THE ARCHIVES OF TRANSPORT

VOL. XXIII NO 2 2011

10.2478/v10174-011-0016-7

The Influence of Damping Changes on Vertical Dynamic Loads of Wheel – Experimental Investigations

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Received March 2011

Abstract

In this paper the results of tests of vertical dynamic wheel loads due to changes in suspension damping level are presented. The changes in damping level were performed using semi-active shock absorbers with by-pass valve. The article presents suspension goals, the definition of dynamic loads of vehicle wheel, testing methods with the information on the used apparatus and sensors. In the end the results are presented, interpreted and compared with other researchers' results.

Keywords: vehicle dynamics, suspension dynamics, dynamic wheel load, semi-active suspension, EUSAMA test, adjustable dampers

1. Introduction

Car suspension as a dynamic system limited only to vertical motions can be considered a dynamic system with two degrees of freedom. This concerns the suspension of one corner of a vehicle and such a model is often called a quarter car model, which formally is a two-degrees-of-freedom model with two masses. One – the upper and bigger mass – represents the part of vehicle body with possible load supported by the suspension, while the other mass represents the wheel and

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elements moving together with the wheel (a brake, some masses of suspension links, tie-rods, shock absorber, spring). Each mass can move separately, hence two degrees of vertical movement freedom for both masses. The first mass is often called sprung mass and the other one is unsprung mass.

During suspension dynamic analysis transmissibility functions are considered. Those functions show the relation between road excitation inputs and selected output variable of suspension dynamics. They can be used to predict a dynamic response of the suspension while driving on a road with specified excitations.

When analyzing the full vehicle vibration model (four wheels model – seven-degree-of-freedom) also response to inertial forces caused by longitudinal and lateral dynamics can be considered. These two types of excitations – road unevenness and inertial forces – are present in everyday vehicle operation.

Two main tasks that vehicle suspension has to manage are vibration comfort and good handling (Rajamani 2006, Wong 2001). Good handling depends on tractive force potential and small roll and pitch angles causing wheel-ground contact angle changes.

When analyzing the quarter car model the following signals can be considered for performance analysis:

– the vertical acceleration of the sprung mass to evaluate passengers comfort,
– tire deflection or dynamic tire (wheel) load to evaluate the road-holding.

When the vehicle sits statically on the level ground, the static load F_{stat} on the tire results from weight distribution on front and rear axles. While traveling over an uneven road dynamic variations in tire load can be observed. Static tire load because of suspension travel causing changes in spring and damping forces of suspension is modified by dynamic tire loads F_{dyn}. Dynamic tire load F_{dyn} can be defined as a difference between the static tire load F_{stat} and temporary wheel load resulting from dynamics situation. That means that when a car is sitting on the level ground, dynamic tire load F_{dyn} equals zero and during suspension travel the dynamic tire load F_{dyn} can be negative or positive. When the value of negative dynamic wheel load equals the static wheel load value ($F_{dyn} = -F_{stat}$), the tire can lose contact with the road. Such a situation means that there is no possibility to produce any longitudinal or lateral tire forces – Figure 1.

2. Methods of Evaluating the Dynamic Wheel Load During Laboratory Tests

A direct measurement tire – pavement normal forces during vehicle drive – is practically impossible (the only possibility is to use sensors mounted in pavement [1]). But those forces can be evaluated by performing measurements of forces in other components of the suspension or wheel [2]. An example is a 6-component

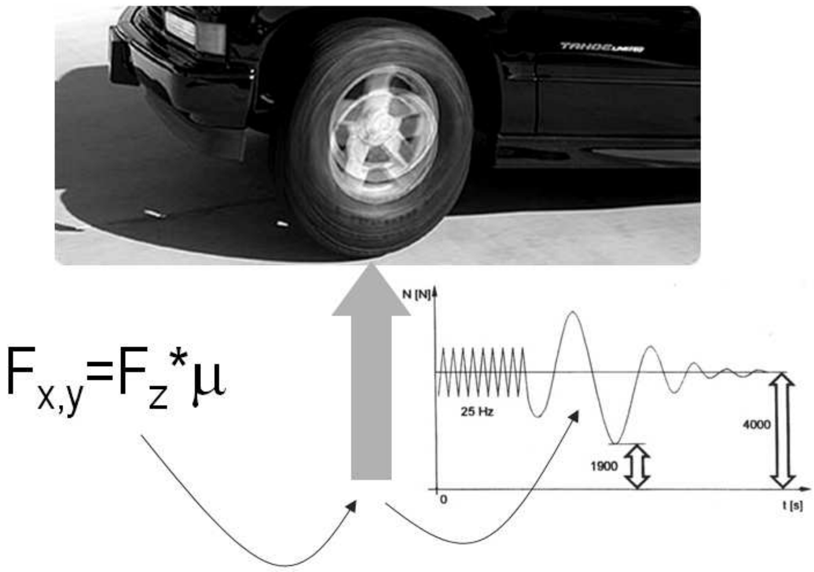

Fig. 1. Wheel load fluctuation and its influence of longitudinal an lateral tire forces

strain gage measuring wheel presented in Figure 2 [3]. But this kind of sensor is very expensive.

