Abstract

The aim of this paper is to present a methodology which was set up to develop a robust controlled flexible multi-body space system, with high precision sensors and actuators. The integration of the structural and the controller design is the key-point for such a procedure.

Special emphasis was placed on the identification of the most suitable advanced controller design methods and on the technique of how to transfer the relevant structural information about the dynamic behaviour of the mechanism into the controller algorithm design.

The appropriate tools and reduction methods were selected in order to condense in a meaningful way the structural dynamics of the mechanism represented by a finite element model, usually huge (hundreds of thousand of degree of freedom) and quite complex, down to a set of state space matrices whose size can be handled by the controller.

The effectiveness of this procedure and the performance of the designed controller were tested and verified on a dedicated breadboard.

1. Introduction

The trend in the design of next generation satellite communication systems is towards real-time services leading to an increase in the peak data rates provided by the satellites. This leads to a requirement for relatively high pointing accuracy for the devices which provide the communication such as optical terminals or high gain antennas. We can find the same trend in the development of instruments for very accurate investigations of space, such as interferometric and spectrometric systems. The order of magnitude of the phenomena which we want to investigate becomes smaller and therefore the maximum error allowable for the instrument has to be reduced.

To correct the errors in the pointing of these mechanisms implies that they have to work in a higher range of frequencies. The controllers in use for spacecraft systems and subsystems usually work in a frequency range below the natural frequencies of the structure. If on one side the controller of i.e. a solar arrays system works typically in the range below 10Hz, on the other side for an optical terminal the range is in the kHz region. The wide operating frequencies range for such new generation of pointing systems therefore interacts with a large number of structural resonances.

For applications where a high performance of the controlled structure is required in combination of a significant overlap with the structural dynamics, the prefixed targets can only be achieved considering a unique integrated system. The design aspects of such integrated system were investigated at Contraves Space in the frame of a dedicated project and the main findings are presented in this paper.

2. ISCS Project

The Integrated Structure and Controller System Project started in Contraves Space in April 98 as internal research and development program. The project was carried out in a close collaboration with the Institut für Mess- und Regeltechnik of the Swiss Federal Institute of Technology of Zürich.

The team working on the project consisted Contraves Space staff members and students carrying out diploma works, supported by the Institute, namely by Prof. Dr. H. P. Geering, head of the above mentioned Institut für Mess- und Regeltechnik, and his collaborators.

The background for the project was the Contraves Space development of a high performance pointing system in the frame of the design of optical inter-satellite link terminals. The need of developing a pointing system with high pointing requirement, and a controller for maintaining the direction of the out-going laser beam within µrad tolerance and for correcting the pointing errors of the mechanism due to different kind of disturbances, led the team to identify risk areas on one side and the mitigation actions on the other.

The fact that for correcting pointing errors the controller has to work in a higher frequency range implies that the controller bandwidth interacts with a large number of structural resonances. Therefore the traditional approach of de-coupling structural dynamics and controller operating range and to design the mechanism and the
controller almost independently (see Figure 2.1) seemed to be not the most suitable one for the kind of application, even if the best known.

The baseline for the optical inter-satellite link terminal development was the traditional approach: the controller is a PI one with pre-tuned parameters where the mechanism structure is assumed to be rigid in the bandwidth of the controller.

The ISCS program was initiated with the primary aim of defining a strategy to develop a robust controlled flexible multi-body space system with high precision sensors and actuators. The key point of the approach we wanted to follow was to consider the controller, the structure, the sensors and the actuator as part of a unique integrated system whose performance are not the sum of the performance of each component, but the result of their best integration in a robust global system (see Figure 2.2).

The design of the structure and of the controller are therefore not any longer almost independent but the mechanical design and modelling have to be carried out taking into account the requirements of controllability of the system under development, and the controller is designed based on the dynamic behaviour of the structure.

The following points have been addressed in the frame of the ISCS project:

- Investigation about the state of the art in the field, with particular regard to structural modelling, controller design methods, characteristics of actuator and sensors.
- How to introduce in an efficient way information about the structural behaviour in the controller design.
- To identify the controller design method more suitable to handle the structural flexibility.
- Breadboard definition of a flexible multi-body structure and realisation of a controller according to the selected procedure.
- Integration of a system model and performance prediction.
- Verification of the performance with the hardware.

