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## Modeliranje in optimiranje vzmetenja vozil Modelling and Optimizing of Vehicle Suspension

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*Razvili smo program za preračun in optimiranje vzmetenja vozil. Modeli vozil so sestavljeni iz togih teles in vzmetnih ter dušilnih elementov. Program najprej izračuna masno in togostno matriko ter matriko dušenja, nato pa izračuna lastne frekvence, lastne oblike in dušenje celotnega sistema. Sledi izračun vseh potrebnih prenosnih funkcij za računanje spektrov pospeškov ter za računanje udobnosti in dinamičnih sil. S programom je mogoče začetne modele vozil preveriti in izboljšati na podlagi primerjave z izmerjenimi spektri pospeškov. Prav tako je mogoče izvesti optimizacijo za najboljšo udobnost ali pa za minimalne dinamične sile na kolesih. Izmerili smo tudi karakteristike profilov naših cest in jih uporabili za vzbujanje na modelu vozila. Rezultati so prikazani na modelu avtobusa B3 090 T s petimi prostostnimi stopnjami.*

*A program for calculation and optimization of vehicle suspension consisting of rigid bodies, spring and damper elements was developed. First, mass, stiffness and damping matrices are constructed; then eigenfrequencies, mode shapes and damping of the whole system are calculated. Afterwards, all the necessary transfer functions for the generation of various spectra are computed. Vehicle ride comfort and dynamic tyre forces are also calculated. Initial models of vehicles can be verified and improved on the basis of comparison with measured acceleration autospectra. Optimization with respect to vehicle ride comfort and dynamic tyre forces can also be carried out. We measured road profiles of our roads, and later used these road profiles as input to our vehicle models. Results are shown on a model of the bus B3 090 T, with five degrees of freedom.*

### 0. UVOD

Preračun in optimiranje vzmetenja vozil je dokaj pomembna naloga pri projektiranju in konstruiranju novih vozil. V svetu poskušajo reševati celotno dinamiko vozil [1] – hkrati obravnavati več stvari, npr.:

- pasivno, polaktivno ali aktivno vzmetenje,
- zaviranje (ABS) in pospeševanje (ASR),
- krmiljenje na vseh štirih kolesih (4WS),
- pogon na vseh štirih kolesih (4WD),
- nihanje in dinamiko motorja.

Pri tem sta mogoča dva načina preračunov:

- v časovnem območju z nelinearnimi modeli vozil,

- v frekvenčnem območju z lineariziranimi modeli vozil.

Nelinearni modeli so običajno dokaj preprosti, čeprav novejši programi za nelinearno dinamiko (npr. ADAMS) zmorejo tudi bolj zapletene modele. Kljub temu pa je postopek preračunov dokaj zpletten in počasen. Pri linearnih modelih, ki jih rešujemo v frekvenčnem območju ni tolikšnih omejitev glede velikosti problemov, saj lahko ti modeli obsegajo nekaj sto prostostnih stopenj ali pa celo nekaj tisoč (modeli MKE).

### 0. INTRODUCTION

The calculation and optimization of vehicle suspension is quite an important task in designing new vehicles. Attempts are made to solve the entire vehicle dynamics [1], i.e. to deal with several problems simultaneously :

- passive, semi-active or active suspension,
- braking (ABS) and accelerating (ASR),
- four wheel steering (4WS),
- four wheel drive (4WD),
- vibrations and dynamics of the engine.

Here, two approaches are possible :

- calculations in the time domain with non-linear vehicle models,
- calculations in the frequency domain with linear vehicle models.

The non-linear models are usually rather simple, though the modern programs for non-linear dynamics (e.g. ADAMS) are also capable of more complicated models. Nevertheless, the procedure of calculations is quite complicated and time-consuming. With linear models solved in the frequency domain, there are no such restrictions as to the magnitude of the problems: such models can include several hundred or even several thousand degrees of freedom (FEM models).

Glede na to, da imajo naša vozila samo pasivno vzmetenje, se za zdaj ukvarjamo samo s to vrsto vzmetenja. Uporabili smo modalni način. Prikazali bomo gradnjo modelov vozil v frekvenčnem območju, preračun lastnih frekvenc in oblik nihanja, računanje prenosnih funkcij, računanje odzivov zaradi neravnin cestišča, preverjanje in izboljšavo modela in optimiranje parametrov vzmetenja vozila.

## 1. OPIS POSTOPKA REŠEVANJA

Razvili smo program za preračun dvodimenzijsionalnih modelov vozil, sestavljenih iz togih teles, ki so med seboj povezana z vzmetnimi oziroma dušilnimi elementi. Najprej je treba podati vse potrebne podatke za masne točke in toga telesa, nato podatke za vzmetne in dušilne elemente, sledijo povezave med posameznimi elementi. Na koncu je treba izbrati še točke na vozilu, v katerih želimo izračunati odzive. Nato program po vseh teh podatkih sam sestavi masno in togostno matriko ter matriko dušenja. Sledi izračun lastnih frekvenc in lastnih oblik nihanja. Značilno za modele vozil je, da so sistemi precej močno dušeni in zato dušenje ni sorazmerno. Izračun lastnih frekvenc in lastnih oblik, pa tudi različnih prenosnih funkcij, poteka po postopku, opisanem v [2]. Spektre odzivov izračunamo po naslednjem obrazcu za sisteme z več vstopi in izstopi (sistemi MIMO – Multi Input Multi Output) [3]:

$$G_{yy} = H_{y1}^2 G_{11} + H_{y1}^* H_{y2} G_{12} + H_{y2}^* H_{y1} G_{21} + H_{y2}^2 G_{22} \quad (1).$$

