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Decrease of the Vibration Load Level on the Tractor Operator Working Place by Means of Using of Vibrations Dynamic Dampers in the Cabin Suspension

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Abstract

This paper is devoted to description of the method for decreasing of the vibration load level on the tractor operator working place using vibrations dynamic dampers in the cabin suspension. Standard cabin vibration isolators of tractors of DT and VT series include the monolith rubber block and have limited capabilities to change its elastic parameters and adapt in the course of the impact of vertical and lateral vibration loads with various frequencies. Elastic-damping properties of these vibration isolators doesn't allow one to provide comfortable conditions for the operator's work because their abilities of vibration damping aren't sufficient especially at low-frequency vibrations. Dynamic vibration dampers installed in the cabin suspension were proposed by authors for decreasing of the vibration load level on the operator's workplace. 3D- model of the caterpillar chassis, suspension, engine, cabin and operator's seat of the tractor was created in the software "Universal mechanism" for performing of the computational research. This model was used to conduct research of sprung mass vibration for tractor motion with and without hook load at 3rd and 7th gears on the smooth surface and on testing areas with single, periodic and random bumps. Research results undoubtedly indicate the best vibroprotective properties of cabin suspensions with dynamic vibration isolators.

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Keywords: tractor cabin; suspension; standard vibroisolator; dynamic vibration damper.

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1. Introduction

Tractors power ability and motion speeds constantly increase and it results in increasing of dynamic loading of chassis and transmission details and rising of level of vibrations generated by this details. Vibration loads impact negatively on tractor details and units, environment and operator. It results in higher fatigability of an operator and increasing of control mistakes number which influence on the productivity of tractor aggregate. During lasting impact of vibrations operator's professional diseases develop quite often [8, 9, 10, 12]. Therefore, in modern tractors, high attention is concentrated on the protection of operator from noises and vibrations generated by engine, chassis and operating machines.

Operator protection, decreasing of vibration load of aggregates, control and automation systems are implemented by means of various frame, cabin and seat suspensions. In cabin suspensions of domestic tractors spring resilient elements, elastomer elements and its combinations are mostly used. Its constructions, locations and ways of connection to the frame or transmission case, floor or cabin's post are different for all machines [1, 2, 3, 9].

In the cabin suspension of tractors DT and VT series produced by VGTZ elastomeric vibroisolators [8, 9, 12] are used (fig. 1).

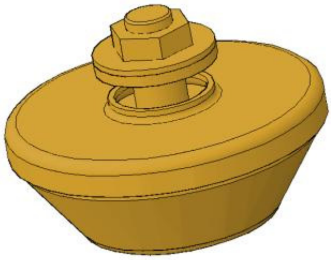

Fig. 1. Cabin vibroisolator of tractors DT and VT series

Improvement of its constructions and its elastic-damping characteristics hasn't been done during design of machines new generations. Material of elastomer and construction of vibroisolator hadn't been changed during all period of tractors producing. Vibroisolators include monolith rubber block and have limited opportunities to change its elastic parameters adaptively during acting of vertical and lateral vibration loads with various frequencies [1-7, 10, 12].

It is showed in the range of scientific works that the elastic-damping properties of this vibroisolators don't allow to provide comfortable conditions for operator work because their abilities of vibration damping aren't enough especially at low-frequency vibrations. Decreasing of vibration load level on tractor operator's working place by means of using of vibrations dynamic dampers in a cabin suspension (in other words – dynamic absorbers or dynamic vibroisolators). Dynamic vibration dampers installed in the cabin suspension were proposed by authors for decreasing of vibration load level on operator workplace. These dampers represent devices included in the structure of vibrating system for damping of vibrations with specific frequencies and partially absorb vibration energy. They consist of additional masses connected to main mass of vibrating system due to elastic-damping elements. Value of additional mass and elastic and damping parameters of damper are calculated at design stage in that way to provide maximal effective damping of vibrations of main mass with specific frequencies [9, 11, 13].

Several variants of dynamic vibroisolators schemes were proposed by authors. These schemes differ in the quantity of additional masses (one, two or three) and type of connection between additional masses and main mass due elastic-damping elements (parallel or in series) (fig. 2).

