Simulink Model to Optimize Cab Suspension

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Abstract. An important feature in designing a truck's cab is the driver's comfort. Through cabin architecture it is intended to optimize its behaviour in all modes of operation of the vehicle.

Experimental studies to improve comfort parameters are laborious, require significant time and costs.

The study proposes and uses a model developed in the Simulink Matlab program, which takes into account the elastic system of the truck and the cabin suspension system. The simple model allows the changement of the input parameters - road parameters - and the variation of the elastic and damping characteristics of the cabin to improve the behaviour and comfort.

The results obtained by simulation are compared with experimental data. Suspension performance was analyzed for two type of road obstacle: a threshold and a step-down.

The study of the optimization of the cabin mounting system was achieved by modifying the values of the elastic characteristics and the damping in the model.

The model is also likely to be developed and used to study the dynamic behaviour of truck structures in different working regimes.

Keywords: Cab comfort · Simulink half-truck model
Suspension optimization

1 Introduction

In the field of freight transport, trucks are increasingly used. Comfort in the truck cab is an objective of continuous interest for truck manufacturers. Cab suspension and chassis suspension are constantly improved to help achieve good vehicle behavior and enhanced driving comfort on all road types.

The optimization of truck suspension system parameters can be done analytically and experimentally. Experimental studies require a great deal of testing, costs and time. An analytical study eliminates some of these inconveniences.

The specialized literature presents a wealth of analytical models, from the simplest to the most complex, with good results in finding, optimizing and improving the truck's suspension solutions: the quarter car model [8], the half-car model [9, 10] and the full (complex space) model [2–4, 8, 10]. It is not indicated to model the vertical truck dynamics using only one quarter vehicle model. The half-car and the complex models take into account the suspension elements' positioning and their characteristics. In the

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process of improving the drive comfort, the most important function of the truck model is to describe the motion of the cabin caused by road irregularities and other disturbances.

The paper aims at studying a tractor's behavior by measuring the parameters of the vertical displacement of the wheels, chassis and cabin and of the chassis and chassis pitch movement using a 6-degree plan model.

The model is made in the Simulink module in Matlab and takes into account the unsprung and sprung masses and the elasto-damping systems of the tire, chassis and cab [1–3]. Tires were considered elastomer-damping systems characterized by stiffness and damping coefficients [10]. The front and rear suspension systems for the chassis and cabin have been modeled as passive spring and shock absorber systems [3, 5, 6]. A step type signal is entered for modeling and simulating the road irregularity [3]. The parameter values entered in the model are similar to those used in such studies [7, 10].

In earlier works, the authors developed an FEM model for studying and analyzing the dynamic behavior of the truck structure [11]. The FEM model required much work and time to generate results.

The 6-degree plan model presented in this work has saved time, delivering results that are close to experimental ones.

2 Mathematical Model

The half truck model with 6 degree of freedom (DOF), as considered in this paper, is shown in Fig. 1. It takes into account the vertical motion of the front and rear wheels, of the chassis and cab and of the chassis and cabin pitch motion. Tires, chassis and cab systems are considered as spring and damper system.

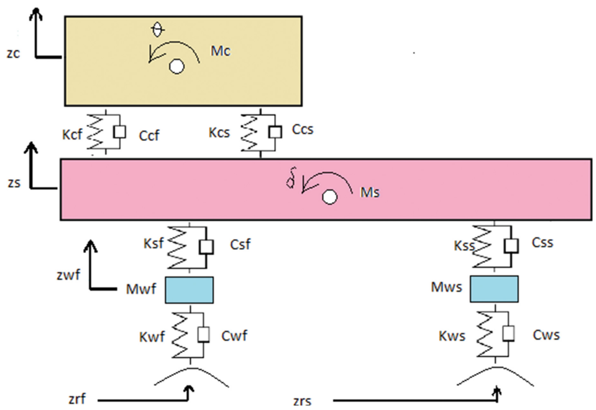

Fig. 1. The mathematical half truck model.

