A Contribution to Optimizing the Power Train Suspension

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ABSTRACT
Contemporary passengers motor vehicles have to achieve a great deal of comfort requirements. Problems of noise and vibration reduction in vehicles may be solved by choice of optimal power train (PT) suspension. In this paper an analytical method is developed. Also, a global model of both vehicle and PT is developed and analyzed. Experimental investigations (laboratory and in field conditions) were done to verify the model. The adopted model was initially used to form the optimization method. Modified Hooke Jeeves method of optimization was combined with results obtained in the experimental research and modeling to form the optimization method. Obtained results from the optimization were a base for PT reconstruction of the analyzed vehicle.

1. INTRODUCTION
Motor vehicles have to satisfy a great deal of ride comfort criteria. Velocities of passengers cars are greater and greater causing increase of mechanical loads. Problems of isolation of noise and vibration are greater so their solution needs detailed investigation. PT suspension is one of the systems that can be called an "absorber of noise and vibration." Elastic suspensions have to obey both necessary comfort and good functions of assemblies and vehicle.

In the papers by Simić and Demić, (1990) and Toljskij, (1976) these problems were analyzed in detail. Bearing in mind the deficiency of these papers, a global model of vehicle with PT was formed including the influence of ride comfort elements. Experimental investigations used to verify the model represent the basis for a method of parameter identification of PT suspension elements.

2. DYNAMIC MODEL
Different kinds of model can be used to analyze oscillatory movements of PT, (Demić, 1990, etc.). Linear models can be used to evaluate the influence of elasto-damping characteristics of mounts on vertical vibrations. Linear models with few masses are useful to investigate the influence of road roughness on vertical oscillations of PT, (Simić and Demić, 1990).

Other models are appropriate for analyses of angular and vertical vibrations of PT. There are many possibilities for reduction of vibration and noise in vehicles as well as in appropriate PT suspension. This includes choice of optimal number, position and elasto-damping characteristics of the PT.

Problems can be solved by use of experimental empirical and analytical methods. It is usual in practice to employ all of the above-mentioned methods.

In this paper an analytical method was used. For getting improved suspension for defining analytical methods, experimental results were used, as well as producers recommendations (FAMOS 1991, NAMI, 1983).

Depending on the problem to be solved, one can find in the references various mechanical models of the PT and the vehicle.

The PT as a solid body in space has six degrees of freedom. It was connected to the car body across N mounts with known non linear characteristics. Weight and dimensions of the mounts were neglected, (Simić and Demić, 1990, Toljskij, 1976).

The car body is considered as a solid plate resting on the ground through the suspension system, unsprung mass and tyres.

For a description of the oscillatory movements of the PT in this paper, the following reference frames were used: ground fixed coordinate frame OXYZ and moving coordinate systems TXYZ and T_pXYZ fixed for car body and PT respectively, (Pantović, Lukić, 1995). Moving coordinate systems coincide with geometrical axes and their origins with centers of gravity.

Adopted coordinate systems and their axes are not principal axes of car body and PT and its geometry axes, shown on Figure 1.

The space model, according to generalized coordinates, describes the oscillatory movements of vehicle elements.

Adopted generalized coordinates are:

- q_1 – vertical displacement of front left unsprung mass,
- q_2 – vertical displacement of front right unsprung mass,
- q_3 – vertical displacement of rear left unsprung mass,
- q_4 – vertical displacement of rear right unsprung mass,
- q_5 – straight line motion of vehicle,
- q_6 – lateral motion of vehicle,
- q_7 – vertical motion of vehicle,
- q_8 – roll of car body,
- q_9 – pitch of car body,
- q_{10} – yaw of car body,
- q_{11} – straight line motion of PT,
- q_{12} – lateral line motion of PT,
- q_{13} – vertical line motion of PT,
- q_{14} – roll of PT,
- q_{15} – pitch of PT,
- q_{16} – yaw of PT.

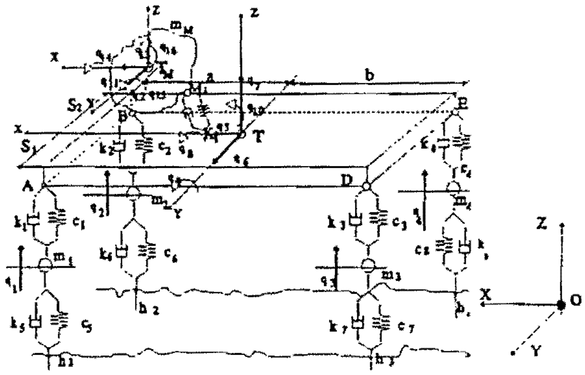

Figure 1. Dynamic model

Differential equations of an oscillatory movement are written according to Delambert's principal:

$$\bar{F}_R^a + \bar{R}_R + \bar{F}_R^i = 0 \tag{1}$$

and

$$\bar{M}_o^{F_R^a} + \bar{M}_o^{R_R} + \bar{M}_o^{F_R^i} = 0 \tag{2}$$

where:

- F_R^a – principal vector of active forces which act on system,
- R_R – principal vector of reactions,
- F_R^i – principal vector of inertial forces,
- M_o^{FR} – principal vector of moment of active forces that act on system according to fixed pole O,
- $M_o RR$ – principal vector of moment of reactions for fixed pole O,
- M_o^F – principal vector of moment of inertial forces for pole O.