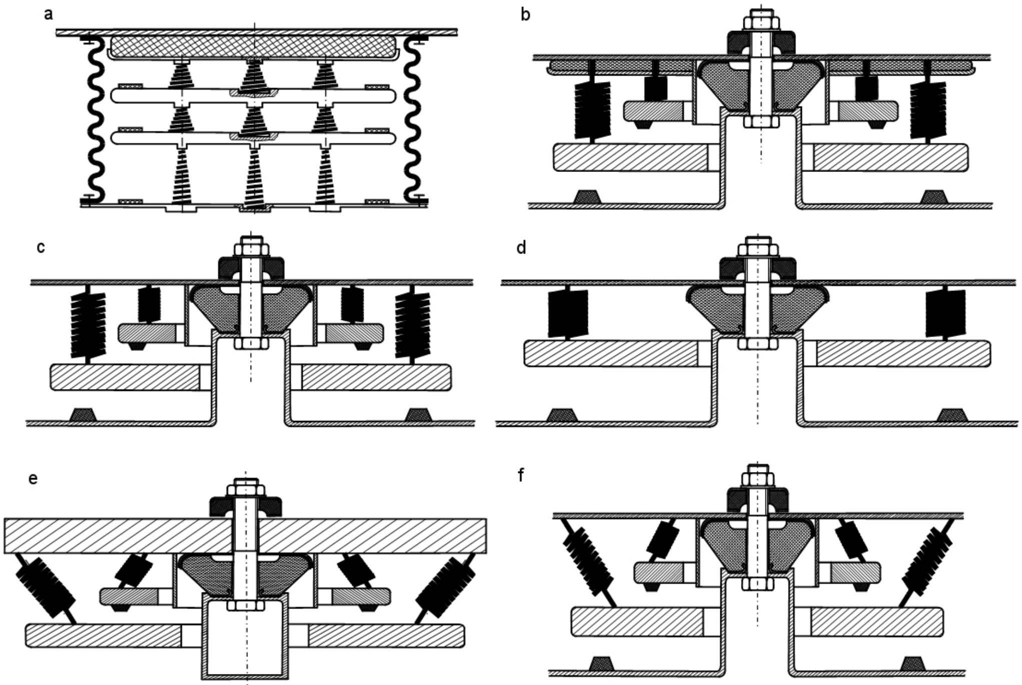

Fig. 2. Schemes of dynamic vibroisolators: (a) three masses in line and conic springs; (b) three parallel masses, elastomer and standard vibroisolator; (c) two parallel masses and standard vibroisolator; (d) one mass and standard vibroisolator; (e) two parallel masses, cabin's mass, standard vibroisolator and springs installed angularly; (f) two parallel masses, standard vibroisolator and spring installed angularly

For verification of efficiency of dynamic vibration dampers, scheme presented on fig. 3 was chosen as main [13].

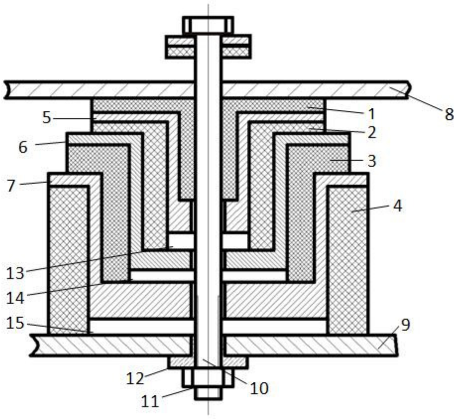

Fig. 3. Scheme of proposed engineering solution of vibroisolator

It consists of [13] coaxially located between sprung object 8 (cabin) first 1, second 2, third 3 and fourth 4 elastic-damping elements and first 5, second 6 and third 7 moveable masses. Pack of elastic-damping elements and masses located between them is fastened with central bolt 10 and screw 11 and bush 12 in that way there are gaps 13, 14, 15 between movable masses 5, 6, 7 and supporting base 9 and at the same time movable masses can move on axle in range limited by these gaps and elastic deformation of elastic-damping elements connected with these masses.