	Measured value	Unit	Measurement range
Force	Force F_x	kN	-20...20
	Force F_y	kN	-15...15
	Force F_z	kN	-20...20
Torque	Torque M_x	kN·m	-4...4
	Torque M_y	kN·m	-4...4
	Torque M_z	kN·m	-4...4
Rim size	-	inch	14...19

Fig. 2. An example of 6-component strain gage measuring a wheel for passenger vehicles, light weight carbon fiber design [1]

The other method used to estimate dynamic wheel load is the one used during Periodic Motor Vehicle Inspections suspension tests. This method is used for determining shock absorber performance through a vibrating platform test bench, as recommended by the European Shock Absorbers Manufacturer Association – EUSAMA (1976).

In this method the platform with wheel placed on it vibrates with a constant amplitude and at a variable frequency from 25 to 0 Hz. The force between the wheel and the platform is measured by a force sensor placed on the platform during the frequency decreasing phase. According to the test results, the minimum measured dynamic force is compared with the static force (weight of the vehicle quarter), and the efficacy of the system can be defined as [4]:

$$EU = [(F_{min=}F_{stat} - -F_{dyn})/F_{stat}] * 100\%$$

This minimum wheel vertical force value F_{min} is measured for unsprung mass resonance frequency range.

It is also possible to estimate the level of dynamic wheel loads F_{dyn_max} or its proportion to static wheel load F_{dyn_max}/F_{stat}.

There is also a relation between dynamic wheel loads and tire deflection or between spring deflection used to produce force excitation in BOGE method for evaluating dampers wear level. These relations can be used for estimation dynamic wheel load.

3. Test Bench

When testing shock absorbers, according to Recommendation TS-02-76 [5] issued by EUSAMA a special test bench is used.

During the test wheel of tested vehicle corner suspension rests on a platform of the unit which induces force (vibration exciter). At the beginning of test the static measurement of F_{stat} is performed and the EU value is 100%. After that the force inducing system is actuated. The platform attains vibrations of amplitude ± 2...4 mm at the frequency of about 25 Hz (the used test bench had ± 3 mm amplitude and 24,45 Hz frequency). After the shut-down of the drive system at a frequency of unsprung mass resonance the minimum dynamic tire-to-platform contact force (F_{min}) is measured [6]. The amplitude of platform vibration is the same all the time (only at the end it is not full), the frequency decreases with time corresponding to the inertia of rotating masses of test stand and masses of tested suspension and of course to suspension damping efficiency.

There is a force transducer mounted between platform supporting wheel and exciting mechanism (Fig. 3). Using electronic measurement and signal processing system it is possible to obtain values of dynamic tire-to-supporting platform contact force F_{min}.

Test bench diagram is presented in Figure 3a and photography of test bench without supporting platform (disassembled to make a picture) is presented in Figure 3b. The electric motor, flywheel and shaft with a support bearing are presented. The shaft is connected with eccentrically mounted supports of platform supporting vehicle wheel. Supports of a platform are guided to make the only vertical motion with amplitude of ± 2...4 mm.

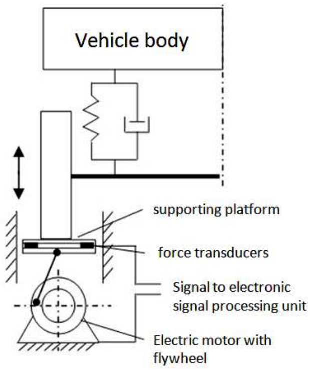

Fig. 3. Test bench working with EUSAMA principle: a) diagram, b) photography of partially disassembled test bench – from the bottom – view of electric motor, flywheel and shaft with a support bearing and with force transducers

The platform supporting wheel is made of aluminum to minimize the inertial forces of excitation system with its mass. A recorded displacement of platform during one full test procedure are presented in Figure 4.

At the end of platform vertical displacement signal time history it is visible that not full amplitude was realized. It was because the kinetic energy of flywheel was already too small to lift the platform with a wheel acting on it with static weight of one corner of a vehicle. This limitation makes it impossible to test suspension at frequencies of body resonance. The whole test is based on unsprung mass resonance frequency analysis.

