Active Lateral Suspension for High Speed Trains

A Step towards the Mechatronic Bogie

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24.10.01

Summary

In a joint project of the DaimlerChrysler AG and Bombardier Transportation an active lateral suspension system for high-speed trains has been developed by using a closed mechatronic design cycle. The concept has been successfully tested in a prototype.

Encouraged by these results and following the mechatronic design principles, a first and realistic Mechatronic Bogie concept has been developed.

Abstract

In the field of rail technology, there are continuously rising requirements concerning riding comfort, running safety, and speed from the side of the railway operators. These requirements are opposed by the fact that the condition of the tracks is getting worse and maintenance is becoming expensive. In view of this conflict, conventional suspension concepts are quickly at their limits. Like in the automobile industry, active measures are therefore becoming more and more interesting. To meet the very conflicting requirements for introducing active components in an economical way, a full mechatronic design cycle process must be followed.

In the project ”Active Bogies” which was financed by the Bombardier Transportation Business Unit Bogies and carried out in the DaimlerChrysler Research and Technology department, a new concept for an active lateral suspension for high speed trains has been developed and implemented in a prototype.

The hardware consists of a hydropneumatic lateral actuator in each bogie. With the aid of these actuators, it is possible to influence the three degrees of freedom lateral movement, yaw, and roll movement. Passive air springs are used as vertical suspension, because a fully active suspension system would consume too much energy, as the static load of the car body has to be held.

In order to control this complex system, a new straightforward mechatronic design cycle has been developed. This cycle starts with the modelling and model reduction process. In a second step,

the model parameters are identified with the help of real measurements. Based upon this validated model, the controller is developed automatically. Its parameters are also determined in a single step. As a by-product, a model-based online diagnosis helps to detect hardware failures. By using this approach, it was possible to optimise the controller structure and adapt the parameters prior to the first implementation in the prototype in a very secure and static closed loop process.

First tests with this prototype on the roller rig in Munich show the high potential of this technology. The concept succeeds to fulfil all requirements by the Deutsche Bahn AG, concerning energy consumption, the restriction of lateral movement, running into the curve, and the lateral comfort values.

The Mechatronic Bogie will become Reality!

Increasing demands in railway business appear to force contradicting development paths: How to offer higher performance like increased running speed and improved passenger comfort while reducing first cost, maintenance effort - both on the vehicle and the infrastructure side - and lead time? The solution will be the Mechatronic Bogie!

At a first glance this statement will appear to give an unattainable vision which will fail on reality – this paper explains why the Mechatronic Bogie will become reality, and not at some date far in the future but on the rather short term.

MECHATRONICS is the integrated design and development philosophy of a system being formed by mechanical, electrical, and control components taking into account from the beginning their interdependencies and interfaces. Traditionally the first step in product development is to design the mechanical system. Then the electrical – or other active - components will be designed, and finally the controller is generated which shall actively influence the behavior of the entire system. In contrast the mechatronic approach regards all requirements simultaneously, not sequentially, thus leading to a mechanical structure which is well prepared for the electrical systems functioning in and on it and which does not interfere with the activity of the controller.

In a rough sense the main point is the system design approach looking for the purpose of each component in the sense: Does it transmit power or is its purpose control, synchronization and coordination? Identifying the first area with mechanical and actuation elements and the second one with electronics, communication and control increases design freedom for the flexible generation of controlled motion [ASHLEY97].

This mechatronic design approach will now be applied on a bogie, which is primarily characterized by the mechanical side. The required functions of carrying and suspending the car body, following the guideway formed by the rails, and transmitting traction and braking forces between vehicle and track determines the lay-out; it is the “incarnation” of running stability and safety. But increasing demands concerning passenger comfort or reduced impact on the tracks require – within the mechanical design process - additional or improved *passive* components, leading to complex and sometimes complicated configurations of bogies and suspension and therefore to additional weight. The move to *active* systems was of rather limited benefit yet since the first ones have been introduced after the mechanical design was largely finished: Switchable dampers yielded some improvement by allowing a first step towards adaptation, but the advantage of e.g. hold-off devices has to be produced in a passive environment which as a whole is not well prepared.

The goal of the new approach called mechatronics [KORTÜM98] is to re-design traditional mechanical components through the introduction of controlled active elements thus reducing complexity of the mechanics nearly to the minimum by allowing the highest effectivity and efficiency of electrical subsystems activated by controllers. Each subsystem can function at its optimum since the overall system concept from the very beginning takes its requirements into account. This concurrent engineering process will lead to new solutions of integrated functional elements replacing several traditional add-ons. E.g. the tasks of the separate devices semi-active lateral damper and hold-off device will be performed by one optimized actuator. On the other hand the attempt to reduce the heavy structures of rolling stock by applying light-weight designs in car body and bogies introduces the severe hazard of structural vibrations – this problem can hardly be solved by passive means under all operational conditions, but is a possible task for state-of-the-art actuators and advanced control algorithms emerging from a synergetic systems consideration. This is the design freedom offered by mechatronics.

Historical Overview: Active Systems in the Railway Industry

The main drivers for the development of the mechatronic design methodology have been robotics and spacecraft technology. Exact performance of complex multidimensional movements required from the beginning closed real-time feed-back loops, i.e. the common development of kinetic structures and cybernetic control, especially when being performed by light-weight structures whose elasticity has to be taken into account and actively compensated.

Starting mainly with servo technology as simple stand-alone applications in the 1970s, the introduction of information technology in the 80s led to the embedding of microcontrollers into mechanical systems. Numerically controlled machines, robots and automotive applications like antilock-braking systems [NIEMAN99] or electronic engine control became widespread and well-known. A further stage is marked by adding communication technology in the 1990s allowing mechatronic systems to act in large and fast networks.

Those requirements mentioned above for robotics and space vehicles do not play a comparably important role in railway operation. Here the safe movement of a rather massive structure with high speed over a guideway which only provides tiny contact areas is the major issue. The techniques applied have to be robust to withstand the severe environmental conditions acting on the bogie and its components for a very long period, and they have to be rather easily maintainable. Hence the introduction of active systems was a rather reluctant process.

The first controllable system concerning the bogies of railway vehicles has been the slip-slide protection to allow the full exploitation of the traction and braking effort of high-power locos by adhesion control [BAUER86] [LANG93] [SCHREI99]. In comparison: The above mentioned anti-blocking system well-known in the automotive sector only controls the braking performance. An example apparent to passengers is the active tilting functionality – first realized in the Italian „Pendolino“ train in 1967 [FIAT98] - allowing to increase running speed on strongly curved tracks. Other progress which is less obvious are the introduction of hold-off devices [ALLEN94], the development of switchable – or more general: variable – damping characteristics [ROTH95], e.g. for coarsely adapting the running behavior to straight or curved tracks, leading to a pneumatic attempt for an active lateral suspension [MARUYA98], and the field of electrically controlled braking which has just started to emerge.

Neither of these systems have been developed using a real mechatronic approach! They all form add-ons on a predefined mechanical environment which on the one-hand side forms the fully passive safety-related fall-back solution but on the other hand limits the benefit of the new devices. Main obstacles are e.g. weight and cost of the mechanical structures and active components due to the requirement on fail-safety, and the traditional development process looking at mechanics, electrics and control separately.
They have to be overcome within the railway industry by installing a new design and engineering culture, i.e. by

- strengthening complete system competence and
- building-up of mechatronic know-how, and concerning suppliers by the
- development of appropriate actuator technologies.

How will the Mechatronic Bogie look like?

Since the customer requirements form a multidimensional world, the solution giving the most customer benefit will differ in several aspects from traditional products:

- High performance in terms of speed and uncompensated lateral acceleration
- Higher passenger comfort
- Low impact on the environment and especially the track
- Low investment cost
- Easy to maintain
- Long maintenance intervals
- “Intelligent” in the sense that it monitors its own condition to allow maintenance when necessary

These continuously increasing and more and more conflicting customer demands can only be fulfilled if the limitations of the traditional design process will be overcome. The need is to not longer focus on individual (passive) elements to fulfil some specific functions but to look at the system as a whole which shall serve the customer’s desires as a whole. This leads to the need for multifunctional elements and to look at their interactions. A multifunctional element must be placed at a position where it optimally influences a number of physical degrees of freedom. Realistic mechatronic concepts can be found in the first place in the carbody suspension and the steering system of the axles.

Car body suspension

A car body suspension must consist of at least two lateral, two vertical and two roll springs. In such a configuration the lateral springs influence the lateral, roll, sway, and yaw behavior of the carbody. An active lateral suspension could therefore take-over the role of the lateral suspension and the lateral damper plus the inter-vehicle suspension and dampers, which are usually necessary to improve the yaw behavior of a vehicle. Thus a better overall ride quality will be realized while reducing mechanical complexity.

Axle steering

Due to the wheel (and rail) profiles, the axles have a natural tendency to follow the track and to align themselves in a radial position in curves. However with increasing speed this behavior leads to highly dynamic and unstable movements. In order to stabilize an axle at high speeds it must be stiff guided and strongly coupled to the car body, in most cases by yaw dampers. The latter leads to a strong mechanical excitation of the car body structure and to a very high structural noise transmission.
In contrast to this an active steering device for axles could provide full steering ability in curves and simultaneously stabilize the axle without the requirement for yaw dampers. An immediate benefit in the reduction of wheel/rail wear and forces, ride comfort improvement and noise reduction would result from the introduction of such a system with even lower mechanical complexity.

There are definitely more areas in railway vehicle design where mechatronic principles could successfully be applied, but at present the two proposed solutions appear to fulfill for the first time the very conflicting requirements in an economical way. In addition, through these two functionalities an almost full bogie condition monitoring system is automatically installed.

Controller Design

The heart of each mechatronic system is the design of its controller. Traditional controller design methods can only yield sub-optimal results in terms of performance and economics. Only intelligent model-based controllers can handle the required multi-functionality together with condition monitoring in the right way. Applying different optimization strategies the resulting system can be individually configured for the specific needs of different customers.

Synthesis

Looking at the existence and availability of actuator technology on the market, semi-active dampers and electro-mechanical drives are already state-of-the-art and can easily be introduced in any bogie type.

Therefore the first mechatronic bogie configuration will consist of [Fig. 1 and 2]:

- two axles, individually controlled by one actuator per axle
- conventional primary suspension incl. vertical dampers
- bogie frame
- two air springs, one anti-roll bar, one semi-active lateral damper, two vertical dampers, optionally hold-off device and semi active vertical dampers.
- air spring leveling valves, electronically controlled
- overall condition monitoring

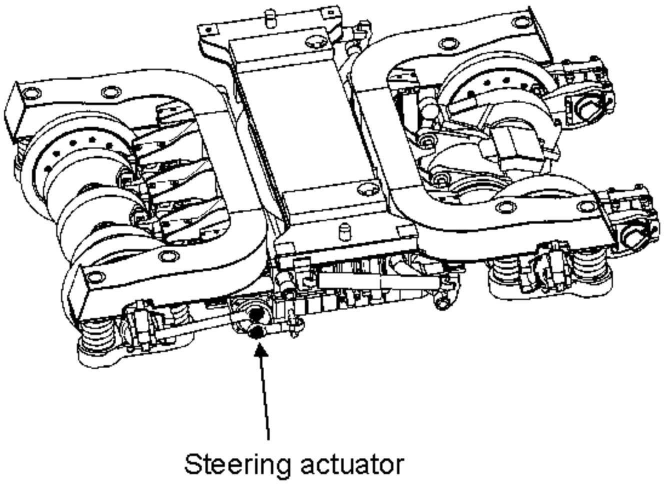

Fig.1: General Layout of the first Mechatronic Bogie

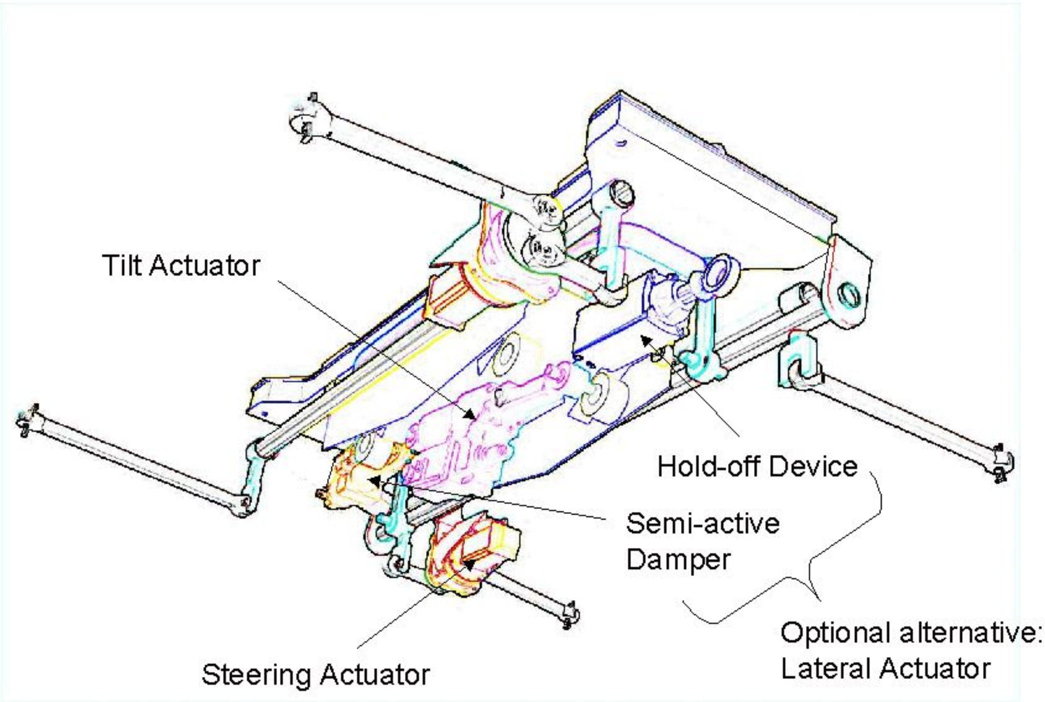

Fig. 2: Design Study: Optional Actuators of the first Mechatronic Bogie

All components, electronic ones as well as mechanical ones, will be designed for “plug and play” and will have a self configuration and self diagnostics capability.

Using train communication systems to share information between different bogies and even vehicles of a train set, track condition monitoring can be realized in combination with a system which provides geographical position information like e.g. GPS.

A Step into Realisation

In order to examine the potential of a mechatronic bogie design and to show the capabilities of an entirely new control design method an active lateral suspension system has been realised. The system has been successfully tested on the roller rig in Munich. A detailed description of this development is given in the following paragraphs.

The Active Lateral Suspension System – An Introduction

In the last few years, demands in the rail engineering sector to improve the comfort and safety of passengers have advanced into areas which do not seem feasible on the basis of conventional bogie engineering. As a result - and like in the automotive sector - active measures are very much gaining ground, particularly in order to be able to cope technically with increasing speeds and greater comfort demands (Fig. 3).

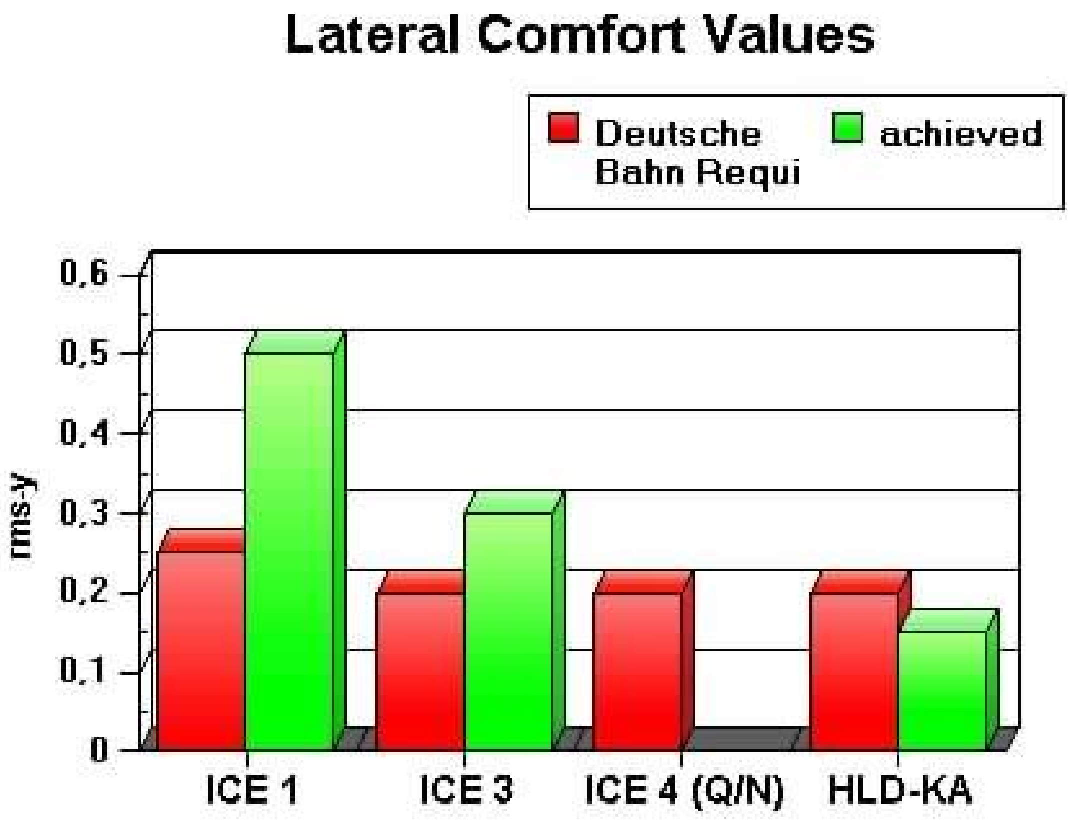

Fig. 3: Increasing comfort demand profile of Deutsche Bahn (ICE 1, ICE 3: conventional concepts, ICE 4: with tilt technology, HLD-KA: Project prototype)

Active suspension systems for rail vehicles have been discussed in this context as part of scientific investigations for more than twenty years. However, because of the much higher system complexity compared with the car, up to now, the only implementations in practice have been made in the sector of tilt technology and the application of quasi-static adjustment systems.

