Active Damping of Vibrations in Elevator Cars

A dissertation submitted to the

SWISS FEDERAL INSTITUTE OF TECHNOLOGY
ZURICH

for the degree of
Doctor of Technical Sciences

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1997
Leer - Vide - Empty
Abstract

In modern high-speed elevators, tight specifications concerning ride comfort are dictated by the market. These specifications cannot be fulfilled with the conventional passive vibration damping systems currently in use. The goal of this dissertation was to develop an active damping system to achieve a ride comfort in elevators not reached before by passive means. In comparison with previously known solutions, the solution presented in this work utilizes a reduced actuation configuration, an instrumentation concept adapted to the structural resonances of the car, and a multivariable robust controller design which maximizes the controller effectiveness and ensures system stability.

The system developed comprises eight actuators, six accelerometers, eight position sensors, a signal conditioning unit, and power electronics. It is controlled by a real-time computer utilizing a digital signal processor. A laboratory test rig was designed and constructed to allow testing the active damping system fitted to a full-scale high-speed elevator car. The unevenness of the two guide rails, which is the main source of vibrations, is emulated using eight hydraulic cylinders, so that no vertical travel of the car is necessary.

A multivariable system identification procedure in the frequency domain was developed to obtain a linear time-invariant system model with six command force inputs, six acceleration outputs, and six position outputs. For each input signal, the system is reduced to a subsystem with a single input and twelve outputs. Complex curve-fitting algorithms are used to identify the parameters of the numerator and the denominator of all 72 transfer functions. The subsystems identified are then combined to form a global multivariable model.

A first controller loop utilizes the acceleration outputs to suppress the vibrations. Due to asymmetric load distributions in the cabin, the orientation of the car with respect to the guide rails may change to the extent that for one or more actuators there may not be enough actuation distance. In this case, strong blows in these actuators are expected. This would dramatically reduce the effectiveness of the active
damping system. To correct the car orientation, a second controller loop utilizing position outputs is used.

The position measurements are taken between the car and the guide rails rather than relative to an earth-fixed coordinate system. Nulling the position error signals thus means forcing the car to follow an average line of the two rail profiles. To avoid a conflict between the acceleration and the position loops, the acceleration controller was designed to become active only in the frequency range where humans are sensitive to vibration, which starts at about 1 Hz, while the bandwidth of the position loop is limited to slightly below that value.

Using the $H_\infty$ method, the two controller loops are designed as robust multivariable controllers. The acceleration controller is designed first, followed by the position controller. The model used for the design of the position controller is that of the system already controlled by the acceleration controller. Due to the variations in payload, the acceleration controller is not allowed to invert the plant dynamics in the frequency range in which performance is achieved. For this reason, a non-inverting $H_\infty$ design using the GS/T weighting scheme was used.

The controller effectiveness was tested using rail profiles previously measured in a real building and emulated by the hydraulic excitation system at a travel speed of 6 m/s. The vibration reduction factor achieved is around 5 as compared to a passive damping system, based on a standard judgement criterion. The controller is capable of correcting the car orientation caused by asymmetric loads without reducing the effectiveness of the acceleration loop. In addition, this controller effectively counters other disturbance forces that act directly on the elevator car.

Based on the laboratory version of the active damping system developed in this work, it is possible to realize an industrial product fulfilling and even surpassing the market demands concerning ride comfort.
Kurzfassung


Eine erste Reglerschlaufe benutzt die Beschleunigungsmessungen, um die Schwingungen zu vermindern. Eine Schiefbeladung der Kabine beeinträchtigt die Orien-

Da die Positions messungen zwischen dem Aufzug und den Führungsschienen aufgenommen werden, bedeutet eine Positionsfolge, dass der Regler den Aufzug in die Mitte zwischen den zwei Führungsschienen bringt, welche für die Schwingungen verantwortlich sind. Um einen Konflikt zwischen dem Beschleunigungsregler und dem Positionsregler zu vermeiden, wird der Wirksamkeitsbereich der ersten Reglerschlaufe auf jenen Bereich beschränkt, in welchem der Mensch auf Schwingungen empfindlich ist. Dieser Bereich fängt bei 1 Hz an. Entsprechend wird die Bandbreite der Positionsschlaufe auf einen Wert knapp darunter eingestellt.


Um die Wirksamkeit des Reglers zu bestimmen, wurden echte, im Aufzugs schacht gemessene Schienenprofile mit dem hydraulischen Anregungssystem für eine Fahrtgeschwindigkeit von 6 m/s emuliert. Der Schwingungsunterdrückungs faktor basierend auf einem standardisierten Beurteilungsverfahren erreichte einen Wert von ungefähr 5 verglichen mit einem passiven System. Der Regler korrigiert eine wegen schiefer Beladung der Kabine resultierende Aufzugsneigung, ohne die Qualität der Schwingungsunterdrückungsqualität zu verschlechtern. Er ist auch wirksam gegen die anderen Störkräfte, die direkt auf den Aufzug wirken.

Es lässt sich feststellen, dass auf der Basis des entwickelten Labormusters ein industriell verwendbares Produkt realisiert werden kann, welches die Markt anforderungen bezüglich Laufruhe erfüllt und sogar übertrifft.
Acknowledgments

At first I would like to express my gratitude to Prof. Dr. Hans Peter Geering, head of the Measurement and Control Laboratory of the Swiss Federal Institute of Technology in Zurich (ETH), who agreed to supervise this dissertation and who supported this research project with all possible means. The degree of confidence and autonomy I enjoyed under his leadership gave the whole project the best chance to overcome the various difficulties. I am also very grateful to Prof. Dr. Gerhard Schweitzer, head of the Robotics Laboratory of the same school, who agreed to serve as co-examiner of this thesis. His comments and suggestions were extremely helpful.

I would also like to thank the management of the company Schindler Management Ltd. of Switzerland for its continuous support of this project. In this context I would like to thank Mr. Dietrich Wegener, Vice-Director Research and Development, Ms. Jean Smith who was responsible for the coordination with the Schindler Elevator Corporation in the USA for two years, and Mr. Josef Husmann who assisted this project from the beginning to the end. His deep understanding of the elevator car dynamics and his keen interest in the project were essential to reaching a good solution from the technical as well as the economic point of view.

It is important to note that this project was financially supported by the Committee for the Promotion of Scientific Research (KWF)\(^1\) of the Swiss Federal Department of Economics during the first three years. This support provided a good basis for the project and thus increased the chances of its success.

My deepest gratitude goes to Dr. Stephan Hepner, who helped me to initiate the project at the very beginning and who continued his support for the first two years. I am also indebted to my colleagues at the Measurement and Control Laboratory for their very splendid comradeship. I would like to thank Dr. Esfandiar Shafai for his feedback concerning system identification. My special thanks go to Dr. Urs Christen, who greatly helped me with his vision, criticism, and analytical insights.

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1. Now Committee for Technology and Innovation (KTI)
concerning controller design. For the useful discussions and the helpful remarks I would also like to thank Dr. Hans Musch, Mr. Carlos Cuéllar, Mr. Roberto Cirillo, Mr. Michael Simons, and Mr. Bernard Mettler.

I would also like to thank Mr. Georges Bammatter for his patience and support concerning the computer systems, Mr. Oskar Brachs for his contribution to the hardware of the test rig, and finally Ms. Brigitte Rohrbach for her continuous help throughout the project.
Contents

Abstract 3

Kurzfassung 5

Acknowledgments 7

1 Statement of the Problem 15

1.1 Motivation 15
1.2 High-speed elevator cars 15
1.3 Sources of vibrations in high-speed elevator cars 16
1.4 The passive damping system 19
1.5 The need for an active damping system 19
1.6 Requirements on the active system 21
1.6.1 Degrees of freedom controlled 21
1.6.2 Target performance 21
1.6.3 Constraints imposed by the construction 22
1.6.4 Safety measures 22
1.6.5 Cost limitations 22
1.7 Necessity for innovation 22
1.7.1 State of the art 22
1.7.2 New features of the system developed 24
1.8 Future potential 25

2 The Approach to the Solution 27

2.1 Preliminary investigations 27
### 2.1.1 The control problem

#### 2.1.2 Hardware availability

### 2.2 Important decisions

#### 2.2.1 Project phases

#### 2.2.2 Construction of a laboratory test rig

#### 2.2.3 Control signals

#### 2.2.4 Identification of the guide rail profiles

### 3 Concept of the Active Damping System

#### 3.1 Impact of design constraints on the system configuration

##### 3.1.1 Actuation

##### 3.1.2 The active roller guide shoe

##### 3.1.3 Accelerometer

#### 3.2 System configuration

##### 3.2.1 Actuators and sensors

##### 3.2.2 Hardware scheme

#### 3.3 Component selection for the active damping system

##### 3.3.1 Actuator

##### 3.3.2 Actuator drive amplifier

##### 3.3.3 Accelerometer

##### 3.3.4 Position sensor

##### 3.3.5 Analog signal processing

##### 3.3.6 Real-time computer

##### 3.3.7 Host computer

#### 3.4 Future development of the active damping system

### 4 Laboratory Test Rig

#### 4.1 Concept of the test rig

##### 4.1.1 Purpose

##### 4.1.2 Experimentation

##### 4.1.3 Expansion capabilities

#### 4.2 Mechanical assemblies

##### 4.2.1 The elevator car

##### 4.2.1 Elevator car suspension frame
5 System Identification 63

5.1 System configuration 63
5.1.1 The mechanical construction 63
5.1.2 Actuators and sensors 63
5.1.3 The control system 66

5.2 Mathematical modeling and identification 67
5.2.1 System dynamics 67
5.2.2 Requirements for the system model 69
5.2.3 System identification versus modeling 70

5.3 Model structure for identification 70
5.3.1 Proposed model 70
5.3.2 Physical guidelines 72
5.3.3 Effect of parameter variation 75

5.4 Identification method 75
5.4.1 Time and frequency domain identification algorithms 75
5.4.2 The complex curve fitting algorithm 76
5.4.3 Identification sequence 77

5.5 Identification experiment 78
5.5.1 The excitation signal 78
5.5.2 The excitation and measurement program 81

5.6 Identification results 83
5.6.1 The identified transfer functions 83
5.6.2 The identified model 83
5.6.3 Model validation 83

5.7 Comments and conclusion 83
6 Controller Design

6.1 Design requirements 87
6.1.1 Controller structure 87
6.1.2 The control design problem 87
6.1.3 Acceleration loop specifications 90
6.1.4 Position loop specifications 91

6.2 Control design method 92
6.2.1 Selection of the controller design method 92
6.2.2 Inverting and non-inverting $H_\infty$ designs 93
6.2.3 Algorithms for controller design and manipulation 97

6.3 Acceleration control design 97
6.3.1 The acceleration feedback loop 97
6.3.2 Weighting functions for the acceleration loop 97
6.3.3 The acceleration controller 102
6.3.4 The acceleration-controlled system 103
6.3.5 Order reduction of the acceleration controller 104

6.4 Position control design 106
6.4.1 The position feedback loop 106
6.4.2 Weighting functions for the position loop 106
6.4.3 The position controller 110
6.4.4 Order reduction of the position controller 111

6.5 Conclusions 111

7 Controller Realization and Results 113

7.1 Controller realization 113
7.1.1 Controller discretization 113
7.1.2 Real-time program 113

7.2 Criteria to judge the performance 114
7.2.1 Human sensitivity to vibrations and the K signal 114
7.2.2 Measurement processing 118
7.2.3 Permissible K values 118

7.3 Excitation for controller testing 119

7.4 Testing results of the controller 121
7.4.1 The testing sequence 121
1 Statement of the Problem

1.1 Motivation

As the worldwide competition in the market of high-speed elevators increased remarkably during the last decade, the ride quality denoted by the strength of the lateral cabin vibrations rose as an important sales argument. With its well established tradition in the construction of elevators, the company Schindler of Switzerland arrived at the conclusion that the limits of what a passive vibrations reduction system could provide were almost reached [28] and that alternatives should be evaluated.

In January 1992, Schindler Management Ltd., the Measurement and Control Laboratory of the Swiss Federal Institute of Technology in Zurich (ETH), and the Swiss Federal Committee for the Promotion of Scientific Research (KWF)\(^1\) reached an agreement to start a research and development project having as goal the design, realization, and testing of an active vibration damping system for high-speed elevator cars. The system is capable of reducing the lateral vibrations of the elevator car to target levels not reached before by the conventional passive means.

1.2 High-speed elevator cars

A high-speed elevator car has a travel speed that equals or exceeds 2.5 m/s. Typical speeds are 3.15, 4.0, 5.0, 6.3, 8 m/s, or even more. Beside the limitations of the drives, the sudden atmospheric pressure difference associated with rapid change of altitude affects the passengers and represents the limit of what the travel speed may be allowed to reach. The elevator car needs also a certain acceleration and deceleration time. Due to the limited height of a building, the maximum speed is thus only reached for short time periods.

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1. Now Committee for Technology and Innovation (KTI)
Figure 1.1 shows a high-speed elevator car in a hoistway. The car is composed of the passenger cabin fixed to a cabin frame using a set of rubber pads. The cabin frame is the part of the car connected to the traction by the suspension cables. The contact between the cabin frame and the two T-profile guide rails fixed to the hoistway is achieved by three guide rollers integrated in a mechanical construction called the roller guide shoe at each of the four corners of the frame. The guide rollers are kept in contact with the guide rails by prestressing the helical springs that press the rollers against the guide rails.

1.3 Sources of vibrations in high-speed elevator cars

The vibrations sensed by the elevator passengers are those of the cabin. Regarding the cabin as a rigid body, vibrations occur in the six degrees of freedom of its body fixed coordinate system (cf. Figure 1.1). The degrees of freedom (DOF) of the vibrations dealt with in this work are the two translational DOF along the $x_c$ and $y_c$ directions as well as the three rotational DOF $\phi_{x_c}$, $\phi_{y_c}$ and $\phi_{z_c}$ around the $x_c$, $y_c$, and $z_c$ axes, respectively. Vibrations in the direction of the elevator's travel ($z_c$ directions) have other sources and hence are of a different nature than the rest. They are not treated in this work.

The sources of vibrations in the whole elevator car are divided into high-frequency and low-frequency sources.

**High-frequency disturbance sources**

1) Unevenness of the guide rails

   As supplied by the producers, the guide rail profiles are not completely straight. Due to the inaccurate mounting of the guide rails to the hoistway walls and the different coefficients of thermal expansion of the wall and the guide rail, the resulting rail profile is even worse. The quality of the rails varies greatly from one building to another. As explained later in Chapter 7, it is reasonable to expect a rail excitation spectrum up to 10 Hz.

2) Aerodynamic forces

   The motion of the elevator car inside the hoistway results in aerodynamic forces that also affect the ride comfort. The magnitude and the frequency content of
Figure 1.1: A high-speed elevator car equipped with a passive damping system
these forces are dependent on the travel speed. They are influenced by the
dimensions and the inside shape of the hoistway, the number of cars running
and their locations, as well as the number of counter weights and their locations.
The mechanism of generation of these forces is therefore very complicated.
While there is not enough information available about their excitation spectra,
it is reasonable to assume that they are not wider than those of the rail excita-
tions.

3) Other high-frequency sources

The movement of the passengers inside the cabin and the forces exerted by the
door opening mechanism are relatively minor sources of high-frequency distur-
ances which act on the car directly.

Low-frequency disturbance forces

1) Asymmetric cabin loading

The presence of a large concentrated load inside the elevator cabin which is not
placed on the line joining the centre of gravity with the suspension point cause
moments that change the car orientation.

2) Suspension cables

The suspension cables represent another source of slowly changing forces that
act on the elevator car. The magnitudes of these forces are proportional to the
cable length and hence to the position of the car along the height of the
hoistway.

Another important classification of the disturbance sources is their point of occur-
rence. The rail disturbances affect the cabin through the roller guide shoes, while
the other sources either affect the cabin frame or the cabin directly. Accordingly,
the classification of the disturbance forces as either rail excitation forces or direct
car forces will be adopted in this work.

The rail excitation and the aerodynamic forces are the most important disturbance
sources. Slowly varying disturbances do not produce any vibrations which are
sensed by the passengers. However, since they will affect the active damping
system to be installed in the elevator car, they must be taken into consideration.
1.4 The passive damping system

As seen in Figure 1.1, the roller guide shoe is the connection between the guide rails and the cabin frame. Each of the three rollers contained in such a unit is pressed against the rail by a prestressing force in the corresponding spring. Aside from the material used to cover the rollers, these springs are the main element responsible for reducing the excitations produced by the guide rail which affect the cabin frame and hence the cabin.

The cabin is suspended inside its frame and is cushioned with rubber pads on the bottom and at its top side edges. These rubber pads serve mainly to reduce the high-frequency vibrations generated by the motion of the rollers. Additionally, they are useful as an acoustic isolation of the cabin from the frame. The number of these pads and their stiffness are determined by the mass of the cabin including the payload.

1.5 The need for an active damping system

The limits of what a passive damping system can achieve are determined by the stiffness of the springs, the number of the rubber pads and their characteristics, as well as their distribution. The rubber pads are effective against the higher frequency disturbances (roughly speaking higher than 10 Hz), while the springs are effective against lower frequency disturbances. Since the important part of the excitation spectrum lies between 1 and 10 Hz, the characteristics of the springs are the limiting factor for the damping capabilities of the passive system. Softer springs reduce the corner frequency of the passive filter they form. Their damping effect thus starts at lower frequencies which means that the passive system becomes more effective. However, softening the springs is associated with a larger side motion of the car to the extent that some parts of its structure such as the safety brakes may start to touch the guide rails. The mechanical stoppers of the springs are thus adjusted to allow just the tolerable deviation from the initial position. For this reason the effectiveness of the passive damping system is always limited and other alternatives must be considered in order to fulfill harder requirements on the vibrations level in the cabin.

With a better quality of guide rails it is possible to reduce the vibrations level of the cabin considerably. Since the accurate mounting of the guide rails to the hoistway
walls is a very labor-intensive and hence costly job, this option is not economically feasible. Also the qualified labor necessary for this job is not available everywhere. According to experience, the quality of the guide rails deteriorates after an expensive and time consuming realignment. In a period of about one year the quality of the rails returns to what it was before the realignment. The elasticity of the building where the elevator is mounted in the presence of large dynamic air loading is a source of continuous change in the rail profiles. The day-to-day and the year-to-year thermal cycles are other sources. It is therefore not practical to try to stabilize the rail profiles.

According to the studies done by Schindler Management Ltd. [28], the effectiveness of a passive system may be optimized by a better selection of the properties of the passive elements, their location, and their numbers. Constructional improvements in the roller guide shoes may bring some effect, but both of these methods are not able to introduce a substantial improvement to the ride quality regarding the required target specifications.

Another candidate damping system is a semi-active system which uses solenoid valve controlled hydraulic shock absorbers connected in parallel to the springs. A technique which is used nowadays in automobile suspension systems where hydraulic shock absorbers are the main damping element. The promised performance of such a system lies within the limits of what an optimized passive system may provide. Another problem with this system is that its cost is not necessarily lower than that of a fully active system. The components used are not cheap and almost the same cost is required for sensors and for the real-time computer as for an active system.

An active damping system utilizes an external source of energy to counteract the effect of the disturbances. It is composed of a multitude of actuators driven by a real-time computer which realizes a control algorithm in response to the inputs from a multitude of sensors distributed in the elevator car. This system changes the dynamic characteristics of the elevator car and is therefore capable of substantially improving the ride comfort.

It should be noted that the limits of what an active damping system can achieve are determined by hardware and software considerations, such as the specifications of the actuators and the sensors or the capabilities of the real-time computer. Another
important point in this context is the controllability of the elevator car as a dynamic system that might contain uncontrollable or difficult to control structural modes. This is determined by the structure of the car, the number of actuators and sensors, and their locations within the car.

1.6 Requirements on the active system

1.6.1 Degrees of freedom controlled

The active damping system must be capable of suppressing the vibrations of the cabin in all five degrees of freedom previously defined in Section 1.3. It has to reduce the total disturbance effect on the cabin regardless of the origin of its components.

1.6.2 Target performance

The present elevator cars equipped with a passive damping system suffer from horizontal vibrations of the cabin floor which are concentrated in the range between 1 and 5 Hz. However, sensed vibrations continue to exist up to 10 Hz. In the frequency range higher than 10 Hz the excitation is remarkably weaker.

According to the specifications set by Schindler Management Ltd. in [54], the acceleration magnitudes in the cabin in the frequency range from 1 to 10 Hz must be reduced by a factor of 10. Starting from 10 Hz, this factor is allowed to reduce linearly until it reaches unity at 20 Hz.

Having achieved this target, it could be concluded that the travel comfort required is assured. However, the sensitivity of human beings to vibrations is acceleration dependent only in the frequency range between 1 and 2 Hz. In the frequency range above 2 Hz, the passengers are more sensitive to the velocity [62]. Therefore, the acceleration cannot be directly taken as a measure of performance. As will be explained later in Chapter 7, a certain processing of the acceleration is required to get a more suitable performance measure for the evaluation of the ride comfort.

Another important requirement on the active system is the limitation on the allowable side motion of the car relative to the two guide rails. Knowing that the displacement of the springs utilized by the passive system is limited to a few milli-
meters (3 to 5 mm), the active system should keep the car inside these limits even for the worst case of asymmetric loading of the cabin.

1.6.3 Constraints imposed by the construction

Regarding the market potential for the active damping system, it is clear that the largest portion of the demand will be the modernization and the retrofit of existing elevators. This condition encourages the conception of the active damping system as an accessory kit suitable to be fitted to new as well as to old cars with minimum structural modification.

1.6.4 Safety measures

The main safety requirement on the active system is that its operation and its malfunctioning do not block or hinder the existing safety measures of the elevator car. This can be achieved when the active system is mounted parallel to the passive system. This way at least the performance of the passive system is guaranteed in cases of partial or full failure of the active system. For this reason the passive damping system should be kept unchanged after the installation of the active system.

