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DEPENDENCE BETWEEN VIBRATION LOAD ON THE DRIVER AND ACCOMMODATION SCHEME OF THE TWO-AXLE TRUCK

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Keywords: automobile, composition with hood, composition without hood, amplitude-frequency response, acceleration, driver's seat, oscillation.

Abstract. The selection of the accommodation scheme of the truck depends on the vehicle's function, engine type, power train, suspension design, etc. In its turn, the accommodation type significantly affects almost all the major performance data of the vehicle, including its stability and drivability, smooth ride, off-road capacity, etc. Each accommodation scheme available today has its benefits and drawbacks. The article provides their brief overview as well as considers the question of their influence on the vibration load of the driver, which affects both the driver's health and his combat efficiency. The article features an analytical model and a corresponding mathematical model and gives the expression of the frequency and range characteristics to determine the position of the driver's seat (the task is solved linearly). Further on, the authors consider the example of its composition for the two options of the seat position within the base conditioned by the accommodation scheme type.

ЗАВИСИМОСТЬ ВИБРОНАГРУЖЕННОСТИ ВОДИТЕЛЯ ОТ КОМПОНОВОЧНОЙ СХЕМЫ ДВУХОСНОГО ГРУЗОВОГО АВТОМОБИЛЯ

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Ключевые слова: автомобиль, капотная, бескапотная компоновочные схемы, амплитудно-частотная характеристика, ускорения, сиденье водителя, колебания.

Аннотация. Выбор компоновочной схемы грузового автомобиля зависит от его назначения, типа двигателя и силовой передачи, конструкции подвески и др. В свою очередь, компоновка оказывает существенное влияние практически на все основные технико-эксплуатационные свойства машины: устойчивость-управляемость, плавность хода, проходимость и пр. Каждая из существующих на сегодняшний день компоновочных схем грузового автомобиля имеет свои преимущества и недостатки. В статье приводится их краткий анализ, а также рассматривается вопрос их влияния на вибронагруженность водителя, которая оказывают воздействие не только на его организм как таковой, но и на условия наблюдения и стрельбы, т.е. на боевую эффективность. Составлены расчетная схема и соответствующая математическая модель, получено выражение амплитудно-частотной характеристики для определения ускорений на сиденье водителя (задача решается в линейной постановке), рассмотрен пример ее построения для двух вариантов расположения сиденья в пределах базы, обусловленных вариантом компоновочной схемы.

The smooth ride of the vehicle is defined as its ability to move in the specified time of operational speeds without exceeding the actual norms of vibration load on the driver, passengers, cargo transported, its construction elements as well as without significant jolts or bumps. Apart from increasing the driver and passenger comfort thus improving their well-being, the improvement in the smooth ride entails the increase in the average speed of the movement, in the runlife, cargo safety, reduction in the fuel consumption and cost of transportation as this is one of the main performance characteristics of the vehicle.

It is well-known that there exist accommodation schemes of trucks [1, 2]: conventional where the engine is located above the front-wheel axle, whereas the driver's cabin is behind the engine; semi-conventional where the engine is located above the front-wheel axle and the driver's cabin is protruded onto the engine; and cab over engine scheme where, as the name suggests, the engine is placed above the front-wheel axle and the cabin is located above the engine or with the engine placed behind the front-wheel axle and the cabin is placed in front of the engine. They all possess their advantages and disadvantages and the selection of a particular scheme for a particular vehicle depends on its functionality, engine type and power train, the number of axles, their location within the base, suspension type, etc. [3] However, the scheme also has a significant effect on the performance data of a mobile vehicle, including its weight, off-road capability, smooth ride, active and passive safety, driver's working conditions, cargo safety, etc.