As part of the "Active Bogies" project, an active lateral suspension system for high speed trains has been developed on the basis of the active suspension system for cars [ABC99] developed by Daimler Chrysler, which differs from previous concepts in all aspects and creates the technological basis for achieving the comfort and safety standards demanded and also opens up further improvement potential for future generations of trains. A key element of the development process was a new straight-forward concept for designing control system structures and adapting their parameters which allows a systematized controller designed without the need for iterative controller adjustment [STREIT96].

Conflicts and Restrictions

Bogie design is governed by a large number of conflicting restrictions and requirements. On the one hand, the standard of comfort has to be maintained or raised at increasing top speeds and in spite of a deteriorating rail tracks. At the same time the observing of the clearance gauge is of prime importance. Another requirement is to minimize the maximum lateral displacement of the car body that mainly occurs when running into a curve, because this opens up the possibility of building the car body wider and thereby offers the passenger an increased level of comfort. In the "Active Bogies" project, the maximum admissible lateral stroke of the actuators was limited to ± 4 cm. In addition , the lateral stroke shall be fully available to absorb track disturbances, if the train is curving steadily.

It is obvious that the demand to reduce the maximum lateral stroke and increase riding comfort are diametrically opposed.

If active measures are chosen for increasing comfort, this in turn creates another restriction, i.e. to minimise the energy used. In the "Active Bogies" project, an upper energy limit of 5 kW was agreed with Deutsche Bahn AG. Summing up, the points of conflict are as follows:

- Increasing the comfort potential, in spite of higher speeds and deteriorating rail tracks,
- Minimising the actuator stroke required with an upper limit of ± 4 cm
- Minimising the power consumption with an upper limit of 5 kW

Determining the Hardware Concept

The definition of a suitable hardware structure according to the boundary conditions and requirements described above must be based on the specific evaluation guidelines of the railway. ISO 2631 [ISO2631] represents the two most important criteria for assessing vertical and lateral comfort (Fig. 4).

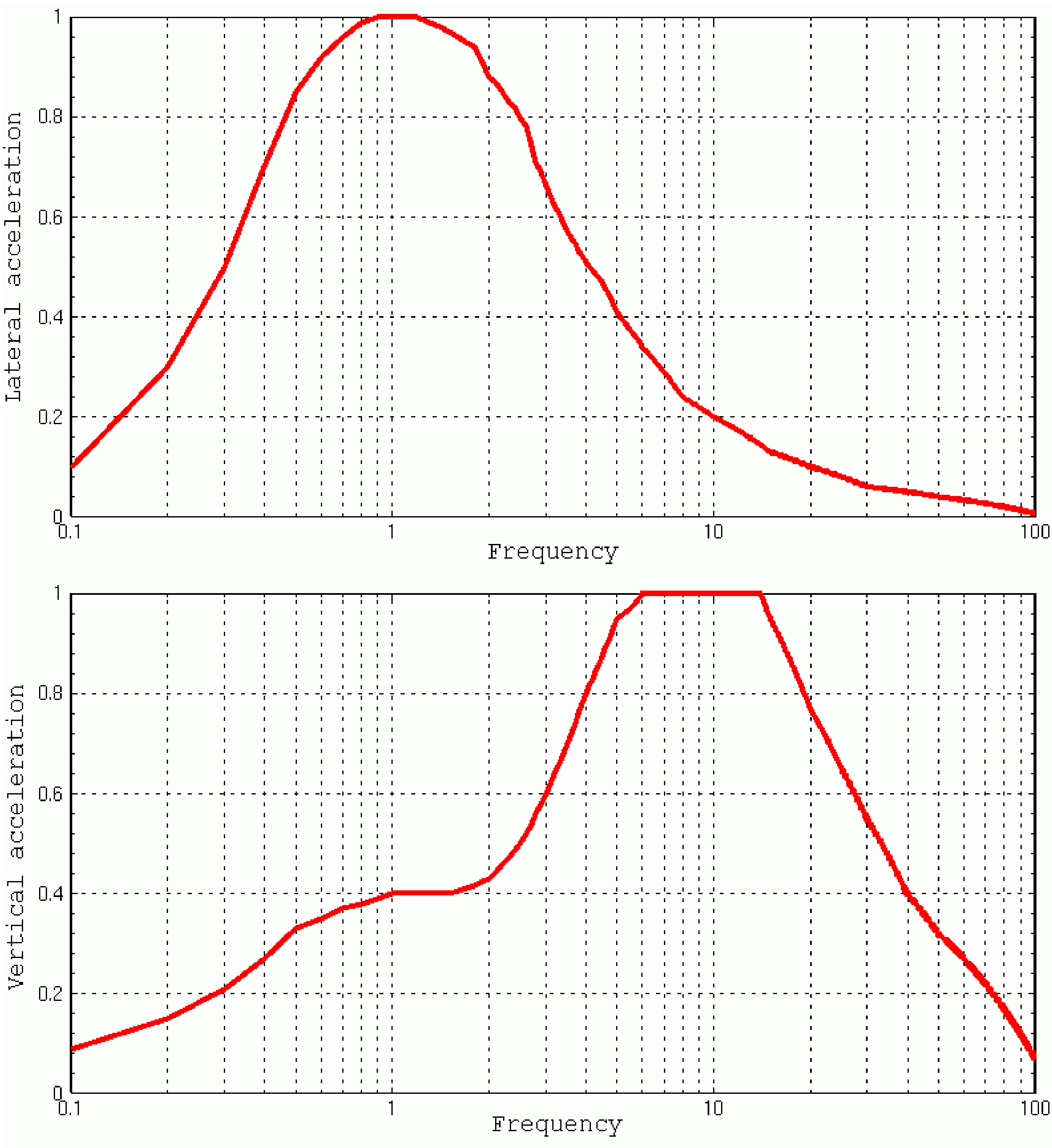

Fig. 4: Comfort evaluation function according to ISO 2631 of Deutsche Bahn AG

In this case, a frequency-related distribution and weighting of the accelerations occurring at the car body provides an index that will allow an objective evaluation of the sensitivity of passengers. It must be pointed out that the weighting curve maxima for lateral and vertical comfort occur at different frequencies. Whereas the vertical comfort weighting curve assesses vibrations around 6–10 Hz as negative maximum, the lateral comfort assessment has a maximum at 1 Hz. It is obvious that vertical vibrations have to be controlled in a range between 6-10 Hz in order to achieve a significant improvement in comfort compared with the passive system, whereas controlling the lateral acceleration of the car body can already have a significant effect in the range around 1 Hz. If we assume the general correlation, that the energy to be consumed increases proportionally to the speed of movement of an actuator and therefore controlling the vertical direction must be much more energy-consuming than controlling the lateral direction, this leads directly to the conclusion of improving riding comfort by means of active control of the lateral direction while at the same time leaving the vertical movement passive. This creates a favourable ratio between the energy to be consumed and the comfort improvement achieved.

Various actuator concepts are suitable for this purpose. The project opted for hydro-pneumatic actuators, which, on the one hand, can achieve the passive spring/damper properties and at the same time generate active forces by supplying hydraulic fluid (Fig. 5). Since the energy density of hydraulic actuators is very high, further advantages are created in terms of space requirements.

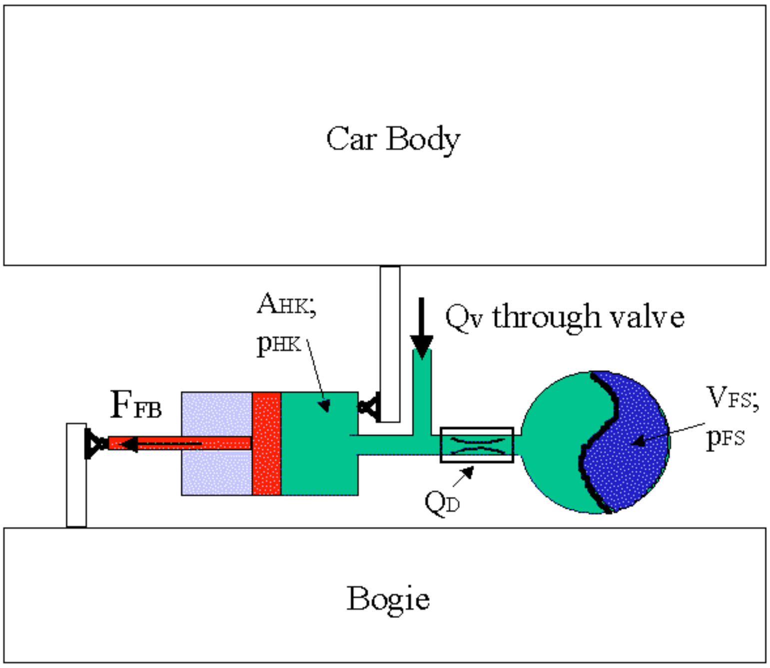

Fig. 5: Actuating principle of a hydro-pneumatic actuator

Description of the technical implementation

The car body is controlled by four lateral hydro-pneumatic actuators (Fig. 6 and 7). The actuators are arranged in pairs per bogie, but the system lay-out leads to the effect of having two actuators operate per car body.

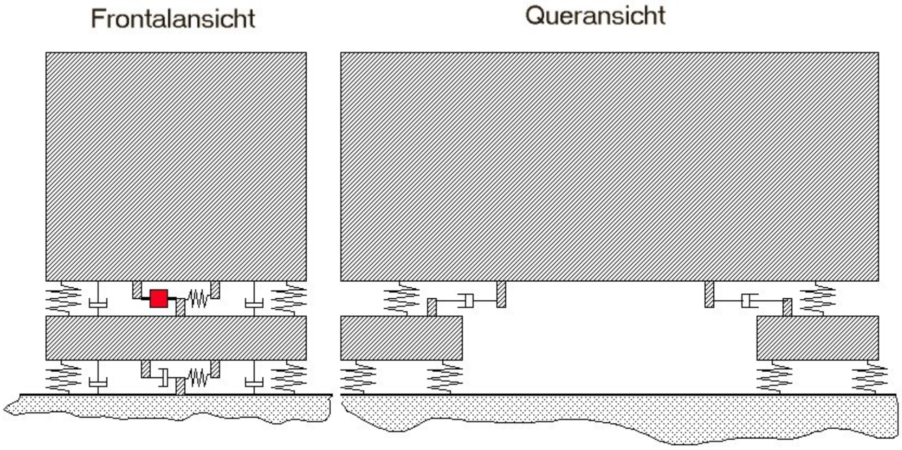

Fig. 6: Principle diagram of the actuator arrangement (lateral actuators combined)

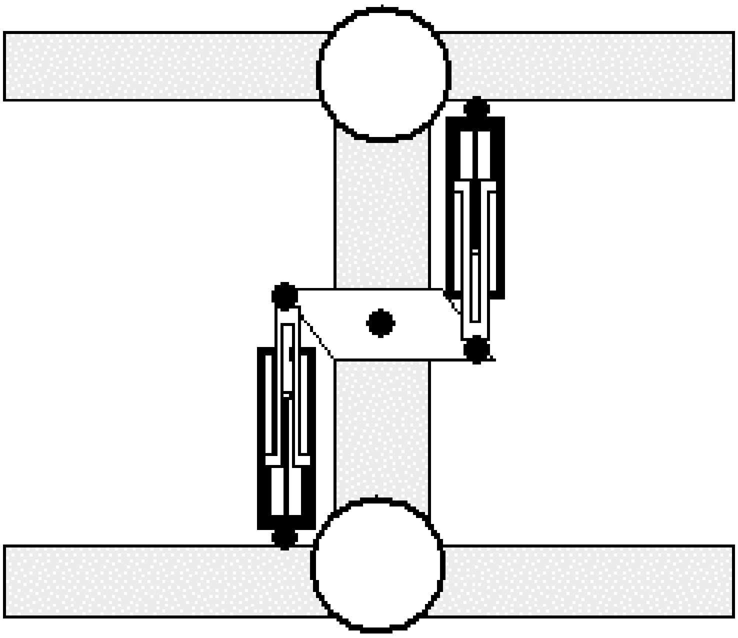

Fig. 7: Top view of actuator arrangement. Actuators operating anti-clockwise generate torque freedom. Only secondary springs are shown.

The car body is supported vertically by four air springs. With this actuator arrangement, it is possible to influence the lateral and yawing movement. However, since the structural boundary conditions prevent the actuators from being applied directly or very close to the centre of gravity, a synchronous action of the actuators via the lever arm “z_wk_akt” illustrated in Fig. 8 creates a rolling movement at the same time. The rolling movement and lateral movement of the car body are therefore linked. Another problem is that now only two actuators are available for controlling three degrees of freedom and therefore, from the control point of view, the system has to be designated as not being fully controllable.

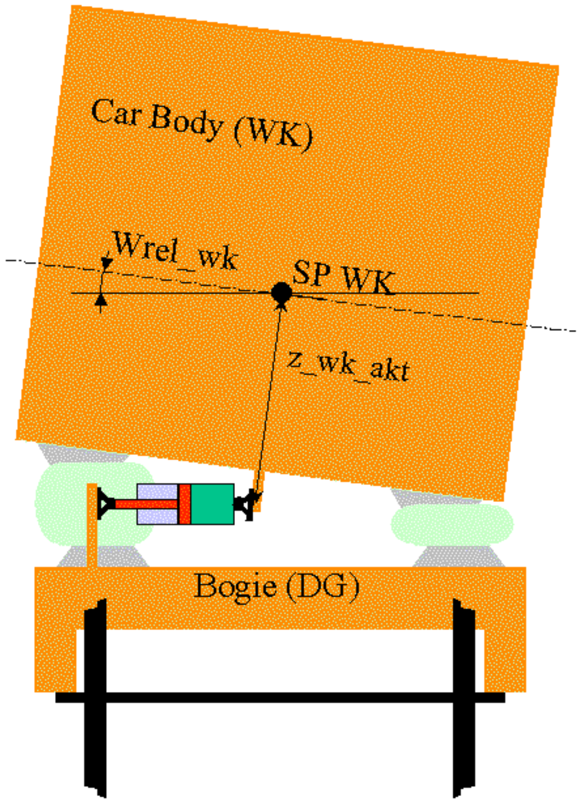

Fig. 8: Excitation of rolling movement by lateral actuators

Therefore, the aim of the controller design is to define a controller structure which takes into account the linking of lateral and rolling movements and, together with yawing movement control, allows an optimum point of application from the point of view of comfort and energy.

The Model

One of the principal tasks of controller development consists of modelling the complete system. At the same time the model has to concentrate on the basic modes of the car body and the bogie movement. Therefore, it is assumed that the car body is a rigid body. The hydro-pneumatic actuators are detailed modeled with non-linear elements such as overlaps in the valves or pressure dependence of the actuator stiffness. On the other hand, the wheel-rail contact is not represented. Therefore, the movements of the wheel sets form the model input.

The main tasks of the complete model are:

- better understanding of the system,
- providing access for deriving optimum controller structure through specific model reduction,
- checking the functionality of the controller regarding the performance objective and also observing the restrictions imposed.

In order to be able to arrive at valid conclusions on the basis of this model, the model parameters are determined with the aid of parameter identification. This process, together with a statistical evaluation of the parameter variations of identification results with different manoeuvres and also a parameter sensitivity analysis, guarantees the prognosis capability of the model.

If the equation of motion for the car body and the bogies is formulated in the co-ordinates (x, y, z, w, n, ψ), we have a non-linear system with three linked matrix differential equations of second order and therefore a system with $18 \cdot 2 = 36$ state variables. The model was implemented in Matrixx [MATX96]. Further non-linear elements were added.

Deriving the Reduced Model

The model already discussed includes frequencies and modes which are explicitly not taken into account in the control system. Therefore, to define the actual controller, the complete model is reduced until only the model properties relevant for the control system remain.

In a first step, the non-linear effects, like friction and the non-linear spring stiffnesses in the actuators, are eliminated. The number of model frequencies is reduced in a second stage. To do this, the bogie mass and also its damping is assumed to be zero. In this way, the reduced equations of motion for the bogies can be substituted in the car body equation of motion. Figure 9 shows the comparison of the frequency spectra of the vertical and lateral acceleration between the linear complete model and the frequency-reduced model with a representative stochastic excitation of the bogies.