1.6.5 Cost limitations

Since this work is striving from the beginning to develop a marketable product, the cost of the active system as an option must remain attractive for the customers. Respecting the margins of the market, the feasibility of the solution was a subject of many investigations that influenced the configuration of the active system as well as the selection of the components.

1.7 Necessity for innovation

1.7.1 State of the art

Upon analyzing the current stand of research and development done by other elevator producers ([56], [57], and [58]) one can see schemes for systems in elevator cars using acceleration measurements for the active damping of vibrations.
and position measurements for the correction the car orientation. However, the following remarks were observed:

Concerning the hardware:

- The actuation is done at the roller guide shoes utilizing two actuators for each degree of freedom. An actuator with a large bandwidth is responsible for the active damping and an actuator with a low bandwidth is responsible for the correction of the car orientation.

- The actuator responsible for the active damping is composed of a controlled current coil facing a permanent magnet. The gap between the coil and the magnet is variable according to the position of the roller. This leads to a non-linearity that needs to be compensated using an additional position sensor integrated in the actuator.

- Five accelerometers mounted on the cabin are used to provide the signals for active damping.

Concerning the design of the controller:

- Although the proposed systems should be applicable to all concerned degrees of freedom of the cabin, the proposed controller schemes may be only used for decentralized controller design.

- The mathematical models used for controller design regard the elevator car as a single rigid body. The structural resonances are not considered and there is no sign of model verification.

The problems with the proposed systems are summarized as follows:

- The proposed schemes with two actuator for each degree of freedom lead to great complications of the control. The nonlinear behavior of the dynamic actuator is another problem adding to this complexity. In addition, the mechanical construction of the roller guide shoe is expensive.

- As explained later in Chapter 5, mounting the accelerometers on the cabin leads to a strong limitation of the controller performance due to the elasticity between the cabin and its frame.
For such a complicated system with structural resonances the absence of model-based controller design leads to instability (Chapter 5).

Due to the strong coupling between the different degrees of freedom of the cabin it is not possible to utilize a decentralized scheme to control the car. According to experimental experience, the danger of instability of such a system is high.

The proposed controller designs contain a lot of unnecessary nonlinearities which do not allow the judgement of stability. The employment of additional filters in the various loops without an analysis of their effects endanger the system stability.

For these reasons, it was necessary to concept a new system hardware and to adopt controller design methods capable of solving the problem as explained below.

1.7.2 New features of the system developed

As may be seen in Chapters 3, 5, and 6, the solution proposed here has the following new features which allow it to achieve the required goals:

Concerning the hardware:

- The system utilizes one actuator for each degree of freedom. The actuator is capable of providing the required force for the vibration damping as well as for the correction of the car orientation. The relationship between the input and output of the actuator is linear.

- The accelerometers are mounted on the cabin frame and not on the cabin in order to increase the performance capabilities of the controller.

Concerning the design of the controller:

- Model based control for both acceleration and position loops is employed to guarantee system stability. The models are validated with measurements to ensure that they correctly represent the system dynamics. They also consider all possible coupling effects between the system inputs and outputs.

- Multivariable robust controller design methods are used to ensure the stability of the controlled system and the achievement of the required performance. Using
the $H_\infty$ controller design method allows to shape the controlled loops according to the requirements without the need of separately added filters.

1.8 Future potential

Although active vibrations damping has become a known practice in many sophisticated industrial applications like aerospace structures and advanced robotics [14] the solution presented in this work is believed to be one of the very early commercially feasible applications of this discipline to be adopted for this class of products.

The system configured makes good advantage of the latest developments in the fields of sensor technology, actuation, and digital signal processing. The controller itself is based on recent developments in automatic control theory and the design tools used were not available a decade ago.

Successful prototyping, series production, marketing, and maintenance of the active damping system will establish this product in the international market and will pave the way for other similar mass production applications.
2 The Approach to the Solution

2.1 Preliminary investigations

At the beginning of this project, a series of investigations was carried out to define the strategy to be adopted to solve the problem and to draw the guidelines for the solution [20], [21], [24], and [37]. These investigations were supported by the information supplied by Schindler Management Ltd. and were guided by the feedback received from the various component suppliers.

2.1.1 The control problem

The most essential and elementary question upon the start of the project was to understand the control problem at least in its simplest form. Starting from the mathematical modeling of the elevator car developed by Schindler Management Ltd. for the purpose of simulation [28], a simple linear model which describes the car dynamics in one degree of freedom was derived. For this model simple classical controllers with acceleration and position feedbacks were designed and simulated [20]. Based on that work, it was possible to write the specifications for the various components necessary for the realization of the control system.

As a preparation for the case of a multivariable system, other models were derived and controllers similar to those described in [20] were designed [24]. However, the complex control problem was defined after the construction of the test rig and after the identification and control experiments with the real elevator car in the laboratory.

2.1.2 Hardware availability

Parallel to the investigation of the control problem, the realization of the control system had to be ensured. The availability of the hardware components fulfilling the required specifications at a reasonable price had to be checked. The most important hardware components which needed to be searched for were:
1) Actuator

Finding a suitable actuator at a reasonable price was the first obstacle to be removed in order to give the project a good chance of success. Beside the relatively high maximum force required, the time constant of the actuator should be very small. The allowable weight of the moving part and the desired compactness imposed additional constraints. Moreover, a direct-drive actuator was preferred in order to avoid play and to guarantee fast response.

2) Accelerometer

The main difficulty with the accelerometer is its price. Inertial sensors are rather expensive components. It was thus not easy to find a low-cost sensor fulfilling the required specifications as for example a low measurement range of ±1 g. Recent developments in the area of micro-machining allowed the production of suitable accelerometers at a reasonable price.

3) Real-time computer

A model which accurately represents some parts of the structural dynamics of the car is necessarily of high order. Any model-based controller must therefore be of higher order too, requiring a rather powerful real-time computer. Although an expensive real-time computer is an acceptable solution for a laboratory version of the active system, it is essential to use a low-cost one in the case of a commercial product. The relatively new introduction of low-cost digital signal processors in the market was encouraging in this respect.

4) Other elements

The other hardware components needed for the active system such as the position sensor, the signal conditioning, or the power electronics were more or less standard devices which did not represent any special difficulty, neither in the specifications nor in the price.

2.2 Important decisions

The important decisions taken at the early and middle stages of the work will be stated here together with their justifications and consequences.
2.2.1 Project phases

This project started as a feasibility study to investigate the possibility of developing an active damping system for high-speed elevator cars. Based on the encouraging results of that study [20], it was decided to proceed with the project up to the development of a laboratory version of the active system. A prototype phase was also considered before series production could take place. The main phases of the project were thus defined as the development phase, the prototype phase, and then the production phase.

During the development phase, the active damping system was incrementally built and tested. This allowed to proceed with limited financial resources at the beginning and gave the possibility for low-cost experimental feedback. Therefore, the active system has started with a one-degree-of-freedom configuration, followed by a three-degree-of-freedom version before being expanded to the final configuration with its five DOF. The development of the system identification and controller design procedure experienced corresponding phases before reaching the final form.

2.2.2 Construction of a laboratory test rig

As it is usual for such a research and development project, the first studies done in this work concentrated on mathematical modeling, simple controller designs, and simulations. Once the decision was made to proceed towards the development of a functioning active damping system, the need for a testing facility was identified. The two possibilities for such a facility were either to test the developed active system directly on an elevator car running in a hoistway (test tower) or to build a laboratory test rig where the elevator car does not travel in the vertical direction and the guide rail profiles are emulated using vibration exciters. After a thorough investigation, the second option was favored for the following reasons:

- With a sufficient dynamic range of the excitation forces used to emulate the rail profiles in the laboratory test rig the main source of vibrations in the car is considered. An asymmetric loading of the elevator car in the test rig may be used to check the capability of the active damping system of rejecting low-frequency disturbances. The system can also be tested against high-frequency direct car disturbances. The effect of friction caused by non-rotating guide rollers is alleviated by teflon strips covering their circumference. The influence of the rotation
of the guide rollers on the system dynamics is negligible due to their relatively small masses and limited angular speeds.

- The availability of the test facility during the whole period of the work could not be guaranteed in case of a test tower due to its continuous occupation by other projects.

- Building a test rig at the Measurement and Control Laboratory would permit to take advantage of the hardware and software infrastructure already available.

- The experiments with the car in a test rig would be far more comfortable than in a test tower due to the easy handling of the car and the possibility to emulate infinite travel duration.

- Changes in the travel speed and rail profiles are easily done by modifying the software.

- Safety measures in case of the test tower experimentation presented a considerable handicap in comparison with the test rig.

Upon completion of the experiments in the test rig a prototype of the active damping system is to be fitted to an elevator car running in a test tower.

2.2.3 Control signals

During the phase of early investigations the inputs and the outputs of the control system had to be defined. Due to the suspension of the elevator car in the hoistway without any reference that indicates the car position with respect to an earth-fixed coordinate system, the only measurements available for vibrations damping must be provided by inertial sensors. As gyroscopes are very expensive to be considered for this application, accelerometers are the only reasonable sensors to provide the signals for active damping of the vibrations.

As stated in Chapter 1, the active system must correct the car orientation with respect to the guide rails in addition to vibration damping. To give an indication of this orientation, it is necessary to measure the position between the car and the two guide rails at several reference points.
2.2.4 Identification of the guide rail profiles

One of the important ideas evaluated during the early phase of the project was the use of identified guide rail profiles as input signals to the controller [51]. The profiles are identified during a “learning travel” that takes place during the start-up of the active system and might be repeated periodically for calibration. The algorithms to do this are those of a nonlinear observer capable of simultaneously estimating the states as well as the input signals [26]. As described in [37], it is possible to identify the second time derivative of the rail deflection (the curvature), but not the deflection itself. The deflection of the rail is needed to estimate the spring forces. However, it cannot be obtained by a double integration of the curvature due to sensor bias. The deflection remains unknown.

Even when the rail deflection is known, a controller based on this information is very sensitive to changes in the elevator speed and can only reduce the effect of the rail disturbances but not the effect of the direct forces on the car. The controller based on acceleration feedback remains then necessary. As the studies have shown, the acceleration controller alone is sufficient to achieve the required performance and the identified rail profiles would not enhance the ride quality. The controller would be more complex and the measurements of the car velocity and position would be also needed. For these reasons it was decided to exclude the identification of the rail profiles from the control strategy.
3 Concept of the Active Damping System

3.1 Impact of design constraints on the system configuration

The requirements on the active damping system summarized in chapter 1 have influenced its concept in the way described below.

3.1.1 Actuation

Considering size, weight, cost, and noise requirements, it is evident that electrical devices are the only possible type of actuators that might be used in such an application. An actuator used for active vibration damping may be based on one of the three following principles:

1) A dynamic shaker fastened to a point on the body required to reduce its vibrations. It actuates a moving counter mass so that the base of the shaker will be as still as possible. In the case of the elevator cabin, a number of shakers may be fitted to some selected points.

2) For the particular case of the elevator car, electrical magnets that replace the guide rollers or work in parallel to them can be used to generate forces between the cabin frame and the two guide rails ([2] and [55]). The whole system is controlled so as to suspend the car on magnetic cushions.

3) A motor that exerts force between the body required to reduce its vibrations and another body that already exists around. For example, a linear motor can be integrated between the cabin and its frame or between the cabin frame and the guide roller.

Using shakers does not permit to exert constant or low frequency forces on the controlled mass. Therefore, the correction of the car orientation, which is an important requirement on the active system, is not possible.
The second idea was not found to be realistic for the following reasons:

- Due to the small available guide rail area, the forces produced by the magnets are not sufficient. In addition, the relative vertical speed between the magnetic field and the guide rail, caused by the elevator car travel, leads to considerable energy losses in the material of the guide rail. This presents another reason for limiting the force exerted.

- A total replacement of the guide rollers by magnets is not possible, since safety requirements are not satisfied in case of a failure of the active system.

- Besides, there is no reason to expect better performance or less production price if this solution is adopted.

The third option is then the only reasonable one. However, an actuator between the cabin and its frame is also not capable of correcting the car orientation with respect to the guide rails. For this reason the actuators must be integrated between the guide rollers and the car frame.

### 3.1.2 The active roller guide shoe

The need to design the active damping system as an accessory kit for elevator cars already in service has encouraged to limit the mechanical changes in the car to the replacement of only a few parts. The passive roller guide shoes are then replaced by active roller guide shoes which contain the actuators.

As depicted in Figure 1.1 (Chapter 1), the active roller guide shoe has one x-guide roller and two y-guide rollers. Each of the guide rollers is connected to the car frame by a rocking lever and is kept in contact with the guide rails by a prestressing spring force. A schematic drawing of the active roller guide shoe is shown in Figure 3.1. The moving part of the x-actuator is fixed to the outer edge of the rocking lever of the x-guide roller. The movement of this roller is thus forced against or away from the guide rail. The outer edges of the two rocking levers of the y-guide rollers are connected by a horizontal connecting rod so that the mechanism has a single degree of freedom actuated by the y-actuator. The position sensors necessary for the correction of the car orientation with respect to the guide rails are integrated in the assembly parallel to the springs.
Since the original roller guide shoe construction with the springs is conserved, safety requirements are fulfilled. The active roller guide shoes are designed to be interchangeable with the passive ones.

### 3.1.3 Accelerometer

Due to the cost requirements on the system, it is necessary to keep the number of accelerometers at the minimum necessary. This is equal to the number of degrees of freedom of the cabin to be controlled. However, mounting the accelerometers on the cabin itself has an important drawback, as explained in Chapter 5 below (Section 5.1.2). A collocation between the actuators and the sensors is very favored for the control of the system [27]. It is achieved when the accelerometers are mounted near and parallel to the actuators. In this case eight accelerometers are
required instead of five. This obviously raises the costs of the active system. Therefore, another scheme using only six accelerometers has been adopted, with the accelerometers mounted on the cabin frame. The reason why six accelerometers are needed rather than only five is explained in Chapter 5 below.

3.2 System configuration

3.2.1 Actuators and sensors

Figure 3.2 shows the location of the actuators and accelerometers on the elevator car as described above. The position sensors are mounted according to Figure 3.1 but are not shown in the simplified active roller guide shoes. In this configuration the active damping system has eight actuators, eight position sensors, and six accelerometers.

3.2.2 Hardware scheme

The hardware of the active system can be divided into three main groups. The first group includes the actuators and sensors and is integrated in the mechanical structure of the elevator car according to Figures 3.1 and 3.2. The second group contains the following electronic units (cf. Figure 3.3):

1) Voltage-to-current converters (V/C) used to transform the voltage signals of the accelerometers into current signals to insure a disturbance-free transfer up to the interface unit of the real-time computer. The transfer of the signals in the form of current is necessary due to the long distances between the sensors and the computer. The position sensors already have built-in V/C converters.

2) Analog signal conditioning (ASC) for all measured signals including current to voltage converter (C/V), anti-aliasing filter, and amplification is necessary to prepare the signal for the analog to digital converter (A/D).

3) Power amplifiers (Amp.) are used to supply the energy required to drive the actuators. They are of the pulse-width modulated type and contain a current feedback loop in order to drive the actuators in a force command mode.

4) Power supply for the whole system.
The third group of hardware belongs to the computer system. It includes the following units:

1) The real-time computer responsible for the realization of the controllers and for the measurements needed for system identification.

2) The interfaces of the real-time computer including the A/D converters as well as the digital-to-analog (D/A) converters which provide the command signals to the actuators.

Figure 3.2: Positions of the actuators and accelerometers on the elevator car
3) The host computer which communicates with the real-time computer, prepares the various real-time applications, and is used for all off-line analysis and evaluation of the results.

The hardware scheme of the whole system is shown in Figure 3.3.

3.3 Component selection for the active damping system

3.3.1 Actuator

Actuator type

As explained above, the actuators are integrated in the roller guide shoes between the guide rollers and the cabin frame. The simplest and cheapest version of the actuator is a permanent magnet which moves between two stator coils connected in series (cf. Figure 3.1). The moving part is fastened to the outer end of the rocking lever and has a limited rotary stroke so as to keep a linear relationship between the current in the coils and the force generated on the moving part. The only drawback of this design is the unstable lateral position of the moving part between the two coils due to the very high side forces tending to attract this part to one of the coils. To overcome this difficulty, guide rollers are used to keep the moving part in a middle position between the two coils.

Actuator specifications

With the requirements on the active system given in [54] and considering the mechanical construction shown in Figure 3.1, the actuator specifications can be defined. As a result of the close cooperation with the actuator manufacturer, it was possible to develop a suitable actuator. The important part of its specifications is summarized in Table 3.1 below. The lever ratio between the actuator and the roller is 3:1 making the nominal force on the roller equal to 900 N.

3.3.2 Actuator drive amplifier

The drive amplifier of the actuator is a four-quadrant current converter produced by the same manufacturer. It is configured specifically to fulfill the actuators power and safety requirements. It has an analog current feedback loop with a PID
Figure 3.3: Hardware scheme of the active damping system
controller which allows to use the actuator in a force command mode. This means that the input voltage to the amplifier is interpreted as the command force value required from the actuator. The current supplied to the actuator is pulse width modulated with a switching frequency of 23 kHz. The nominal average current is 6.6 A, with a possible peak of 22 A. Thermal damage is prevented by a safety switch disconnecting the unit when the dissipated energy exceeds a certain limit.

Table 3.1: Actuator specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal thrust</td>
<td>300</td>
<td>N</td>
</tr>
<tr>
<td>Peak thrust</td>
<td>800</td>
<td>N</td>
</tr>
<tr>
<td>Nominal power</td>
<td>104</td>
<td>W</td>
</tr>
<tr>
<td>Peak power</td>
<td>740</td>
<td>W</td>
</tr>
<tr>
<td>Stroke</td>
<td>30</td>
<td>mm</td>
</tr>
<tr>
<td>Thrust constant</td>
<td>40</td>
<td>N/A</td>
</tr>
<tr>
<td>Electrical time constant (force time constant)</td>
<td>12.4</td>
<td>ms</td>
</tr>
<tr>
<td>Mass of the moving part</td>
<td>1.0</td>
<td>kg</td>
</tr>
<tr>
<td>Stator mass</td>
<td>7.5</td>
<td>kg</td>
</tr>
</tbody>
</table>

3.3.3 Accelerometer

Due to their low cost, continuously improved characteristics, and compactness, micro-machined piezo-accelerometers are now dominating the market. In all possible types of these accelerometers the displacement of a micro-machined mass relative to a casing is taken as a measure of the acceleration. The mass is connected to the casing through a cantilever beam with a certain elasticity, the movement taking place in a damping environment. However, the accelerometers differ according to their measurement principles as follows:

1) In a piezo-electric accelerometer, the electric charge accumulated between the mass and the casing is measured.
2) The piezo-resistive accelerometer contains a strain gauge on one of the cantilever surfaces to measure the deflection as a function of the acceleration using a resistive bridge.

3) The piezo-capacitive accelerometer measures the distance between the mass and the casing by determining the capacitance between the two surfaces.

The piezo-electric accelerometer was excluded from the evaluation for two main reasons: first, it is not capable of measuring accelerations below a certain frequency and secondly, the measured signal is interrupted in cases of overload and sometimes even upon start-up. The other two types are suitable for active damping systems, with the main difference that the second type needs temperature compensation while the third type is the most accurate one. At the time of the construction of the test rig, piezo-resistive accelerometers were the only type available at an acceptable price fulfilling the specifications. Over the past two years, the technological development in micro-machining together with the widespread use of accelerometers in mass produced systems such as automobile suspensions, anti-lock systems, and air bags encouraged manufacturers to produce low-cost piezo-capacitive accelerometers with a far better performance than the piezo-resistive types, even at a lower price.

The key specification of the accelerometer for this application is its measurement range. Considering the accelerometers offered on the market it is usually easier to find devices with higher measurement ranges, like for instance ±20 g or more, than with lower measurement ranges, like ±1 g or less. As stated in [54] the active damping system is required to reduce the peak-to-peak acceleration signals to below 7 mg. According to experience with high-rise elevators the maximum acceleration measured before the installation of the active system ranges from 40 to 100 mg depending on the conditions of operation and the quality of the rails. In addition to the acceleration resulting from the vibrations, the accelerometer measures a certain component of the earth’s gravitation. A single degree of tilt is responsible for about 17 mg of additional acceleration. With a few degrees of tilt it is reasonable to assume that the absolute value of the maximum measured signal will be less than 200 mg. Since high amplification of the signal would increase the noise level and deteriorate the signal-to-noise ratio, it is not advisable to amplify the measured acceleration signal too much. The ideal measurement range of the accelerometer will then be ±0.5 g.
At the time of the construction of the test rig, accelerometers with a measurement range below ±2 g were not available at a suitable price. Capacitive low-cost accelerometers with a measurement range of ±1 g began to appear in the last two years only. The sensors used in the laboratory version of the active system in the test rig have a measurement range of ±2 g. An amplifier integrated with the V/C unit is adjusted to limit the range of the transmitted signal to ±1 g only.

Another sensitive feature of the accelerometer is its bandwidth. As stated in [54], the acceleration controller is required to achieve performance in the frequency range between 1 and 20 Hz. For the stability of the system it is necessary to have good measurements up to 60 Hz with very little phase error. The accelerometer selected can be modeled as a second-order mass/spring system with a damped natural frequency of about 450 Hz, and a damping factor of about 0.4. Significant variations of these two parameters from one sensor to another are common but the phase and the magnitude errors remain small in the relevant frequency range.

The accelerometer selected is the model 3140 offered by Euro Sensor in England. It is a piezo-resistive accelerometer which has its built-in temperature compensation. The effects of non-linearity, scale factor error, bias stability, and noise on the quality of the signal for identification or control are minor. It is expected that a piezo-capacitive accelerometer would be even better in this respect.

