ACTIVE SUSPENSION DESIGN FOR A LARGE SPACE STRUCTURE GROUND TEST FACILITY

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INTRODUCTION

The expected future high performance requirements for Large Space Structures (LSS) enforce technology innovations such as active vibration damping techniques e.g. by means of structure integrated sensors and actuators. The implementation of new technologies like that requires an interactive and integrated structural and control design with an increased effort in hardware validation by ground testing.

During the technology development phase generic system tests will be most important covering verification and validation aspects up to the preparation and definition of relevant space experiments. For many applications using advanced designs it is deemed necessary to improve existing testing technology by further reducing disturbances and gravity coupling effects while maintaining high performance reliability. A key issue in this context is the improvement of suspension techniques.

The ideal ground test facility satisfying these requirements completely will never be found. The highest degree of reliability will always be obtained by passive suspension methods taking into account severe performance limitations such as non-zero rigid body modes, restriction of degrees of freedom of motion and frequency response limitations. Passive compensation mechanisms, e.g. zero-spring-rate mechanisms, either require large moving masses or they are limited with respect to low-frequency performance by friction, stiction or other non-linear effects.

With active suspensions these limitations can be removed to a large extent thereby increasing the range of applications. Despite an additional complexity which is associated with a potential risk in reliability their development is considered promising due to the amazing improvement of real-time control technology which is still continuing.

THE ACTIVE SUSPENSION TEST SETUP

The basic idea of an active suspension device is the combination of a bias spring designed to support the weight load at the hinge, augmented by an actuator which in a defined range compensates for the spring stiffness via positive displacement feedback (Figure 1). The artificial negative stiffness adds up with the passive one to an overall stiffness close to zero, without the penalty of excessive weight or undesired additional eigendynamics. The potential advantages in performance are as follows:

- the overall stiffness is easily adjustable by choosing an appropriate gain factor,
- the relative error of the resulting differential dynamic forces can be kept small by using appropriate precision control hardware and software,
- an increased dynamic range becomes feasible owing to the reduced overall moving suspension mass.

As a reference test item a beam shaped truss structure has been selected which can be regarded as a typical substructure for LSS or at least as a basic element of potential space experiments. The principal test setup, as depicted in Figure 2, comprises three functional levels which logically are strictly separated:

- the active suspension mechanism with local control of associated suspension rods,
- the active vibration control which "sees" an ideal truss beam under approximately zero-g conditions,
- reference sensors and stimulation actuators for test evaluation.

In contrast to passive methods the active suspension permits optimal decoupling of bending from torsional degrees of freedom and the full compensation of pendulous rigid body modes. As shown in Figure 3 this becomes possible by equidistant clustered suspension units with the horizontal ones exhibiting negative overall stiffness to counteract gravity induced pendulous forces. Using kinematic decoupling algorithms, this technique can also be applied for non-symmetrical cross-sections. A practical advantage is the shortness of the suspension cable which is limited primarily by mechanical design tolerances, e.g. with respect to the radial load of the linear guidance bearing.

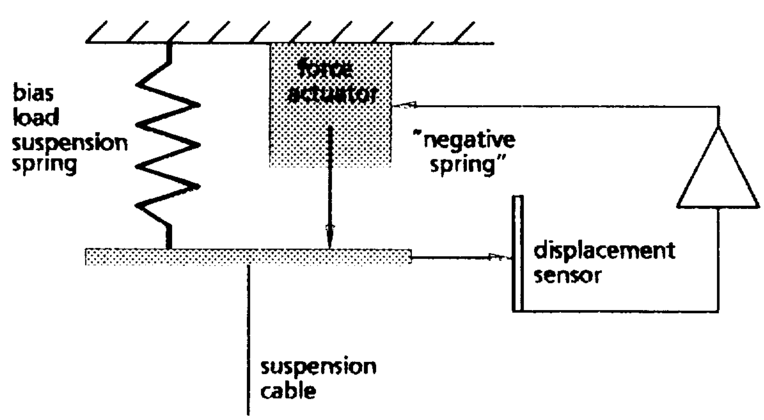

Figure 1. Active suspension principle.

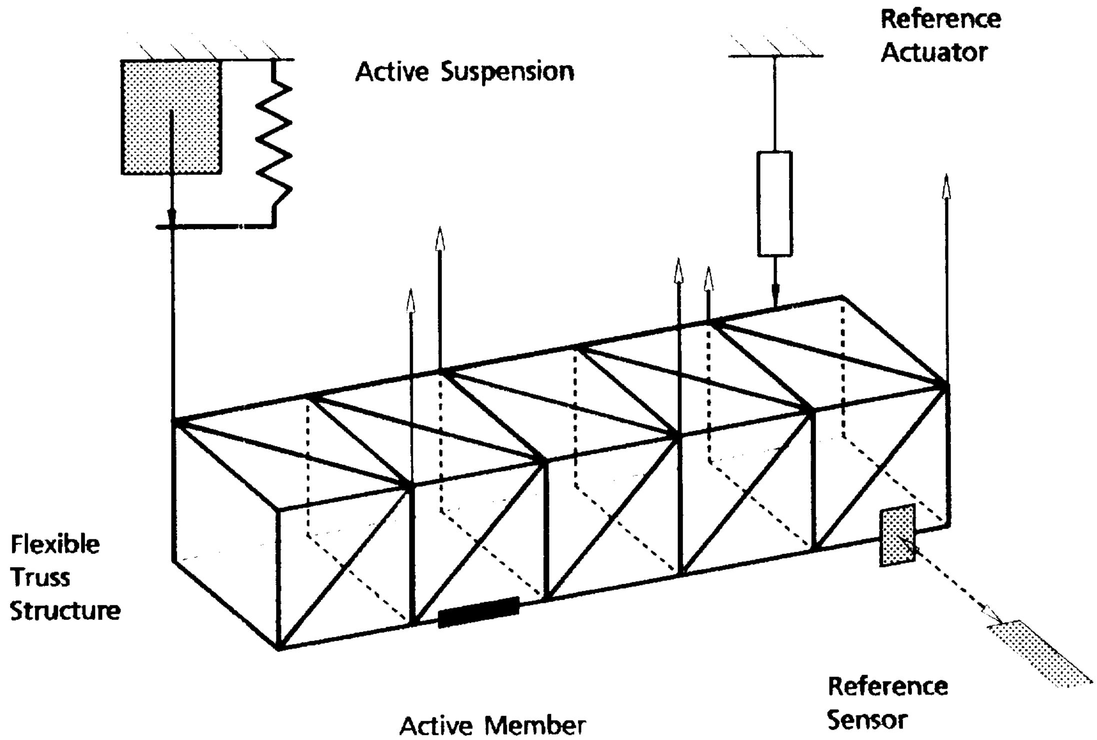

Figure 2. Principle configuration of test setup.