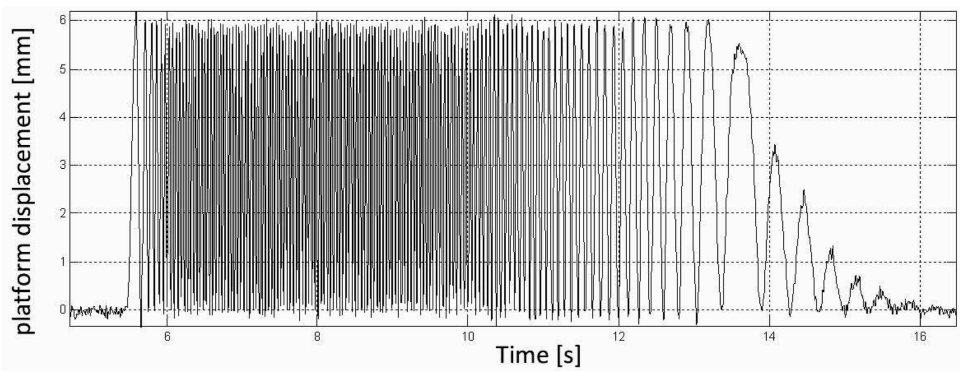

Fig. 4. Time history of platform vertical displacement during one full test according to UESAMA test procedure

4. Tests Results

Tests were performed for rear axle suspension (twist beam) of a passenger car. Its body weight was increased by the weight of measurement equipment and two batteries. In the second test case additionally the weight of three passengers sitting on rear seats was added.

Static wheel loads – tire-platform vertical contact forces were for the cases 3090 N and 3983 N respectively.

Tests were performed for 8 different levels of control current starting from 0 A value, corresponding to the largest damping and with a sequence of next increasing values – 0,43 A; 0,62 A; 0,8 A; 1,0 A; 1,2 A , 1,4 A i 1,6 A (smallest damping). During every test EUSAMA EU indicator and the value of resonance frequency of unsprung mass were recorded.

The results of all the tests after the calculation of average value were collected and are shown in the Table 1. As the characteristics of shock absorber is nonlinear, the values of damping ratio γ [8] were calculated for three different suspension deflection speeds. A nonlinearity of damper's characteristics occurs in higher values of the damping coefficient and the damping ratio for low values of damper deflection speed and lower values for higher speeds. The values for low velocities are almost two times higher than for higher velocities.

Table 1

Results of tests performed with test bench working with EUSAMA principle

Mass of one axle of vehicle	Sprung mass of one corner of vehicle	Control current of by-pass valve	$\gamma-$ damping ratio			Verbal descri-ption of damping level	EUSAMA Indicator: minimum (F_{dyn}/F_{stat})	Resonance frequency of unsprung mass
			γ for 0,2 m/s	γ for 0,5 m/s	γ for 0,8 m/s			
kg	kg	A	–	–	–		–	Hz
I case of vehicle load								
635	282.5	0	0.7	0.49	0.37	“hard”	0.70	19
630	280	0.43	0.57	0.45	0.35		0.64	16
635	282.5	0.62	b.d.	b.d.	b.d		0.62	16
625	277.5	0.8	0.4	0.35	0.27	“medium”	0.60	15.5
635	282.5	1.01	b.d	b.d	b.d.		0.58	16
635	282.5	1.21	0.35	0.27	0.22		0.56	16
640	285	1.4	0.28	0.24	0.19		0.53	15
625	277.5	1.6	0.25	0.18	0.16	“soft”	0.51	15
II case of vehicle load								
815	372.5	0	0.56	0.38	0.3	“hard”	0.64	16
810	370	0.8	0.31	0.26	0.2	“medium”	0.55	15
815	372.5	1.6	0.21	0.16	0.13	“soft”	0.46	14,5

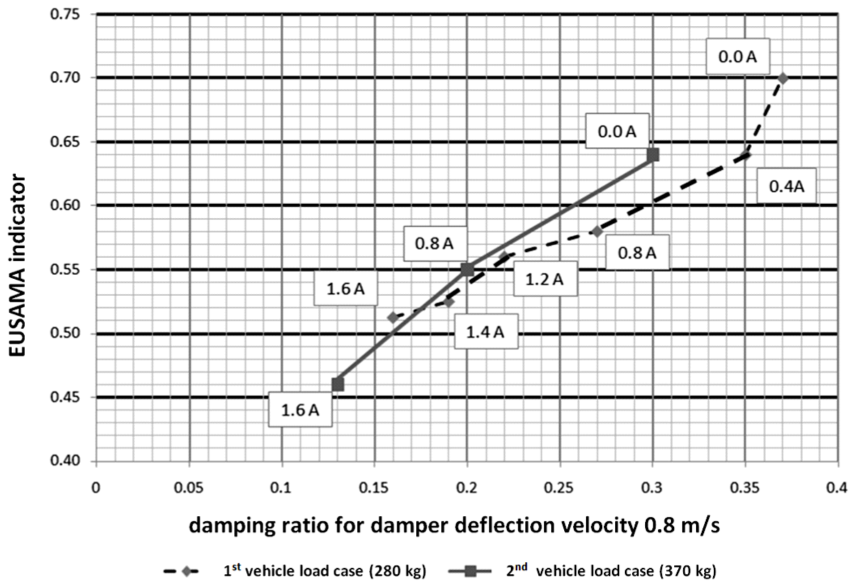

Fig. 5. Relation between damping ratio and EUSAMA indicator with variations of control current level and vehicle load