3. Flexibility and Controllability

From the structural point of view, a pointing system is a multi-body system which consists of joints and structural parts which are most of the time very lightweighted and therefore also flexible.

The usual approach for design such a mechanism is to fulfil requirements like mass, first eigenfrequency, pointing range, rate, acceleration and accuracy, but usually we don’t have requirements about mode shapes and interval between resonances.

To characterise the dynamic behaviour of a structure, the finite element modelling technique is the most widely used. With computational power nowadays available it is possible to generate and solve very detailed models with hundreds of thousands degrees of freedom. Finite element modelling is a standard procedure but to use the output of a FEM as input for the controller design the following aspects have to be carefully addressed:

- The accuracy of the model parameters like eigenfrequencies, damping ratio and mode shapes. In the case of a multi-body system it is heavily dependent on the joints modelling.
- How to reduce the size of the mathematical model to an order which can be handled by the controller. The order allowable for the controller is often restricted by the computational power, and this is in particular true for space applications.

The design of the controller is one of the most important element to achieve the demanded accuracy out of a pointing system. In order to understand the difficulties we have to deal with when we are considering a flexible system instead of a rigid one, the following points have to be considered:

- To measure directly the necessary output signal for the feedback control is not always possible in case of a flexible structure. Indeed, in a pointing mechanism the number of sensors is in general limited and their location in most of the cases is not the most appropriate for giving directly the required information. If we are interested in the pointing
error for instance, and we cannot put a sensor on the pointing device, the structural flexibility between the location of the sensor and the pointing device that we want to control might be the source of non appropriate correcting actions.

- To be able to make corrections the position of the actuators and the deformation in the resonances have to be compatible.
- The eigenmodes are often closely spaced which makes difficult to distinguish one from the other;
- The damping ratio in the case of a multi-body system is usually small and that is source of instability;
- The already mentioned inaccuracy of the modal parameters.
- The dynamics of a multi-body system might change for different configurations.

4. Selection of the controller design method

In controlling dynamic systems the well known PID-controller is widely used in industry, and tuning PID controllers is one of the most common tasks for control engineers. For SISO systems a satisfactory control can be achieved by established tuning rules. This method can be applied to MIMO systems but it results less effective because the number of tuning parameters increases.

Due to increasing demand of flexible structure with high accuracy, not only for space application, consistent research activities have been carried out. The result is that, because of the complexity of this kind of systems and of the amount of information related with the flexibility to be accounted, multivariable controller design methods are preferred [1], [2].

Controller design methods such as robust controller, model predictive controller, fuzzy logic controller have been widely developed in the past decades but their application in space has fund a limiting factor in the restriction on computational power availability.

Having reviewed the different above mentioned design methods and their range of applicability, the robust design methods was judged the most suitable for the application. A robust controller presents the advantage of guaranteeing a certain insensitivity to uncertainties on the parameters of the mathematical model of the to be controlled system.

The objective of a multivariable feedback controller is to achieve specific aims for the controller plant, such as good reference tracking or disturbance rejection. Within robust controller design methods the desired performance and robustness of the system are achieved shaping specified transfer function derived from the mathematical model of the plant (i.e. complementary sensitivity $T$, sensitivity $S$). In comparison to classical control methods like PID, modern control design such $H_\infty$ or $H_\infty$ enable the engineer to shape closed loop transfer functions. Unfortunately different objectives require different closed-loop functions to be shaped and these functions most of the time can be shaped independently. This means that different objectives cannot be met simultaneously. Moreover the functions cannot be shaped in the same way over the whole frequency domain as, for instance, $\lim_{s \to \infty} S(s) = 1$ and $\lim_{s \to \infty} T(s)=0$. The feedback design is always a trade-off of conflicting requirements over different frequency ranges. At low frequencies, disturbance rejection and tracking are important objectives therefore the sensitivity $S$ should be small. At high frequencies, complementary sensitivity has to roll fast enough in order to guarantee sufficient noise attenuation as well as robustness against multiplicative perturbations.

