Calculated evaluation of the efficiency of dynamic vibration isolators of the tractor cab suspension system

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Abstract. The article gives a brief analysis of vibration-protective properties of standard rubber monoblock vibration isolators used in the systems of cabin suspension of domestic tracked tractors. The advantages of dynamic vibration dampers are described. The dynamic model of the tractor with standard and dynamic vibration isolators is described. The article presents the review of the methods of performed design researches and the analysis of their results.

Introduction

The quality of the cabin's suspension system has a significant impact on the vibration load of the traction vehicle operator's workplace. In these systems of domestic tracked agricultural tractors in most cases monoblock rubber vibration isolators are used (Fig. 1). They are installed between the frame and the floor of the cabin, have the most simple design and low cost [1, 2].

During operation, a wide range of dynamic influences is transmitted to the tractor frame from the chassis system due to changes in traction resistance, tractor speed and direction of movement, from vertical, longitudinal and transverse angular oscillations of the scaffolding on the suspension and from rewinding of the tangential track. The frame also receives both high-frequency and low-frequency effects from the engine mounted on it, caused by changes in its speed and generated torque, as well as by the motor oscillations on its elastic supports. The task of the cab suspension system vibration isolators is to prevent this flow of disturbances and to provide the most complete vibration isolation of the cab floor as possible. Practice shows [3]–[18] that vibration isolators made of elastomer have not bad enough vibration-protective properties at the vibrations of cabins with high and medium frequencies and significantly worse protect the operator at low-frequency vibrations. In order to protect against such vibrations, a large, at least tens of millimeters, elastic course of the vibration isolator should be provided, while elastomeric vibration isolators provide an order of magnitude less elastic course. The best vibration protection properties are provided by vibration isolators with pneumatic elements with nonlinear elastic
Fig. 2. Schemes of dynamic vibration isolators: a — with three consecutively arranged masses and conical springs; b — with three parallel masses, elastomer and standard vibration isolator; c — with one mass and standard vibration isolator; d — with two parallel masses, cabin mass, standard vibration isolator and springs installed at an angle; e — with two parallel masses, standard vibration isolator and springs installed at an angle; f — with three parallel masses, elastomer and standard vibration isolator; g — with three parallel masses, cabin mass, standard vibration isolator and springs installed at an angle.
and damping characteristics, but they also do not provide the most effective protection in the entire range of frequencies of operational vibration effects, but only in a certain band of frequencies, which can be changed, if the software control the pressure in the cavities of the pneumatic element, that is, its rigidity. Such control systems are developed, applied on the best samples of foreign equipment and are gradually introduced into domestic equipment. They have a complex design and quite a high cost. It is possible to unite positive qualities of vibroisolators from an elastomer and vibroisolators with pneumoelement at the expense of use of the devices providing dynamic damping of fluctuations.

**Schematic solutions for dynamic vibration isolators**

Dynamic vibration dampers (Fig. 2) are devices introduced into the vibrating system to dampen oscillations with certain frequencies and for partial absorption of vibrational energy [8, 9].

They include additional masses associated with the main oscillating mass of the system through elastic damping elements. The additional masses of the damper and the elastic and damping properties of its components are designed to absorb the vibrations of the main mass at certain frequencies as effectively as possible. The authors propose several variants of dynamic vibration isolators with one, two and three additional masses connected to the main mass through elastic damping elements in parallel or in series (Fig. 2). Elastomeric elements are installed in these vibration isolators to damp high frequency vibrations, and even two such elements are installed in scheme b. The vibration isolator in the scheme a is equipped with conical screw protrusions, allowing to obtain nonlinear elastic characteristics. In vibration isolators on e and f circuits, the springs are set at an angle, which ensures that both vertical and lateral loading influences can be absorbed. In order to prevent hard impacts that may occur during operation when the moving masses are in contact with each other, the vibration isolators are equipped with dynamic stroke limiters. All vibration isolators, except those shown in diagram d, are equipped with two additional masses. The value of these masses, as well as the rigidity of their elastic bonds are calculated from the condition of damping of vibrations of a certain frequency band. These can be sufficiently wide bands in the spectrum of vibration effects on the frame, which are most intensely manifested operational loads. Thus, vibration isolators provide effective protection in two usually low-frequency ranges, in which monoblock elastomeric vibration isolators are ineffective. The number of moving masses can be large; in this case, vibration damping is provided in a wider frequency range, but the overall dimensions of the insulator are increased, which becomes problematic when installed between the frame and the floor of the cabin.