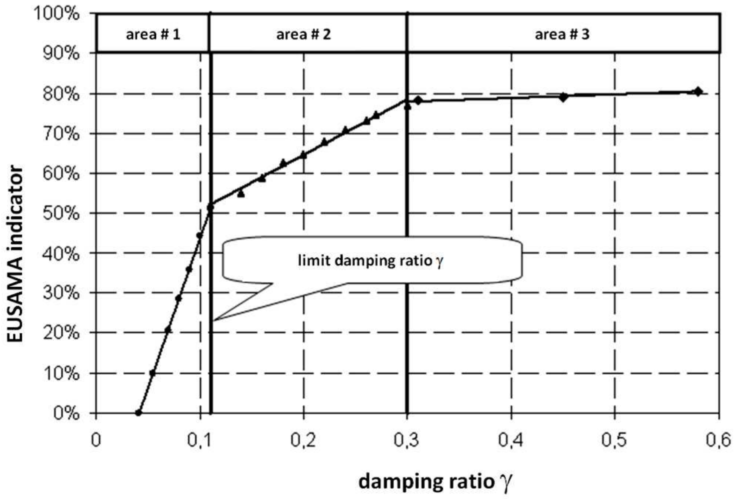

Fig. 6. Changes of EUSAMA indicator caused by damping ratio changes due to shock absorber wear [2]

In the paper [10] damper characteristic was analyzed in detail to take into consideration the influence of changes in control current of shock absorber by-pass valve, deflection speed and vehicle mass on damping ratio values. It was stated that although the shock absorber has adjustable damping force, the force changes are within the limits of usual values of damping ratios in vehicle suspensions (0.2–0.4) with 0.2 value possibly optimal for comfort and 0.4 possibly optimal for handling (depends on particular car and situation) [10].

Tests made with more loaded vehicle (II load case) yielded results similar to less loaded vehicle (I load case) but with a lower damping ratio. It is caused by the influence of sprung mass on a damping ratio and its influence on suspension dynamics.

The obtained test results coincide with the results presented by other researchers testing the influence of damping ratio decrease caused by shock absorber wear on values of EUSAMA indicator [4]. Similar relation is presented in other papers [7].

Generally all these results show variations of conditions of suspension functioning due to changes of vehicle operation conditions. The results confirm earlier results obtained during the study of the influence of vehicle load variations on actual value of damping ratio and its influence on changes in transmissibility's functions in area of comfort and safety [9].

The possibility of measurement of tire deflection with the use of additional laser displacement sensor mounted on a wheel enabled to estimate the relation between tire deflection and wheel dynamic load. Analysis of obtained results allowed to estimate tire vertical stiffness. The results of measurements and estimation are presented in Table 2.

Table 2

Tire deflections and estimated tire stiffness values

Control current of by-pass valve	Wheel displacement min [mm]	Wheel displacement max [mm]	travel [mm]	amplitude [mm]	EUSAMA indicator	F_{dyn}/F_{stat}	F_{dyn} [N]	Tire stiffness [kN/m]
0 a	-5.18	4.46	9.64	4.82	0.7	0.3	927	207
0.8 a	-6.04	5.26	11.3	5.65	0.6	0.4	1236	235
1.6 a	-6.95	6.25	13.2	6.6	0.51	0.49	1514	242

5. Conclusions

The presented tests of suspension with adjustable dampers and its results provide knowledge about the range of dynamic wheel load variations estimated with use of methodology and apparatus used during Periodic Motor Vehicle Inspections.

It was stated that the available range of damping ratio variation remains within the range of commonly used values of damping ratios from comfortably to sports tuned suspensions.

Applying shock absorbers with adjustable damping allowed to investigate the influence of changes of damping ratio on EUSAMA indicator with exactly the same vehicle configuration, its suspension condition and vehicle placement on test bench, which is not so obvious during tests performed with a new set of tested shock absorbers every time due to the necessity of suspension disassembly.

The results confirm that tests used to evaluate shock absorber condition during Periodic Motor Vehicle Inspections are not precise methods due to rather a wide distribution of possible vehicle suspension parameters even for similar vehicles.

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