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SEMI-ACTIVE SUSPENSION FOR A SEAT WITH A CONTROLLED ELECTROMAGNETIC DAMPER

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Abstract

In vehicle suspension systems, the amplification of vibrations at resonance is considered the main problem. Secondary problems such as vibrations of wheel masses can be solved by conventional means like the 'three-mass configuration'. For the elimination of the main resonance, electronically controlled dampers must be applied. At the Delft University a new semi-active system with a rotational electromagnetic damper is developed. This system is studied theoretically and experimentally with deterministic and random excitations. The theoretical simulation model includes filtering of measured acceleration for obtaining the absolute velocity signal, the delay in the electromagnetic system and the locking through the frictional force. In a possible application for a sprung seat, the resonance can be eliminated. As result of non-linearities, some anomalies are encountered: subharmonic vibrations, medium-frequency disturbances and chaotic motions.

Keywords: semi-active suspension, random excitation, chaos.

1. Suspension Systems in Vehicles and the Choice of Models

The principal aim of sprung suspensions in vehicles is the isolation of the sprung mass from vibrations caused by passing irregularities of the road. Due to resonance of mass-spring systems, there is always a region where the disturbing vibrations are amplified instead of being attenuated. One example is given in *Fig. 1.* In a cabin of a truck, the spectrum of vertical acceleration measured on the floor is given in solid line. A sprung seat intended to protect the driver against vibrations has a natural frequency of 1.5 Hz. The acceleration spectrum of the mass of the sitting person is given in dashed line. For frequencies below 2 Hz, the sprung seat amplifies the vibrations. The typical situation in trucks is shown: the resonant frequency of the driver's seat is near the peak in the spectrum of input acceleration so that a high resonant peak occurs. The isolating effect at higher frequencies is good, nevertheless the R.M.S acceleration value of the isolated mass is higher than that of the input.

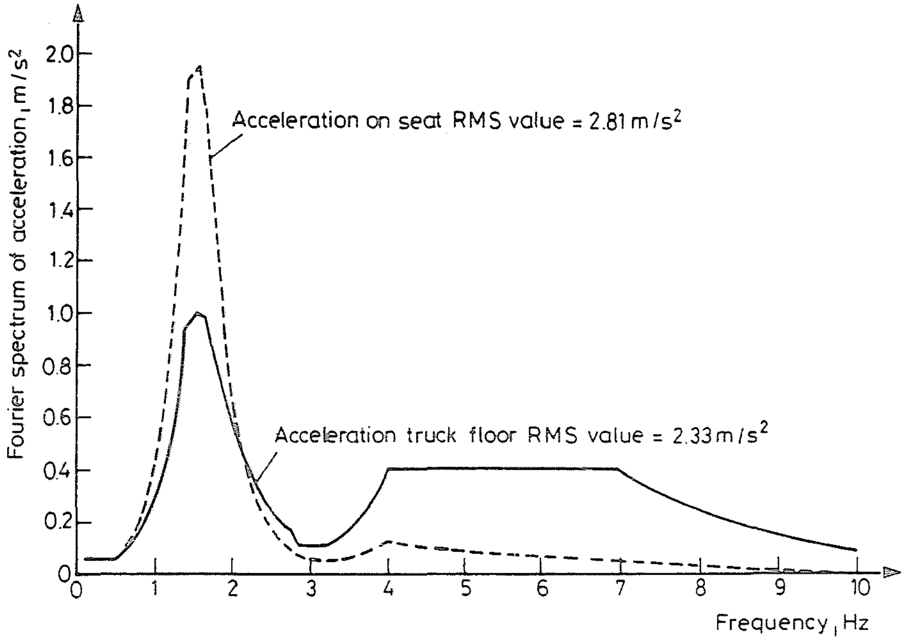

Fig. 1. Vertical acceleration on truck floor and on seat

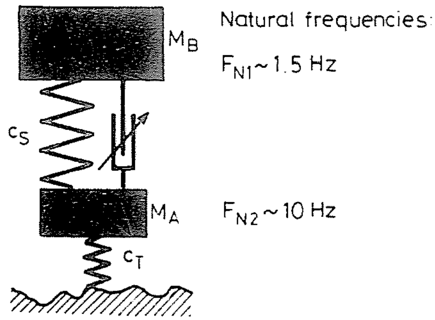

Fig. 2. The 'quarter-car' model

In many theoretical works, the possible improvements of road vehicle suspensions is studied on the two-mass system shown in *Fig. 2.* In this model called usually the 'quarter-car model', two resonant frequencies are present: at 1–2 Hz for the sprung mass and at about 10 Hz for the unsprung mass. By using active or semi-active suspension systems, the resonance at the natural frequency of the sprung mass can be avoided. However, the problem of damping the unsprung mass resonance leads to a compromise

