly integrated with the structural parts of the platform. This makes the deployment mechanism very dependent on the design of the platform and a design revision of the platform could also have an impact on the deployment mechanism.
- The integration with the platform makes the mounting of the boom mechanism to the rest of the satellite a little bit tricky.

7 DESCRIPTION OF THE DESIGN

At the start of my work the mechanism seemed to be very simple but after a while it turned out to get more and more complicated the more details that were studied. This project is not unique in that way however.

The belt transmission with its wheels, shafts, bearings and motor package is a quite conventional thing to design except for the demands on magnetic cleanliness and performance at low temp and vacuum. The storing of the probe and the mechanism that folds out the eye that the cable runs through is a more delicious mechanical problems. The whole design is shown in figure 8-1. My ideas around the different details are briefly mentioned here in chapter eight.

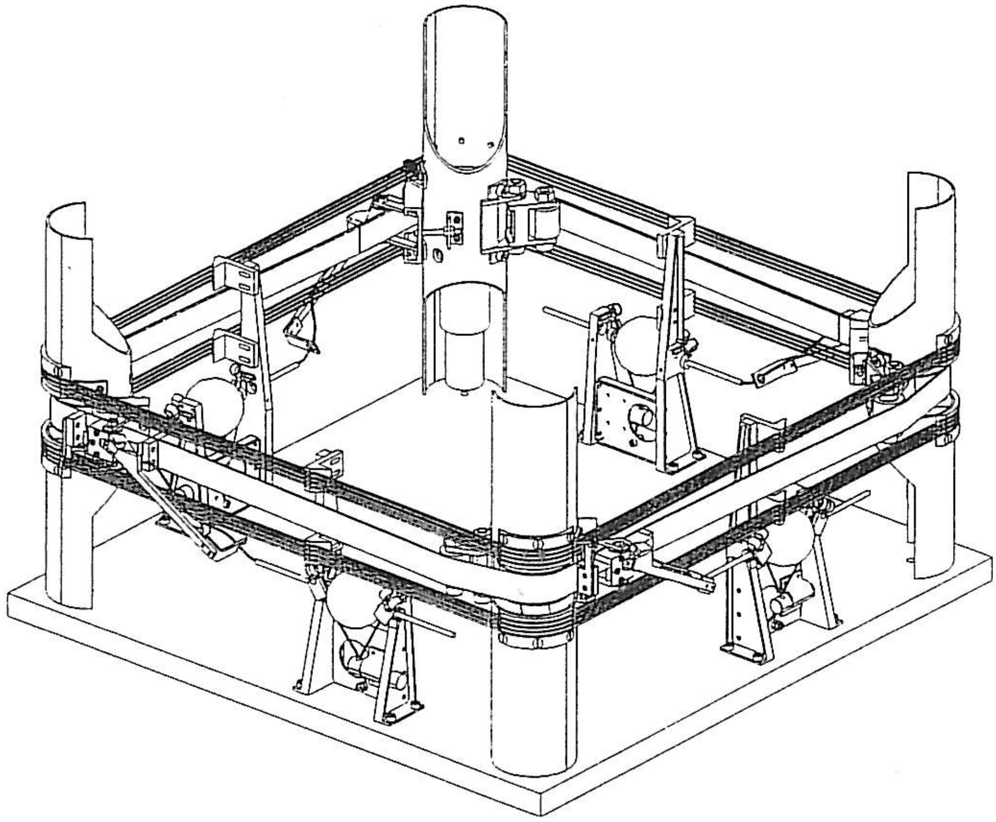

Figure 7-1. The wire boom mechanism assembly (one deck is removed from the satellite structure).

7.1 Storage of the Probes

The probes have to be stored inside the retracted solar panels during the launch. This is done by the probe brackets. The probes are fixed in these brackets by their probe stubs. These stubs are in the first place designed to refine the electric field measurements but they do also provide a simple way to hold the probes during the launch. The probes are lying in plastic seats on the brackets and they are fixed by strained kevlar wires. Between the wires and the stubs there is a plastic plate that distributes the pressure on the stubs. The kevlar wires can be cut off by pyrotechnic guillotines. When this is done the plastic plates automatically lifts up from the stubs by torsion springs that revolve the plates 100° from their initial position. By this rotary motion the kevlar wires are out of the way and they can not disturb the release of the probes.

One more function is included in the probe brackets: The wound up wires have to be stabilized between the corner bars of the satellite body. Otherwise oscillations may be excited during the launch and they will oscillate like guitar strings. Because the wound wires are very close to each other they would rub against each other. This is avoided by the wire holders that keep the wires in place. One of the cantilevers of the probe bracket has been extended and the wire holders are attached to this cantilever.

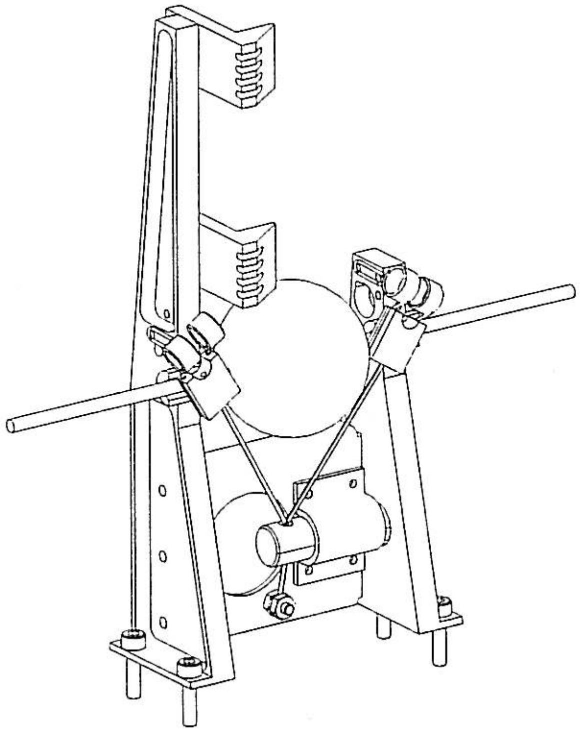

Figure 7.2 The probe bracket with probe.

7.2 Cable Guide Eyes

The eyes for guiding the cable have some different tasks. Firstly, the cable shall be lifted up from the groove in the "wire wrap profiles", see section 7.4 "details for the winding of the wires". This is done by the first eye that is mounted on the belt. This eye sweeps over the wire wrap profile just above the groove where the cable is wound and lifts up the cable. There must be one more eye per wire boom, because these four eyes are all positioned differently over the belt width to be able to pick up the cable from the grooves. The four deployed wire booms have to lie in the same spin plane of the satellite and this can only be done with one more eye per wire boom that guides the cables to a common spin plane. The second eye has to be positioned so that the cable runs in a smooth curve between the two eyes. This can be achieved if this second eye is placed at a distance out from the first eye. The belt is lying on the outer surface of the satellite and as the second eye shall be placed at a distance out from the first eye (and from the belt) it has to be foldable in some way to fit inside the solar panels when the satellite is packed for launch. This is done by the folding mechanism and this mechanism also provides a step by step release of the probes from their storage.

The position of the eyes in unfolded sate was determined by preparatory friction tests, see chapter 10. This geometry has however been modified a bit to make the folding movement possible.

7.3 Folding Mechanism

The folding mechanism is designed to move the outer cable eye from a folded down position to a working state position. The desired motion is simply a move of the outer eye from a position next to the probe bracket to a position outside the belt at a distance of 50-100 mm. This motion is achieved by a linked arm. The mechanism that folds out the second eye has to be driven in some way, this is done by the movement of the belt. In that way the folding movement is fully controlled and is not depending on a spring or any other kind of buffer that can jam if the friction is too big. The motion is controlled by two pins that run in a fixed guide rail. The pins are attached to two arms that bring a shaft to rotate and when the shaft rotates the arm with the eye will rise and finally after revolving of 90° it is locked.

At the same time as the arm rises 90° it rotates 30° in a direction perpendicular to the first rotation. This rotation provides the arm to stop in a direction perpendicular to the outer surface of the satellite even though the initial position was at an angle of 30 degrees to the longitudinal direction of the belt.

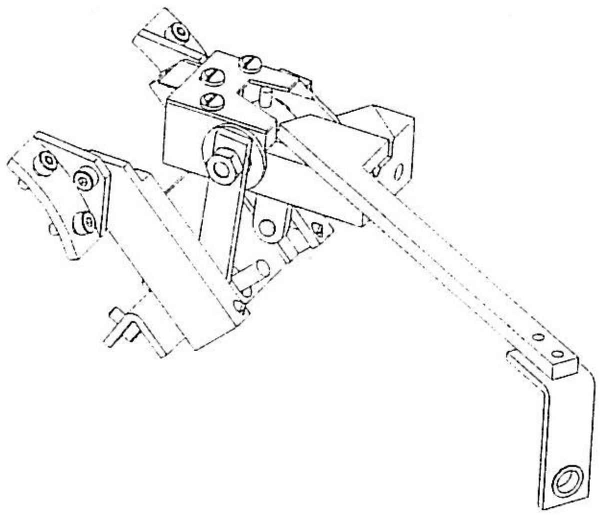

Figure 7.3 The folding mechanism.

7.4 Components for the Winding of the Wires

The cables have to be fixed in some way when they are wound on the satellite, this is done by the "wire wrap profiles" and the "wire holders". The wire wrap profiles are placed on the corner posts of the satellite and they fix the cable both longitudinally and transversally. This is done by narrow grooves that squeeze the cable when it is strained over the corner.

Between the corner posts the wire holders support the wires. The wire holders are shaped to support the wires at the middle of their free lengths and in that way increase their resonance frequency. Without the wire holders the vibration of the cables can cause problems during the launch. In figure 7.2 the wire holders are shown mounted on the top of the probe bracket.

7.5 Belt Transmission

The design of the wire boom system has intruded on the design of structural parts in the satellite, this because of the high degree of integration between the boom mechanism and the structural parts of the platform. The design of the corner posts of the satellite has been ruled by the wire boom system. The boom cables are wound round the posts and the corners have to be smoothly curved to not give the boom cables too narrow radii. Therefore the posts are made of an aluminum tube with a circular profile with outer diameter 50 mm. The pulley wheels are mounted on bearings inside the tubes, concentrically to the tube profile. They are mounted on stationary shafts and the shafts are rigidly supported to the tubes in both ends of the shafts. This design makes the shaft to a reinforcement of the tube. This is good and maybe necessary because the tube is weakened just where the pulley wheel is mounted. Half of its circumference is removed because the belt has to pass through it in to the pulley wheel inside the tube. The design with stationary shafts makes the bearings less sensitive for angular mismatches. One of the pulley wheels has to be driven and this shaft can of course not be stationary. This wheel axis is the output shaft from the gear head and it does not reinforce the tube in the same way as the stationary shafts. Therefore it may be necessary to increase the stiffness of this post in some other way.

The toothed belt is reinforced with kevlar fibers. This kevlar fiber has a different thermal expansion factor from the aluminum structure of the satellite. Probably the belt and the structure will also have different temperatures. This fact will cause a mismatch between the belt and the structure, the belt can get slack or it can get to strained. To avoid this I have decided to use a mechanical belt strainer. This belt strainer is a small pulley wheel that is suspensioned on a spring. It is positioned just by one of the four pulley wheels next to one of the corner posts. The belt strainer wheel can flex in and out and in that way it keep the belt tight.

The thermal expansion coefficient (α) of the kevlar reinforced belt was specified by the manufacturer to be 70- $150*10^{-6}$ [K^{-1}]. This is a quite high value compared to the aluminum structure ($\alpha=23*10^{-6}$ [K^{-1}]). The design temperature interval of the belt is -40°C to +40°C. This means a maximum length difference of 18 mm for the 1500 mm long belt. The thermal expansion factor of the belt is considerable higher in magnitude than the thermal expansion factor of aluminum, therefore the structure (aluninum) is considered as unaffected by the temperature changes. The upper limit of the thermal expansion factor gives a worst case and the maximum length difference of the belt 18 mm. The belt strainer has to be able to compensate the length difference of 18 mm. This considerable expansion can not be compensated for by just one belt strainer because of the limited available space. Therefore a system of four belt strainers has been designed.

However, when the belt had been delivered, measurements showed that the thermal expansion factor was less than the specified. In fact just $7*10^{-6}$ [K^{-1}]. This means that the system with four belt strainers is much too oversized. I have decided to use only one of the belt strainers. One belt strainer is able to compensate for a length difference of 4.5 mm and the actual length difference is estimated to two mm.

Calculations of the thermal expansion and shortening of the belt as well as the geometry of the belt strainers are enclosed in appendix E.

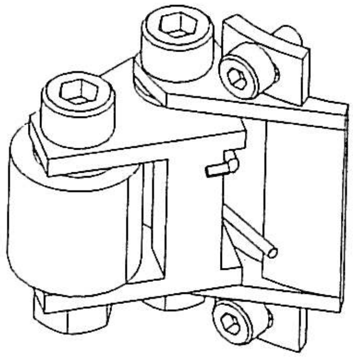

Figure 7.4 The belt strainer.

7.6 Gear motor

The selection and design of the drive system is not included in the degree project but a preliminary selection has been done. The drive system with its motor and gear is planned to be a stepper motor with a reduction gear head. There are several reasons to use a combination of stepper motor and gear head:

- A stepper motor is more suitable than a DC-motor because it has a lower magnetic moment than a DC-motor in the same range of output torque.
- A stepper motor with sufficient output torque to drive the belt without a reduction gear should be quite heavy and it should have a too big diameter to fit in the corner beam. It should also have a too big magnetic moment.
- If it is possible to find a non magnetic gear head from the shelf from any manufacturer, a sufficient output torque can be produced without causing too harmful magnetic moment with a stepper motor-gear head combination.