The mass of cabin and chassis (sprung mass) are noted by Mc and Ms while the mass of front/rear wheels and axle (un-sprung mass) are noted by Mwf/Mws. The notations zc, zs and zwf/zwr represent the vertical displacement of cabin, chassis and front/rear wheel respectively. By θ and δ are noted the pitch of cabin and chassis respectively. The notations Kcf, Kcs, and Ccf, Ccs, denote the stiffness and damper parameters of the cab, while Ksf, Kss and Csf, Css denote the stiffness and damper parameters of the chassis. For wheels, Kwf, Kws and Cwf, Cws represent their stiffness and dampers parameters. Iyyc and Iyys denote the inertia moment for the cabin and chassis respectively about y axe of the tires. The notation r and p denote the front and rear cab suspension elements' distance from the centre of gravity of the cabine, while s and d represent the distances of the front and rear suspension elements of the chassis to the chassis centre of gravity. The longitudinal position of front chassis suspension to the gravity centre of cabin was noted by d. The zrf/zrf was the profile of the treadmill on which the truck is moving.

2.1 Model Properties

The values of parameters described in the previous section and used for modelling are considered around the values of a real truck used in experimental determination. These are presented in Table 1.

Table 1. Parameters of the system.

Model parameters					
No.	Parameter	Value	No.	Parameter	Value
1	Mc	1300 [kg]	13	Kws	$1.1 * 10^6$ [N/m]
2	Ms	2050 [kg]	14	Cwf = Cws	$2 * 10^3$ [Ns/m]
3	Mwf = Mws	500 [kg]	15	Iyyc	1100 [kgm^2]
4	Kcf	$120 * 10^3$ [N/m]	16	Iyys	7000 [kgm^2]
5	Kcs	$80 * 10^3$ [N/m]	17	p	1.04 [m]
6	Ccf	10000 [Ns/m]	15	r	1.05 [m]
7	Ccs	6000 [Ns/m]		s	1.46 [m]
8	Ksf	$17 * 10^4$ [N/m]	16	t d	2.34 [m] 0.15 [m]
9	Kss	$15 * 10^4$ [N/m]	17	v_1	10 km/h
10	Csf	$75 * 10^2$ [Ns/m]	18	v_2	30 km/h
11	Css	$45 * 10^2$ [Ns/m]	19	v_3	50 km/h
12	Kwf	$1.2 * 10^6$ [N/m]Kws			

2.2 Model Properties

Using Newton's law, for the half truck model there have been described the equations of motion (1), (2)…(5) that are presented bellow:

$$\begin{aligned} & Mc * \ddot{z}_c + K_{cf}\left[z_c - \theta * \mathrm{p} - z_s + \delta * (\mathrm{p} + \mathrm{d} + \mathrm{s})\right] \\ & + C_{cf}\left[\dot{z}_c - \dot{\theta} * \mathrm{p} - \dot{z}_s + \dot{\delta} * (\mathrm{p} + \mathrm{d} + \mathrm{s})\right] \\ & + K_{cs}\left[z_c - \theta * \mathrm{r} - z_s + \delta * (\mathrm{s} + \mathrm{d} - \mathrm{r})\right] \\ & + C_{cs}\left[\dot{z}_c - \dot{\theta} * \mathrm{r} - \dot{z}_s + \dot{\delta} * (\mathrm{s} + \mathrm{d} - \mathrm{r})\right] \\ & = 0 \end{aligned} \tag{1}$$

$$\begin{aligned} & Iyyc * \ddot{\theta} - K_{cf}\left[z_c - \theta * \mathrm{p} - z_s + \delta * (\mathrm{p} + \mathrm{d} + \mathrm{s})\right] * p \\ & - C_{cf}\left[\dot{z}_c - \dot{\theta} * \mathrm{p} - \dot{z}_s + \dot{\delta} * (\mathrm{p} + \mathrm{d} + \mathrm{s})\right] * p \\ & - K_{cs}\left[z_c - \theta * \mathrm{r} - z_s + \delta * (\mathrm{s} + \mathrm{d} - \mathrm{r})\right] * r \\ & - C_{cs}\left[\dot{z}_c - \dot{\theta} * \mathrm{r} - \dot{z}_s + \dot{\delta} * (\mathrm{s} + \mathrm{d} - \mathrm{r})\right] * \mathrm{r} = 0 \end{aligned} \tag{2}$$