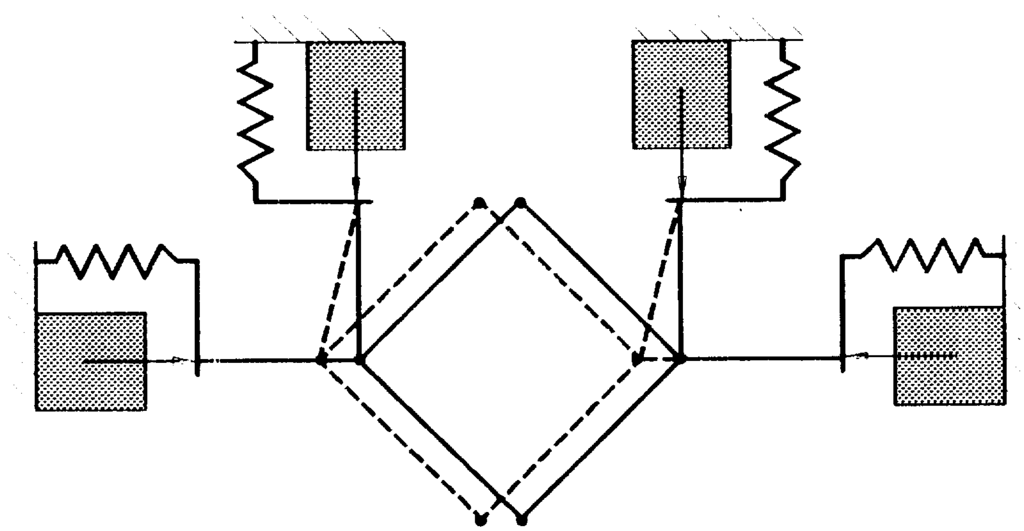

Figure 3. Optimal decoupling from gravity forces by an active suspension.

With such a compensated test setup new applications become feasible, since the suspension rods no longer have to be attached to long distance fixtures usually mounted at the lab ceiling. With a moving base it will be possible to investigate structure dynamics and control under large angle motions, e.g. slewing maneuvers. Although full gravity compensation by this suspension technique is approximative, depending on the number of hinge points, this new testing approach will help to enhance the design validation relevance for many applications.

DESIGN REQUIREMENTS FOR AN ACTIVE SUSPENSION UNIT

There are three basic functions which the suspension unit has to support with a defined accuracy:

- the passive weight compensation being free from hysteresis and acting with limited relative stiffness about equilibrium,
- the static compensation of the spring characteristic with low residual stiffness and noise,
- the dynamic compensation of moving mass inertial forces depending on the required operational bandwidth.

The interaction of forces and the compensation loops are shown in the functional diagram of Figure 4. The active functions have different associated frequency bands. While static compensation requires low-pass filtering to prevent measurement noise from deteri-

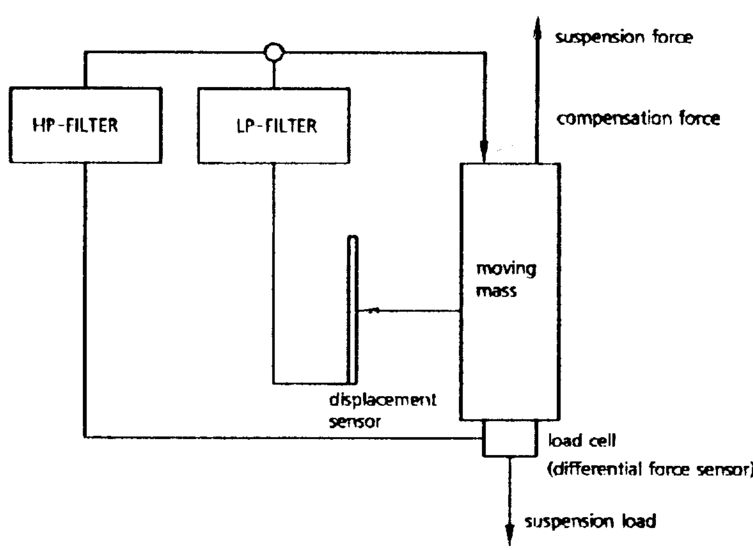

Figure 4. Active suspension force compensation loop

orating performance, the dynamic compensation is operating via high-pass filter to exclude the bias weight load measurement from feedback.

The system requirements to be satisfied by the basic functions are defined as to comprise a large number of applications which are likely to come in the foreseeable future. A large class of generic LSS control validation models including unscaled but moderately sized space experiments [1] are assumed to be covered by the following requirements:

Table 1. Requirements for an Active Suspension Unit

bandwidth	<30 Hz	(structural damping assumed with no additional active vibration damping required for higher frequencies)
load per hinge	100 N	(may be higher on expense of bandwidth)
max. displacement	± 3 cm	(including open-loop tests)

The size of the suspension unit is largely defined by the required weight load and displacement range. This affects the passive spring device and the related force interaction with the actuator. But respective specifications depend on the selected design approach and hence not directly on the test item itself.

The dynamic environment however has a direct impact on the design. Respective actuator reaction force compensation requirements depend largely on the test operation mode:

- Closed-loop tests:

  Amplitudes are small with active vibration damping and hence not considered critical with respect to the dynamic feedback of inertial suspension forces.

- Open-loop tests:

  Depending on the stimulation mode, i.e. on the distribution of input energy over the structural modes large vibration amplitudes may occur. Therefore this operation mode is regarded most critical.