Methods of spectral analysis allow marking out of basic frequencies from the specter of driving forces frequencies. At these basic frequencies sprung mass gets maximal vibration energy. Usually, it is the lowest frequencies of specter lying in the range from 0 up to 50 Hz, or as maximal up to 100 Hz [8]. On practice number of frequencies taken into account usually is limited to 2-3 (f_{o1}, f_{o2}, f_{o3}). It is believed that driving forces with higher frequencies have significantly low vibration energy [8].

Vibroisolator (fig 3) provides cabin vibration damping with three basic frequencies of specter [13]. Scheme on fig. 4 clarifies the principle of its structure and operation. There one can mark out (together with sprung cabin) 4 partial systems. First consists of spring mass 1 with elastic-damping element 2, second consists of mass 3 and elements 2 and 4, third consists of moveable mass 5 and elements 4 and 6, fourth consists of mass 4 and elements 6 and 8.

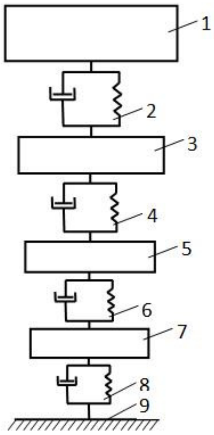

Fig. 4. Scheme of four-mass oscillating system

In second, third and forth partial systems values of moveable masses and stiffness of elastic-damping elements are chosen in that way to provide corresponding of three partial frequencies $f_{\text{п}1}$, $f_{\text{п}2}$, $f_{\text{п}3}$ of these systems and three basic frequencies f_{o1}, f_{o2}, f_{o3} of frequencies specter of operational driving forces on cabin from ground 9 (fig. 3b). Thus, in accordance with the theory of oscillations when frequency of driving force of base is equal to first basic frequency f_{o1} then movable mass 3 will oscillate with high amplitude and oscillation amplitudes of masses 5 and 7 and also springing mass 1 (this is very important) will be minimal. Accordingly when the frequency of driving force of base is equal to the second basic frequency f_{o2} then movable mass 5 will have high amplitude but masses 3 and 7 and also springing mass 1 will oscillate lightly. The same way when the frequency of driving force of base is equal to the third basic frequency f_{o3} then movable mass 7 will have high amplitude but masses 5 and 7 and also springing mass 1 will be quite motionless.

Three-dimensional model [11, 12] of tracked chassis, suspension, engine, cabin and operator seat of the tractor was created in software “Universal mechanism” for performing of computational research (fig. 4a).

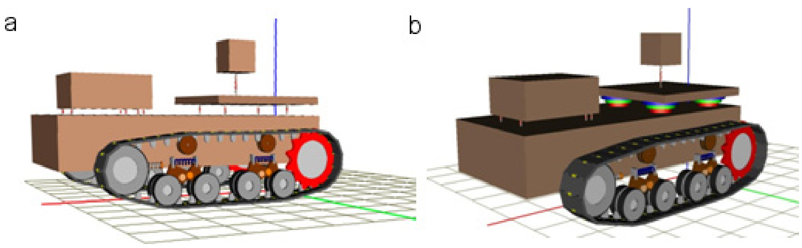

Fig. 5. (a) Tractor 3D-model with standard damper; (b) Tractor 3D-model with dynamic dampers

On this model researches of vibration of sprung mass were made for tractor motion with and without hook load at 3rd and 7th gears on a smooth surface and on testing areas with single, periodic and random bumps [11, 12].

Examples of oscillograms obtained are presented on fig. 6a. As long as accelerations of operator's workspace during vertical and lateral vibrations are the most important for operator's work conditions, oscillograms of cabin and seat accelerations are presented on fig. 6b. Graphs for standard isolators are marked with number 1 and for dynamic vibration isolator - with number 2.