Differential equations of movements for a dynamic model were given in (Pantović Lukić, J. 1995). Differential equations are nonlinear, with random time function of excitations. They were solved numerically, by the Runge-Kutta method. Time domain was 5 s with integration steps of 10^{-3} s, which is appropriate for the frequency domain from 0.2 to 500 Hz, (Demić, 1990, Toljskij, 1976). This frequency domain is of special interest in investigation of the noise and vibration influence on ride and acoustic comfort, (Toljskij, 1976).

3. EXCITATION

Dynamic models of vehicle and PT in the mentioned references (Simić and Demić, 1990, Toljskij, 1976, etc.) did not consider the excitation from both road roughness and from the engine. Also, the influence of PT vibrations was neglected.

In the paper road excitations were considered as a random stationary function. Determinations of road roughness characteristics were done according to Simić and Radonjic, 1975.

PT vibrations caused by the engine can be due to:

- unbalanced inertial forces of translatory and rotational masses,
- gas pressure forces,
- nonuniform engine torque and
- different piston masses, etc.

Oscillatory movements of PT can be caused by: inertial forces of translatory masses and inertial forces of a rotary masses.

Determination from exciting engine was made by thermodynamics and kinetic numeric calculations of internal combustion engines. Results were used as input data for solving the system of differential equations and in the parameter identification method.

4. EXPERIMENTAL VERIFICATION OF DYNAMIC MODEL

In order to verify the model developed above, experiments were performed which were aimed at measuring the translatory vibrations of PT mounts, by running the engine in a stationary vehicle from idling speed to 3000 rpm. Translatory vibrations of PT and car body on the point of connection with the mount were measured by means of B&K and HMB acceleration transducers. Experiments were performed in the laboratory (from idling speed to 3000 rpm, on the roller) and on a horizontal, good concrete road.

The passenger car used in the experiments had characteristics given in (Pantović Lukić, 1995. Verification of the model needed approximately the same conditions under simulation as well as experiments.

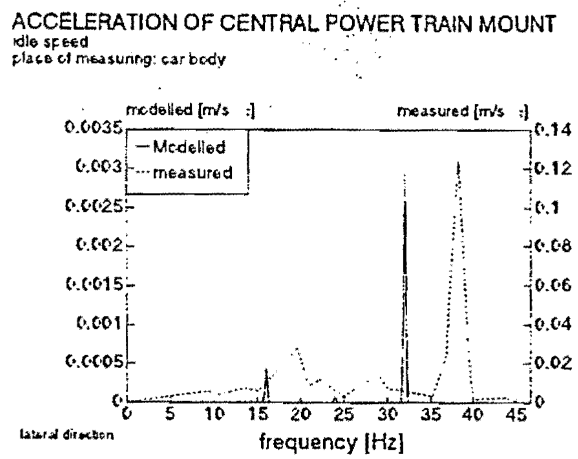

Figure 2. Comparing results obtained by modeling and measuring

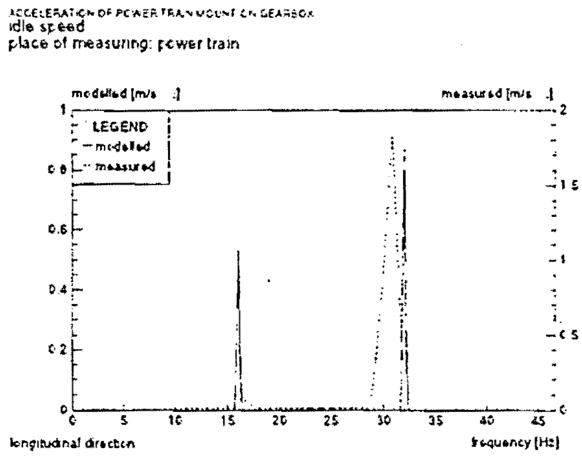

Figure 3. Comparing results obtained by modeling and measuring

Figures 2 and 3 compare the results of experiment and simulations at the point of connection between mount and PT. Also, there are shown comparison of experimental and simulating results at the point of connection between mount and car body (figures 2 and 3). From the figures we can conclude that there is good agreement of results from the dominant frequency. Values of amplitude are similar for measurements on the engine but on the car body they have not the same order of values. That is why the car body was modeled as a rigid body. All figures show that the dominant frequency corresponds with the second order harmonic of the engine.