Thus, for instance, the conventional scheme results in the reduction of the height of the vehicle, and therefore, the mass center (which, in its turn, positively impacts the stability and drivability of the vehicle), the simplification of the control system, better passive safety achieved through energy absorption by the engine part in case of a frontal collision, however, this results in lower active safety as the front view becomes worse while axial outline dimensions are increased. Its advantages also consist in the convenience of getting into and out of the vehicle and the accessibility of the engine. The semi-conventional type results in the reduction of axial outline dimensions and the base, the volume of the body increases, which is a benefit, however, this impedes access to the engine and the layout of control mechanisms. In case of cab over engine type the axial outline dimensions and the base are further reduced, which enhances the maneuverability of the vehicle and the front view, yet the access to the engine is significantly hampered, and, most importantly, passive safety is likely to worsen.

In terms of the crew protection from explosion in case of driving on a land mine, the specialists are divided: some argue that the multi-functional vehicles are used with both conventional and cab over engine designs. It is generally accepted that if the engine compartment is protruded forward, it may act as a buffer, however, in case of cab over engine layout, the whole power of the explosion will directly affect the cabin. However, if we consider Typhoon military trucks, we can see them available in both layouts (fig. 1) [4].

It is well-known that the oscillations of the vehicle affect the well-being of the driver and the passenger. The sensitivity to these oscillations depends on their frequency, intensity, direction and length. A human organism is adapted to oscillations with the frequency, range, and acceleration similar to the values that occur when walking at an average speed, i.e. the oscillations with the frequency of 1.5 – 2 *Hz* with the acceleration below 0.4*g*. Humans are most sensitive to vertical oscillations in the range of frequencies from 4 to 8 *Hz* and to horizontal oscillations in the range of 1-2 *Hz*.

The oscillations with the frequency of 3 – 5 *Hz* cause nausea, vascular disorders, the frequencies of 4 – 11 *Hz* may cause resonance vibrations of the head, stomach, intestine, and liver.

Fig. 1. Typhoon Military Trucks

The features and parameters of oscillations affect not only the body, but also the conditions of observation and shooting, i.e. combat efficiency. The so-called commensurability ratio that takes into account the probability of the combat vehicle victory and that equals the likelihood of the victory of the vehicle operated by the unfatigued crew divided by the likelihood of the victory of the vehicle operated by the fatigued crew can reach 2.3 [5].

Let us consider the way the accommodation of the driver's seat (which is directly linked to the accommodation scheme) influences his working conditions from the perspective of oscillation processes.

A qualitative comparison allows us to obtain frequency and range characteristics of accelerations affecting the driver's seat depending on its location within the vehicle base, all other terms being equal. There exist different biodynamic models of a human. The simplest one is the single-mass model where the driver and the spring part of the seat are considered as a single concentrated mass. The corresponding analytical model is provided in Figure 2, where M, m is spring and non-spring weight of the vehicle; m_c is the weight of the driver and the mobile part of the seat; y, ξ, η are vertical movements of spring and non-spring elements of the vehicle and driver respectively; c_p, k_p are the wheel rate of the stiff element and the ratio of the vicious resistance of the vehicle suspension; c_c, k_c are the wheel rate of the stiff element and the ratio of the vicious resistance of the seat suspension respectively; c_{sh} is the normal stiffness of tires; $q(t)$ is the kinematic impact from the road surface.

It is possible to simplify this model because of the absence of interdependence between the oscillations of the front and rear masses. As the weight of the driver and the movable part of the seat is significantly lower than the weight of the front and rear spring weights of the vehicle, we can first obtain frequency characteristics of the front and rear parts, then the frequency characteristics of the position of the driver's seat located at the distance of x_c (see Fig. 2) and, considering it as a source of oscillations of the seat with the driver, we can get the frequency and range characteristics of the accelerations affecting the driver's seat.

