THE EFFECT OF A TRAILER ON CAR VIBRATIONS

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More and more passenger cars running in road traffic haul house trailers or freight trailers. House trailers hauled by passenger cars offer a cheap and comfortable accommodation for those touring by car, while the use of freight trailers suits the transport of lightweight goods.

The production of trailers is technically relatively simple, therefore also in countries producing no passenger cars such as Hungary various types of trailers are built in lesser or greater series.

In the design trailers are generally considered as independent units and one performs the necessary strength, shock and stability calculations on them as on special vehicles. However, the trailer and the prime mover constitute together a single vehicle set which presents various different requirements in the trailer design. In the technical literature few studies can be found on the joint dynamic conditions of passenger cars and trailers. In this domain professor Gnadler and his colleagues performed extensive research concerning the effect of the trailer on the braking and stability of the passenger car [1], [2]. No works on the special subject of analysing the trailer's effect on vibrations can be found in the available literature.

The theoretical examination of the dynamics of this problem is interesting in itself but at the same time it may be of direct practical use in selecting or designing the spring characteristics of the trailer (type of tyres, spring stiffness, characteristics of dampers, etc.).

From the aspect of simulating the vibrations a passenger car hauling a trailer is equivalent to a three-axle articulated bus or to a semitrailer, therefore the theoretical and practical conclusions drawn here may be of use in the examination of these vehical types too, and vice versa: test methods of articulated vehicles and semitrailers can be utilized in analysing passenger cars coupled to trailers. Internationally recognized fundamental results have been achieved on the vertical vibrations of articulated buses and semitrailers by Prof. L. Ilosvai [3]. Stability and braking of semitrailers have been analysed by E. Jánosdeák [4], M. Mitschke [5] and A. Slibar [7].

Initial conditions and method of analysis

The vibrational analysis was made on a vehicle combination constituted by a passenger car of type Lada VAZ 2101 and by a Bastei type van. The vibrational model of the passenger car and the van are seen in Fig. 1, while their parameters are contained in the appendix and in Table I. The parameters of the passenger car were not altered in the analysis, while in the case of the trailer three other variations were examined besides the rated values of the factory.

In this way vibrations of altogether four models were simulated. Model **A** is based on the rated parameters of the single motor vehicle, model **B** on those of the single motor vehicle and the van, model **C** on those of the single motor vehicle and a van having only a dry friction damper, model **D** on the single motor vehicle and on the van without vibration damper.

In building up the models the following assumptions were used:

— the vehicle combination was supposed to be a system of five masses with six degrees of freedom i.e. we employed a plane model of rigid car body. In the technical literature plane models are generally accepted for analysing vertical vibrations. The investigations of Prof. L. ILOSVAI [3] proved that

