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Determination of Vibrations Caused by the Road Profile in Case of a Fire Truck

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Abstract: In Hungary, the quality of lower class roads is inadequate in many cases. Road irregularities cause different vibrations in vehicles. It is important to examine the vibrations generated in vehicles by a given road section as they can lead to a reduction of lifetime or even an immediate failure of suspensions and other structural components. Recently several fire trucks with special vehicle body have also experienced structural failures which have possibly been caused by vibrations from road irregularities. To prove our assumption field measurements were carried out with the Csepel-Metz 755-10 heavy duty fire truck on a selected road section. In this paper the results of numerical simulations and field measurements with a special fire truck is presented. The aim of our research is to draw attention to the importance of the topic and to contribute to the safe operation of fire-fighting vehicles.

Keywords: Fire Truck, Field Measurement, Road Profile, Vibrations, Numerical Simulation

I. INTRODUCTION

The examination of vibrations generated by the operation of various machines is an important area of engineering, there are several studies dealing with the topic [1]. Vibrations can affect the operation of manufacturing equipment and the quality of the manufactured product, but can also cause injuries to the operator [2]. The harmful vibrations of vehicles can reduce the lifetime of different components and harm the health of the passengers [3]. Nowadays many electronic components, various control panels are used in vehicles, and their lifetime can be reduced by harmful vibrations, resulting in malfunctions [4]. The examination of various vibrations on vehicles was mostly carried out with passenger cars [5]-[7], different purpose trucks [8]-[11] or motorcycles [12]-[13] but not with fire-fighting vehicles. Therefore the examination of the vibrations occurring during the operation of a special fire truck is an important, current task [14]. By studying the literature available we have also found that vehicle vibration measurements and tests were performed mostly in laboratory conditions or test tracks [8]-[13],[15]. Field experiments with scale 1:1 were performed in a smaller number [16]-[17] and not with fire trucks with special suspension and vehicle body. The first step in our research was the measurement of the road profile, which was followed by mathematical modelling and numerical simulation of the suspension system. Taking the simulation results into account, we designed and carried out field measurements with the Csepel-Metz 755-10 fire-fighting vehicle. Accelerometer was mounted on the chassis, and measurements were carried out on the selected road section. Vibrations were examined on the vehicle body containing special fire-fighting equipment in order to prevent equipment failures and consequent malfunctions in the future.

II. ROAD PROFILE MEASUREMENT

In Hungary lower class roads are often in really bad condition, travelling through them badly loads vehicles. The test site was located relatively close to our University, in Nép street in Győr. The selected road section is a 2x1 lane, 6 m wide, flexible track structure with compaction asphalt pavement, classified as VI. category with 1500-3000 daily passing vehicles [18]. This road section was chosen because the traffic there is big enough and the condition of the pavement is very bad, therefore the speed limit is 30 km/h there. The shape of the road profile was determined using a digital laser meter. Leica Disto S910 Laser Distance Meter was used. The points were measured in the width of the wheels of the vehicle.

A 100 m section was selected on the road on which the field measurements were carried out later on. Geodetic measurement method was used to determine the profile of the road section. There are different height measurement methods, of which trigonometric height measurement was applied. The measurement device detects the vertical inclination, measures the direct distance between the gauge and the object, and determines the vertical distance from the horizontal base line by trigonometric calculation. The principle of measurement is shown in Fig 1.