Pri tem pomenijo:  $G_{11}$ ,  $G_{22}$  – spektra vzbujanja na prednjem oziroma zadnjem kolesu,  $G_{12}$  – križni spekter med vzbujanjem na prednjem in zadnjem kolesu,  $H_{y1}$ ,  $H_{y2}$  – prenosni funkciji med vzbujanjem na prednjem oziroma zadnjem kolesu in odzivom v točki y. Izračunati je treba vse potrebne prenosne funkcije za računanje spektrov pospeškov, pomikov in relativnih pomikov na vozilu. Iz spektrov lahko izračunamo udobnost v različnih točkah na vozilu, dinamične sile na kolesih in dejanske vrednosti pomikov vzmeti.

## 2. PROFILI CESTIŠČ

Običajno se za vzbujanje zaradi neravnin cestišč upošteva naslednji izraz:

$$G_{11} = A(\Omega_0)(\Omega/\Omega_0)^{-W} \quad (2).$$

Bearing in mind that our vehicles feature only passive suspension, we are for the time being dealing only with this kind of suspension. We used a modal approach. The development of vehicle models in the frequency domain, calculation of eigenfrequencies and mode shapes, calculation of transfer functions, calculation of responses due to unevenness of the road surface, inspection and improvement of the model and optimization of the model and optimization of suspension parameters is presented.

## 1. NUMERICAL PROCEDURE FOR SOLVING 2D VEHICLE MODELS

We developed a programme for calculation of 2D vehicle models built of rigid bodies interconnected by spring and damper elements. It is first necessary to specify all the required data for mass points and rigid bodies, then data for spring and damping elements and connections between individual elements. Finally, it is necessary to select the points on the vehicle at which the responses should be calculated. With all these data to hand, the program will set up a mass and stiffness matrix and the damping matrix by itself. Next comes the calculation of eigenfrequencies and mode shapes. It is characteristic of vehicle models that the systems are damped considerably, so damping is not proportional. The calculation of complex eigenfrequencies and mode shapes, as well as of various transfer functions is carried out according to the procedure described in [2]. The autospectra of responses are calculated according to the following formula for systems with several inputs and outputs [3].

Here:  $G_{11}$  and  $G_{22}$  are autospectra of the excitation of front and rear wheels, respectively,  $G_{12}$  is the cross spectrum between excitation on front and rear wheels.  $H_{y1}$  and  $H_{y2}$  are transfer functions between excitation on front and rear wheels respectively and the response at point y. It is necessary to calculate all the required transfer functions for the calculation of autospectra of accelerations, displacements and relative displacements. From the autospectra, it is possible to calculate the ride comfort at several points on the vehicle, dynamic forces on the vehicle and effective values for spring displacements.

## 2. ROAD PROFILES

Due to unevenness of the road surface, the following formula is usually considered for road excitation:

Pri tem so:  $A(\Omega_0)$  in  $w$  — konstanti, odvisni od vrste in kakovosti cestišča,  $\Omega$  — krajevna frekvenca z enoto (cikl/m),  $\Omega_0 = 1$  cikl/m. V literaturi so podane vrednosti za te konstante za povprečne evropske ceste [4], [5]. Za naše ceste teh konstant nismo poznali. Ker pri našem postopku potrebujemo točne vrednosti teh konstant, smo se odločili sami izmeriti profile naših cest. Zgradili smo t.i.m. »peto« kolo. Najprej smo izmerili prenosno funkcijo tega kolesa. Na hidravličnem vibratorju smo izmerili odziv v sredini kolesa (pospeške) v odvisnosti od pomikov podlage. Ker je »peto« kolo dokaj linearen sistem, velja preprosta zveza med spektrom vzbujanja  $X(f)$  in spektrom odziva  $Y(f)$ :

$$Y(f) = X(f) H(f) \quad (3).$$

Tako dobimo prenosno funkcijo petega kolesa —  $H(f)$ . Pozneje smo izmerili pospeške med vožnjo po različnih cestah in določili spekture pomikov podlage po naslednji formuli:

$$G_{yy}(f) = G_{xx}(f) * H(f)^2 \quad (4),$$

pri tem pomenita:  $G_{xx}(f)$  — spekter neravnin cestišča,  $G_{yy}$  — spekter pospeškov v sredini kolesa. Po metodi najmanjših kvadratov smo določili konstanti  $A(\Omega_0)$  in  $w$  v izrazu (2). V našem primeru smo dobili naslednji vrednosti:

$$A(\Omega_0)$$

dober asfalt	1 E - 8	2,5	good asphalt
slab asfalt	5 E - 7	2,5	bad asphalt

### 3. RAČUNANJE UDOBNOŠTI IN DINAMIČNIH SIL

#### 3.1 Udobnost v vozilu

Na udobnost v vozilu vpliva več dejavnikov. Najpomembnejši so naslednji:

- nihanja — tresenje na tleh vozila, na sedežu, krmilnega kolesa, naslona za roke itn.,
- hrup v notranjosti vozila,
- temperatura in gibanje zraka ter vlažnost v prostoru — topotna udobnost,
- ergonomičnost, kako so oblikovani sedež, naslon za roke, kakšna je preglednost itn..