The drive system on the prototype is a gear motor from Minimotor SA, Agnio, Switzerland [ref.15]. This is a DC-motor with a five stage planet type gear head supplied as one unit. Both the motor and the gear head have too high magnetic moment to be used in the flight unit. The gear head can maybe be made sufficiently magnetically clean by changing the steel housing to a housing made of stainless steel. The motor has to bee changed, presumably to a stepper motor. A suitable stepper motor is the P110 from Escap, La Chaux-de-Fonds, Switzerland [ref.14]. The calculations of torque and gear ratios for the motor-gear head combination are presented in appendix B.

8 Materials Selection

Two different approaches could be applied for materials selection in space applications. The easiest way is to choose a simple material that has a well documented usage in space applications, such as aluminum, austenitic stainless steel or any other material according to the ESA paper "Data for selection of space materials" [ref. 1]. If a material with very specific properties is needed or a commercial fabricated product is used the material may not be documented in ref. 1. This means that it is necessary to verify that the material is suitable in a space application by tests. In the boom mechanism this concerns the material in the toothed belt, to some extent some of the other plastic details, the cable and the cable guide eyes.

8.1 General Demands

Here is a list of four general demands to consider when selecting materials for the Astrid II satellite components:

1. A general demand on all surfaces of the satellite, which are exposed to the space environment, is that they have to be electrically conductive. The boom mechanism is located at the outside of the satellite with the thermal shield on the inside of the mechanism, consequently the surface of the boom mechanism has to be electrically conductive. The demand on conductivity is created to prevent the surfaces from getting electrostatic charged and therefore spoiling the E-field measurements. A resistivity of 10^8 Ω/m^2 or less will be sufficient.

2. Another very important demand is magnetic cleanliness, preferably all parts have to be non magnetic. Total magnetic cleanliness is not possible to achieve but there is a limit of the total allowed magnetic moment of the whole satellite. This value is 125 mNm^2 and the maximum allowed magnetic moment of the different experiments onboard is distributed by their mass share of the total mass of the satellite. This means that the total allowed magnetic moment of the E-field experiment (electronics and mechanics) is about 25 mNm^2. The demands on magnetic cleanliness depends on the mangnetometric measurements that will be performed by an instrument onboard.

3. The materials used must be sufficiently resistive to the destructive environment factors in space e.g. cosmic radiation.

4. The materials must not emit gases in vacuum.

8.2 Materials for General Mechanical Components

In most constructive elements aluminum has been used, or stainless steel for components with high mechanical load. Aluminum is very good considering weight and magnetic cleanliness. Stainless steel is more problematic considering magnetic cleanliness. Austenitic stainless steels are almost nonmagnetic apart from ferritic stainless steels that are magnetic. The trade name "acid proof steel" often means austenitic stainless steel. A problem with austenitic stainless steels is that machining can cause phase changes in the material and make it magnetic. Therefore all parts made of austenitic stainless steel have to be examined in a magnetometer.

8.3 Material in Screws and Fasteners

The screws and nuts in the mechanism are standard components made of austenitic A4 70 (ISO designation). Titanium has also been discussed as an alternative. This alternative has been put off so far because of the difficulties to find a supplier. As for the general mechanical components mentioned above the magnetism of the fasteners varies between specific components and the magnetism of the fasteners have to be measured in a magnetometer in each specific case. This has not been done concerning the fasteners in the prototype. So the parts in the prototype that will be used in the flight unit have to be disassembled to measure the magnetic properties of the fasteners, alternatively the total magnetic moment of the assembled parts can be measured.

8.4 Materials in the Belt

The toothed belt is a quite sophisticated component and has to be purchased from a commercial manufacturer. This reduces the number of available of materials. The selected belt is delivered by Aratron AB and it is manufactured by Breco in Germany. The belt consists of two different materials, kevlar reinforcement in a matrix of Polyurethane. Kevlar has been used before in space applications and is documented in ref. 1. Polyurethane has also been used in space applications before but not this particular make so it has to be tested.

The belt has been tested in vacuum to see if it emits any gases. The evacuated gases in this test are analyzed in a mass spectrograph and the spectrogram has been enclosed in appendix D. This spectrogram shows the emitted gases after 30 minutes vacuum pumping and after 16 hours vacuum pumping, there is also a graph showing the mass distribution of the evacuated gases when there is no belt in the vacuum chamber, this graph is used as a reference. When the belt has been in the vacuum chamber for about 30 minutes the total pressure is 40 micro bar and the belt emits gases with two different molecular masses: 19, 43 and 55 g/mole. These gases have not been further analyzed. After 16 hours the total pressure is 4 micro bar. This is the same pressure as for the reference with empty chamber. If the spectrograms from this measurement and from the reference is compared you can see that the graphs are almost identical. We can see a big peak at the molecular mass 18 g/mole, this is water. The similarity in appearance between these two graphs indicates that the belt doesn't emit any gases. The change in appearance of the spectrogram for the belt indicates that the belt emits gases in the initial stage but after a while this will decay. Small differences in the spectrograms can bee seen but this is not considered a serious problem and the emitted gases from the belt have been considered acceptable.

The resistance to radiation for the polyurethane plastic in the belt has not been tested and no specifications have been found. The frequent use of polyurethane in other forms in space applications indicates that the radiation resistance should be acceptable. However long term resistance will not be required because the belt is driven once and will not be moved after the booms have been deployed. The booms are deployed in the beginning of the satellites life and at that time the belt has not yet been exposed very long to radiation, the dose is then low. It doesn't matter if the belt gets stiff by the radiation after a while, the only demand is that the belt with its attachment devices does not get so brittle by the radiation that it will be broken into pieces.

The polyurethane matrix of the belt is not conducting, so the belt has to be coated with a conductive material to meet the general demand nr. 1.

8.5 Selecting Plastic Type for Other Plastic Details

For several components, metal will not be feasible as construction material. These components are the "wire wrap profiles", the "wire holders" and the fixture for the probe in the probe bracket, see figure 7.2. These components have to be made of a soft material to not damage the wire and probe. There are some more mechanical demands on these details: The "wire wrap profile" has to be made of a material with a quite rough surface because the cable should be squeezed in the groove and not be able to slip. The demands on the probe fixture parts is quite different, they must have low friction to ensure that the probes do not jam when they should be let out.

The plastic for the probe fixation is easy to select: PVDF (polyvinylidenfluorid) has been used by the Swedish Space Corporation several times before in space applications and it meets all the demands for the fixation parts. PVDF is a fluoroplastic. The wire holders are manufactured in the same material because the demands on this component are purely geometrical and the material properties are of secondary importance.

The "wire wrap profiles" can not be made of the same PVDF plastics because its friction is too low and the cables would slip in the grooves. Actually a PTFE (Teflon) has been used. This type is in fact a fluoroplastic and has the same oily surface as all fluoroplastics but the used quality is reinforced with chopped glass fibers that gives it the desired rough surface. PTFE has a well documented use in space and the glass reinforcement is inert and stable and should not cause any problems.

All these plastic components are partly exposed to the space environment and thus their surfaces have to be coated with an electrically conductive material. There are however conflicting demands on the surfaces that are in contact with the boom cable, these surfaces should be isolating or must not be connected to the rest of the satellites surface (electrically ground level), see next chapter. Probably these surfaces have to be exceptions to the demand on electrical conduction, they are quite small and hopefully this will be no problem. The rest of the plastic components, that are not in contact with the boom cable, will be coated with a conductive material not yet decided which.

8.6 Selecting the Boom Cable

The boom cable is selected mostly by electrically demands but the cable must be compatible with the boom deployment mechanism. The most interesting parameters from the mechanical point of view is the flexibility of the cable and its friction properties together with the cable guide eyes that it runs through. These parameters are discussed in chapter 8.7 "Material in the cable guide eyes" and in chapter 10 "Preparatory friction tests". The electrical demands and the discussions in this chapters have ended up in the selection of cable RG 178 B/V from Habia Cable. This is a single conductor coax with a braided jacket in silver plated copper. The copper conductor is surrounded by a Teflon insulator and the outer diameter of the cable is roughly 1.2 mm. Several tests of this cable have indicated an ultimate tensile strength of 70 N. The spring constant of the braid is much higher than that of the conductor, such that the mechanical forces are carried by the braid rather than by the tiny conductor.

8.7 Material in the Cable Guide Eyes

There are two conflicting demands concerning the conductivity of the components in the boom mechanism that are in contact with the boom cable in retracted and in deployed state. These components are the so called wire wrap profiles on the corner bars of the satellite, the wire holders and the eyes that the cable runs through. The surfaces of these parts have to be isolators because of the fact that the boom cable is boot-strapped i.e. the jacket of the cable is forced to a specific electrical potential. If the cables are conducted to the satellite structure the boot-strap will be impossible. Consequently all surfaces that are in contact with the boom cables have to be isolators or they have to be isolated from the rest of the satellite. At the same time these surfaces have to be conductors and held at the same electrical potential level as the rest of the satellite.

These two demands are in obvious opposition to each other and there is no good compromise. One way to come to a solution is to manufacture this objects in a material with a resistivity acceptable for both demands Another way is to make these surfaces as small as possible and manufacture them in an isolating material. Because if they are very small the static charging hopefully will be no problem and we are not forced to coat these components with a conductive material.

Concerning the cable guide eyes the second way is probably the simplest way to come to a satisfying solution because the eyes can be made quite small. There are some other demands concerning the material in the eyes so the second way makes the choice of material easier (with fewer demands).
Other demands on the material in the cable guide eyes are:

- Low friction in combination with the boom cable.
- Non magnetic.
- Resistant to the destructive factors in space e.g. cosmic radiation.
- It must not emit gases in vacuum.

The tribological (the discipline of friction and wear) point of view has been discussed with Thorvald Eriksson at the Department of Machine Design/Machine Elements at KTH. Here follows a short summary:

It is not recommendable to use metallic materials in both cable jacket and eyes. This material combination can cause the materials to stick together in vacuum. The cable jacket is made of a metallic material, consequently the eye should not be made of a metallic material.
The short sliding length makes the theories of dry lubricated slide bearings unusable because this theory relies on that the surfaces abrade in the beginning of the life cycle and the soft, plastic material which has been created serves as lubrication. This problem differs from "traditional tribology problems" in the following ways:

- The contact force is very low.
- The operation time is very short.
- The slide length is very short.
- The movement is linear and in one direction.

Most of these facts make this problem less critical than many other tribological problems.

The material in the cable jacket is silver plated copper. Silver is a soft and easily sheared material. A simple rule of thumb is that a soft easily sheared metal should be combined with a firm non metal to give the lowest possible friction. Therefore a ceramic material would be good for the eyes. All demands are satisfied by a ceramic material (except for the ambiguous demand on the conductivity).

A problem with ceramic material is that they are difficult to machine, on the other hand prefabricated ceramic bushings and rings are available as standard components for different purposes. I have found a top eye from a fishing rod made of the electrical isolator aluminumoxide which fits the demands on material properties and geometry.

8.8 Material in Slide Bearings

The folding mechanism and the belt transmission contain several slide bearings. These bearings are manufactured by Glacier, type DU-bearings. These bearing bushings have a well documented usage in space applications. The sliding surfaces are made of a fluoroplastic-lead composite and the frame is made of bronze, this satisfies the demands of magnetic cleanliness.

The sliding surfaces in the guide rails for the folding mechanism have to be coated with a low friction material. The DU-material is available as a bronze sheet covered with the fluoroplastic-lead composite. This DU-sheet is suitable as low friction liner in the guide rail.

8.9 Material in Torsion Springs

The belt strainer and the probe fixation devices contain torsion springs. The relevant material demand on these springs besides the elastic properties is the magnetic cleanliness. The selected materials are the two austenitic steels: SS 2570, that is a non magnetic acid proof steel and SS 2347 that is also an acid proof steel with very low magnetism [ref.16]. SS 2570 is used in the belt strainers and SS 2347 is used in the probe fixation devices.

8.10 Selecting Adhesives

Some components of the deployment mechanism have to be fastened with adhesives. The cable guide eyes have to be fixed in the holes in the eye holders and in the guide rails the DU-liner has to be glued. The selected adhesive is Araldite AV 238M/HV998 (100/40 pbw). This epoxy type adhesive is documented in ref. 1 and has a well documented usage in space applications.

9 NECESSARY TESTS, A DISCUSSION

To make sure that the boom deployment works properly it is very important that the friction between the boom cable and the cable guide eyes is not too big. Therefore this friction has been studied in some tests before the work with the design of the mechanism was started. This was done to assure that the chosen concept was practically feasible. These preparatory friction tests are described in chapter 10.

The development of the deployment mechanism can be separated in four stages namely: the preparatory tests mentioned above, the selection of materials, design and verifying tests. The verifying tests are the tests that have to be done with a prototype of the deployment mechanism to prove that it will work properly. The straight forward way to do this would have been to test the mechanism in weightlessness over and over again until you are sure that the mechanism works properly. When this is not possible we have to identify critical parts of the design and the deployment procedure and test these one by one. The verifying tests are:

- Mounting of all components and a simple function test of the belt transmission and the folding mechanism. The wire booms are not mounted in this test. This test is done in room tempered air and the four folding mechanisms are tested with regard to preparations for launch (the possibility to in a as simple and safe way as possible put the deployment mechanism in launch configuration) and function of the folding mechanisms.

- The second test that has to be done is a function test of the boom deployment. This test is done in room tempered air and one boom is deployed with the centrifugal force of the satellites rotation substituted by the gravitational force. The boom deployment test is started by with the release of the probe from its storage in the probe bracket and finished when the boom cable has been deployed its total length.

- The mechanism has to be tested at the extreme temperatures of its working range i.e. at -40°C and at +40°C. The critical point in this interval is the lower point where the cables and the belt get stiffer and the friction in sliding bearings increases. The deployment of the cable through the eyes and the flexibility of the cable has already been tested at this temperatures so the first thing we need to test is if the belt transmission and the folding mechanisms work properly at -40°C. A critical thing is if the belt strainers are able to compensate for the expansion and contraction of the belt due to temperature changes and if the folding mechanisms stay unaffected of these changes of length of the belt. It is of course very interesting to see if any bearing or any sliding surface will jam at low temperatures. Though the deployment of cable through the eyes already have been tested at low temperatures a complete deployment test as described above should be done at -40°C to see if the cool and stiff boom cables will be straightened sufficiently by the centrifugal force on the probes.