3.3.4 Position sensor

Position sensor type

As evident from the construction sketch of the active roller guide shoe shown in Figure 3.1, the position sensor measures the distance between the vertical support and the rocking lever. To avoid mechanical wear, it is advantageous to use a contactless sensor. The type chosen is the CK40 produced by Hans Turck GmbH & Co. in Germany. It is an inductive sensor which contains a coil activated by a high-frequency oscillating supply (typically 120 kHz). When a metallic object is located in the vicinity of the sensor, it absorbs part of the oscillator system energy. The resulting attenuation in the amplitude of the coil voltage is a measure of the distance between the sensor and the metallic object. An electronic circuit produces a signal proportional to this attenuation, linearizes it, and amplifies it. A first-order low-pass filter is then used to reduce the noise level of the output.
**Position sensor specifications**

The bandwidth of the position control loop will be far below that of the acceleration control. For this reason, the bandwidth of the position sensor is allowed to be correspondingly lower than that of the accelerometer. The noise level of the output may then be kept at a very low level. The bandwidth of the sensor chosen is 80 Hz and its noise level is about 1.6 μm peak-to-peak.

The measurement range of the sensor is 7 mm (from 4 to 11 mm). The maximum linearity error is ±3% of full scale and the temperature error is less than ±5% of the same value. The output of the sensor is available as a voltage or as a current signal.

### 3.3.5 Analog signal processing

Before the A/D conversion takes place, the signals of all fourteen sensors are transformed back to voltages. Each of them is then filtered with a second-order analog Chebyshev low-pass anti-aliasing filter. The range of the final signals fed to the A/D converter is then ±10 V.

The most important characteristic of the analog signal processing channels is that up to a frequency of 60 Hz they do not produce any remarkable phase or amplitude errors. For this reason the bandwidth of the anti-aliasing filter is fixed to 200 Hz and its order is not allowed to exceed two. Concerning the V/C and the C/V units, their bandwidths are high enough so that their influence on the amplitude and the phase is even smaller than that of the anti-aliasing filters.

### 3.3.6 Real-time computer

**Requirements**

The real-time computer is the instrument with which component testing, system identification, and controller realization are performed. Due to the intensity of these tasks during the laboratory phase of the project, it is very advantageous to keep the programming and debugging effort of the real-time computer to a minimum. At the time of the construction of the test rig, the final requirements on the real-time computer were still unknown and it was therefore wise to use a powerful device capable of fulfilling the control requirements in their most demanding case.
Real-time computer for the laboratory version of the system

The real-time computer for the laboratory version of the active damping system is based on a digital signal processor (DSP) as described in [8]. It is composed of a single processor board which contains a DSP capable of 40 million floating point operations per second (DS1003 Parallel DSP Board based on Texas Instruments TMS320C40). A separate multi I/O board (DS2201) provides the following interface to the active system hardware:

1) A/D Converters

20 channels, multiplexed to be sampled by five 12-bit converters. All channels are equipped with simultaneous sample-and-hold circuits to ensure that all sampled values belong to the same time instant. The conversion time is 32.5 µs for all channels and the input range is ±10 V.

2) D/A Converters

Eight bipolar 12-bit units with 4 µs settling time. The output range of all channels is also ±10 V.

3) Other interfaces

In addition to 16 programmable digital I/O lines, the board contains six pulse-width modulated outputs, not used by the active system.

The DSP board communicates with the multi I/O board through a 32-bit parallel bus which allows a data transfer rate of 20 MB/s. For the communication with the host computer, both boards are mounted on a PC/AT bus connected to the host computer via Ethernet. The board may be equipped with up to 2 M words of 32-bit on-board SRAM memory, intended for data acquisition. Through an external connector this memory can be expanded to up to 5 M words. However, in the configuration used for the test rig, only 512 k words are used.

The DSP board, the multi I/O board, an interface board for the PC/AT bus, and an Ethernet board are all mounted in a separate expansion box (PX20) which contains its own power supply. This expansion box has a total of 20 PC/AT slots, only four of which are occupied. The remaining 16 slots are reserved for future expansion.
Software and programming

The biggest advantage of the real-time computer chosen is the ease of its programming. Another advantage is that its software is compatible with the program MATLAB [38], the main software package used in this project for system analysis and controller design.

A real-time application is programmed graphically in SIMULINK [39], the graphical simulation environment of MATLAB. By using an additional software package of the same producer [40], a C-code representing the SIMULINK window containing the real-time application can automatically be generated. The window contains additional graphical elements supplied by the real-time computer manufacturer which make reference to the specific hardware used for inputs and outputs. The C-code is optimized for minimum computational power, compiled, and linked. The resulting object code is then downloaded on the DSP board. The whole process is carried out automatically upon the activation of a single command. An on-line progress report of the process, showing any error messages, is active until the successful start of the DSP. The automatically generated C-code is clearly commented and may be modified manually, if needed.

Following the download of a real-time application, it is possible to change any of the numerical parameters shown in the graphical presentation. A special real-time interface is used [11] which allows activating, deactivating, or changing the parameters of the application without a recompiling or downloading again. In order to get measurements during the run of a real-time application, an additional software facility [9] allows specifying the variables and also the parameters to be measured during the run of the application. Upon completion, the measured data are transferred from the real-time computer to the host computer where they can be saved directly in MATLAB format.

Examples of real-time applications for system identification and controller realization are shown in Figures 5.7 (Chapter 5) and 7.1 (Chapter 7), respectively.

3.3.7 Host computer

The host computer is needed for the programming of the real-time computer and for the communication with the whole system. It is also the computer used for system identification, controller design, and performance evaluation.
The possibility of operating the real-time computer from a SUN Sparc station allowed the use of any of the available work stations at the Measurement and Control Laboratory as a host computer. These machines use the operating system SunOS (Release 4.1.3_U1) and are all connected together through a local server. The real-time computer communicates with the host computer via Ethernet. A Sun Sparc Station LX was used as a host computer with the possibility of utilizing other more powerful machines in a remote login mode in case intensive calculations were needed for analysis or C-code optimization.

3.4 Future development of the active damping system

The active damping system presented in this chapter is the one developed for the experiments in the laboratory test rig. For the following project phases the actuators and sensors should remain the same. The electronic components might need some modifications. The computer system used in the laboratory test rig is comfortable to program and possesses great computational capabilities, but it is expensive and not ideal for either a test tower version of the system or for a prototype. For the later project phases the computer system can be selected only after the final definition of the computational power required.
4 Laboratory Test Rig

4.1 Concept of the test rig

4.1.1 Purpose

The test rig provides the validation facility necessary for the hardware of the active damping system during its development. The fact that the elevator car in the test rig does not move in the vertical direction makes the experiments more comfortable. This is very useful during the first phase where intensive experimental work is needed to find the most suitable controller design procedure. The experiments are carried out with a full-scale elevator car suspended in the test rig. This ensures that the dynamics of the real system are not changed and that the solution is not far from the one suitable for the elevator car in a hoistway.

Aside from the role it plays during the primary phase of the project, the test rig continues to have the following functions after the successful completion of the first phase:

- It allows the examination of the dynamic behavior of any type of elevator car. This may be used to modify actual designs.

- It may be used for testing the components intended for the following project phases (e.g., electromagnetic compatibility tests).

- It may be used to develop and test new types of controllers before they are applied to an elevator car in a hoistway. This is particularly important since the stability of the controlled system has to be checked experimentally.

4.1.2 Experimentation

A full-scale high-rise elevator car equipped with the active damping system is suspended in the test rig the way it is suspended in a hoistway (cf. Figure 4.1). The experimentation with the elevator car includes the following:
Figure 4.1: The test rig
1) Testing of the active system hardware.

2) System identification to determine the mathematical model needed for controller design.

3) Testing of the designed controller and determination of its performance.

To test the active damping system, vibration exciters are used to emulate the excitation caused by the rail unevenness during elevator travel.

4.1.3 Expansion capabilities

One important feature of the test rig is the fact that it allows carrying out the tests for an increasing number of degrees of freedom of the car (cf. Section 2.2.1). An expansion of the test rig to add exciters for the emulation of the direct car forces is also possible. However, such an expansion has not been found necessary as yet.

4.2 Mechanical assemblies

4.2.1 The elevator car

The elevator car suspended in the test rig is a medium-sized high-speed car produced for the European and the Asian-Pacific markets. The structure of the car is shown in Figure 1.1 (Chapter 1); its main features are summarized in Table 4.1 below. The nominal speed of the car is 5 m/s but it may be increased to 6 m/s or beyond depending on the operating conditions.

4.2.1 Elevator car suspension frame

The suspension frame is a steel construction used to suspend the elevator car in the test rig in the same way as it is suspended in a hoistway, but with a limited length of the suspension cables. It also provides mounting stages for the excitation units used to emulate the effect of rail unevenness during the elevator travel. The important specifications of the suspension frame are summarized in Table 4.2; its appearance is presented in Figure 4.1.

The most important feature of the suspension frame is its rigidity. An insufficiently rigid frame would vibrate in response to the reaction forces between the car and the
excitation units and the active damping system would be influenced by the vibrations of the car as well as those of the frame.

### Table 4.1: Main features of the elevator car in the test rig

<table>
<thead>
<tr>
<th>Feature</th>
<th>Value</th>
<th>Unit</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Payload</td>
<td>1600</td>
<td>kg</td>
<td>21 persons</td>
</tr>
<tr>
<td>Mass of the empty cabin</td>
<td>1800</td>
<td>kg</td>
<td></td>
</tr>
<tr>
<td>Mass of the cabin frame</td>
<td>800</td>
<td>kg</td>
<td>lower yoke 395 kg upper yoke 275 kg each side stile 65 kg</td>
</tr>
<tr>
<td>Cabin dimensions</td>
<td></td>
<td></td>
<td>width depth height</td>
</tr>
<tr>
<td>Distance between upper and lower roller guide shoes</td>
<td>3.94</td>
<td>m</td>
<td>also called wheel base</td>
</tr>
<tr>
<td>Height of the cabin frame</td>
<td>3.15</td>
<td>m</td>
<td>from upper to lower yoke</td>
</tr>
<tr>
<td>Distance rail to rail</td>
<td>2.07</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Mass of the roller</td>
<td>5.00</td>
<td>kg</td>
<td>equivalent mass of the roller and the rotating lever</td>
</tr>
<tr>
<td>Roller prestress force</td>
<td>200</td>
<td>N</td>
<td>x-direction</td>
</tr>
<tr>
<td></td>
<td>150</td>
<td>N</td>
<td>y-direction</td>
</tr>
<tr>
<td>Roller play to stopper</td>
<td>4.0</td>
<td>mm</td>
<td>x-direction</td>
</tr>
<tr>
<td></td>
<td>3.5</td>
<td>mm</td>
<td>y-direction</td>
</tr>
</tbody>
</table>

### Table 4.2: Main specifications of the elevator car suspension frame

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
<th>Unit</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Width</td>
<td>2.60</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Depth</td>
<td>2.60</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Height</td>
<td>6.50</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Mass</td>
<td>2500</td>
<td>kg</td>
<td>approximately</td>
</tr>
</tbody>
</table>
Since the conditions in the test rig are to be kept as close as possible to those in the hoistway, it is important to limit the effects of short suspension cables. Short suspension cables produce pendulum forces which may alter the dynamics of the car. Nevertheless, it is not possible to increase the height of the test rig beyond a certain limit due to the limited space of the laboratory and in order to assure the stability of the whole construction.

As seen in Figure 4.1, the structure is formed by four vertical columns and three horizontal stages. The lower and the middle stages are designed to support the excitation units, while the upper stage is made for the suspension of the car. Each of the three stages may be shifted upwards or downwards for limited distances in constant steps. This allows adapting the suspension frame to the different cars that might be examined later. The whole construction is fastened by nuts and bolts only in order to allow future modifications. The possibility of adding a fourth stage between the lower and the middle ones exists for potential future demands such as more rigidity or the need to mount additional devices. In order to increase the rigidity for the same weight, the suspension frame is constructed with squared hollow steel sections. For simplicity, identical sections (20 cm x 20 cm) were used for all columns and beams.

4.2.2 The hydraulic exciters

Exciter type

Beside the high requirements on its positioning accuracy, resolution and repeatability, the exciter needed for the emulation of the rail profiles must be capable of producing high force amplitudes at a wide dynamic range. The exciter chosen is a hydraulic cylinder driven by a four edge servo valve which has a rotating valve body actuated by a stepper motor through a V-belt. The rotary motion of the valve body results in the application of the hydraulic pressure to the cylinder which moves linearly and closes the valve opening upon its motion through a screw and nut mechanism. The stepper motor is coupled to an incremental encoder to measure the position. With the stepper motor as a driver and the mechanical feedback in the cylinder assembly, the unit is designed for open-loop operation, thus eliminating the need for an additional feedback position controller.
Exciter specifications

The excitation system is required to emulate the unevenness of the rail profiles at the four contact points between the elevator car and the two guide rails (cf. Figure 4.1). Since a guide rail has two independent profiles, each contact point has two degrees of freedom to be driven by two different exciters. Eight exciters are thus needed according to the scheme of Figure 4.2.

Table 4.3: Specifications of the hydraulic exciter

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
<th>Unit</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum stroke</td>
<td>120</td>
<td>mm</td>
<td>The stroke is limited to ±10 mm by the software for safety</td>
</tr>
<tr>
<td>Repeatability of the position</td>
<td>5.0</td>
<td>μm</td>
<td>According to [21] 10 μm is a sufficient accuracy of the exciter position</td>
</tr>
<tr>
<td>Resolution of the position</td>
<td>2.5</td>
<td>μm</td>
<td></td>
</tr>
<tr>
<td>Stiffness of the cylinder</td>
<td>450</td>
<td>N/μm</td>
<td></td>
</tr>
<tr>
<td>Maximum amplitude at 6.6 Hz</td>
<td>1000</td>
<td>μm</td>
<td>Refer to the remarks below concerning the dynamic behaviour of the cylinder</td>
</tr>
<tr>
<td>10 Hz</td>
<td>700</td>
<td>μm</td>
<td></td>
</tr>
<tr>
<td>20 Hz</td>
<td>218</td>
<td>μm</td>
<td></td>
</tr>
<tr>
<td>40 Hz</td>
<td>47</td>
<td>μm</td>
<td></td>
</tr>
<tr>
<td>Maximum static force</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inward</td>
<td>11,700</td>
<td>N</td>
<td>The inside cylinder areas are different from one side to the other</td>
</tr>
<tr>
<td>Outward</td>
<td>14'500</td>
<td>N</td>
<td></td>
</tr>
<tr>
<td>Maximum acceleration</td>
<td>2</td>
<td>m/s²</td>
<td></td>
</tr>
<tr>
<td>Maximum velocity</td>
<td>0.08</td>
<td>m/s</td>
<td></td>
</tr>
<tr>
<td>Working pressure</td>
<td>120</td>
<td>bar</td>
<td></td>
</tr>
</tbody>
</table>

Concerning the exciter force, it is important to know the required maximum static force needed and to have a good idea about the required shape of the frequency response of the output position. As the exciter is used to emulate a rail profile, it is also important to determine the accuracy, the repeatability, and the resolution of its position. To define the specifications of the exciters, a detailed study was carried out at the beginning of the project [21]. The important specifications of the cylinder
Figure 4.2: Hardware scheme of the excitation system
chosen are summarized in Table 4.3. The following points are interesting to note about the dynamics of the cylinder:

- The force supplied by the hydraulic cylinder is far higher than the load resisting its motion. For this reason the frequency response between the command position and the actual position of the cylinder is not affected by the load.

- The stepper motor is the limiting element of the dynamic response rather than the cylinder itself.

- Below 6.6 Hz the limits on the maximum possible position amplitude of the cylinder commanded by a sine wave is determined by the maximum possible velocity of the stepper motor. Above 6.6 Hz this limit is given by the maximum possible acceleration.

Excitation unit

Four identical excitation units are needed for the emulation of the rail profiles. As shown in Figure 4.2, each excitation unit uses two hydraulic cylinders. In order to protect the cylinder against side forces, it is necessary to integrate it in a linear guide mechanism as shown in Figure 4.1. Each excitation unit moves a small piece of the guide rail in two perpendicular directions in a horizontal plane.

Due to the high forces generated, a hydraulic cylinder may not be driven all the way to the end of its stroke. In addition, any large motion of the rail may damage the roller guide shoe. For these reasons it is necessary to limit the stroke. The signal used to detect this motion is supplied by a contactless inductive limit switch that produces an alarm signal if the cylinder motion exceeds the range of ±10 mm from the zero point defined. An identical limit switch is used for the initialization of the cylinder, i.e., to position the cylinder before the start of the excitation.

4.3 The control system

The hydraulic cylinders employed in the test rig as exciters are originally produced for machine tools. Consequently, the control systems available for them are intended for applications such as copying machine tools or punching machines, where a simple velocity trajectory is to be followed. For this reason, these control
Laboratory Test Rig 55

systems are not suitable for application in the test rig. A special development was therefore necessary.

The hardware scheme of the control system for the excitation is presented in Figure 4.2. To emulate the rail excitation, a host computer is used to generate reference position trajectories for the eight exciters based on measured rail profiles. The generated trajectories are then downloaded on a real-time computer controlling the motion. The real-time computer converts each trajectory into a set of command signals to the stepper motor of the corresponding exciter. These control signals are generated by timer modules which control the pulse train to the stepper motor drivers.

The control of the exciters is realized in an open-loop mode; it is therefore useful to keep track of the actual position of the units. The counter modules receive the signals from the incremental encoders coupled to the stepper motors and sum them to provide the actual motor angle. By monitoring the actual positions of the motors it is possible to make sure that they follow the command signals and that the required cylinder motion is reached. This position observation is also very useful during the experimental determination of the maximum possible acceleration and velocity of the stepper motor. Since the load on the stepper motor is unknown and probably variable, the maximum allowed acceleration and velocity are unknown and have to be determined experimentally.

The real-time computer is also responsible for monitoring the limit switches for safety. In addition, it controls the initialization of the exciters before the start and after the stopping of motion. The actions of the real-time computer are controlled and monitored by the host computer from a special user's interface.

4.3.1 Requirements on the control system

The requirements on the control system for the emulation of the rail excitation are summarized as follows:

- The reference trajectories for each of the eight exciters are given in the form of position deflections versus time. They are loaded from the host computer to the memory of the real-time computer using a special user's interface.

- The sampling frequency of the control system is 200 Hz for all exciters. This allows to realize 40 Hz sine waves with sufficient accuracy (cf. Table 4.3).
- The maximum length of a file containing a reference trajectory is 4000 points (16-bit words). With the sampling frequency of 200 Hz the maximum period of non-repeated travel is 20 seconds. For an elevator speed of 6 m/s this is enough to emulate an elevator travel of 120 m.

- The files are repeated from the beginning after they terminate. The cylinders are then driven in a continuous operating mode until a stop command is issued by the host computer. The whole system may operate non-stop for several hours.

- All exciters are synchronized. This means that the position reference points in all eight files which belong to the same time period are realized simultaneously.

- It is possible to stop and start any or all of the cylinders at any time by a corresponding command from the host computer. After a stop command is issued, the stopped exciters are automatically repositioned to their zero points.

- It is possible to observe the real travelled trajectories as measured by the incremental encoders. The data files are transferred from the real-time computer to the host computer upon the activation of the corresponding command. The data transfer does not block the system functionality.

- In case of an alarm signal from any of the limit switches the whole system stops. The position of an exciter which lies outside the safety limits may then be checked and corrected by issuing the proper commands from the host computer.

- The user’s interface to operate the system is contained in a single window animated by the host computer. It is possible to activate all system control commands from this window (e.g., the loading of the different files, the initialization of the exciters, the start, the stop, or the importation of the real travelled trajectories).

### 4.3.2 Real-time computer and interfaces

The real-time computer of the excitation system is a VME-based system which has one processor board with a MOTOROLA 68040 microprocessor. Two interface boards provide mounting connectors for a total of eight standard interface modules (cf. Figure 4.2). The modules used for this application are the following:

1) Four timer modules; each of them supplies the frequency pulses necessary to drive two independent stepper motors.
2) Two counter modules; each of them receives the pulse trains from four independent incremental encoders. The module sums the pulses to provide the actual position of each of the four stepper motors.

3) One digital I/O module tracing the signals from the sixteen limit switches used for safety and initialization.

The operating system of the real-time computer is described in [41]. The compiled program realizing the control of the exciters and ensuring the system safety resides on the programmable read-only memory of the real-time computer. The host computer communicates with the real-time computer through a serial port RS232.

4.3.3 Host computer and software

One of the important requirements on the excitation system is that it is operated from the same host computer which operates the active damping system during the laboratory phase of the project. The same SunSparc Station LX (Section 3.3.7) is used as the host computer for the excitation system. The rail profiles are to be prepared using the same software program MATLAB [38] used for controller design, realization, and evaluation. The rail profiles are then saved in a directory as a set of MATLAB-compatible variables before being loaded on the real-time computer. The actual travelled trajectories monitored by the real-time computer are transferred to the host computer in the same format.

The control program and the user’s interface are both programmed in OBERON [50] which offers a lot of flexibility concerning the realization of real-time applications, especially in the programming of the user’s interface [59]. It allows to execute the different control commands from the host computer in real-time. Some examples of these commands are given below:

- Load a trajectory file for one of the exciters from the host computer to the real-time computer
- Activate the cylinders used in the experiment
- Initialize the active cylinders
- Start / stop the active cylinders
- Get the actually traveled trajectory of the stepper motor.
In order to keep track of the execution of these commands, dedicated messages appear on the user’s interface of the host computer. These indicate the accomplishment of start, stop, or initialization commands. They also indicate an emergency stop and give an alarm should a stepper motor lose track of the required trajectory.