5. Structural inputs for the controller design

Transfer functions are a common way to represent the dynamic behaviour of a modelled system, to be used for the controller design. The required number of transfer functions increases rapidly with the number of actuators and sensors present in the system. In addition, one set of transfer functions for each configuration of the multi-body system have to be derived. The handling of this amount of data results therefore difficult.

Multivariable systems are often described by the state space matrices and the multivariable controller design methods are based on such a representation. Therefore a method to efficiently represent the structural information in state variable matrices $A$, $B$, $C$ and $D$ has been searched for.

The reduction the information of finite element model with hundreds of thousand of degrees of freedom in a limited number of state variables maintaining the accuracy of the information was carefully addressed in the frame of the ISCS project. A dedicated software, ADAMS [3], was selected in order to generate the state matrices out of the FEM model.

A first step towards the reduction of the order of the model takes place while importing the FEM into the ADAMS Flex tool [3]. Each body of the system is reduced using the Craig Bampton method [4] and calculating the mass and stiffness matrices for the DOFs of the boundary points. The minimum set of boundary nodes consists of:
- connecting points in the flexible multi-body system;
- points where the sensors are located;
- points where actuators are located.

In addition, a number of generalised modes are selected to calculate the modal matrix. The number of modes has to be chosen to not exceed a fixed maximum error between the value of the natural frequencies of the assembled reduced and not reduced models.

Once the Craig Bampton reduced models of each body are available, they can be imported into the ADAMS
Flex tool and the joints connecting the bodies can be defined. The dynamics for the assembled multi-body flexible system can be re-calculated and compared with the original FEM one. The set of input variables, typically the actuator forces, and of the output ones such as displacements of the sensors positions, can be defined and the set of state space matrices for the system being linearised around a certain operating point are derived. The state variables, selected by the software, are used to describe the kinematic degrees of freedom (displacement and velocities) and to represent modal degrees of freedom (modal generalised coordinates and first time derivatives of the modal generalised coordinates).

In ADAMS it is possible to further reduce the size of the output state space matrices disabling static modes of higher order which do not effect significantly the accuracy of the dynamics in the range we want to design the controller for. In addition modal reduction algorithms based on energy reduction methods, deformation and shape methods are available. An advantageous feature of this software is the possibility of defining the kinematics for the bodies of the system and automatically generating the state space matrices for defined operating points.

6. The ISCS breadboard

The ISCS project was intended to set-up and test a procedure for designing controllers operating in a bandwidth which overlaps the range of the structural eigenfrequencies of a multi-body flexible system. Therefore a representative breadboard was designed in order to:
- integrate the design of the structure and of the controller;
- carefully address the modelling aspects;
- design different controllers with the main purpose of:
  - disturbance rejection
  - tracking of reference signals up to 100 Hz;
- investigate how to reduce the structural model to be taken into account in the controller design;
- integrate a system model for performance prediction.
- Evaluate the robustness of the designed controller with regard to unbalance of the mass distribution and to eigenfrequency uncertainty.

7. Mechanical design of the breadboard

The design of the structural breadboard was driven by the following aspects:
- to have a two axis multi-body pointing system;
- flexibility of the structure to have many resonances below 100 Hz;
- no closely spaced eigenmodes;
- clearly defined uncoupled mode shapes;
- flexibility concentrated in one part of the system to have a good understanding of the behaviour and to be able to exaggerate the problems;
- possibility to change the position of the centre of gravity;
- to re-use components already available from previous optical terminal breadboards (i.e. the actuators).

The designed breadboard, presented in Figure 7.1, has two rotating axes. The angular position of each axis is measured by an encoder. An optical measurement realised by the means of a Position Sensitive Detector (PSD) mounted at the end of the flexible arm allows to measure the position of a selected point with regard to an external reference. A telescope was designed to achieve a PSD field of view of \( \pm 1° \).