Also, differences between experimental and simulated results are caused by: nonuniformity of tyres (very important on the roller) (Demić, 1993), impossibility to completely repeat the conditions, and deviation of simulation of conditions from experiment.

5. OPTIMIZATION METHOD

The Hook Jeeves method (e.g. Bunday, 1984) was modified to make it possible to solve complex problems of optimization of the positions and characteristics of PT mounts, including the design constraints. The modification consisted of using the method of "external penalty functions," whenever the computed values of the optimization parameters left the respective allowed design ranges, and using a selective increment of the optimizing parameter in the first iteration step.

In the case analyzed the following objective function was used:

$$z=\alpha \ddot{z}_{V_{eff}}+\beta \ddot{z}_{a_{eff}}-\gamma\sum_{i=1}^{n} I_{m_{eff}}(i) \tag{3}$$

where:

- $\ddot{z}_{veff}$ – RMS values of PT vertical acceleration,
- $\ddot{z}_{aeff}$ – RMS values of PT angular acceleration for crankshaft axes,
- $I_{meff}(i)$ – RMS values of mechanical impedance of i^{th} mount,
- N – number of the mounts and
- α, β and γ – empirical coefficients that determine the influence rank.

The biggest problem in defining objective functions is the choice of rank coefficient. There are no recommendations in the references for choice of α, β and γ, so in the paper the influence of this parameter on objective function is considered.

The objective functions used reduce as much as possible the vertical, angular vibrations and internal noise of the vehicle. Minimization of noise can be modeled by maximizing the mechanical impedance in the points where mounts are connected, (Toljskij, 1976). It is well known that mechanical impedance is proportional to noise.

The objective function has to satisfy two opposite aims: to reduce vertical and angular accelerations and to maximize mechanical impedance in the points where mounts are connected. It included an increase of mount damping and decrease of vibration in the low frequency domain and in the high frequency domain noise increased.

Software was developed for solving the problem of parameter identification and it was automatically stopped when:

- difference between two successive objective functions is less than 1*e-8 and
- difference between two near values of optimum parameter is less than $1*e^{-3}$.

The initial optimization step is determined according to Demić, 1990

$$k[j]=\frac{x[j]}{100} \tag{4}$$

Rank coefficients were taken into account on the base of the initial values of maximum acceleration level. Also, choice of rank parameters depend on speed of vehicle, exploitation and initial conditions. Coefficient Y has the greatest influence on the objective function. Initial values of mechanical impedance in every point of connection are much higher than initial values of acceleration of C.O.G. of PT.

A favourable choice of coefficient γ makes it possible to equalise the level of influence, shown in figure 4.

For α and β rank coefficient relative ratio of 10^1 order was adopted and it was 1/50000 for γ, which is shown as the appropriate value according to Figure 4.

Chosen values $\alpha = 10$, $\beta = 100$ and $\gamma = 1/50000$ are accepted as satisfactory because they take into account the relative ratio of RMS values of $\ddot{z}_{veff}$, $\ddot{z}_{aeff}$ and I_{meff}. For different exploitation regimes it could be advocated to take different rank coefficients because both vehicle and PT have passive suspensions.

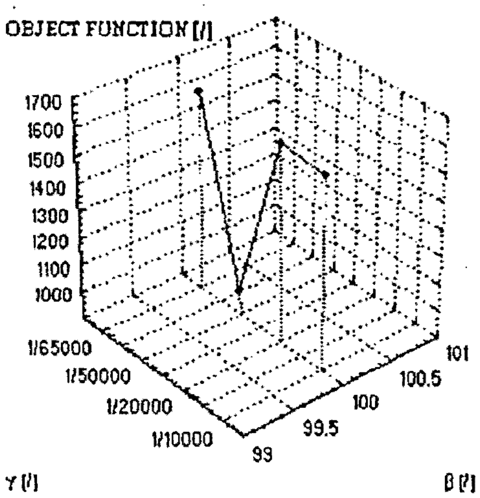

Figure 4. Rank coeficients influence

6. RESULTS OF OPTIMIZATION

The aim of the work is to improve PT suspension in order to obtain minimization of noise and vibration. Optimization was held for passenger cars with a fourstroke four cylinder engine sprung in three points. The engine is placed laterally, with front drive wheels.

Physical and construction constraints of parameters are caused by free space and motor equipment, and are given in (Pantović Lukić, 1995.)

The speed has a significant influence on noise and vibration in vehicles with passive suspension systems. The optimization is thus performed for the range of the most relevant speeds.