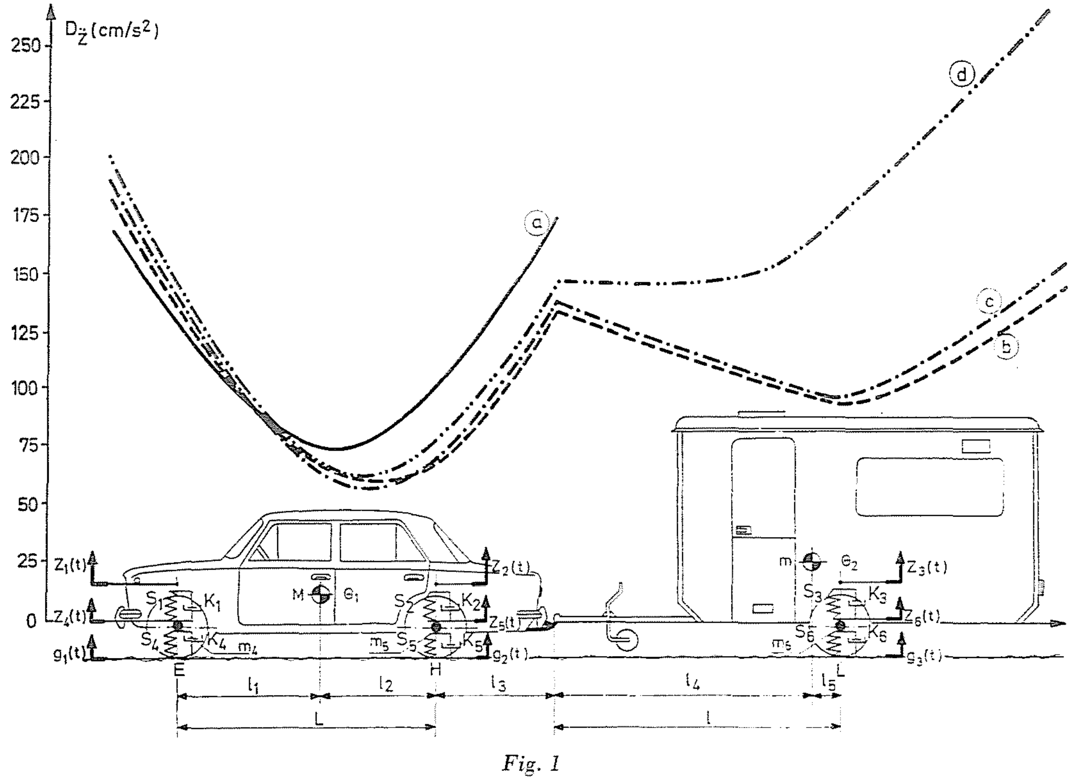

Fig. 1

plane models are suitable for the exact description of road vehicle vibrations due to moderate excitation;

— the damping caused by the internal friction of the tyres was taken into account;

— the non-linearities due to wheels rebouncing from the road surface (causing a change in the spring and damping characteristics of the tyres) were taken into consideration;

— the non-linearities arising from the butting on the rubber brackets built into the wheel suspension were also considered;

— the vibration dampers were replaced by their characteristics measured on the wheel, that is nonlinear and asymmetrical characteristic curves were employed in the model;

— the Coulomb (dry) friction between the elements of the wheel suspension has been taken into account.

The models built with all these starting conditions were tested by stochastic road excitations simulated by a digital computer based on a published road section spectral density function [6]. The model was driven on an asphalt road of medium quality at a speed of 50 km/hour, the road section had a standard deviation of 1 cm.

The dynamical behaviour of each model is described fairly correctly by a system of second-order nonlinear time-dependent differential equations.

The system of differential equations of the vibrational model of the single motor vehicle is the following:

$$\begin{bmatrix} M_1 & M_{12} & 0 & 0 \\ M_{21} & M_2 & 0 & 0 \\ 0 & 0 & m_4 & 0 \\ 0 & 0 & 0 & m_5 \end{bmatrix} \begin{bmatrix} \ddot{Z}_1 \\ \ddot{Z}_2 \\ \ddot{Z}_4 \\ \ddot{Z}_5 \end{bmatrix} + \begin{bmatrix} K_1((\dot{Z}_1 - \dot{Z}_4)) \\ K_2((\dot{Z}_2 - \dot{Z}_5)) \\ K_4((\dot{Z}_4 - \dot{g}_1)) - K_1((\dot{Z}_1 - \dot{Z}_4)) \\ K_5((\dot{Z}_5 - \dot{g}_2)) - K_2((\dot{Z}_2 - \dot{Z}_5)) \end{bmatrix} +$$

$$+ \begin{bmatrix} S_1((Z_1 - Z_4)) \\ S_2((Z_2 - Z_5)) \\ S_4((Z_4 - g_1)) - S_1((Z_1 - Z_4)) \\ S_5((Z_5 - g_2)) - S_2((Z_2 - Z_5)) \end{bmatrix} = 0 \tag{1}$$

where:

$$\vartheta^2 = \frac{\Theta_1}{M}; \qquad M_1 = M\frac{l_2^2 + \vartheta^2}{L^2}; \qquad M_2 = M\frac{l_1^2 + \vartheta^2}{L^2};$$

$$M_{12} = M_{21} = M \cdot \frac{l_1 l_2 - \vartheta^2}{L^2}$$

$K_i((\dot{Z}_{rel}))$ — nonlinear damper characteristics
$(i = 1, 2, 4, 5)$ (Table I, $\dot{Z}_{rel} \equiv \Delta\dot{Z}$);

$S_i((Z_{rel}))$ — nonlinear spring characteristics
$(i = 1, 2, 4, 5)$ (Table I, $Z_{rel} \equiv \Delta Z$).

It is well known from the technical literature that in case of $l_1 l_2 = \vartheta^2$ the vibrating system decomposes into two independent subsystems i.e. the front and the rear parts of the vehicle perform independent vibrations.