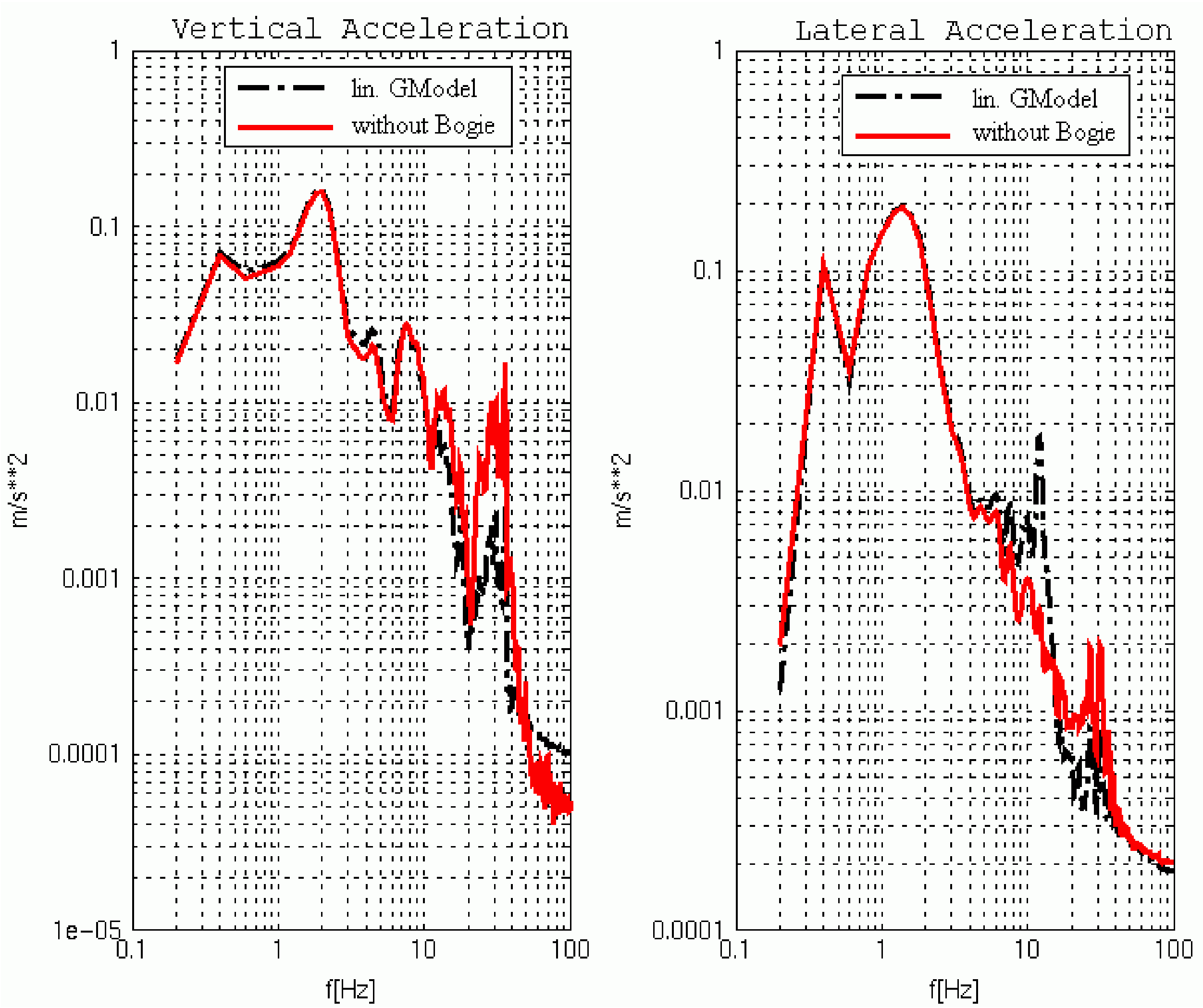

Fig. 9: Model comparison of a linear complete model with a linear model without the bogie dynamics

It can be seen that a very good model correspondence exists up to approximately 5 Hz. The reduced model equations have the following form:

$$\left[T_{RM\_N2}s^2 + T_{RM\_N1}s + I\right]\begin{pmatrix} x \\ y \\ z \\ w \\ n \\ \psi \end{pmatrix} = \left(K_{RM\_Z1}s + K_{RM\_Z0}\right)\begin{pmatrix} u_{RSL}^{DG1} \\ u_{RSL}^{DG2} \end{pmatrix} + \left(K_{RM\_P} + \frac{1}{s}K_{RM\_I}\right)Q$$

However, a modal reduction of the model to those eigenmodes that are to be influenced by the control system is necessary in a next step.

In a first test, all the non-relevant modes are eliminated directly from the equations of motion except the yawing movement and the lateral movement. However, a simulation comparison (Fig.

10) of a transformed system of this type shows that it is no longer possible to represent the characteristic modes.

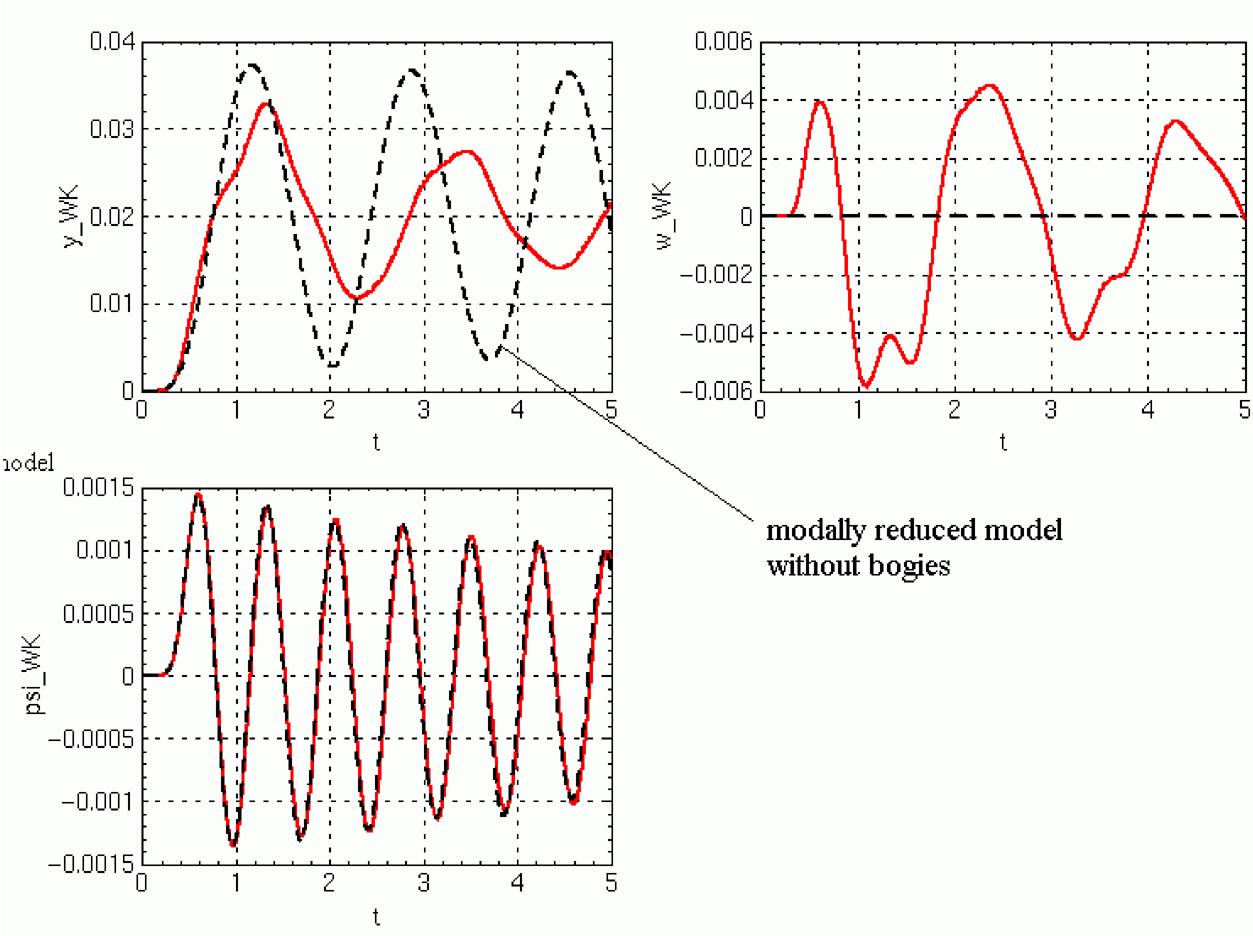

Fig. 10: Simulation comparison after a simple modal model reduction to a system with lateral and yawing movement. The rolling movement is not illustrated in the modally reduced model

This is also confirmed by the eigenvalue analysis (Fig. 11). The reduced model (represented by crosses) has eigenvalues that do not correspond to the linear complete model.

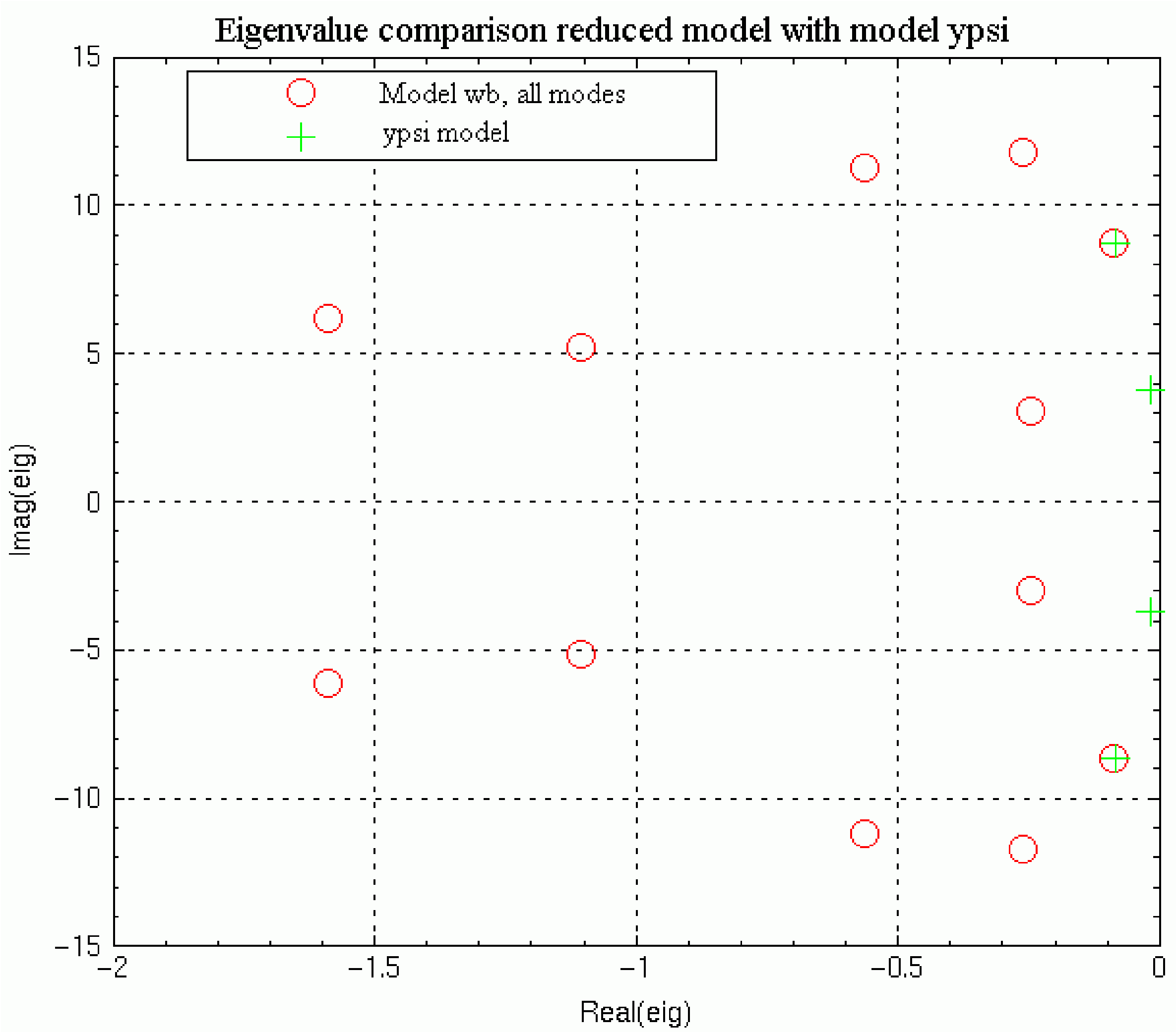

Fig. 11: Eigenvalue comparison after simple modal model reduction to a system with lateral and yawing movement

Therefore, a **modal integration** is suggested where the rolling movement is integrated into the lateral movement. The basic idea behind this suggestion is to regard the rolling movement as part of the lateral movement and integrate it into this mode. The modal integration can be formally defined as follows:

$$f(w, y, \psi) = R \Rightarrow f_1(y, \psi) + f_2(w) = R_1 \text{ and } f_3(w) = f_4(y, \psi) + R_2$$

$$f_1(y, \psi) + f_2\left(f_3^{-1}\left(f_4(y, \psi) + R_2\right)\right) = R_1 \Rightarrow$$

This transformation gives a matrix equation with only two modes.

The system order of the resultant lateral movement eigenmode therefore increases by the order of the rolling movement dynamics and reflects the correlation between the linked modes exactly. A control system designed for this mode stabilises rather than controls the rolling movement, because this mode is represented by internal state variables. However, if the lateral movement as a whole is stabilised by a control system, this also applies to all the internal eigenforms.

The simulation and eigenvalue comparison (Fig. 12 and Fig. 13) confirms the success of this model reduction; the modes relevant for the control system are accurately reflected.

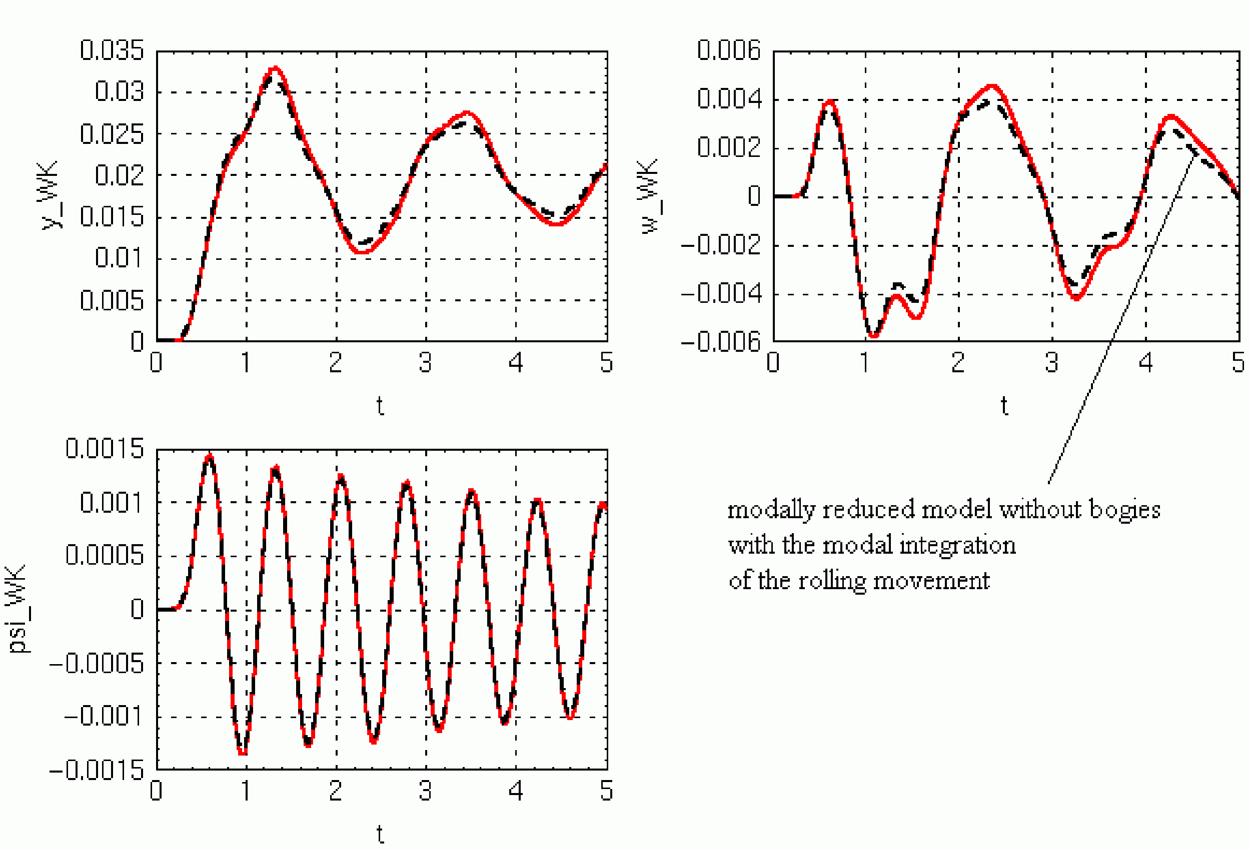

Fig. 12: Simulation comparison after model reduction with modal integration of the rolling movement into the lateral movement

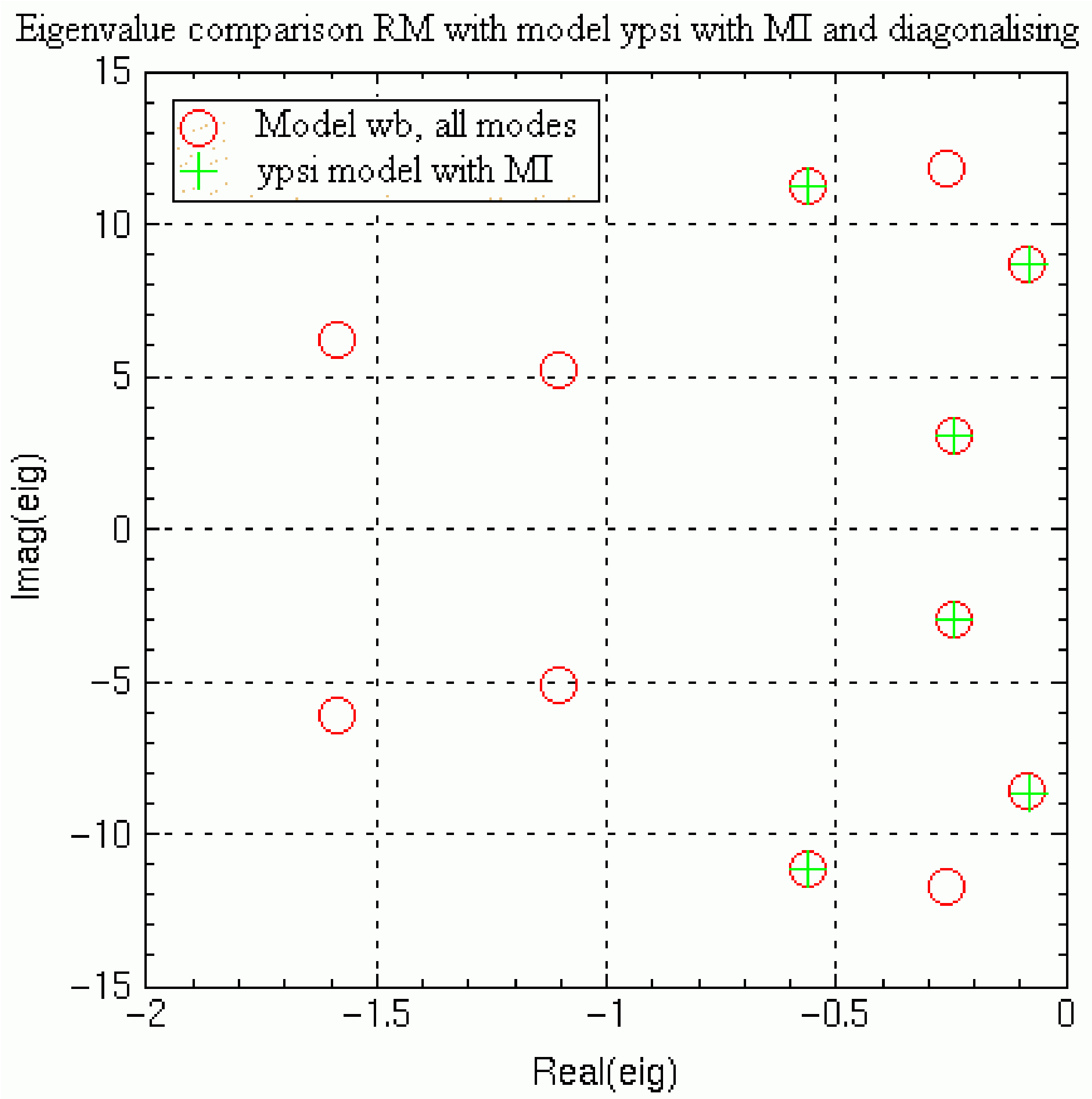

Fig. 13: Eigenvalue comparison with modal integration of the rolling movement into the lateral movement

This is obviously the model on which the necessary control structure can be based. The formal matrix structure is as follows:

$$\left[T_4 s^4 + T_3 s^3 + T_2 s^2 + T_1 s + T_0\right]\begin{pmatrix} y \\ \psi \end{pmatrix} = \left[K_3 s^3 + K_2 s^2 + K_1 s + K_0\right]\begin{pmatrix} u_{RSL}^{DG1} \\ u_{RSL}^{DG2} \end{pmatrix} + \left(K_{2D} s^2 + K_D s + K_P + K_I \frac{1}{s}\right) Q$$

In the last step, a scalar representation can be found for the two remaining modes if we just consider the correlation between the oil volume flow and the relevant state variable and eliminate the remaining fast zeros from the differential equations.

