Road roller operator’s vibroprotection system improvement

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Abstract. The article reflects the results of the theoretical and experimental studies aimed to reduce dynamic influences on a road roller operator’s workplace by changing the design parameters of the ‘supporting surface – road roller – operator’ dynamic system. The results of the experimental studies on the modernized hinged tie rods construction are presented. The rigidity coefficients of the springing elements of the hinged tie rod ends are determined. The dependence of the rigidity coefficient on the deformation value of the hinged tie rod ends is graphically presented. A comparative analysis of the experimental studies results aimed at determining the value of the dynamic influences on an operator’s workplace (a cabin floor and a seat) carried out with a road three-axle three-wheeler roller with plant and modernised hinged tie rods. The transmission coefficients from the internal combustion engine to the cabin floor and the operator’s seat are obtained at small, medium and high engine speeds, with a vibration exciter switched on and off. The conclusions on the legal use of the modernised hinged tie rod structure as a component of the vibroprotection system of a road roller operator have been drawn.

1. Introduction
In the last decade, when the issues related to the improvement of road construction machines productivity through the improvement of the working bodies’ structural design, the increase of capacity, the introduction of remote control systems have achieved the maximum possible results, the issues related to the improvement of the ergonomic indicators in construction equipment have come to the fore. The problems of reducing dynamic influences, vibration in particular, and creating comfortable working conditions for construction and road machines operators are one of the main areas in ergonomics, and consequently one of the current matters in modern mechanical engineering.

One approach to this task is the use of passive vibroprotection [1]. Under this approach, reducing the dynamic effects may be achieved by reducing the couplings between a source and an object, thereby reducing the transmission coefficient by placing a vibrating insulator device between them [2].

Previously, the scientists such as V V Bolotin, M Z Kozlovskii, N I Ivanov [1] were involved in solving the problems related to the protection of road construction machines and hand-held equipment operators from vibration. S B Eliseev, M M Gaitsgori, A A Khodakova and others’ works were devoted to the design improvement of the seats and cabin suspensions in self-propelled machines. Y Yunshi, F Zhongxu., Ch.Shbin, J Selech, W Konrad, R Dawid studied the changes in the design of a working body of a vibrating road roller to increase productivity and the issues of the reliability of these machines. E A Kozinov, V P Gerger and V V Sovrasov are working on optimizing the parameters of the vibration protection systems of mobile machines and structures. At present, G M Kutkov, V N Sorokin, O V Fominova, Y V Koshelev are developing the effective vibration protection systems[1].
Despite numerous studies in this area, the improvement of the ergonomic performance of road rollers is supposed to be a largely unexplored issue.

2. Problem setting
The task is to reduce the amount of dynamic influences on an operator’s workplace by changing the design parameters of the ‘supporting surface – road roller – operator’ dynamic system.

3. Theory
A mathematical model [3,4] for the analysis and synthesis of the ‘supporting surface – road roller – operator’ dynamic system parameters was developed. The basis of the mathematical model is a generalized calculation scheme of a complex ‘surface to be treated – road roller – operator’ dynamic system (figure 1). The calculation diagram is a 3-mass system and reflects the characteristics of the system [3,5].

![Figure 1. ‘Supporting surface – road roller – operator’ calculation diagram.](image)

The generalized calculation scheme of the ‘supporting surface – road roller – operator’ dynamic system is a system with three masses [2,6]:
– $m_1$ mass includes the mass of the drive rollers, frame and engine. The mass centre of the first unit shall be located at $O_1$ [7,8];
– $m_2$ mass includes the mass of the rear roller, frame, rotating mechanism, 25% part of the operator’s mass. The mass centre of the second unit shall be located at point $O_2$ [8];
– $m_3$ mass includes the mass of the seat, 75% part of the operator’s mass. The mass center of the third unit is at point $O_3$ [8].

Gravity forces are formed in the gravitational field of the mass, as shown in the calculation diagram with $m_i g\hat{F}_j (\vec{F}_2, \vec{F}_3, \vec{F}_4)$ vectors [8].