The worst case is obtained with all participating modes oscillating in-phase, e.g. immediately after an impulsive stimulation. More relevant for open-loop damping measurements however are sinusoidal and pass-band noise tests which permit extended measurement duration. These tests exhibit less overshoot and hence require less maximum dynamic compensation forces.

Assuming that the modal frequency distribution of a truss beam will be not too much different from that of an Euler-Bernoulli beam the assessment of the required dynamic compensation forces can be done analytically. The worst case in terms of response amplitude is obtained for the suspension unit at the tip end of the beam with collocated stimulation. At this location then all mode shapes add up with a non-zero contribution as indicated in Figure 5.

The dynamic compensation force is derived from the required stimulation amplitude permitting a reasonably accurate evaluation of the structural motion. For wide-band excitation the unequal energy distribution with respect to the structural modes requires the full utilization of the admitted displacement range in order to include the low vibration frequencies. For single frequency excitation on the other hand the required peak acceleration or displacement depends on the evaluation instruments used. In both cases, the structural damping is not relevant since the stimulation input will always have to be adapted to the required measurement output.

The undefined input stimulation quantity is eliminated from the beam model equations by utilizing the expected output value as outlined above. The analysis for the three types of stimulation, which is omitted here, yields to the following results for the respective specific compensation forces:

- impulsive: $\ddot{y}_{max} < 45 \frac{m}{s^2} \approx 4.6g \quad (t = \Delta t)$
- white noise: $\ddot{y}_{max} < 1g \quad (3\sigma)$
- sinusoidal: $\ddot{y}_{max} < 1g$

These requirements are not very challenging for common actuators and motors available today. The low-noise performance however requires a more or less non-conventional design at the expense of available output power. This becomes evident from the detailed design analysis.

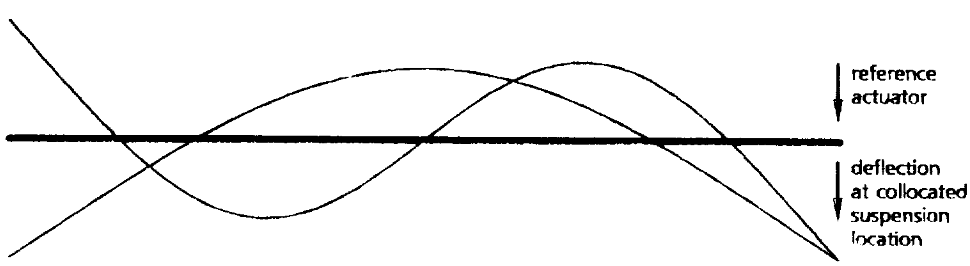

Figure 5. Beam model for dynamic analysis.

DESIGN CONCEPT

The minimization of disturbance and noise associated with the active suspension design requires precision technology. On the other hand high precision hardware may be extremely expensive, which then becomes a practical realization problem, particularly if many of these suspension units are required for an LSS control test setup. With the increasing capability of modern real-time control processors, however, which are available at even decreasing cost, this problem can be solved. A non-linear characteristic, e.g., can more easily be calibrated by respective functions in the processor than by complex mechanical adjustment procedures.

The design is determined by the method used for weight support and by the interaction with the compensation actuator. The basic solution for the passive weight support is a torsional spring. It is superior to a coil spring in terms of preloading capability and less sensitive with respect to the dynamic interference with the suspension forces. A design problem is however the force diversion, as the variation of the effective lever arm with the angle of rotation has an influence on the resulting suspension force vector.

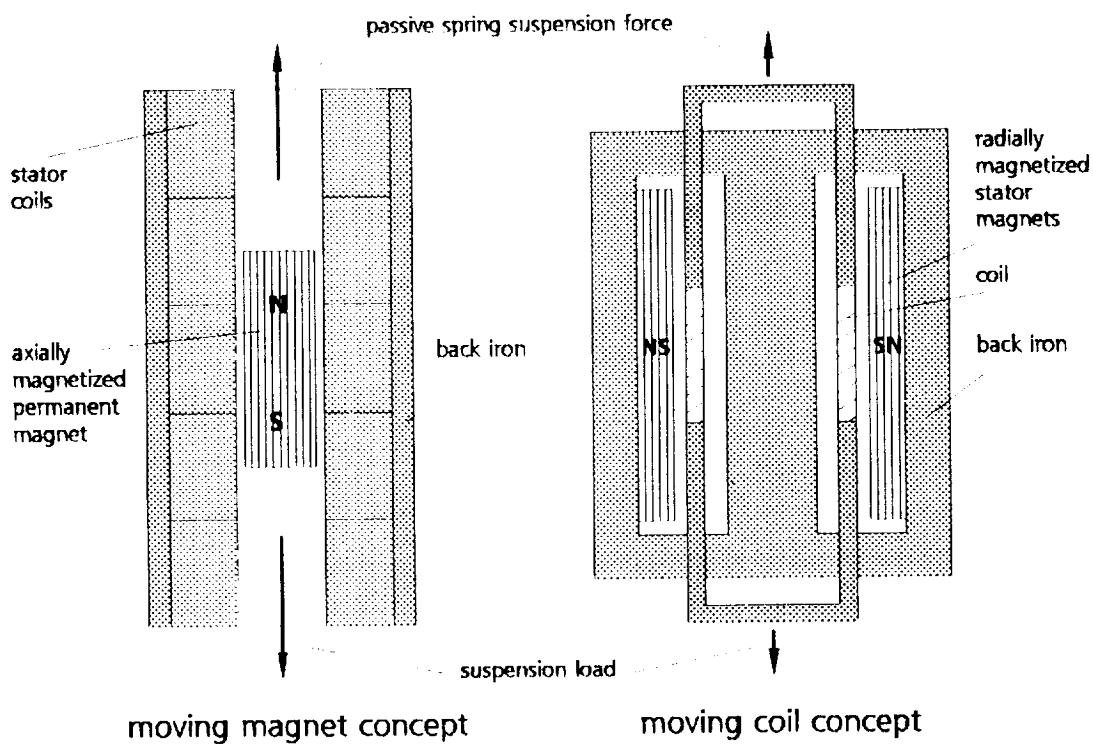

Figure 6. Electrodynamic actuator design options.