- A fourth test to do is a vibration test. In this test the mechanism is tested for sensitivity to the vibrations during the launch. The complete mechanism is mounted on a satellite mock-up with mass dummies and the whole assembly is vibrated in an oscillator with a spectrum that simulates the oscillations during the launch. The interesting things in this test are in the first place to see what happens with the flexible parts of the mechanism: the wound and prestressed cables, the belt, the belt strainers and the folding mechanisms when the satellite is oscillated. Secondly we test the function of the mechanism after the oscillation to make sure that no parts have jammed. This test is a simple function test of the belt transmission and the folding mechanisms.

- After this tests it is time to decide if the mechanism is ready for launch or if the test schedule shall proceed with a complete deployment test in weightlessness. Such a test can be done in cooperation with ESA in a test aircraft that during a parabolic flight provides very low gravitation for about twenty seconds. Twenty seconds is a little too short time to perform a thorough test but it should be enough to test the critical moment of the deployment when the probes are released from their storages and the folding arms are folded out.

10 Preparatory Friction Tests

It was important to study the friction at the low temperatures that the mechanism can be exposed to in the space environment. Another important factor was how vacuum affected the friction. It is well known that vacuum can cause two metal surfaces to stick together. The very best would have been if the vacuum and temperature tests could be done at the same time in the same test. Unfortunately this was not possible with the equipment available at the Alfvén laboratory. The tests have instead been separated in one series of vacuum tests and one series of low temperature tests. Possible effects depending on the combination of temperature and vacuum can consequently not be studied.

To start with, some tests of the friction in room temperature has been done, partly to test the equipment and partly to get references for comparison of the tests in vacuum and at different temperatures. In the first place the tests have been done to show if any difficulties will appear when the cable is deployed. In the second place the choice of materials and geometries that minimize the friction have been tested. Thirdly the minimum pulling force in the cable that makes sure that the cables don't jam has been determined.

To study the friction phenomena during the deployment of the boom cable a test rig has been designed, see figure 10-1. This test rig can be used for tests in a vacuum chamber and for tests in a temperature camber. The test rig has been designed to simulate the conditions in space during the deployment. This is done by letting a weight pull the cable through two eyes positioned approximately like the cable guide eyes on the deployment mechanism. The centrifugal force on the probes is accordingly simulated by the gravitational force on the weight. The mass of the weight can be varied to simulate probes with different weight, different rotational speeds of the satellite and different stages of the deployment. The movement of the cable trough the eyes is attended by a linear guide driven by an electric motor. Between the cable and the linear guide there is a force gauge, this measures the pulling force in the cable. By measuring this force by the force gauge when the cable is deployed and comparing it to the gravitational force in the other end of the cable one can calculate a resisting coefficient (μ) quite analogous to the coefficient of friction. How this is done is shown in appendix A.

This test, together with the electrical demands, is the foundation for the selection of a suitable boom cable.

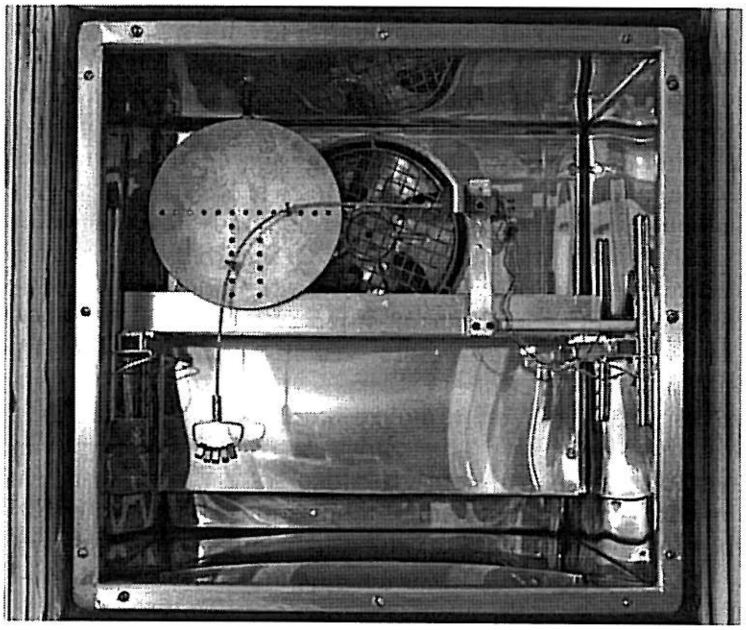

Figure. 10-1. The friction test rig in the temperature chamber.

10.1 Tests in Room Tempered Air

Purpose:
To show which problems that can appear during the deployment and to show which geometry that are suitable. The tests in room tempered air is also done to find suitable materials and the most feasible type of cable construction from the mechanical point of view. Another purpose with this test is to get references for comparison of the vacuum and the temperature tests. This approach makes it possible to do the tests in vacuum respectively at low temperature less extensive, more like sample tests that show the differences between the conditions in room tempered air and the conditions in vacuum and at different temperatures.

Performance:
A quite long series of tests has been made with different geometries, with different pulling forces and with three different cables. All these cables are conceivable in a flying boom system and they have the specifications according to table 10-1.

	cable 1	**cable2**	**cable3**
Description	coax. with one conductor	coax. with eight conductors	coax. with one conductor
Manufacturer	Habia Cable	Weizmann Consulting	Habia Cable
Designation	RG 178 B/v		
Diameter (mm)	1,2	2,2	2,0
Mass (g/m)	2,5	8,0	10,0

Table 10-1. Specifications of the available cables.

Result:
A suitable geometry of the cable guide eyes was found. The rest of the tests were done with this geometry. The result of this tests is a numerical value of the resisting coefficient (μ) for different values of the pulling force. The resisting coefficient is calculated as described in appendix A. These results are presented in the graphs in figure 10-3. In the graphs there are two curves, one describes the upper limit of the resisting coefficient the other describes the lower limit.

Figure 10-3 shows that all cables have a resisting coeff. μ of 0.2-0.3 at high pulling forces in the cable. At low forces the difference between the cables is more obvious. The thin cable (cable 1) appears to have a low μ (less than 0.4) down to forces around 50 mN. The other cables have a resisting coeff. of about 1 at this low pulling force. For $\mu=1$ the deployment probably fails.

The difference in resisting coefficient depends on the difference in bending stiffness. The thicker cables have a high bending resistance with hysteresis due to internal friction in the cable. The resisting coefficient that is used here takes in consideration all forces that resist the movement, not just the friction in the contact points. When the cable is moving through the eyes it has to bend and then be straightened again. This bending consumes energy and of course more energy the stiffer the cable is. The more energy that is consumed, the bigger the resisting force is.

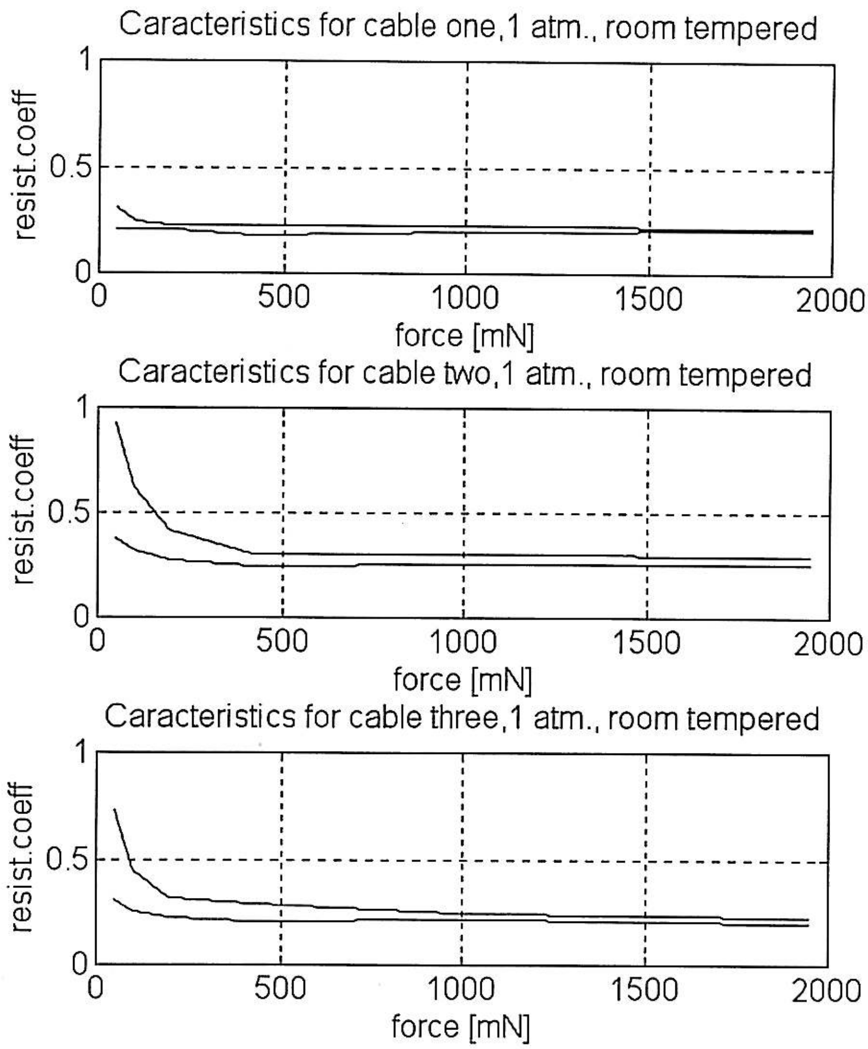

Figure 10-3. The resisting coefficient for the different cables in room tempered air.

10.2 Tests in Vacuum

Purpose:
To determine the difference between deployment in room tempered air and deployment in vacuum.

Performance:
The tests in vacuum are more complicated than the tests in air and were therefore done with a selection of weights and they are compared with the tests in room tempered air. All three cables were tested in vacuum. The tests in vacuum were done in the same rig as the tests in air. The pressure in the vacuum camber was $7*10^{-4}$ Pa.

Results:
The results from the tests in vacuum are shown in figure 10-3. The basis of this graph is not so extensive: two tests with cable 1 and cable 3 and three tests with cable 2.

Also in the vacuum tests cable one had the lowest friction at low pulling force (50 mN). The resisting coefficient does newer exceed 0.3, which is a bit surprising because it is a lower value than in air. In other respects the results of tests in vacuum are quite similar to the results of the tests in air.

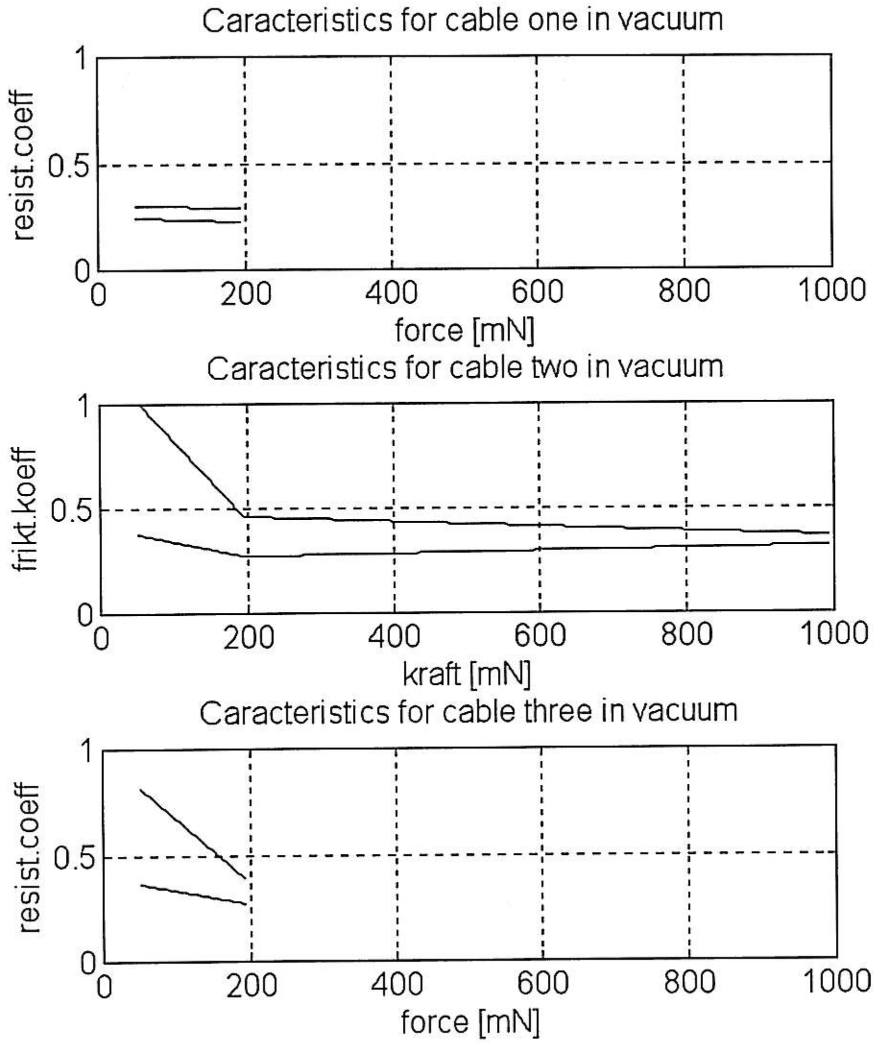

Figure 10-3. The resisting coefficient of the cables in vacuum, two samples with cable one and cable three, three samples with cable two.