$$\begin{aligned} & Iyys * \ddot{\delta} + K_{cf}\left[z_c - \theta * \mathrm{p} - z_s + \delta * (\mathrm{p} + \mathrm{d} + \mathrm{s})\right] * s \\ & + C_{cf}\left[\dot{z}_c - \dot{\theta} * \mathrm{p} - \dot{z}_s + \delta * (\mathrm{p} + \mathrm{d} + \mathrm{s})\right] * s \\ & + K_{cs}\left[z_c - \theta * \mathrm{r} - z_s + \delta * (\mathrm{s} + \mathrm{d} - \mathrm{r})\right] * t \\ & + C_{cs}\left[\dot{z}_c - \dot{\theta} * \mathrm{r} - \dot{z}_s + \delta * (\mathrm{s} + \mathrm{d} - \mathrm{r})\right] * t \\ & - K_{sf}(z_s - \delta * \mathrm{s} - z_{wf}) * s - C_{sf}(\dot{z}_s - \delta * \mathrm{s} - \dot{z}_{wf}) * s \\ & + K_{ss}(z_s - \delta * \mathrm{t} - z_{wf}) * t + C_{ss}(\dot{z}_s - \delta * \mathrm{t} - z_{wf}) * t = 0 \end{aligned} \tag{3}$$

$$\begin{aligned} & M_{wf} * \ddot{z}_{wf} - K_{sf}\left(z_s - \delta * \mathrm{s} - z_{wf}\right) - C_{sf}\left(\dot{z}_s - \dot{\delta} * \mathrm{s} - \dot{z}_{wf}\right) + K_{wf}\left(z_{wf} - z_{rf}\right) \\ & + C_{wf}\left(\dot{z}_{wf} - \dot{z}_{rf}\right) = 0 \end{aligned} \tag{4}$$

$$\begin{aligned} & M_{ws} * \ddot{z}_{ws} - K_{ss}\left(z_s + \delta * \mathrm{t} - z_{ws}\right) - C_{ss}\left(\dot{z}_s - \dot{\delta} * \mathrm{t} - \dot{z}_{ws}\right) + K_{ws}\left(z_{ws} - z_{rs}\right) \\ & + C_{wf}\left(\dot{z}_{ws} - \dot{z}_{rs}\right) = 0 \end{aligned} \tag{5}$$

3 Simulink Model

To simulate the vibrational behavior of the truck, a model file was created in the Simulink module of Matlab. It contains the solution of the 2nd order-derived equations system, describing the movement of the component parts of the model shown in Fig. 1. Vertical displacement of the suspended and un-suspended mass and of the pitch motion of the chassis and cabin were taken into account. The Simulink model is shown in Fig. 2.

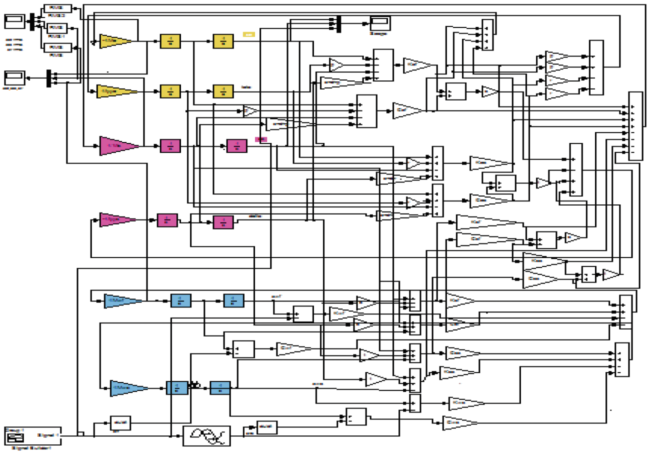

Fig. 2. Simulink model.

