IDENTIFICATION OF CAR SUSPENSION SYSTEM PARAMETERS ON THE BASIS OF EXPLOITATIONAL MEASUREMENTS

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Summary

The paper concerns exploitational identification of structural parameters of the Toyota Camry suspension system. Analysis was carried out by means of the output-only nonlinear system identification method for vibration accelerations recorded along the centre line of absorber while driving over a bumpy road profile. As the measure of accuracy of estimated suspension parameters the percentage relative error of car body mass estimation was assumed. Presented methodology can be used for the purposes of model based diagnostics and maintenance planning.

Keywords: exploitational (output-only) identification, nonlinear system, car suspension system.

EKSPLOATACYJNA IDENTYFIKACJA PARAMETRÓW UKŁADU ZAWIESZENIA SAMOCHODU OSOBOWEGO

Streszczenie

W pracy przedstawiono rezultaty eksploatacyjnej identyfikacji parametrów układu zawieszenia samochodu osobowego Toyota Camry. Badania przeprowadzono z zastosowaniem eksploatacyjnej metody identyfikacji układów nieliniowych na podstawie charakterystyk czasowych przyspieszeń drgań zmierzonych wzdłuż osi amortyzatora podczas jazdy po wyboistej nawierzchni. Jako miarę dokładności estymacji parametrów rozpatrywanego układu zawieszenia przyjęto procentowy błąd względny estymacji masy karoserii. Zaprezentowana w pracy metoda badawcza znajduje zastosowanie w diagnostyce realizowanej w oparciu o model układu nieuszkodzonego oraz w planowaniu przeglądów.

Słowa kluczowe: identyfikacja w warunkach eksploatacyjnych, układ nieliniowy, układ zawieszenia samochodu osobowego.

1. INTRODUCTION

Vast majority of real mechanical systems is nonlinear to a certain degree. Nonlinear industrial systems can be classified as systems designed for work in nonlinear ranges of dynamic characteristics and systems of nonlinear properties resulting from fault appearance. Although the sources of nonlinear properties can vary [1, 2, 3, 4], all the nonlinear systems exhibit some common properties. In general, they do not follow the superposition principle and exhibit complex phenomena unusual for linear systems, such as jumps, self-induced and chaotic vibrations, changes in natural frequencies resulting from changes in excitation amplitudes and coexistence of many stable equilibrium positions. In view of these properties, classical identification methods can not be used for the purposes of nonlinear system identification.

For many years linearization methods [5, 6] were the only methods used for the purposes of nonlinear system identification. In the following years the concept of nonlinear normal modes was introduced [7, 8] while for weakly nonlinear systems perturbation theory was developed [9, 10]. Recently, mechanical systems are designed for work in nonlinear ranges of dynamic characteristics taking advantage of phenomena characteristic for nonlinear systems [11, 12].

The first research into nonlinear system identification methods goes back to the seventies [13, 14]. Then identification of single degree-of-freedom systems with various types of nonlinearities was considered. Multiple degree-of-freedom identification methods have been elaborated over last 15 years.

The methods considered above are called 'classical' nonlinear system identification methods and are usually classified into a few basic groups: linearization methods, time domain methods, frequency domain methods, modal methods, time–frequency analysis methods, methods based on neural networks, wavelet transform methods, structural model updating. Unfortunately, the range of practical applications of theses methods is limited due to strong requirements concerning linear system behaviour around any operating point and necessity of performing input (excitation) measurement.

In the paper, for the purposes of identification of car suspension parameters, application of the exploitational nonlinear system identification method was proposed. Selection of such a method is motivated by the fact that, on the contrary to the classical nonlinear system identification methods, it requires neither the knowledge of excitations acting on the system of interest nor linear system dynamic behaviour in a broad frequency range around any operating point [15, 16]. The possibility of system identification in the exploitational conditions is of key importance since the measurement of operational exciting forces (e.g. tire – road contact forces) is difficult or impossible to carry out and diagnostics of many industrial systems, for the economical or technological reasons, should be carried out during the normal work.

2. ASSUMED IDENTIFICATION METHOD

Identification of the considered suspension system parameters was performed by means of the nonlinear system identification method [15, 17], the algorithm of which combines restoring force, boundary perturbation and direct parameter estimation techniques (Fig. 1). Since the method requires neither input measurement nor linear system behaviour around an operating point, it can be used for both nonlinearity detection and parameter identification of nonlinear systems working under operational loads, the measurement of which is difficult or impossible to carry out.