This produces the following differential equations in this specific case:

$$y_{rel}^{WK} = \frac{0.0422s^2 + 0.057s + 1}{0.00086s^5 + 0.0014s^4 + 0.1165s^3 + 0.0628s^2 + s} Q_y$$

and

$$\psi_{rel}^{WK} = \frac{1}{0.0132s^3 + 0.0022s^2 + s} Q_\psi$$

The transfer functions illustrated in Fig. 14 occur in the frequency range.

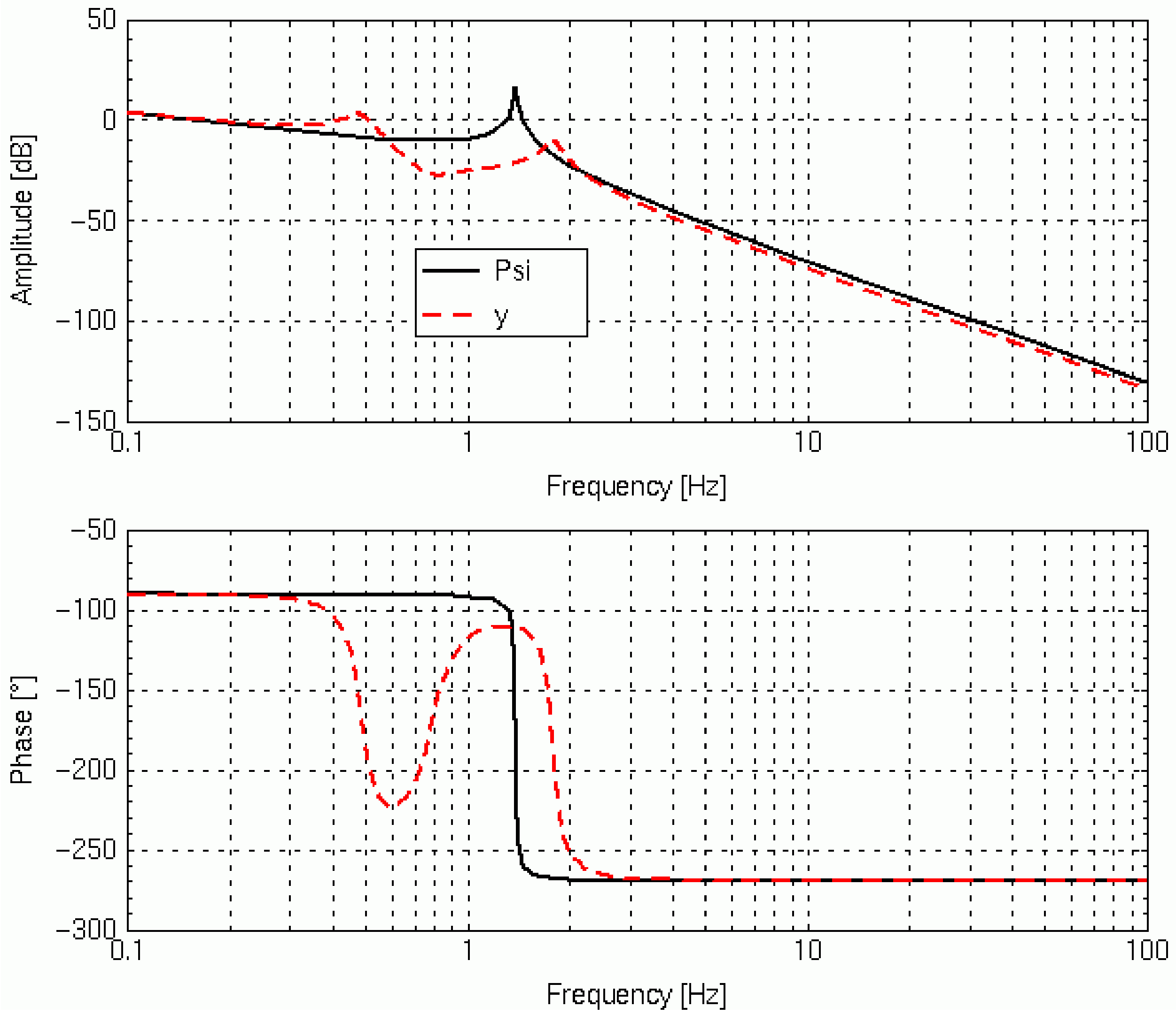

Fig. 14: Bode's diagram of the scalar transfer functions for y and psi

Control System Concept

A suitable control system concept must guarantee that

- on the one hand, disturbances resulting from the irregularity of the tracks can be eliminated in the interests of improving comfort,
- on the other, following the line of the track is guaranteed and
- further restrictions, like limiting the spring travel and observing the clearance gauge, are taken into account.

An integral control design concept is described below which determines the controller structure and parameters on the basis of the model and leads to an ideal interaction of the passive system with the controller. The principle considerations are discussed first.

The basic concept of the complete design is based on the principle of influencing the dominant modes of the car body by the control system, whereas higher frequency modes, essentially resulting from the movements of the bogies, are only determined by the passive properties of the actuators. This principle mainly results from performance and energy considerations and automatically leads the need for integrating higher order filters in the controller.

The problem already discussed, i.e. that only two actuators act on three degrees of freedom, is counter-acted by using the equation of motion for the lateral displacement with integrated rolling dynamics for the controller design. Without this special consideration of the link between the lateral movement and the rolling movement, a higher frequency control system in terms of lateral comfort, and according to ISO Guideline 2631 [ISO2631] in particular, automatically leads to an excitation of the rolling movement. Consequently, up to now, known lateral comfort control systems have been operating at low frequencies in order to avoid this excitation of the rolling movement. However, this also means that these control systems remain far away from the lateral comfort level illustrated here.

The centering and guiding of the car body in relation to the bogies is guaranteed by low frequency control of the relative lateral displacement and yawing position of the car body, whereas the improvement in comfort is achieved through the higher frequency control of the absolute lateral and yawing acceleration of the car body. Which signals are specifically used has to be deduced from a precise analysis of the manoeuvres to be made. Consequently, the control system will have four paths in principle:

- Two paths which are responsible for the position control of the relative lateral and yawing movement of the car body, and
- two paths which counteract the absolute yawing and lateral acceleration of the car body.

Naturally, position control and inertial acceleration control are diametrically opposed. Therefore, the purpose of the design is to minimise this area of conflict and thereby guarantee optimum effectiveness of the individual controller components with regard to performance and energy consumption.

Since the lateral and yawing position control system partly undertakes the centring task, the design of the secondary springs can be softer in the lateral direction. Through this easily feasible measure, particularly in the case of air springs, fewer disturbances from the lateral position differences of the tracks are transferred to the car body via the passive bogie–air spring–car body path, which, in turn, means that the active system has to absorb fewer disturbances at the car body.

Controller Design Process

An important success factor when designing the active lateral comfort control system was the development of a controller design concept that functions exclusively to the straight-forward principle, manages without any manual parameterisation, is extremely robust and can be used very flexibly in the design phase. This new controller design procedure forms the nucleus of the design process. It supplies modelbased and analytical the controller structure and the parameters that are adapted to the problem. In spite of the complex controller structures generated, the controller design process guarantees an extremely easy operation for the developer. In this case, the controller can be parameterised fully automatically according to the known rail vehicle data. The concept can be used equally well with scalar control circuits, matrix systems and also non-linear systems and therefore represents an instrument which allows the designing of very highly developed controllers with automatic parameter setting.

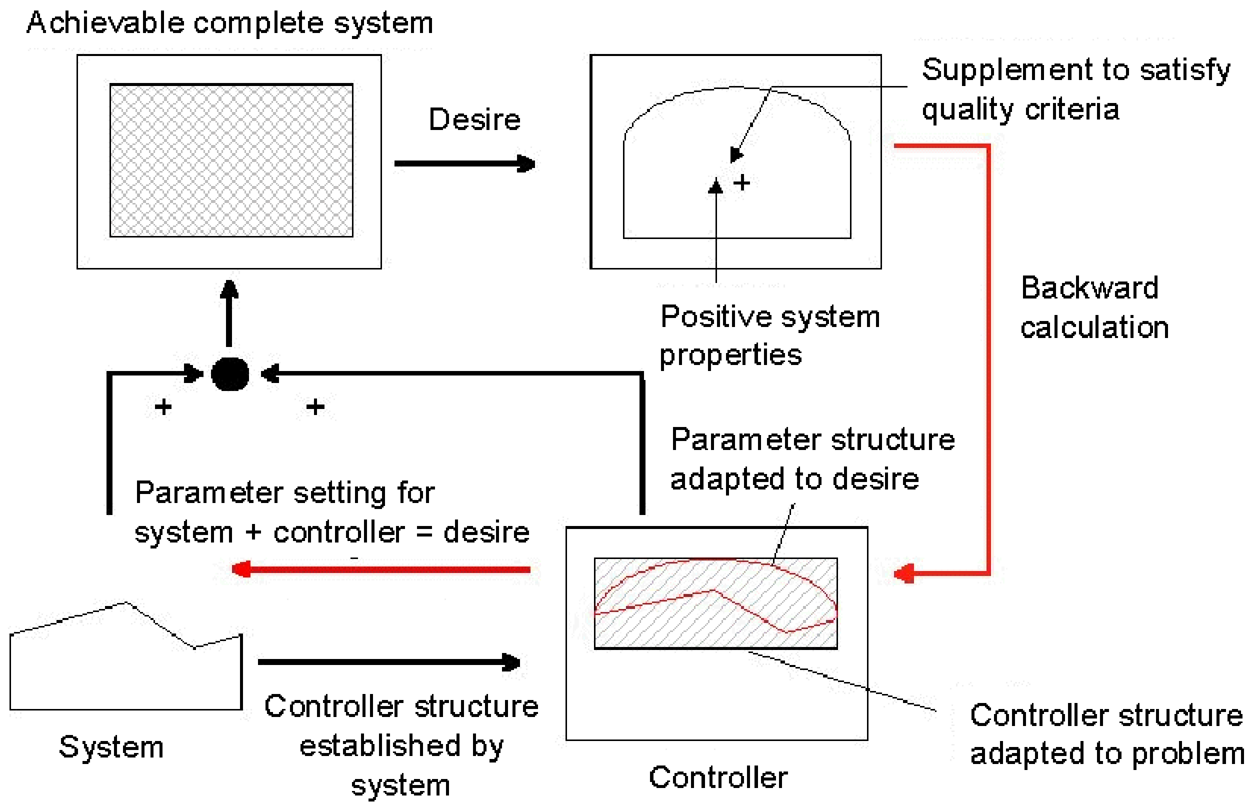

Fig. 15: Fully automated controller design

First of all, the detail model is generated as part of a system analysis which is reduced to the essential system properties for the control system in a second stage. Based on real measuring data, the model is then validated by means of identification. This is the deciding basis for the necessary data and model certainty required in the following controller design stage. The controller design process then takes place in the following stages (Fig. 15):

1. The minimum structure of the controller is determined by the definition of the structure of the system.
2. Through the interaction of system and controller, a potential implementation area is created that can be defined completely by setting the controller parameters.
3. The user can freely define a wish behaviour within the potential area.

4. Favourable passive properties are integrated into the desire so that the controller and system are linked.
5. The controller parameters are set by a back calculation to reflect exactly the desire of the user.

A structural adjustment of the controller to the hardware described can be made in an ideal way with the integral controller design suggested here. In this case, all the modes which are described by the two derived movement differential equations for the lateral and yawing movement are controlled. This means that with this principle, the rolling movement, which has been formulated as part of the lateral movement, is also further stabilised by the control system.

Control System Structure Implemented

As already described, the controller design process is based on the structure of the system to be controlled. The minimum nominator order of the controller is determined by the order of the denominator of the relevant system reduced by one. In the case of higher order systems, this automatically means that higher derivations are used in the control system. Therefore, in order to be able to implement these derivations, the denominator order of the controller must be at least higher by one than the nominator order. Higher filter orders may be chosen in order to reduce further the disturbing influence of higher frequency vibrations that result from car body structure vibrations, for example. Special requirements, which, in this specific case, have to be formulated when controlling the absolute lateral and yawing acceleration of the car body, mean that a shifting in the nominator is necessary. This means that the number of coefficients in the nominator is determined according to the above specifications, but from the shift position onwards, the nominator is split and the coefficients are displaced further in the direction of higher derivations.

In the specific case, this means that the lateral control system must have at least 4 nominator coefficients, whereas the yawing movement only needs one controller with at least 2 nominator coefficients. The filter order is specified as 4 in each case, so that the denominator polynomials will be of the eighth and sixth order. The first two coefficients of the two nominators are used for adjusting the levels of the degree of freedom concerned. Together with the denominator filter function, this guarantees a low frequency positioning and guiding of the car body in relation to the bogies. The controller is split from this nominator position onwards, because the higher derivations are used for comfort control purposes. Therefore, the absolute yawing and lateral acceleration of the car body are the input signals.

The requirement that a constant lateral acceleration during static running through a curve, but also the more or less constant first derivation of the lateral acceleration occurring when running into a curve, should have a minimum effect on the movement of the car body in relation to the bogies applies to the following for the structure of the lateral acceleration controller:

- that these signal portions must be omitted on the one hand and, on the other,
- that the acceleration must be controlled through the use of at least the next higher derivation stages of the lateral acceleration.

Since the control system also acts directly on the valve and thereby creates an integrating effect of the system, a further shift is necessary to satisfy the requirements indicated. The total shift of two positions occurs for the lateral acceleration control system, which, under consideration of the measured second derivation of the car body lateral acceleration, leads to a controller part which uses the filtered second and third derivation of the lateral acceleration signal. By shifting, the desired gain in comfort can be achieved on the one hand and on the other, the relative stroke restriction can be maintained even when running into a curve.

Similar considerations can be made for controlling the absolute yawing acceleration. Since, with static running through a curve, a constant yawing speed occurs, only the yawing acceleration may be included in the comfort control system. If we also consider the integrating effect of the system, the additional shift, together with the second derivation of the yawing acceleration that already exists, leads to a control part which uses the filtered first derivation of the yawing acceleration.

The control signals calculated in this way are distributed among the actuators taking the geometric conditions into account. Volume flows demanded directly by the controller for the rear and front bogie flow in or out of the actuators depending on the sign. If, as in the present case, two actuators are connected so that they oppose each other, the controller output signal determined in this way will be distributed with different signs to the right hand and left hand actuator. The inverse valve characteristics have to be taken into account to the discussed controller output. A pressure independence of the spring stiffnesses of the lateral actuators can be achieved through additional measures.

The controller structure developed in this way can be parametrised using the process already explained.

In order so to have the entire actuator stroke available to absorb lateral disturbances even when running through a curve, a nominal value has to be calculated for the relative car body position which, during steady curving, means that the actuators move to their middle position. The critical factor for this consideration is the static rolling angle of the car body resulting from running through the curve.