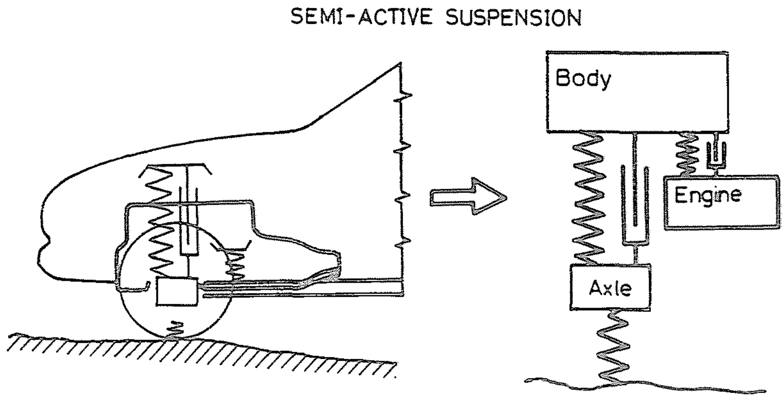

Fig. 3. The three-mass model

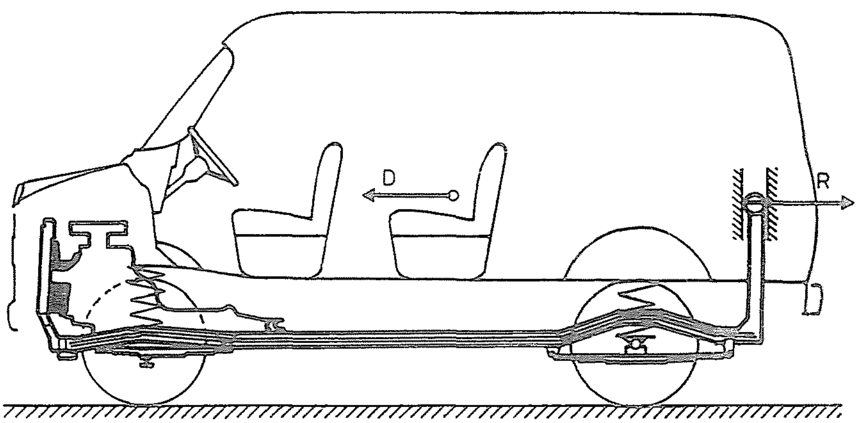

Fig. 4. Experimental vehicle

between ride comfort, suspension clearance and unsprung mass damping. The vibration isolation for the sprung mass is then partly sacrificed for the damping of the wheel vibrations.

In many cases, the two-mass quarter car model is not realistic: especially with the engine mass mounted on the vehicle frame or body, natural frequencies between 2 and 10 Hz are present. The three-mass system shown in *Fig. 3* should be used when modelling the engine end (usually the front end) of a road vehicle.

The three-mass configuration gives the possibility of placing damping elements between the unsprung mass and the engine mass. For the opti-

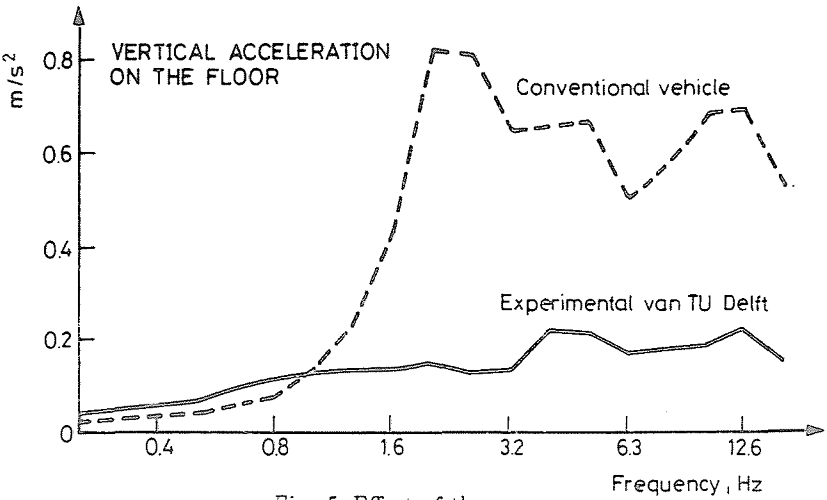

Fig. 5. Effect of the sprung mass

mization of such system, the wheel vibrations of the sprung mass can be damped effectively without affecting the ride comfort in the sprung mass.

The effectiveness of a three-mass configuration was proved theoretically and experimentally at the Delft University in the past. In *Fig. 4*, the experimental vehicle is shown schematically. The suspension of the sprung mass was a very low-frequency (passive) suspension. The effect on the acceleration of the sprung mass is shown in *Fig. 5* — a spectacular improvement has been reached.

In a new, advanced three-mass system, the vibration isolation for the sprung mass could be realized as a semi-active or active suspension. This suspension could be optimized without a compromise regarding the vibrations of the sprung mass.

We can conclude that it can be worth while optimizing active and semi-active suspensions using a one-mass theoretical model. Results can be used for vehicles with a three-mass configuration as well as for single isolated masses in vehicles such as sprung seats for persons or isolators for vibration-sensitive payloads.

2. Systems with an Electromagnetic Force Generator

At the Delft University, the use of an electromagnetic semi-active damper has been studied. The advantage of the semi-active damping against a fully active one is the low energy demand: the only energy needed is for the controller and for the activation of the force element, which works dissipatively. Systems with controlled hydraulic dampers need fast-acting electromagnetic valves, which are expensive and not capable of reducing the damping force to zero when needed: there is always a damper force caused by the oil flow through orifices and it is not easy to provide for a very great flow section in the valve. A force generator with an electromagnet acting on a friction surface gives the possibility of controlling the force by the magnitude of the electric current through the magnetic coil, independently of the relative velocity.