4.3.4 Power electronics

The stepper motor used to drive the valve body of the hydraulic exciters is a five-phase bipolar constant current motor. As used here, the motor operates in a high resolution mode of 1000 steps/revolution (half step mode). The V-belt used to drive the valve body is mounted so that one step is equivalent to the resolution of the exciter, i.e., 2.5 µm. A stepper motor driver board is required to amplify the pulses produced by the timer modules of the real-time computer. The board operates from a 60 V DC supply and consumes around 1.5 A. The switching frequency of the on-board chopper is 20 kHz, with an output current level of 1.3 A per phase. The eight boards needed are mounted in two identical power electronic racks (cf. Figure 4.2). Each of them contains a power supply and is fitted with the proper connectors to the real-time computer as well as to the motors.

4.4 Emulation of rail excitation

4.4.1 Preparation of the rail profiles for excitation

As seen in Section 4.3, the four values of the rail deflections $x^r_g, y^r_g, x^l_g$ and $y^l_g$ (cf. Figure 4.3) can be measured along the height of the hoistway. Knowing the travel speed and the vertical distances between the measurement points, it is possible to generate a time-dependent position trajectory for each rail profile. The time shift between the rail profiles at the upper rollers and those at the lower rollers is calculated by dividing the distance between the upper and the lower rollers by the travel speed.

To implement the rail profiles on the excitation system the following processing steps are needed:

1) Four rail deflections are generated as a function of time for the given travel speed.
2) In most cases, high-pass filtering is necessary to limit the maximum and the minimum position deflections in a trajectory to realistic values. The measurements of rail profiles in a hoistway may be influenced by slowly varying errors which cause large low-frequency deflections. Real rail deflections are expected to lie in the range of ±4 mm. Large deflections may damage the roller guide shoes as they force the guide rollers to move beyond their stoppers.

3) The rail side velocity and side acceleration must be controlled to ensure that they lie within the limits of what the stepper motor can achieve. A procedure to bring these values within allowable limits, with a minimum change in the frequency content of the files, is described in [42].

It is also possible to manipulate the given rail profiles to change their frequency content or even to generate totally synthetic rail profiles. Synthetic rail profiles are useful if specific excitations are desired, such as sine waves or other particular shapes. Details of the procedure to program rail profiles for emulation and a complete description of the whole excitation system are documented in [42].

### 4.4.2 Example of rail excitation

The data used here as examples of rail profiles result from measurements in a real hoistway and contain the four deflections defined in Figure 4.3 for 2063 points equally spaced along the height of the hoistway. The distance between two succes-
sive points is 6 cm making a total travel distance of 123.72 m. As explained later in Section 7.3, a travel speed of 6 m/s is a good choice for testing the active system. Figure 4.4 shows the four position trajectories generated from the measured rail profiles for a travel speed of 6 m/s after they are prepared according to the procedure described in Section 4.4.1. The frequency contents of these profiles are presented in Section 7.3 (cf. Figure 7.4).
Figure 4.4: Example of rail excitation at 6 m/s
5 System Identification

5.1 System configuration

5.1.1 The mechanical construction

Figure 5.1 shows a simplified multi-body system of the elevator car equipped with the active damping system. The components of this system are the cabin, the cabin frame, and the equivalent moving masses of the actuators including the guide rollers. The utilization of elastic elements in the construction, such as rubber pads and helical springs, is responsible for the existence of different vibration modes between these bodies. Although it is not possible to control all of these modes with the proposed set of actuators and sensors, it is still important for the stability of the controlled system that their dynamics are described by the mathematical model used for the design of the controller.

5.1.2 Actuators and sensors

Based on the configuration of the active system presented in Chapter 3 (cf. Figure 3.2) it is possible to draw the scheme of the actuators and sensors as shown in Figure 5.2. The four x-guide rollers are kept in contact with the guide rails by prestressing the corresponding springs. This arrangement remains the same after the integration of the actuators LM1R, LM1L, LM4R and LM4L parallel to the springs. To ensure proper functioning of the active system, the rollers are not allowed to lose contact with the rails. For this reason the actuators coupled to these rollers should not apply pull forces with large amplitudes. Hence, two actuators are needed to generate forces in both directions, with only one of them active at a time. From the point of view of the controller the system remains linear, with the two actuator pairs LM1R and LM1L and LM4R and LM4L equivalent to the two actuators LM1 and LM4, respectively, as seen in the equivalent scheme of Figure 5.2.

The signals from the two position sensors PS1R and PS1L are subtracted to provide a single signal representing the deviation from the middle point between
Figure 5.1: Simplified multi-body system of the elevator car equipped with the active damping system

The two rails. The resulting signal may be considered as being produced by a single equivalent sensor PS1. The same is true for the other position sensor pair, PS4R and PS4L, which is considered equivalent to the sensor PS4. The equivalent scheme in Figure 5.2 with six actuators, six accelerometers, and six position sensors is the basis for system identification and controller design tasks to be implemented later.
Due to the utilization of rubber pads to connect the cabin to its frame, elasticities between the actuators and the accelerometers exist, if the accelerometers are mounted on the cabin. This results in a considerable loss of phase in the frequency responses from the actuator input to the accelerometer output. The controller for the loop formed by those two elements must have more differential nature leading to an increased sensitivity to noise. Thus the system becomes more difficult to control. This is the reason why the accelerometers were mounted on the cabin frame rather than on the cabin itself. The torsional elasticity around the vertical axis between the upper and lower yokes of the cabin frame necessitates the use of six accelerometers instead of only five.
5.1.3 The control system

System inputs and outputs

As shown in Figure 5.3, the inputs to the system are the six command force signals, while its outputs are the six acceleration signals and the six position signals. The subtraction of the signals from the position sensors PS1R and PS1L as well as PS4R and PS4L is done by the real-time computer. The same applies for the splitting of the output signals F1 and F4 to command the actuator pairs LM1R, LM1L and LM4R, LM4L, respectively (cf. Figure 7.1).

The control loops

The controller’s duty can be divided into two different tasks. The first one is to suppress the vibrations by way of acceleration measurements, while the second task is to correct the orientation of the car using the measurements from the position sensors. It is therefore logical to represent the controller in block diagram form as
being composed of two cascaded loops with combined force command signals at the output (cf. Figure 6.1).

*Controller’s coordinate system*

The inputs and outputs of the controller can be selected to be compatible with the degrees of freedom of the elevator cabin (cf. the coordinate system of Figure 1.1). This choice requires physical modeling of the system which was not adopted here. The proposed controller is based on a model which has its inputs in the actuators coordinate systems and its outputs in the sensors coordinate systems. The input and output signals of the controller are all in Volts.

### 5.2 Mathematical modeling and identification

#### 5.2.1 System dynamics

*The multi-body system*

The multi-body system of the elevator car shown in Figure 5.1 has a multitude of vibration modes. This includes the modes of the whole body composed of the cabin and its frame moving between the rollers, the modes belonging to the motion of the cabin relative to its frame, as well as the modes of the guide rollers with the moving part of the actuators as mass spring systems mounted on the cabin frame. These latter modes are the limiting factor for the controller bandwidth since they represent the resonances of the actuators.

*Structural dynamics*

Due to the flexibility of its mechanical components, the elevator car possesses an infinity of structural modes. The structural modes important for the design of the controller are those originating from the flexibility of the roller guide shoes and those resulting from the torsional elasticity of the cabin frame around the vertical axis. Since their influences are present in the frequency range of interest, their dynamics must be included in the system model.
Actuator dynamics

The actuator can be modeled as a linear solenoid motor (Section 3.3.1). However, it is driven by a pulse width modulated amplifier which has a built-in analog PID controller with current feedback.

To determine the dynamic behavior of the actuator and its driver, the frequency response between the input voltage representing the command force and the output voltage supplied by a force sensor mounted on the actuator was measured. The actuator was prevented to move in order to null the effects of inertia and friction on the exerted force. The results show extremely flat amplitude and phase responses up to 100 Hz. The amplitude ripple is less than 1 dB, while the phase fluctuation is less than 3°. Beyond 100 Hz, it is the dynamics of the structure rather than that of the actuator which shows resonances in the frequency range between 200 and 400 Hz. Since the frequency range of the required model lies between 0 and 60 Hz, the dynamics of the actuator and its driver as one unit may be considered negligible.

Accelerometer dynamics

The accelerometers used in this project are of the piezo-resistive type. They can be modeled as a mass spring system with a certain resonance frequency. This yields a second-order system model. The values supplied by the manufacturer for some of the system parameters, such as the natural frequency and the damping coefficient are subject to considerable uncertainty, especially the damping coefficient (Section 3.3.3). Therefore, it was important to use an accelerometer which has a natural frequency sufficiently higher than the dynamic range relevant for the system model (450 Hz). The accelerometer dynamics are then negligible in the modeling.

Position sensor dynamics

Referring to Section 3.3.4, the dominant dynamic behavior of the position sensor is that of the first-order low-pass filter connected in series to its output signal. It is therefore possible to model this sensor as a first-order low-pass element. The bandwidth of the position sensor used here is 80 Hz. Due to the low bandwidth of the position loop, the dynamics of this sensor are not relevant for the system model.
Dynamics of the measurement chain

Voltage signals from the sensors are converted into current signals for a disturbance-free transmission from the measurement point to the real-time computer. Before the A/D conversion, these signals are reconverted into voltages. The bandwidth of the V/C and C/V converter pair is higher than 1 kHz. This is high enough to neglect any amplitude attenuation or phase loss which might arise during the conversion. Following the C/V conversion, the signals are filtered by the anti-aliasing filters in order to assure correct sampling of the data. These filters are designed as second-order low-pass Chebychev filters with a bandwidth of 200 Hz. This bandwidth results in negligible phase loss and amplitude attenuation in the frequency range relevant for modeling.

5.2.2 Requirements for the system model

The requirements for the system model are dominated by the fact that it is a model needed for controller design rather than for simulation only. Therefore, it must be a linear time-invariant model. Strict conformity between the modeling assumptions and reality is very important. The fact that the elevator car is a dissipative system implies stability [25]. In addition, the model must reflect all other known physical features of the system, such as the absence of time delays, the existence of minimum as well as nonminimum-phase behavior, and the shape of the various frequency responses.

The order of the model remains one of the most important features to be defined. The higher the model order, the higher the order of the resulting controller. A high-order controller demands more real-time calculation effort and may not be implementable on the real-time computer available. A high-order model might also lead to numerical problems during the controller design. Nevertheless, the model must be capable of describing the important system dynamics, so that the desired performance can be achieved without endangering system stability. This trade-off between model accuracy and simplicity is a key problem always encountered in the application of control theory. The model must also be capable of describing all possible couplings between the various inputs and outputs of the system.
5.2.3 System identification versus modeling

In order to get a mathematical model of the system suitable for controller design there are two main methods:

1) Model building by deriving the equations of motion for the multi-body system using the Newton-Euler method [53] and by combining the result with the physical models of the other subsystems such as actuators, amplifiers, and sensors. The physical parameters, such as the masses, dimensions, and time constants, must then be determined, either from known information or by identification of the various subsystems.

2) Given a certain model structure defined by the order, the relative order, and the shape of the frequency response, the input and output signals are processed to estimate the non-physical parameters of the model.

As can be seen in Figure 5.1, the dynamic system of the elevator car is too complicated to be modeled by the first method. The complexity of the system and the large number of degrees of freedom involved do not allow the derivation of a reliable physical model with a reasonable effort. Furthermore, such a model is not capable of describing the structural dynamics of some parts of the car which need to be considered in the design of the controller.

Since the active damping system is designed as a modernization kit for elevators already in service, the elevator cars concerned differ in their construction and parameters, so that the modeling effort must be repeated for each new car. System identification to be carried out on-site for each new car will be the only reliable method to obtain a system model in a reasonable time. For this reason, the second method was chosen to obtain the required system model.

5.3 Model structure for identification

5.3.1 Proposed model

The proposed structure for the overall system model is given in (5.1). This includes a transfer function from each of the six actuator command forces to each of the twelve outputs, as indicated in Figure 5.3. Each individual transfer function being
known, the overall system model is known. However, since these transfer functions share many common parameters, the identification job becomes easier.

\[
\begin{bmatrix}
A_1(s) \\
A_2(s) \\
A_3(s) \\
A_4(s) \\
A_5(s) \\
A_6(s)
\end{bmatrix} =
\begin{bmatrix}
G_{a11}(s) & G_{a12}(s) & G_{a13}(s) & G_{a14}(s) & G_{a15}(s) & G_{a16}(s) \\
G_{a21}(s) & G_{a22}(s) & G_{a23}(s) & G_{a24}(s) & G_{a25}(s) & G_{a26}(s) \\
G_{a31}(s) & G_{a32}(s) & G_{a33}(s) & G_{a34}(s) & G_{a35}(s) & G_{a36}(s) \\
G_{a41}(s) & G_{a42}(s) & G_{a43}(s) & G_{a44}(s) & G_{a45}(s) & G_{a46}(s) \\
G_{a51}(s) & G_{a52}(s) & G_{a53}(s) & G_{a54}(s) & G_{a55}(s) & G_{a56}(s) \\
G_{a61}(s) & G_{a62}(s) & G_{a63}(s) & G_{a64}(s) & G_{a65}(s) & G_{a66}(s)
\end{bmatrix}
\begin{bmatrix}
F_1(s) \\
F_2(s) \\
F_3(s) \\
F_4(s) \\
F_5(s) \\
F_6(s)
\end{bmatrix}
\]

where

- \(F_i(s)\) Laplace transformation of the \(i\)th force command signal
- \(A_i(s)\) Laplace transform of the \(i\)th acceleration signal
- \(P_i(s)\) Laplace transformation of the \(i\)th position signal
- \(G_{a_{ij}}(s)\) Transfer function from \(F_j(s)\) to \(A_i(s)\)
- \(G_{p_{ij}}(s)\) Transfer function from \(F_j(s)\) to \(P_i(s)\)

During the first experiments with the system each one of the actuators was excited separately in order to determine the frequency responses of the six diagonal transfer functions with acceleration output \(G_{a_{11}}(s), G_{a_{22}}(s), G_{a_{33}}(s), G_{a_{44}}(s), G_{a_{55}}(s),\) and \(G_{a_{66}}(s)\). The most important observation of these transfer functions is that they all have three main resonances in the frequency range between 0 and 60 Hz. This is explained by the fact that each actuator is exerting force between its equivalent moving mass and the cabin frame (cf. Figure 5.1). The cabin frame itself is connected to the mass of the cabin through the elastic connection of the rubber pads. Considering the elasticity of the tire material existing between the roller and the guide rail, one can think of the actuator as influencing a system of three masses connected in series by elastic elements. Since the three masses oscillate relative to
each other and relative to the guide rail, three resonances dominate the dynamic behavior of this system. A sixth-order model is then capable of describing its dynamics.

Excitation with a pure sine wave having a frequency in the vicinity of these three resonances shows that the first resonance belongs to the oscillation of the cabin and its frame as a single mass between the rollers, that the second resonance describes the vibration mode between the cabin and its frame, and that the third resonance originates from the actuator moving part.

5.3.2 Physical guidelines

System submodels

The model presented in (5.1) can be divided into six submodels, each of which having a single input and twelve outputs. With $F_1$ as the input, the model is reduced to the first column. The same applies to each of the other five inputs. Each subsystem can be modeled by a common system matrix describing its dynamics. Hence, all of the transfer functions from a certain input to all of the outputs must have the same denominator.

The denominator of the diagonal transfer function should contain the most important poles of this subsystem and is then taken as the common denominator for all transfer functions belonging to the same subsystem. In fact, there could be other poles which are cancelled in the denominator of the diagonal transfer function by very near zeros in its numerator and that these poles are not cancelled in other transfer functions belonging to the same subsystem. The measurement of the whole set of transfer functions does not show any strong indication for such a cancellation. The influences of nonlinearities and other deviations from the assumed behavior make it inadvisable to overinterpret the frequency responses measured (cf. Appendix A.4).

Model order

It is reasonable to adopt a sixth-order model for each of the six available subsystems. This allows the description of the three main modes defined above.
Under the assumption of linearity, the six subsystems are connected in parallel to form an overall multivariable system model with a minimal order of 36.

**Frequency response at low frequencies**

The transfer functions with acceleration output have a double differentiator behavior which dominates the response from 0 Hz up to the first resonance. For the transfer functions with position output, the response from 0 Hz up to the first resonance is that of a constant amplification factor.

**Relative order of the transfer functions**

As can be shown by simple multi-body modeling, the relative order of the acceleration transfer functions is zero in case of no elasticity between the actuator and the accelerometer. A single elasticity between the actuator and the accelerometer leads to a phase loss of 90°. The relative order of this transfer function is one. Two elasticities lead to a phase loss of 180°, meaning that the relative order is two, and so on.

Considering the active roller guide shoe with the two heavy actuator stators mounted on its cover (cf. Figure 3.1) and with the relatively weak vertical support between the cover and the base, it is reasonable to expect elasticities between the actuator and the accelerometers mounted on the cabin frame.

The frequency response measurements from the diagonal transfer functions with acceleration output are the less affected by friction and other nonlinearities. Their phase responses may be trusted to check the existence of elasticities between the actuators and the accelerometers. Referring to the results in Appendix A.4, the actuator resonance reduces the phase to −180° in all of these transfer functions. This means that there are two elasticities between each actuator and its nearest accelerometer. This result has been confirmed with the experimental observation that a proportional controller for the loop formed by any of the diagonal transfer functions with acceleration output leads to a strong excitation of the actuator resonance such that the system becomes practically unstable. This confirms that the value of the phase angle is near −180° in the vicinity of the actuator resonance and that the relative order of the transfer functions with acceleration output is two.

The position sensors are mounted very near to the actuators (cf. Figure 3.1), so that they can be considered as measuring the actuator stroke directly. In this case there
is no elasticity between the actuator and the position sensor and the relative order of the transfer functions with position output is two (cf. Appendix A.5).

**Coupling**

The model proposed above is capable of describing all couplings which may exist between the various inputs and outputs of the system. However, the structure of each individual car determines the transfer functions, where coupling occurs and how significant it is.

**Minimum-phase and nonminimum-phase transfer functions**

In such a complex system having a large number of inputs and outputs with considerable degrees of coupling between them, the measured signals are influenced by the translation and rotation of the various bodies. It is therefore reasonable to expect that many of the transfer functions in (5.1) are nonminimum-phase. This applies in particular to the off-diagonal transfer functions influenced by the rotations of the cabin and its frame.

A theoretical determination of the transfer functions that possess nonminimum-phase behavior needs a physical model of the system which is not available here. For this reason these transfer functions have to be determined experimentally. A nonminimum-phase transfer function expresses itself in the shape of its phase response as well as in the form of its step response [31]. In the case of an odd number of positive zeros, the step response shows an inverse response. Large friction forces may cause this inverse response to appear like a time delay. Since the system does not have any noticeable time delay, a delayed response is interpreted as a sign of nonminimum-phase behavior.

Appendix A.1 contains measurements of all step responses of the system for acceleration and position outputs. The inverse response is easy to recognize for the nonminimal-phase transfer functions with position output. For the nonminimal-phase transfer functions with acceleration output, the steady-state value of the output signal is always zero. The inverse response is therefore not easy to recognize. The measured step responses shown in Appendix A.1 are used together with the frequency responses in the Appendices A.4 and A.5 to decide whether a transfer function is minimum or nonminimum-phase.
5.3.3 Effect of parameter variation

The only physical parameter in the elevator car system which is a subject of considerable variations during operation is the cabin mass. For the elevator car in the test rig, the mass of the empty cabin is 1800 kg and its payload can reach a maximum of 1600 kg, i.e., nearly 90% of the cabin mass.

For each of the system transfer functions the dynamics are dominated by the three vibration modes defined in Section 5.3.1 above. The first mode describes the motion of the cabin and its frame as one mass between the guide rollers. The natural frequency of this oscillation can be approximated as the square root of the elasticity of the guide roller spring divided by the mass. By calculating this value for the mass of the empty cabin as well as the fully-loaded cabin the difference in this natural frequency is about 20%. As the second resonance belongs to the vibration mode between the cabin and its frame, the value of the corresponding natural frequency will be near to the square root of the stiffness of the rubber pads divided by the mass of the lighter body, which is the cabin frame. For this reason, the effect of the variation in the payload on the second mode will be even less than its effect on the first mode. As the third resonance belongs to the equivalent moving mass of the actuator vibrating between the tire and the spring, the natural frequency of this mode is almost not affected by the variation of the cabin mass.

The expected changes in the system dynamics due to the maximum possible variation in the payload do not pose a problem for a controller that does not invert the plant dynamics. Furthermore, the maximum payload is equivalent to the mass of twenty-one persons standing on the cabin floor. Due to the small area of this floor (3.325 m²), this situation is very uncomfortable and hence unlikely. For this reason, the nominal plant to be considered for identification and controller design is the elevator car with an empty cabin.

5.4 Identification method

5.4.1 Time and frequency domain identification algorithms

The methods of system identification can be divided into time domain and frequency domain methods. Examples of algorithms available for time domain identification are the autoregressive exogenous (ARX), the autoregressive moving
average exogenous (ARMAX), and the maximum likelihood (MLL). The first two
methods are well explained in [35] with applicable algorithms implemented in [36].
An algorithm for the third method is available in [43]. The methods of frequency
domain system identification are based on the experimental determination of the
system frequency response. A complex curve fitting algorithm is then utilized to
obtain the parameters of the numerator and the denominator of the transfer function
to be identified ([1], [34], [46], [48], and [52]).

After a thorough investigation of both types of identification methods the frequency
domain method was adopted. The main reasons for this choice are as follows:

- It allows to impose the known physical characteristics of the system such as
  stability, shape of the frequency response, minimum or nonminimum-phase
  responses, etc.

- It permits efficient control of the model order which is necessary to avoid
  overmodeling and undermodeling.