To minimize both friction effects and overall moving mass a linear motor design has been selected delivering the compensation forces. The optimal design in terms of noise and repeatability is the electrodynamic actuator which in contrast to ordinary DC-motors is characterized by

- absence of hysteresis effects which is a prerequisite to achieve high linearity performance,
- virtually noise free performance since commutation or cogging effects are excluded,
- direct force control without using high-authority control loops.

In principle two types of electrodynamic actuators are possible as illustrated in Figure 6:

1. A permanent magnet is moving in the center of controlled stator coils.

   Mechanically, this design is simple since the moving part is a concentrated mass in the center, collocated to the external load force vector. With control applied to the stator, there are no electrical leads to the moving part. Moreover, a modular adaptation to an arbitrary working range is easily possible by adding further stator coils. However, there is a long distance to the magnetic return path which decreases the force to input power efficiency.

2. An actuator coil is moving in the air gap of a radial magnetic field which is completely guided in an iron feedback.

   This design is mechanically more complex since the air gap enclosing the moving actuator coil must be narrow to minimize the magnetic stray field. The efficiency however is by principle exceeding that of the moving magnet concept.

Problems which are common to either approach are design limitations due to the non-linearity of the magnetic field characteristic. Moreover, to avoid stick-slip effects, the moving part has to be integrated into special linear contactless bearings. Hence a special development is necessary, taking into account the specific drawback features in a trade-off analysis.

Moving Magnet Actuator Concept

Due to the wide air gap associated with this approach, output force is largely dependent on displacement. As shown by the typical characteristic in Figure 7, force rapidly decreases with increasing displacement from a maximum. That is defined by the tip end of the permanent magnet being located approximately in the center of the excited coil. With increasing ratio of coil length vs. displacement the decay becomes smoother, but at the expense of a reduced effectiveness of force vs. electrical input power.

The solution to this problem is already indicated by Figure 6, where the moving magnet option is illustrated by an assembly of axially stacked coils which are simultaneously controlled depending on the relative magnet position. The efficiency of this device is dependent both on the geometrical dimensions of magnet and coil and on the control method.

Force controlled motion over a long distance may be very useful also for other applications such as handling objects under zero-g conditions. Potential benefits are in addition to the inherent modularity the accuracy and reproducibility of the exerted force.

Optimal Electromechanical Design

The optimal design of a moving magnet actuator is not straightforward due to the complex electromagnetical field distribution over a large air gap. Hence the axial force, which according to Ampère's law results from the cross-product of local field vector and the current through the stator coil windings, requires the solution of a volume integral of the form

$$F_z = G_\varphi \int_V B_r dV \tag{4.1}$$

with

G_φ = current density with respect to the coil cross-section

B_r = radial component of the emerging magnetic flux density

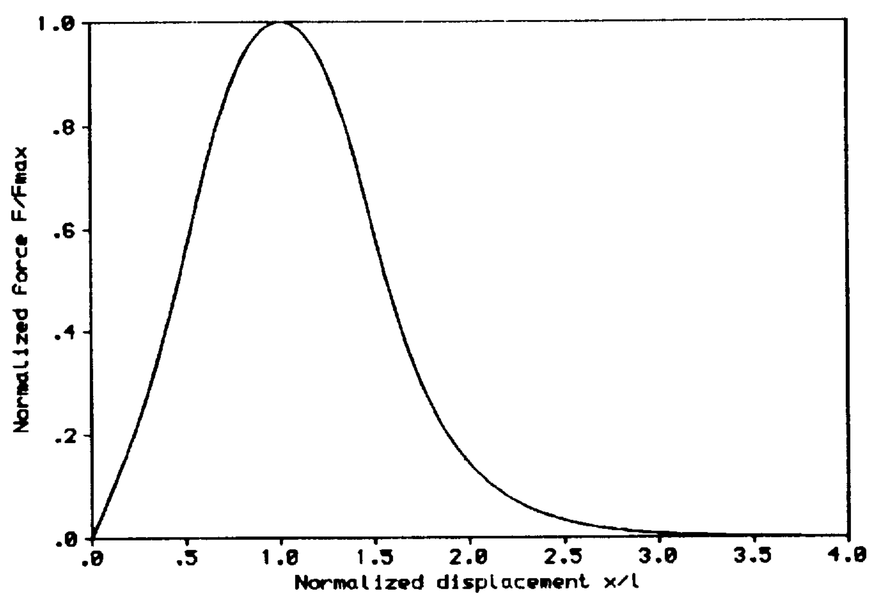

Figure 7. Coil/magnet force vs. displacement characteristic.

V = volume enclosed by the coil winding

To obtain a more general solution specific design parameters shall be eliminated from evaluation of Eq. (4.1). Assuming rare earth magnets, a remanent flux density of 1.2 T can be expected which is about the maximum to be achieved for modern NdFeB-magnets. Specific coil wiring data are replaced by more general parameters yielding an expression for the current density which is only dependent on coil geometrical dimensions and input power:

$$G_\varphi = \frac{1}{2}\sqrt{\frac{\sigma}{\rho}\,\frac{P}{(r_o^2 - r_i^2)l}} \tag{4.2}$$

with

$\sigma = \sigma_d d^2$ = normalized wiring constant

σ_d = Ampère turns per unit cross sectional area for wire diameter, d

ρ = specific ohmic wire resistance

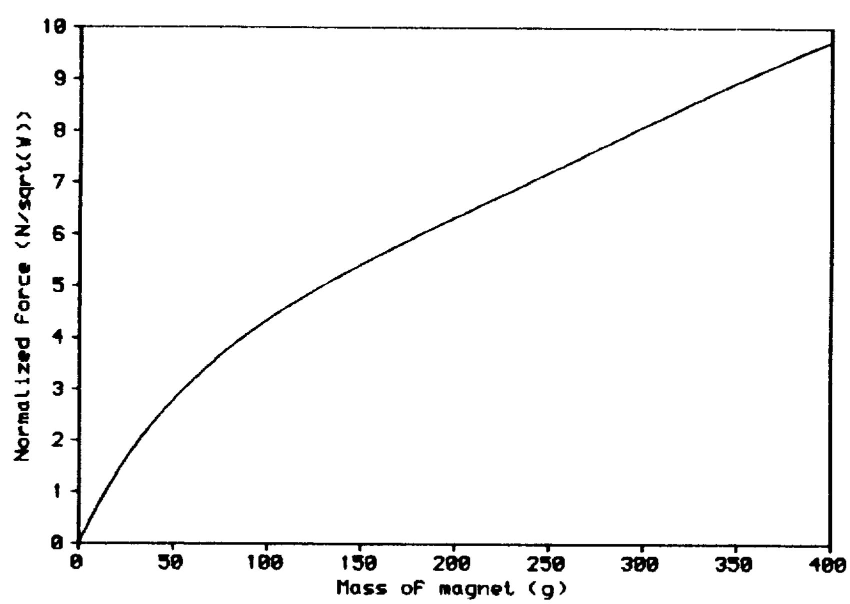

Figure 8. Optimal design diagram.

