STUDY OF DYNAMICS OF CAR SUSPENSION MODEL: DYNAMIC ELIMINATION OF ROAD SURFACE VIBRATION

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Abstract. This paper presents a two-degrees-of-freedom system to describe the dynamics and behaviour of a vehicle driving on a road profile. It represents a simplified approximation of a quarter-car system, where the kinematical excitation is sourced from road roughness profiles. The description is provided using analytical and numerical approaches. Eliminating undesirable vibrations in an automobile is crucial for ensuring ride comfort and passenger safety. One of the most effective solutions is implementing vibration dampers, which play a key role in reducing the transmission of unwanted vibrations to the vehicle's body. Vibration dampers, as components of the suspension system, absorb energy from vibrations generated by road surface irregularities or other dynamic factors such as acceleration, braking, or steering maneuvers. With appropriately tuned damping characteristics, they effectively minimize vibration amplitudes and prevent their transmission to the vehicle body and interior. As a result, dampers not only enhance the vehicle stability and improve the tire-road contact but also reduce noise levels and vibrations perceived by passengers. The use of advanced dampers, such as shock absorbers with adaptive damping characteristics, further enables dynamic adjustment of their performance to current driving conditions. This allows for even more effective elimination of undesirable vibrations, translating into a higher level of comfort and increased durability of the suspension system and other vehicle components. This paper discusses the anti-resonance phenomenon that occurs when appropriate dampers and spring characteristics are introduced into the model. The results demonstrate the advantages of applying such dampers to avoid damaging conditions for the suspension during operation.

Keywords: quarter car, absorption of vibration, suspension system, road surface.

Introduction

The suspension system of a motor vehicle, along with the braking system, is one of the most important systems for ensuring safety [1-3], stability, and travel comfort [4-6]. One of the most effective solutions is the implementation of vibration dampers, which play a key role in reducing the transfer of unwanted vibrations to the vehicle body. Vibration dampers absorb the energy of vibrations generated by road surface unevenness or other dynamic factors resulting from the road situation, such as acceleration, braking, or turning manoeuvres. Gogola [7] and Borowiec et al. [8; 9] conducted studies that analysed the effect of road profile unevenness on the generated vehicle vibrations. Other researchers [10-12] indicate a correlation between the road profile and the energy consumption of the vehicle. The vehicle suspension system is heavily loaded, especially during braking manoeuvres [13; 14]. With appropriately tuned damping characteristics, they effectively minimize vibration amplitudes and prevent their transfer to the vehicle body and interior. As a result, shock absorbers not only increase the vehicle stability and improve the tire-road contact [15-18], but also reduce the level of noise and vibrations felt by passengers [19-21]. The use of advanced shock absorbers, such as shock absorbers with adaptive damping characteristics [22], additionally enables dynamic adjustment of their performance to current driving conditions [8; 10].

As mentioned, vibration damping is very important from the point of view of driving safety as well as travel comfort, but vibrations also constitute significant energy for use in the vehicle. Energy occurring in the vehicle suspension system can be recovered in energy harvesting systems. Based on the literature study, it can be concluded that in addition to thermal energy harvesting from the exhaust system of the combustion engine [23-25], the damping energy from the vehicle suspension reaches significant values [26-30]. Due to the growing interest in the effective flow of energy in the vehicle from various sources, including the harvesting of so-called wasted energy, suspension systems can have a significant contribution to the energy balance of the vehicle. Therefore, research related to the mechanical energy of vibrations occurring in the vehicle suspension systems, their damping and conversion into useful energy is gaining new importance.

This paper presents a quarter-car model designed for numerical analysis of the operation of shock absorbers. performance. This model facilitates simulations of dynamic adjustments to shock absorber operation under varying driving conditions, leading to more effective vibration isolation, enhanced ride comfort, and improved durability of suspension components and other vehicle parts.

Materials and methods

In the paper, a two-degree-of-freedom vehicle model is assumed (Fig. 1a, b). For simplification, the tire damping is neglected, and it is assumed that the vehicle moves on a road at a constant speed v, where the irregularity profile is described by a harmonic function y_0. In the mechanical system, the excitation from the road surface is transmitted to the unsprung mass m_u via a tire with known stiffness k_t, and further through the suspension spring system with stiffness k_s to the sprung mass m_s. The adopted stiffnesses of the system are equivalent stiffnesses of the front suspension and tires.