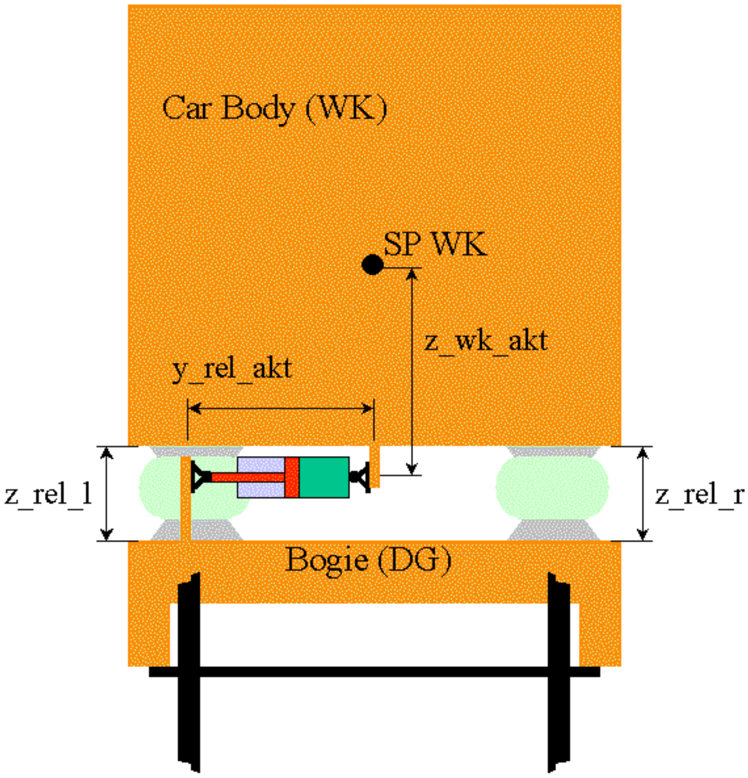

Fig. 16: The y position of the centre of gravity of the car body can be calculated from the stroke of the lateral actuator and vertical strokes

A corresponding nominal position can be calculated for the lateral displacement of the car body [Fig. 16] by a back calculation. The requirement that the lateral displacement of the actuators should be equal to zero while static cornering results in the following after a brief calculation.

$$y^{soll}_{rel\_WK} = -z_{WK\_akt} w_{rel\_WK}$$

This nominal value only affects the position circuit of the lateral movement (Fig. 17) via a slow low pass filter. This requirement is met in this way at low frequency without a negative effect on acceleration control. Requirements concerning the gauge clearance can be met by compromise mean values.

The controller described can be constructed as follows:

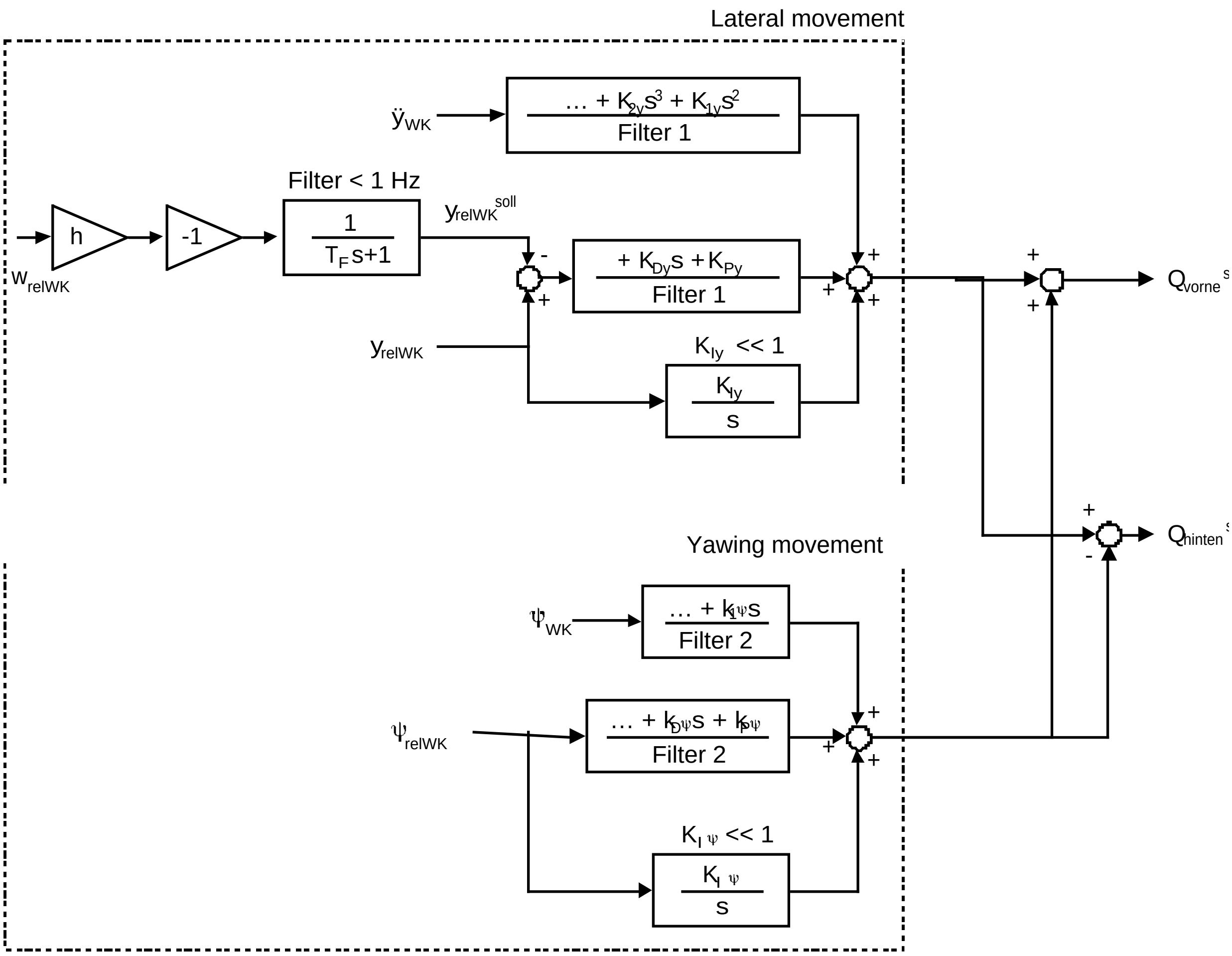

Fig. 17: Controller structure

Sensors

The controller described requires the following measuring data.

- inertia lateral and yawing acceleration of the car body at the centre of gravity, and
- the relative lateral displacement of the car body in relation to the two bogies and also their relative yawing and rolling angles.

Since, for example, a direct measurement of the lateral movement of the car body's centre of gravity is not possible, this entity must be determined from other measured values.

To do this, the relative yawing angle of the car body in relation to the bogie is calculated in the first stage from the elongations of the lateral actuators, which contain a displacement sensor:

$$\psi_{rel\_WK} = \frac{y_{rel\_v} - y_{rel\_h}}{l_v + l_h}$$

The yawing acceleration can be determined with the aid of two lateral acceleration sensors on the car body in a similar way:

$$\ddot{\psi}_{WK} = \frac{\ddot{y}_v - \ddot{y}_h}{l_v + l_h}$$

In the next stage, the relative rolling angle is determined from vertical relative stroke measurements at the air spring positions taking into account the lateral distance of the displacement sensors.

$$w_{rel\_WK} = \frac{z_{rel\_vr} - z_{rel\_vl}}{l_{q\_v}}$$

The measurement can be supported by adding vertical displacement sensors to the rear bogie.

The relative lateral displacement of the car body can then be calculated from the elongations of the lateral actuators and the two previously calculated relative rolling and yawing angles (Fig. 16).

$$y_{rel\_WK} = \frac{y_{rel\_v} + y_{rel\_h}}{2} - z_{WK\_akt} w_{rel\_WK} - \psi_{rel\_WK} \frac{l_v - l_h}{2}$$

The car body acceleration can be calculated using similar considerations. Two lateral acceleration sensors attached to the car body supply the car body lateral acceleration value at the centre of gravity if the signal portions resulting from the rolling angle and the rolling and yawing acceleration of the car body are removed.

To do this, the absolute rolling acceleration is determined first of all by vertical acceleration sensors attached to the car body.

$$\ddot{w}_{WK} = \frac{\ddot{z}_{vr} - \ddot{z}_{vl}}{l_{vr\_VBS} + l_{vl\_VBS}}$$

Again vertical acceleration sensors on the back of the car body can support the measurement.

If it is also taken into account that the gravity value supplies a portion of the measured lateral accelerations on the car body via the absolute rolling angle of the car body then the car body lateral acceleration can be determined with approximation of the absolute rolling angle through the filtered relative rolling angle.

$$\ddot{y}_{WK} = \frac{\ddot{y}_v + \ddot{y}_h}{2} - z_{WK\_akt} \ddot{w}_{WK} - \ddot{\psi}_{WK} \frac{l_v - l_h}{2} - \frac{g}{2} w_{rel\_WK}$$

The two lateral actuators working against each other also make it necessary to have one pressure sensor per actuator. This allows the sum of the two pressure signals to be controlled in order to prevent mutual building up of the force levels acting in opposite directions. In the uncontrolled case, such a twisting would lead to an uncontrolled change in the spring stiffnesses and, in an extreme case, result in damage to the system. On the other hand, the spring stiffness of the hydro-pneumatic actuators can be specifically influenced by controlling this kind of twisting.

Parameter Identification

An extension of the vectorial Maximum Likelihood algorithm [KARMAR84] is used as the identification algorithm, to which the following points have been added:

- Linear dependencies in the parameters are recognised automatically and these parameters are eliminated from the direction calculation.
- The measuring variables are further normalised, which facilitates the mathematics considerably.
- Parameters can be linked automatically.
- Parameters which reach the limits specified for them in the course of an identification process are automatically eliminated from the direction calculation if a first direction assessment indicates that these parameters need to be shifted further beyond the limits.

The identification tool produced in the course of the project is able to determine the parameters quickly and robustly even when faced with this complex problem. All in all, the identification has to determine approximately 60 parameters, for example the

- masses and inertias
- position of the centre of gravity
- spring stiffness of air springs and anti-roll bar
- all hydraulic parameters
- signal offsets and integrator initial conditions.

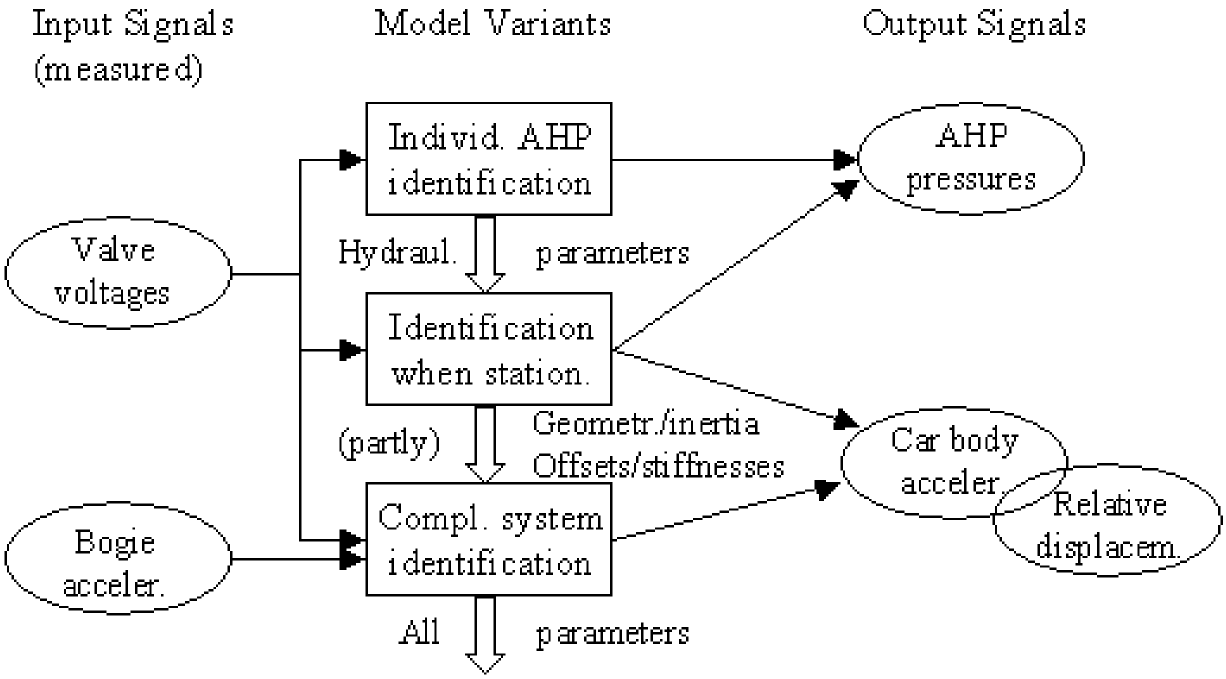

Fig. 18: Graduated identification process

As illustrated in Fig. 18, the parameter identification process takes place in three steps, which differ according to the size of the identification models, and takes into account the boundary conditions that occur with this particular problem. The preliminary stage represents the identification of the individual hydro-pneumatic actuators based on measurements obtained on an actuator test rig. Only hydraulic parameters are identified at this stage. On the one hand, this identification step increases the model certainty and on the other, it is possible to document and freeze the hydraulic parameters in the complete model in advance.

The second identification step consists of the autonomous excitation when the train is stationary, because with the prototypes in the early stage, it is not possible to imposese real excitations through track irregularities. A big disadvantage of his purely lateral excitation is that not all the degrees of freedom can be excited and therefore not all the parameters can be determined, but have to be partly fixed or linked.

The model quality required for the controller design can only be achieved in a third stage, which is the complete system identification. Real track irregularity profiles can now either be imposed during test runs or on the roller rig of Deutsche Bahn AG in Munich [ROLLP85]. All the parameters required can be determined on the basis of these measurements. Since, as explained above, the identification model cannot take into account any track excitations, these bogie accelerations have to be used as the input signal.

Fig. 19 shows the correspondence achieved between the measurement and simulation with a sinus sweep imposed on the wheel sets.

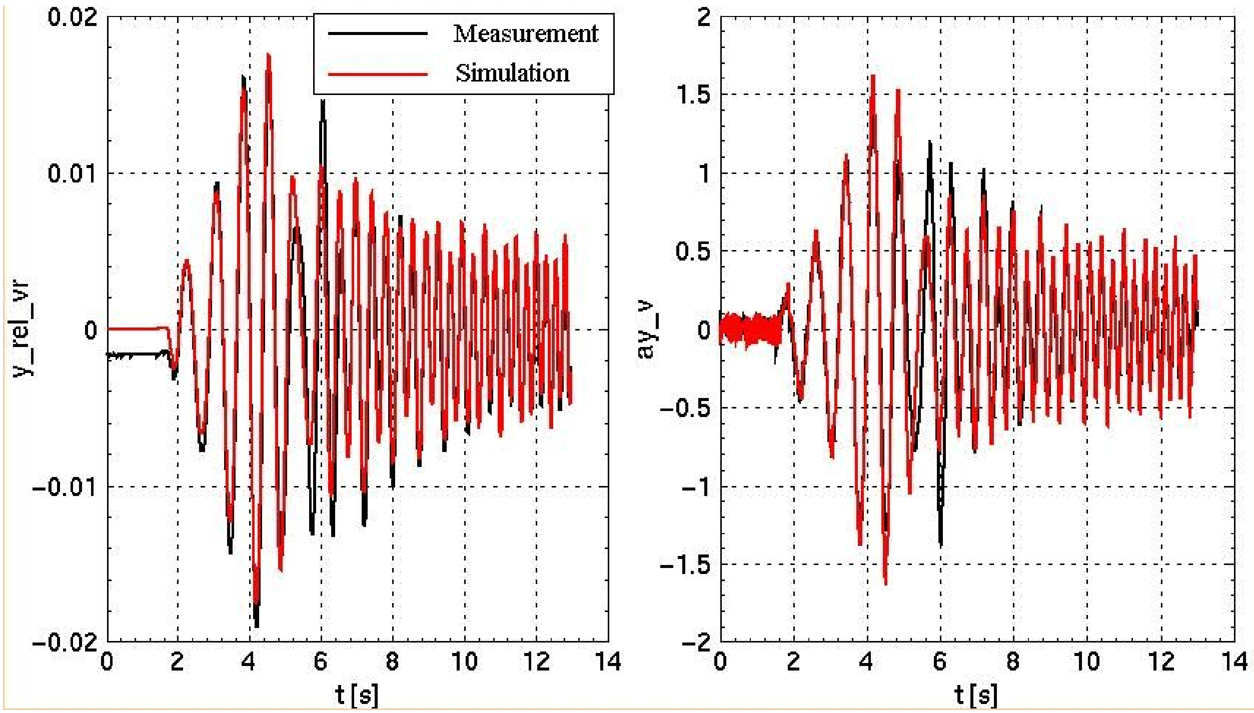

Fig. 19: Identification result (based on an example of lateral stroke and acceleration)

In order to be able to estimate the prognosis capability of the model, the parameter identification is carried out for different excitation modes (Fig. 20). The resulting parameter variances indicate whether the model is able to reflect the reality for different load cases. The variance ranges of the identified parameters determined are included in the following parameter sensitivity analysis, when it is investigated whether the controller is in a position to maintain the quality criteria required with the remaining model uncertainties.