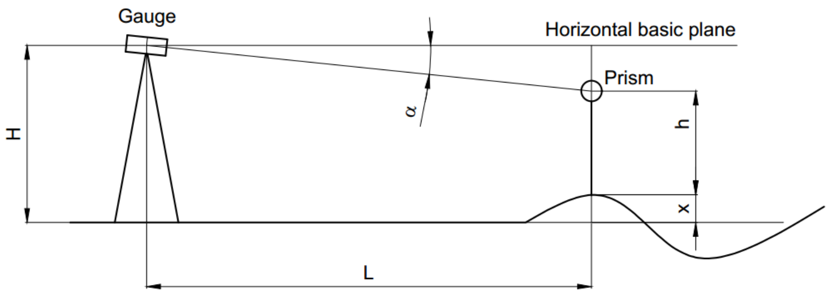


Fig 1.Trigonometric height measurement

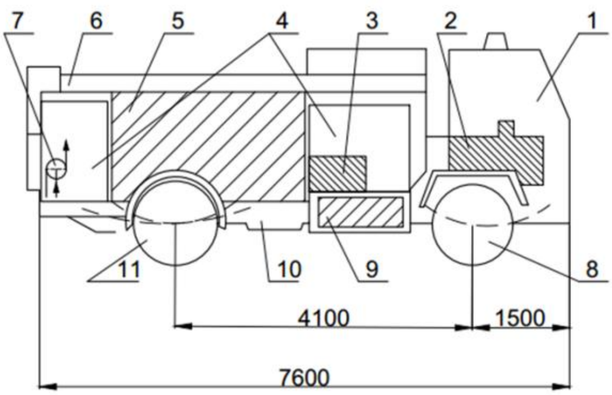
The vertical distance is calculated with the following formulas

$$x = H - L \cdot \tan(\alpha) - h \quad (1)$$

Absolute value of *x* is calculated by the gauge, therefore it will always be a positive number. The measured road profile was implemented to the simulation later on.

III. TECHNICAL PARAMETERS OF THE TESTED VEHICLE

The vehicle chosen for our tests is a heavy duty fire truck manufactured in Hungary. The vehicle was built with a ladder frame and rigid front and rear axles. Both axles are equipped with leaf springs and an auxiliary leaf spring for permanent load is installed at the rear axle. Hydraulic shock absorbers on both sides provide the proper road holding. The vehicle is powered by a CUMMINS 6CT 8.3 refueling four-stroke diesel engine that drives the FP-2418 fire pump through a separate drive. The special fire truck vehicle body, which contains the 5000-liter water tank and the 500-liter foam tank, as well as the 2400 liter / min water pump and the storage compartments for fire-fighting equipment and hoses is secured with a separate subframe to the chassis. The 220-liter fuel tank is assembled to the chassis on the right and the battery compartment on the left. The driver's cabin is fixed to the main chassis with special rubber mount. The vehicle is equipped with a 6 + 1-speed synchronous transmission system. The structure of the vehicle is showed in Fig 2.



1. Cabin, 500 kg
2. Engine + transmission: 2400 kg
3. Foam tank 500 l, 500 kg
4. Firefighter equiment, 500 kg
5. Water tank 5000 l, 5000 kg
6. Superstructure, 1800 kg
7. Pump, 200 kg
8. Front axle, 620 kg
9. Fuel tank + accumulator, 200 kg + 100 kg
10. Chassis, 3200 kg
11. Rear axle, 1480 kg

Fig 2. Csepel D755-10 fire truck

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IV. NUMERICAL SIMULATION

As we are interested in the vibrations of the vehicle body a simplified full-vehicle model was created with 3 degrees of freedom (Fig. 3).

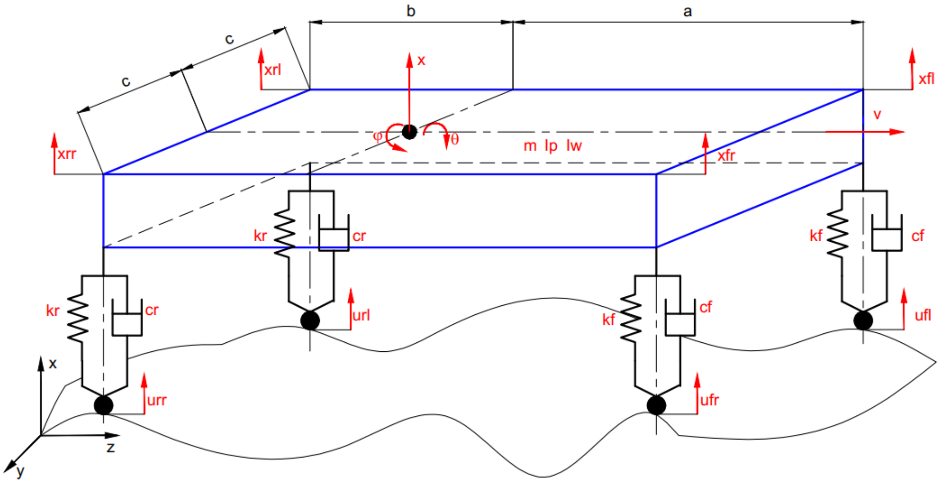

Fig. 3. Dynamic model of the vehicle

The 3 degrees of freedom (DOF) of the vehicle are listed in the following table.