Pri numeričnem računanju udobnosti se največkrat upoštevajo le vibracije. Nekateri kombinirajo hrup in vibracije v enoten kriterij udobnosti. Vpliv vibracij na človeka je pri različnih frekvencah različen. V našem primeru smo udobnost računali po mednarodnem standardu ISO 2631 [6] in po priporočilih VDI 2054 [7]. Tukaj je merilo za udobnost uteženi dejanski pospešek, ki ga izračunamo takole:

Here  $A(\Omega_0)$  and  $w$  are constants, depending on the kind and quality of road surface.  $\Omega$  is the frequency with the units (cycles/m),  $\Omega_0 = 1$  cycle/m. The values for these constants for average European roads [4], [5] are reported in literature. For our roads, however, these constants were not known. Since in our procedure exact values of these constants were required, we decided to measure the profiles of our roads by ourselves. We built a so called »fifth wheel«. First we measured the transfer function of the »fifth wheel«. By means of an hydraulic vibrator, we measured the response in the middle of the wheel (accelerations) depending on the displacements of the base. Since the fifth wheel is a rather linear system, a simple connection between the spectrum of excitation  $X(f)$  and spectrum of response  $Y(f)$  applies :

$$Y(f) = X(f) H(f) \quad (3).$$

Thus the transfer function of the fifth wheel —  $H(f)$  is determined. Later we measured the accelerations during operation on different roads and determined the autospectra of displacements according to the following formula:

$$G_{yy}(f) = G_{xx}(f) * H(f)^2 \quad (4),$$

Here  $G_{xx}(f)$  is the autospectrum of unevennesses of road and  $G_{yy}$  the autospectrum of accelerations in the wheel centre. According to the method of the least squares we determined the constants  $A(\Omega_0)$  and  $w$  in formula (2). In our case, the following constants were obtained:

$$w$$

dober asfalt	1 E - 8	2,5	good asphalt
slab asfalt	5 E - 7	2,5	bad asphalt

### 3. CALCULATION OF RIDING COMFORT

#### 3.1 Riding comfort in vehicle

The riding comfort in the vehicle is influenced by several factors. The most important are:

- vibrations — on vehicle floor, on seat, vibrations of steering wheel, armrest, etc.,
- noise in the vehicle interior,
- thermal comfort — temperature and movement of air as well as ambient humidity,
- ergonomic properties — design of seat, armrest, visibility, etc.

In numerical calculations of ride comfort, in most cases only the vibrations are considered. In some cases, the noise and vibrations constitute a uniform criterion of riding comfort. The influence of vibrations on people varies with different frequencies. In our case, the riding comfort was calculated acc. to international standard ISO 2631 [6] and VDI 2057 recommendations [7]. The criterion for riding comfort is here the weighed effective acceleration calculated in the following way:

$$a_{\text{eff}}^2 = \int W(f) * S_{yy}(f) df \quad (5),$$

pri tem sta:  $S_{yy}(f)$  – spekter pospeškov,  $W(f)$  – utež pospeškov. Uteži dajejo različni kriteriji udobnosti. Vsak kriterij ima svoje uteži tako za navpična nihanja kakor tudi za vodoravna in vrtilna. Po VDI 2054 lahko izračunamo kriterij udobnosti, imenovan faktor  $K$ . Za navpična nihanja se ta kriterij izračuna:

$$K = 20 \quad (6).$$

### 3.2 Dinamične sile na kolesih

Zaradi neravnin cestišča niha pri vožnji sila med vozilom in tlemi okrog statične vrednosti. Ta odstopek od statične sile so dinamične sile. Ugodno je, da so te sile čim manjše. Dinamične sile vplivajo na varnost vožnje in stabilnost vozila. Pomembne so za učinkovito zaviranje, prav tako pri vožnji v ovinek. Delno vplivajo tudi na poškodbe cestišča, pa tudi vozila samega. Dejansko vrednost dinamičnih sil lahko izračunamo iz spektrov relativnih pomikov med podlago in premo:

$$F_{\text{din}} = K * (y_1 - x_1)_{\text{eff}} \quad (7),$$

kjer so:  $K$  – togost pnevmatike,  $y_1$  – pomik kolesa,  $x_1$  – pomik pnevmatike.

Med vožnjo vozilo ne sme poskakovati, zato mora biti dejanska vrednost dinamičnih sil na pnevmatiki vsaj trikrat manjša od statične sile na kolesu:

$$F_{\text{din}}/F_{\text{sta}} < 1/3 \quad (8).$$

### 4. OPTIMIRANJE UDOBNOSTI IN DINAMIČNIH SIL

Začetni model vozila je treba dobro preveriti in po potrebi izboljšati [8]. Šele potem se lahko lotimo študija, kako posamezni parametri vzmetenja vplivajo na udobnost in dinamične sile. Pri tem lahko narišemo udobnost ali dinamične sile v odvisnosti od določenega parametra vzmetenja. Prav tako lahko spremljamo lastne frekvence, celotne faktorje dušenja ali pa potreben prostor med premo in šasijo. Tako lahko z opazovanjem več diagramov izberemo primerne vrednosti za parametre vzmetenja (togosti vzmeti in dušenje blažilnikov spredaj in zadaj).