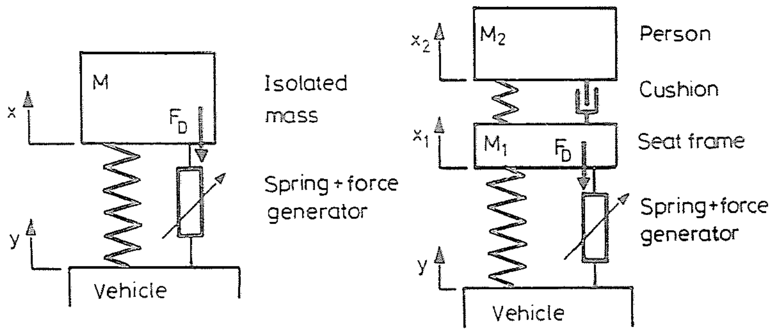

Fig. 6. The studied models

The two models having been studied are shown in *Fig. 6*: a one-mass model for the vibration isolation as a general problem and a two-mass model representing a sprung seat with a sitting person. The control strategy of the force generator is based on the sky-hook principle: the desired force should be proportional to the absolute velocity of the sprung mass. In the case of a fully-active system, this would lead to adding external energy to the moving mass. As the semi-active damper only dissipates energy, the force is set to zero at times when the absolute velocity and the relative velocity have opposite signs. The control law can be written as

$$\text{IF} \quad \dot{x} \cdot (\dot{x} - \dot{y}) > 0 \quad \text{THEN} \quad F_D = k\dot{x} \quad \text{ELSE} \quad F_D = 0 \,. \tag{1}$$

The law is written for the one-mass system; in the two-mass system, the velocity of the lower mass $\dot{x}_1$ is used instead of velocity $\dot{x}$.

The force generator could in principle be designed as a linear friction element using a prismatic part with linear friction surface. For practical reasons, a rotational force generator has been chosen. The linear motion of the sprung mass should be transformed into rotation by a suitable mechanism: a rotating arm, a friction drive, a rack and pinion, a flexible band or a screw could be used.

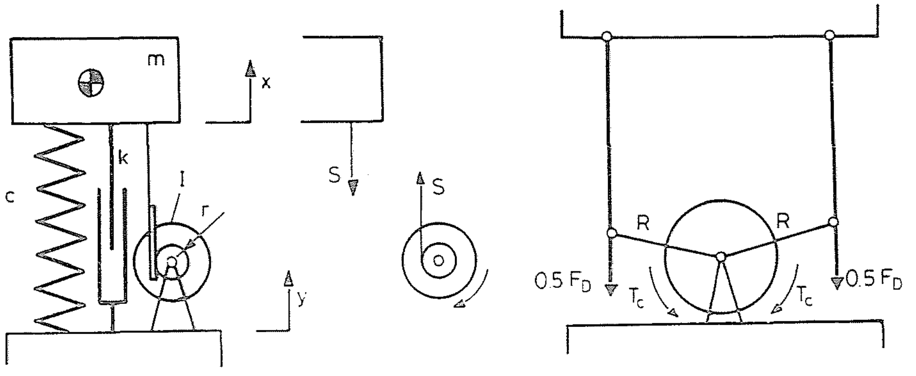

Fig. 7. The rotational force generator

In the left part of *Fig. 7*, the principle of work of the rotational force generator is shown. The moment of inertia I of the rotating mass can influence the dynamic behaviour of the isolated mass. The displacement of mass $(x - y)$ is coupled to the angle of rotation of the rotor. The high-frequency transmissibility can be negatively influenced. It is necessary that the quantity

$$m_{red} = \frac{I}{r^2} \tag{2}$$

be as low as possible. The effect of the rotating mass is calculated in [1].

The rotational force generator chosen for experimental systems is a commercially available electromagnetic clutch. The suitable size was chosen for the maximum desired force necessary for the application under a sprung seat used in a heavy truck on rough roads. The mechanical connection with the sprung mass is designed as a pair of rotating arms. One arm is connected with the body of the clutch containing the electromagnet, the other arm rotates in the direction opposite to the armature. On the right side of *Fig. 7*, the function of the force generator is shown. As both parts of the electromagnetic clutch rotate in opposite direction, reaction forces act on both ends of the swing arms. For a given torque of the clutch, this configuration reduces the angle of rotation of the arms.

The swinging arm linkage was preferred to other types of mechanisms because of its simplicity and reliability.

The dynamic behaviour of the force generator is described in *Fig. 8*. The left side of the figure shows the electric circuit including the controller, the power amplifier and the coil of the electromagnet. With inductance L, the internal resistance of source R_A and the resistance of conductors R_c,

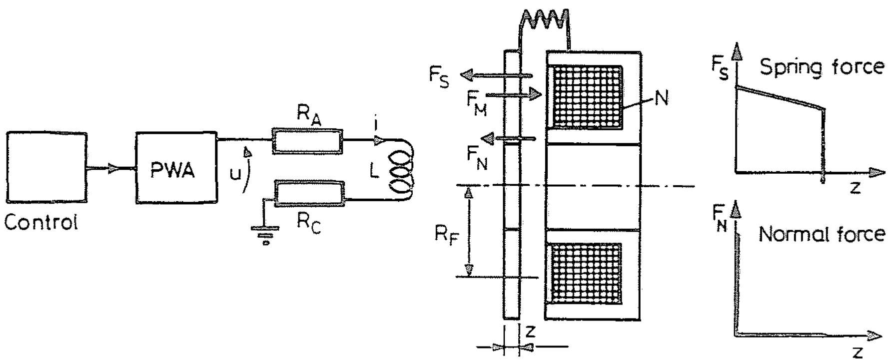

Fig. 8. The force generator

the law for the circuit is:

$$u = (R_A + R_c)i + L\frac{\mathrm{d}i}{\mathrm{d}t}. \tag{3}$$

This differential equation describes the electric current i depending on the applied voltage u. The inductance will cause a delay of the current with respect to the voltage.