5.4.2 The complex curve fitting algorithm

The heart of the identification method applied here is the complex curve fitting
algorithm employed to find the parameters of the numerator and denominator poly-
nomials for an experimentally determined frequency response. The complex curve
fitting algorithm utilized is mainly the one programmed in [33]. It is based on the
contributions presented in [34] and [48] to form a linear least-square minimization
problem for an overdetermined set of linear equations. The least square solution of
the equations set yields the initial values of the parameters. These values can then
be enhanced iteratively using the Gauss-Newton gradient procedure.

The following versions of the algorithm needed to be developed by modifying the
one programmed in [33]:

1) For the identification of the diagonal transfer functions with acceleration
   output, the parameters of a sixth-order denominator and a fourth-order numer-
   ator are identified fulfilling the following conditions:

   - The denominator is stable.

   - The numerator has two roots at 0 Hz.
- All poles are complex conjugate.
- All zeros are complex conjugate.

To ensure the stability of the denominator, the signs of any positive real parts of the its roots are inverted to get an equivalent stable polynomial. The second condition needs a slight modification in the algorithm to force the identification of a numerator whose first two coefficients equal zero. The third and the fourth conditions are reached by correctly selecting the weighting functions involved.

2) For the identification of the off-diagonal transfer functions with acceleration output, the denominator is known and the numerator is fourth-order with two roots at 0 Hz.

3) For the identification of the transfer functions with position output, the denominator is known and the numerator is fourth-order. For the transfer functions $G_{p_{11}}(s)$, $G_{p_{22}}(s)$, $G_{p_{33}}(s)$, $G_{p_{44}}(s)$, $G_{p_{55}}(s)$ and $G_{p_{66}}(s)$ the four zeros are complex conjugate with negative real parts.

In each of the above cases, the input to the algorithm consists of the frequency response data, the order of the two polynomials, and the vector of the weights at the same frequency points as the frequency response data. This vector is generated as the amplitude response of a transfer function defined by the user (cf. Appendix A.3). These transfer functions are selected to influence the identification results, so that the required characteristics of the identified polynomials including minimal and nonminimal-phase behavior are reached.

5.4.3 Identification sequence

Following the excitation of the system the input and output signals of each of the system transfer functions are available in time domain. Using fast Fourier transform (FFT), the frequency response of the different transfer functions as an amplitude and a phase response is constructed from the time domain data [33].

The diagonal transfer functions with acceleration outputs are identified at first. With the identification of these transfer functions, all denominators of the model are known. The identification of all off-diagonal transfer functions may then follow.
After the identification of the individual transfer functions the overall system model is formed according to (5.1). Using balanced truncation [3] it is possible to get the minimal order model in state space presentation to be used for the design of the controller.

5.5 Identification experiment

5.5.1 The excitation signal

To determine the various frequency responses, an excitation signal in the time domain is used as input to each one of the actuators separately, while all output signals are measured.

Requirements for the excitation signal

1) Frequency content:

The model to be identified describes the system dynamics in the frequency range from 0 Hz up to the actuator resonances. These resonances lie between 37 and 52 Hz. At frequencies higher than 60 Hz the system is excited by the vibrations of the hydraulic unit which covers the range from 100 to 320 Hz. These vibrations represent an unknown input noise to the system and the input/output relationship required for the modeling is wrong. Therefore, it is not required to excite the system beyond 60 Hz.

2) Amplitude:

The excitation signal must adequately excite the system without forcing it out of its linear range of motion. This implies that none of the actuators touches either of its ends during the excitation. Since the loads on the different actuators are not the same, the optimal amplitude for each of them is different and has to be determined by trial and error.

3) Signal duration:

The signals to be measured in real-time during the system excitation are the input signal, the six acceleration outputs, and the six position outputs. At each sampling period the values of these signals are stored on the real-time computer. At the end of the excitation run, they are transferred to the host computer where
the identification takes place. Due to the limited memory of the real-time computer (cf. Section 3.3.6), it is important to limit the duration of the excitation signal. The necessary duration of the excitation signal is taken to be ten times the period of the lowest frequency it contains, which is 1 Hz. This means that it is necessary to count on a signal that has a duration of at least ten seconds.

**Design of the excitation signal**

The two main signal types for excitation are the multisine and the binary signals. Since the actuators used do not impose any restrictions on the command signal duration nor on its sign, it is possible to use a multisine signal for the excitation. This gives a better reproduction of the desired power distribution.

The algorithms used to generate the multisine excitation signal are those programmed in [32]. The main feature of this signal is its capability to adequately excite the system in the specified frequency range. The most delicate part of this range is the portion near the first resonance due to the large amplitudes required. In the range above 10 Hz the required amplitudes are much smaller.

A practical problem in the realization of this signal by the real-time computer is the fact that it has to be realized using a look-up table [39]. This type of function has a limited number of points according to the available on-board data memory. If a certain frequency must be included in the signal, it is necessary to use a number of points equal to a certain sampling frequency, usually ten times the highest frequency in the look-up table, multiplied by the signal duration. This means that the higher the frequency content of the excitation signal, the higher the number of its points and the more memory its realization will need.

The solution to this problem is to split the excitation signal into two signals to be added. The first one is realized by the look-up table and contains frequencies up to 10 Hz, while the second signal is a band-limited white noise generated on-line and includes the higher frequencies. The sampling frequency of the look-up table signal then is 100 Hz, and its number of points is one thousand. In the specification of the desired amplitude distribution, the lower frequencies should get a higher share of the energy distribution. According to experience, it was found that a linear reduction of the required amplitude from its maximum value at 1 Hz to zero at 10 Hz is a good choice. The resulting signal based on this distribution is shown in Figure
Figure 5.4: Excitation signal with a frequency content from 1 to 10 Hz

Figure 5.5: Excitation signal utilized for identification
5.4. Figure 5.5 shows the final signal utilized for the excitation which is composed of the signal on Figure 5.4 plus the band-limited white noise. The power spectral density for the excitation signal with and without the band-limited white noise is shown in Figure 5.6.

5.5.2 The excitation and measurement program

The block diagram of the excitation and measurement program is shown in Figure 5.7. The numerous amplifiers serve as valves to direct the excitation signal with the proper amplification factor to the intended actuator. Using the arrangement with the reset integrator, it is possible to restart the excitation whenever desired and to repeat it for an unlimited number of runs. The TRACE facility [9] allows to take the desired measurements of the real-time signals with the required sampling frequency. The experiment is repeated for each of the six command forces while the twelve outputs are recorded. The sampling frequency for all measurement is 1 kHz.
Figure 5.7: Real-time program for excitation and measurement
5.6 Identification results

5.6.1 The identified transfer functions

The parameters of all 72 identified transfer functions are given in Appendix A.2. The transfer functions utilized to generate the weighting vectors used by the complex curve fitting algorithms for the identification of the system parameters are all defined in Appendix A.3.

To verify the identification results in the frequency domain, the frequency responses generated from the identified transfer functions are compared with the corresponding measured ones. Appendix A.4 shows these comparisons for the 36 transfer functions with acceleration outputs. The same information for the 36 transfer functions with position outputs is shown in Appendix A.5.

5.6.2 The identified model

To show the dynamics of a multivariable system, the singular values are plotted versus the frequency. They are obtained from a singular value decomposition of the system transfer matrix at each frequency [3]. Figure 5.8 shows the singular values of the system with the acceleration outputs, while Figure 5.9 shows the singular values for the position outputs.

5.6.3 Model validation

The quality of an identified model is usually checked by time domain validation. A comparison between the time response of the identified transfer functions with the output signals from the system is a good measure of the model quality. Since the identification is done in the frequency domain, no new measurements are required for the validation in the time domain (cross validation). The validation results, for the acceleration and position outputs, are shown in the Appendices A.6 and A.7, respectively.

5.7 Comments and conclusion

The following comments on the identified model are based on the results shown in the Appendices A.1 through A.7.
Figure 5.8: Singular values of the identified model with acceleration outputs

Figure 5.9: Singular values of the identified model with position outputs
Model order

A higher quality of matching between the measured and the identified frequency responses is attainable if the order of the identified transfer functions is increased. An increase in the order of each of the subsystems from six to eight increases the final system order from 36 to 48 which has a strong impact on the design procedure. The resulting order of the acceleration controller increases accordingly and the design procedure is more vulnerable to numerical errors. A sixth-order model for each of the subsystems is considered to be a good compromise between desirable accuracy and effective applicability.

Range of validity of the identified model

The identified model describes the elevator car dynamics from 0 Hz up to the actuator resonances occurring between 37 and 52 Hz. These are the highest frequencies important for the design of the controller. The inclusion of actuator resonances in the system model is very important for the stability of the controlled system, since they are associated with the phase reduction from zero to $-180^\circ$ (cf. Appendix A.4).

Coupling between the inputs and outputs of the system

The strength of the coupling between the various inputs and outputs of the system can be quantified by the amplitude responses of the corresponding transfer functions. The higher the coupling, the higher the magnitude of these transfer functions. This may be seen clearly in the results in the Appendices A.4 and A.5. The step responses contained in Appendix A.1 provide another useful indication of the coupling strength.

Quality of the identified transfer functions

It should be noted that the limiting factor for the quality of the identified transfer functions is the quality of the frequency response data rather than the accuracy of the curve fitting algorithm. It was therefore not necessary to program any other algorithm based on another method (e.g., the algorithms described in [18] or in [49]). Considering the construction of the frequency response data from time domain input and output signals, the algorithms available in [33] and based on FFT
are found adequate. There was no need to develop any other algorithms based on other techniques (e.g., the one described in [19]).

Quality of the time domain results

Referring to Appendices A.6 and A.7, it is clear that the higher the influence of the actuator on a certain output signal, the better the time response of the identified transfer function is. For the transfer functions with very weak input-output coupling, the results in the frequency domain as well as in the time domain are rather irrelevant. It is then possible to assume no coupling between those inputs and outputs. However, since their influence on the numerical accuracy of the controller design algorithm was not noticeable, these transfer functions were not eliminated from the model.

Conclusion

The identification method presented here is considered to be the most appropriate one for the given system fulfilling the modeling requirements and leading to a model useful for the design of the controller. The model obtained forms the basis for the controller design documented in Chapter 6.
6 Controller Design

6.1 Design requirements

6.1.1 Controller structure

The primary goal of the controller is to suppress the vibrations in the frequency range between 1 and 20 Hz [54]. As stated in Chapter 3, acceleration feedback is used to achieve this goal. The secondary goal of the controller is to correct the orientation of the car in the presence of asymmetric cabin loading using position feedback. This is needed to optimize the car position and attitude relative to the two guide rails in order to provide sufficient actuation distance for each actuator. The mechanical blocking of the actuators is then prevented. The controlled system therefore has two distinct feedback loops for the acceleration and the position measurements. Both of these loops are designed according to their own requirements for the bandwidth, loop shape, and performance.

The selected structure of the control system is shown in Figure 6.1. It is composed of a first loop for acceleration feedback and a second loop for position feedback. The acceleration controller is designed and tested first, before the design of the position controller begins. The separation of the design problem into two cascaded loops is a great simplification, since it reduces the effects of the numerical errors which are expected when dealing with such a high-order multivariable system.

6.1.2 The control design problem

The control design problem dealt with here has many particular features which must be taken into consideration in the selection of the design method. These features are summarized in the following points:
1) Low-frequency disturbances

The accelerometer measurements contain a certain zero error which is not known precisely. Furthermore, they are strongly influenced by the earth’s gravitation. A tilt angle of one degree, which is very common for an accelerometer mounted on the car, results in a measured acceleration error of about 17 mg. Knowing that 40 mg may represent the peak acceleration in a typical high-speed elevator not equipped with an active damping system and that the same value in a bad elevator may reach just 100 mg, the strong influence of low-frequency disturbances on the controlled system is evident. The measured acceleration due to the earth’s gravitation changes its value with the tilt angle of the sensor and hence with that of the elevator car. The change in the car attitude is caused by the variation in the center of gravity of the payload. It is therefore considered as a low-frequency disturbance. The same classification applies for the sensor zero error which is mainly temperature dependent.

2) High-frequency disturbances

Due to the proportionality of its amplitude to the square of the frequency, the acceleration measurement is very sensitive to high-frequency disturbances. Although the measurement noise of the sensor itself is very low (cf. Section 3.3.3), the system
Controller Design

is excited by many high-frequency input noise sources, such as the actuator drivers or the vibrations produced by the hydraulic pump. These disturbances are present up to 350 Hz and are not sufficiently suppressed by the anti-aliasing filter. If the acceleration controller does not adequately suppress the vibrations in this range, the high-frequency disturbances are amplified leading to instability.

3) Conflict between the position and the acceleration loops

The acceleration measurements represent the absolute acceleration of the mounting points in inertial space. Hence, they can be used directly in a feedback loop to suppress the vibrations. The position measurements correspond to the relative positions between the cabin frame and the guide rollers. Neglecting the tire elasticity, those measurements may be considered as the distances between the cabin frame and the two guide rails. Nulling the position error signals means that the elevator car is forced to follow the average line of the two guide rail profiles. This will deteriorate the ride quality in the car since the guide rail profiles are the main source of vibrations. For this reason the position feedback, though necessary for the functionality of the active system, is conflicting with the acceleration feedback. This conflict may result in a considerable deterioration of the vibration suppression. This is a serious problem since the position feedback loop produces command force signals to the actuators with far higher amplitudes than the acceleration loop. The solution to this problem is to limit the bandwidth of the position controller to a frequency lower than the active-band of the acceleration controller. In this way, a frequency domain separation between the effectiveness ranges of the two loops is achieved. The position controller is thus defined to dominate the command force signals in the frequency range below 1 Hz, where humans are not sensitive to vibrations, while the acceleration controller takes over in the frequency range above 1 Hz.

4) Stability of the controlled system

The primary goal of any controller is to ensure the stability of the control system. Although the elevator car itself is a dissipative system, a feature implying stability [25], it can easily be destabilized if the controller does not take care of the lightly damped actuator poles where the phase crossover of the open loop system occurs. These poles are easily excited by the command signals since they do not lie at
frequencies sufficiently higher than the frequency range where the controller is required to achieve performance. The most efficient way not to excite those resonances is to include their inversion in the controller dynamics. However, the controller is not allowed to invert the system dynamics in the lower frequency range where vibration suppression is required. In case of an inversion, the changes in these dynamics caused by the variations in the payload result in an imperfect pole-zero cancellation. This means that the dynamics thought to be cancelled still exist leading to a loss of performance.

5) Performance limitation

The acceleration controller has a relatively narrow active-band in the frequency range where performance is required and a deep stop-band in the frequency range of the actuator resonances. Since the frequency range separating the active-band from the stop-band is not sufficiently large, a certain amount of contradiction exists in the formulation of the performance requirements for this controller. On the one hand, it is required to be active up to 20 Hz, while on the other hand it is not allowed to produce any signal in the frequency range from 37 to 52 Hz. The fact that 37 Hz is not sufficiently higher than 20 Hz makes it difficult to achieve both goals simultaneously. This is an important source of limitation for the achievable controller performance.

6.1.3 Acceleration loop specifications

1) Performance

According to the specifications given by the industrial partner, the controller is required to reduce the acceleration magnitudes in the frequency range from 1 Hz to 10 Hz by a factor of 10. Starting from 10 Hz, this factor is allowed to reduce linearly until it reaches unity at 20 Hz [54].

2) Loop shape

In order not to react to low-frequency disturbances and not to excite any of the high-frequency disturbances, the acceleration controller must have a band-pass frequency response to be effective from 1 to 20 Hz only. Experience has shown, that a high-pass behavior with a second-order slope is required at frequencies lower
than 1 Hz, whereas a low-pass behavior with a second-order slope is required at frequencies higher than 20 Hz.

3) Stability and robust performance

As explained in Section 6.1.2, the controller is not allowed to invert the dynamics in the frequency range where it is required to achieve performance. Referring to Chapter 5, this is the range in which the first two resonances occur in each of the available transfer functions. These resonances are affected by the changes in the payload of the cabin which is subject to continuous variation during operation. On the other hand, the controller has to invert the actuator dynamics in order to ensure that they will not be excited by the control signals.

6.1.4 Position loop specifications

1) Performance

The position control loop is active in the frequency range from 0 to 1 Hz. The loop gain at these very low frequencies must be high enough to ensure a full exploitation of the available maximum forces of the actuators.

2) Loop shape

In the vicinity of their cross-over frequency the singular values of the position controller must have a declining slope order of at least three in order to ensure a minimal interference with the acceleration loop. For the following reasons, this controller should not possess poles that lie at 0 Hz:

a) The realization of a controller possessing poles at 0 Hz necessitates a mechanism to avoid numerical overflow during the start-up phase or when the loop is not functioning properly. In case of a simple PI or PID controller, this is achieved using common anti-windup techniques. In the case of a multivariable robust controller, this is more complicated and adds to the difficulties of the design and implementation procedures.

b) The aim of the position controller is not to eliminate the position errors but just to optimize the car inclination so that each actuator has sufficient actuation distance. The most important aspect therefore is not the nulling of the position
error signals but to ensure that the maximum available actuator force will be utilized to maintain the optimal car attitude.

3) Stability and robust performance

In comparison with the acceleration controller, the bandwidth of the position controller is limited to a considerably low frequency (1 Hz). Therefore, the effect of the position controller on the system dynamics should be minimal. This allows a large order reduction of this controller without either endangering the system stability or reducing the performance achieved by the acceleration loop.

6.2 Control design method

6.2.1 Selection of the controller design method

Due to the strong coupling between most of the degrees of freedom of the system, a multivariable design method must be utilized. Although it is possible to control each single degree of freedom of the car separately, it is not possible to control the whole system using a multitude of SISO controllers in a decentralized way assuming decoupling between the degrees of freedom. Such a control system becomes unstable, as experiments have shown.

As stated in Section 6.1.3, the required acceleration controller must have a certain band-pass loop shape in the frequency range between 1 and 20 Hz which cannot be reached by common pole placement [61] or by linear quadratic optimal controller design methods [15].

The LQG/LTR robust controller design method [16] offers robust stability but it has the tendency to invert the plant and therefore does not guarantee robust performance.

The two robust control design methods found suitable for application here are the \( H_2 \) and the \( H_\infty \) methods [16]. Besides robust stability, both methods allow a considerable amount of freedom in the selection of the desired loop shape of the controlled system by appropriate choice of the involved weighting functions.

The \( H_2 \) method minimizes the \( H_2 \) norm of the closed loop transfer function, while the \( H_\infty \) method minimizes its \( H_\infty \) norm. Experiments with \( H_2 \) and \( H_\infty \) controllers
have shown that there is practically no difference in the performance achieved. The algorithms available for the $H_\infty$ method [3] provide a good measure of the design success, since they show the value of $\gamma$ after each iteration step (cf. (6.4) below). For this reason the $H_\infty$ method has been adopted.

The $\mu$-synthesis robust design method [16] allows to treat the influence of the variation in the physical parameters of the system on the plant dynamics. It was not found suitable for this controller design problem since it requires physical modeling of the system. Another disadvantage of this design method is the typically high-order of the resulting controller. As the analysis in Chapter 5 has shown, the effects of the changes in the payload, which is the only physical parameter capable of changing during operation, on the system dynamics are limited, and there is no necessity to apply $\mu$-synthesis.

6.2.2 Inverting and non-inverting $H_\infty$ designs

The $H_\infty$ controller design procedure may or may not result in controllers which include an inversion of the plant, depending on the problem formulation and the shape of the weighting functions used [5].

A plant-inverting $H_\infty$ problem formulation

As an example for a plant-inverting $H_\infty$ design, the so called $S/KS/T$ problem formulation, in which the sensitivity $S$ and the complementary sensitivity $T$ are weighted, will be analyzed here.

For a given plant $G$ connected as shown in Figure 6.2 to the three dynamic weighting functions $W_e$, $W_y$, and $W_u$, the relationship between the input signal $w$ and the three output signals $z_e$, $z_u$, and $z_y$ is given by:

$$
\begin{bmatrix}
  z_e \\
  z_u \\
  z_y
\end{bmatrix} = [T_{zw}] \cdot [w] 
$$

(6.1)

The transfer matrix $T_{zw}$ between the input and the output of the augmented system including the controller $K$ is:
We(s) \quad \text{ze(s)}
Wu(s) \quad \text{zu(s)}
Wz(s) \quad \text{zw(s)}

Figure 6.2: S/KS/T Scheme for the design of an H\(_\infty\) controller inverting the plant

The same equation may be written as:

\[
T_{zw} = \begin{bmatrix}
W_e \cdot [I + G \cdot K]^{-1} \\
W_u \cdot [I + G \cdot K]^{-1} \\
W_y \cdot G \cdot K \cdot [I + G \cdot K]^{-1}
\end{bmatrix}
\] (6.2)

The same equation may be written as:

\[
T_{zw} = \begin{bmatrix}
W_e \cdot S_e \\
W_u \cdot K \cdot S_e \\
W_y \cdot T_e
\end{bmatrix}
\] (6.3)

with \(S_e\) and \(T_e\) as the sensitivity and the complementary sensitivity, respectively, for the loop breaking point \(e\).

The aim of the H\(_\infty\) algorithm is to design a controller \(K\) which results in the transfer matrix \(T_{zw}\) satisfying the following inequality:

\[
\|T_{zw}\|_{\infty} \leq \gamma
\] (6.4)

where \(\gamma\) is a design parameter usually set to unity.

Equation (6.3) shows that the controller leads to a sensitivity \(S_e\) which is more or less equal to the inversion \(W_e^{-1}\) of its dynamic weighting and to a complementary sensitivity \(T_e\) that is very near to the inversion \(W_y^{-1}\) of its dynamic weighting. With
a proper selection of the dynamic weights $W_e$ and $W_y$, it is possible to impose the desired frequency response on the sensitivity and the complementary sensitivity. However, it is important to guarantee that the augmented plant, composed of the plant $G$ and the dynamic weighting functions $W_e$, $W_y$, and $W_u$, satisfies the necessary conditions for this design method [16].