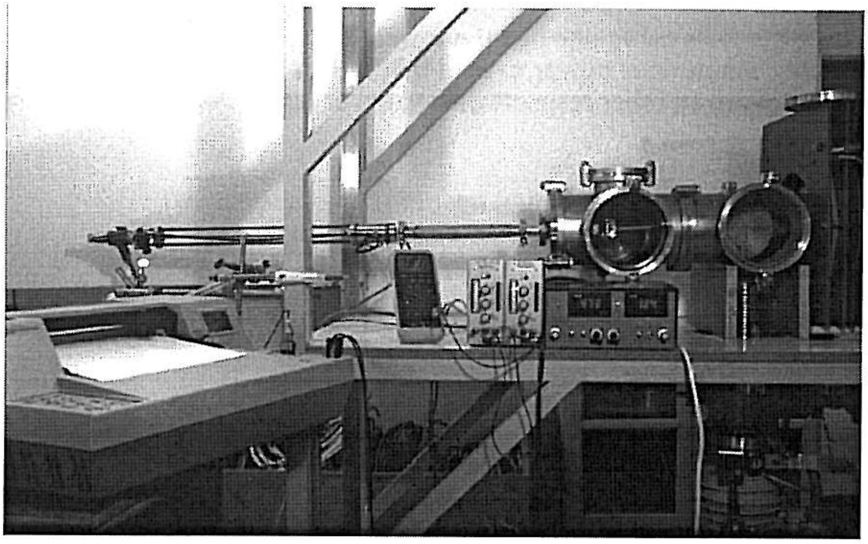

Figure 10-2. The test rig in the vacuum chamber.

10.3 Tests at Low Temperatures

Purpose:
To show the difference between deployment in room tempered air and deployment in cold air.

Performance:
As for the tests in vacuum the tests at low temperatures are based on a sample of parameter values from the tests at room temperature. All three cables were tested in the same test rig that was used for the other tests, this time located in a climate chamber at a temperature of -60°C. For each cable two tests were done, with pulling forces 50 mN and 200 mN respectively.

Results:
The tests at low temperature differ a lot compared to the tests at room temperature, the resisting coefficient is much higher. Cable one has the lowest μ, lower than 0.6 in all tests. Maximum resisting coefficient for cable 3 is about 0.8 and for cable 2 about 1.4. This value is of course too high, $\mu > 1$ means that the cable has to be pushed through the eyes. Accordingly cable one has the lowest resisting coefficient also at low temperatures.

The explanation to why cables two and three have so high resisting coefficients can maybe be found in their increase in stiffness at low temperatures. This assumption can be verified by some simple bending tests of the cables at low temperatures. This is described in the following chapter.

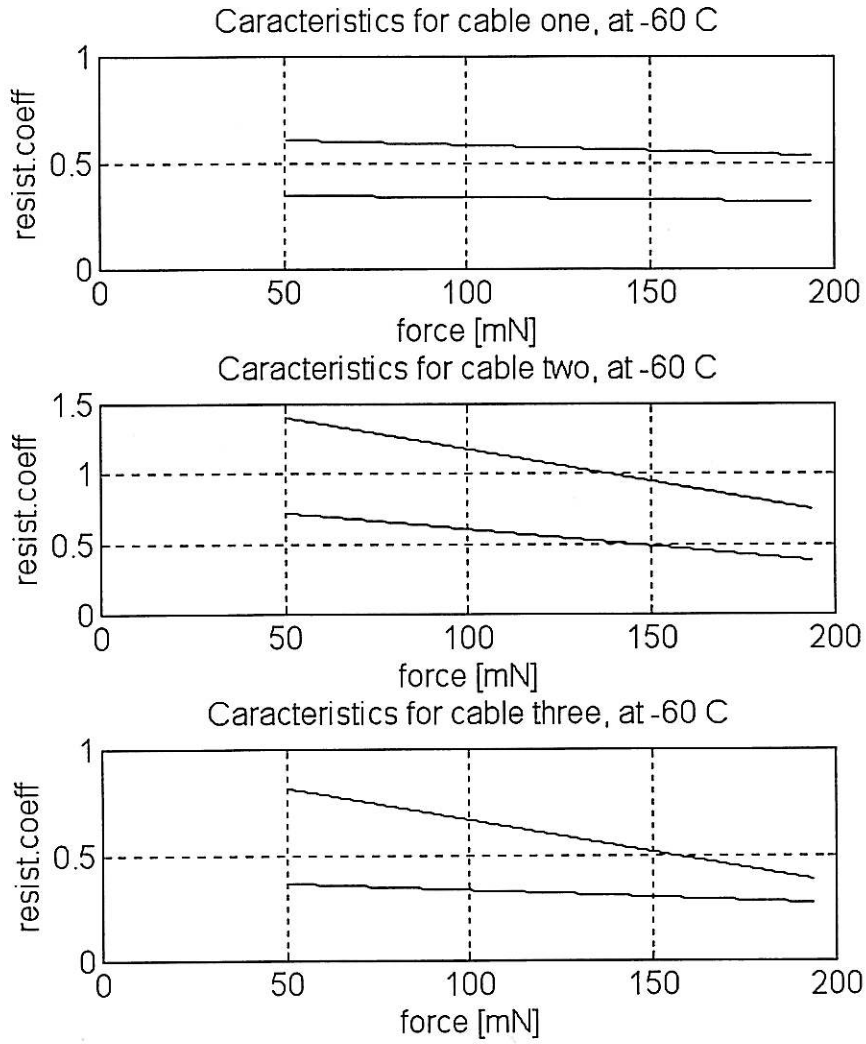

Figure 10-5. The resisting coefficient of the cables at -60°C. Observe the scale of the x-axis.

10.3.1 Bending resistance of the cables

Purpose:
To investigate in which way the temperature affects the bending resistance of the cables.

Performance:
The test rig was modified to be able to measure the bending force when a cable is bent like a beam with one fixed end. The applied force is plotted as a function of the movement of the outer tip of the cable. This test was done with all three cables, first at room temperature and then at -60°C. The test is controlled by the deformation and is in progress until the cable is bent so much that it slips off the eye that controls the deformation.

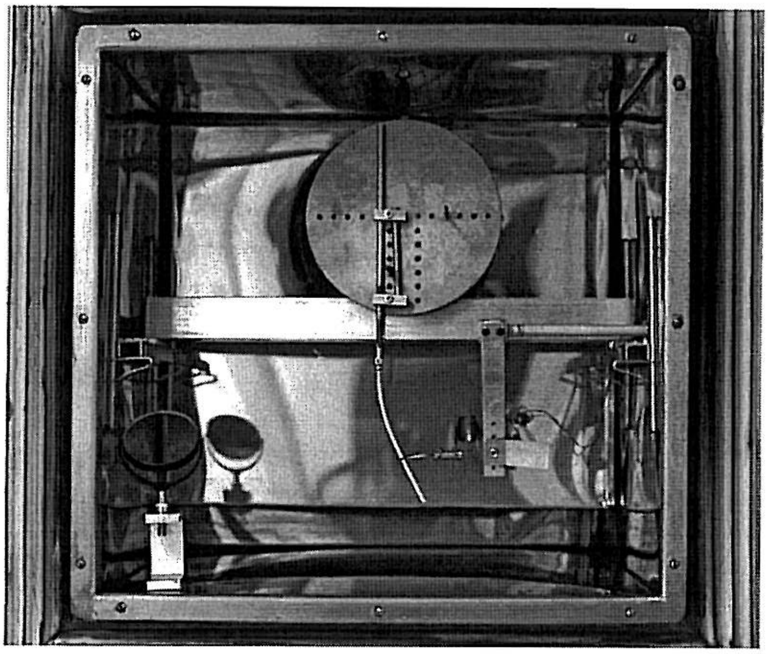

Figure 10-6. Test rig for investigation of the stiffness of the cable at different temperatures.

Results:
The plots show that the characteristics of the cables is not in accordance with the linear force-deformation function for an elastic beam. The applied force increases, when the cable is bent, up to the elastic bending limit of the cable. When this deflection is reached the force is constant independent of the deflection, a plateau of the deflection-force plot.The cables can not be considered as elastic but rather having two different stiffness components: One elastic that gives the force-deformation curve a linear appearance. And one plastic stiffness that gives a constant resistance against deformation independent of the degree of deformation and that gives a permanent deformation that remains after unloading.

In this analysis these two stiffness components are not considered separately but we search for a simple quantitative measure that characterizes the stiffness. We simply use the maximum measured force during the deflection to describe the stiffness.

In table 10-2 this maximum force (the elastic bending limit) is shown for the three cables at 20°C and at -60°C. Cable one is much more flexible than the other two. Cable two however changes stiffness very little, it has only 22% difference in stiffness between the low and the high temperature.

	Cable 1	**Cable 2**	**Cable 3**
Maximum force at 20 C (mN)	14	86	62
Maximum force at -60 C (mN)	20,5	105	134
Percentage stiffness difference	46%	22%	116%

Table 10-2. The stiffness of the wires at different temperatures.

10.4 Summary of the preparatory friction tests

In all the tests cable one seems to be the most suitable alternative. In the friction tests at -60°C cable one is the only practical useful alternative, because the other two causes too high resistance against movement.

A very important thing to determine is the minimum centrifugal force that gives a reliable deployment. After this friction tests we are able to specify this minimum force. It is a matter of judgement how big the safety margins should be. It is a fact that cable one has newer show any tendency to jam (μ=1), not even a the lowest pulling forces. In other words, we can not see any lower limit for the centrifugal force. This is of course very good but, how strange it even may seem, a problem when you are about to determine a minimum centrifugal force. We can't do it the conventional way to multiply the force when the cable jams with a safety factor. We have to do a judicious decision when we settle the minimum force. The bigger the centrifugal force is the safer the deployment is. We can not accept any risk that the deployment should fail on account of too small centrifugal force. Therefore we specify the minimum centrifugal force to 200 mN, this is the highest tested pulling force in the vacuum and temperature tests of cable one.

There is however a more critical stage in the real deployment than in the preparatory friction tests: When the cable guide eyes pass the corners of the satellite. There the cable is wound in a radius around the corner posts of the satellite. When this part of the cable is about to be deployed it has to be lifted up from the groove in the winding profile and then it has to be bent in a radius in the opposite direction to how it was wound. These more critical circumstances in the actual deployment in space than in the tests motivate the pulling force of 200 mN that considering the graphs from the tests seems quite high.

11 VERIFYING TESTS

I have already mentioned which verifying tests I consider necessary in chapter 9. Due to the limited time schedule of this degree project one of this tests has not yet been performed. It is the vibration test that has been postponed. The test in weightlessness that is mentioned in chapter 9 has not been performed either but I doubt its significance because the short time of weightlessness that can be provided during a parabolic flight (about 20 seconds). Twenty seconds is enough to test the first critical moment in the deployment; when the folding mechanisms are working, but to test the complete deployment procedure we have to rely on the function test where one boom cable is completely deployed with the gravitational force acting instead of centrifugal force. My point of view is that it is not worth the effort to do the test in weightlessness.

11.1 Function Test of Belt Transmission and Folding Mechanisms

Purpose:
To verify if the components in the deployment mechanism fit together and to show if the folding mechanisms have any tendency to jam.

Performance:
The belt transmission is assembled and mounted on a platform mock-up. The motor package used consists of the gear head suggested in appendix B: gear head 30/1 from Minimotor AS with gear ratio 1526:1 combined with a standard DC-motor from Minimotor AS. The folding mechanisms are mounted on the belt and on the corner posts but the boom wires are not put in place. The folding mechanisms with their guide rails are possible to adjust in many ways e.g. the folding movements of the four mechanisms must be synchronized, these are controlled by one adjustment. All these adjustments are done.

During the test the belt is driven and the folding arms are folded out. This test is repeated four times. The guide rails on the corner posts are not supposed to be adjusted to fit only one of the folding mechanisms on the belt. The folding mechanisms shall be able to work in all four possible positions. Therefore this test is done four times, one in each position.

The DC-motor current is measured continuously when the folding arms are folded out. This is done to analyze if there is any tendency that the mechanisms should jam. The torque on the motor shaft can easily be calculated from the motor current. A jam in the mechanisms will cause an increase in the torque on the motor shaft and can easily be detected by measuring the current. The motor current is registered by a graphic plotter with a current-time scale.

Result:
The four guide rails can be adjusted in such a way that all four mechanisms can work in all four positions.

In the first test the mechanisms were jamming. The problem was a part of the guide rail that had been designed in an improper way. The guide arms of the folding mechanisms were impacting on the guide rail during their motion. A dimensional modification of the guide rail solved this problem and the folding mechanisms showed no tendency to jam when the guide rails had been modified.

The motor current was quite constant during the four tests, about 80-110 mA. The motor voltage was 6.8 V, implying the power dissipation 544-748 mW. The motor efficiency is about 40% in this load case so the mechanical output power was about 250 mW. Finally the output torque of the motor was about 0.6 mNm and the output torque of the gear head was about 0.5 Nm. All according to Minimotor SA product catalogue [ref.15].

11.2 Boom Deployment Test in Air at Room Temperature

Purpose:
To show if any problems occur during a wire deployment.

Performance:
The deployment test is done in room tempered air and one boom is deployed with the centrifugal force of the satellites rotation substituted with the gravitational force on a weight in the end of the boom. The boom deployment test is started by the release of the probe from its storage in the probe bracket and finished when the boom cable has been deployed its total length. The mass of the weight in the end of the boom is 20g, so the pulling force in the wire is 200 mN. This is the specified minimum pulling force in the wire, see chapter 10.1.5 "Summary of the preparatory friction tests" and chapter 12.1 "Required rotational speed and probe masses".

Result:

Some observations were done during the deployment test:

- When one of the folding mechanisms (number 2) is passing the corner post the boom cable is touching the folding arm for a moment. Fortunately this causes no problems in the deployment of the cable, the only concern is that the cable jacket is conducted to the satellite body and it is not possible to do any measurements by this probe for a short while. Probably no measurements will be performed during the boom deployment and then this problem will be no problem!