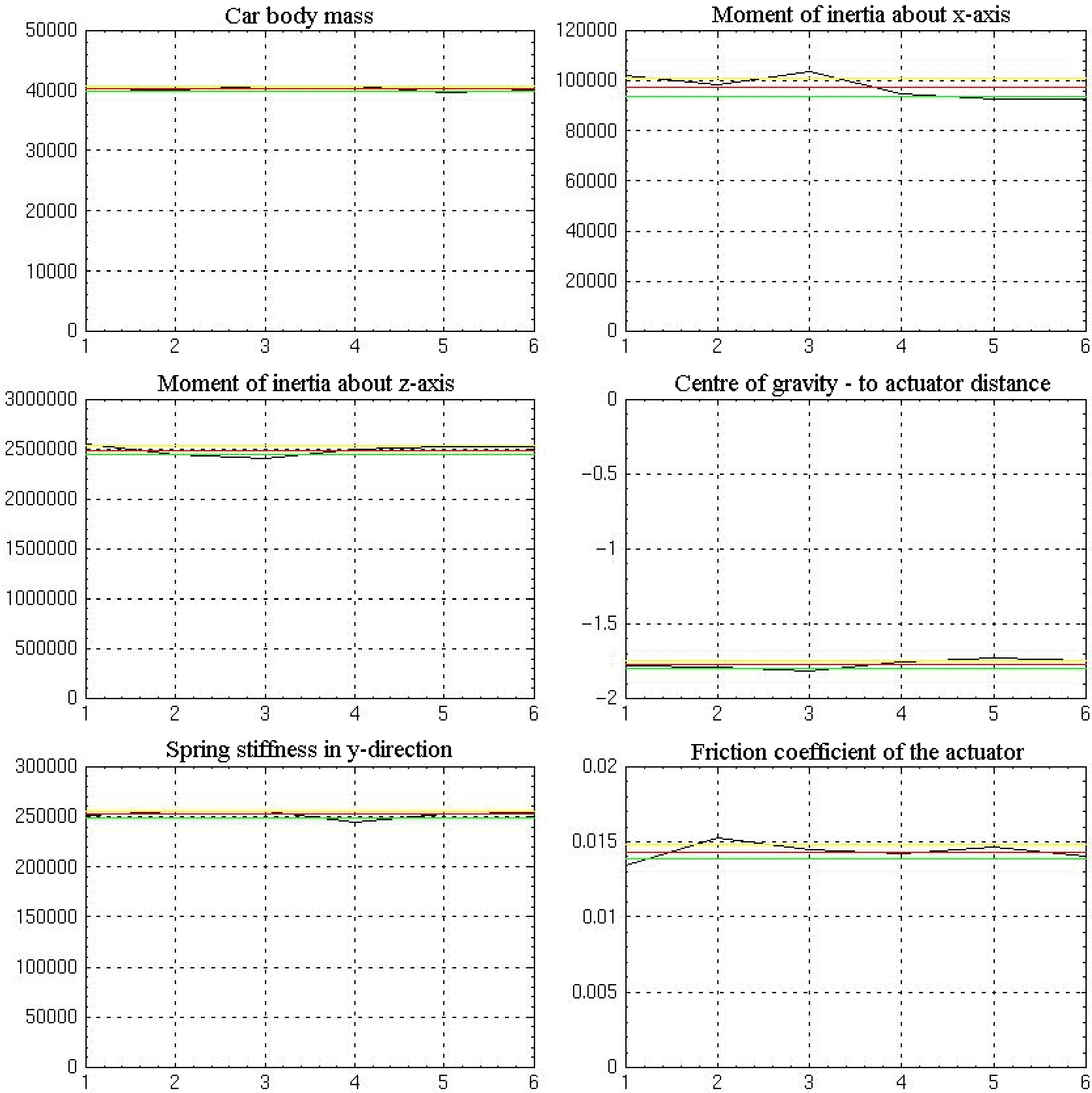

Fig. 20: Parameter differences when identifying six different measurements

Parameter Sensitivity Analysis (PSA)

The parameter sensitivity analysis is an important tool for determining in advance the robustness of the control system developed. In this case, the relative influence of the model parameters with fixed control parameters on the performance of the complete system is checked. Fig. 21 shows the influence of the maximum lateral acceleration occurring on the car body during simulation as an example. This analysis reveals which parameters

- are particularly critical and therefore must be determined with particular care, or
- must be explicitly integrated in the control system via adaptation, or
- must be made uncritical by changing the construction of the hardware, or
- can be used to improve the overall system performance.

The example illustrated show through relative sensitivities close to one in Fig. 21 that the length of the vertical lever arm between lateral actuators and car body centre of gravity has a decisive effect on the maximum lateral acceleration and therefore on lateral comfort. The moment of inertia about the vertical axis is also important.

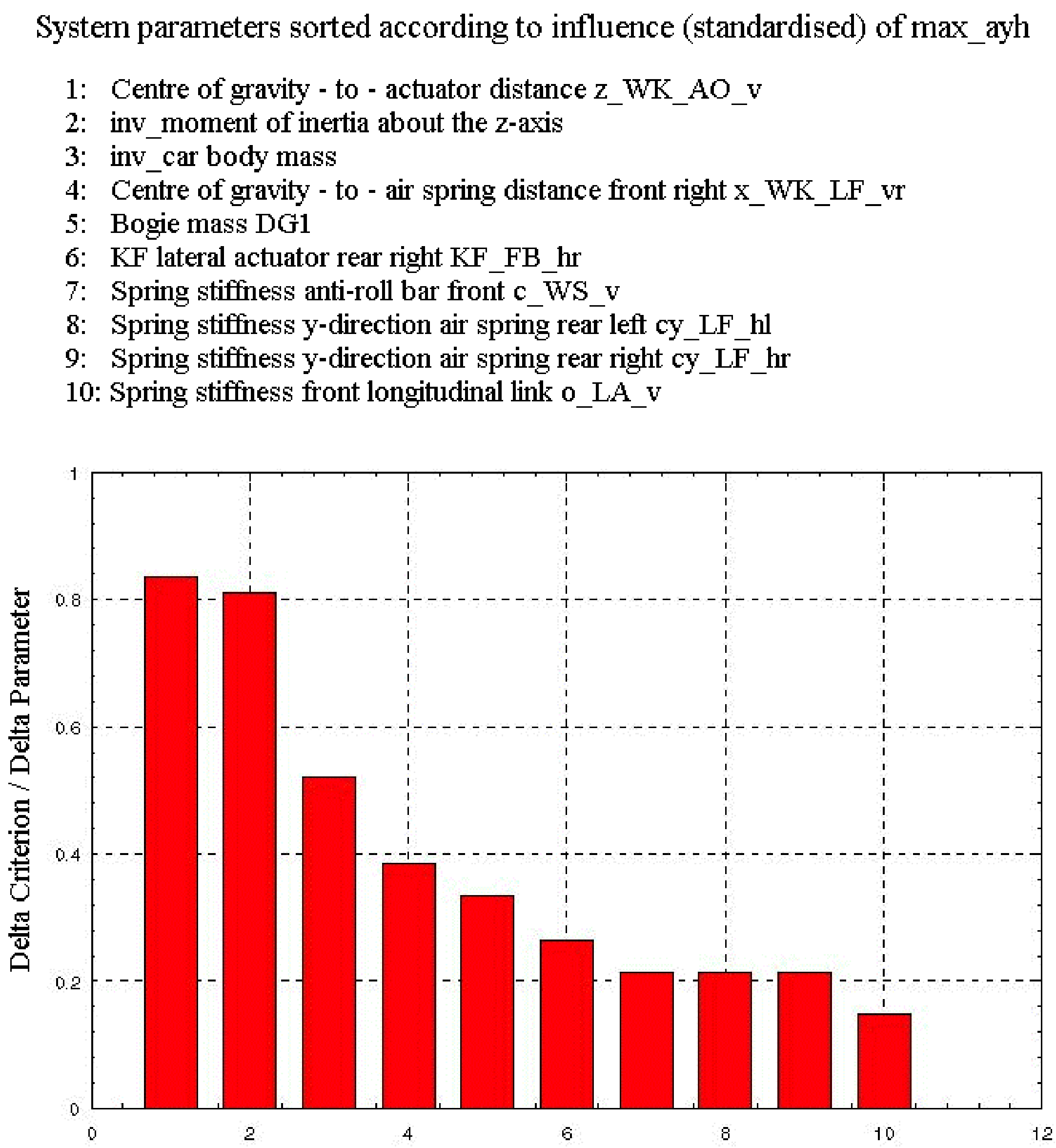

Fig. 21: PSA relating to maximum lateral acceleration

Another analysis can be made by integrating the parameter variances determined from the statistical evaluation of the model parameters identified into the parameter sensitivity analysis. The quality criterion for the control system in this case must be that the control system can guarantee meeting all the requirements in spite of the assumed parameter uncertainties.

Safety and Diagnostic Concept

Operating safety plays an essential role in railway engineering. Since the additional hardware also creates additional risks, new ways have to be adopted for diagnostics and troubleshooting. The usual observation of the overshooting of measured value limits is not in a position in some cases to detect faults (see Fig. 32) and, depending on the system, has a relatively long response time.

Therefore, a model-based diagnostic process was developed during the project which is based on the evaluation of the models derived for the controller design. Consequently, it is ensured on the basis of the validated design model that the system to be controlled behaves during operation as was assumed at the time of the controller design. Deviations from this “nominal behaviour” are recognised as faults first of all and can be used in the case of a redundant sensor design for a diagnosis through cross-comparisons between parallel models. Test results are described in the section entitled Results.

The model-aided diagnostics concept is now explained using a simple scalar system, but one which has structural similarities with the present problem. Let us assume a system with the output variable y and the controller output Q and also the interference variable r.

$$\left(a_2 s^2 + a_1 s + a_0\right) y = \frac{1}{s} Q + \left(a_1 s + a_0\right) r$$

The following are measured:
the acceleration $\ddot{y}$
the relative stroke $y_{rel} = y - r$ and
the acceleration of the interference signal $\ddot{r}$.

The controller output variable Q is known. The differential equation can be transcribed through the direct use of the measuring information available:

$$a_2 \ddot{y} + a_1 \dot{y}_{rel} + a_0 y_{rel} = \int Q dt$$

and

$$a_2 \ddot{y} + a_1 \dot{y}_{rel} + a_0 y_{rel} - \int Q dt \overset{!}{=} 0$$

A criterion is already found here which disappears with the correct – i.e. assumed – system behaviour. In real use, no exact correspondence can be achieved between the basic model and reality. Therefore, this equation is formulated as a residuum:

$$a_2 \ddot{y} + a_1 \dot{y}_{rel} + a_0 y_{rel} - \int Q dt = RES_1 \overset{!}{<} G_1$$

If the limit G_1 is exceeded, it is assumed that the system is no longer functioning correctly. However, it cannot be diagnosed which fault actually exists. If parameter errors are ruled out, then either the acceleration measurement or the relative stroke measurement are incorrect or the controller output has not been correctly converted.

In order to delimit the problem more precisely, a second parallel model is set up. The original differential equation can be transcribed by differentiating twice:

$$a_2 \ddddot{y} + a_1 \dddot{y} + a_0 \ddot{y} = \dot{Q} + a_1 \dddot{r} + a_0 \ddot{r}$$

The measuring equations can now be taken into account again.

$$a_2 \ddddot{y} + a_1 \dddot{y} + a_0 \ddot{y}_{rel} = \dot{Q} + a_1 \ddot{r}$$

This model representation can be converted to a second residuum equation:

$$a_2 \ddddot{y} + a_1 \dddot{y} + a_0 \ddot{y}_{rel} - \dot{Q} - a_1 \ddot{r} = RES_2 \overset{!}{<} G_2$$

The measuring equation $y_{rel} = y - r$ can be used as a further equation. The third residuum equation is created by differentiating twice.

$$\ddot{y}_{rel} = \ddot{y} - \ddot{r}$$

and

$$\ddot{y}_{rel} - \ddot{y} + \ddot{r} = RES_3 \overset{!}{<} G_3$$

$\ddot{y} = \ddot{y}_{rel} + \ddot{r}$ can be eliminated in the first equation by this measuring equation. The fourth residuum equation is then derived from

$$a_2 \dddot{y} + a_1 \ddot{y}_{rel} + a_0 \dot{y}_{rel} = Q$$

$$a_2 \left(\dddot{y}_{rel} + \dddot{r}\right) + a_1 \ddot{y}_{rel} + a_0 \dot{y}_{rel} = Q$$

and

$$a_2 \left(\dddot{y}_{rel} + \dddot{r}\right) + a_1 \ddot{y}_{rel} + a_0 \dot{y}_{rel} - Q = RES_4 \overset{!}{<} G_4$$

The dependencies of the four residuum equations are now entered in a table [Table A].

	$\ddot{y}$	$\ddot{r}$	y_{rel}	Q
RES_1	X		X	X
RES_2	X	X	X	X
RES_3	X	X		X
RES_4		X	X	X

Table A: Dependencies of the four residuum equations

A certain sensor malfunction f can be reliably concluded from the table as a function of exceeding the residuum. It is agreed that overshooting a residuum is evaluated with logic 1 (e.g. RES_1) and undershooting the limit, with logic 0 (e.g. $\overline{RES_1}$). Therefore, the malfunctions are shown as follows [Table B]:

$f(\ddot{y})=$	$RES_1 \bullet RES_2 \bullet RES_3 \bullet \overline{RES_4}$
$f(\ddot{r})=$	$\overline{RES_1} \bullet RES_2 \bullet RES_3 \bullet RES_4$
$f(y_{rel})=$	$RES_1 \bullet RES_2 \bullet \overline{RES_3} \bullet RES_4$
$f(Q)=$	$RES_1 \bullet RES_2 \bullet RES_3 \bullet RES_4$

Table B: Fault diagnostic equations

The process described shows that a detailed diagnosis of individual sensor errors and also the recognition of hardware faults on-line is possible in principle. The process also gives rise to the consideration of whether the reliability in operation of active systems compared with passive systems without sensors does not even increase through on-line diagnosis.

Results

The active lateral comfort control system has been tested on the rolling test rig in Munich (Fig. 22).

Fig. 22: Prototype vehicle on the roller test rig at DB AG in Munich

The performance and functionality of the system was demonstrated on the basis of numerous manoeuvres – performed under different conditions - and predefined criteria. The test programme includes the following track qualities [Table C]:

Abbre-viation	**Section**	**Characterisation**
RO	Rheda-Oelde	Elder track with high frequency faults; Wavelengths up to 70 m; permitted up to 200 km/h
GH96	Göttingen-Hannover Synthetically extended to 100 m Wavelength	New track with marked low frequency disturbances; Minimum excitation frequency at 300 km/h = 0.833 Hz
GH98	Göttingen-Hannover Synthetically extended to 150 m Wavelength	New track with marked low frequency disturbances; Minimum excitation frequency at 300 km/h = 0,555 Hz

Table C: Test manoeuvres

The sections were “travelled” at speeds of 150, 200, 250 and 300 km/h.

To document behaviour with stationary cornering, the two sections GH96 and RO were also travelled with an banking of 150 mm so that it was possible to simulate a 1 m/s^2 static lateral acceleration. The manoeuvres were performed at running speeds of between 200 and 300 km/h.

Dynamic curving behaviour was simulated on the roller test rig by superimposing an increasing banking on the Rheda-Oelde track section. The transition curve was designed so that the maximum lateral acceleration of 1 m/s^2 occurred after 2 seconds at a speed of 300 km/h. The ramp process was carried out at a constant rolling speed.

Figure 23 illustrates first of all the amplitude spectrum of the lateral acceleration signal evaluated when travelling on the straight track of the Rheda-Oelde section. The behaviour of the controlled and also the uncontrolled system is illustrated as a comparison. The amplitude is reduced significantly over the entire relevant frequency range up to 3 Hz.

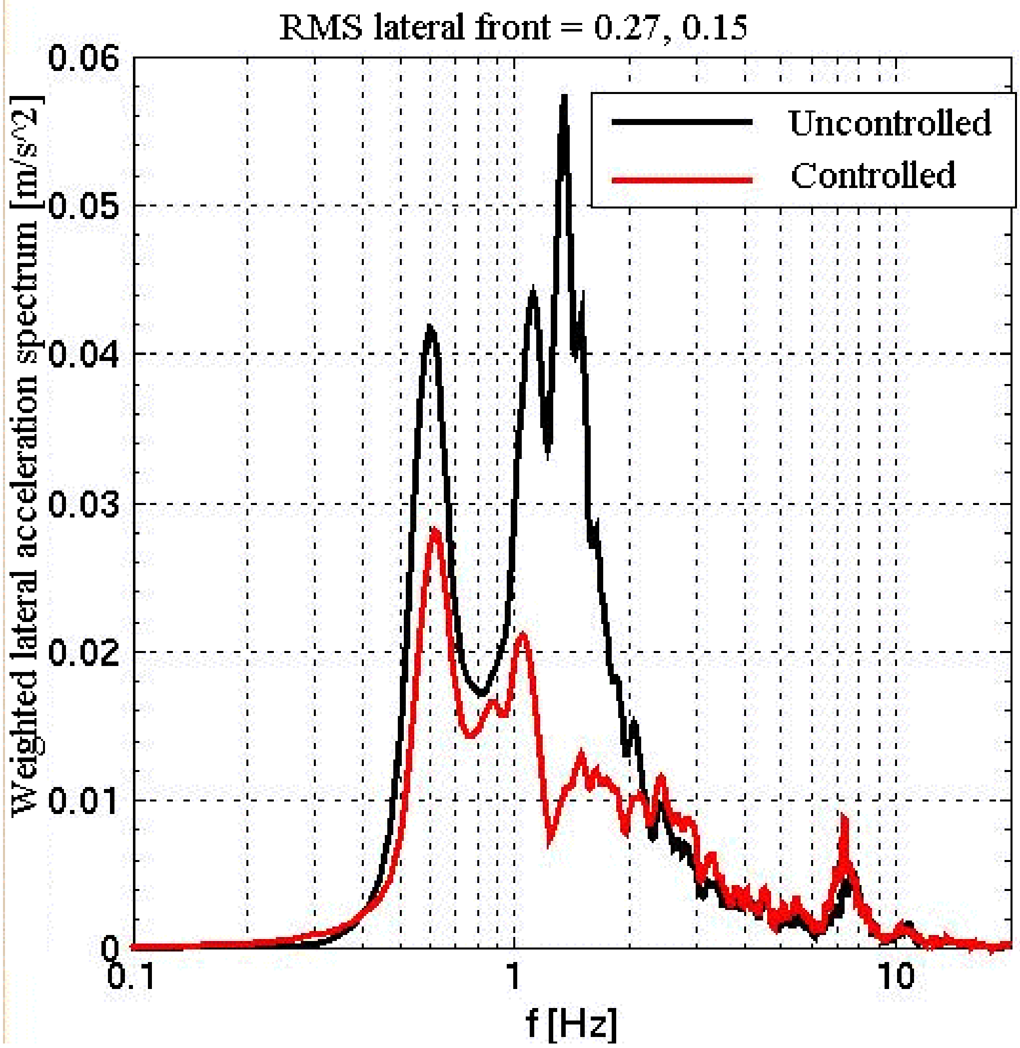

Fig. 23: Results gained on the Munich test rig (amplitude spectra, straight track)

The same applies to the evaluated lateral acceleration spectrum illustrated in Fig. 24, which occurs on the Rheda-Oelde section with a simulated running into the curve. It can be clearly seen that the amplitudes in the frequency range around 1 Hz are drastically reduced by the control system.