The complete design specifications can be given in the two weights for the sensitivity $W_e$ and the complementary sensitivity $W_y$. The weighting $W_u$, which is selected to be a small constant value, does not influence the resulting controller but is necessary to fulfill the mathematical conditions for the design algorithm.

The condition given in (6.4) with $\gamma = 1$ applies for all three components of $T_{zw}$. Due to the small value of $W_u$, the condition involving the second component is always satisfied. The condition involving the first component may be analyzed to prove the plant-inverting nature of this problem formulation (cf. [5], p. 88).

*An $H_\infty$ problem formulation without inversion of the plant*

It is nevertheless possible to formulate the $H_\infty$ controller design problem so that the controller does not include an inversion of the plant dynamics which lie in the frequency range where performance is achieved. An inversion of the plant dynamics lying outside this frequency range is not harmful but rather useful since it can be used to invert the dynamics of the actuators.

In the weighting scheme shown in Figure 6.3, the sensitivity multiplied by the plant $G \cdot S_u$ (identical to $S_e \cdot G$ since $S_u = [I + K \cdot G]^{-1}$) and the complementary sensitivity $T_e$ are weighted. It is therefore called the GS/T scheme.

The resulting transfer matrix between the input and the output of the augmented plant with the controller is then given as follows:

$$
\begin{bmatrix}
    z_u \\
    z_y
\end{bmatrix} =
\begin{bmatrix}
    -W_u \cdot K \cdot G \cdot [I + K \cdot G]^{-1} \cdot W_y \\
    [I + G \cdot K]^{-1} \cdot G \cdot W_y
\end{bmatrix}
\begin{bmatrix}
    W_u \cdot K \cdot [I + G \cdot K]^{-1} \cdot W_r \\
    G \cdot K \cdot [I + G \cdot K]^{-1} \cdot W_r
\end{bmatrix}
\begin{bmatrix}
    v \\
    r
\end{bmatrix}
$$

(6.5)

which is equivalent to
Figure 6.3: GS/T scheme for an $H_\infty$ controller design without plant inversion

\[ \begin{bmatrix} z_u \\ z_y \end{bmatrix} = \begin{bmatrix} -W_u \cdot T_u \cdot W_v & W_u \cdot K \cdot S_e \cdot W_r \\ S_e \cdot G \cdot W_v & T_e \cdot W_r \end{bmatrix} \begin{bmatrix} v \\ r \end{bmatrix} \]  

(6.6)

with:

\[ T_{zw} = \begin{bmatrix} -W_u \cdot T_u \cdot W_v & W_u \cdot K \cdot S_e \cdot W_r \\ S_e \cdot G \cdot W_v & T_e \cdot W_r \end{bmatrix} \]  

(6.7)

The elements of $T_{zw}$ which include $W_u$ satisfy the minimization condition of (6.4) for $\gamma = 1$ since $W_u$ is chosen as a constant of a very low value. It is only necessary for a non-singular $H_\infty$ problem formulation. The function $W_v$ is used to weight the sensitivity multiplied by the plant $S_e \cdot G$, while $W_r$ is used to weight the complementary sensitivity $T_e$. The resulting $H_\infty$ controller does not include an inversion of the plant dynamics which lie in the frequency range where performance is achieved (refer to [5], p. 92f). This frequency range is specified by the two weighting functions $W_r$ and $W_v$. However, the controller inverts the plant dynamics which lie outside this frequency range. By limiting the range of the required performance to a frequency below the resonances of the actuators, the resulting controller contains an inversion of these resonances although it does not invert any other part of the plant dynamics. The desired requirements concerning performance and stability are then achieved.
6.2.3 Algorithms for controller design and manipulation

The algorithms for the $H_\infty$ control design as well as for the further manipulation of the controllers obtained, such as balancing and order reduction, are those programmed in [3]. Those algorithms have proven to be reliable for such a complex problem dealing with a high-order system which has a large number of inputs and outputs.

The controller design algorithms in [3] are mainly for continuous-time systems leading to continuous-time controllers. On the other hand, the identification procedure adopted provides a system model which is also continuous-time. Moreover, the available discrete-time $H_\infty$ algorithm in [3] first transforms the given discrete-time system into a continuous-time system and then designs a continuous-time controller to be discretized. The discrete-time design is thus not distinguishable from the continuous-time one. For these reasons, it was found more convenient to utilize a controller design method for continuous-time systems and then to discretize the resulting controller before the realization on a digital computer (refer to [5] Lemma 2.4 and remark following it (p. 39f)).

6.3 Acceleration control design

6.3.1 The acceleration feedback loop

As stated above, first the acceleration controller is designed, implemented, and tested, followed by the position controller. Using the model identified in Chapter 5 as basis for the design, the singular values of the plant with the six command force signals as inputs and the six acceleration measurements as outputs are shown in Figure 5.8.

6.3.2 Weighting functions for the acceleration loop

The weighting scheme used for the acceleration loop is shown in Figure 6.4. Only the acceleration outputs of the plant $G_p$ are fed back, and the number of inputs is equal to the number of outputs. The acceleration feedback controller $K_a$ found by the $H_\infty$ algorithm must satisfy the conditions given in (6.4) for the transfer matrix $T_{zw}$ of the augmented plant for $\gamma = 1$ (cf. (6.7)).
With a deliberate selection of the two weighting functions $W_v$ and $W_r$, it is possible to achieve the desired band-pass nature of the acceleration controller according to the specifications summarized in Section 6.1.3. However, the following additional points must be respected in any choice of the two weighting functions:

- The two functions are not allowed to conflict. As it is known that the relationship between the sensitivity and the complementary sensitivity for all frequencies is described by

$$S_e + T_e = I \quad (6.8)$$

the weighting functions $G \cdot W_v$ and $W_r$ for $S_e$ and $T_e$, respectively, should not intersect above the 0 dB line ([5], p. 110).

- Since the controller's order is equal to the order of the augmented plant, it is important to utilize weighting functions with the lowest possible order that allows the fulfillment of the specifications.

- A high-order slope of the weighting functions results in a resonance at the crossover frequency of the loop gain. As experience has shown, $W_r$ is more influential than $W_v$. It is therefore not advisable that it has a slope order higher than three at any frequency range ([5], p. 111 and [63]).

- The loop gain of the weighting function $W_v$ is a direct measure of the performance demanded from the controller. Due to the small frequency band of the
acceleration control, it is not possible to increase the performance required without increasing the slope order. That in turn increases the sensitivity resonance at the crossover frequency of the controlled system.

The acceleration control presented here uses identical weighting functions for all channels. The transfer matrices $W_v$, $W_r$, and $W_u$ are all diagonal with the same transfer functions $W_{dv}$, $W_{dr}$, and $W_{du}$ along their diagonals. Actual values of these transfer functions are as follows:

$$W_{d_v} = \frac{k_v \cdot s}{(s-p_{v1}) \cdot (s-p_{v2}) \cdot (s-p_{v3}) \cdot (s-p_{v4})}$$

$$k_v = 4.1163 \times 10^{-12} \quad p_{v1} = -0.5 \cdot 2\pi$$
$$p_{v2} = -0.5 \cdot 2\pi$$
$$p_{v3} = -(3 - 3i) \cdot 2\pi$$
$$p_{v4} = -(3 + 3i) \cdot 2\pi$$

$$W_{d_r} = \frac{k_r \cdot (s-z_{r1}) \cdot (s-z_{r2}) \cdot (s-z_{r3}) \cdot (s-z_{r4}) \cdot (s-z_{r5})}{(s-p_{r1}) \cdot (s-p_{r2}) \cdot (s-p_{r3}) \cdot (s-p_{r4}) \cdot (s-p_{r5})}$$

$$k_r = 1.667 \times 10^6 \quad z_{r1} = -(0.3 - 0.5i) \cdot 2\pi \quad p_{r1} = -(1 \times 10^{-5} - 1 \times 10^{-5}i) \cdot 2\pi$$
$$z_{r2} = -(0.3 + 0.5i) \cdot 2\pi \quad p_{r2} = -(1 \times 10^{-5} + 1 \times 10^{-5}i) \cdot 2\pi$$
$$z_{r3} = -40 \cdot 2\pi \quad p_{r3} = -1 \times 10^4 \cdot 2\pi$$
$$z_{r4} = -(40 - 40i) \cdot 2\pi \quad p_{r4} = -(1 \times 10^4 - 1 \times 10^4i) \cdot 2\pi$$
$$z_{r5} = -(40 + 40i) \cdot 2\pi \quad p_{r5} = -(1 \times 10^4 + 1 \times 10^4i) \cdot 2\pi$$

$$W_{d_u} = 5 \times 10^{-5}$$

Figure 6.5 shows the singular values of all three weighting functions used for the design of the acceleration controller given here. It also shows the singular values of the plant with acceleration output $G_a$ multiplied by $W_v$ which is a good measure of the performance demanded. The singular values of $G_a \cdot W_v$ indicate that the highest
Figure 6.5: Singular values of the weighting functions for the acceleration controller

Figure 6.6: Singular values of the transfer matrix $T_{zw}$ resulting from the acceleration controller design
**Figure 6.7**: Singular values of the acceleration controller $K_a$

**Figure 6.8**: Singular values of the sensitivity $S_e$ and the complementary sensitivity $T_e$ of the system with acceleration controller
required performance is in the frequency range of the first group of resonances, while in the frequency range of the second group of resonances, less performance is demanded. The third group of resonances, which belongs to the actuators, lies adequately below the \( W_v \) lines, so that its inversion occurs. It is not possible to achieve the same performance in the frequency range of the second group of resonances as in that of the first group, since the second group lies very near to the actuator resonances inverted by the controller (cf. Section 6.1.2).

The \( W_r \) function is chosen to have an inverted band-pass structure between 0.5 and 40 Hz. The low-frequency portion has a second-order slope and the high-frequency portion has a third-order one. This is necessary to obtain the required slope order of the controller in both frequency ranges. For the \( W_v \) function, it was not found necessary for its low-frequency portion to have a second-order slope, as the case with \( W_r \), and that a first-order slope is sufficient. In this way an unnecessary rise of the controller’s order is avoided.

Regarding the singular values of the system model with acceleration outputs (cf. Figure 5.8), it is clear that those values are increasing with the frequency up to the third group of resonances. This tendency to rise is due to the fact that the amplitude of the acceleration is proportional to the square of the frequency. This makes it more difficult to limit the bandwidth of the controlled system to 20 Hz. For this reason, the high-frequency portion of the \( W_v \) weighting has a third-order slope, conforming with the slope order of \( W_r \) at the same frequency range. Both slope orders are rather high presenting no advantage with regard to the numerical difficulties while implementing the design algorithm. The singular values of the function \( G_a \cdot W_v \) in Figure 6.5 show how the bandwidth of the controlled system is limited with the proposed choice of \( W_v \).

### 6.3.3 The acceleration controller

Figure 6.6 shows that the maximum value of any of the twelve singular values of \( T_{zw} \) does not exceed unity. This is an indication of the fulfillment of the assumption made in Section 6.2.2. The group of singular values with higher values is the one related to the performance achieved. It corresponds to the functions \( S_e \cdot G \cdot W_v \) and \( T_e \cdot W_r \). The group with lower values corresponds to the functions multiplied with \( W_a \) which has a constant low value.
Figure 6.7 shows the singular values of the acceleration controller $K_a$ resulting from the $H_\infty$ design. They indicate that the desired band-pass nature of this controller has been successfully achieved. Concerning the response at very low frequencies, the controller has a high-pass nature with a second-order slope up to 0.05 Hz, which is sufficient to avoid the feedback of low-frequency disturbances. The amplification of the controller reduces gradually from between 20 and 40 dB at 0.05 Hz until it reaches 0 dB in the frequency range between 7 and 15 Hz. Due to the necessity to limit the order of the weighting functions, steep changes in the performance are not possible. For this reason, the gain of the controller must decline gradually. It is thus not possible to achieve the same controller effectiveness in the frequency range from 0.05 to 15 Hz. Complicated weighting functions are not only disadvantageous because they produce higher order controllers, but also because they result in a complicated problem formulation which is vulnerable to numerical errors during controller design. The amplification of the controller at higher frequencies is quite low in order to suppress the high-frequency disturbances. In the frequency range between 37 and 52 Hz, the controller has the desired stop-band. This stop-band results from the inversion of the third group of resonances which belongs to the actuators. As required, the controller does not include any inversion of the low-frequency system dynamics.

6.3.4 The acceleration-controlled system

Figure 6.8 shows the sensitivity $S_e$ and the complementary sensitivity $T_e$ of the system controlled with the acceleration controller $K_a$. A low sensitivity at a certain frequency range indicates controller effectiveness and means that disturbance rejection is present. The lowest value of the sensitivity lies between $-7$ and $-23$ dB. It occurs at the frequency range of the first resonance, which is the range where the maximum performance is required. Between this range and the range where the controller is no longer effective (15 Hz), the sensitivity tends gradually to unity. Accordingly, the controller effectiveness at the frequency range of the second resonance is smaller than at the first resonances. Due to the utilization of high-order weighting functions, a slight resonance at the crossover frequency must result ([5], p. 111). However, this resonance is tolerable as long as its value is below a factor of two, it happens in a frequency range where there is no strong excitation, and where it is not required to achieve performance.
6.3.5 Order reduction of the acceleration controller

Before the acceleration controller can be realized, its order is reduced according to the following sequence of tests and operations:

1) At first, the stability of the controller itself as a dynamic system must be checked. This may be verified by ensuring that the maximum value of all real parts of its poles is negative.

2) In order to construct the \( W_r \) weighting as a proper function, it is necessary to use the same number of zeros as poles. For this reason, each of the diagonal elements of \( W_r \) includes three poles that are selected to lie far above the performance range. In the design presented here these three poles lie at 10000 Hz. The \( H_\infty \) algorithm produces controllers that contain a copy of these poles. The resulting controller therefore has twelve poles located at 10000 Hz. These are not necessary either for the performance, nor for the stability. Therefore, they should be eliminated for controller realization on a digital controller with a limited real-time calculation power. In order to eliminate these poles it is possible to divide the controller into two parallel systems. The first one includes the dynamics up to the limits of realization, for instance 200 Hz, while the second one includes the dynamics at higher frequencies (cf. [5], Appendix A.2). The system that includes the higher dynamics can be replaced by its static gain. The two systems then can be regrouped to form a controller that does not include poles faster than 200 Hz. The resulting controller has an order of 78 instead of 90.

3) A further order reduction of the controller is possible using balanced truncation [44]. For the controller presented here it is possible to reduce the order from 78 to 66 without sacrificing any of its originally desired features. Figure 6.9 shows the singular values of the acceleration controller before and after the final order reduction. Clearly, they are almost identical in the frequency range of interest. The corresponding singular values of the system sensitivity and complementary sensitivity are plotted in Figure 6.10. However, the reduced-order controller must remain stable and result in a stable closed loop system. Otherwise, this order reduction is not allowed.

4) It is very important for the implementation of the controller that its poles do not lie at high frequencies, so that they can be realized by a real-time computer with
Figure 6.9: Singular values of the acceleration controller $K_a$ with full and reduced order.

Figure 6.10: Singular values of the sensitivity $S_e$ and the complementary sensitivity $T_e$ of the system with the reduced-order acceleration controller.
a reasonable sampling frequency. Based on experience, it is a good practice to select the sampling frequency at least eight times higher than the highest frequency of the controller poles. The same applies for the controller zeros which do not lie at infinity. The controller designed above has the highest damped frequency of its poles at 48 Hz and that of its zeros at 56 Hz. Both allow a good realization with a sampling frequency of 600 Hz.

6.4 Position control design

6.4.1 The position feedback loop

The position controller is a cascaded loop to be designed after the activation of the acceleration controller. The same $H_\infty$ problem formulation will be used for the system controlled with the acceleration controller as a new plant with six inputs and six outputs. The augmented plant is shown in Figure 6.11 with the position controller $K_p$ to be designed. The singular values of the plant for the position controller design are shown in Figure 6.12.

6.4.2 Weighting functions for the position loop

In order for the position controller not to conflict with the acceleration controller, the bandwidth of the position loop must be limited to below 1 Hz. It is also important to make sure that the loop gain declines with a sufficient slope order in the frequency range higher than its crossover frequency. In the selection of the weighting functions for the position loop, the same limitations as those mentioned in Section 6.3.2 must be respected. The main difference between the two loops is that the position control has a low-pass frequency response instead of the band-pass frequency response of the acceleration control.

In the design of the position controller shown here, the specifications summarized in Section 6.1.4 were used as guidelines for the selection of the weighting functions. Referring to Figure 6.13, the weighting $W_v$ is selected as a third-order low-pass transfer function in order to ensure a sufficiently steep attenuation of the controller’s effectiveness at the frequencies higher than its performance range. For the weighting $W_r$, a similar second-order weighting scheme suffices. The three multivariable weighting functions $W_v$, $W_r$, and $W_u$ are all diagonal with identical
Figure 6.11: $H_{\infty}$ Weighting scheme for the position controller

Figure 6.12: Singular values of the plant with position outputs and command force inputs after the activation of the acceleration controller
Figure 6.13: Singular values of the weighting functions for position controller design

Figure 6.14: Singular values of the transfer matrix $T_{zw}$ resulting from the position controller design
Figure 6.15: Singular values of the position controller $K_p$

Figure 6.16: Singular values of the sensitivity $S_e$ and the complementary sensitivity $T_e$ of the system with position controller
transfer functions $W_{dv}$, $W_{dr}$, and $W_{d}$ along their diagonals. Actual values of these transfer functions used for the design presented here are summarized below.

$$W_{dv} = \frac{k_v}{(s-p_{vl}) \cdot (s-p_{v2}) \cdot (s-p_{v3})}$$  

$$k_v = 2.6508 \times 10^{-15} \quad p_{vl} = -0.1 \cdot 2\pi \quad p_{v2} = -0.1 \cdot 2\pi \quad p_{v3} = -0.1 \cdot 2\pi$$  

$$W_{dr} = \frac{k_r \cdot (s-z_{r1}) \cdot (s-z_{r2})}{(s-p_{r1}) \cdot (s-p_{r2})}$$  

$$k_r = 1.25 \times 10^7 \quad z_{r1} = -0.1 \cdot 2\pi \quad p_{r1} = -5 \times 10^3 \cdot 2\pi \quad z_{r2} = -0.1 \cdot 2\pi \quad p_{r2} = -5 \times 10^3 \cdot 2\pi$$  

$$W_{d} = 5 \times 10^{-5}$$

As in the case of the acceleration controller, a certain amount of trial and error is necessary to find optimal low-order weighting functions resulting in a good position controller.

### 6.4.3 The position controller

For the weighting functions shown in Figure 6.13 it is possible to reach an $H_\infty$ controller leading to a transfer matrix $T_{zw}$ which has an $H_\infty$ norm less than or equal to unity. In Figure 6.14, the resulting singular values of $T_{zw}$ are plotted. The singular values of the position controller are shown in Figure 6.15. It results in the sensitivity and the complementary sensitivities shown in Figure 6.16.

The design of the position control involves a system of order 102 with six inputs and six outputs. This increases the numerical difficulties during the design. The
order of the resulting controller is equal to the order of the system plus the sum of the orders of all weighting functions, amounting to 132.

6.4.4 Order reduction of the position controller

Similar to the acceleration controller, the position controller has some very high frequency poles produced by the $H_\infty$ algorithm due to the existence of high frequency zeros in $W_r$ (cf. (6.13)). Contrary to the acceleration controller, it is possible to do the necessary order reduction of the position controller in a single step using balanced truncation. The final order of the position controller is 42 (Figure 6.17) and it still fulfills the requirements for the position loop. The sensitivity and the complementary sensitivity resulting from this controller are shown in Figure 6.18. It is clear that this order reduction deteriorates the shape of the position controller at the frequency range higher than 10 Hz. However, this deviation from the original shape is tolerable since the gain of the position controller at these frequencies is adequately low. The highest damped frequency of the poles of the final position controller lie at 14.6 Hz.

6.5 Conclusions

The control design problem presented in this chapter is a large-scale problem in terms of the high system order, the large number of inputs and outputs, the existence of two cascaded loops, and the particularity of the desired shape of each of them. Beside the considerable amount of care required in the selection of the weighting functions, the controller designed must be experimentally evaluated on the test rig to ensure the fulfillment of all design requirements. The next chapter documents the testing results of the controller designed here. A quantification of its performance is also given.
**Figure 6.17:** Singular values of the position controller $K_p$ with full and reduced order

**Figure 6.18:** Singular values of the sensitivity $S_e$ and the complementary sensitivity $T_e$ of the system controlled with the reduced order position controller
7 Controller Realization and Results

7.1 Controller realization

7.1.1 Controller discretization

Following the design procedure described in Chapter 6 both the acceleration and the position controllers are available as linear time-invariant continuous-time systems. Although the real-time computer is capable of realizing these controllers as time-continuous systems using real-time simulation, it is more convenient to discretize the controllers and realize them as time-discrete systems. Discrete-time realization requires a minimum of real-time calculation effort and eliminates the dependency of the results on the integration algorithm chosen. The algorithm used to discretize the controllers is programmed in [3]. It uses a standard bilinear transformation [47] to convert a continuous-time system to an equivalent discrete-time one.

The real-time computer implements both controllers as a single application with the same sampling rate. The poles of the acceleration controller lie at higher frequencies than those of the position controller. The highest frequency pole of the acceleration controller lies at 48 Hz, and its highest frequency zero is at 56 Hz. The real-time computer allows a sampling period of 1.5 ms for the whole application meaning that the sampling frequency is 666 Hz. The order of the acceleration controller is 66 and that of the position controller is 42.