A input signal was introduced for modeling the tread deviation and was denoted by zrf and zrs. The height of the unevenness is 20 cm and the tractor's speed is 10 km/h. Thus, by the first second, the front axle of the tractor goes over the irregularity, while the rear axle passes over it after another 1.4 s (Fig. 3).

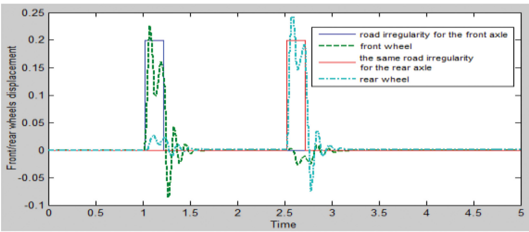

Fig. 3. Front/rear wheel displacement passing over the irregularity.

4 Experimental Determination

Experimental determinations consisted in recording the vertical accelerations of the cabin and chassis of a truck when driving at constant speed on the road.

For the recorded values, the mean square acceleration for the chassis and cab was calculated. These are shown in Fig. 4. The diminished values of the cab accelerator are observed, compared to those of the chassis and of the wheel.

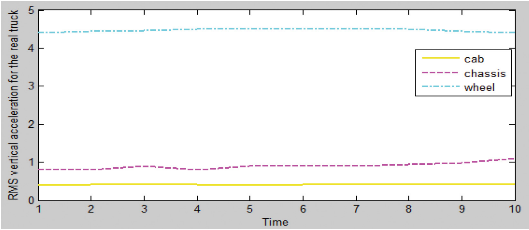

Fig. 4. RMS vertical acceleration of the wheels, chassis and cab vs time, for real truck.

5 Results

The objective of the study was to highlight the efficiency and influence of the cab suspension system on cabin comfort for the Simulink model. Suspension performance was analyzed for two type of road obstacle: a threshold and a step-down (Fig. 5). Vertical acceleration of the front wheel, chassis and truck cabin obtained by simulating the passing over irregularity are shown in Fig. 6.

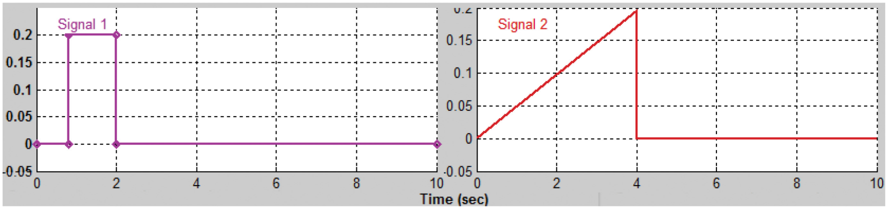

Fig. 5. Type of road obstacle used in the simulation (threshold-signal 1 and step-down-signal 2).

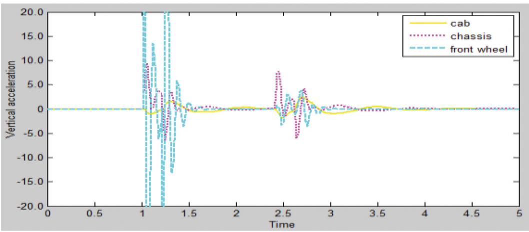

Fig. 6. Vertical acceleration of wheel, chassis and truck cabin vs time, obtained by simulating the passing over irregularity - signal 1 (for the front wheel).

Cab vertical acceleration has lower values than those of the chassis, which have in their turn lower values than those recorded on the wheel. At the same time, a lower maximum amplitude of the cab acceleration is found at passing the rear axle of the truck over the obstacle than in the case of the front axle. In terms of vertical acceleration, its cabin value is lower than that of the chassis. RMS values of vertical acceleration, shown in Fig. 7, for both cabin and chasis, are comparable to those obtained by experimental determinations on the real truck (Fig. 4).