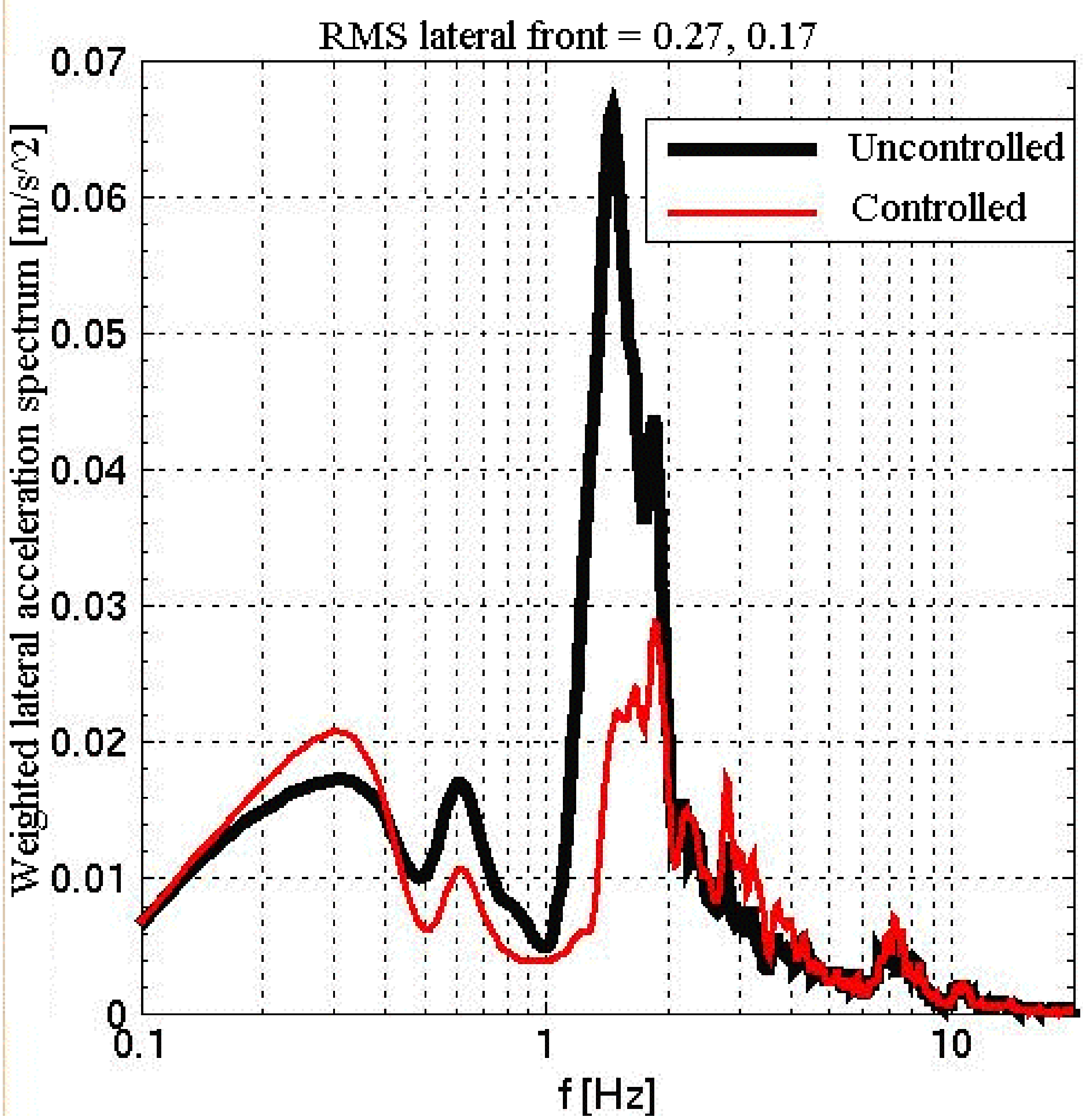

Fig. 24: Results gained on the Munich test rig (amplitude spectra, running into the curve)

Definite improvements in comfort were achieved on all the track sections investigated, as documented below on the basis of the typical rail evaluation ORE-B 153 [ISO2631].

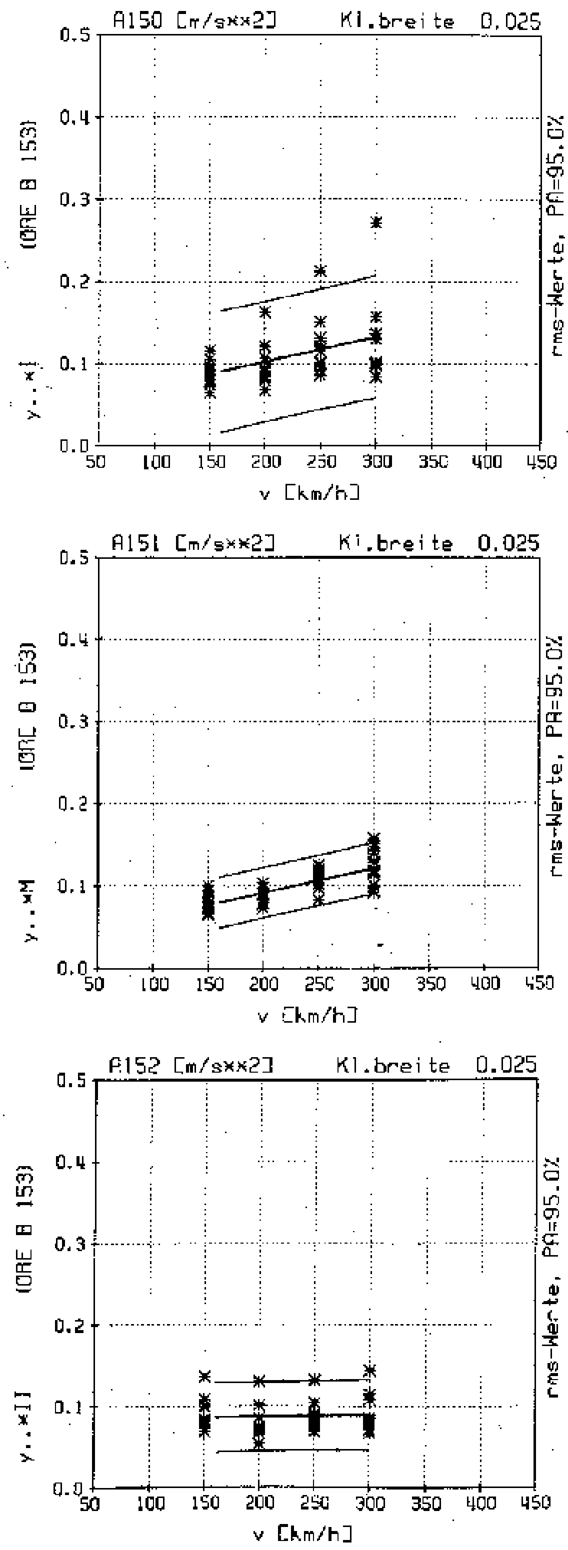

Fig. 25: rms values, active, Rheda-Oelde without dynamic ramp

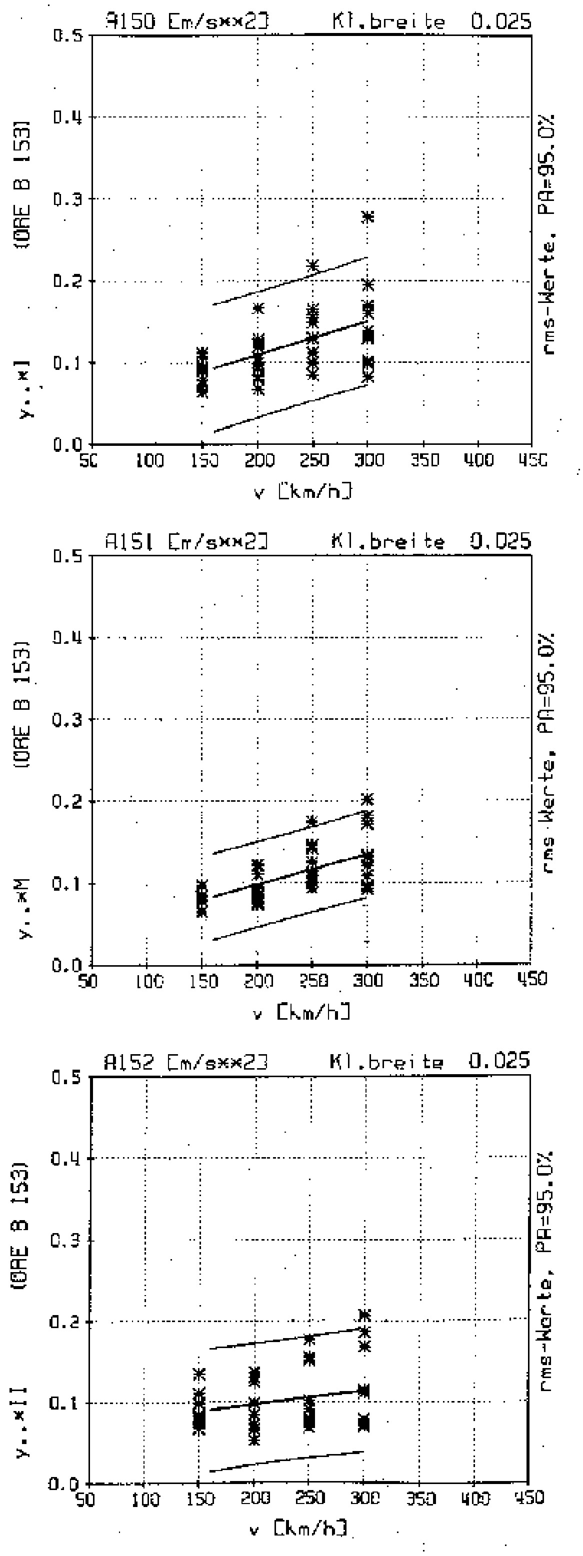

Fig. 26: rms values, Rheda-Oelde, with dynamic ramp

Fig. 25 and 26 show the rms values for the Rheda-Oelde section for four different speeds

- once on a straight track (Fig. 25),
- once with dynamic ramp (Fig. 26),

at the front, centre and back of the car body respectively. rms values fall below 0.2 for all speeds and measuring points and are only slightly worse even when running through a curve.

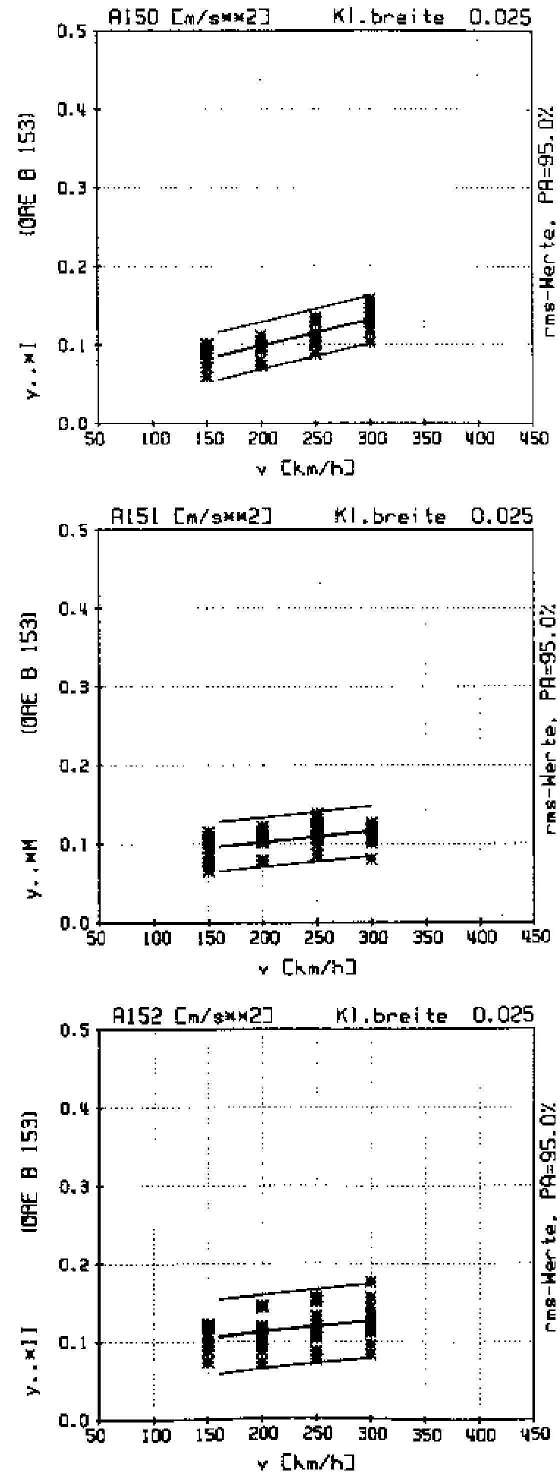

Fig. 27: Active rms values, GH 98

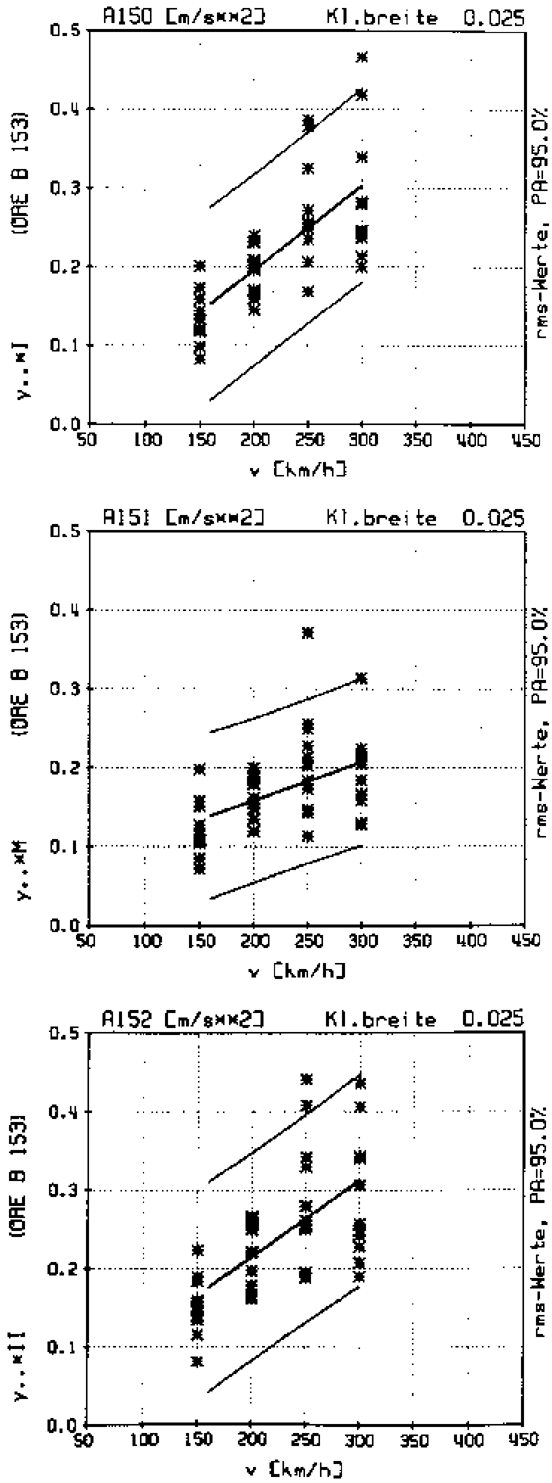

Fig. 28: Passive rms values, GH 98

The same applies to the Göttingen-Hannover 98 track (Fig. 27). In this case, rms values below 0.2 are achieved throughout. In comparison, Fig. 28 shows the comfort values for a journey on the same section with passive lateral suspension. The comfort values measured here are inferior by a factor of at least 2.

Fig. 29 shows the lateral stroke of the front actuator when running into the curve on the Rheda-Oelde section. The train enters the curve at time t=14s. The controller quickly pulls the actuator back into the zero position so that the entire lateral stroke is available for absorbing the lateral track disturbances when running through the curve. At approximately t=34s, the curve changes to a straight track. The lateral stroke required never exceeds 4 cm throughout the manoeuvre. In comparison, the ICE 3 requires a lateral stroke of approximately 9 cm for the same manoeuvre.