In the middle part of *Fig. 8*, the electromagnet is shown schematically. The magnetic force is given as

$$F_M = \frac{AN^2 \left[\dfrac{i}{\left(\frac{z}{\mu_0} + K_s\right)}\right]^2}{2\mu_0} \tag{4}$$

where A is the cross-section of the magnetic circuit, N — the number of windings, μ_0 — the magnetic permeability, z — the variable magnetic gap and K_S — a constant of the magnetic circuit. With varying gap z, the inductance is

$$L = \frac{AN^2}{\frac{z}{\mu_0} + K_S}. \tag{5}$$

The equation of motion of the armature with mass M_A is

$$M_A\ddot{z} = -F_M + F_S + F_N. \tag{6}$$

The damper force developed by the force generator on the sprung mass is

$$F_D = \frac{F_N \mu_F r_F}{r}, \tag{7}$$

where μ_F is the friction coefficient and r — the length of the swing arm.

The spring force F_S and the normal reaction F_N are given as functions of z graphically on the right side of *Fig. 8*.

The control equation (1) together with the equations of the force generator (3)–(7) represent the mathematical model of the real controller. This set of non-linear equations together with the equations of motion of the moving masses form the model of the semi-active suspension system.

3. The Mathematical Modelling of the Suspension System

For the two models given in *Fig. 6*, the equations of motion can be formulated by using the Newton's law. The equations for the two-mass system are given in [1].

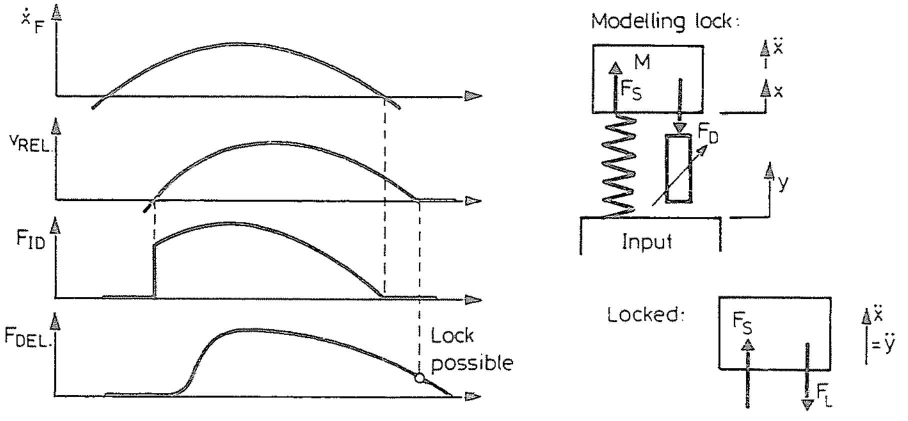

Fig. 9. The locking of the clutch

In both systems, a special phenomenon may occur. The normal reaction force in the friction clutch can be somewhat delayed with respect to the ideal control law (1).This sometimes causes locking of the clutch. The possibility of locking can be seen from the example of time histories of velocities and forces shown in *Fig. 9*. The signal of the absolute velocity $\dot{x}_F$ is plotted on the top. This signal does not describe exactly the absolute velocity, there is a time lag due to filtering (pseudo-integrating) the measured acceleration of the mass. The filtering is described in [1]. Signal $\dot{x}_F$ is one of the two input signals of the controller acting according to *Eq.* (1). The second input of the controller is the relative velocity v_{rel} plotted in the second graph. The third graph shows the ideal damper force F_{ID} at the output of the controller. Due to delay in the force generator caused by the electric circuit and by the inertia of the armature, the real force F_D is delayed. There is a possibility of F_D having non-zero value at the moment when the absolute velocity has already passed the zero and the relative velocity approaches the zero value. Then the relative motion can be locked for a certain period of time.

For the formulation of the conditions of locking, we use the right-hand side of *Fig. 9*.

We introduce the Boolean variables:
LFL — lock flag
SZF — speed zero flag
NSC — no slide condition.

For the acceleration of mass M, we can write

$$\text{IF LFL} = 1 \text{ THEN } \ddot{x} = \ddot{y} \text{ ELSE } \ddot{x} = (F_S - F_D)/M, \tag{8}$$

$$\text{LFL} = \text{SZF AND NSC}, \tag{9}$$

$$\text{IF } |\dot{x} - \dot{y}| < e \text{ THEN SZF} = 1 \text{ ELSE SZF} = 0. \tag{10}$$

In locked state $\ddot{x} = \ddot{y}$ so that $M\ddot{y} = F_S - F_L$. (11)

Locking force is $F_L = F_S - M\ddot{y}$ (12)

$$\text{IF } |F_D| > |F_D| \text{ THEN NSC} = 1 \text{ ELSE NSC} = 0. \tag{13}$$

Mass M represents the single mass in the one-mass model and the lower mass m in the two-mass model.

The possibility of locking the relative motion is a non-linear phenomenon which causes changes in the number of degrees of freedom. In locked state the one-mass system has zero degree of freedom, the two-mass system has only one degree. The equations of motion together with the control law (1) and locking model (8)–(13) describe the dynamic motion of the suspension system.

The non-linear system of equations can be integrated numerically in the time domain. The block-based simulation program TUTSIM has been used for this purpose. *Eqs.* (8)–(13) were represented by logical blocks.

Various types of input signals have been used. This included two types of deterministic signals: a harmonic function with various frequencies and a transient input simulating a truck riding over a single obstacle — a groove in the road. Another class of input is represented by random signals. These were obtained as the sum of a great number of harmonic functions with random phase angles, the amplitudes of components being chosen to model the measured spectrum of acceleration in a road truck.