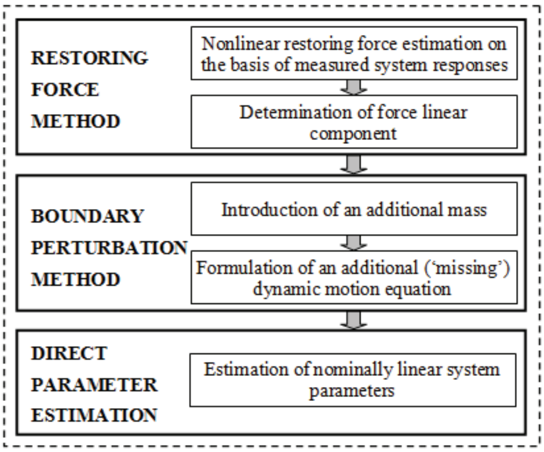

Fig. 1. Algorithm of the applied exploitational nonlinear system identification method

In the first step of the considered algorithm, system dynamic motion equations are formulated and nonlinear restoring forces are reconstructed on the basis of measured system responses. In the following step, identified nonlinear restoring forces are eliminated from the formulated dynamic motion equations that describe balance of forces acting on the considered system. Since the excitation remains unknown, the number of unknown parameters is greater than the number of system dynamic motion equations that can be formulated. In order to provide an additional dynamic motion equation and determine absolute values of system parameters the boundary perturbation method is used. The method consists in introducing an additional mass altering dynamic behaviour of the considered system and retaking measurements of modified system masses accelerations. In the final step values of system parameters are estimated.

3. IDENTIFICATION OF CAR SUSPENSION PARAMETERS

The research was carried out for the rear left suspension system of Toyota Camry, version 2.2 LE, 2001 (Fig. 2).

In the course of the identification experiment, the measurements of acceleration time histories were taken by means of two single-axial piezoelectric sensors placed on the spindle (sensor 1, 'lower', Fig. 3a) and the upper strut connection with the body (sensor 2, 'upper', Fig. 3b). Vibration acceleration time histories measured along the centre line of the absorber were registered with the use of the NI USB-6009 Multifunction Data Acquisition Device (14-bit, 48 kS/s).

Fig. 2. Considered car: Toyota Camry 2.2 LE (2001)

Measurements were taken in two sessions. In the second session, in order to alter system dynamic behaviour, back of the car was loaded with an additional mass $\Delta M_2 \approx 56{,}25$ [kg] by filling up 35 litres of petrol and 30 litres of LPG. Value of ΔM_2 was calculated under assumption that 1 litre of petrol weights about 0,75 [kg] while 1 litre of LPG about 1 [kg]. Example acceleration time histories measured in the considered measurement points are presented in the Fig. 4.

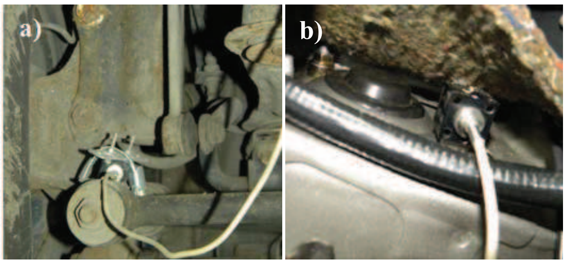

Fig. 3. Location of a) sensor 1 ('lower'), b) sensor 2 ('upper')

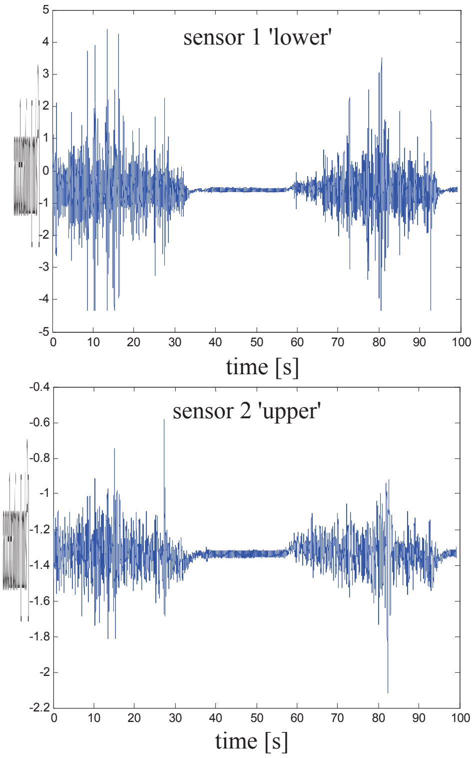

Fig. 4. Example acceleration time histories measured during the second session (for the body mass $M_2 + \Delta M_2$)