Zelo zanimiv je t.i.m. konfliktni diagram, kjer je narisana udobnost (faktor  $K$  ali utežni dejanski pospešek) v odvisnosti od razmerja dinamičnih sil proti statičnim. Konfliktni diagram je dobil ime po

where  $S_{yy}(f)$  signifies the autospectrum of accelerations and  $W(f)$  the weight for accelerations. The weights are defined by different criteria of riding comfort. Each criterion has its own weights for vertical, horizontal and rotational vibrations. Due to VDI 2057 the criterion of riding comfort called the  $K$  factor can be calculated. For vertical vibrations this criterion is:

### 3.2 Calculation of tyre dynamic forces

Due to the unevennesses of the road surface, the force between the vehicle and ground oscillates around the static force during operation. This deviation from the static force is called the dynamic force. The dynamic forces influence the reliability of operation and stability of the vehicle. They enable effective braking and are very important during cornering. They are influenced also partially by damage to the road surface and the vehicle itself. The effective value of dynamic forces can be calculated from autospectra of relative displacements between the base and axle:

where  $K$  is the tyre stiffness,  $y_1$  – displacement of wheel,  $x_1$  – displacement of tyre.

During the operation the vehicle must not jump up and down, therefore the effective value of dynamic forces on the tyre should be at least three times smaller than the static force on the wheel:

### 4. OPTIMIZING THE RIDING COMFORT AND DYNAMIC FORCES

The initial vehicle model should be checked thoroughly and improved, if necessary [8]. Only then should a study of the influence of individual suspension parameters on riding comfort and dynamic forces be made. The riding comfort or dynamic forces can be drawn as a function of certain suspension parameters. We can also pursue these eigenfrequencies, global damping factors or the working space between axle and chassis. By observing a large number of diagrams, we can thus select appropriate values for the suspension parameters (stiffness of springs and damping of shock absorbers, front and rear).

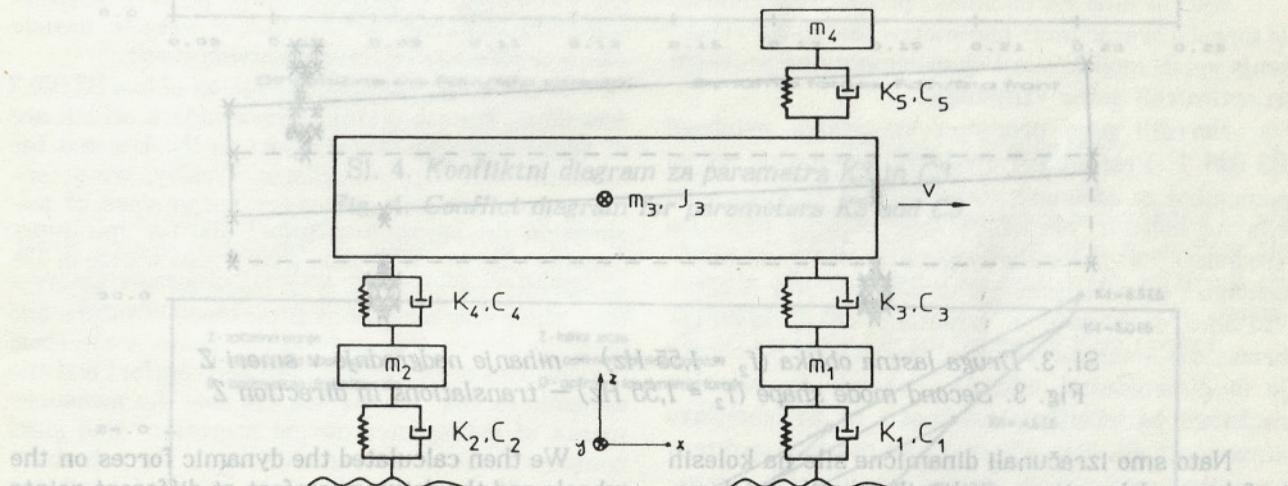
In the so called conflict diagram, the riding comfort ( $K$  factor or the weighted effective acceleration) is drawn as a function of the ratio of dynamic and static forces on tyres. The designation »conflict diagram« can be ascribed to the fact

tem, ker imamo na njem v bistvu dve najugodnejši točki – ena je najugodnejša glede na udobnost, druga pa za dinamične sile. Izbira ene ali druge je odvisna od potreb, npr. pri športnem avtomobilu je treba izbrati najugodnejše točke glede na dinamične sile, pri limuzinah pa glede na udobnost.

Pri nas lahko poiščemo najboljše po enem od obih kriterijev (udobnost, dinamične sile) tudi s popolnoma avtomatskim postopkom. Uporabili smo metodo simpleks za iskanje minimuma funkcije več spremenljivk. Najprej je treba izbrati kriterij optimiranja, nato se izberejo parametri. Prav tako lahko določimo omejitve, ki jih je treba pri optimirjanju upoštevati: npr. omejitve za dinamične sile, delovni prostor vzmeti, statični poves vzmeti, vrednosti parametrov itn..