Impact force shall be applied on the first unit from the engine and vibration exciter [8]

$$\vec{F}_1 = \vec{F}_1^t + \vec{F}_1^v. \tag{1}$$
\( \mathbf{F}_2 \) gravity force shall be applied on the compacted soil from the front rollers. The effect of the ICE, the vibration roller and the mass of the units on the treated surface leads to \( \mathbf{F}_5, \mathbf{F}_6 \) reactions. The value of each of these reactions consists of the sum of two components: static and dynamic [9]

\[
\begin{align*}
\mathbf{F}_5 &= \mathbf{F}^{st}_5 + \mathbf{F}^{d}_5, \\
\mathbf{F}_6 &= \mathbf{F}^{st}_6 + \mathbf{F}^{d}_6.
\end{align*}
\]

A static component depends on the gravity force applied on the relevant part of a road roller [4,9]:

\[
\begin{align*}
\mathbf{F}^{st}_5 &= m_1 g, \\
\mathbf{F}^{st}_6 &= m_2 g + m_3 g.
\end{align*}
\]

The dynamic components depend on the parameters of the supporting surface and shall be determined according to the formulas [1]:

\[
\begin{align*}
\mathbf{F}^{d}_5 &= -k_1 \cdot F_1, \\
\mathbf{F}^{d}_6 &= -k_2 \cdot F_1,
\end{align*}
\]

where \( K_1, K_2 \) are coefficients dependent on the physical-mechanical properties of the material being compacted.

The elastic properties of the ground are characterized with \( c_1, c_2 \) rigidity coefficients and \( b_1, b_2 \) viscous friction coefficients.

The dynamic system design parameters studied in this work are represented in the design model with viscous elastic coupling, the suspension is characterized with \( c_3, c_4 \) rigidity and \( b_3, b_4 \) friction coefficients of an operator’s seat and with \( c_1, c_4 \) and \( b_1, b_4 \) rigidity and viscous friction coefficients of the hinged tie rods [7,10].

The mathematical model was implemented in Mathlab software product. The compiled ‘Road roller 107’ program, which source data are geometric parameters, units mass, rigidity coefficients and resistance coefficients of the elastic elements, enables to determine the influence of structural and operational parameters on the dynamic effects on the cabin floor and road roller operator’s seat [10,11].

In the experimental studies, Ecophysics multi-functional measuring instrument was used, which allows to measure average quadratic values of vibration acceleration, corrected levels of vibration acceleration of octave and third octave spectra. The vibration sensors are installed on an ICE frame, a cabin floor and a road roller seat.

Due to the impossibility of directly measuring engine speed, the speed of the drive shaft of the vibration detector was measured and then converted through the ratio to the engine speed [1].

The average engine speeds correspond to a transport mode and the maximum speeds correspond to a working mode (vibrocompaction). In order to avoid the influence of transition processes in the ‘surface to be treated – road roller – operator’ system, the recording of measurements started not earlier than 40 seconds after the start of the road roller operation and lasted at least 60 seconds. The number of the measurements for the combinations of the operating modes and the condition of the compacted soil is equal to three at least.

The theoretical studies have concluded that the design parameters of a road roller in order to reduce the dynamic effects on an operator’s workplace are legitimate to be used. The method of passive vibration insulation was selected as a method for reducing dynamic effects. The key point of such a method consists in weakening the connection between the source and the object by placing an insulating device between the sources, thereby reducing the transmission coefficient [11].

To confirm the results of the theoretical studies, the experimental models of the upper and lower hinged tie rods were produced (fig. 2a). The difference between the experimental models and the plant hinged tie rods is that the cavities between the containers enlarged to the size of the rods ends are fully filled with elastoplastic (fig. 2b).
The rigidity coefficients of the hinged tie rod ends are determined in accordance with GOST ISO 7743-2013 with ‘LKSM-1K Strength of materials and joints’ laboratory equipment (Fig. 3) [12].

As a result of the research, the compression speed was determined. The limit was up to 250 µm/s. The maximum load was 20 kN, the compression stroke was 0...3 mm, the number of compression cycles was 4. Figure 4 graphically presents the relationship of the compression force to the deformation of the hinged tie rod ends with the sample compressed by 25%.