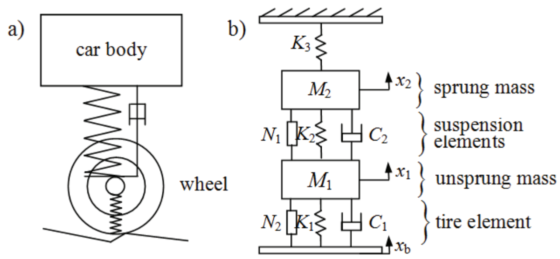

Fig. 5. Assumed quarter car model [17, 18], M_1: sprung mass, M_2: unsprung mass, K_1, K_2: stiffness coefficient of tire and suspension, C_1, C_2: damping coefficient of tire and suspension, x_1, x_2, x_b: displacements of M_1, M_2 and tire patch, N_1, N_2: nonlinear forces

For the purposes of the considered suspension system identification, the quarter car model presented in the Fig. 5 was assumed [17, 18].

Dynamic motion equations formulated for the M_1 and M_2 masses are as follows:

$$\begin{cases} M_1\{\ddot{x}_1\} + (C_1 + C_2)\{\dot{x}_1\} + (K_1 + K_2)\{x_1\} - \\ - C_2\{\dot{x}_2\} - K_2\{x_2\} + N_1 + N_2 = C_1\{\dot{x}_b\} + K_1\{x_b\} \\ M_2\{\ddot{x}_2\} - C_2\{\dot{x}_1\} + C_2\{\dot{x}_2\} - K_2\{x_1\} + \\ + (K_2 + K_3)\{x_2\} = N_1 \\ N_1 = N_1(\{x_1(t)\}, \{x_2(t)\}, \{\dot{x}_1(t)\}, \{\dot{x}_2(t)\}) \\ N_2 = N_2(\{x_1(t)\}, \{x_b(t)\}, \{\dot{x}_1(t)\}, \{\dot{x}_b(t)\}) \end{cases} \tag{1}$$

For the considered suspension system, with the application of the direct parameter estimation technique, the following equations were formulated:

$$K_2\left(1 - \frac{1}{T_{21}(\omega_k)}\right) + K_3 = \omega_k^2 M_2 \tag{2}$$

and:

$$T_{21}(0) = \frac{K_2}{K_2 + K_3} \tag{3}$$

where: $\{T_{21}(j\omega)\} = X_2(j\omega) / X_1(j\omega)$: transmissibility function between the sprung mass M_2 and unsprung mass M_1 (with nonlinear restoring forces subtracted), $X_1(j\omega)$, $X_2(j\omega)$: Fourier transforms of signals $x_1(t)$ and $x_2(t)$, $k = 1, 2, ..., N_f$, N_f: number of spectral lines of useful data, $T_{21}(0)$: transmissibility function evaluated at zero frequency.

Since in practice the exact value of mass M_2 remains unknown, direct parameter estimation method provides two equations with three unknowns - M_2, K_2 and K_3. Therefore, at this stage of the analysis, only ratios of parameters (with respect to the mass M_2) are available.

In order to provide an additional dynamic motion equation, boundary perturbation method was used. The method consists in introducing an additional mass ΔM_2 altering dynamic behaviour of the considered system (linear system natural frequencies) and retaking measurements for such a modified system. An additional equation is as follows:

$$K_2\left(1 - \frac{1}{T_{21}^{'}(\omega_p)}\right) + K_3 = \omega_p^2 M_2 \tag{4}$$

where: $\{T_{21}'(j\omega)\} = X_2'(j\omega) / X_1'(j\omega)$: transmissibility function between the sprung mass $(M_2+\Delta M_2)$ and unsprung mass M_1, $p = 1, 2, ..., N_f'$, N_f': number of spectral lines of useful data.