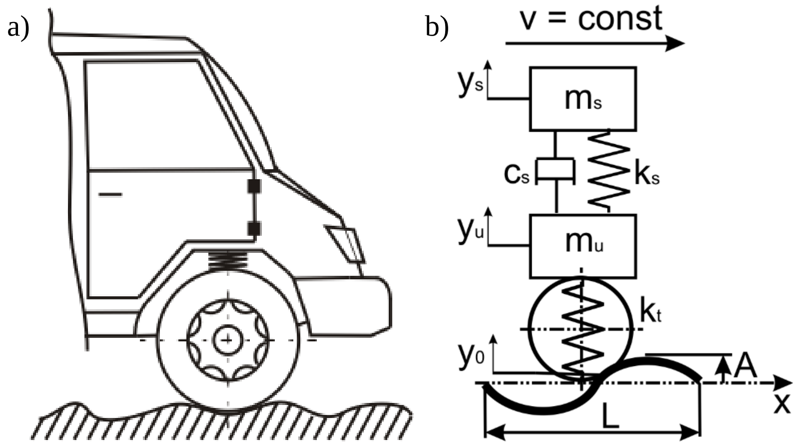

Fig. 1. **Schematic diagram of the front part of the vehicle (a) and a substitute model of the front suspension of the car (b)**

The differential equations of motion for the presented linear model take the form:

$$m_s\ddot{y}_s + c_s(\dot{y}_s - \dot{y}_u) + k_s(y_s - y_u) = 0$$

$$m_u\ddot{y}_u + k_t y_u - c_s(\dot{y}_s - \dot{y}_u) - k_s(y_s - y_u) = F_t. \quad (1)$$

In the model, a force is assumed based on the road profile according to the function: $F_t = k_t y_0 = k_t A\sin(\Omega t)$, where Ω represents the angular excitation frequency of the applied force and is expressed by the relationship (2):

$$\Omega = \frac{2\pi v}{L}. \quad (2)$$

Table 1

Parameters for the vehicle suspension analysis

Value	Parameter
A = 0.01 m	amplitude from road profile
L = 1.8 m	length of the wave
m_s = 560 kg	sprung mass
m_u = 102 kg	unsprung mass
c_s = 50 Ns·m^{-1}	damping coefficient
k_s = 40000 N·m^{-1}	suspension spring stiffness
k_t = 250000 N·m^{-1}	tire stiffness
v = 0-80 km·h^{-1}	range of the car linear velocity

Results and discussion

The results are provided by the quarter car model in Simulink environmental as presented in Fig. 2. In the simulation the numerical integration was used to solve the system of Eqs (1) by the ode 4th Runge-Kutta method at fixed step of time. The path of the driving speed was assumed within the range up to 80 km·h^{-1} because of the presence of both first two resonance zones, which occurred at frequencies

1.24 Hz and 8.50 Hz (see Fig. 3a). The natural frequencies were found by solving the characteristic equation (Eq. 3) of the linear sets of Eqs. (1).

$$\left(-m_s\Omega^2 + k_s\right)\left(-m_u\Omega^2 + k_t + k_s\right) - {k_s}^2 = 0 \tag{3}$$

Additionally, the response amplitudes y_u and y_s were analytically calculated by assumed of no damping component for simplifying calculations Eqs. (4):

$$y_u = \frac{k_t A(-m_s\Omega^2 + k_s)}{(-m_s\Omega^2 + k_s)(-m_u\Omega^2 + k_t + k_s) - {k_s}^2},$$

$$y_s = \frac{k_s k_t A}{(-m_s\Omega^2 + k_s)(-m_u\Omega^2 + k_t + k_s) - {k_s}^2}. \tag{4}$$