r_o , r_i = outer, inner coil radius

l = coil length

P = ohmic power loss

Then, after dividing the output force by the square root of input power, a function is obtained where only the geometrical coil/magnet parameters remain to be optimized. The result, which is computed by iteratively solving the magnetic field equations and the force integral, is shown in Figure 8. The force calculation is based upon the reference position and the ratio of magnet to coil length of two by one as shown in Figure 6, with the two coils at the tip ends of the magnet being simultaneously excited.

The required compensation force, f, depends on the dynamic force to moving mass ratio, q_m, and on the spring restraint force, F_c, yielding:

$$f(m) = F_c + q_m (m + M) \tag{4.4}$$

with M denoting the moving mass without the actuator magnet mass, m; i.e. the equivalent spring mass, attachments, rods etc.

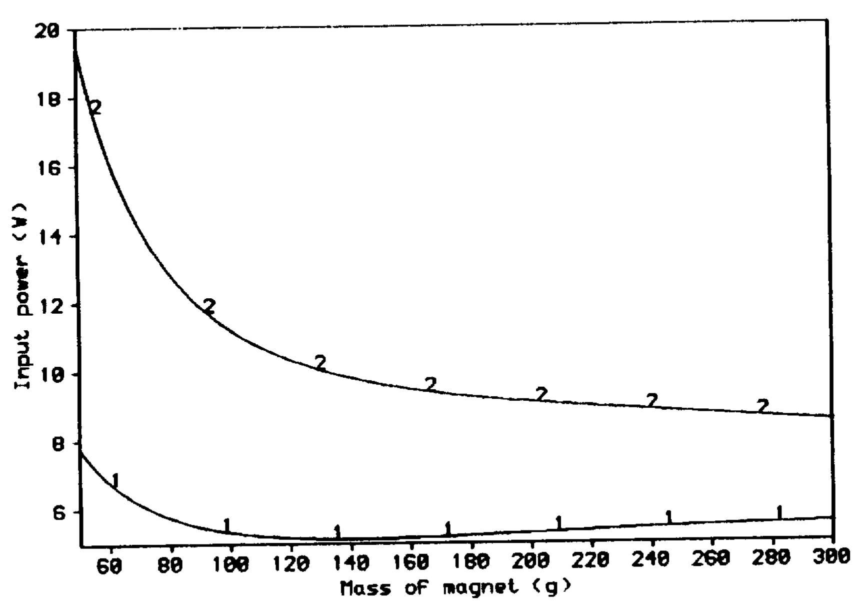

Figure 9. Input power vs. moving mass performance
(graph 1: M = 100g, graph 2: M = 200g).

The most decisive design data to evaluate trade-off solutions are input power and magnet mass. A respective function is obtained by elimination of force using Eq. (4.4). The result is shown in Figure 9 for two different masses, M, and with a maximum spring restraint force F_c of 1 N.

Inspection of these performance diagrams shows at least with respect to the electromechanical voice coil design, that solutions with moderate technical effort in terms of input power and size of the moving magnet can be found to be realized with reasonable costs.

Optimal Gain Control

The actuator force is generated by simultaneously controlling all coils by dedicated wide-band current amplifiers.Their input signals are computed from the force command signal via individual displacement dependent gain factors.

The gain control law is applied to an arbitrary number of coils. A set of four coils, migrating with the magnet position, is actually involved, being controlled by a non-zero gain factor. In the reference position, as shown in Figure 10, coils No. (i + 1) and (i + 3), having maximum force efficiency, are controlled with maximum gain, while all other coils are idle.

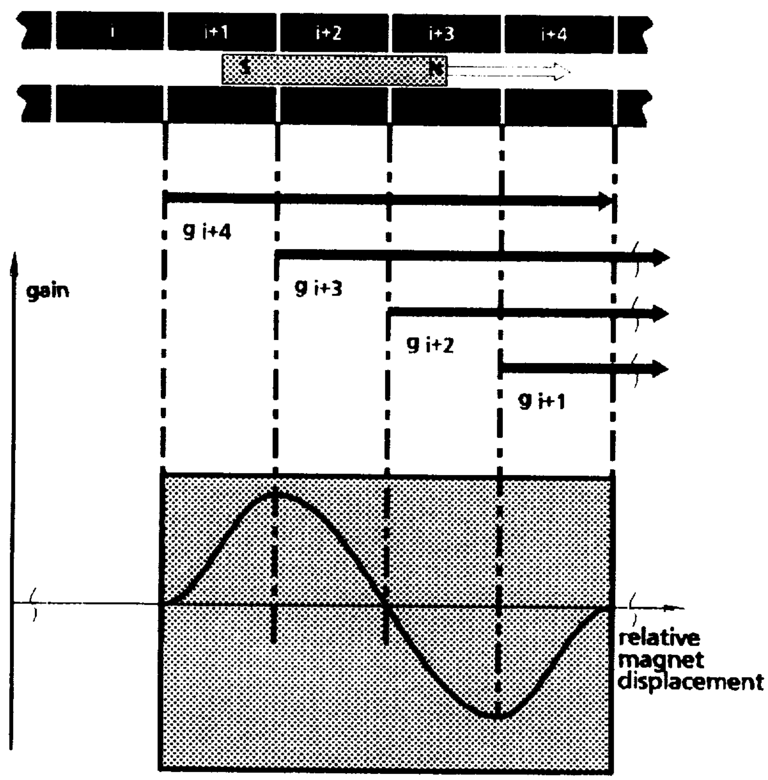

Figure 10. Gain control concept.