The vibrations of the model with six degrees of freedom of the passenger car coupled to a trailer are described by the following system of differential equations:

$$\begin{bmatrix} M_1 & M_{12} & M_{13} & 0 & 0 & 0 \\ M_{21} & M_2 & M_{23} & 0 & 0 & 0 \\ M_{31} & M_{32} & M_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & m_4 & 0 & 0 \\ 0 & 0 & 0 & 0 & m_5 & 0 \\ 0 & 0 & 0 & 0 & 0 & m_6 \end{bmatrix} \begin{bmatrix} \ddot{Z}_1 \\ \ddot{Z}_2 \\ \ddot{Z}_3 \\ \ddot{Z}_4 \\ \ddot{Z}_5 \\ \ddot{Z}_6 \end{bmatrix} + \begin{bmatrix} K_1((\dot{Z}_1 - \dot{Z}_4)) \\ K_2((\dot{Z}_2 - \dot{Z}_5)) \\ K_3((\dot{Z}_3 - \dot{Z}_6)) \\ K_4((\dot{Z}_4 - \dot{g}_1)) - K_1((\dot{Z}_1 - \dot{Z}_4)) \\ K_5((\dot{Z}_5 - \dot{g}_2)) - K_2((\dot{Z}_2 - \dot{Z}_5)) \\ K_6((\dot{Z}_6 - \dot{g}_3)) - K_3((\dot{Z}_3 - \dot{Z}_6)) \end{bmatrix} +$$

$$+ \begin{bmatrix} S_1((Z_1 - Z_4)) \\ S_2((Z_2 - Z_5)) \\ S_3((Z_3 - Z_6)) \\ S_4((Z_4 - g_1)) - S_1((Z_1 - Z_4)) \\ S_5((Z_5 - g_2)) - S_2((Z_2 - Z_5)) \\ S_6((Z_6 - g_3)) - S_3((Z_3 - Z_6)) \end{bmatrix} = 0 \tag{2}$$

where:

$$M_1 = \frac{1}{L^2}(M \cdot l_2^2 + \Theta_1) + \frac{a^2}{l^2}(m \cdot l_5^2 + \Theta_2); \qquad a = \frac{l_3}{L};$$

$$M_{12} = M_{21} = \frac{1}{L^2}(Ml_1 l_2 - \Theta_1) - \frac{a(1+a)}{l^2}(ml_5^2 + \Theta_2);$$

$$M_{13} = M_{31} = \frac{a}{l^2}(\Theta_2 - ml_4 l_5);$$

$$M_2 = \frac{1}{L^2}(Ml_1^2 + \Theta_1) + \frac{(1+a)^2}{l^2}(ml_5^2 + \Theta_2);$$

$$M_{23} = M_{32} = \frac{1+a}{l^2}(ml_4 l_5 - \Theta_2);$$

$$M_3 = \frac{1}{l^2}(ml_4^2 + \Theta_2);$$

$K_i((\dot{Z}_{\mathrm{rel}}))$ — nonlinear damper characteristics
$(i = 1, 2, \ldots, 6)$ (Table I);

$S_i((Z_{\mathrm{rel}}))$ — nonlinear spring characteristics
$(i = 1, 2, \ldots, 6)$ (Table I).

Table I

Spring characteristics of suspension

A B C D		A B C D		B C D
ΔZ [cm]	$S_1((\Delta Z))$ [daN]	ΔZ [cm]	$S_2((\Delta Z))$ [daN]	
−28	−2000	−28	−1800	
−25	−1500	−25	−1250	
−20	−970	−20	− 800	$S_3((\Delta Z)) \equiv S_2((\Delta Z))$
−10	− 470	−10	− 400	
0	0	0	0	
10	470	10	400	
12	530	12	500	
13	1000	15	1000	
$\Delta Z = Z_1 - Z_4$		$\Delta Z = Z_2 - Z_5$		$\Delta Z = Z_3 - Z_6$

Damping characteristics of suspension

A B C D		A B C D		B	C	D
$\Delta\dot{Z}$ [cm/s]	$K_1((\Delta\dot{Z}))$ [daN]	$\Delta\dot{Z}$ [cm/s]	$K_2((\Delta\dot{Z}))$ [daN]	$K_3((\Delta\dot{Z})) \equiv K_2((\Delta\dot{Z}))$	$K_3((\Delta\dot{Z})) = 20\cdot \mathrm{sign}\,(\Delta\dot{Z})$	$K_3((\Delta\dot{Z})) = 2\cdot \mathrm{sign}\,(\Delta\dot{Z})$
−30	−10	−30	−10.0			
−20	− 7.4	−20	− 7.0			
−10	− 3.7	−10	− 3.5			
0	0	0	0			
10	18.4	10	19.40			
20	36.8	20	38.80			
30	50.0	30	52.0			
$\Delta\dot{Z} = \dot{Z}_1 - \dot{Z}_4$		$\Delta\dot{Z} = \dot{Z}_2 - \dot{Z}_5$		$\Delta\dot{Z} = \dot{Z}_3 - \dot{Z}_6$		