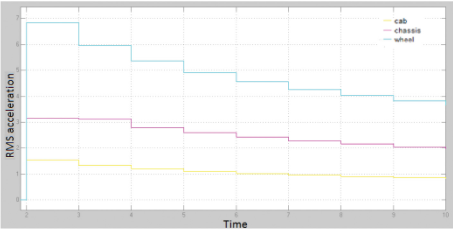

Fig. 7. RMS values of vertical acceleration vs time for the wheel, chassis and cab for the 6 degrees model.

In Figs. 8, 9 and 10 are presented the values for vertical acceleration of the cab, of the chassis and cab pitch acceleration versus time, respectively, to different speeds of the truck (v1, v2, v3) in two case: threshold type obstacle and step-down type obstacle.

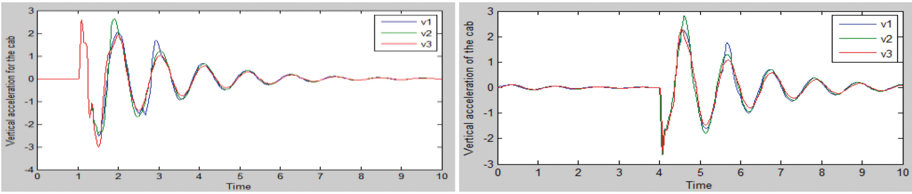

Fig. 8. Cab vertical acceleration vs time for the different speeds of the truck (left-threshold type obstacle; right - step-down type of obstacle).

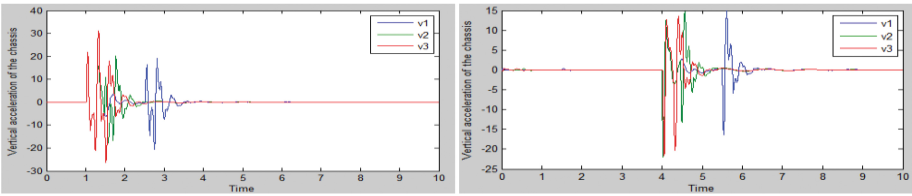

Fig. 9. Chassis vertical acceleration vs time for the different speeds of the truck (left-threshold type obstacle; right - step-down type of obstacle).

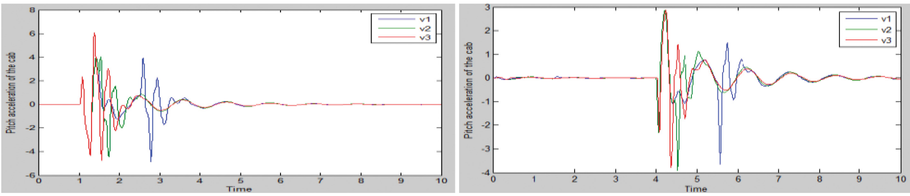

Fig. 10. Pitch cab acceleration vs time for the different speeds of the truck (left-threshold type obstacle; right - step-down type of obstacle).

The study of the optimization of the cabin mounting system was achieved by modifying the values of the elastic characteristics and the damping in the model. The values of vertical acceleration of the cab versus time, are presented in the Fig. 11 for different speeds of the truck. (v1, v2, v3) in the case of the step-down type obstacle for the optimised suspension of the cab.

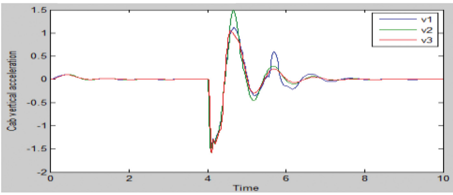

Fig. 11. Cab vertical acceleration vs time for the different speeds of the truck (in the case of step-down type obstacle) for the optimized model.

6 Conclusion

The 6-degree plan model developed in Matlab and used in simulation has compelling results on improving the chassis and cab suspensions of a tractor. The obtained values are close to those obtained in an analogue study using a finite element model. RMS values of vertical acceleration, for both cabin and chasis, are comparable to those obtained by experimental determinations on the real truck. The proposed model is a tool that can easily study the vibrational behavior of the truck in different displacement modes and optimize suspension systems. The Matlab model takes less time and is easier to use to improve driver comfort on the vehicle.

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