7.1.2 Real-time program

The controller program is automatically generated from the corresponding SIMULINK graphical window containing the schematic presentation of the controller [39]. After the selection of the real-time parameters, the C code is generated. It is then compiled and linked to obtain the object code for the real-time computer as a target machine. The object code is then downloaded on the real-time computer. The whole procedure is explained in Chapter 3.
Figure 7.1 shows the SIMULINK window containing the graphical presentation of the controller. It is composed of the acceleration and the position controllers with some additional blocks useful for start-up and testing. For each of the position signals, it is possible to add an offset signal to allow testing with step inputs. A constant value between zero and one in the “On/Off Acceleration” block is multiplied by all output command signals of this controller simultaneously to permit the gradual activation of this loop. The same thing is done for the position controller from the block “On/Off Position.” The controllers are realized as linear discrete-time systems in state space form. For the first and the fourth output channels from the position controller, it is necessary to split the force command signal into two channels, a right channel which transfers only the negative part of the signal, and a left channel which transfers only the positive part of it. In this way, the command force to the four actuators LM1R, LM1L, LM4R and LM4L is limited to thrust force only. This is necessary to ensure continuous contact between the guide rollers driven by these actuators and the two guide rails (cf. Section 5.1.2). Due to the limited amplitudes of the command signals from acceleration controller in comparison with those from the position controller, it is not necessary to do the same signal splitting for the output of the acceleration controller. To protect the actuators against overload, the combined command forces from both controllers are fed through a group of saturation blocks used to limit these signals to lie inside the range of ±300 N.

After the successful download of the program, the on-line parameter tuning is activated. The acceleration controller is activated at first to be followed by the position controller. Using the trace utility [9], it is possible to make real-time measurements of all variables. These signals are necessary for the off-line determination of the controller effectiveness.

7.2 Criteria to judge the performance

7.2.1 Human sensitivity to vibrations and the K signal

To establish a judgment criterion based on a quantitative evaluation of the controller effectiveness, it is necessary to understand and model the human sensitivity to vibrations. The models utilized here are those proposed in [62]. Their utilization for elevator cars is explained in [29].
Figure 7.1: Controller program
Experiments have shown that the human sensitivity to vibrations is frequency-dependent. For example, vibrations at 50 Hz are sensed to be six times weaker than vibrations at 8 Hz with the same amplitude. As a result, the peak-to-peak values of the measured accelerations and their spectra are not the correct measure of comfort. Those experiments led to the definition of the vibrations quality measure called the K value, which is a dimensionless quantity determined from the acceleration [62].

As stated in [62], human sensitivity to vibrations is different in the cases of sitting, standing or lying. The direction of the vibrations plays a role as well. In the case of elevator cars, only horizontal whole-body vibrations in the frequency range between 1 and 80 Hz for standing people is of interest. The most important range is that between 1 and 8 Hz.

To obtain the K signal for a certain channel, a frequency evaluation of the measured acceleration is necessary. A band-pass filter which models the human sensitivity to vibrations may be used for this purpose. The transfer function of this filter as given in [62] is composed of a high-pass and a low-pass member. The transfer function of the high-pass member $H_H(s)$ is defined as:

$$H_H(s) = \frac{s^2}{s^2 + \sqrt{2} \cdot \frac{2 \cdot \pi}{10^{1/10}} \cdot s + \left(\frac{2 \cdot \pi}{10^{1/10}}\right)^2}$$

while the transfer function of the low-pass member $H_L(s)$ for vibrations in the horizontal direction is given by:

$$H_L(s) = \frac{12.5 \cdot s + 12.5^2}{s^2 + \frac{12.5^2}{8} \cdot s + 12.5^2}$$

The K signal in a horizontal direction is then given by:

$$K(s) = H_L(s) \cdot H_H(s) \cdot \frac{1}{0.035} \cdot a(s)$$

with $1/0.035$ as a scaling factor and the acceleration $a(t)$ in m/s$^2$. Figure 7.2 shows the amplitude response of the transfer function $H_L \cdot H_H/0.035$.

It is then possible to calculate the effective value of the K signal as follows:
Another important quantity for the judgment of the ride quality is the vibration strength signal $K_t$, which is a measure of the vibrations sensed by humans. Its maximum value $K_{t_{\text{max}}}$ during elevator travel is another important figure of merit for the ride quality. The $K_t$ signal is a smoothed effective value of the $K$ signal calculated using a first-order filter with a time constant $\tau$ according to:

$$K_t(t) = \sqrt{\frac{1}{\tau} \cdot \int_0^\tau e^{-\xi/\tau} \cdot K(t-\xi) d\xi}$$  \hspace{1cm} (7.5)

For whole-body vibrations in buildings and vehicles, a value of $\tau = 0.125s$ is recommended [62]. Beside the maximum value of the signal $K_t$ and the effective values of the $K$ signal for each channel, the total effective value as defined below can be used as a figure of merit for the whole active damping system:

$$KV_{\text{eff}} = \sqrt{K_{1_{\text{eff}}}^2 + K_{2_{\text{eff}}}^2 + K_{3_{\text{eff}}}^2 + K_{4_{\text{eff}}}^2 + K_{5_{\text{eff}}}^2 + K_{6_{\text{eff}}}^2}$$  \hspace{1cm} (7.6)
7.2.2 Measurement processing

A numerical solution of (7.5) does not make sense due to the great calculation effort required. The procedure to obtain the $K(t)$ and the $K_t(t)$ signals from the acceleration $a(t)$ of a certain channel may be represented by the block diagram of Figure 7.3 including the first-order low-pass filter.

After allowing a certain time for the initial excitation to die out, the effective value of the $K(t)$ signal $K_{\text{eff}}$ as well as the maximum value $K_{\text{tmax}}$ of the $K_t(t)$ signal is taken as a measure for the ride comfort for each channel separately. The scheme of Figure 7.3 is implemented on the computer to calculate the previously defined quantities.

7.2.3 Permissible $K$ values

Vibrations with $K$ values less than 0.1 are almost imperceptible. Values between 0.4 and 1.6 represent acceptable levels, while values between 1.6 and 6.3 are very well sensed. The degree of sensitivity differs from person to person. In [29], the industrial partner in this project set norms for $K$ values in elevators with the categories defined in Table 7.1.

Table 7.1: Permissible $K$ values

<table>
<thead>
<tr>
<th></th>
<th>Premium</th>
<th>Standard</th>
<th>Basic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Effective value</td>
<td>$K_{\text{eff}}$</td>
<td>0.6</td>
<td>0.9</td>
</tr>
<tr>
<td>Peak value</td>
<td>$K_{\text{tmax}}$</td>
<td>1.3</td>
<td>2.2</td>
</tr>
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</table>
7.3 Excitation for controller testing

Rail profiles for excitation

To allow testing the controller, it is required to produce an excitation very near to that caused by actual rail profiles during the elevator travel in a hoistway. Based on ride quality comparisons of travel with the active damping system with those without it, a measure of effectiveness of the active damping system was quantified.

As explained in Chapter 4, the hydraulic excitation system allows the emulation of the unevenness of the two guide rails. This suffices to generate the actual rail excitation during elevator travel.

Choice of the travel speed

The measured rail profiles are provided as deflections along the height of the hoistway. To generate the trajectories used for the excitation emulation during elevator travel, the travel speed must be known. The frequency content of the rail excitation is directly proportional to this speed. The higher the travel speed, the wider the bandwidth of the excitation. To insure that the control system remains effective for the largest possible excitation spectrum and that the resonances in the sensitivity and the complementary sensitivity resulting from the controller design are tolerable, it is important to use a high travel speed to test the system. A travel speed of 6 m/s may be considered to be quite high but also realistic for a test.

Frequency content of the rail profiles

For the four rail trajectories generated it is possible to calculate and plot the frequency spectra. The spectra of the four rails are depicted in Figure 7.4. They clearly show that the excitation is effective up to about 7 Hz. Since human beings are most sensitive to vibrations in the range between 1 and 8 Hz [62], this is quite acceptable. This range must lie in the domain of effectiveness of the controller designed.
Figure 7.4: Spectra of the guide rail excitations at a travel speed of 6 m/s
7.4 Testing results of the controller

7.4.1 The testing sequence

The response of the elevator car with the active system functioning must be compared with the corresponding response with passive system only for the same excitation. The quantitative measures for this are the six $K_r(t)$ signals, their maximum values $K_{r\text{max}}$, the effective values $K_{\text{eff}}$ of the $K$ signals, and finally the total effective value $K_{V\text{eff}}$ as a figure of merit for the effectiveness of the whole active system.

The acceleration controller is first tested alone, with the position controller deactivated. The same test is then made with the position and the acceleration controllers functioning. Based on the results of the two tests, it is possible to make sure that the position controller does not deteriorate the ride quality reached by the acceleration controller alone. After the success of the first two tests, the effectiveness of the position controller is tested. The elevator car is loaded asymmetrically to cause the mechanical saturation of some of the actuators. Two further tests are carried out with this load, the first one for the system controlled with the acceleration controller alone and the second test for the system controlled with both controllers. The system controlled with both controllers must be capable of correcting the car orientation as well as suppressing the vibrations. The excitation trajectories are identical for all four tests to allow the comparison of the results (cf. Figure 4.4).

7.4.2 Experimental results of the active system

Figure 7.5 shows the test results for the empty car with the acceleration controller alone, while Figure 7.6 shows those for the same car with both controllers. The arrangement of the six figures follows the scheme of the accelerometer signals defined in Chapter 5 (Figure 5.3). The transient period of 3 seconds was omitted from all results.

In order to test the effectiveness of the position controller the cabin is asymmetrically loaded, so that the resulting tilt of the car will force at least one of the actuators to hit its stoppers during the travel. Due to the big mass of the car and the relatively small cross section of the cabin, a large weight needs to be applied on the far most edge of the cabin. For the results given here, six average persons (total
weight of 430 kg) were standing on the right edge of the cabin roof. This was sufficient to force the four actuators LM1R, LM1L, LM4R and LM4L (Figure 5.2) to hit their stops during the travel.

In the case of the asymmetrically loaded car with the acceleration controller only, the performance compared with the empty car was significantly worse. The moments where the actuators hit their stops are visible in the signals shown in Figure 7.7. They are associated with a considerable abrupt increase of the \( K_T(t) \) signals. This performance deterioration is eliminated after the activation of the position loop as shown in Figure 7.8.

The maximum values \( K_{T_{\text{max}}} \) of the \( K_T(t) \) signals for each channel are listed in Table 7.2 for all cases examined. Table 7.3 contains the effective values \( K_{\text{eff}} \) of the \( K \) signals for all channels, while Table 7.4 contains the total effective values \( KV_{\text{eff}} \) calculated according to (7.6) for each experiment. The vibration reduction factor of the whole system is defined as the ratio of the value of \( KV_{\text{eff}} \) without the active system and its value with the active system.

### 7.4.3 Comments

The acceleration controller alone reaches a satisfying vibration reduction factor of 5.01. The fact that the additional activation of the position loop does not lead to a performance deterioration means that the bandwidth of the position controller does not interfere with that of the acceleration controller. With both loops in action, the vibration reduction factor is 5.30. This is even better than with the acceleration controller alone. The reason for this is that the amplitudes of the rail excitation in the very low frequency range are not high. The position loop adds then to the travel comfort.

The necessity of the position controller is proven by the deterioration of the ride quality to a vibration reduction factor of 2.67 when the car is loaded asymmetrically and the acceleration controller only is activated. The activation of the position controller thus saves the performance in that it allows to reach a vibration reduction factor of 4.67, which is not far from the level obtained with the empty car.
7.5 Conclusions

The successful test of the controller proves the validity of the whole active damping system. A vibration reduction factor of 5 is ensured under normal operating conditions. The system is capable of preventing actuator saturation in cases of asymmetric loading of the car.

With the results presented in this chapter the goals of the laboratory phase of the project have been reached and the decision to proceed with the project towards an industrial product has sufficient justification.
Figure 7.5: $K_x$ Signals of the empty car with acceleration controller only
Figure 7.6: $K_1$ Signals of the empty car with acceleration and position controllers
Figure 7.7: $K_r$ signals of the asymmetrically loaded car with acceleration controller only
Figure 7.8: $K_t$ Signals of the asymmetrically loaded car with acceleration and position controllers
Table 7.2: Values of $K_{t_{\text{max}}}$

<table>
<thead>
<tr>
<th>Accelerometer</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empty car</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No active system</td>
<td>2.79</td>
<td>3.99</td>
<td>5.01</td>
<td>2.64</td>
<td>2.67</td>
<td>4.76</td>
</tr>
<tr>
<td>Acceleration controller only</td>
<td>0.76</td>
<td>0.57</td>
<td>0.70</td>
<td>0.57</td>
<td>0.91</td>
<td>0.79</td>
</tr>
<tr>
<td>Acceleration and position controllers</td>
<td>0.67</td>
<td>0.48</td>
<td>0.55</td>
<td>0.57</td>
<td>0.75</td>
<td>0.63</td>
</tr>
<tr>
<td>Asymmetrically loaded car</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No active system</td>
<td>4.68</td>
<td>3.28</td>
<td>5.23</td>
<td>3.63</td>
<td>3.16</td>
<td>6.03</td>
</tr>
<tr>
<td>Acceleration controller only</td>
<td>3.25</td>
<td>1.33</td>
<td>3.67</td>
<td>1.89</td>
<td>1.22</td>
<td>1.09</td>
</tr>
<tr>
<td>Acceleration and position controllers</td>
<td>0.85</td>
<td>0.50</td>
<td>0.71</td>
<td>0.65</td>
<td>1.02</td>
<td>0.67</td>
</tr>
</tbody>
</table>

Table 7.3: Values of $K_{\text{eff}}$ for all channels

<table>
<thead>
<tr>
<th>Accelerometer</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empty car</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No active system</td>
<td>1.34</td>
<td>1.67</td>
<td>2.00</td>
<td>1.37</td>
<td>1.25</td>
<td>1.76</td>
</tr>
<tr>
<td>Acceleration controller only</td>
<td>0.34</td>
<td>0.27</td>
<td>0.26</td>
<td>0.26</td>
<td>0.41</td>
<td>0.29</td>
</tr>
<tr>
<td>Acceleration and position controllers</td>
<td>0.36</td>
<td>0.25</td>
<td>0.26</td>
<td>0.27</td>
<td>0.35</td>
<td>0.27</td>
</tr>
<tr>
<td>Asymmetrically loaded car</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No active system</td>
<td>1.54</td>
<td>1.51</td>
<td>1.95</td>
<td>1.40</td>
<td>1.47</td>
<td>2.14</td>
</tr>
<tr>
<td>Acceleration controller only</td>
<td>0.87</td>
<td>0.31</td>
<td>0.60</td>
<td>0.88</td>
<td>0.48</td>
<td>0.33</td>
</tr>
<tr>
<td>Acceleration and position controllers</td>
<td>0.46</td>
<td>0.27</td>
<td>0.31</td>
<td>0.29</td>
<td>0.48</td>
<td>0.29</td>
</tr>
</tbody>
</table>
Table 7.4: Value of $K_{V_{eff}}$ and the vibration reduction factor

<table>
<thead>
<tr>
<th>Accelerometer</th>
<th>$K_{V_{eff}}$</th>
<th>Vibrations Reduction Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empty car</td>
<td></td>
<td></td>
</tr>
<tr>
<td>No active system</td>
<td>3.90</td>
<td></td>
</tr>
<tr>
<td>Acceleration controller only</td>
<td>0.77</td>
<td>5.03</td>
</tr>
<tr>
<td>Acceleration and position controllers</td>
<td>0.73</td>
<td>5.30</td>
</tr>
<tr>
<td>Asymmetrically loaded car</td>
<td></td>
<td></td>
</tr>
<tr>
<td>No active system</td>
<td>4.15</td>
<td></td>
</tr>
<tr>
<td>Acceleration controller only</td>
<td>1.54</td>
<td>2.69</td>
</tr>
<tr>
<td>Acceleration and position controllers</td>
<td>0.88</td>
<td>4.67</td>
</tr>
</tbody>
</table>
Closing remarks

Upon ending this thesis the following closing remarks are stated:

– An active damping system for the lateral vibrations of elevator cars was developed, and a laboratory version was successfully tested.

– The hardware design of the active system respects the economical aspects of its production later as a modernization kit for elevator cars already in service.

– The system uses two control loops, one for vibration suppression using acceleration feedback and the other for the correction of the car orientation using position feedback. Since the frequency ranges of activity of the two loops are not overlapping, the two loops are not in conflict with each other.

– Model based controller design was proven to be essential in order to consider the effects of the structural dynamics.

– Due to the difficulty of modeling, system identification was adopted. Since the position sensors are not mounted on the same points as the accelerometers, identification was necessary for the transfer functions with position output in addition to the transfer functions with acceleration output.

– The need for a multivariable controller design method and the necessity of flexibility in the shaping of the acceleration and position loops recommended the use of the $H_{\infty}$ robust design method.

– The performance achieved based on a standardized judgment criterion was five times better than a passive damping system. Better performance would be possible if the following modifications were done (cf. Section 6.3):

  - Stiffening the active roller guide shoes (cf. Figure 3.1), so that the effect of the elasticities between the actuators and the accelerometers on the controller performance is neutralized.

  - Stiffening the connection between the cabin and its frame in order to reduce the influence of the second group of resonances on the controller design.
The system identification and controller design procedures developed in this work are suitable to be carried out on-site for each elevator car to be fitted with the active damping system. The man-hour required is estimated to be acceptable allowing an economical start-up of the system.

This work paves the way for the prototype phase, before the small series production may be started.
Appendices

A.1 System step responses

The letters "M" and "N" indicate transfer functions judged to have a minimum or a nonminimum-phase response, respectively. The letter "W" signifies transfer functions with very weak coupling.

In all curves, the input step signal is shown just to indicate the start of the excitation. Its numerical value is manipulated and does not belong to the same vertical scale as the system output.
A.2 The identified system parameters

The following mathematical presentation is used to define the parameters of all identified transfer functions of the system:

\[ G(s) = \frac{k \cdot (s-z_1 \cdot 2\pi) \cdot (s-z_2 \cdot 2\pi) \cdot (s-z_3 \cdot 2\pi) \cdot (s-z_4 \cdot 2\pi)}{(s-p_1 \cdot 2\pi) \cdot (s-p_2 \cdot 2\pi) \cdot (s-p_3 \cdot 2\pi) \cdot (s-p_4 \cdot 2\pi) \cdot (s-p_5 \cdot 2\pi) \cdot (s-p_6 \cdot 2\pi)} \]

The values of all eleven parameters for each of the 72 identified transfer functions of the system are listed in the following two tables.
Parameters of the identified transfer functions with acceleration output:

<table>
<thead>
<tr>
<th>Input F1</th>
<th>Input F2</th>
<th>Input F3</th>
</tr>
</thead>
<tbody>
<tr>
<td>p1 = -0.9944 + 1.5855i</td>
<td>p1 = -0.6185 + 1.6410i</td>
<td>p1 = -0.5840 + 1.7949i</td>
</tr>
<tr>
<td>p2 = -0.9944 - 1.5855i</td>
<td>p2 = -0.6185 - 1.6410i</td>
<td>p2 = -0.5840 - 1.7949i</td>
</tr>
<tr>
<td>p3 = -2.3006 + 13.613i</td>
<td>p3 = -2.9061 + 14.241i</td>
<td>p3 = -2.5817 + 13.354i</td>
</tr>
<tr>
<td>p5 = -2.000 + 41.000i</td>
<td>p5 = -4.3000 + 51.000i</td>
<td>p5 = -6.113 + 42.406i</td>
</tr>
<tr>
<td>p6 = -2.000 - 41.000i</td>
<td>p6 = -4.3000 - 51.000i</td>
<td>p6 = -6.113 - 42.406i</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ga_{11} k = 3.4825 \times 10^4</th>
<th>Ga_{12} k = 1.0858 \times 10^3</th>
<th>Ga_{13} k = -1.177 \times 10^3</th>
</tr>
</thead>
<tbody>
<tr>
<td>z1 = 0</td>
<td>z1 = 0</td>
<td>z1 = 0</td>
</tr>
<tr>
<td>z2 = 0</td>
<td>z2 = 0</td>
<td>z2 = 0</td>
</tr>
<tr>
<td>z3 = -0.6674 + 10.886i</td>
<td>z3 = 1.9062</td>
<td>z3 = -0.22946</td>
</tr>
<tr>
<td>z4 = -0.6674 - 10.886i</td>
<td>z4 = -47.194</td>
<td>z4 = -24.226</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ga_{21} k = 8.8452 \times 10^3</th>
<th>Ga_{22} k = 7.1325 \times 10^4</th>
<th>Ga_{23} k = -1.0254 \times 10^4</th>
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</thead>
<tbody>
<tr>
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<tr>
<td>z2 = 0</td>
<td>z2 = 0</td>
<td>z2 = 0</td>
</tr>
<tr>
<td>z3 = -2.7133 + 4.3661i</td>
<td>z3 = -2.4358 + 7.8502i</td>
<td>z3 = 3.0012</td>
</tr>
<tr>
<td>z4 = -2.7133 - 4.3661i</td>
<td>z4 = -2.4358 - 7.8502i</td>
<td>z4 = -12.621</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ga_{31} k = 7.2608 \times 10^3</th>
<th>Ga_{32} k = -2.0036 \times 10^4</th>
<th>Ga_{33} k = 4.1156 \times 10^4</th>
</tr>
</thead>
<tbody>
<tr>
<td>z1 = 0</td>
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<td>z1 = 0</td>
</tr>
<tr>
<td>z2 = 0</td>
<td>z2 = 0</td>
<td>z2 = 0</td>
</tr>
<tr>
<td>z3 = -0.5570 + 10.022i</td>
<td>z3 = -3.9038</td>
<td>z3 = -1.4263 + 7.7748i</td>
</tr>
<tr>
<td>z4 = -0.5570 - 10.022i</td>
<td>z4 = 4.2710</td>
<td>z4 = -1.4263 - 7.7748i</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ga_{41} k = -1.5639 \times 10^4</th>
<th>Ga_{42} k = -4.6680 \times 10^3</th>
<th>Ga_{43} k = 4.6005 \times 10^3</th>
</tr>
</thead>
<tbody>
<tr>
<td>z1 = 0</td>
<td>z1 = 0</td>
<td>z1 = 0</td>
</tr>
<tr>
<td>z2 = 0</td>
<td>z2 = 0</td>
<td>z2 = 0</td>
</tr>
<tr>
<td>z3 = 2.345 + 10.324i</td>
<td>z3 = 3.9796 + 3.2704i</td>
<td>z3 = 0.40296 + 6.6433i</td>
</tr>
<tr>
<td>z4 = 2.345 - 10.324i</td>
<td>z4 = 3.9796 - 3.2704i</td>
<td>z4 = 0.40296 - 6.6433i</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ga_{51} k = -2.4572 \times 10^3</th>
<th>Ga_{52} k = 7.7449 \times 10^3</th>
<th>Ga_{53} k = 5.3820 \times 10^3</th>
</tr>
</thead>
<tbody>
<tr>
<td>z1 = 0</td>
<td>z1 = 0</td>
<td>z1 = 0</td>
</tr>
<tr>
<td>z2 = 0</td>
<td>z2 = 0</td>
<td>z2 = 0</td>
</tr>
<tr>
<td>z3 = -3.8323 + 5.0899i</td>
<td>z3 = 18.458 + 4.1678i</td>
<td>z3 = 9.6655 + 7.2219i</td>
</tr>
<tr>
<td>z4 = -3.8323 - 5.0899i</td>
<td>z4 = 18.458 - 4.1678i</td>
<td>z4 = 9.6655 - 7.2219i</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ga_{61} k = -4.7300 \times 10^3</th>
<th>Ga_{62} k = 4.4288 \times 10^3</th>
<th>Ga_{63} k = -2.5669 \times 10^3</th>
</tr>
</thead>
<tbody>
<tr>
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<td>z1 = 0</td>
</tr>
<tr>
<td>z2 = 0</td>
<td>z2 = 0</td>
<td>z2 = 0</td>
</tr>
<tr>
<td>z3 = -2.0035 + 5.7501i</td>
<td>z3 = 4.0058</td>
<td>z3 = 10.304</td>
</tr>
<tr>
<td>z4 = -2.0035 - 5.7501i</td>
<td>z4 = -50.868</td>
<td>z4 = -61.309</td>
</tr>
<tr>
<td>Input F4</td>
<td>Input F5</td>
<td>Input F6</td>
</tr>
<tr>
<td>---------</td>
<td>---------</td>
<td>---------</td>
</tr>
<tr>
<td>p1 = -0.4468 + 1.5417i</td>
<td>p1 = -0.5798 + 1.8932i</td>
<td>p1 = -0.8943 + 2.1821i</td>
</tr>
<tr>
<td>p2 = -0.4468 - 1.5417i</td>
<td>p2 = -0.5798 - 1.8932i</td>
<td>p2 = -0.8943 - 2.1821i</td>
</tr>
<tr>
<td>p3 = -1.1349 + 8.7344i</td>
<td>p3 = -1.3060 + 8.7631i</td>
<td>p3 = -2.8829 + 11.412i</td>
</tr>
<tr>
<td>p4 = -1.1349 - 8.7344i</td>
<td>p4 = -1.3060 - 8.7631i</td>
<td>p4 = -2.8829 - 11.412i</td>
</tr>
<tr>
<td>p5 = -6.000 + 52.000i</td>
<td>p5 = -5.0687 + 37.828i</td>
<td>p5 = -6.000 + 40.000i</td>
</tr>
<tr>
<td>p6 = -6.000 - 52.000i</td>
<td>p6 = -5.0687 - 37.828i</td>
<td>p6 = -6.000 - 40.000i</td>
</tr>
</tbody>
</table>