The analyses show that for this group of passenger car characteristic speeds are 30-45 m/s with use on asphalt roads of good quality and in these analyses the optimization was carried out for these conditions.

Since the velocity on roads is usually limited to 120 km/h, the authors considered it appropriate to optimize the PT suspension system parameters for velocity of 30 m/s, based on what, as optimum values, were adopted values given in Pantović, Lukić, 1995. The suggested optimization method was done for three groups of parameters.

In Figures 5 and 6 the results of optimization are compared with initial values. Amplitudes of angular and vertical accelerations are reduced. Maximum amplitudes of vertical accelerations are in the frequency of second order harmonics of engine. Vehicle speed was 30 m/s, IV gear ratio and n_e of crankshaft was 68.932 s^{-1}. Analyzing results of optimization, it can be concluded that the lowest objective function defines the sought minimum of the object function.

Finding, the global minimum of the objective function is solved by using a large number of combinations of the initial values of the PT suspension parameters.

Finally, optimization of the PT suspension is by three hydraulic mounts of which elasto-damping characteristics and positions are defined with respect to the adopted coordinate systems given previously.

The method described gave suggestions for redesigning the PT suspension system. By this method we can obtain the nearest minimum of the objective function. It depends on initial values of optimization parameter, velocity, excitation, regime etc. It means that the vehicle would have a different behaviour in each situation, but on the other hand suspensions of PT and vehicle are suboptimal. The ideal solution would be found by using an active suspension system of the PT and vehicle, whose characteristics would compensate all differences of suggested suspension systems of PT and vehicle.

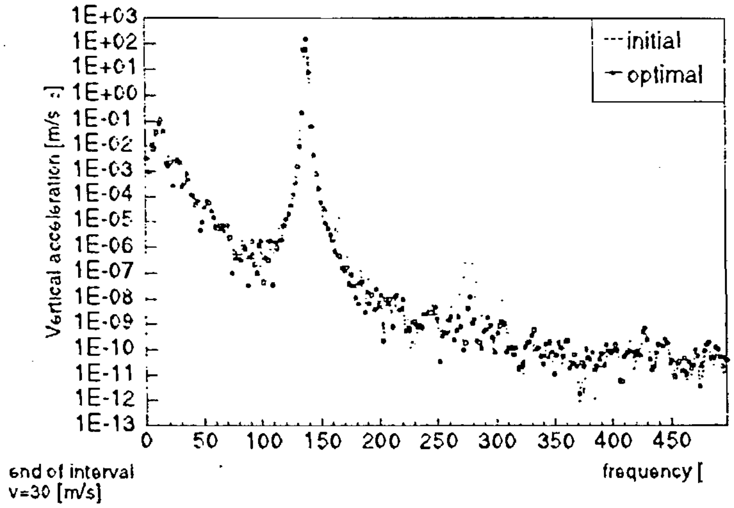

Figure 5. Results of optimization

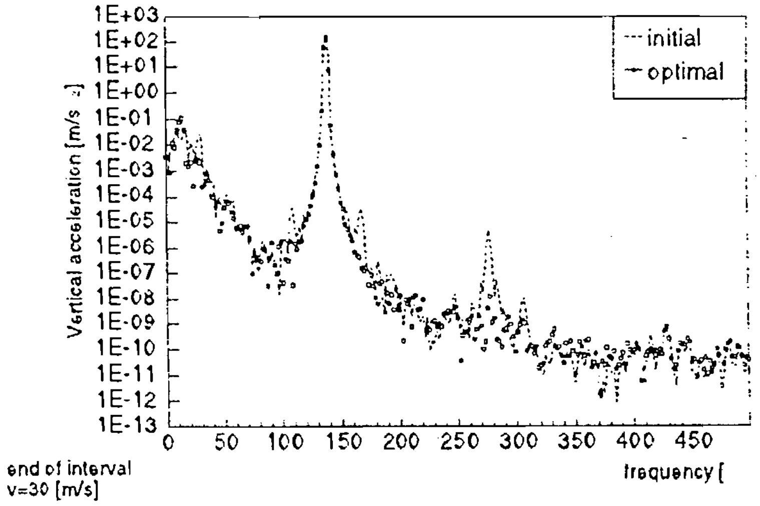

Figure 6. Results of optimization

7. CONCLUSIONS

- Adopted dynamic model of PT and vehicle is advocated as shown by its experimental results. So it could be used for forming an analytical optimization method.
- Results show that the existing PT suspension system needs redesigning. It is necessary to construct suspension with three hydraulic mounts, which give a reduction of vibration and noise in the whole vehicle.
- The best solution of the considered problem is an active suspension system of PT and vehicle.

8. ACKNOWLEDGEMENT

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