- The wire holders have some burrs from the manufacturing in the grooves where the cable is lying. These burrs obstruct the wire when it is about to be lifted from the groove, it does not ruin the deployment but it makes it look a bit unreliable. However the burr is easy to remove and the problem is out of the way.

- The angle of the first of the cable guide eyes could be changed so the direction of the centerline of the eye is parallel to the cable running through it, this makes the cable sliding easier through the eye. The centerline of the eye is parallel to the direction of the belt and it could be changed about 30° to follow the average direction of the cable.

- When the probe is released from the probe bracket and the folding mechanism not yet has been driven, the probe is hanging quite close to the outer cable guide eye and the boom wire is bent in a 90°-curve through the outer cable guide eye. Some kind of support for the probe stub should be mounted on the folding arm to keep the probe in a position where the cable is not bent in this sharp curve. This support should fill a very important function when the probe is released from the probe bracket. Without the support the probe release is quite violent: the probe is brought into a pendulum motion by the centrifugal force and is bumping into the satellite body. The support will catch the probe in its pendulum motion and prevent the motion. When the folding mechanism is driven the folding arm is revolved 90° and this support end up in a position where it does not affect the cable deployment. This support has not yet been designed.

11.3 Boom Deployment Test in Air at -35°C

Purpose:
To show if any problems occur during a wire deployment at the lower extreme temperature.

Performance:
The test is performed in the same way as the boom deployment test in room tempered air. The test is performed four times, once with each boom.

The desired temperature was the lower extreme temperature in the working range of the mechanism, -40°C. No temperature chamber with this performance was found so the tests had to be done at -35°C. The tests were performed in the laboratory of the Department of Energy Technology, Division of Applied Thermodynamics and Refrigeration, KTH.

Figure 11-1. The prototype in the temperature chamber.

Results:
The four problems that occurred in the deployment test in room tempered air were also seen in this test, because the imperfections had not been eliminated.

Some other imperfections in the boom mechanism where also observed at this low temperatures:

- Firstly the gear head 30/1 from Minimotor AS jammed because the lubrication in the gearbox became to viscous. The driving package is specified in a range down to

-30°C and -35°C was too much for it. It is a bit hazardous to use a multi step planetary gear head with that high reduction (1521:1) at this low temperature. The matter that make this gear head so sensitive to viscous lubrication is the low torque in the first gear stage (5 mNm). When the lubrication starts to get a bit sluggish this is enough to stop the rotation in this first stage. To be able to perform this test the gear head was heated by a kanthal wire wound around the gear head. This solution is not realistic in the flight unit. To solve this problem one can maybe replace the oil with another more suitable type, for example the Dupont Krytox grease, but it is probably safest to choose a gear head with lower reduction and a motor with higher output torque.

- Secondly there was a problem with one of the guide rails. One of the low friction liners that are glued to the guide rail came loose during the test. This low friction liner is a bronze sheet covered with DU-material (chapter 8.7) and is glued to the aluminum guide rail with a epoxy type adhesive. This adhesion is obviously insufficient but no alternative has been found yet. A hypothesis to why this liner came loose is the different thermal expansion factor (α) of bronze and aluminum (α_{Al}=23.8*10^{-6} [K^{-1}] , α_{Bronze}=17.5*10^{-6} [K^{-1}]). This difference will cause shear strain in the boundary between aluminum and bronze.

- A third observed problem was the stiffness of the cables caused by the low temperature. This stiffness does not affect the deployment procedure but it does affect the deployed cable. The problem is that the shape that the cables are in when the booms are in launch position will residue to some extent even in deployed mode. The cable is wound over the corner posts and is forced to a complicated shape in the folding mechanism when it is in launch configuration. The cold and stiff cable is not fully straightened by the 200 mN pulling force when it has been deployed. On the other hand the maximum pulling force during the deployment is about 500 mN according to the deployment sequence outlined in chapter 12, but the boom wires have not been tested with 500 mN pulling force. The bends caused by the radius of the corner post is actually not a problem because they get almost straight by the pulling force but the bends caused in the folding mechanism are sharper and are not completely straightened by the pulling force. These permanent bends cause the probes to hang a bit oblique, the probe stubs are not pointing in an exact radial direction, a maximum deflection of about 3° has been observed. Maybe this can cause bad accuracy in the measurements, this has to be determined by the scientists that use the data from the E-field measurements. This problem was also seen during deployment at room temperature but not to this extent. The shapes of the deployed booms are showed in the photos in figure 11-1, there one can see the permanent bends of the boom wires.

The stiffness of the cable affects the accuracy of the position of the probe and the angle of the probe stub. The accuracy of the position of the probes can be estimated by using the elastic bending moment limit of the cable. A deviation from the ideal probe position demands a bending moment in the wire in the attachment point between wire and satellite body. The maximum bending moment in the cable is the elastic bending moment limit of the cable. Accordingly: the maximum deviation of a probe is the deviation that causes the moment in the attachment point equal to the elastic bending moment limit. The elastic bending limit of the cable has already been measured in connection to the preparatory friction tests in chapter 10 (10.1.4 Bending resistance of the cables), these results can be used here. The deviation of the probes from their ideal positions is mentioned in chapter 5. "Further studies and additional design work to be carried out" consequently this analysis has not been done yet.

The motor current was quite constant during the four tests, about 70-100 mA. This value is in fact lower than the current through the motor in the tests in room tempered air (chapter 11.1). The cause for this low value is probably the heated gearhead, a temperature sensor on the gearhead housing showed 45°C. The conclusions of this are:

- The increase in mechanical resistance due to the low temperature of the belt transmission and folding mechanism is neglectable.
- The mechanical resistance (and the mechanical efficency) of the gearhead is strongly influenced by the temperature.

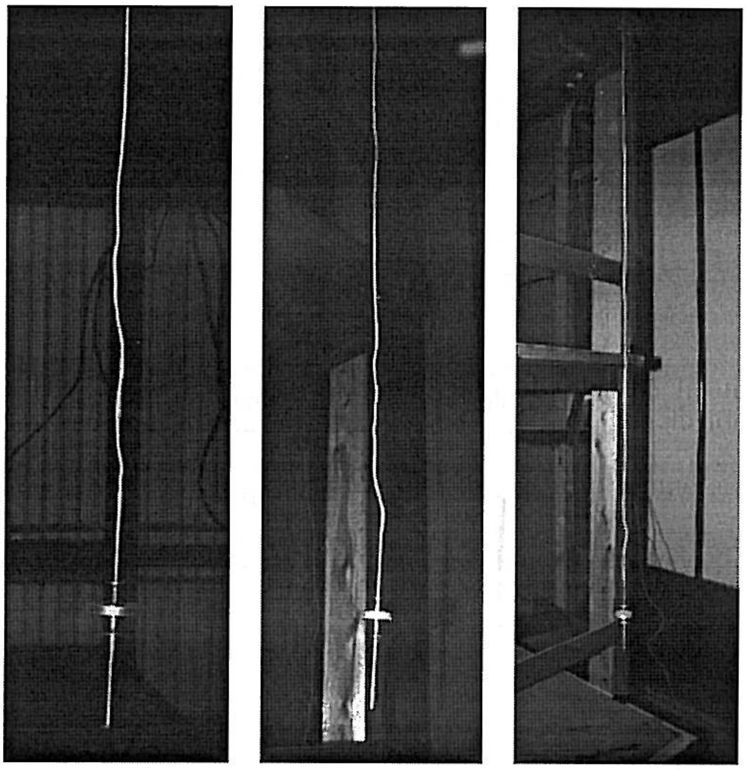

Figure 11-2. The shape of the cables deployed at -35°C.

12 Characteristics of the Deployment Procedure

In section 10.1.5 "Summary of the preparatory friction tests" the minimum centrifugal force was set to 200 mN. 200 mN is a require specification and in this chapter a suitable deployment procedure is outlined. It is just a proposal based on the approximated moment of inertia of the platform: 1 kgm^2.

The deployment procedure consists of to stages: the release of the probe from the probe bracket and the deployment of the boom wire. The probes should be released at a rotational speed as low as possible not to get damaged. Then rotational speed can be increased before the wire deployment starts.

Factors determining the centrifugal force on the probes are of course the probe masses and the rotational speed of the satellite. The probes can be made quite light, approx. 25 g, but we have to afford some extra mass here because otherwise the rotational speed has to be very high to get the desired centrifugal force. We specify the probe mass to 70 g, this is a reasonable value. If we use the demand for minimum centrifugal force 200 mN we get a demand for the rotational speed as a function of the position of the probes. This function is described in the lower curve in figure 12-1. In figure 12-1 we can also see a curve that describes the obtained rotation speed for the satellite if the initial speed is 31 r.p.m.. This second curve is calculated from the postulation that the initial rotation speed should give the probes a centrifugal force of 200 mN when they are positioned in their inner position (right by the satellite platform). Then when the booms are deployed no torque is applied to the satellite body, so the moment of momentum for the whole system is constant. When the probes are deployed the moment of inertia for the whole satellite has been increased, consequently the rotational speed will decrease.

As figure 12-1 shows, the obtained rotational speed is higher than the required rotational speed up to the probe radius 2.8. The radius from rotation center to the fully deployed probes are 3.3 m i.e. the obtained rotation is just enough for the demand on the centrifugal force.

The calculations of the obtained rotation are based on that the moment of inertia of the platform is 1 kgm^2. This is an estimation based on the value for Astrid I and the analysis has to be modified when a more accurate value is known.

A calculation of the obtained pulling force in the probe wires has been done and the result is shown in figure 12-2. These calculations are also based on a moment of inertia of 1 kgm^2 for the platform.

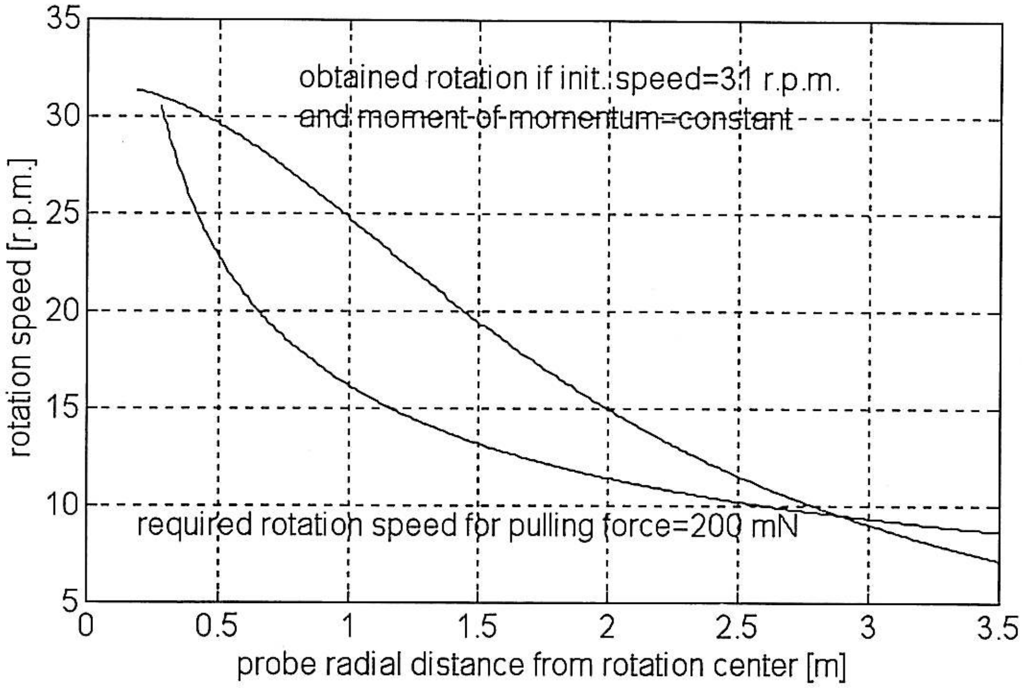

Figure 12-1. Required rotational speed and obtained rotational speed during deployment.

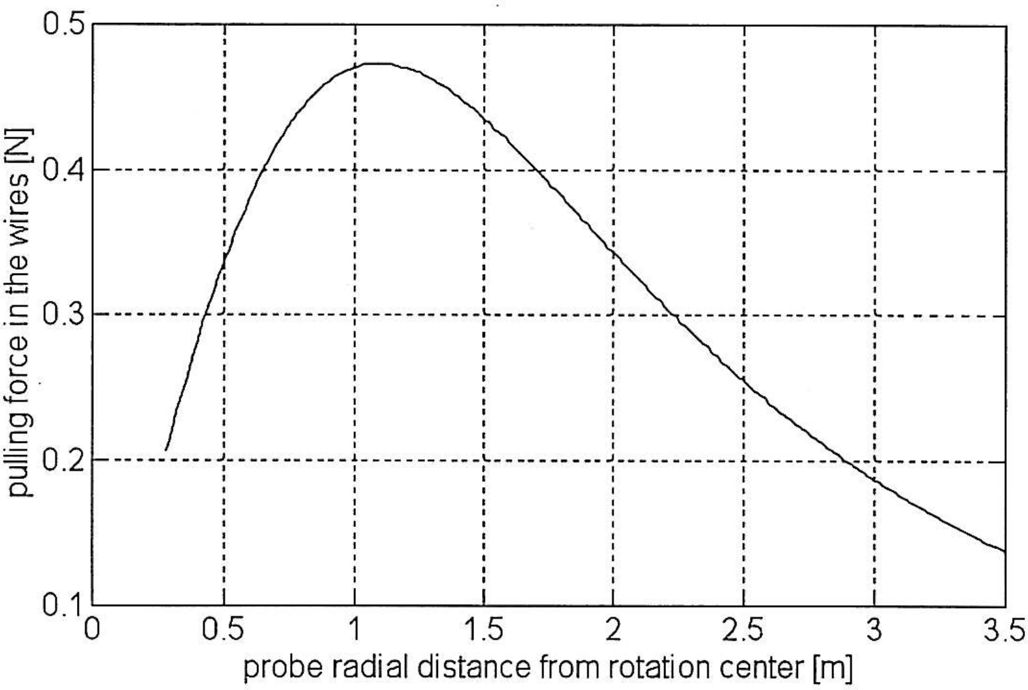

Figure 12-2. Pulling force in wires during deployment.