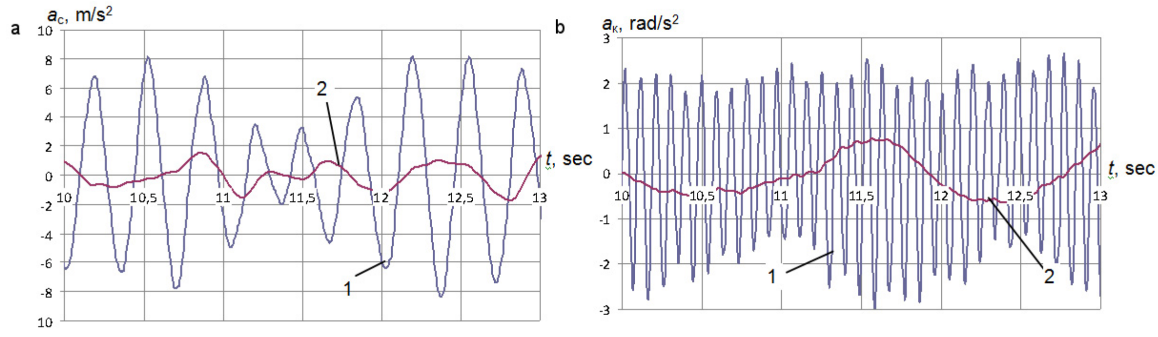

Fig. 6. (a) oscillograms of seat vertical accelerations for periodic bump - 7th gear with hook load; (b) oscillograms of cabin angular accelerations - random bump 3rd gear without hook load.

Comparison of all oscillograms obtained shows that installations of dynamic vibroisolators decreases vertical and angular accelerations of cabin and seat at motion regimes considered (table 1 and fig. 7). At this time [12]:

Table 1

Seat vertical vibrations frequency, Hz	2		3		10
aw st/aw dyn	1,5		3,5		4
Cabin vertical vibrations frequency, Hz	4	7	11	17	18
aw st/aw dyn	1,7	2,5	8,4	9,6	10,0
Angular cabin and seat vibrations frequency, Hz	3	5	11	14	17
aw st/aw dyn	2,5	2,8	3,6	4,9	8,0

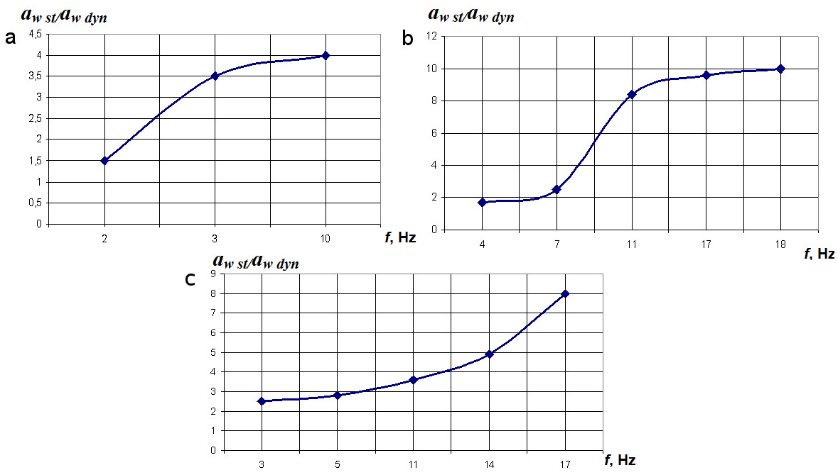

Fig. 7. Comparison of efficiency of cabin suspension with standard and dynamic vibroisolators: (a) decreasing of seat vertical acceleration amplitudes; (b) decreasing of cabin vertical acceleration amplitudes; (c) decreasing of seat and cabin angular acceleration amplitudes

- seat vertical accelerations at frequency 2 Hz became 1.5 times lower, at frequency 3 Hz – 3.5 times, at frequency 10 Hz – 4 times;
- cabin vertical accelerations at frequency 4 Hz became 1.7 times lower, at frequency 7 Hz – 2.5 times, at frequency 11 Hz – 8.4 times, at frequency 17 Hz – 9.6 times, at frequency 18 Hz – 10 times;
- cabin and seat linear-angle accelerations at frequency 3 Hz became 2.5 times lower, at frequency 5 Hz – 2.8 times, at frequency 11 Hz – 3.6 times, at frequency 14 Hz – 4.9 times, at frequency 17 Hz – 8 times;

Thus research results indicate undoubtedly better vibroprotective properties of cabin suspensions with dynamic vibration isolators. Consequently, using of these systems provides decreasing of vibration loads on tractor operator's workspace.

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