Table I Degrees of Freedom

x	vertical displacement of the vehicle body
φ	roll of the vehicle body
θ	pitch of the vehicle body

The equations of motion describing the system's dynamics can be obtained with free-body diagrams [19].

$$m\ddot{x} = F_1 + F_2 + F_3 + F_4 - mg \quad (2)$$

$$I_p\ddot{\varphi} = (F_1 + F_3)a - (F_2 + F_4)b \quad (3)$$

$$I_w\ddot{\theta}(F_3 + F_4)c - (F_1 + F_2)c \quad (4)$$

where

$$F_1 = \begin{cases} (u_{fl} - x_{fl})k_f + (\dot{u_{fl}} - \dot{x_{fl}})c_f > 0 \\ 0, else \end{cases} \quad (5)$$

$$F_2 = \begin{cases} (u_{rl} - x_{rl})k_r + (\dot{u_{rl}} - \dot{x_{rl}})c_r > 0 \\ 0, else \end{cases} \quad (6)$$

$$F_3 = \begin{cases} (u_{fr} - x_{fr})k_f + (\dot{u_{fr}} - \dot{x_{fr}})c_f > 0 \\ 0, else \end{cases} \quad (7)$$

$$F_4 = \begin{cases} (u_{rr} - x_{rr})k_r + (\dot{u_{rr}} - \dot{x_{rr}})c_r > 0 \\ 0, else \end{cases} \quad (8)$$

where

$$x_{fl} = x + a\varphi - c\theta \quad (9)$$

$$x_{rl} = x - b\varphi - c\theta \quad (10)$$

$$x_{fr} = x + a\varphi + c\theta \quad (11)$$

$$x_{rr} = x - b\varphi + c\theta \quad (12)$$

The mass parameters of the vehicle were taken from manufacturer's catalog. The parameters of spring stiffness and the damping coefficients were taken from literature [20]. The parameters used for simulation are listed in Table II.

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Table II. Parameters used for simulation

Name	Notation	Value	Unit
mass of vehicle body	m	16500	kg
longitudinal moment of inertia of vehicle body	I_p	58000	kgm^2
cross directional moment of inertia of vehicle body	I_w	17500	kgm^2
half of the vehicle width	c	1.1	m
rear axle distance from centre of mass	b	0.613	m
front axle distance from centre of mass	a	3.486	m
rear suspension spring stiffness	k_r	400000	N/m
front suspension spring stiffness	k_f	200000	N/m
rear suspension damping coefficient	c_r	4000	Ns/m
front suspension damping coefficient	c_f	2000	Ns/m

Numerical simulations were carried out with Maple. *ODE45* numerical algorithm was used for calculations with step 0.01 s [21]. The speed of the vehicle was 30 km/h, the same as in case of field measurement. The measured road profile as kinematic excitation signal was implemented as a spline polinom. First the excitation signal on the tire was calculated with interpolation of the measured road profile points, converting the horizontal distance among points into time as a function of speed. This data was imported to Maple where the spline-polinoms were calculated, which were used as functions in the simulation ($u_{rr}(t)$, $u_{rl}(t)$, $u_{fr}(t)$, $u_{fl}(t)$).

As mentioned before the carried fire-fighting equipment is the most important part to be protected. Therefore in this paper only the vibrations of the vehicle body is examined. Another paper is planned, which will deal with a more detailed numerical examination of the vehicle. The acceleration of the point where the accelerometer was installed was calculated (see Fig. 5) with the following equation.

$$\boldsymbol{a_B} = \boldsymbol{a_s} + \frac{d}{dt}(\boldsymbol{\omega x r_{BS}}) \tag{13}$$

where $\mathbf{a_B}$ is the acceleration at the selected point, $\mathbf{a_S}$ is the acceleration at the centre of mass, $\boldsymbol{\omega}$ is the angular velocity and $\mathbf{r_{BS}}$ is the distance vector.

The frequency spectrum diagram from the simulation can be seen in Fig. 4.