Starting out from this initial condition with the magnet moving to the right, the adjacent coils No. (i + 2) and No. (i + 4) are increasingly included into the magnetic driving field generation, while the rating of No. (i + 1) and (i + 3) is decreasing. This follows from the individual gain functions, g_{i+k}, which are sections of the overall gain function, shifted by the relative magnet position. After one coil length displacement, control continues periodically with the new set of coils controlled with maximum gain. The continuous change of gain factors is illustrated by corresponding arrows above the gain function graph.

The primary goal of gain control is an output force which is strictly proportional to the input command and independent from magnet displacement, x. This requires the four partial gain functions to satisfy for all locations, x

$$\sum_{j=i+1}^{i+4} g_j(x)\, F_j(x) = F_o = \text{const.} \tag{4.5}$$

with F_j representing the respective partial force characteristic.

In addition to this basic functional requirement a smooth variation of gain with displacement is desirable to avoid any AC transients from entering control. This applies in particular for the end of the gain function when a coil is dismissed from control. At this location therefore a zero derivative is provided to achieve a "soft commutation" performance.

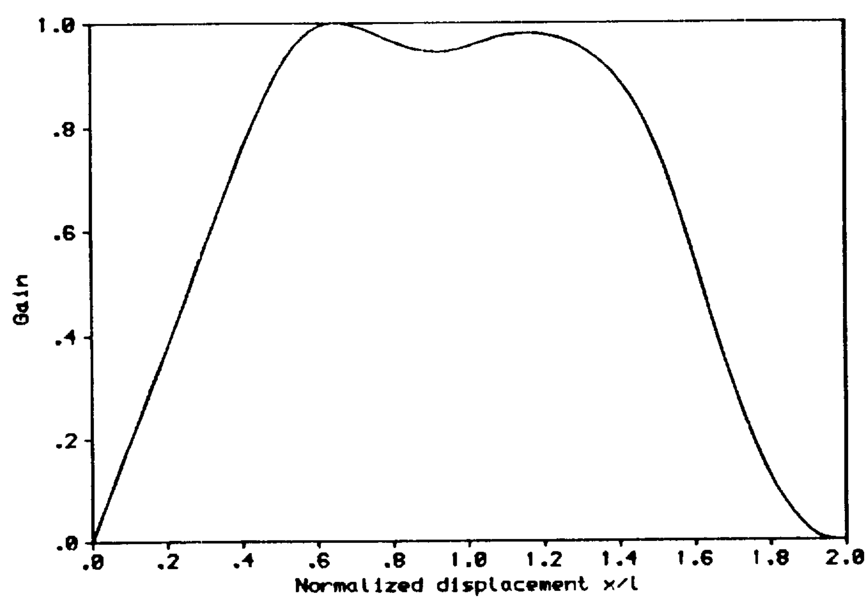

Figure 11. Typical gain characteristic.

All of these conditions are set as constraints for a quadratic optimization. The minimization function has the form

$$J = \int_{-2l}^{2l} \left[k_1 g(x)^2 + k_2 g''(x)^2\right] dx \tag{4.6}$$

with the second derivative of the gain function being included to minimize the average curvature. The weight factors k_1 and k_2 have to be selected by a trade-off to achieve acceptable smoothness on account of the average input power. A typical result is shown in Figure 11. For this design, maxima are found outside the reference position at x/l = 1. Accordingly, the average increase of input power as compared to that required for the reference position, is about 20 percent.

Moving Coil Actuator Concept

For most standard applications this concept would be preferred due to the higher efficiency as mentioned above. However, to enable a wide frequency band application, the coil AC resistance has to be low enough to avoid excessive input control voltages. This can only be realized by coil sectors opposed to multipole magnets with the individual coils not enclosing the center back iron [2].

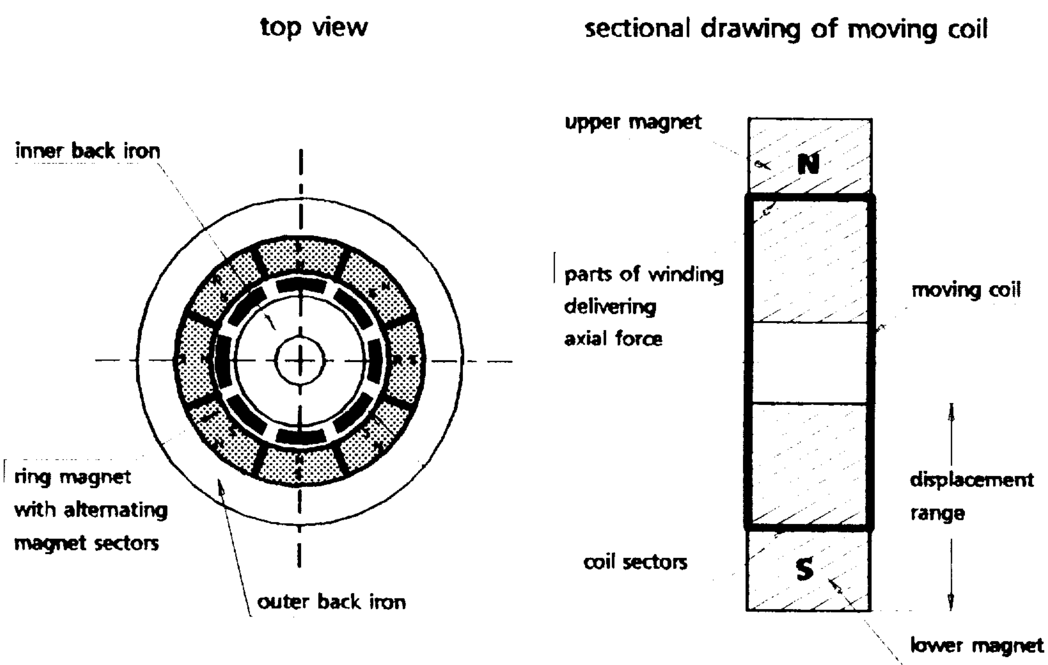

Figure 12. Multipole moving coil actuator.