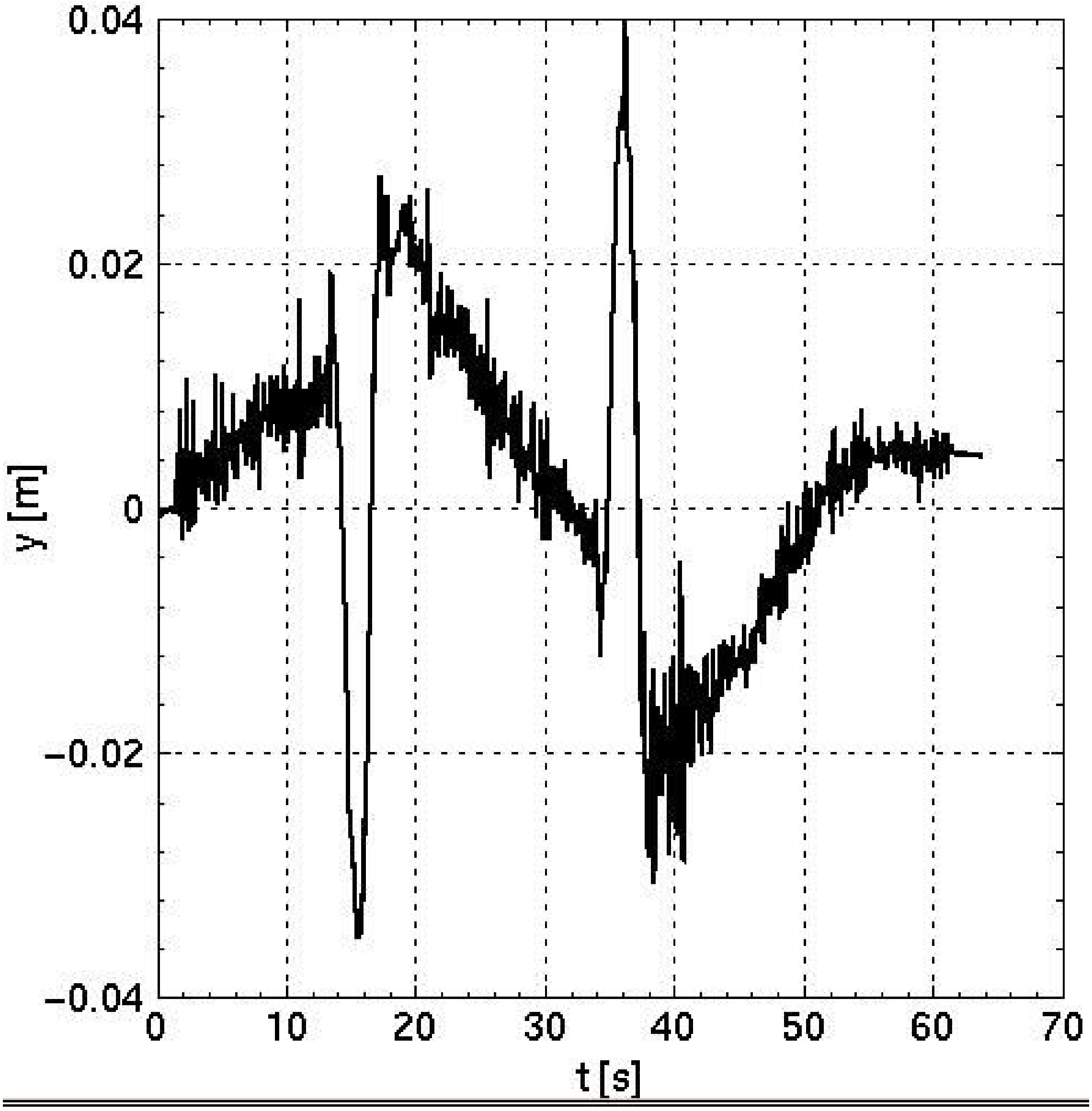

Fig. 29: Lateral stroke required when running into the curve

Fig. 30 shows the net energy consumption of the controller used on all the relevant track qualities. The mean required power remains below 5 kW even at high speeds. It rises only slightly above this limit when running into the curve on the Rheda-Oelde section, far right in this figure.

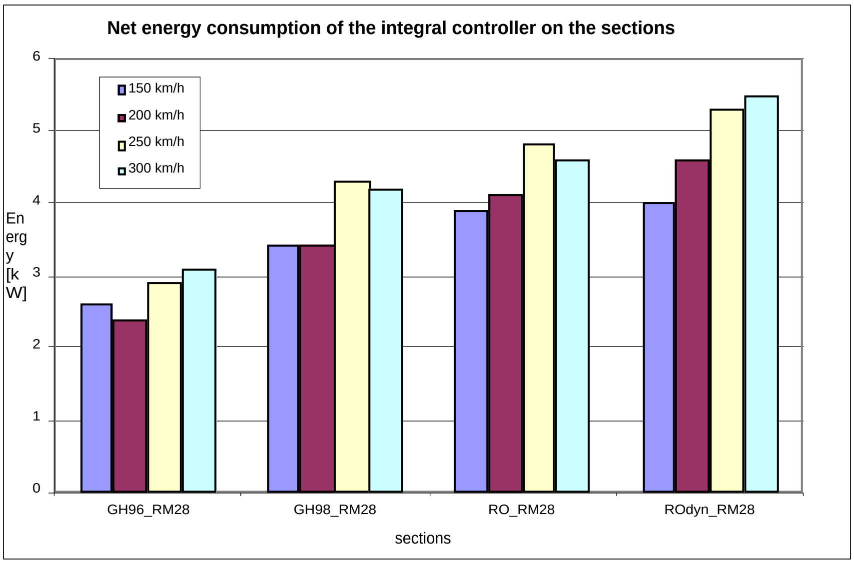

Fig. 30: Net energy consumption of the controller

Figure 31 shows the comfort values again for some typical, series-relevant manoeuvres in the comparison between the controlled and uncontrolled system. It should be noted that these very good results can be achieved with the very low energy consumption shown in Fig. 30 and also maintaining the lateral stroke restriction.

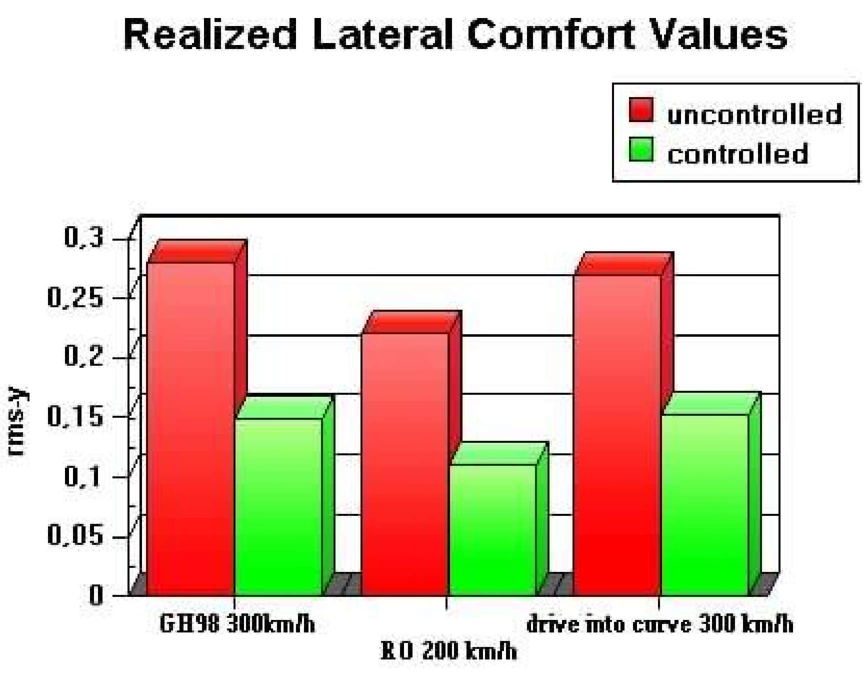

Fig. 31: Comfort evaluation – controlled/uncontrolled

An example of the above mentioned fault diagnosis closes this section. At first Fig. 32 illustrates the effects of the occurrence of a fault if the model-aided troubleshooting system is de-activated. After approximately 0.4 seconds, the relative stroke sensor fails because the power supply plug is pulled out and supplies zero constantly. With the controller in operation, this results in a continuous rise in the actuator force and therefore the total pressure in the system. If it is not switched off, the increasing actuator pressure may damage the hardware. The limit monitors will only recognise the fault when a defined maximum actuator pressure is exceeded.

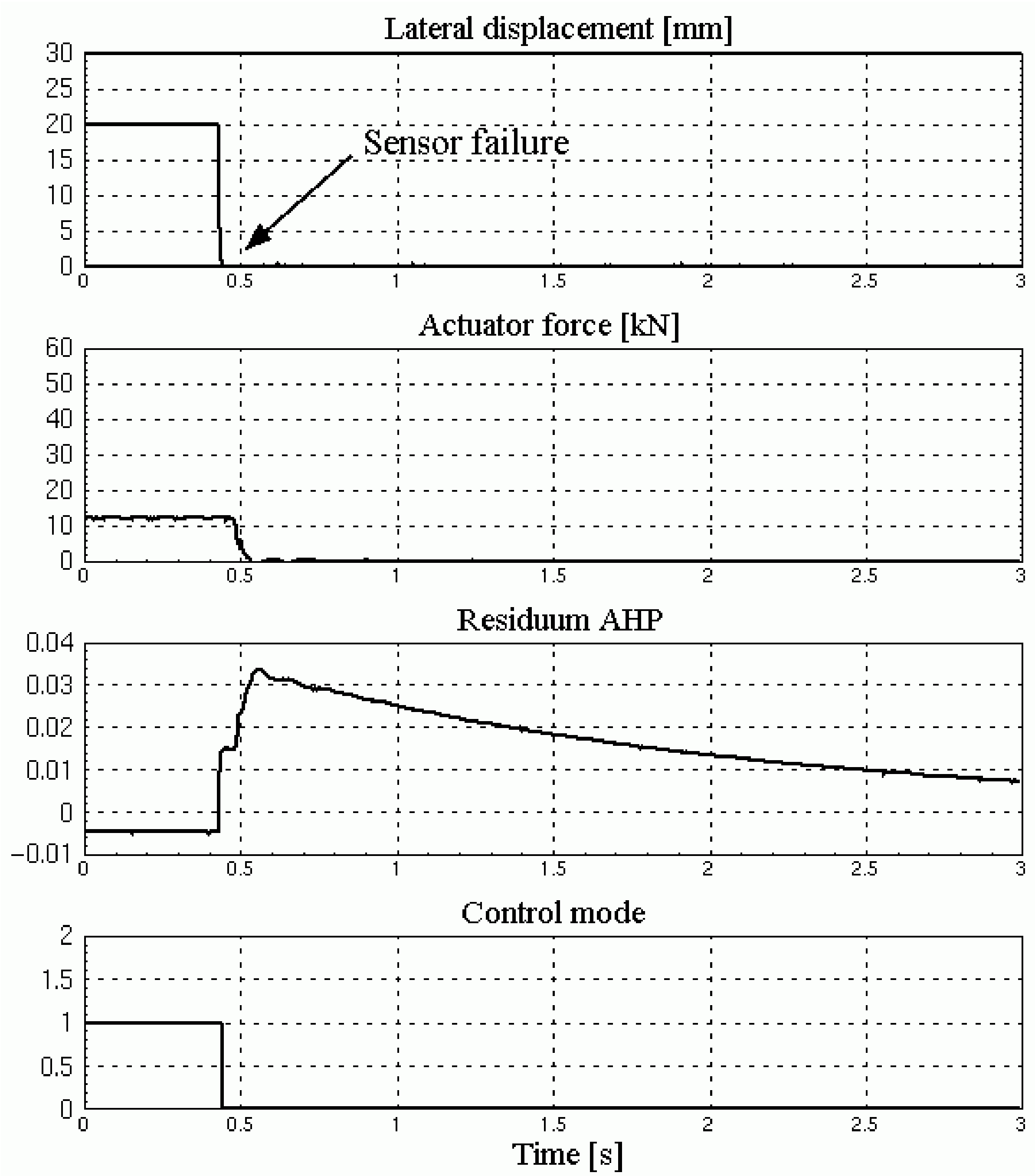

Fig. 32: Without model-aided fault recognition, relative movement sensor fails. Fault not recognised by limit monitor

The situation is different with the model-aided fault recognition system in use as illustrated in Fig. 33. In this case, as soon as the fault occurs, the actuator model residuum increases immediately, because the actuator behaviour deviates from that of the reference model. The limit is exceeded after a few milliseconds. The controller is de-activated and the actuators are depressurised. Therefore, the defective build-up of pressure in the actuator is prevented at a very early stage.

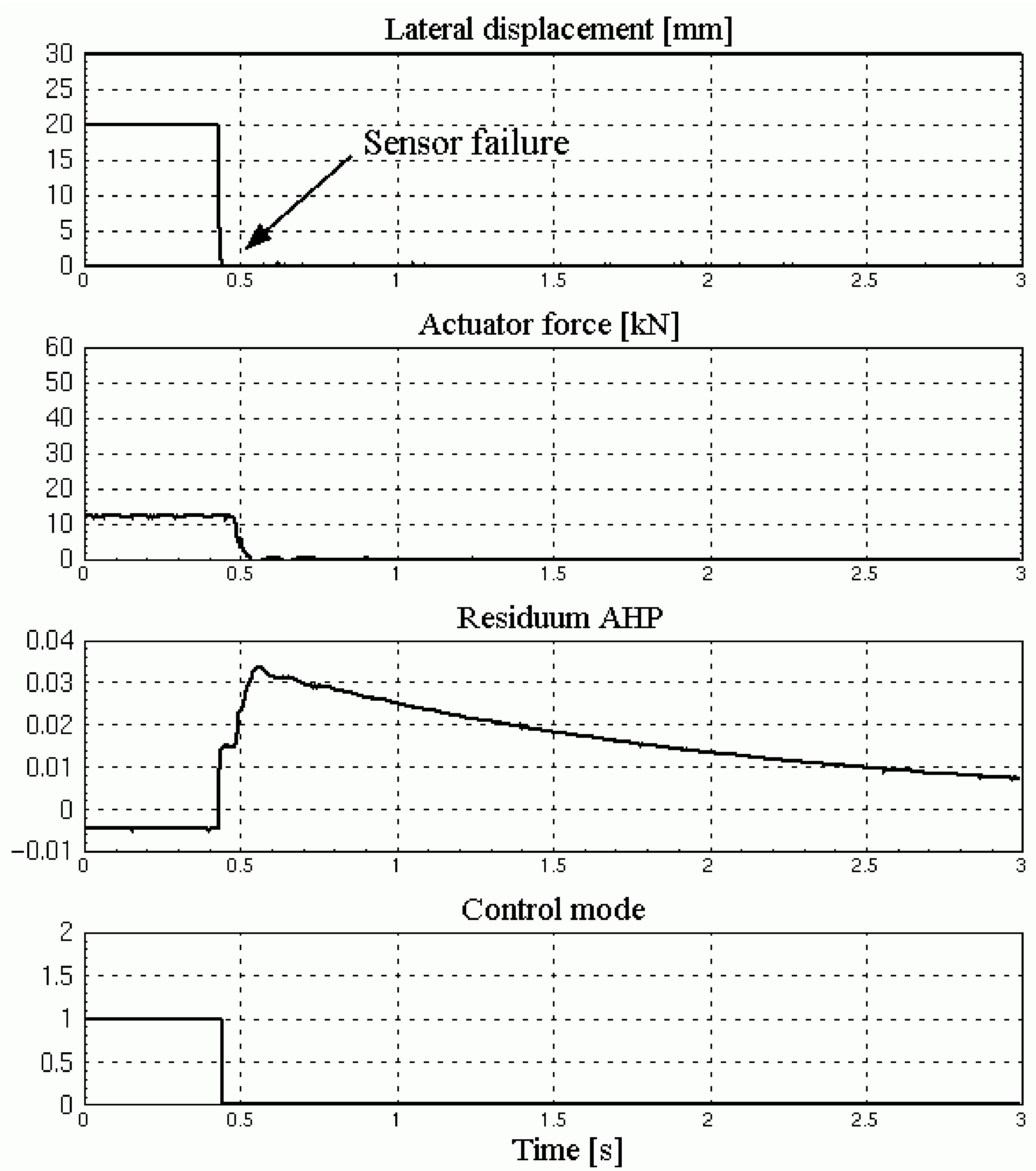

Fig. 33: With model-aided fault recognition. Relative movement sensor fails. Fault recognised immediately

To sum up, it was possible to achieve the following results by implementing the new integral control concept described:

- A model of the prototype, which has been validated on the basis of parameter identification with real measuring data, is available (Fig. 19).
- Through the control system, a reduction of the comfort values to 50% compared with the passive design can be achieved for all series-relevant track qualities (Fig. 23 to 28).
- The power required by the controlled actuators remains below 5 kW net for all sections (Fig. 30).
- Even when running into a curve, the maximum lateral stroke required remains below 4 cm (Fig. 29).

Conclusion

In the "Active Bogies" project DaimlerChrysler Research&Technology, working with colleagues at Bombardier Transportation and with the assistance of Mannesmann/Rexroth, was able to present an active bogie system in just 20 months. This success can be regarded as a breakthrough in bogie technology. In this case, the implemented solution not only has a theoretical character, but it has been validated with real bogies mounted under a real vehicle, and its concept is already close to a series solution in many respects.

One reason for the short development time is the consistent application of modelbased tools and their continuous adjustment to the hardware reality. It has been shown that even a controller of this complexity can be completely designed and parameterised in the computer without having to make any additional adjustments during the operation of the prototype. Subsequent improvements have been exclusively triggered by model improvements. This analysis and design technology path opens up a wide field of savings - both in time and cost - but also creates a deeper understanding of the properties of the product and creates an opportunity for wide-reaching improvement processes that are computer-aided but do not lose contact with reality.

This methodology, applied in combination with advanced actuator technologies in a systems oriented bogie concept development, opens the potential for real Mechatronic Bogie structures with economical character. Thus a new generation of bogie configurations – especially for high speed applications – has just been launched.

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Formula symbols

X	Co-ordinates in the direction of travel; front is positive (according to DIN 7000)
Y	Co-ordinates in lateral direction; left is positive (according to DIN 7000)
Z	Co-ordinates in vertical direction; up is positive (according to DIN 7000)
W	Angle co-ordinates, rolling; clockwise direction positive (according to DIN 7000)
N	Angle co-ordinates, pitching; front down is positive (according to DIN 7000)
Psi	Angle co-ordinates, yawing, front left is positive (according to DIN 7000)
Q	Mass inertia
K	Constant
A	Area
C	Spring stiffness
D	Damping
R	Local vector in space
V	Volume
P	Pressure
U	Input variable
T	Transformation matrix with geometric dependencies
K	Model matrix
Gmodell	Complete model
RM	Reduced model
MI	Modal integration
SP	Centre of gravity
Q	Oil volume flow
F	Force
L	Length
Lq	Lateral length

Indices

WK	Car body
DG	Bogie (DG1 is front)
Akt	Actuator
Rel	Relative movement
V	Valve
D	Throttle
FS	Spring memory
R	Right

L	Left
PF	Primary spring
RSL	Wheel set bearing
HK	Main chamber, actuator
V	Front
H	Back
VBS	Vertical acceleration sensor