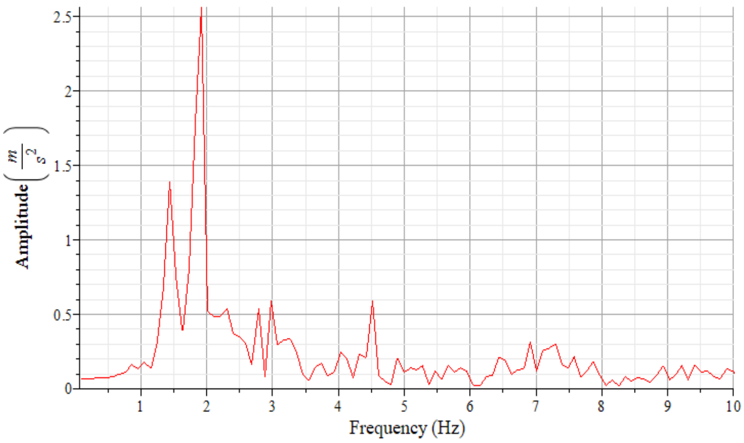

Fig 4. Spectrum diagram of acceleration

It can be seen that the peak acceleration takes place at 1.8 Hz. The eigenfrequencies of this vehicle can be calculated with solving the following quadratic eigenvalue problem [22]:

$$Q(\lambda) = \lambda^2 M + \lambda C + K \tag{14}$$

where M is the mass matrix, K is the stiffness matrix. and C is the damping matrix. The matrices are the following:

$$M = \begin{bmatrix} m & 0 & 0 \\ 0 & I_p & 0 \\ 0 & 0 & I_w \end{bmatrix}$$

$$K = \begin{bmatrix} -2k_f - 2k_r & -2ak_f + 2bk_r & 0 \\ -2ak_f + 2bk_r & -2a^2k_f - 2b^2k_r & 0 \\ 0 & 0 & (-2k_f - 2k_r)c^2 \end{bmatrix}$$

$$C = \begin{bmatrix} -2c_f - 2c_r & -2ac_f + 2bc_r & 0 \\ -2ac_f + 2bc_r & -2a^2c_f - 2b^2c_r & 0 \\ 0 & 0 & (-2c_f - 2c_r)c^2 \end{bmatrix}$$

The eigen frequencies of the vehicle are 1.13 Hz, 1.45 Hz and 1.68 Hz respectively.

The peak acceleration takes place at 1.8 Hz, which is close to one of the eigen frequencies, therefore resonance might occur.

V. FIELD MEASUREMENT

The purpose of the measurement is to record the vibration on the vehicle body. Brüel & Kjaer 3560-C Frontend Accelerometer sensors and Pulse Labshop 18 software were used. Before the measurement an accelerometer sensor was installed at the vehicle body as it can be seen in Fig 5.

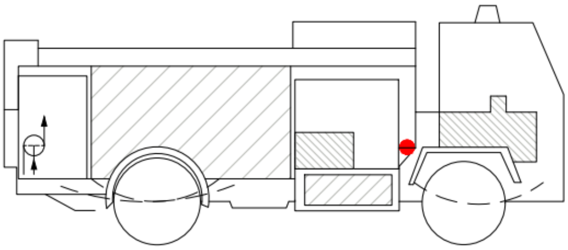

Fig. 5. Figure: Placement of vibration sensor on the vehicle

The data acquisition unit and the control computer were placed in the vehicle cabin. After putting the instruments into operation we started the four-stroke Diesel engine of the vehicle and measured the idle speed vibrations. It was found that the instruments were working properly, then the field measurement followed. The front and end of the selected road section were marked with paint on the road. The speed of the vehicle was 30 km/h, because of speed limit.

VI. MEASURED DATA ANALYSIS

In this article only the measurement on the vehicle body is discussed. Another paper is planned in the future, which will deal with a more detailed field measurement results. As a first step, the idle vibrations generated by the four-stroke Diesel engine built into the vehicle were measured. This step was necessary to identify and separate the frequencies of vibrations generated by the engine from the vibrations of the vehicle body. The idle speed based on manufacturer's data is 600 1 / min, ie 10 rev / s A four-stroke internal combustion engine with 6 cylinders causes theoretically 30 Hz vibrations. The measured frequency peak is accordingly 29.5 Hz, which can be seen in Fig. 6. In addition harmonics can also be seen, which are multiples of this frequency.