The selected sensors were:
- Position Sensitive Detector 2L4 device from SiTek with accuracy of 3 per mill. Additional errors of up to 1 per mill. are due to the sampling.
- Optical incremental encoders from Heidenhain, with a resolution of about 350 \( \mu \)rad. With a simple 4-fold interpolation a resolution of 87.3 \( \mu \)rad has been achieved.

The two axes are actuated by two brushless DC torque motors from ETEL. The mean torque of respectively 0.2 Nm and 0.04 Nm is available for the azimuth and the elevation axis and a peak torque of a factor 3 higher than the mean value can be delivered.

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8. Structural modelling

The dynamic behaviour of the multi-body system was calculated by finite element analysis using the NASTRAN 70 solver. The finite element model of the breadboard includes the jig on which the breadboard is mounted for testing. This jig is rather flexible in order to be able to apply vibration to the upper plate by the means of a shaker. The finite element model is not further detailed as its development followed standard procedures. The calculated non-zero eigenmodes of the system up to 100 Hz are presented in Table 8.1 and the mode shapes in Figure 8.1.
In a second step the three main bodies constituting the system were reduced using the Craig-Bampton technique. A significant reduction of the degrees of freedom took place. Around 160,000 physical DOFs were reduced to 102 physical DOFs and 45 modal coordinates.

The reduced models of the three bodies were imported in ADAMS and the multi-body system re-built by defining the joints. The result of the dynamic analysis carried out in ADAMS Flex once the model has been re-built is presented in Table 8.1. The error introduced with the described procedure is less than 2% for the first 7 eigenmodes.

The modelling flow is shown in Figure 8.2.

With ADAMS three different state space models, a rigid, a flexible and a disturbance input model, were generated for simulation and controller development.

The rigid body model consists of two input variables (torque forces), four output variables (the two encoder signals and the two PSD displacements) and four state variables (the azimuth and elevation angles and angular velocities). This model was mainly used to check the modelling of the system with time dependent response for the model variables.

The flexible-body model consists of the same two inputs and four outputs as the rigid model but the dynamic behaviour of the plant and the flexibility of the parts are taken into account. The state vectors consist of 54 elements. Four states are used to describe the two kinematics degrees of freedom. The other 50 states represent 25 modal degrees of freedom. The number of states unfortunately is too big and it would require too much computational power. Therefore the flexible model has been reduced to a 22 states model using the method of balanced residualisation. The singular values plot of the reduced model presented in Figure 8.3 shows that an accurate description of the plant up to the frequency bandwidth of interest is available.

The disturbance input model consists of three input variables, four output variables and 54 state variables.
This model has an extra input variable to take into account an external disturbance applied to the mounting plate, as shown in Figure 8.4.

![Figure 8.4 - Working point of the external disturbance](image)

### 9. Controller design aspects and robust controller

The controller was designed for:
- Rejecting external disturbances applied to the mounting plate. The controller must be able to reject those disturbances which excite the various eigenfrequencies. The controller task is to damp the influence of vibration on the PSD signal.
- Tracking a reference signal up to 100 Hz.

The design of the controller went through several steps. The first controller designed only the PSD signals were fed-back and the encoder signals, which might be different from the PSD ones because of the flexibility, were ignored. The main disadvantage of that controller is that without the encoder position information some states cannot be observed and therefore certain eigenmodes are not accounted for. The controller was designed using a GS/T weighting [5] scheme as shown in Figure 9.1.

![Figure 9.1 - GS/T scheme](image)

Two simulation results are presented in Figure 9.2 - 9.3. The first is the response to a reference signal step of 1 mrad for both PSD signals. The PSD follows the reference signal but the PSD azimuth has an overshoot of 67% and a 5% settling time of 3s. The PSD elevation has an overshoot of 20% and 5% settling time of 2.9s.

In the second simulation a sinus of 14Hz as reference input was used in order to determine if the controlled system can follow a reference input at resonance frequency. The azimuth PSD angle follows the reference signal almost perfectly. The PSD elevation angle has a phase shift of 3 degree and an amplitude overshoot of 15%. By examining the azimuth plots of the encoder position and PSD angle, it can be concluded that the controller takes into account the information on the dynamics of the system.