Since at this stage three equations with three unknowns can be formulated, it is possible to estimate the absolute values of demanded parameters.

In the next step of analysis, on the basis of carried out measurements of system responses to exploitational excitation, transmissibility functions of the initial and modified system were estimated ($\{T_{21}\}$ and $\{T_{21}'\}$, respectively). Comparison of estimated functions $\{T_{21}\}$ and $\{T_{21}'\}$ is presented in the Fig. 6. As the result of system modification, resonant frequencies of transmissibility function were shifted towards lower values.

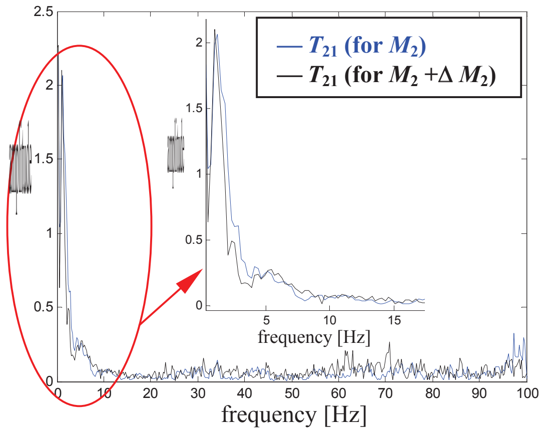

Fig. 6. Comparison of estimated trans-missibility functions $\{T_{21}\}$ (for M_2) and $\{T_{21}'\}$ (for $M_2 + \Delta M_2$)

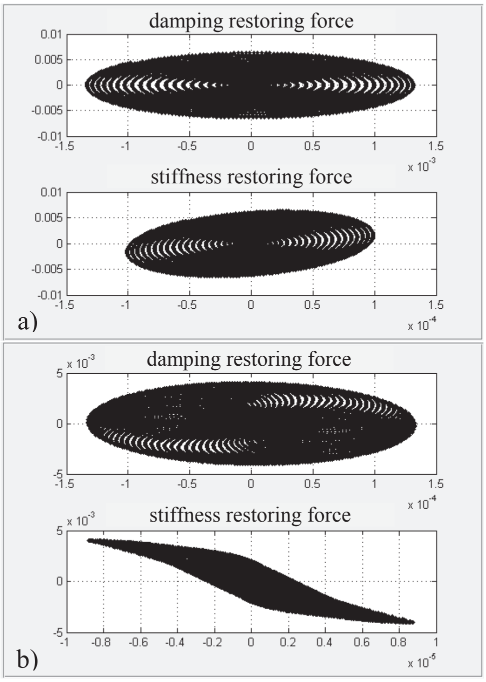

Fig. 7. Damping and stiffness restoring forces identified for frequencies a) f = 0,7813 [Hz], b) f = 4,6875 [Hz]

In order to verify linearity of the considered system, dynamic motion equation formulated for the sprung mass M_2 was written in the form:

$$M_2\{\ddot{x}_2\} = -C_2(\{\dot{x}_2\} - \{\dot{x}_1\}) - K_2(\{x_2\} - \{x_1\}) - \\ - K_3\{x_2\} + N_1(\{x_1(t)\}, \{x_2(t)\}, \{\dot{x}_1(t)\}, \{\dot{x}_2(t)\}) \quad (5)$$

expressing relation between acceleration of the sprung mass M_2 and relative velocity or relative displacement between masses M_1 and M_2. On the basis of measured vibration accelerations time histories, by means of time-domain integration, vibration velocities and displacements in the considered measurement pointes were determined. Before each integration, constant components were removed from the analyzed signals. Selected characteristics of estimated stiffness and damping restoring forces are presented in the Fig. 7.

In the considered frequency bandy, reconstructed damping restoring forces are linear while the vast majority of stiffness restoring forces reconstructed for resonant areas of $\{T_{21}\}$ function display histeretic properties.

Estimation of the considered suspension system parameters was carried out for frequencies and magnitudes of identified 'peaks' of transmissibility functions $\{T_{21}\}$ and $\{T_{21}'\}$ (Table 1) with the application of the created software [19].