Ga_{14} k = -1.2955 \times 10^4  

z1 = 0  
z2 = 0  
z3 = 1.165 + 8.1232i  
z4 = 1.165 - 8.1232i

Ga_{15} k = -6.9749 \times 10^2  

z1 = 0  
z2 = 0  
z3 = 1.7960 + 3.6898i  
z4 = 1.7960 - 3.6898i

Ga_{16} k = -5.9733 \times 10^2  

z1 = 0  
z2 = 0  
z3 = 21.335  
z4 = 0.21106

Ga_{24} k = 7.7148 \times 10^3  

z1 = 0  
z2 = 0  
z3 = 1.2187 + 2.7866i  
z4 = 1.2187 - 2.7866i

Ga_{25} k = 1.1266 \times 10^4  

z1 = 0  
z2 = 0  
z3 = -0.6308 + 6.5107i  
z4 = -0.6308 - 6.5107i

Ga_{26} k = -2.4380 \times 10^3  

z1 = 0  
z2 = 0  
z3 = -4.9363 + 18.212i  
z4 = -4.9363 - 18.212i

Ga_{34} k = 3.9015 \times 10^3  

z1 = 0  
z2 = 0  
z3 = 1.1790 + 3.4583i  
z4 = 1.1790 - 3.4583i

Ga_{35} k = -3.1399 \times 10^3  

z1 = 0  
z2 = 0  
z3 = -0.7589 + 10.489i  
z4 = -0.7589 - 10.489i

Ga_{36} k = 1.5209 \times 10^4  

z1 = 0  
z2 = 0  
z3 = -0.8743 + 9.2981i  
z4 = -0.8743 - 9.2981i

Ga_{44} k = 7.4626 \times 10^4  

z1 = 0  
z2 = 0  
z3 = -0.8712 + 5.4535i  
z4 = -0.8712 - 5.4535i

Ga_{45} k = 3.6145 \times 10^3  

z1 = 0  
z2 = 0  
z3 = 1.7108 + 3.4388i  
z4 = 1.7108 - 3.4388i

Ga_{46} k = -7.3617 \times 10^3  

z1 = 0  
z2 = 0  
z3 = 0.56706  
z4 = 4.5385

Ga_{54} k = -7.0667 \times 10^3  

z1 = 0  
z2 = 0  
z3 = 2.2919  
z4 = 10.728

Ga_{55} k = 5.4867 \times 10^4  

z1 = 0  
z2 = 0  
z3 = -1.4424 + 3.4946i  
z4 = -1.4424 - 3.4946i

Ga_{56} k = -1.7373 \times 10^4  

z1 = 0  
z2 = 0  
z3 = 5.4079  
z4 = 5.6332

Ga_{64} k = -5.7218 \times 10^3  

z1 = 0  
z2 = 0  
z3 = 2.0202 + 4.5448i  
z4 = 2.0202 - 4.5448i

Ga_{65} k = -8.300 \times 10^3  

z1 = 0  
z2 = 0  
z3 = -0.1738  
z4 = 3.8811

Ga_{66} k = 8.5841 \times 10^4  

z1 = 0  
z2 = 0  
z3 = -1.3161 + 5.3597i  
z4 = -1.3161 - 5.3597i
### Parameters of the identified transfer functions with position output:

<table>
<thead>
<tr>
<th>Input F1</th>
<th>Input F2</th>
<th>Input F3</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p_1 = -0.9944 + 1.5855i$</td>
<td>$p_1 = -0.6185 + 1.6410i$</td>
<td>$p_1 = -0.5840 + 1.7949i$</td>
</tr>
<tr>
<td>$p_2 = -0.9944 - 1.5855i$</td>
<td>$p_2 = -0.6185 - 1.6410i$</td>
<td>$p_2 = -0.5840 - 1.7949i$</td>
</tr>
<tr>
<td>$p_3 = -2.3006 + 13.613i$</td>
<td>$p_3 = -2.9061 + 14.241i$</td>
<td>$p_3 = -2.5817 + 13.354i$</td>
</tr>
<tr>
<td>$p_4 = -2.3006 - 13.613i$</td>
<td>$p_4 = -2.9061 - 14.241i$</td>
<td>$p_4 = -2.5817 - 13.354i$</td>
</tr>
<tr>
<td>$p_5 = -2.000 + 41.000i$</td>
<td>$p_5 = -4.3000 + 51.000i$</td>
<td>$p_5 = -6.113 + 42.406i$</td>
</tr>
<tr>
<td>$p_6 = -2.000 - 41.000i$</td>
<td>$p_6 = -4.3000 - 51.000i$</td>
<td>$p_6 = -6.113 - 42.406i$</td>
</tr>
</tbody>
</table>

| $G_{p_{11}} k = 1.3000 \times 10^4$ | $G_{p_{12}} k = 4.9296 \times 10^2$ | $G_{p_{13}} k = -5.6938 \times 10^1$ |
| $z_1 = -0.2323 + 8.3451i$ | $z_1 = 0.9212 + 13.348i$ | $z_1 = -2.8238 + 5.2694i$ |
| $z_2 = -0.2323 - 8.3451i$ | $z_2 = 0.9212 - 13.348i$ | $z_2 = -2.8238 - 5.2694i$ |
| $z_3 = -3.9614 + 22.110i$ | $z_3 = -1.2691$ | $z_3 = 11.349$ |
| $z_4 = -3.9614 - 22.110i$ | $z_4 = 28.983$ | $z_4 = 71.749$ |

| $G_{p_{21}} k = 2.0634 \times 10^3$ | $G_{p_{22}} k = 7.0720 \times 10^3$ | $G_{p_{23}} k = -6.4538 \times 10^2$ |
| $z_1 = 0.79448$ | $z_1 = -2.4815 + 6.5116i$ | $z_1 = -2.3473$ |
| $z_2 = -1.0442$ | $z_2 = -2.4815 - 6.5116i$ | $z_2 = 7.8507$ |
| $z_3 = -0.0724 + 15.049i$ | $z_3 = -7.8684 + 22.098i$ | $z_3 = 29.548 + 16.024i$ |
| $z_4 = -0.0724 - 15.049i$ | $z_4 = -7.8684 - 22.098i$ | $z_4 = 29.548 - 16.024i$ |

| $G_{p_{31}} k = 1.7189 \times 10^2$ | $G_{p_{32}} k = -8.3991 \times 10^2$ | $G_{p_{33}} k = 9.1408 \times 10^3$ |
| $z_1 = 3.6314$ | $z_1 = -2.1572$ | $z_1 = -1.145 + 6.1049i$ |
| $z_2 = 11.137$ | $z_2 = 11.359$ | $z_2 = -1.145 - 6.1049i$ |
| $z_3 = 6.5463 + 21.474i$ | $z_3 = 19.135$ | $z_3 = -6.1745 + 18.594i$ |
| $z_4 = 6.5463 - 21.474i$ | $z_4 = 43.230$ | $z_4 = -6.1745 - 18.594i$ |

| $G_{p_{41}} k = 7.8957 \times 10^3$ | $G_{p_{42}} k = -2.6024 \times 10^2$ | $G_{p_{43}} k = -5.9663 \times 10^1$ |
| $z_1 = 0.9944 + 1.5855i$ | $z_1 = -2.1335$ | $z_1 = 5.4305 + 2.1928i$ |
| $z_2 = 0.9944 - 1.5855i$ | $z_2 = 2.7499$ | $z_2 = 5.4305 - 2.1928i$ |
| $z_3 = 2.000 + 41.000i$ | $z_3 = 3.8323 + 24.084i$ | $z_3 = 4.7257$ |
| $z_4 = 2.000 - 41.000i$ | $z_4 = 3.8323 - 24.084i$ | $z_4 = 64.472$ |

| $G_{p_{51}} k = -1.5715 \times 10^3$ | $G_{p_{52}} k = 3.4006 \times 10^2$ | $G_{p_{53}} k = 4.5505 \times 10^2$ |
| $z_1 = 2.6902 + 1.4542i$ | $z_1 = 18.786$ | $z_1 = -7.1161$ |
| $z_2 = 2.6902 - 1.4542i$ | $z_2 = 5.3904$ | $z_2 = 15.871$ |
| $z_3 = 14.002 + 1.4398i$ | $z_3 = 0.70044 + 43.202i$ | $z_3 = 11.166 + 13.100i$ |
| $z_4 = 14.002 - 1.4398i$ | $z_4 = 0.70044 - 43.202i$ | $z_4 = 11.166 - 13.100i$ |

<p>| $G_{p_{61}} k = 2.9607 \times 10^1$ | $G_{p_{62}} k = 8.2252 \times 10^3$ | $G_{p_{63}} k = 1.1346 \times 10^2$ |
| $z_1 = 1.9509 + 1.3335i$ | $z_1 = 0.16634$ | $z_1 = 6.5205 + 30.839i$ |
| $z_2 = 1.9509 - 1.3335i$ | $z_2 = 4.7589$ | $z_2 = 6.5205 - 30.839i$ |
| $z_3 = 27.014$ | $z_3 = 2.2188 + 22.496i$ | $z_3 = 2.8981$ |
| $z_4 = -89.326$ | $z_4 = 2.2188 - 22.496i$ | $z_4 = 105.41$ |</p>
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<th>Input F6</th>
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<td>$p_1 = -0.5798 + 1.8932i$</td>
<td>$p_1 = -0.8943 + 2.1821i$</td>
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<td>$p_2 = -0.5798 - 1.8932i$</td>
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<td>$p_3 = -1.3060 + 8.7631i$</td>
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<tr>
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<td>$p_4 = -1.3060 - 8.7631i$</td>
<td>$p_4 = -2.8829 - 11.412i$</td>
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<td>$p_6 = -5.0687 - 37.828i$</td>
<td>$p_6 = -6.000 - 40.000i$</td>
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<td>$G_{p_{16}} k = -4.5893 \times 10^2$</td>
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<td>$z_1 = 5.2286$</td>
<td>$z_1 = 5.4094 + 9.5605i$</td>
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<tr>
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<td>$z_2 = 5.1866$</td>
<td>$z_2 = 5.4094 - 9.5605i$</td>
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<td>$z_3 = 19.310 + 18.504i$</td>
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<tr>
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<td>$z_2 = -3.8003 - 18.346i$</td>
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<td>$z_2 = 4.6134$</td>
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<tr>
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<td>$z_3 = -0.1587 + 14.931i$</td>
<td>$z_3 = 16.091 + 44.252i$</td>
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<tr>
<td>$z_4 = 3.6405 - 8.6859i$</td>
<td>$z_4 = -0.1587 - 14.931i$</td>
<td>$z_4 = 16.091 - 44.252i$</td>
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<td>$G_{p_{64}} k = 3.5706 \times 10^2$</td>
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<td>$z_1 = -1.4880 + 4.9363i$</td>
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</table>
A.3 The weighting functions used for the identification

The following mathematical definition applies for all transfer functions utilized to generate the weighting vectors utilized by the complex curve fitting algorithms for the identification of the individual transfer functions of the system:

\[
W(s) = \frac{k \cdot \prod_{i=1}^{m} (s - zi \cdot 2\pi)}{\prod_{j=1}^{n} (s - pj \cdot 2\pi)}
\]

In the following two tables the values of all poles and zeros are listed for all weighting functions utilized. The amplification parameter \( k \) did not have an influence on the identified parameters. Therefore, it was always set to unity.
Parameters of the transfer functions used to generate the weighting vectors

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<th>Input F2</th>
<th>Input F3</th>
</tr>
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<td>( W_{a_{12}} )</td>
<td>( W_{a_{13}} )</td>
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<td>no zeros ( p_1 = -0.1 + 2.0i )</td>
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</tr>
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</tr>
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<td>( p_3 = -0.1 + 11.0i )</td>
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<tr>
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<td>( p_4 = -0.1 - 11.0i )</td>
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<tr>
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<td>( W_{a_{23}} )</td>
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<td>no zeros ( p_1 = -0.1 )</td>
<td>no zeros ( p_1 = -0.1 )</td>
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for the transfer functions with acceleration output

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<td>no zeros $p1 = -0.1 + 2.0i$</td>
<td>no zeros $p1 = -0.1 + 3.0i$</td>
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<td>$p2 = -0.1 - 2.0i$</td>
<td>$p2 = -0.1 - 3.0i$</td>
</tr>
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<td>$p3 = -0.1 + 10.0i$</td>
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<td>$p4 = -0.1$</td>
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<td>$W_{a65}$</td>
<td>$W_{a66}$</td>
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<td>no zeros $p1 = -0.1 + 2.0i$</td>
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<td>$p3 = -0.1$</td>
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<td>$p4 = -0.1$</td>
<td>$p4 = -0.1$</td>
</tr>
<tr>
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<td>$p4 = -0.1$</td>
<td>$p4 = -0.1$</td>
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<td>$p5 = -0.001 + 40.0i$</td>
<td>$p5 = -0.001 + 40.0i$</td>
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<td>$p6 = -0.001 - 40.0i$</td>
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### Parameters of the transfer functions used to generate the weighting vectors

<table>
<thead>
<tr>
<th>Input F1</th>
<th>Input F2</th>
<th>Input F3</th>
</tr>
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<tbody>
<tr>
<td>( W_{p11} )</td>
<td>( W_{p12} )</td>
<td>( W_{p13} )</td>
</tr>
<tr>
<td>no zeros ( p1 = -1.0 + 10.0i )</td>
<td>no zeros ( p1 = -0.1 + 10.0i )</td>
<td>no zeros ( p1 = -0.1 )</td>
</tr>
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<td>( p2 = -0.1 - 10.0i )</td>
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<tr>
<td>( W_{p21} )</td>
<td>( W_{p22} )</td>
<td>( W_{p23} )</td>
</tr>
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<td>no weighting was useded</td>
</tr>
<tr>
<td>( p2 = -0.01 - 3.0i )</td>
<td>( p2 = -0.1 + 6.0i )</td>
<td></td>
</tr>
<tr>
<td>( p3 = -2.0 + 18.0i )</td>
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<td>( p4 = -2.0 - 18.0i )</td>
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<td>( W_{p33} )</td>
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<td>no weighting was useded</td>
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<td>( W_{p42} )</td>
<td>( W_{p43} )</td>
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<td>no zeros ( p1 = -0.1 )</td>
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<tr>
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<td>( p2 = -0.1 - 2.0i )</td>
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</tr>
<tr>
<td>( z3 = 0 )</td>
<td>( p3 = -0.1 + 40.0i )</td>
<td></td>
</tr>
<tr>
<td>( p4 = -0.1 - 40.0i )</td>
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<td></td>
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<tr>
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<td>( W_{p52} )</td>
<td>( W_{p53} )</td>
</tr>
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<td>( z1 = -0.1 )</td>
<td>no zeros ( p1 = -0.1 )</td>
</tr>
<tr>
<td>( p2 = -0.1 )</td>
<td>( p1 = -20 )</td>
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</tr>
<tr>
<td>( p3 = -0.1 )</td>
<td></td>
<td></td>
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<tr>
<td>( W_{p61} )</td>
<td>( W_{p62} )</td>
<td>( W_{p63} )</td>
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<tr>
<td>no zeros ( p1 = -0.1 )</td>
<td>no zeros ( p1 = -0.1 + 3.5i )</td>
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</tr>
<tr>
<td></td>
<td>( p2 = -0.1 - 3.5i )</td>
<td></td>
</tr>
<tr>
<td></td>
<td>( p3 = -4.0 + 40i )</td>
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<tr>
<td></td>
<td>( p4 = -4.0 - 40i )</td>
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</tbody>
</table>
for the transfer functions with position output

<table>
<thead>
<tr>
<th>Input F4</th>
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<tbody>
<tr>
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<td>$WP_{16}$</td>
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<td>no zeros $p1 = -0.1$</td>
</tr>
<tr>
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<td>$p2 = -0.1 - 2.0i$</td>
<td>$p2 = -0.1$</td>
</tr>
<tr>
<td>$p3 = -1.0 + 40.0i$</td>
<td>$p4 = -1.0 - 40.0i$</td>
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<td>$WP_{24}$</td>
<td>$WP_{25}$</td>
<td>$WP_{26}$</td>
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</tr>
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<td>was useded</td>
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<td>$p1 = -30$</td>
</tr>
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<td>$WP_{34}$</td>
<td>$WP_{35}$</td>
<td>$WP_{36}$</td>
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<td>no weighting</td>
<td>no weighting</td>
</tr>
<tr>
<td>was useded</td>
<td>was useded</td>
<td>was useded</td>
</tr>
<tr>
<td>$WP_{44}$</td>
<td>$WP_{45}$</td>
<td>$WP_{46}$</td>
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<td>$z1 = 0$</td>
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<td>$p1 = -20$</td>
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<tr>
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<td>$p3 = -0.1$</td>
<td>was useded</td>
</tr>
<tr>
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<td>$WP_{64}$</td>
</tr>
<tr>
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<td>$p5 = -0.1$</td>
<td>$WP_{65}$</td>
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<td>$WP_{65}$</td>
<td>$WP_{66}$</td>
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<td>no weighting</td>
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<tr>
<td>was useded</td>
<td>was useded</td>
<td>was useded</td>
</tr>
</tbody>
</table>
A.4 Frequency responses of the acceleration

Each page contains the transfer functions from one of the inputs to the six accelerometer outputs. The placement of the six figures on a page follows the arrangement of the sensors shown in Figure 5.3.
A.5 Frequency responses of the position

Each page contains the transfer functions from one of the inputs to the six position outputs. The placement of the six figures on a page follows the arrangement of the sensors shown in Figure 5.3.
A.6 Validation of acceleration outputs
A.7 Validation of position outputs
References


Ayman Hamdy was born on March 13th, 1962 in Giza, Egypt, where he received his entire education. In July 1984, he obtained his B.Sc. with honors in mechanical engineering from Cairo University, Faculty of Engineering. Following his graduation he worked as a research assistant in the Department of Aeronautics at the same University, from which he received his M.Sc. degree in Aeronautics Engineering in July 1988. In July 1989, he received a Swiss federal scholarship to pursue postgraduate studies at the Automatic Control Laboratory of the Swiss Federal Institute of Technology in Zurich. Between October 1989 and July 1991, he completed a postgraduate study with certificate in mechatronics. In February 1992 he joined the Measurement and Control Laboratory of the Swiss Federal Institute of Technology in Zurich as a research and teaching assistant where he pursued his doctorate study in automatic control.