## 5. REZULTATI ZA AVTOBUS B3 090 T

Razvili smo dinamični model avtobusa B3 090 T s petimi prostostnimi stopnjami (sl. 1).



Sl. 1. Model avtobusa TAM B3 090 T s petimi prostostnimi stopnjami

Fig. 1. Model of the bus TAM B3 090 T with five degrees of freedom

Preglednica 1: Nedušene in dušene lastne frekvence, faktorji dušenja in lastne oblike

Table 1: Undamped and damped eigenfrequencies, damping factors and mode shapes

Mod Mode	$f_0$ Hz	$f_d$ Hz	faktorji dušenja damping factors	oblika nihanja mode shape
1	1,12	1,10	0,215	pokimavanje avtobusa – $R_y$ bus pitching
2	1,55	1,50	0,254	nihanje nadgradnje v smeri Z translation of body in Z direction
3	2,45	2,10	0,386	nihanje sedeža motion of seat in Z direction
4	10,45	10,23	0,204	nihanje zadnje preme motion of rear axle
5	12,47	11,91	0,296	nihanje sprednje preme motion of front axle

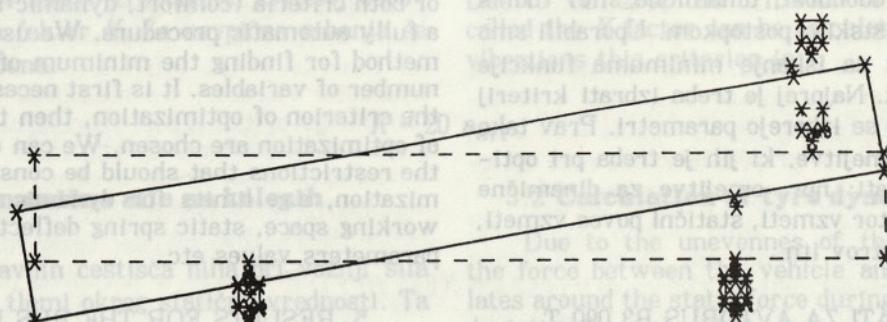
that it has two optimal points – one is the optimum with regard to comfort and the other with regard to dynamic forces. The selection of one or other depends on needs, e.g. in a sports car it is necessary to select the optimum with regard to dynamic forces, in a limousine, however, the optimum with regard to comfort.

We can find the optimum with regard to one or both criteria (comfort, dynamic forces) also by a fully automatic procedure. We used the simplex method for finding the minimum of function of a number of variables. It is first necessary to select the criterion of optimization, then the parameters of optimization are chosen. We can also determine the restrictions that should be considered in optimization, e.g. limits for dynamic forces, spring working space, static spring deflection, limits for parameters values etc.

## 5. RESULTS FOR THE BUS B3 090 T

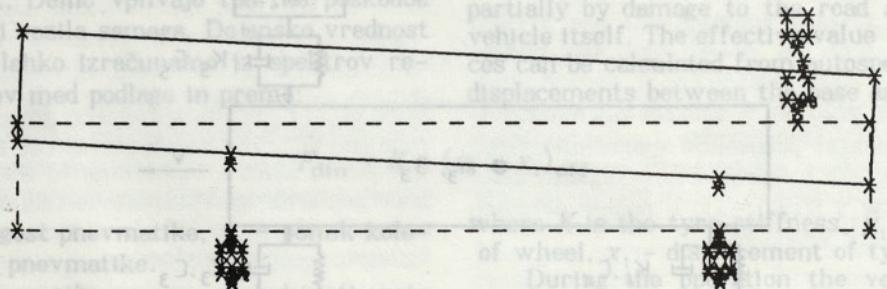
We developed a dynamic model of the bus B3 090 T with five degrees of freedom (figure 1).

Model ima tri točkovne mase (prednja in zadnja prema ter sedež z voznikom), togo telo (nadgradnja) ter pet vzmetnih oziroma dušilnih elementov. Najprej smo izračunali nedušene in dušene lastne frekvence ter celotne faktorje dušenja in pripadajoče lastne oblike nihanja (preglednica 1). Prvi dve lastni obliki sta prikazani na slikah 2 in 3.



Sl. 2. Prva lastna oblika ( $f_1 = 1,12 \text{ Hz}$ ) — pokimavanje avtobusa

Fig. 2. First mode shape ( $f_1 = 1,12 \text{ Hz}$ ) — pitching mode



Sl. 3. Druga lastna oblika ( $f_2 = 1,55 \text{ Hz}$ ) — nihanje nadgradnje v smeri Z

Fig. 3. Second mode shape ( $f_2 = 1,55 \text{ Hz}$ ) — translations in direction Z

Nato smo izračunali dinamične sile na kolesih in faktor udobnosti na različnih mestih avtobusa. Na sliki 4 je prikazan konfliktni diagram pri hkratnem spremenjanju parametrov sprednjega vzmetenja — togosti vzmeti in dušenja spredaj. Na sliki 5 je prikazan konfliktni diagram pri hkratnem spremenjanju parametrov zadnjega vzmetenja — togosti vzmeti in dušenja zadaj. Na teh slikah je označeno začetno stanje (Z), najboljša udobnost (U) in najboljše dinamične sile (D), s črtkano črto pa meja največjih mogočih dinamičnih sil. Oba konfliktna diagraoma sta narejena za vožnjo po slabem asfaltu.