4. Results of Simulations

The results of simulations with harmonic excitation are given in [1]. A special phenomenon has been found: subharmonic vibrations with frequency many times lower than the excitation frequency sometimes occur. The resonant amplification at the natural frequency of the mass can be avoided. Because of the non-linearities of the system, the resonant behaviour has to be studied with inputs representing excitation in vehicles.

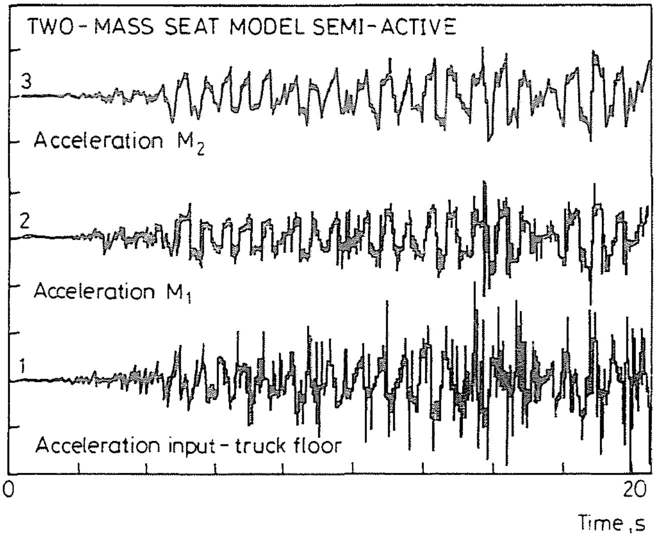

Fig. 10. System response to random excitation

The response of a two-mass system to random excitations is shown in *Fig. 10.* In the lower part of the graph, the input function — acceleration of a truck floor — is shown as a function of time. The second curve is for the acceleration of the lower mass, at the top is the acceleration of the second mass — in this case the sitting person. The reduction of accelerations can be easily seen. Higher frequency vibrations which are present in the second curve, are strongly reduced in the upper curve.

The semi-active system presented in *Fig. 10* can be compared with a passive system in *Fig. 11.* The natural frequency of 1.5 Hz can be seen as the main component of the acceleration of the upper mass — an amplification of this component is present.

Fig. 12 shows spectra of accelerations for random excitation representing the acceleration of the truck floor. The input signal spectrum is plotted in a solid line. This spectrum has a maximum at 1.5 Hz and an

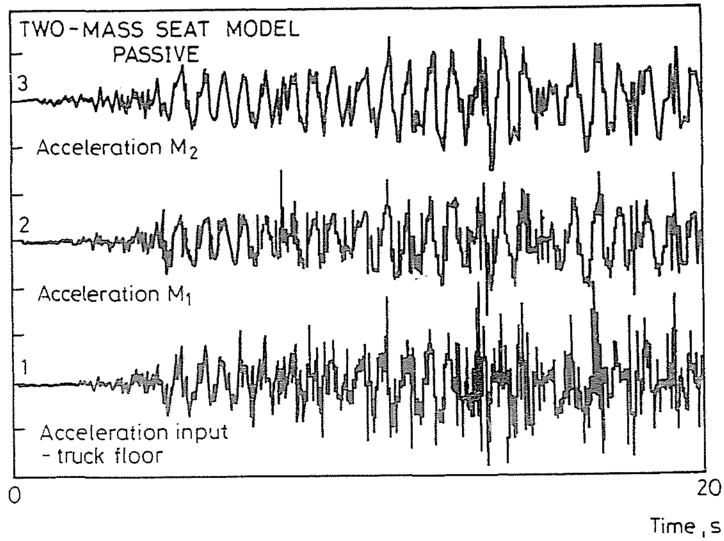

Fig. 11. Response of a passive system

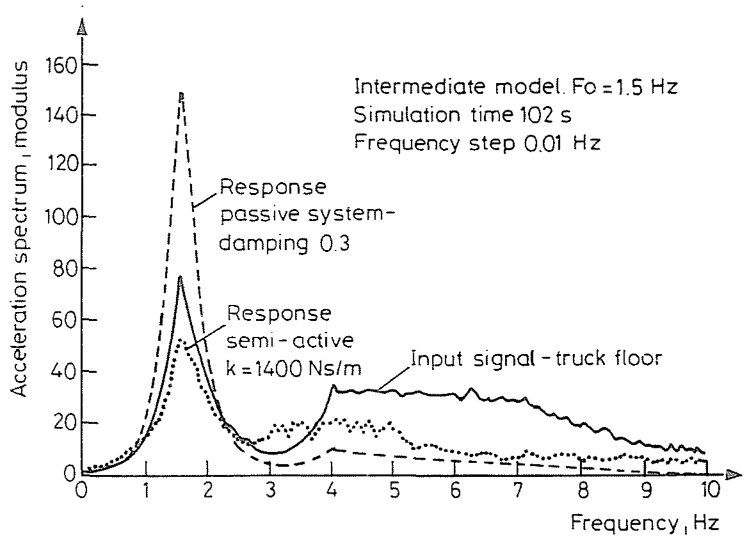

Fig. 12. Transmission of vibrations through passive and semi-active systems

almost constant value between 4 and 7 Hz. The spectrum of the response of a passive system is shown by a dashed line. The peak at 1.5 Hz has very high values due to the amplification at resonance which by coincidence lies practically in the same frequency domain as the maximum of the input spectrum. The dotted line represents the response spectrum with a semi-active system. There is no resonance peak at 1.5 Hz, nevertheless at