A design problem arises with the desired range of displacement. As shown schematically in Figure 12, there is only a relatively short part of the winding delivering an axial force component. Moreover, two multipole stator magnet arrangements each extended over the full displacement range plus coil width must be provided ending up in a rather lengthy and complex mechanical design.

For DC applications a less complex design can be utilized employing one set of radial magnets and a simple cylindrical moving coil as shown in Figure 13. Feasibility studies have shown that this approach can be successfully used for the bias load suspension replacing the supporting spring and thereby removing spring restraint forces and mechanical interface design problems. For the specific design envisaged the radial magnets are replaced by a set of stacked rectangular magnets for cost reasons. They are attached to the circumference of the back iron. The moving part has ribs attached to a center shaft which is supported by an air bearing. By linking all external forces to the center shaft bending torques can be minimized.

As a result of electrodynamic model investigations a total moving mass of about 400 g and a required input power of 26 W is necessary to balance a 100 N external load. The main design problem here is the back iron flux saturation. Despite local saturation effects due to the non-uniform flux distribution, a satisfactory design is obtained with a moving coil diameter of about 14 cm and an air gap and magnet depth both measuring 1 cm.

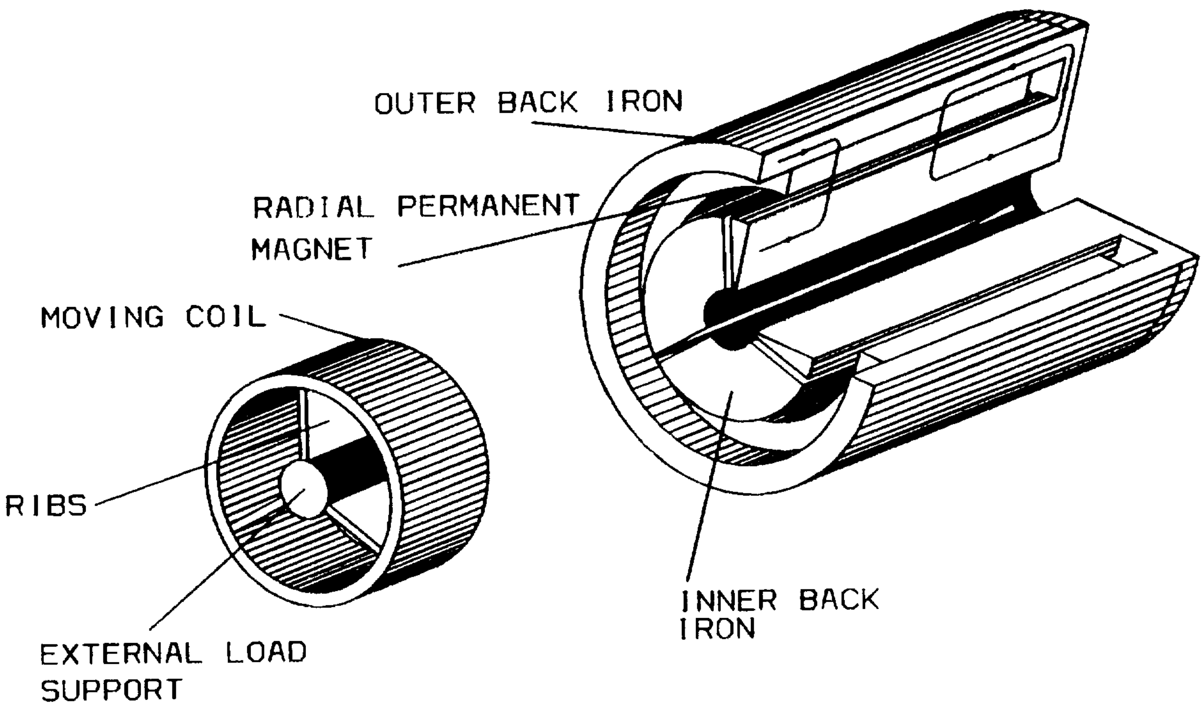

Figure 13. Moving coil actuator with unipolar stator magnet.

The obvious advantage of this approach is the virtual absence of displacement dependent forces with a computed error of a few tenths of a percent. This is achieved at the expense of a large moving mass. Extrapolating the results obtained so far it should be possible to achieve also larger forces. The additional weight penalty in terms of required actuator moving mass is estimated not to exceed about five percent of the supported load. The design limit, which is far beyond the design parameters under consideration here, is given by the increasing DC power with the associated heat transfer problem and by a potential demagnetization of the stator magnets.

DESIGN OF A LABORATORY MODEL

The development of the active suspension test facility is done stepwise starting out with prototype functional laboratory models. The first model which has been realized is a low-cost