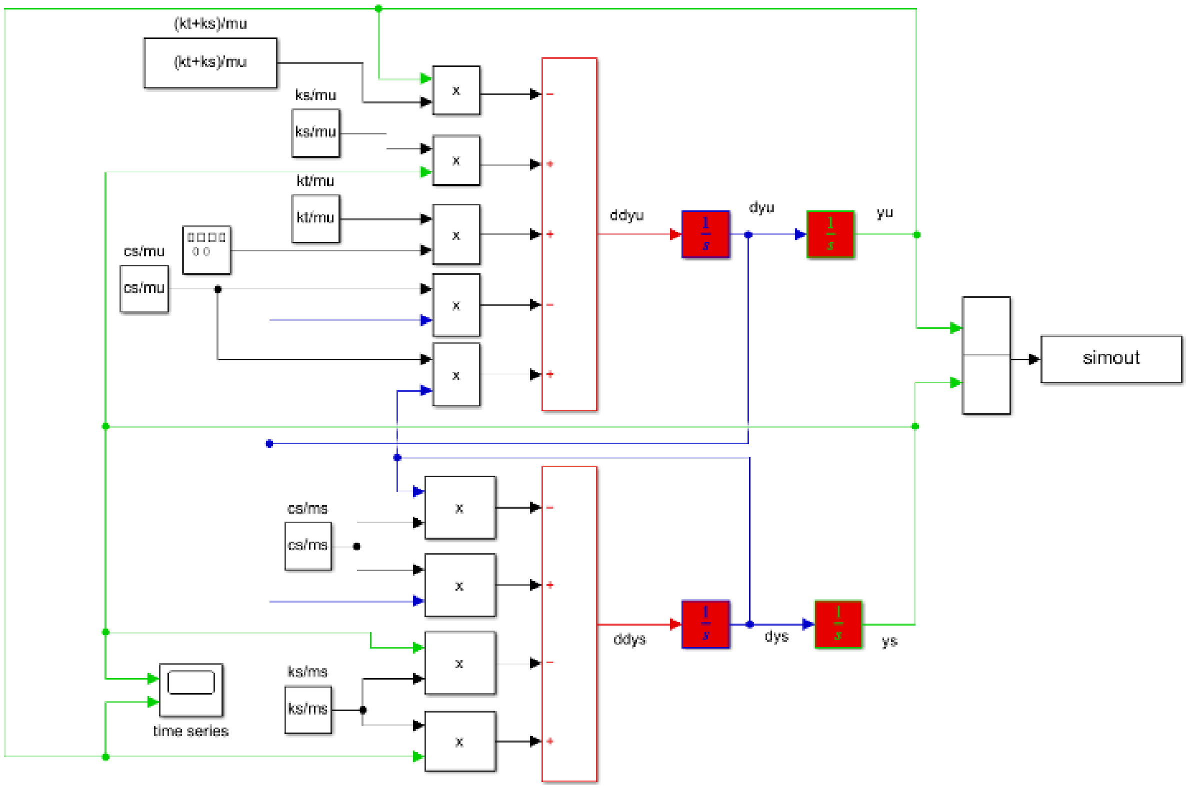

Fig. 2. **Simulink diagram of the "quarter car" model**

The amplitude – frequency characteristic in Fig. 3b) presents the dynamic behaviour of the system at excitation amplitude $A = 0.01$ m obtained both, by analytical and numerical approaches. It appeared in resonances peaks one can easily notice, the sprung mass is subjected to increasing output amplitude to dangerous value and it has to be damped by a dashpot c_s. In Fig. 4 a-d) is presented a different behaviour of the suspension system at significant driving speeds chosen for comparison. Both resonance zones reveal the increasing of the output amplitudes, while the smaller driving speed approximately 8.06 $\text{km}\cdot\text{h}^{-1}$ affects more harmful amplitude at the second mass m_s. It is reverse at higher speed at 55.08 $\text{km}\cdot\text{h}^{-1}$, which corresponds to the second resonance, then the amplitude of the sprung mass is decreased relative to the unsprung mass m_u.

Moreover, a driving speed is detected where the conditions are significantly different. The first is while the beating effect occurs. It appears near the first resonance at 8 $\text{km}\cdot\text{h}^{-1}$ which suddenly amplifies the output amplitude of the second mass. Such conditions of work are very unfavourable and should be avoided or eliminated by damper activation.

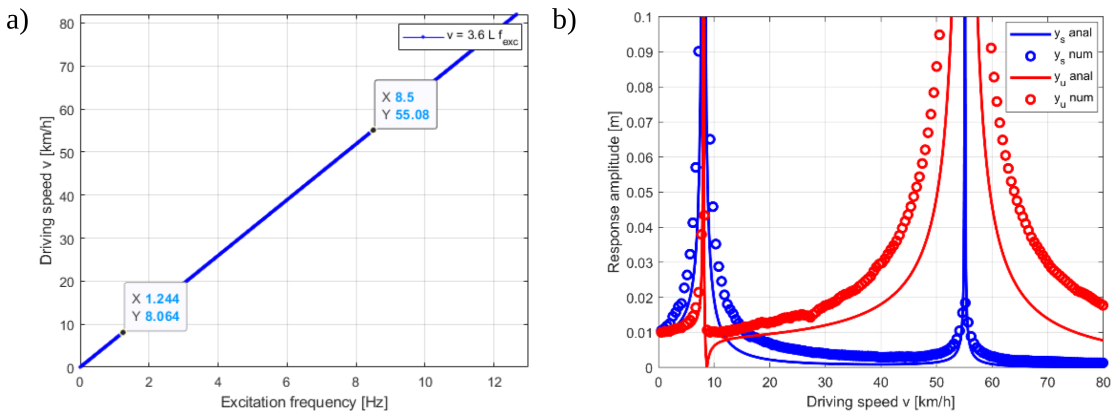

Fig. 3. **Relation between the resonance frequencies and driving speed (a) and the amplitude-frequency characteristics for the adopted data (Tab1) (b)**