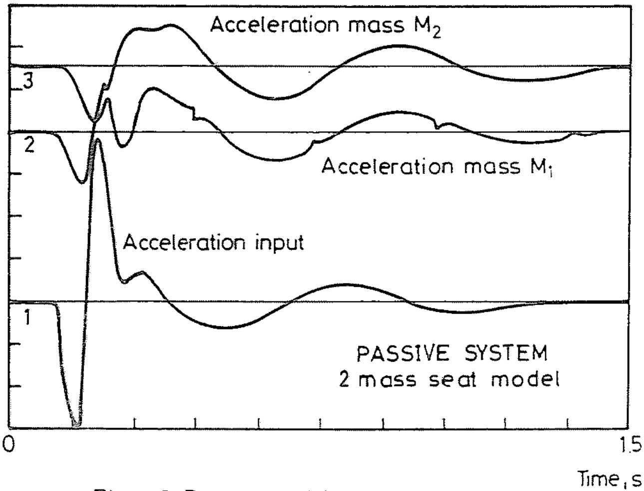

Fig. 13. Response with a semi-active damper

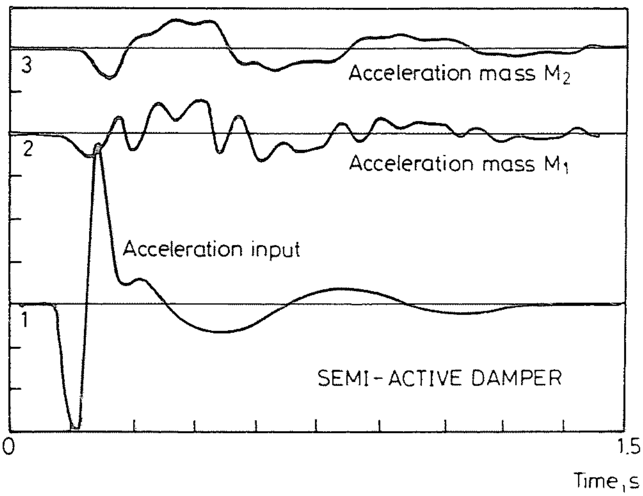

Fig. 14. Response of a passive system

higher frequencies the values lie higher than those of the passive system. Moreover, the response of the semi-active system is higher than the input at 2.5–3.5 Hz.

Other results for random excitation can be found in [1]. The amplification at natural frequency of the sprung mass can be avoided by a suitable choice of parameters. At higher frequencies, the acceleration components can sometimes be higher than those with a passive system. In general,

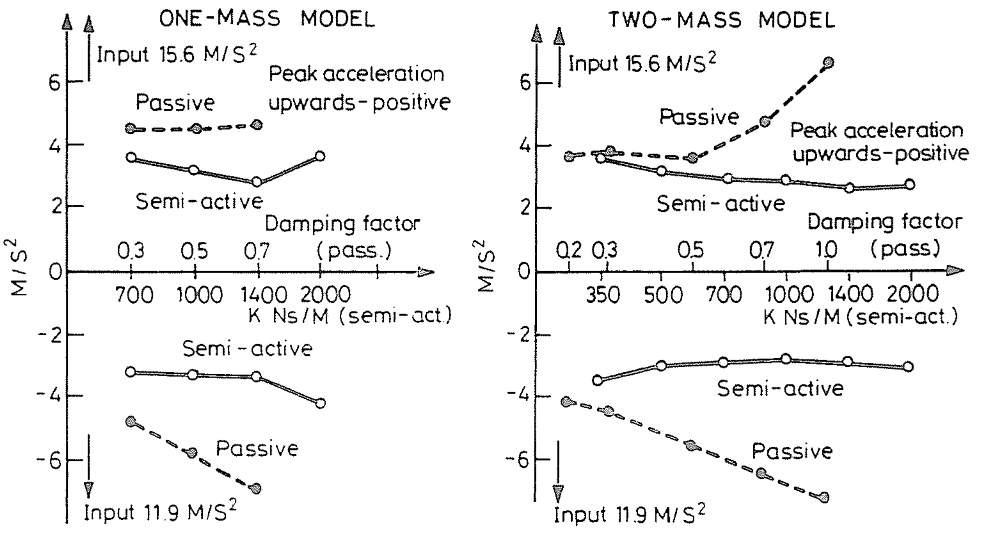

Fig. 15. Extreme values in the two systems

the semi-active suspension improves the vibration isolation considerably as compared with passive suspension systems.

Another realistic input is the acceleration of the floor of a vehicle passing a single obstacle. Simulations with this input have been done and results have been compared with a passive system. The input function was the vertical acceleration of a floor of a truck passing a groove. In *Fig. 13*, the response of a two-mass system with a semi-active damper is shown. The lowest curve is for the input acceleration, in the middle is the acceleration of the lower mass m_1, the upper curve is for the upper mass m_2.

These results can be compared with a passively damped system. *Fig. 14* shows accelerations in a passive system which are higher than those with the semi-active damping.

The quality of the vibration isolation can be judged by comparing maxima and minima of accelerations on passing the obstacle. In *Fig. 15*, the extreme values both in positive and negative directions are given. The semi-active damping is compared with a passively damped system. The horizontal axis is for the damping parameters of both types of damping. It can be clearly seen that the semi-active system gives better results. The results for the passive system approach the values of the semi-active system only for low damping values. In that case, however, the free vibrations of the passive system will continue for a considerable time whereas the vibration of a semi-active system will damp out quite quickly.