Tyre spring characteristics

A B C D	A B C D	B C D
$S_4((\Delta Z)) = \begin{cases} 400\ \Delta Z; & (\Delta Z < 1.65) \\ 660; & (\Delta Z \geq 1.65) \end{cases}$	$S_5((\Delta Z)) = \begin{cases} 420\ \Delta Z; & (\Delta Z < 1.62) \\ 679; & (\Delta Z \geq 1.62) \end{cases}$	$S_6((\Delta Z)) \equiv S_5((\Delta Z))$
$\Delta Z = Z_4 - g_1$ [cm] S_4 [daN] $Z_S = 1.65$ [cm]	$\Delta Z = Z_5 - g_2$ [cm] S_5 [daN] $Z_S = 1.62$ [cm]	$\Delta Z = Z_6 - g_3$ [cm] S_6 [daN] $Z_S = 1.55$ [cm]

Tyre damping characteristics

A B C D

$$K_4((\Delta\dot{Z})) \equiv K_5((\Delta\dot{Z})) \equiv K_6((\Delta\dot{Z})) = \begin{cases} 0.12 \times \Delta\dot{Z}; & (if\ \Delta Z \leq Z_S) \\ 0; & (if\ \Delta Z > Z_S) \end{cases}$$

K_4, K_5, K_6 [daN]

Similarly to model **A**, we analysed also in the case of models **B**, **C** and **D** with six degrees of freedom the possibility of the vibrating system to decompose into independent vibrating systems.

The vibrating system decomposes if:

$$\left.\begin{aligned} M_{12} &= M_{21} = 0 \\ M_{13} &= M_{31} = 0 \\ M_{23} &= M_{32} = 0 \end{aligned}\right\} \tag{3}$$

Introducing the notations $\Theta_1 = M\vartheta_1^2$ and $\Theta_2 = m\vartheta_2^2$ the conditional system of equations becomes:

$$\left.\begin{aligned} \frac{M}{L^2}(l_1 l_2 - \vartheta_1^2) - \frac{a(1+a)\,m}{l^2}(l_5^2 + \vartheta_2^2) &= 0 \\ \frac{a \cdot m}{l^2}(\vartheta_2^2 - l_4 \cdot l_5) &= 0 \\ \frac{(1+a)\,m}{l^2}(l_4 l_5 - \vartheta_2^2) &= 0 \end{aligned}\right\} \tag{4}$$

According to the second and third equations of (4), a neccessary condition for the arise of independent vibrating systems is:

$$l_4 l_5 = \vartheta_2^2 \tag{5}$$

Using (5) and $a = l_3/L$ we can express l_3 from the first equation:

$$l_3 = -\frac{L}{2} \pm \left[\left(\frac{L}{2}\right)^2 + \frac{l_1 l_2 - \vartheta_1^2}{l_5} \cdot \frac{l\,M}{m}\right]^{0,5} \tag{6}$$

The dynamical meaning of Eq. (5) is that in case of its realisation the trailer and the tractor vibrate independently. Note, however, that in this case (and assuming identical excitations) the tractor and the single motor vehicle do not vibrate identically since masses M_1, M_2, M_{12} and M_{21} in Eqs. (1) and (2) are different.

If besides equality (5) also Eq. (6) for the coupling point l_3 between the trailer and the tractor is satisfied then the vibrating system of six degrees of freedom decomposes to three independent vibrating systems of two degrees of freedom. In the case of decomposition into independent vibrating systems Eq. (6) implies also:

1) The drawhead can get behind the rear axle of the hauling vehicle only if: $l_1 l_2 > \vartheta_1^2$

2) If $l_1 l_2 = \vartheta_1^2$ holds for the single motor vehicle then

$$l_3 = 0 \quad \text{or} \quad l_3 = -L.$$

In other words, if the front and rear vibrating systems of the motor vehicle perform a priori independent vibrations then the complex system with the trailer will decompose to three independent vibrating systems only if the point of coupling i.e. the drawhead is above one of the axles of the car.

This is a practically unaccomplishable requirement for a passenger car unless a special construction is employed. On the other hand, the designer of semitrailers has to cope with this requirement since $l_4 l_5 = \vartheta_2^2$ is possible for an adequate trailer design.

3) Provided $l_1 l_2 < \vartheta_1^2$ (which is the most likely case) then on the basis of (6), the drawhead must be located between the axles of the hauling vehicle, again a realisable requirement for semi-trailers.

Observations and conclusions drawn from the computer simulation

For the examination of the four models a program of digital simulation has been written in ALGOL 60 language for the computer ODRA 1204 of the Faculty of Transport Engineering of the Technical University, Budapest.

The most important results fo the simulation have been compiled in Table II and plotted in Figs. 1 to 7.