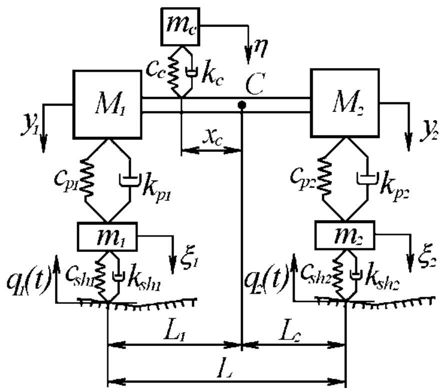

Fig. 2. Analytical model

The analytical model given may be described by the following system of differential equations:

$$\begin{cases} M_j \ddot{y}_{pj} + c_{pj}\left(y_j - \xi_j\right) + k_{pj}\left(\dot{y}_j - \dot{\xi}_j\right) = 0; \\ m_j \ddot{\xi}_j - c_{pj}\left(y_j - \xi_j\right) - k_{pj}\left(\dot{y}_j - \dot{\xi}_j\right) + c_{shj}\left(\xi_j + q_j(t)\right) + k_{shj}\left(\dot{\xi}_j + \dot{q}_j(t)\right) = 0. \end{cases}$$

Where the index j = 1 for the front part, and j = 2 for the rear part.

We accept that $q(t)$ is a harmonic function with the cyclical frequency ν. The differential equations found in the system are linear, thus we can apply Laplace transformations to them [6] and present them as a system of algebraic equations with complex variables $\bar{y}_j(s)$, $\bar{\xi}_j(s)$, $\bar{q}_j(s)$ where $s = a + bi$. With zero initial conditions it can be presented as:

$$\begin{cases} \left(M_j s^2 + k_{pj} s + c_{pj}\right)\bar{y}_j(s) + \left(-k_{pj} s - c_{pj}\right)\bar{\xi}_j(s) = 0; \\ -k_{pj} s \bar{y}_j(s) + \left(m_j s^2 + \left(k_{pj} + k_{shj}\right)s + c_{pj} + c_{shj}\right)\bar{\xi}_j(s) = \left(-k_{shj} s - c_{shj}\right)\bar{q}_j(s). \end{cases}$$

As the kinematic impact is described by a harmonic function with the cyclical frequency ν, the forced oscillations will also have the similar frequency of ν. Therefore, we can replace the complex variable s by its imaginary component νi. As a result we get the equation system related to unknown frequency characteristics $\bar{y}_j(\nu i)$, $\bar{\xi}_j(\nu i)$ corresponding to the function $y_j(t)$, $\xi_j(t)$:

$$\begin{cases} \left(-M_j \nu^2 + k_{pj}\nu i + c_{pj}\right)\bar{y}_j(\nu i) + \left(-k_{pj}\nu i - c_{pj}\right)\bar{\xi}_j(\nu i) = 0; \\ -k_{pj}\nu i \bar{y}_j(\nu i) + \left(-m_j \nu^2 + \left(k_{pj} + k_{shj}\right)\nu i + c_{pj} + c_{shj}\right)\bar{\xi}_j(\nu i) = -k_{shj}\nu i - c_{shj}. \end{cases}$$

In this system for the sake of convenience we accept the singular range of the outer impact, i.e. $\bar{q}(\nu i) = 1$.

Having resolved the equation system, we get the frequency characteristics of the movements of spring weights with the singular outer kinematical impact:

$$\bar{y}_j(\nu i) = \frac{A1_j A0_j + B1_j B0_j}{A0_j^2 + B0_j^2} + i\frac{A0_j B1_j - A1_j B0_j}{A0_j^2 + B0_j^2},$$

where $A0_j = M_j m_j \nu^4 + \left(-k_{pj}k_{shj} - c_{pj}m_j - M_j c_{pj} - M_j c_{shj}\right)\nu^2 + c_{pj}c_{shj}$;

$B0_j = \left(-k_{pj}m_j - M_j k_{shj} - M_j k_{pj}\right)\nu^3 + \left(c_{pj}c_{shj} + k_{pj}c_{shj}\right)\nu$;

$A1_j = k_{pj}k_{shj}\nu^2 - c_{pj}c_{shj}$; $B1_j = \left(-c_{pj}k_{shj} - k_{pj}c_{shj}\right)\nu$.