13 STRUCTURAL, MECHANICAL PROPERTIES

Most parts in the deployment mechanism have very clearly defined mechanical function but no clearly defined, and often very low, mechanical loads. In the design process it is often the main problem to get the correct mechanical function of the part and the structural strength is not deeply analyzed. The loads caused by the vibrations during the launch are in many cases the decisive load case and in these cases I have used an estimated load case. This is a static acceleration load of 30 g applied in an arbitrary direction. The probe bracket and the probe stubs have been dimensioned in this way.

The estimated 30 g load case is insufficient in some cases, the flexible parts that can get into resonance caused by the vibrations during the launch have to be further analyzed. Parts with a resonance frequency in the same order of magnitude as the structural resonance of the platform probably runs a big risk to get excited into resonance. It is desirable with component resonance frequencies exceeding the structural resonance frequency. The structural resonance frequencies of the platform are estimated to be in the interval 40-80 Hz and there are different resonance frequencies in different directions. The wound cables and the toothed belt are very flexible components that can get excited into resonance and then oscillate like guitar strings. The resonance frequency of the wound cables has been increased by a support at the middle of their free length, the "wire holders". With this support the lowest resonance frequency of the cables is about 180 Hz when they are tensioned by 20 N. The resonance frequency of the toothed belt varies between 50 and 100 Hz as the tension of it varies in the interval 30-100 N, this is of the same order as the resonance frequency of the platform and not desirable. Maybe the belt tension has to be increased to over 80 N to get the resonance frequency higher than 80 Hz. The belt tension is set by the spring in the belt strainer. A vibration test will be done to check the vibration amplitude of the belt and decide necessary belt tension.

14 SPECIFICATIONS

The boom wire system compiles with all performance parameters of this section after being subjected to the tests specified in chapters 10 and 11.

14.1 Temperature

The system will survive the following temperatures, operational and non-operational, without performance degradation:

Mechanism working temp-limits: -40°C to +40°C *

Deployed boom temp. limits: not confirmed

* with exception of the driving package used in the prototype which has the working range -30°C to 100°C.

14.2 Performance Parameters

14.2.1 Deployment Speed

The nominal deployment speed is 5 mm/s with the driving package used in the prototype and the speed can be varied within the range of 0 mm/s to 10 mm/s by voltage variation.

14.2.2 Deployment Loads

The mechanism will deploy the wire boom cable under the following cable tensile load conditions:

Load range: 0.2 to 2 N

Angle of deployed wire boom cable with respect to the wound boom cable (this angle varies as the cable guide eyes move from one corner to the other): -45° to +45°.

Angle of deployed wire boom cable with respect to the spin plane: -10° to +10°.

14.2.3 Deployment Control

The mechanism is capable of starting and stopping wire deployment at any time and to maintain any deployed length with motor power off.

14.3 Physical Characteristics

14.3.1 System Mass Properties

The total system mass is approx. 1.77 kg.

14.3.2 Wire Boom Cable Properties

The maximum length of one deployed cable is about 3.05 m (depends on the platform size, this value is valid if the length between the centers of the corner posts is 354.2 mm).

The jacket of the boom cable is electrically insulated from the probe, deployment mechanism and platform in deployed state. In launch mode there is no insulation between the jacket and the deployment mechanism. The jacket is conducted to the folding mechanism and to the jacket of another boom cable in launch mode.

14.3.3 Magnetic Properties

The total magnetic moment of the components in the **folding** mechanisms, not including the driving package, the belt transmission, the screws in the mechanism, the boom wires and the probes, is about 2 mAm^2. The main cause to the magnetic moment are the shafts in the folding mechanism that are made of steel SS 2343. The driving package used in the prototype has a magnetic moment of about 18 mAm^2.

15 REFERENCES

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APPENDIX A

Method for Calculating the Resisting Coefficient

All the tests results from the test rig can be represented by diagrams with an appearance in principle similar to the curve in figure A.1. The upper curve shows the pulling force in the cable as a function of the position when the cable is deployed through the eyes and the lower curve shows the force as a function of the position when the cable is pulled back again through the eyes (restored). In fact the voltage from the force gauge is shown but the gauge has a almost linear force-voltage characteristic in the measuring range. So the voltage can easily be scaled to a force. The span between these two curves shows the difference in pulling force in the cable at deployment respective restoration. This span can be used to calculate a resisting coefficient. Somewhere between the two cures we can draw a horizontal line that represents the gravitational force acting on the weight that hangs in the cable. The difference between this level and the lower curve is the force that works against the deployment.

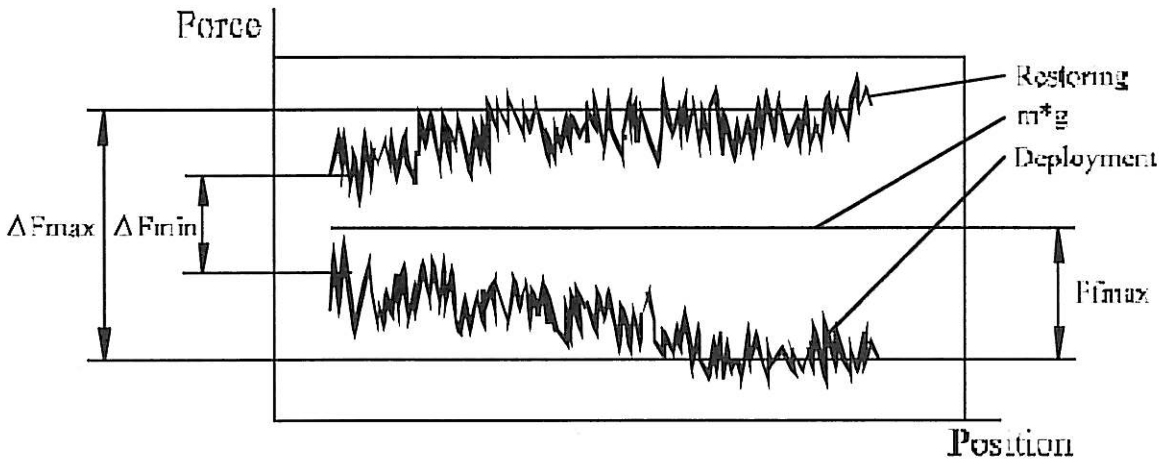


Figure A.1. A typical plot from a friction test.

The interesting thing in Figure A.1 is as mentioned above the span between the deployment curve and the line for the gravitational force, F_{fmax}. This difference shows the size of the force that resists the deployment. A problem is that it is difficult to determine the position of the mg-line in the diagram in some tests. The force gauge has a zero point drift. At low temperatures the zero point drifts and it is not possible to calibrate the force gauge in the climate chamber. In other words: the zero point of the voltage from the force gauge does not coincide with the zero point of the measured force. The amplification factor of the force gauge is not affected by the temperature, so relative changes in the measured force are shown correctly in the graphs. These circumstances leads us to calculate the resisting coefficient in the following way:

The curves for deployment and restoring appear to be quite symmetric around the mg-line. Therefore it is a good approximation to write:

$$F_{f\max} = \frac{\Delta F_{\max}}{2} \qquad \text{expression A.1}$$

We calculate a resisting coefficient μ_{max} in a way analogous to the normal friction coefficient:

$$\mu_{\max} = \frac{F_{f\max}}{mg} \qquad \text{expression A.2}$$

Expression A.1 and expression A.2 yield:

$$\mu_{\max} = \frac{\Delta F_{\max}}{2mg} \qquad \text{expression A.3}$$

In analogy we get:

$$\mu_{\min} = \frac{\Delta F_{\min}}{2mg} \qquad \text{expression A.4}$$

APPENDIX B

Preliminary Motor/Gear Head Selection

This is an outline to selection of motor/gear head combination and not a definite choice. More studies and tests must be done to qualify a drive system for the flight unit.

The torque load on the gear head shaft is caused by the friction in the belt transmission. This friction is very difficult to determine, the conditions in space are complicated to predict and this does not make the determination easier. We have to trust tests and measurements on prototypes. The first test is a simple force-measurement on the demonstration model. This model has not the authentic pulley wheel bearings and it has not even the belt strainers but these force measurements at least give a picture of the order of magnitude of the friction.

The measurement of the torque required to drive the belt transmission is done with a simple dynamometer. This is connected to the belt and the force is applied parallel to the belt. The force that brings the belt in motion is measured. The tension force of the belt is a parameter of vital importance for the torque needed to drive the belt. A fair value of this tension force is 75 N. This tension gives the measured necessary force (F) for driving the belt: 12.5 N. By multiplying the force with 2 we get an estimate of the necessary torque for driving the belt transmission with the folding mechanisms and belt strainers mounted. This makes F=25 N.

The pitch diameter of the pulley wheel is 41.38 mm so the necessary torque is:

$$M = 25\frac{41.38}{2} = 517\,mNm$$

Another important parameter is the rotational speed of the pulley wheels. A suitable belt speed is 5 mm/s, this gives the rotational speed (n) 2.3 r.p.m.

Selection of Gear Head

The gear head should reduce the rotational speed of the motor, approx. 3000 r.p.m. to the rotational speed of the pulley wheel, 2.3 r.p.m. (the rotational speed of 3000 r.p.m. is a typical nominal speed for a small stepper motor). This gives the gear ratio $\approx$1300:1.

The second parameter for selecting gear head is the output torque, as estimated above. An additional safety factor of 5 has been applied (the factor 2 used above is not really a safety factor, it is a factor that helps us to estimate the actual torque). This gives the output torque on the gear head shaft: M=2585 Nmm.

The manufacturer of commercial gear motors; Minimotor AS has a big gap in their product range just at this output torque: between type 23/1 with 700 Nmm and type 30/1 with 4.5 Nm. Select: type 30/1. This is available with a reduction ratio 1526:1 and is available for vacuum usage as option. The gear head 30/1 is a five stage planet type gearbox and has a specified efficiency of 55%. Unfortunately it is manufactured in magnetic steel but maybe it can be modified e.g. with housing in stainless steel. The gear head has in tests shown a tendency to jam because the lubrication in the gearbox becomes to viscous at low temperatures. In cold temperatures the gear head efficiency will decrease, down to zero at about -30°C when it is totally jammed. If the gear head 30/1 will be used in the flight unit the standard lubrication can be substituted by the Dupont

Krytox grease, which is a lubrication with a well documented usage in gearboxes in spacecrafts [ref.4].

Selection of Stepper Motor

With gear head 30/1 we get the motor parameters:

speed:

$$n = 2.3 \cdot 1526 = 3510\, r.p.m.$$

torque:

$$M = \frac{2585}{1526} \cdot \frac{1}{0.55} = 3.08\, mNm$$

(gear head efficiency = 55%)

The stepper motor manufacturer Escap, Switzerland has a suitable stepper motor with these characteristics. It is the P 110 motor. With a drive circuit called P110-064-2.5 it generates the output torque 5 Nmm at 3000 r.p.m.

Result :From a mechanical point of view a suitable motor-gear head combination would be the Minimotor AS gear head 30/1 with gear ratio 1526:1 and the Escap stepper motor P 110-064-2.5. The discussion above is a guideline for motor package selection. The final selection taking into account vibration, vacuum etc. is outside the scope of this degree project.

APPENDIX C

Preparations for Analysis of Oscillations in the Boom Wires

Oscillations in the boom wires are initiated when the booms are deployed and the probes are moving out from the satellite center. This oscillation is in the rotation plane and oscillations in the rotation plane is a bigger problem than oscillations out of the rotation plane. For out of plane motions, the centrifugal force on the probe is a vector that remains parallel to the spin plane as the wire oscillates. However, for in plane motions the centrifugal force vector changes direction as the wire oscillates, such that it is always pointed radial outward. This makes the force that bring back the boom in its state of equilibrium less in the in plane motion and this motion is more unstable.

The oscillations have to be damped and this damping is done by the internal damping of the wire booms and by the nutation dampers that are placed in the platform. The wire boom damping may be related to inelastic flexure in the cables. This damping mainly takes place in the root attachment point of the cable because the bending moment, and hence the bending, is greatest there.

It has to be determined is if the internal damping of the wire booms is sufficient to moderate initiated oscillations. Such an analysis has to be done to be sure that nothing unexpected happens during and after the deployment. In this appendix I suggest a method to analyze the damping coefficient in the cables, I also outline a method to derive the equation of motion for the wire booms.

Method to Experimentally Estimate the Damping Coefficient of the Cable

A test arrangement consisting of a physical pendulum is used. The physical pendulum is a weight that hangs in a piece of boom cable. The damping tests should be performed with wire tensions and deflection angles approximating flight conditions to avoid ambiguities or the need for extrapolation. The tangential velocity is probably so high that aerodynamic drag on the wire and mass is significant, such that meaningful tests should be performed in a vacuum chamber. As the damping is greatest at the root attachment point of the wire the tests do not have to be done with full length wires, maybe with wire length 0.5-1m.