The first column of Table II. contains the vibration parameter symbols (for detailed explanation see the Appendix). The second column of the Table shows the vibration characteristics of the single passenger car. The other columns contain the corresponding characteristics of models **B C** and **D**.

The simulation has led to several technically useful inferences such as:

1) The vibration acceleration in the superstructure above the front axle of the passenger car is increased by the trailer, especially by one of a low damping.

2) The vibration acceleration in the superstructure above the rear axle of the passenger car is much reduced by the trailer even by one of a minimum of damping.

3) The vibration acceleration of the trailer superstructure depends considerably on the damping of the wheel suspension.

The vibration acceleration in a trailer with a minimum of damping will be about twice as in one with a damper, a hint to care for vibration damping in designing the suspension of trailers, contrary to the actual design approach to omit dampers from nearly all trailers and from most vans. Our examination demonstrates that if in the design only the Coulomb friction arising in the suspension is relied on for damping, then both the superstructure and the suspension of the trailer will be exposed to a considerable surplus stress.

4) The vibration accelerations in the superstructure of both the passenger car and the trailer vary characteristically along the longitudinal axis of the vehicle. Fig. 1 illustrates properly the standard deviation of the vertical

Table II

Vibration parameters	A	B	C	D
$D(\ddot{z}_1)$ [cm/s²]	132.58	139.1	139.58	141.08
$D(\ddot{z}_2)$ [cm/s²]	107.86	71.2	70.69	78.49
$D(\ddot{z}_3)$ [cm/s²]	—	95.9	100.3	180.87
$r(z_1, z_4)$	0.559	0.575	0.574	0.576
$r(z_2, z_5)$	0.501	0.478	0.479	0.446
$r(z_3, z_6)$	—	0.594	0.526	0.437
$D(\alpha)$ [rad]	0.01149	0.01290	0.01295	0.01358
$r(z_1, z_2)$	0.032	−0.185	−0.191	−0.2561
$D(\beta)$ [rad]	—	0.00887	0.00918	0.01061
$r(z_2, z_3)$	—	0.385	0.346	0.425
ST_{eff1} [%]	14.34	15.95	16.00	16.15
ST_{eff2} [%]	16.28	14.29	14.28	14.57
ST_{eff3} [%]	—	7.95	9.44	16.09
$r(z_4, g_1)$	0.970	0.966	0.966	0.966
$r(z_5, g_2)$	0.963	0.964	0.964	0.962
$r(z_6, g_3)$	—	0.988	0.983	0.953
S_{1eff} [daN]	82.42	91.90	92.32	93.41
S_{2eff} [daN]	65.21	62.05	61.88	65.48
S_{3eff} [daN]	—	49.89	56.41	106.02
K_{1eff} [daN]	27.93	28.15	28.19	28.37
K_{2eff} [daN]	28.13	28.96	29.03	29.28
P_1 [W]	61.7	63.30	63.51	64.17
P_2 [W]	68.9	63.54	63.54	65.16
$P_ö$ [W]	143.8	189.50	179.33	149.89

vibration accelerations at given geometrical points of the car bodies of each of the four vehicle complexes. We see that the trailer reduces further the vibration minimum of the passenger car and the most comfortable point shifts towards the rear axle of the motor vehicle.

5) The relation between the motion of the axle and that of the superstructure above it has been examined by computing the correlation coefficient.

$$r(z_i, z_j) = \frac{M[Z_i \cdot Z_j] - M[Z_i]\,M[Z_j]}{D(Z_i) \cdot D(Z_j)} \tag{7}$$

(where M and D stand for the expected value and the standard derivation resp.) The trailer is seen to have only little effect on the relation between the displacements.

6) The trailer affects slightly (increases a little) the standard deviation of the angular displacement of the nodding of the passenger car superstructure. The same is demonstrated by the variation of the correlation coefficient between the displacements above the axles of the passenger car body. The negative sign refers to the opposite motion i.e. to the nodding motion of the car body of models **B** **C** and **D**.

7) To characterize numerically the connection between the wheel and the ground a stability index was introduced.

$$ST_{\mathrm{eff}\,i} = \frac{D(Z_i - g)}{Z_s} \cdot 100\ [\%] \tag{8}$$

where Z_i — displacement coordinate of the given axle,

g — road excitation on the given axle;

Z_s — static depression of the tyre on the given axle.

The lower the value of $ST_{\mathrm{eff}\,i}$ is, the better the given wheel adapts to the road section i.e. the more stable the vehicle motion is. The variation of the stability index suggests that:

7/1 the stability of the front axle is slightly impaired by the trailer ($ST_{\mathrm{eff}1}$);

7/2 the stability of the rear axle is slightly improved ($ST_{\mathrm{eff}2}$);

7/3 the stability of the trailer axle is, however, much affected by the rate of vibration damping. The stability will be halved by removing the vibration damper ($ST_{\mathrm{eff}3}$).