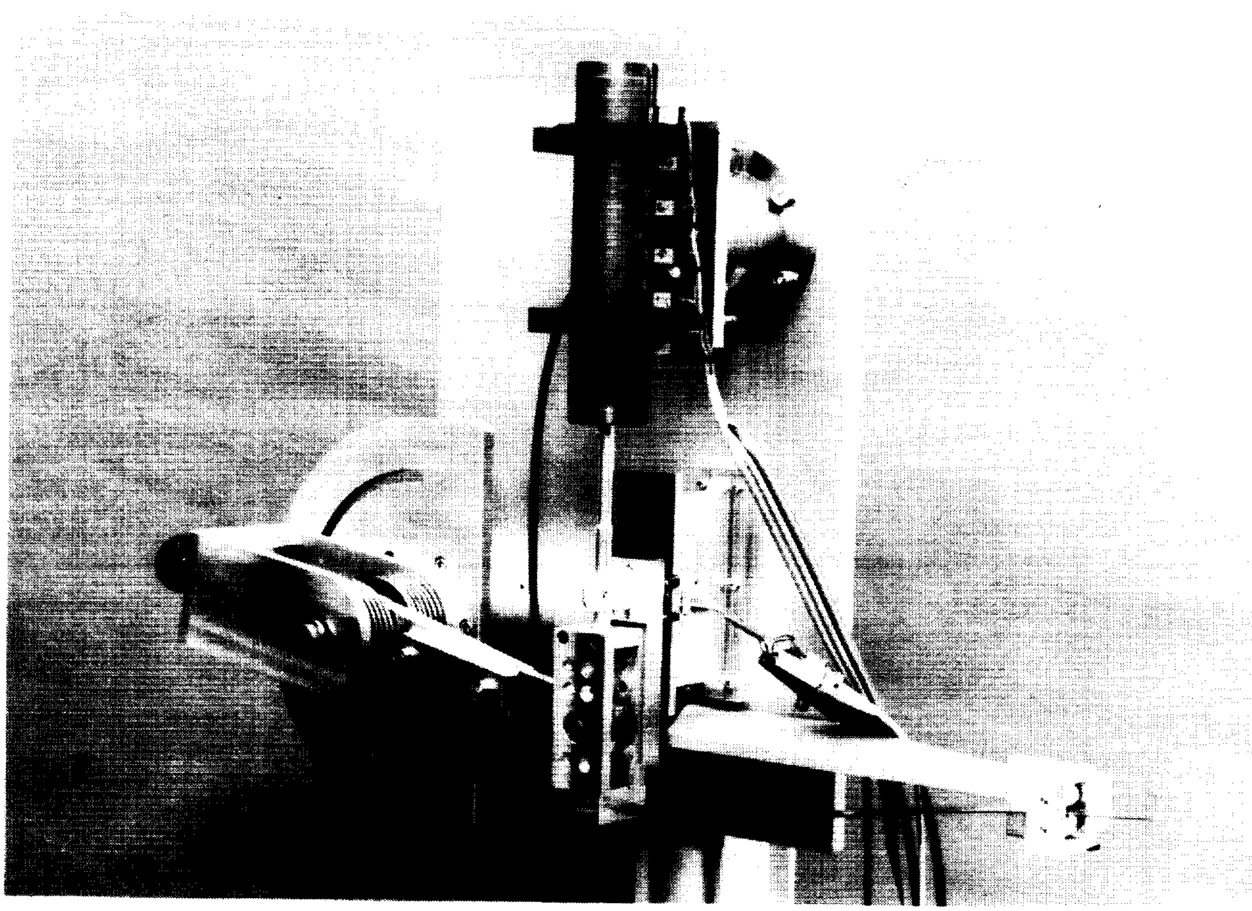

Figure 14. First functional test setup.

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test stand composed of very basic mechanical elements to verify the moving magnet design concept.

Mechanical Test Setup

The functional test setup employs a 6 coil actuator, a torsional spring with a preload to be set manually by a lever and a linear ball bearing sled to guide the moving permanent magnet. The load attached to the sled is an adjustable spring-mass system. As shown in Figure 14, a fairly large force diversion lever arm is provided which is equipped with a ball bearing length compensation mechanism.

The displacement sensor is a simple off-the shelf optical incremental device. With the coil housing mounted to a piezoelectric force transducer, the force exerted on the coil can be measured while the bearing friction forces remain excluded.

Control Processor and Electrical Interface

The control hardware is based upon a high-speed digital signal processor (DSP) with associated fast I/O interface hardware. For system development it is used with a PC/AT host. In the final version it can be operated stand-alone under control of an on-chip EPROM. The main features of a typical control board currently being used is as follows:

- 16 bit fixed point DSP, 160 ns cycle time, 4 k words of memory,
- 6 pulse-width modulated outputs, high resolution & accuracy,
- two 16 bit incremental sensor I/F's with noise filters,
- various high speed ADC's and DAC's,
- 16 bit selectable I/O ports.

Special current control amplifiers have been provided to be controlled by the pulse-width modulators via an analog switch interface. Direct digital current switching has been avoided in favor of a continuous high-impedance control to suppress damping currents from the moving magnet induction voltages.

The control S/W design is supported by a development system permitting direct input of the matrices associated with the standard linear dynamic equations usually being applied for control design. Moreover, the implementation of non-linear functions is supported, realized by table look-up. In addition, on-line tracing of dynamic variables is possible. By this means, S/W development costs can be reduced to a large extent.

Test Results

The first problem to be solved has been the implementation of the gain characteristic. To this end, the force vs. displacement characteristic of each single coil has been determined.

Although no special manufacturing or wiring technique has been used the measurements showed satisfying agreement with the electromagnetic model calculations. The results were obtained by evaluation of the AC force on the stator housing due to a sinusoidal coil stimulation.

After implementation of the gain characteristic the compensated system was stimulated by a constant current input command while the moving magnet was freely oscillating with the spring-mass eigenfrequency. Although the compensation seemed to work properly, no final results are available yet since the current amplifiers started drifting. An appropriate electronics re-design is necessary to remove this malfunction.

The next step before determination of the frequency response characteristic will be the implementation of an air bearing in conjunction with the moving-coil bias load compensation. With the dynamic feedback included this will be the final functional test to validate or modify the concept if necessary before entering the first series production.

CONCLUSIONS

Active suspension tests are valuable for CSI design and validation problems, particularly if optimal dynamic zero-g simulation on ground is required. The technology becomes practically feasible with the advent of modern signal processors which are able to realize complex control tasks in real-time within the frequency range of controlled structures.

Test equipment design problems can be effectively reduced by this technology which decreases also the overall system cost. However, high accuracy results can only be achieved by a precision actuator for dynamic force compensation. The testing requirement of large structural deflection amplitudes for open-loop reference tests leads to the development of a special electrodynamic force actuator. A feasible solution in terms of reasonable mechanical effort is the moving magnet actuator although it is not very effective with respect to the required input power. A linear gain control method, including soft commutation, has been developed and successfully tested using an experimental suspension test setup. A potential spin-off application is the utilization of the force transducer as a manipulator for handling objects under zero-g conditions.

The moving coil actuator in a special configuration has been selected as a candidate to replace the bias weight suspension spring. Feasibility studies have shown that this is possible under the given operational conditions with a moving mass weight corresponding to about 5 percent of the suspended load. Considering the almost zero stiffness force and a further reduction of mechanical calibration effort, this approach has been selected as a promising alternative.

During the subsequent development steps the suspension unit will be optimized on a subsystem level before starting with the initial series production of a few suspension clusters for system level verification.

Due to the approximative gravity compensation in a wide frequency range with the active suspension approach, LSS control verification and system level validation should become increasingly relevant. However, by no means well approved standard test methods will be ruled out, since the achievement of an optimum result will always require a well determined combination of complementary tests.

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