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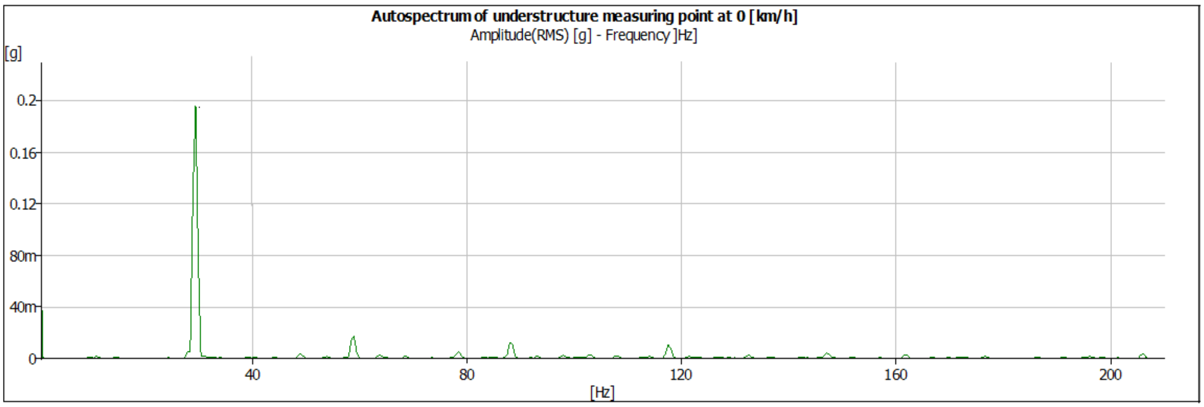

Fig. 6. Idle speed acceleration spectrum measured on chassis

Fig 7. shows the result of the field measurement at vehicle speed 30 km / h. The peak frequency 55.6875 Hz comes from the higher engine speed (1114 1/min). The highest amplitude frequency component is 2.125 Hz, which is generated by the excitation from the road through the suspension.

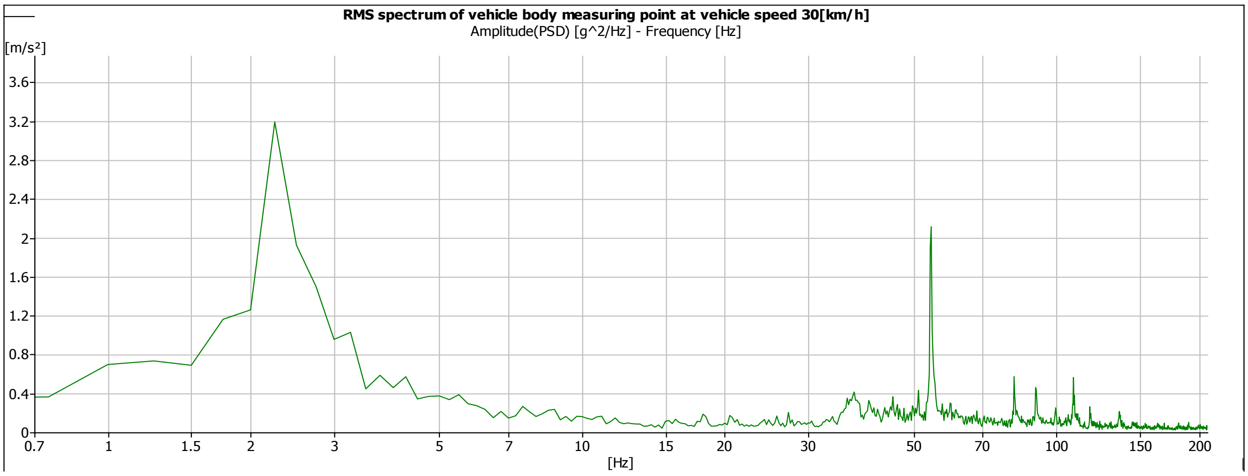

Fig. 7. Acceleration spectrum measured on chassis at vehicle speed 30 km/h

The occurring peak frequency similarly to the results evaluated from numerical simulations is close to one of the natural frequencies, which means there is a great chance for resonance in case of low quality roads.

VII. CONCLUSION

The vibration occurred from a low quality road were examined with numerical simulations and field measurements. A simple fire truck suspension model was created, which is used to examine the vibrations of the vehicle body. From the numerical simulation and the field measurement it can be observed that the peak acceleration was at 2 Hz, which is close to one of the eigen frequencies of the vehicle. Therefore it can be concluded that poor pavement quality might cause the breakdown or damage of certain structural elements of fire trucks. However further measurements and numerical simulations are required to clarify our statement, which will be the task of further research. We consider it important to emphasize the importance of field measurements as they are essential to discover phenomena during the operation of fire trucks. With them an accurate, realistic computer simulation can also be created, which can be used in the practice effectively. With the presented results we want to draw the attention to the importance of the topic and to help the safe operation of fire-fighting vehicles.

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