8) One can get information on the roadability of the vehicle from the examination of the correlation coefficient between the motion of the given axle and the change of the road section. In our case a considerable positive correlation can be demonstrated, somewhat reduced by the trailer. A trailer of poor damping reduced the correlation by about 3%.

9) The dynamical stresses in the spring components of the suspension can be concluded on from the scatter of the spring force values. The trailer causes the dynamic spring load of the front suspension to increase by 10% and that of the rear suspension to decrease by 3%. Thus the change will be opposite to that of the static spring stresses. A lower damping will considerably increase the dynamic stresses in the trailer's spring components.

10) The trailer causes a slight increase of stresses in the vibration dampers of the passenger car.

11) The average power-absorption of the vibration dampers was computed from the well-known relation

$$P_i = \frac{1}{T} \int_0^T K_i(\dot{Z}(t)) \cdot \dot{Z}(t)\, dt \tag{9}$$

where T is the real time of simulation.

Hauling the trailer caused an increase of 2—3% of the power absorption in the damper of the front suspension and a reduction of 4—5% in the rear one. Obviously (and supported also by the numerical data) the vibration damper applied in the trailer adds to the overall power demand of the vehicle.

This power excess needs to be drained to improve the stability and the riding comfort.

12) The distribution functions of the vibration characteristics offer a possibility to draw further conclusions (Figs. 2 through 7).

In conformity with the precedings the distribution function $F_{\ddot{Z}_1}(x)$ of the vertical vibration acceleration of the superstructure part above the front axle changes only slightly (Fig. 2). The distribution functions $F_{\ddot{Z}_2}(x)$ (Fig. 3.)