To obtain the frequency characteristics in the place of the driver's seat fixation it is necessary to take into account the frequency characteristics for front and rear spring parts of the vehicle as well as the link between the y-coordinate of the road surface micro-profile under front and rear wheels

$$q_2(t) = q_1(t - \tau),$$

where τ is the time taken to cover the distance that equals the vehicle base:

$$\bar{y}_C(\nu i) = \frac{L_2 + x_C}{L}\bar{y}_1(\nu i) + \frac{L_1 - x_C}{L}\bar{y}_2(\nu i)\left[\cos(\nu\tau) - i\sin(\nu\tau)\right].$$

The equation of the oscillations affecting the driver's seat is as follows:

$$m_c\ddot{\eta} + c_c(\eta - y) + k_c(\dot{\eta} - \dot{y}) = 0.$$

Having completed the above-mentioned procedures, we get the expression of the corresponding frequency characteristics and then the expression of the frequency and range characteristics of accelerations affecting the driver's seat using the expression of the location of the seat fixture as a source of the oscillations:

$$H_{\ddot{\eta}}(\nu)=\sqrt{\frac{C1^2+D1^2}{C0^2+D0^2}}\left|\bar{y}_C(\nu i)\right|\nu^2,$$

where $C0=-m_c\nu^2$; $C1=c_c$; $D0=k_c\nu$; $D1=k_c\nu$.

By way of example we can obtain frequency and range characteristics of the accelerations affecting the driver's seat calculated for the following initial data: M_1 = 2200 *kilos*, m_1 = 550 *kilos*, c_{p1} = 270 *kN/m*, c_{sh1} = 1280 *kN/m*, M_2 = 6000 *kilos*, m_2 = 900 *kilos*, c_{p2} = 870 *kN/m*, c_{sh2} = 3250 *kN/m*, c_c = 25 *kN/m*, k_{p1} = 6.54 *kNs/m*, k_{sh1} = 1.59 *kNs/m*, k_{p2} = 15.667 *kNs/m*, k_{sh2} = 3.22 *kNs/m*, the relative damping factor of the seat oscillation is γ_c = 0.15, L = 4 *m*, L_1 = 2.5 *m*, L_2 = 1.5 *m*, m_c = 65 *kilos*.

The curves corresponding to the frequency and range characteristics are given in Figure 3, where the solid line for x_c = 1.5 *m*, and the broken line for x_c = 2.5*m*, i.e. when the seat is located above the front axle of the vehicle. As one can see from the comparison of the two curves, from the point of view of oscillations during the movement of the vehicle the driver's working conditions will be more favorable when the driver's seat is placed at a distance from the front axle.

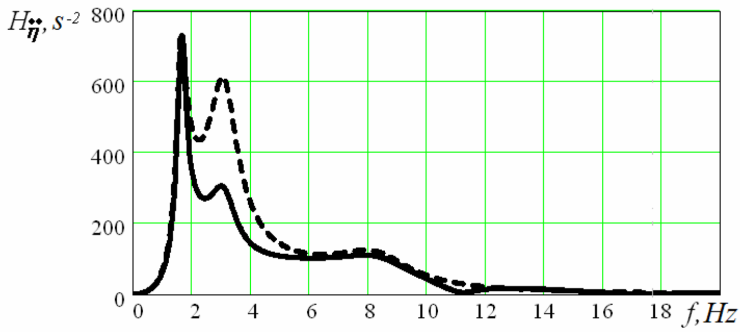

Fig. 3. Frequency and range characteristics of accelerations affecting the driver's seat

The oscillations are mostly caused by the micro-profile of the supporting surface on which the vehicle is moving. When researching the vibration load on the driver, in case of using simplified mathematical models it is expedient to use the so-called spectral approach that allows to obtain the spectral density of accelerations on the driver's seat that characterizes the distribution of acceleration dispersion by the frequency. To determine it, it is necessary to multiply the spectral density of the micro-profile by the square of the received frequency and range characteristics of the acceleration on the driver's seat.

This is certainly only a qualitative comparison of two location schemes of the driver's seat determined by the accommodation type of the vehicle. However, it has demonstrated how the comfort of the driver can change only because of the seat placement, which initially seems to be an insignificant parameter.

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