Na začetnem modelu avtobusa smo izvedli optimiranje udobnosti (faktorja  $K$ ) in dinamičnih sil na kolesih. Izbrali smo avtomatski način optimiranja, pri katerem smo dovolili spremenjanje štirih parametrov, in sicer:

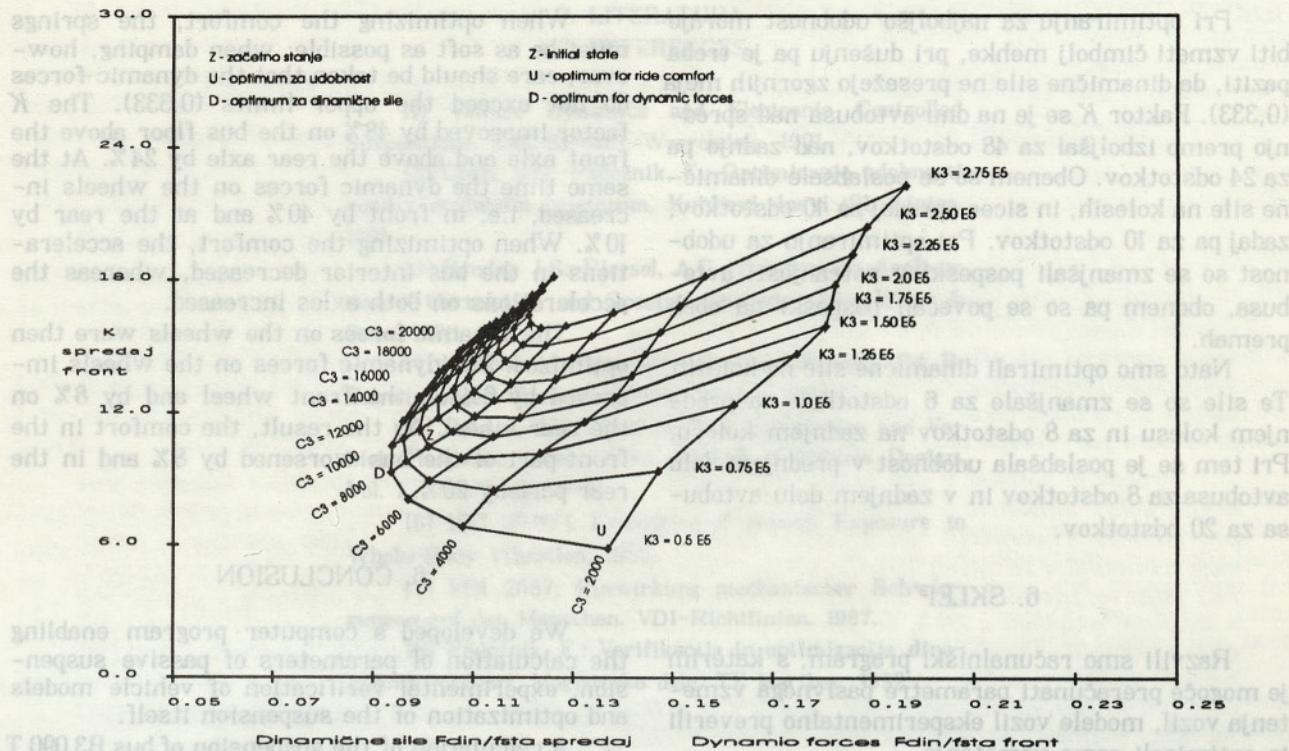
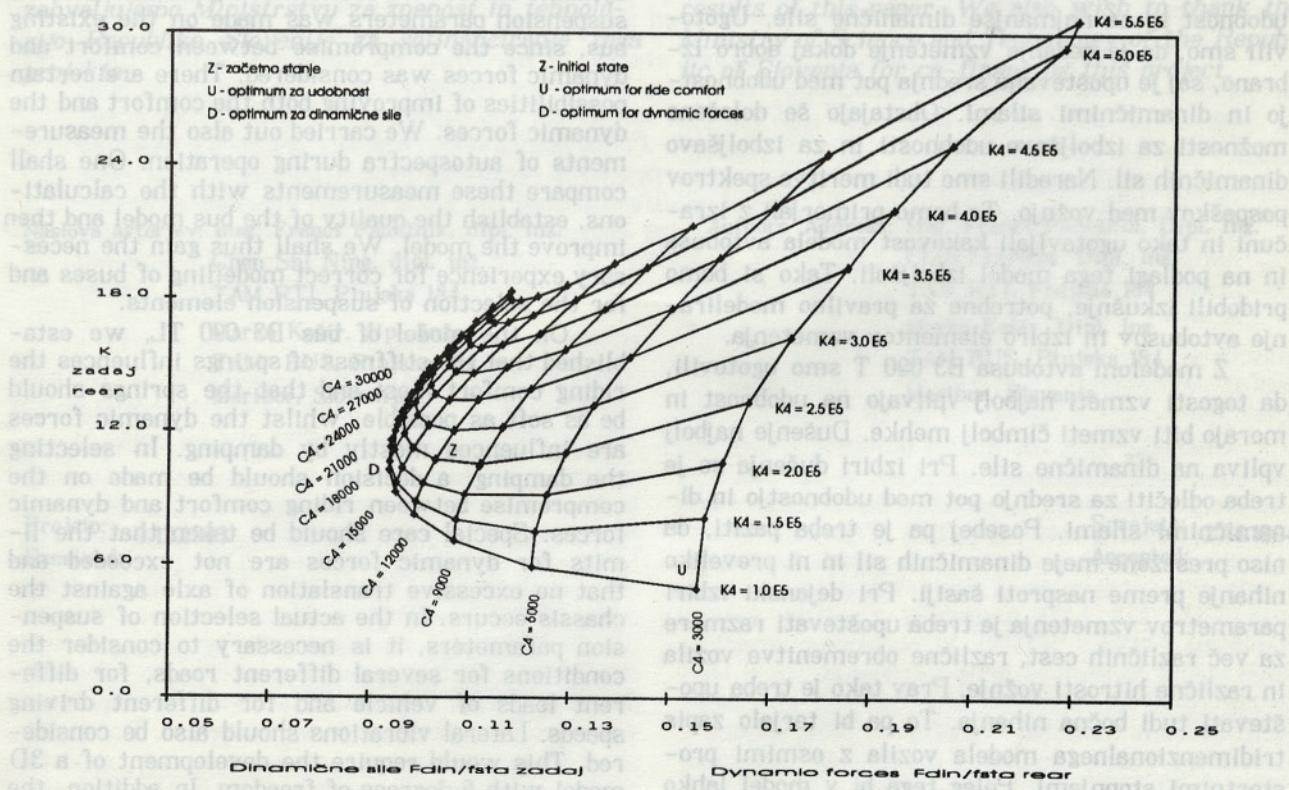
- $K_3$  — togost prednje zračne vzmeti,
- $C_3$  — dušenje obeh prednjih blažilnikov,
- $K_4$  — togost obeh zadnjih zračnih vzmeti,
- $C_4$  — dušenje obeh zadnjih blažilnikov.