For a pendulum of length (L) having tip mass (m), and neglecting the wire mass, the oscillation energy may be established by the change in potential energy (m g z) associated with the extreme half angle excursion (ϕ) [ref.8].

$$U = mgL(1-\cos\Phi)$$

Comparing two successive swings, the change in energy may be related to the respective maximum amplitudes. Define: $d\phi_n=\phi_n-\phi_{n+1}$

$$dU_n = mgL\left[(1-\cos\Phi_n)-(1-\cos\Phi_{n+1})\right]$$

$$dU_n = mgL\left[\cos(\Phi_n - d\Phi_n)-\cos\Phi_n\right]$$

Invoking a trigonometric identity:

$$\frac{dU}{dn} = CmgL^2\ dUn = mgL\left[2\sin\frac{1}{2}\left(\left(\Phi_n - d\Phi_n\right) + \Phi_n\right)\sin\frac{1}{2}\left(\Phi_n - \left(\Phi_n - d\Phi_n\right)\right)\right]$$

$$dU_n = mgL\left[2\sin\left(\Phi_n - \frac{1}{2}d\Phi_n\right)\sin\left(\frac{1}{2}d\Phi_n\right)\right] \approx mgL\Phi d\Phi$$

Using a small angle approximation and neglecting the second term:

$$\frac{dU}{d\Phi} = mgL\Phi$$

The logarithmic decrement method can be used here to evaluate experimental data. This implies a relation of the form:

$$\frac{d\Phi}{dn} = C\Phi$$

Invoking the chain rule provides an expression for the energy dissipated per cycle as a function of the amplitude:

$$\frac{dU}{dn} = \frac{dU}{d\Phi}\frac{d\Phi}{dn} = (mgL)(C\Phi)$$

$$\frac{dU}{dn} = CmgL^2$$

To experimentally estimate C we can bring the pendulum in motion and measure the amplitude as function of the number of cumulative cycles.

The Equation of Motion for a Wire Boom

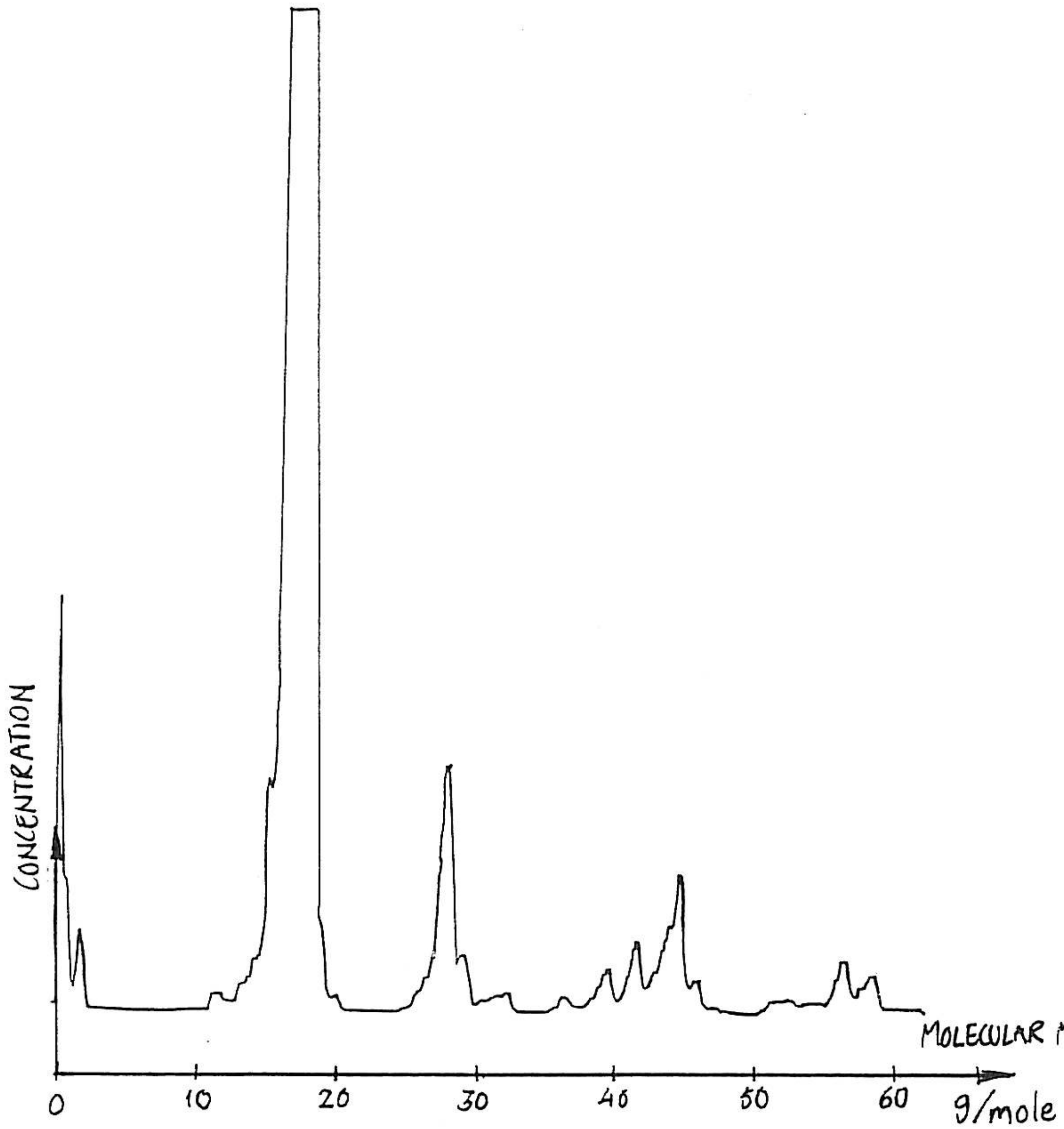
The motion of an oscillating boom cable is not described by the equation for a physical pendulum. Firstly the force field is not parallel as the gravitational force field, secondly the wire boom has no fix suspension because if a wire boom starts to oscillate the satellite platform starts to oscillate too with the same period but with its phase shifted 180° because of the rule of the constancy of the moment of momentum. Thirdly, the distributed mass in the wire boom affects the motion. The four wire booms have the same natural frequency and they can oscillate in the same phase or in different phases. The most common mode is perhaps the one when all four booms are oscillating in the same phase. This mode is exited by spin/despin maneuvers of the satellite and by the Coriolis force caused by simultaneous deployment.

To start with the equation of motion for one wire boom has to be derived. To get a complete description of the motion it has to be combined with three similar equations and an equation of motion for the satellite platform. This gives five coupled parameters: the four pendulum angles for the booms plus the angular position for the satellite platform but the interesting mode is probably the one mentioned above, when all four booms are oscillating in the same phase. This system can be solved with the numerical value of C inserted. The decay of the oscillation after an initial excitation can be studied in a simulation and then it has to be determined if the damping is sufficient.

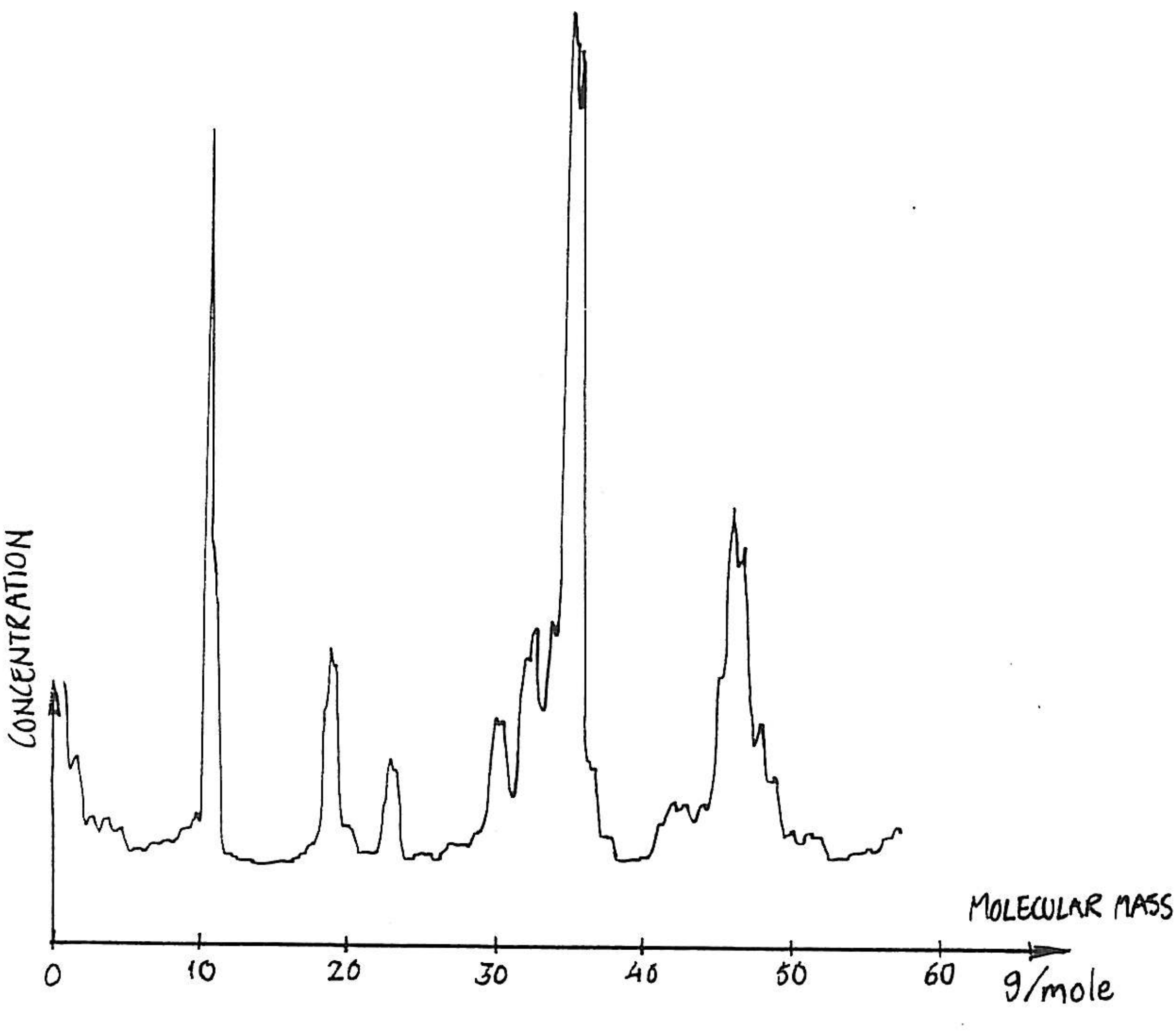
APPENDIX D

Mass Spectrograms of the Emitted Gases from the Belt

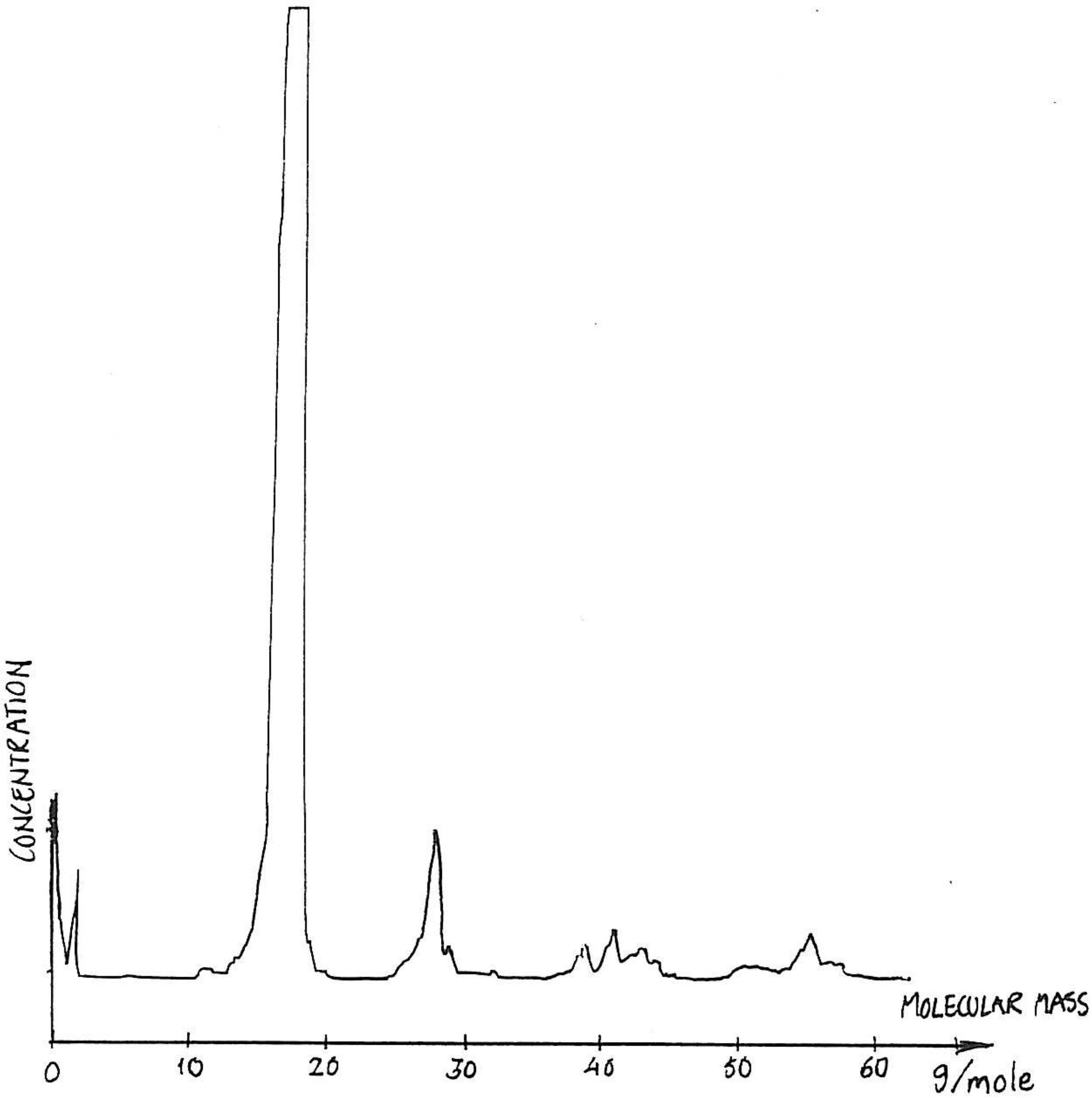
Mass distribution of evacuated gases, empty vacuum chamber.



Mass distribution of evacuated gases after 30 minutes vacuum pumping of the toothed belt.



Mass distribution of evacuated gases after 16 minutes vacuum pumping of the toothed belt.