5. Results of Experimental Work

The semi-active damping was tested in a laboratory using hydraulic actuators for the generation of desired vertical input motion. Both one-mass and two-mass systems were tested. The two-mass system was a sprung seat for a truck where the normally mounted hydraulic damper was replaced by the electromagnetic damper.

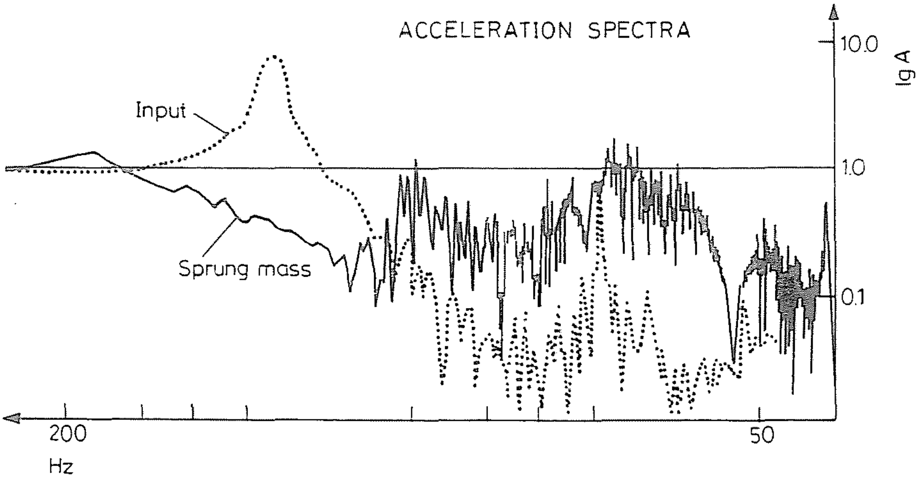

Fig. 16. One-mass system spectra

Some results of the experiments were already given in [1]. In *Fig. 16*, the results of the randomly excited one-mass system are shown. The spectra of accelerations for the input as well as for the sprung mass are plotted as functions of frequency.

The input was a simulated acceleration signal of a truck floor travelling along a cobblestone road. The spectrum of the input signal has a maximum at approximately 1.4 Hz. The semi-active system works very satisfactorily at frequencies lower than 2 Hz so that the peak at 1.4 Hz in the input has no effect on the sprung mass. There is a weak maximum at 0.4 Hz, which will not present a real problem. At frequencies higher than 3 Hz, however, the components of the output are quite high. This phenomenon is caused by the switching action of the force generator: the damper force often jumps from zero to a considerable value at the moment of polarity change in the velocity signal. With the two-mass system of the sprung seat, the effect of switching is less serious: the damper force acts on the frame of the seat and the higher-frequency components are filtered out to some extent by the mass of this frame. A possible measure to reduce the higher frequency vibrations would be a modification of the control strategy by putting a limit on the speed of change in the damping force.

6. Chaotic Motion with a Non-linear Spring

During experimental work in the laboratory, a special behaviour was observed on the one-mass test system. In certain circumstances, periodic vibrations with high amplitude originated suddenly and continued for some time in an almost stationary way. These vibrations were observed with a random input signal with low settings of the damping constant of the semi-active damping system.

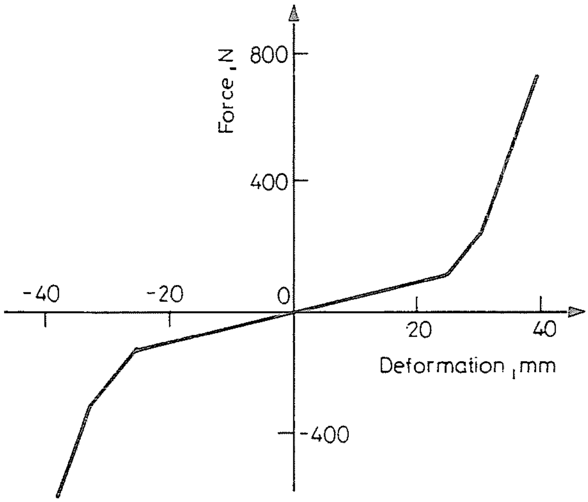

Fig. 17. Spring characteristic

The special feature of the experimental model is the presence of rubber springs acting like stops limiting the length of stroke of the suspension.

The motion was periodical with a frequency 2 Hz, which is higher than the natural frequency of the mass on the spring. The rise in frequency can be explained by the influence of progressive stops: at higher amplitudes when the stops act periodically, the effective spring stiffness increases.

In order to study this vibration more deeply, the real characteristic of the spring was included in the simulation model. The characteristic of the spring with stops measured on the test model is shown in *Fig. 17.*

The simulations were done with two random input signals: the first signal had an acceleration spectrum of the floor of a truck cabin with a maximum at 1.5 Hz, the second signal was with a uniform acceleration spectrum between 0.5 Hz and 7 Hz.