![Figure 9.2 – Simulation results with steps of 1 mrad as reference input.](image)

![Figure 9.3 – Simulation results for reference tracking of a sine wave](image)

In a second step the four signals, PSD and encoders, were used in the feed-back to have all the eigenmodes observable. The encoder acceleration is controlled rather than the position, but instead of using a true derivative of the encoders signals a quadratic lead compensation with two zeros at the origin is included in the feed-back. The result is a feed-back that passes high frequency transients but washes out steady state and low frequency signals. In that way the amplification of
noise is avoided because signals with high frequency are put through unchanged.

The simulation performed as with the previous controller shows that:
- In the case of 1 mrad input step, the PSD azimuth has an overshoot reduced to 38% and a 5% settling time of 1.9 s. The elevation PSD has an overshoot of 42% and a 5% settling time of 1.75 s.
- In case of sinusoidal input, the PSD azimuth follows perfectly the reference input and the elevation angle has a reduced phase shift and an overshoot of 40%.

The main goal of the controller is to keep a specific PSD position even when the structure is excited by external vibrations. The encoder therefore should take any position to achieve the desired PSD output. For decoupling the feed-back of the PSD and the encoder signal, a two-degree-of-freedom GS/T weighting scheme was used. This weighting scheme (Figure 9.4) allows to shape individually the closed-loop transfer functions of each signal used for the feed-back. [5].

![Figure 9.4 – 2dof GS/T scheme](image)

Due to the fact that for good tracking and damping performance an ideal controller utilises the maximum actuator resources and produces commands that overshoot the saturation limits of the actuators, effects such as controller wind-up and entering limit cycles might occur. To counteract these effects an anti-wind-up controller was implemented. With the AWC the state variables of the $H_\infty$ controller are influenced in a way that the output signal decreases instead of going into saturation. The simulation with the AWC shows that the controller outputs become much smaller than without it (See Figure 9.5 and 9.6).

![Figure 9.5 – Azimuth control torque with (blue) and without AWC](image)

Due to the influence of the Anti-Windup on the state variables of the controller, the quality of the estimated states decreases and the active damping of the structure degrades. The tests with the breadboard showed the same results. Any distortion of the observer-states significantly affects the output. Because in the AW scheme the output is fed-back to the states of the controller and the saturated output signal acts on the plant, estimation mistakes are generated. To get better performances, the implemented $H_\infty$ controller is partitioned in such a way that the saturated control signal is fed-back. The controller designed according to the scheme in Figure 9.7.

![Figure 9.7 – Controller scheme](image)

A simulation was performed applying a step of 40 mrad to the reference signal: in Figure 9.8 are shown the elevation and azimuth torque resulting from the developed controller and the one without saturation control. In Figure 9.10 are shown the corresponding PSD signal outputs.

The modelling of the bearing friction was an important point addressed to have reliable prediction of the damping performance of the controller. A very simple friction model was implemented: it is assumed that the steady rolling friction torque is reached immediately without any slope and that it does not depend on the angular displacement of the bearing. By choosing such a model the friction is overestimated by some degree but we have the advantage of simplicity. The only information needed for this approach is the angular velocity, being the phenomena regulated by the following formula:

$$T_p = T_D - T_s \cdot \text{sgn} \alpha$$
where \( T_P \) is the torque acting on the plant, \( T_D \) is the torque produced by the drivers and \( T_S \) is the steady rolling friction torque. The values of the steady rolling friction torque were determined by the CABARET software [6].

The damping factors related with the first 4 eigenmodes were measured and predicted with and without the friction model. The results are shown in Figure 9.10 and 9.11 for the azimuth and the elevation axis respectively. Even with such simple model the prediction improved dramatically.

The damping factors measured and predicted with and without friction model are shown in Figure 9.10 and 9.11 for the azimuth and the elevation axis respectively. Even with such simple model the prediction improved dramatically.

The breadboard and the test set-up are shown in the Figure 10.1 in the configuration that measurements were performed.