The bus model consists of three point masses (front and rear axle and seat with driver), a rigid body (superstructure) and five spring and damping elements. First, we calculated undamped and damped eigenfrequencies and global damping factors and corresponding mode shapes (table 1). The first two mode shapes are shown in figs. 2 and 3.

We then calculated the dynamic forces on the wheels and the driving comfort at different points on the bus. Figure 4 shows the conflict diagram by simultaneously varying the parameters of front suspension — stiffness of spring and damping in front. Figure 5 shows the conflict diagram by simultaneously changing the parameters of rear suspension. Designated in these illustrations are the initial state (Z), optimum for comfort (U) and optimum for dynamic forces (D), and the limit for maximum possible dynamic forces by the broken line. Both conflict diagrams are drawn for driving on bad asphalt.

On the initial bus model we carried out the optimization of comfort ( $K$  factor) and dynamic forces on the wheels. We selected the automatic mode of optimization by changing four parameters, i.e.:

- $K_3$  — stiffness of front air spring,
- $C_3$  — damping of both front shock absorbers,
- $K_4$  — stiffness of both rear air springs,
- $C_4$  — damping of both rear shock absorbers.

Sl. 4. Konfliktni diagram za parametra  $K_3$  in  $C_3$ Fig. 4. Conflict diagram for parameters  $K_3$  and  $C_3$ Sl. 5. Konfliktni diagram za parametra  $K_4$  in  $C_4$ Fig. 5. Conflict diagram for parameters  $K_4$  and  $C_4$

Pri optimirjanju za najboljšo udobnost morajo biti vzmeti čim bolj mehke, pri dušenju pa je treba paziti, da dinamične sile ne presežejo zgornjih meja (0,333). Faktor  $K$  se je na dnu avtobusa nad sprednjo premo izboljšal za 48 odstotkov, nad zadnjo pa za 24 odstotkov. Obenem so se poslabšale dinamične sile na kolesih, in sicer spredaj za 40 odstotkov, zadaj pa za 10 odstotkov. Pri optimirjanju za udobnost so se zmanjšali pospeški v notranjosti avtobusa, obenem pa so se povečali pospeški na obeh premah.

Nato smo optimirali dinamične sile na kolesih. Te sile so se zmanjšale za 6 odstotkov na prednjem kolesu in za 8 odstotkov na zadnjem kolesu. Pri tem se je poslabšala udobnost v prednjem delu avtobusa za 8 odstotkov in v zadnjem delu avtobusa za 20 odstotkov.

## 6. SKLEP

Razvili smo računalniški program, s katerim je mogoče preračunati parametre pasivnega vzmetenja vozil, modele vozil eksperimentalno preverili in optimirali samo vzmetenje.