Table 1. Parameters of estimated transmissibility functions

	Transmissibility function parameters			
	ω [Hz]	**$T_{21}(\omega)$**	**ω' [Hz]**	**T_{21}'(ω)**
1.	1,3021	2,0654	1,0417	2,0991
2.	2,8646	0,6101	2,3437	0,4953
3.	4,1667	0,2936	3,3854	0,1758
4.	5,2083	0,2629	4,4271	0,2415
5.	-	-	5,4688	0,2778
6.	6,5104	0,1798	5,9896	0,2432
7.	8,5937	0,0718	7,0313	0,1687
8.	9,3750	0,0776	8,0729	0,1163
9.	10,1562	0,0663	9,6354	0,0820
10.	10,6770	0,0807	10,4166	0,0794
11.	11,1979	0,0775	10,9375	0,0834
12.	-	-	11,7187	0,0765
13.	13,0208	0,0429	12,2395	0,0705
14.	-	-	13,2812	0,0744
15.	14,8437	0,0433	14,0625	0,0589

Since the magnitude of the estimated transmissibility function $\{T_{21}\}$ approaches 2,25 as the frequency approaches 0 [Hz], on the basis of the equation (3), the following relation between K_2 and K_3 is obtained:

$$K_3 = -0{,}6 \cdot K_2 \quad (6)$$

According to the owners manual [20], mass of the considered car body:

$$M_{car\ body}^{theoretical} = 1380\,[kg] \quad (7)$$

In the course of the carried out identification experiments, the car was loaded with an additional mass of equipment, fuel and passengers of overall value 230 [kg]. Therefore it was assumed that total mass of the considered car body during measurements amounted to:

$$M_{car\ body}^{total(real)} = 1610\,[kg] \quad (8)$$

Taking into account mass distribution (0,59 front axle and 0,41 rear axle), it was assumed that mass concentrated on the considered rare left suspension system [19]:

$$M_2^r = 0{,}5 \cdot 0{,}41 \cdot M_{car\ body}^{total(real)} \left[kg\right] \quad (9)$$

therefore:

$$M_2^r = \frac{0{,}41}{2} \cdot 1610 = 330\ [kg] \quad (10)$$

Mass value estimated with the application of the exploitational nonlinear system identification method equals:

$$M_2^e = 326\ [kg] \quad (11)$$

As the measure of parameter estimation accuracy, the relative percentage errors of M_2 estimation was assumed:

$$Err = \frac{\left|M_2^r - M_2^e\right|}{M_2^r} \cdot 100 \cdot [\%] \quad (12)$$

where M_2^e denotes estimated mass, M_2^r real mass value.

Therefore, on the basis of relation (12), it can be stated that estimated value of mass M_2^e is burdened with the percentage relative error:

$$Err = \frac{|330 - 326|}{330} \cdot 100 \cdot [\%] = 1{,}21\,[\%] \quad (13)$$

Estimated value of the considered suspension system stiffness coefficient equals:

$$K_2^e = 2610\ \text{[N/m]} \quad (14)$$

while the value of stiffness coefficient K_2^z recommended by the producer of spare parts amounts to:

$$K_2^z = 3700\ \text{[N/m]} \quad (15)$$

Estimated value K_2^e of stiffness coefficient is significantly lower than the recommend value K_2^z.

4. CONCLUSIONS AND FINAL REMARKS

The paper concerns identification of structural parameters of car suspension system carried out on the basis of vibration accelerations recorded along the centre line of absorber while driving over the bumpy road profile. Analysis was carried out with the application of the output-only nonlinear system identification method. Accuracy of the estimated suspension parameters was assessed on the basis of the relative percentage errors of the considered car body mass estimation. Obtained results can be influenced by:

- serial connection of shock absorber with elastic elements,
- application of the original part of stiffness coefficient lower than in case of the equivalent (3700 [N/m]) used for tuning up,
- wear of elements,
- clearances, elasticity of connections (e.g. rubber rings),
- estimation of dynamic stiffness coefficient in the whole system operating point instead of static stiffness coefficient,
- higher load (filled trunk),
- inaccurate mounting of sensors (deviation of measurement direction from vertical line, usage of magnet for the 'upper' sensor mounting).

Accuracy of the considered suspension system parameters estimation is influenced by the accuracy of transmissibility function (and parameters of its extrema) estimation. Therefore it is necessary to guarantee high accuracy of sensors fastening (directions of sensor measurement axis and orientation of the coordinate axis should be consistent) while the value of an additional mass ΔM_2 should be reasonably high.

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