The final model developed has an order of 26 and the achieved bandwidth on the breadboard was 60 Hz. Higher bandwidth could not be reached because of the noise on the PSD signal.
The continuous controller designed was digitised for the simulation by bilinear transformation and the sampling time of 0.0005 s. The controller was implemented on SIMULINK on the host PC and the SIMULINK scheme converted into C code by the Real Time Workshop software. On the target computer this code was then executed by MATLAB program xPC target.

Some measurement results are presented.

**Disturbance rejection.**
The disturbance rejection properties have been tested with a shaker as an external source of disturbance. In Figure 10.2 is shown the PSD angle in open and closed loop for an input disturbance at 14 Hz.

![Figure 10.2 - PSD azimuth signals with (blue) and without controller (green) for external disturbance at 14 Hz.](image)

The damping factor achieved for the first 4 eigenfrequencies have been already presented in Figure 9.10 and 9.11.

For comparing the amplitudes of the open-loop plant and of the closed-loop one, the output of a simulation where at 10 s the controller is switched on was run and shown in Figure 10.3.

![Figure 10.3 - Disturbance rejection.](image)

Reference tracking
In Figure 10.4 is shown the response to a step of 40 mrad applied to the PSD azimuth signal (upper picture) and the PSD elevation one (lower picture).

The elevation performance is very good: the overshoot is reduced to 12% and the 5% settling time is 0.2 s. The azimuth performance is more affected by the noise and the overshoot is 26%.

![Figure 10.4 - Reference tracking performance](image)

**Unbalance**
In order to evaluate the robustness of the controller measurements were performed changing the position of the counterweight. It was found out that the azimuth
PSD signal is the most effected by the change of the CoG position. Nevertheless moving the counter weight up to 10 mm did not introduce significant changes in the performance.

11. Conclusions

This paper describes how it was possible to design a robust controller for a flexible multi-body system not having to de-couple the bandwidth of the controller from the frequency range where structural resonances occur. The procedure to reduce the structural dynamic information to a set of state space matrices small enough to be handled for the controller design was addressed. The controller design steps have been discussed and the difficulties and improvements clearly identified and shown. Some of the results of the measurements performed with the breadboard and with the designed controller have been presented and they clearly show the achieved performance.

The successful achievements of this project demonstrate that by integrating the structure and the controller design the required performance of the system can be achieved. It is not necessary to have a very stiff structure, which is very penalising from the mass point of view, and it is possible to damp in a robust way external vibrations which can be coupled with the dynamics of the designed mechanism.

The aim of the project was not to design the optimum controller for the mechanism because the time frame was rather limited. The intent was to clearly identify the procedure to be implemented for designing a high performance multi-body mechanism. More dedicated robustness evaluation and μ-analysis would give a better overview on the insensitivity of the controller to changes of plant parameters.

The presented procedure has been set-up in order to be used during design phase and the breadboard has been used for verification. Standard modelling tools have been used and the suitable tools for time effective calculations have been selected. The limited availability of computational power for space application was one of the main constrain and the presented results have been achieved with a rather simplified controller.

12. Acknowledgements

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13. Acronyms

<table>
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<tr>
<th>Acronym</th>
<th>Description</th>
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<tbody>
<tr>
<td>AWC</td>
<td>Anti Wind-up Controller</td>
</tr>
<tr>
<td>CoG</td>
<td>Centre of Gravity</td>
</tr>
<tr>
<td>DOF</td>
<td>Degree of freedom</td>
</tr>
<tr>
<td>FEM</td>
<td>Finite Element Model</td>
</tr>
<tr>
<td>ISCS</td>
<td>Integrated Structure and Controller System</td>
</tr>
<tr>
<td>MIMO</td>
<td>Multi Input Multi Output</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional-Integral-Derivative</td>
</tr>
<tr>
<td>PSD</td>
<td>Position Sensitive Device</td>
</tr>
<tr>
<td>SISO</td>
<td>Single Input Single Output</td>
</tr>
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</table>

14. References

[2] Balanced Control of Flexible Structures (Deep Space Network Antenna), W. Gaworonski