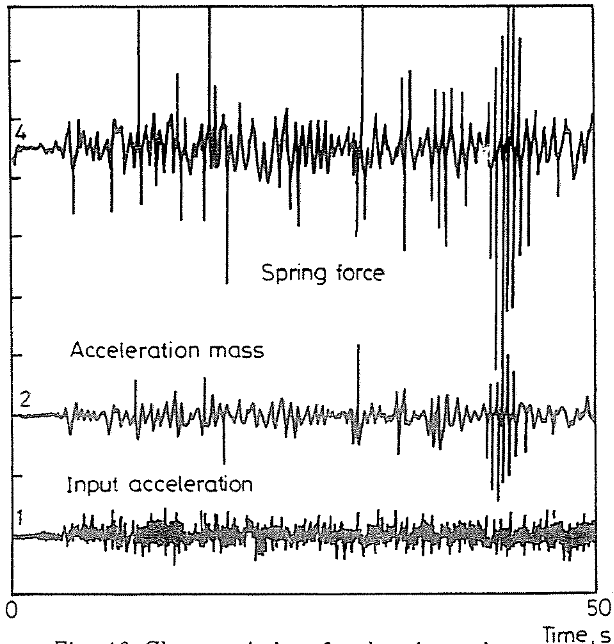

Fig. 18. Characteristics of a chaotic motion

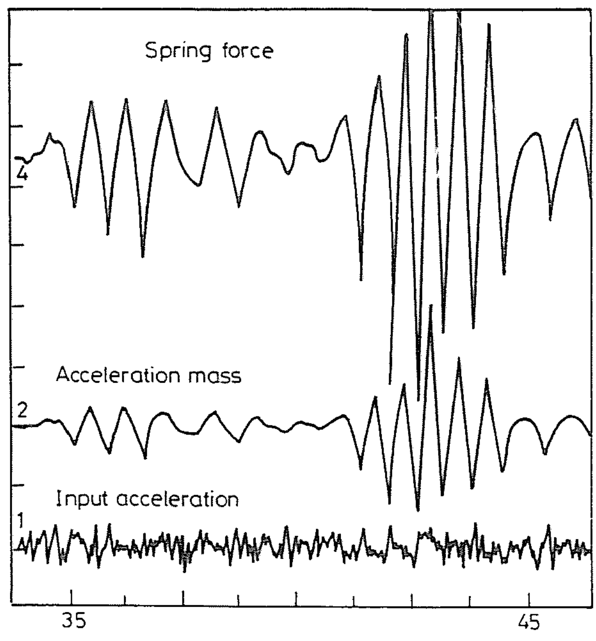

Fig. 19. Chaotic signals

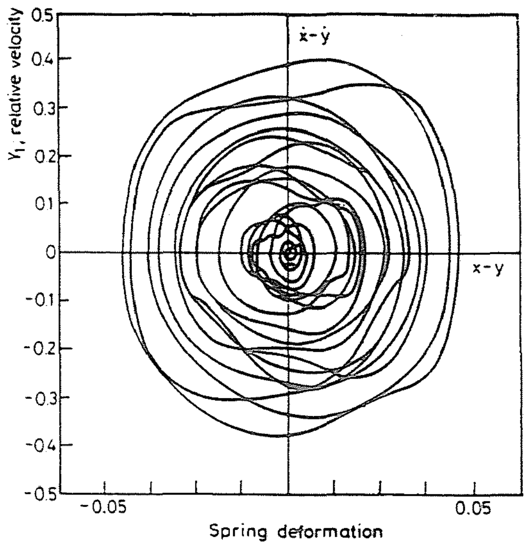

Fig. 20. The phase-plane diagram of the motion

During the simulations, chaotic motion was observed when the damping constant was chosen low, about 20% of its optimal value. The motion occurred with each of the input signals.

One example of the chaotic motion is shown in *Fig. 18*. The plot is for a time period of 50 sec. In the graph, the input acceleration is given by the bottom curve, the acceleration of the mass is shown by the middle curve and the top curve is for the spring force including the action of the stops.

Vibrations with high amplitudes occurred in this case mostly at 40–45 sec.

Fig. 19 shows the region 35–45 sec of the same signal in more detail. From the peaks in the curves for acceleration and for force we can see that the motion reaches the stops at both sides. The frequency of the vibration is 2 Hz.

In *Fig. 20*, the phase-plane diagram of a the motion of the sprung mass is given. The relative displacement $x - y$ is plotted along the horizontal axis, the vertical axis is for the relative velocity. From the shape of the curves, which differs from a circular shape, we can see the limiting action of the progressive spring characteristic due to stops.

7. Conclusions

From the study of semi-active suspension systems with electromagnetic force generator, it seems that this type of suspension can be a useful solution to vibration isolation in vehicles. The main advantage of semi-active damping systems is the capability to eliminate the amplification of input vibrations caused by resonance.

Low energy consumption is a positive feature of these systems in comparison with fully active systems.

The electromagnetic force generator can be of use in applications similar to the studied case, a sprung seat for persons in road- and terrain vehicles. The suspended mass is rather low when compared with sprung masses of passenger cars or commercial vehicles. The possibility of application of this type of force generator to heavier sprung masses would have to be studied separately. The electromagnetic damper can be an economic solution when compared with hydraulic systems with expensive fast control valves.

The function of the electromagnetic damper during tests was satisfying. However, the duration of tests was not long enough for making conclusions about the reliability and possible wear of this device.

The medium-frequency vibrations in this semi-active system were often higher than expected for an electronically controlled damping system. This phenomenon is caused by the fast switching of the damper force and is most striking in the system with one mass. Some improvement in this respect should be possible by the modification of control strategy.

With low values of the damping constant, chaotic motion can occur. This type of high-amplitude vibration will not endanger the practical use of the isolation system studied here.

Practical applications of the system to sprung seats of vehicles seem to be possible. However with respect to the obtained transmissibility characteristics, it is probable that fully active systems will provide - obviously at a higher cost — a better level of vibration isolation than the semi-active ones.

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