Naredili smo preračun vzmetenja avtobusa B3 090 T. Preračunali smo vse parametre, ki so pomembni za udobnost v avtobusu in za dinamične sile na kolesih. Na koncu smo določili najboljše vrednosti za parametre vzmetenja za najboljšo udobnost in za najmanjše dinamične sile. Ugotovili smo, da je sedanje vzmetenje dokaj dobro izbrano, saj je upoštevana srednja pot med udobnostjo in dinamičnimi silami. Obstajajo še določene možnosti za izboljšavo udobnosti in za izboljšavo dinamičnih sil. Naredili smo tudi meritve spektrov pospeškov med vožnjo. Te bomo primerjali z izračuni in tako ugotavljali kakovost modela avtobusa in na podlagi tega model izboljšali. Tako si bomo pridobili izkušnje, potrebne za pravilno modeliranje avtobusov in izbiro elementov vzmetenja.

Z modelom avtobusa B3 090 T smo ugotovili, da togosti vzmeti najbolj vplivajo na udobnost in morajo biti vzmeti čim bolj mehke. Dušenje najbolj vpliva na dinamične sile. Pri izbiri dušenja se je treba odločiti za srednjo pot med udobnostjo in dinamičnimi silami. Posebej pa je treba paziti, da niso presežene meje dinamičnih sil in ni preveliko nihanje preme nasproti šasiji. Pri dejanski izbiri parametrov vzmetenja je treba upoštevati razmere za več različnih cest, različne obremenitve vozila in različne hitrosti vožnje. Prav tako je treba upoštevati tudi bočna nihanja. To pa bi terjalo zapis tridimenzionalnega modela vozila z osmimi prostostnimi stopnjami. Poleg tega bi v model lahko vključili tudi motor vozila. Posebej bi bilo treba natančno izmeriti karakteristike sedeža in pnevmatike.

When optimizing the comfort, the springs must be as soft as possible; when damping, however, care should be taken that the dynamic forces do not exceed the upper limits (0.333). The  $K$  factor improved by 48% on the bus floor above the front axle and above the rear axle by 24%. At the same time the dynamic forces on the wheels increased, i.e. in front by 40% and at the rear by 10%. When optimizing the comfort, the accelerations in the bus interior decreased, whereas the accelerations on both axles increased.

The dynamic forces on the wheels were then optimized. The dynamic forces on the wheels improved by 6% on the front wheel and by 8% on the rear wheel. As the result, the comfort in the front part of the bus worsened by 8% and in the rear part by 20%.

## 6. CONCLUSION

We developed a computer program enabling the calculation of parameters of passive suspension, experimental verification of vehicle models and optimization of the suspension itself.

A calculation of the suspension of bus B3 090 T was done. We calculated all parameters which are of significance for the comfort in the bus and for dynamic forces on the wheels. Finally, we determined the optimum values for parameters of suspension for optimum comfort and for minimum dynamic forces. We found that a good choice of the suspension parameters was made on the existing bus, since the compromise between comfort and dynamic forces was considered. There are certain possibilities of improving both the comfort and the dynamic forces. We carried out also the measurements of autospectra during operation. One shall compare these measurements with the calculations, establish the quality of the bus model and then improve the model. We shall thus gain the necessary experience for correct modelling of buses and for the selection of suspension elements.

On the model of bus B3 090 TL, we established that the stiffness of springs influences the riding comfort most and that the springs should be as soft as possible, whilst the dynamic forces are influenced mostly by damping. In selecting the damping, a decision should be made on the compromise between riding comfort and dynamic forces. Special care should be taken that the limits for dynamic forces are not exceeded and that no excessive translation of axle against the chassis occurs. In the actual selection of suspension parameters, it is necessary to consider the conditions for several different roads, for different loads of vehicle and for different driving speeds. Lateral vibrations should also be considered. This would require the development of a 3D model with 8 degrees of freedom. In addition, the vehicle engine could also be integrated in the bus model. The characteristics of the seat and tyres should be determined with utmost accuracy.

Standard FMVSS-571.220 opisuje potek 7. LITE kusa pasivne varnosti avtobusa za prevoz Š. 7. REF v primeru prečnega vozila. Trenutno je však izdelovalec ter strinjal, da bo pojavljal na avtobusih s tehnologijo Vehicle Dynamics Suspensions. SAE SP-861, preprost preizkus, pri katerem ustreznost konstrukcije vozil z modalnim pristopom.

## 7. LITERATURA

## 7. REFERENCES

- [1] Vehicle Dynamics and Electronic Controlled Suspensions. SAE SP-861. Warrendale, 1991.
  - [2] Chen, S.N.-Padežnik, F.: Optimiranje udobnosti vozil z modalnim pristopom. Kuhljevi dnevi '91, Lipica, 1991.
  - [3] Bendat, J.S.-Piersol, A.G.: Engineering Application of Correlation and Spectral Analysis. John Wiley & Sons, 1980.
  - [4] Mitsche, M.: Dynamik der Kraftfahrzeuge. Bd. B: Schwingungen. Springer-Verlag, Berlin, 1984.
  - [5] Robson, J.D.: Road Surface Description and Vehicle Response. International Journal of Vehicle Design. Vol. 1, 1979/1, pp.25-35.
  - [6] ISO 2631/1: Evaluation of Human Exposure to Whole-Body Vibration, 1985.

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