APPENDIX E

Calculations of the Thermal Expansion of the Belt and Geometry of the Belt Strainers

The temperature range that the belt transmission has to work in is between -40°C and +40°C. This is an estimation that has been made after discussions with Staffan Persson at The Swedish Space Corporation. The changes of length of the belt due to these differences in temperature is calculated and a suitable geometry of the four belt strainers is defined here.

The thermal expansion of the belt is considered to be ruled by the thermal expansion of the kevlar reinforcement. The thermal expansion factor of the kevlar cord is specified by the manufacturer to be 70-150*10^{-6} [K^{-1}]. Measurements on a belt identical to the one in the flight unit has showed that the actual value is about 7*10^{-6} [K^{-1}]. This is a bit surprising but I trust the measured value and in this calculation the value 20*10^{-6} [K^{-1}] is used to be on the safe side. The satellite platform is considered as uninfluenced by the temperature changes. This assumptions make the calculations conservative.

The total change of length in the temperature range is:

$$\Delta L^{-} = L\ \alpha\ \Delta T \approx 1500 \cdot 20 \cdot 10^{-6} \cdot 80 = 2.4\,mm$$

The deployment mechanism is mounted at appprox. 20°C, so the belt has to be allowed to be shortened:

$$\Delta L^{-} = L\ \alpha\ \Delta T^{-} \approx 1500 \cdot 20 \cdot 10^{-6} \cdot 60 = 1.8\,mm$$

The belt strainer was designed in a early stage in the design process. This was before the actual thermal expansion factor had been measured. The design parameter was a total length difference of 18 mm. This length difference was supposed to be compensated with four belt strainers, each strainer able to compensate 4.5 mm length difference. As the actual length difference is about 2.4 mm one of these belt strainers is enough to compensate the length difference.

The available space for the belt strainer is very limited, the distance between the belt and the retracted solar-panel is about 13 mm. This distance is not enough to compensate for 4.5 mm change of length, so the belt strainer has to intrude on the solar-panels. After trial and error a suitable geometry of the belt strainer has been found (see figure E.1). Its allows a length change of approximate 4.7 mm.

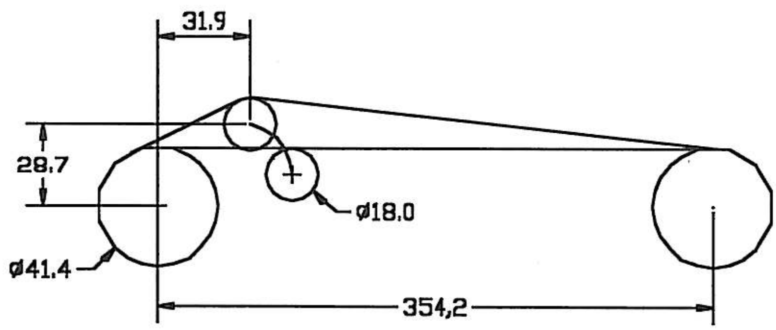


Figure E-1. The geometry of the belt strainer.

Dimensioning the Belt Strainer Spring

The characteristics of the torsion spring in the belt strainer is quite difficult to determine. A suitable belt tension is between 50 N and 100 N, this has been found by tests on the prototype. It is of course good if the tension is relatively constant over the stroke of the belt strainer. The spring in the belt strainer can be prestressed to achieve a small relative variation in belt tension over the strainer stroke.

There is a relationship between the spring torque, the stroke angle of the belt tighter and the belt tension. This relationship is used in the graph in figure E-2, where the required spring torque to obtain the tension 50 N is shown as a function of the stroke angle. The geometry of the belt tighter mechanism is of course very important in this relationship, the geometry used is the one shown in figure E-1. The stroke angle is measured from a position where the belt tighter is fully retracted. The ideal belt tension of 50N over the whole stroke requires the spring characteristic shown in figure E-2. This characteristic is impossible to achieve with a linear spring but two compromises are shown in figure E-2, those are linear characteristics of prestressed springs. The two compromises are torsion springs with wire diameter 1.5 mm respectively 1.8 mm, inside diameter 8 mm and material SS 2570.

The difference between the ideal spring characteristic and the linear ones cause a non constant belt tension. This is shown in figure E-3. The belt tension varies between 40 and 170 N for the 1.5 mm spring and between 30 and 300 N for the 1.8 mm spring. Hence the desired tension was 50 N the best choice must be the 1.5 mm spring.

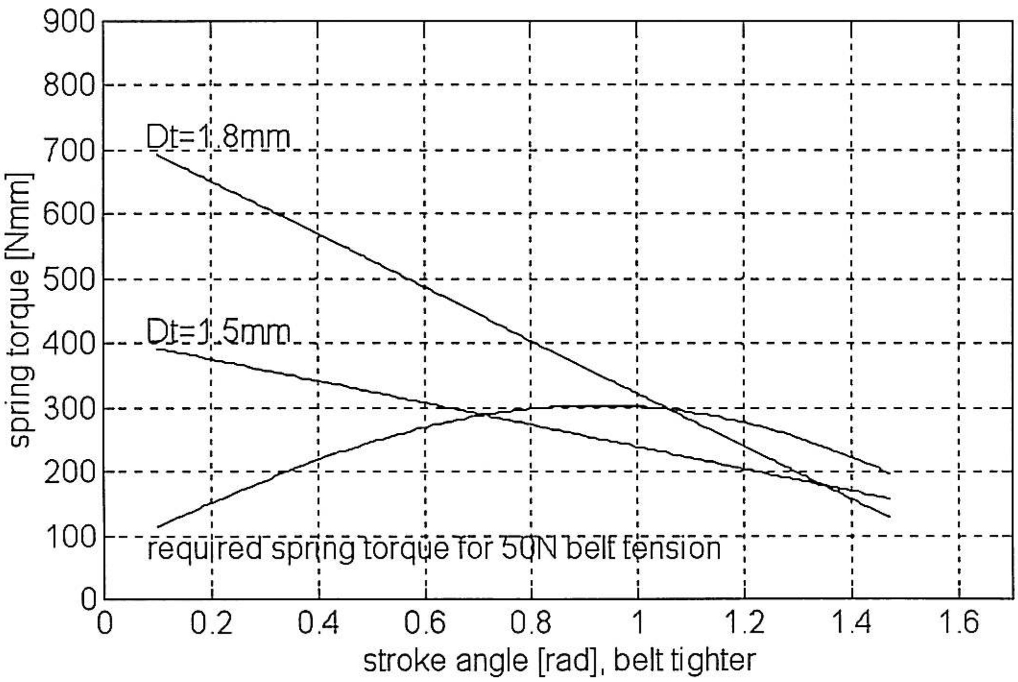


Figure E-2. Required spring torque for 50N belt tension and spring torque for two linear springs.

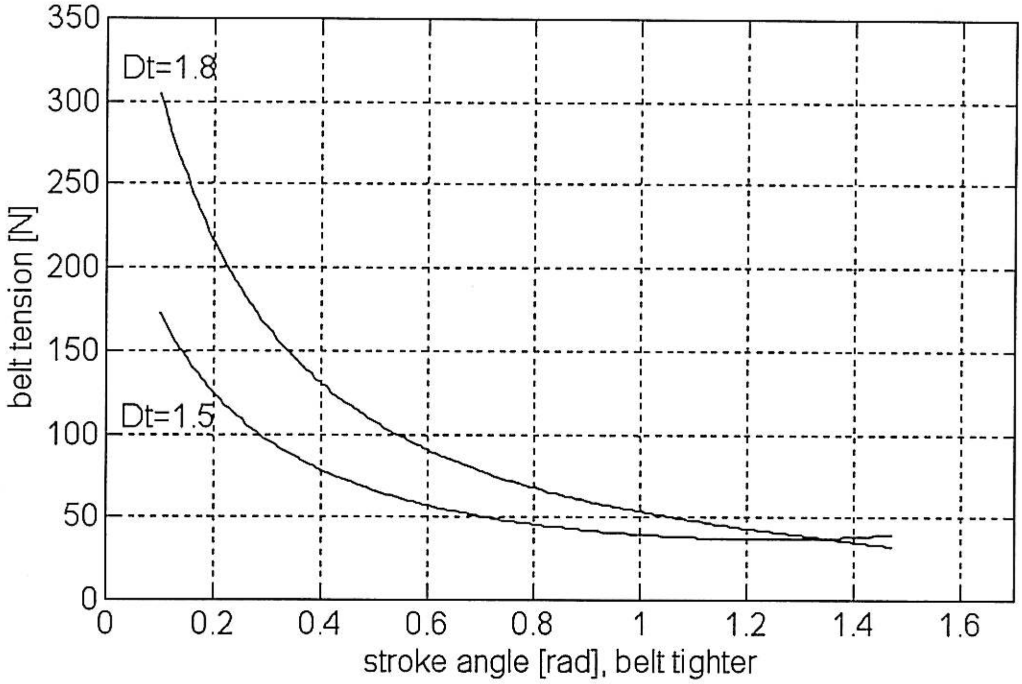


Figure E-3. The belt tension caused by two different torsion springs.

APPENDIX F

Component Specifications

Part nr.	Part name	Quantity	Material	Mat. des.	Mass
0 1	Toothed belt	1	Kevlar+polyuretane		58,5
0 All					58,5
1 01	Foldingblock	4	Aluminium	SIS 4212	10,03
1 02	Folding shaft	4	Stainless steel	SS 2343	4,51
1 03	Foldingarm	4	Stainless steel	SS 2333	15
1 04	Guide arm	8	Stainless steel	SS 2333	2,88
1 05	V-sheet	4	Stainless steel	SS 2333	1,47
1 06	Bolt foldingarm	4	Stainless steel	SS 2343	0,38
1 07	Lockingdevice foldingarm	4	Phosphor bronze		0,3
1 08	Guide pin	8	Stainless steel	SS 2343	0,43
1 09	Guide rail	4	Aluminium	SIS 4212	10,55
1 10	Giderail holder	8	Aluminium	SIS 4212	1,26
1 11	Eye holder	4	Aluminium	SIS 4007	1,12
1 12	Holder angleadjuster	4	Aluminium	SIS 4007	1,35
1 13	Angle adjust screw	4	Stainless steel	SS 2343	1,19
1 14	First eyekeeper short	2	Stainless steel	SS 2333	3,59
1 15	First eyekeeper long	2	Stainless steel	SS 2333	4,08
1 16	Support for folding mechanism	4	Aluminium	SIS 4212	2,78
1 17	Ceramic eye	8	Aluminium oxide		0,11
Bushings	BB0604DU-B	8	Bronze,PTFE+lead		0,26
	MB0205DU-B	4	Bronze,PTFE+lead		1,04
	Mass summation				253,74
	Mass estimation fasteners				29
1 All	Total mass				282,74
2 1	Wire wrap profile	8	PTFE+glass		6,04
2 2	Wire holder	8	Plastic	PVDF	4
Fasteners	M3x5	24			0,35
	M2.5x6	16			0,31
2 All	Total mass				93,68

Part nr.	Part name	Quantity	Material	Mat. des.	Mass
3 1	Short probe bracket	4	Aluminium	SIS 4212	12,39
3 2	Long probe bracket	4	Aluminium	SIS 4212	21,51
3 3	Crossbar probe bracket	4	Aluminium	SIS 4007	9,45
3 4	Probe fixation plate	8	Plastic	PVDF	0,68
3 5	Probe fixation spring, left winded	4	Stainless steel	SS 2347	1,04
3 6	Probe fixation spring, left winded	4	Stainless steel	SS2347	1,04
3 7	Probe fixation seat	8	Plastic	PVDF	0,49
3 8	Guillotine holder	4	Aluminium	SIS 4212	2,5
3 9	Guillotine	4			25
3 10	Fixation wire	8			
	Mass summation				301,08
	Mass estimation fasteners				43
3 All	Total mass				344,08
4 1	Belt tighter holder	1	Aluminium	SIS 4212	6,4
4 2	Belt tighter arm	1	Aluminium	SIS 4212	9,2
4 3	Belt tighter wheel	1	Aluminium		8,54
4 4	Thin spring belttighter	1	Stainless steel	SS 2570	3,48
4 4	Thick spring belttighter	1	Stainless steel	SS 2570	4,42
4 5	Slide bushing	1	Plastic	PTFE	0,51
4 6	Belt tighter arm shaft	1	Stainless steel	SS 2343	8,61
4 7	Belt tighter wheel shaft	1	Stainless steel	SS 2343	7
Bushings	BB0604DU-B	4	Bronze,PTFE+lead		1,04
	Mass summation				52,32
	Mass estimation fasteners				8
4 All	Total mass				60,32
5 1	Motor attachment	1	Aluminium	SIS 4212	19,9
5 2	Gearmotor	1			308
5 3	Driven pulley wheel	1			43,87
Fasteners	M4x6	4			0,6
	M3x8	4			0,43
5 All	Total mass				375,89
6 1	Post for motor	1	Aluminium	SIS 6005	140
6 2	Post without motor	3	Aluminium	SIS 6005	140
6 3	Deck with probebracket	1			
6 4	Deck without probebracket	1			
6 All	Total mass of posts				560

Part nr.	Part name	Quantity	Material	Mat. des.	Mass
7 1	Pulley wheel	3	Aluminium	SIS 4212	40
7 2	Pulley wheel flange	8	Aluminium	SIS 4212	
7 3	Pulley wheel shaft	3	Stainless steel	SS 2343	11,76
7 4	Pulley wheel axis holder	6	Aluminium	SIS 4212	9,41
7 5	Pulley wheel distance bushing	3	Aluminium	SIS 4212	0,87
Bushings	BB0608DU-B	6	Bronze,PTFE+lead		
Fasteners	M3x10	24			0,52
	M5 nut	6			1,06
7 All	Total mass				230,58
8 1	Probe	4			70
8 2	Three meter cable	4			7,5
8 All	Total mass				310
	Total sum				2315,79
	Total sum without corner posts				1755,79