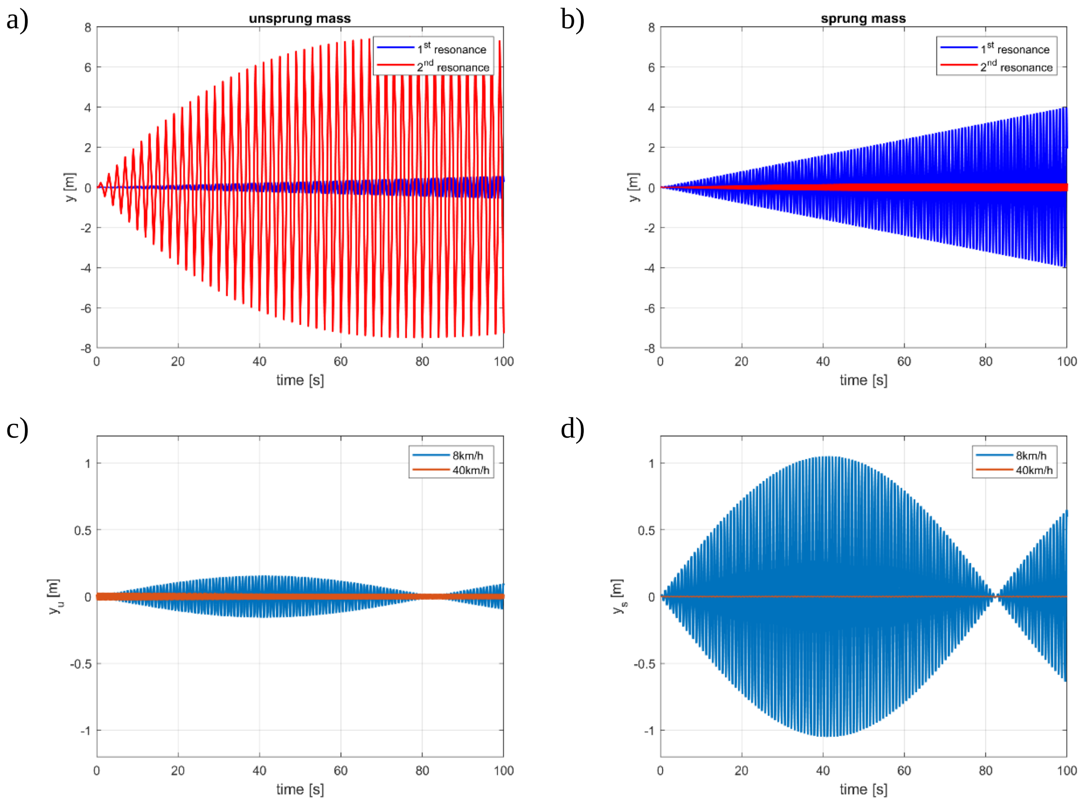

Fig. 4. **Time series of significant excitation frequency of the "quarter car" model**

But the second area which introduces very smooth condition is found at a higher driving speed between the resonance zone within 30-50 km·h^{-1}. It corresponds to an antiresonance effect, and the time history of these vibrations are presented in Figs. 4c) and 4d). One can notice that the energy of vibrations is accumulated by the unsprung mass m_u, here the vibrations are dampening on the sprung mass m_s, which is a very beneficial phenomenon.

The use of simulation tools is very helpful in development work and used in many engineering applications [31-37]. The obtained results of simulation studies show the potential of this model and its adjustment, which can also be helpful when using it for energy harvesting issues in subsequent research tasks.

Conclusions

1. The numerical model of a quarter car is proposed, which supports the estimation of driving conditions of vehicles, especially near the resonance zones.

2. The two degrees of freedom models have the advantage to work as an antiresonance system, which secures the car body away from harmful vibration sources from road profiles.
3. There is observed a beneficial effect known as anti-resonance, which effectively dampens the vibrations of the car body by transferring the energy from the vibrations to the first unsprung mass. The phenomenon of anti-resonance is particularly beneficial at speeds up to 80 $km \cdot h^{-1}$, because at highway speeds the system operates outside the resonant areas, and vibration absorption is not so necessary.

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Author contributions

Conceptualization, J.C. and M.B.; methodology, J.C. and M.B.; software, M.B.; validation, J.C., M.B and A.A.S.; formal analysis, J.C. and A.G; investigation, J.C. and M.B.; writing – original draft preparation, J.C., M.B., A.G. and A.A.S.; writing – review and editing, J.C. and M.B.; project administration, J.C.; funding acquisition, M.B. All authors have read and agreed to the published version of the manuscript.

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