The found dependence of the compression force (F) on the deformation of the elastic element (x) of the hinged tie rod end was approximated with the exponential function

\[ C = 0,718x^{0.57}. \]  

(8)
The data obtained enabled to calculate the dependence of the rigidity coefficient of the hinged tie rod end on the deformation value (fig. 5). The dependence of the rigidity coefficient (C) on the deformation of the hinged tie rod end (x) is also approximated with the exponential function

\[ C = 0.718 x^{0.57}. \]  

As a result of the tests carried out, an average value of rigidity coefficients at 25% samples compression was determined. The value was \( 5 \cdot 10^5 \) H/m, which corresponds to the required rigidity coefficient obtained in theoretical studies [13].

The field experiments were carried out to confirm the results of the study. The experiments were carried out in Siberian State Automobile and Highway University (SibADI) laboratories. The three-axle three-wheeler vibrating roller (DM-107 model) was used as an experimental model. The research for the road roller with plant and experimental rods was carried out in the same conditions of the supporting surface, operating modes, engine speed and vibration exciter.
During the tests, some different operating modes of the road roller were simulated: parking with the engine switched on and vibration-activated; parking with the engine running and vibrating exciter running; a compaction operating mode with the vibration exciter switched on. A variable parameter is internal combustion engine speeds. A range of change is 1300 ... 3225 o/min [14].

4. Results
The results of the experimental studies are presented as the examples in figures 6 and 7. The low speed of the ICE (figures 6a, 7a) corresponds to the road roller parking with the engine and the vibration exciter switched on. The average speed (Figure 6b, 7b) corresponds to the road roller vibrocompaction operating mode, i.e. the engine and vibration exciter are activated.

In order to avoid the effect of transition processes in the compacted ground – road roller - mechanic system, the recording of the measurements started not earlier than 40 seconds after the start of the roller operation and lasted at least 60 seconds. The number of measurements for each of the operating modes combinations and the condition of the compacted soil is equal at least to three [3.15].

During the tests, the installation of the experimental rods on the road roller was indicated to significantly decrease the vibration acceleration value on the cabin floor and operator’s seat within 1 to 125 Hz frequency band. The greatest effect from the installation of the experimental rods is observed in the operating mode for its medium and high engine speeds. 16 Hz is for an average engine speed, from 4 Hz is for a high speed. For the road roller operating mode, the vibration acceleration on the cabin floor is decreased 2.6 times, at 8 and 16 Hz frequencies -approximately 1.5 times, at 31.5 and 63 Hz frequencies - 2.2 times.
The vibration decrease at 1 and 2 Hz frequencies is insignificant. This is due to the impossibility of further reducing the rigidity of the hinged tie rods ends, as they must not only have good vibration-proofing properties, but also ensure the reliability of the control of the road roller movement along a predetermined trajectory [1,16].

The determination of the coefficients for the transmission of the action of the engine and the vibration exciter to the cabin (K21C and K21O) and the mechanic’s seat (K31C and K31O) enabled to prove the effectiveness of the proposed engineering solution. As an example, figure 8 shows the graphical dependences of K21C and K31C transmission coefficients correspond to the plant rods, K21O and K31O coefficients correspond to the road roller with experimental rods.

![Figure 8. Engine transmission factors to the floor of the cab and from the engine to the mechanic’s seat at engine speed of 1700 by/min.](image)

The graph shows the most significant reduction in the transmission coefficient obtained in 8 - 63 Hz frequency band.

5. Conclusion
The experimental studies have confirmed the validity of the proposed mathematical model and the results of the theoretical studies conducted on it. The model of the ‘supporting surface – road roller – operator’ dynamic system can be successfully used for the adequate mapping of the properties of the system parameters and the study of the damping capacity of the structural elements of vibration protection.

The hinged tie rods are proposed as an engineering solution for reducing vibration. The partial modification of the structure enables to reduce the vibration parameters on the paths of its propagation, to absorb and dissipate a part of the energy transmitted by the hinged tie rods at vertical oscillations of the assemblies and units of the road vibration roller.

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