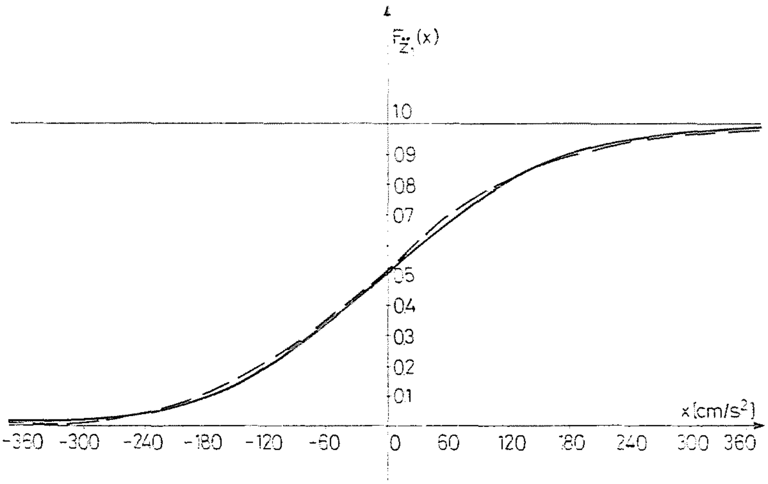

Fig. 2

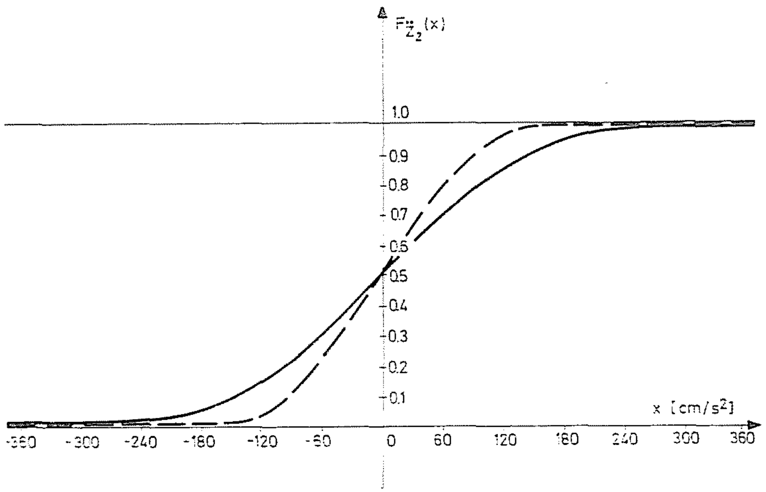

Fig. 3

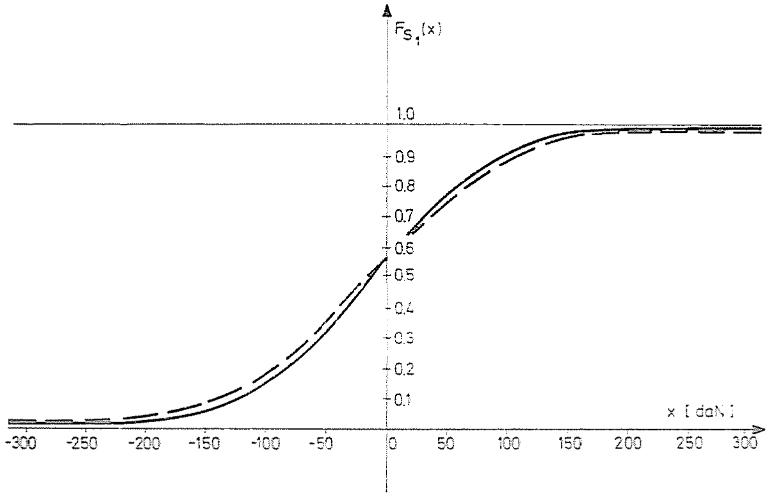

Fig. 4

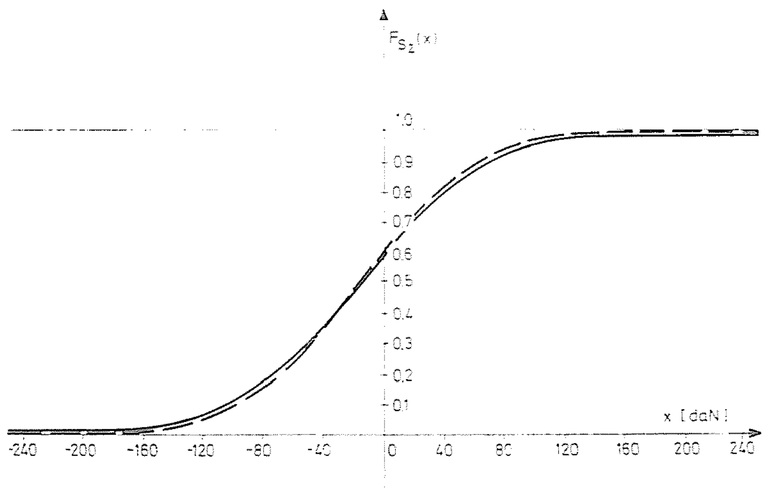

Fig. 5

of the vertical vibration acceleration of the superstructure part above the rear axle show that the probability of higher acceleration values decreases in a case of a trailer.

13) The distribution functions of the dynamic spring loads (Figs.4—5) show that the vibration dampers with asymmetrical characteristics shift the expected values in negative direction. The trailer hardly alters the course of the curves of distribution functions.

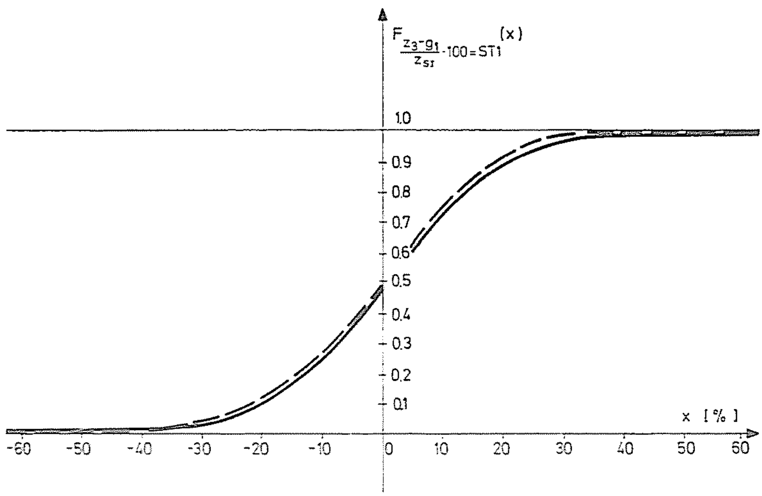

Fig. 6

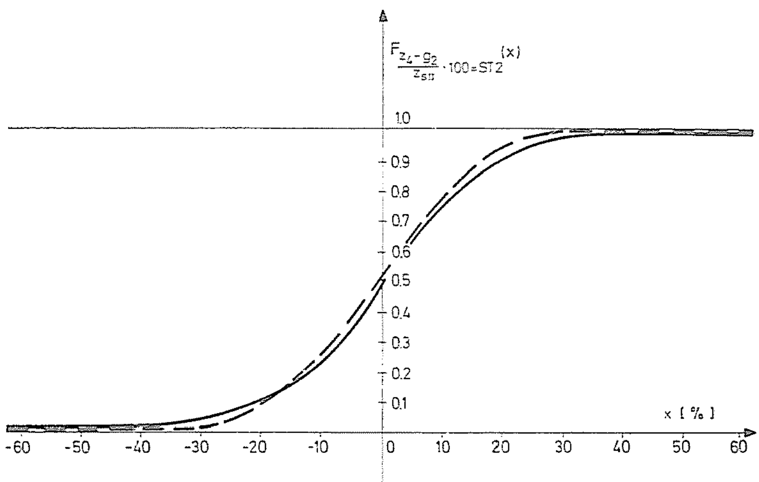

Fig. 7

14) The distribution functions of the stability indices are also only slightly affected by the trailer, their expected values are approximately zero (Figs. 6, 7).

Appendix

Symbol	Unit	Value	Definition
M	kg	1260	Single car body mass
m	kg	600	Van body mass
m_4	kg	60	Front axle mass
m_5	kg	80	Rear axle mass
m_6	kg	50	Van axle mass
Θ_1	kg · cm²	1.8×10^7	Moment of inertia of the single motor car body about its centroid
Θ_2	kg · cm²	8.8739×10^6	Moment of inertia of the van body about its centroid
$\vartheta^2 = \vartheta_1^2$	cm²	15 000	$\vartheta_1^2 = \vartheta^2 = \frac{\Theta_1}{M}$ -square of the inertia radius
ϑ_2^2	cm²	14 790	$\vartheta_2^2 = \frac{\Theta_2}{M}$ -square of the inertia radius
l_1	cm	120.3	Distance of the passenger car body centroid from the front axle
l_2	cm	121.7	Distance of the passenger car body centroid from the rear axle
l_3	cm	110.0	Distance of the rear axle from the drawhead
l_4	cm	247.0	Distance of the drawhead from the van body centroid
l_5	cm	28.0	Distance of the van body centroid from the van axle
L	cm	242.0	$L = l_1 + l_2$
l	cm	275.0	$l = l_4 + l_5$
$Z_1(t)$	cm	—	Displacement of the single motor car body above the front axle
$Z_2(t)$	cm	—	Displacement of the single motor car body above the rear axle
$Z_3(t)$	cm	—	Displacement of the van body above the axle
$Z_4(t)$	cm	—	Displacement of the front axle
$Z_5(t)$	cm	—	Displacement of the rear axle
$Z_6(t)$	cm	—	Displacement of the van's axle
$g_1(t)$	cm	—	Road excitation on the first wheel
$g_2(t)$	cm	—	Road excitation on the middle wheel
$g_3(t)$	cm	—	Road excitation on the van wheel
$\alpha(t)$	rad	—	Angular displacement of the passenger car body about its centroid
$\beta(t)$	rad	—	Angular displacement of the van body about its centroid
$\ddot{Z}_i(t)$	cm/s²	—	Acceleration of the i^{th} displacement coordinate
$D(Z_i$	cm	—	Standard deviation of the i^{th} displacement coordinate
$D\ddot{z}$	cm/s²	—	Standard deviation of the vertical acceleration of body points
$r(z_i, z_j)$	—	—	Correlation coefficient of variables Z_i and Z_j
ST_{effi}	%		$(i = 1, 2, 3)$ Stability of the first, second and third axles
S_{ieff}	daN	—	Effective mean value of spring forces in the i^{th} suspension $(i = 1, 2, 3)$
K_{ieff}	daN	—	Effective mean value of the damping forces in the first and the second suspension $(i = 1, 2)$
P_i	W	—	Effective power absorption in the i^{th} suspension damping $(i = 1, 2)$
$P_{\ddot{o}}$	W		Effective power absorption of all the dampings

Summary

The vibrational behaviour of the vehicle complex of a hauling passenger car and a trailer (van) is accessible to digital simulation. The program lends itself also for the examination of three-axle articulated buses and semi-trailers.

The vibrating system with six degrees of freedom of the vehicle complex is decomposed into independent subsystems at appropriate values of the wheel bases and centroid coordinates.

The trailer exerts a considerable influence on the vibration acceleration of the passenger car superstructure, on the axle stability i.e. on the steadiness of the wheels, on the stresses in the spring components and on the energy absorption of the vibration dampers.

The simulation permits to draw practical-technical conclusions to be taken into consideration in designing trailers or vans. To improve the roadability and the travel comfort as well as to reduce the dynamic stresses in the structural parts of the trailer and the van one has, contrary to the actual practice, to mount dampers also in the trailer and van wheel suspensions.

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