**Dynamic model and calculation methods**

In order to analyze the efficiency of using dynamic vibration isolators in the cabin suspension system with the use of the "Universal Mechanism" program package, a spatial model [19]-[28] of the tracked running system and the systems of suspension of the skeleton, engine, cabin and seat of the tracked agricultural tractor on the basis of Agromash-90TG was created (Fig. 3).

![Fig. 3. The spatial model: a — with standard; b — with dynamic dampers](image-url)
When modeling, we studied the vibrations of the sprung masses during the movement of the tractor with a hook load and without load on all the gears on a flat surface and on the polygons with single, periodic and random irregularities [20, 23].

**Research results**

As a result of a wide range of studies, a complete set of digital oscillograms was obtained, which show the movements, speeds and accelerations of moving masses of machine models with standard and dynamic dampers. Samples of the obtained oscillograms are shown in Fig. 4. Due to the fact that the most important for the working conditions of the operator are the accelerations of the workplace at the vertical and longitudinal angular oscillations, Fig. 4 The oscillograms of changes in cabin and seat accelerations are shown as an example. Acceleration diagrams of standard vibration isolators are marked with number 1, for dynamic — with number 2.

![Fig. 4. Examples of compared oscillograms](image)

**Analysis of research results**

The comparison of the whole set of the obtained oscillograms testifies to the fact that the vertical and longitudinal-angle accelerations of the cabin and seat decrease when installing dynamic vibration isolators of the cabin in all considered cases of motion. Table 1 and the graphs in Fig. 5 for example, the ratios of seat and cabin mass accelerations during installation of standard vibration isolators to accelerations during installation of dynamic vibrations with characteristic frequencies of vibration are given [22]:

![Table 1. The vertical vibration of the seat](table)

| Frequency, Hz | 2 | 3 | 10 |
|--------------|---|---|----|
| $a_c$, rad/c² | 1,5 | 3,5 | 4 |

![Table 1. The vertical vibration of the cabin](table)

| Frequency, Hz | 4 | 7 | 11 | 17 | 18 |
|--------------|---|---|----|----|----|
| $a_c$, rad/c² | 1,7 | 2,5 | 8,4 | 9,6 | 10,0 |

![Table 1. The angular vibration of the cabin and seat](table)

| Frequency, Hz | 3 | 5 | 11 | 14 | 17 |
|--------------|---|---|----|----|----|
| $a_{a_k}$, rad/c² | 2,5 | 2,8 | 3,6 | 4,9 | 8,0 |
– the vertical accelerations of the seat at 2 Hz reduce by 1.5 times, at 3 Hz — by 3.5 times, at 10 Hz — by 4 times;
– the vertical acceleration of the cabin at 4 Hz decreases 1.7 times, at 7 Hz — 2.5 times, at 11 Hz — 8.4 times, at 17 Hz — 9.6 times, at 18 Hz — 10 times;
– the longitudinal-angular acceleration of the cabin and seat at the frequency of 3 Hz is reduced by 2.5 times, at the frequency of 5 Hz — by 2.8 times, at the frequency of 11 Hz — by 3.6 times, at the frequency of 14 Hz — by 4.9 times, at the frequency of 17 Hz — by 8 times.

Fig. 5. The comparison of the efficiency of the cabin suspension systems with the standard and dynamic vibration isolators: a — reduction of the amplitudes of vertical accelerations of the seat; b — reduction of the amplitudes of vertical accelerations of the cabin; c — reduction of the amplitudes of angular accelerations of the seat and cabin

Conclusions
The results of the conducted researches testify to the undoubtedly best vibro-protective properties of cabins suspension systems with dynamic vibration isolators. When using such systems in almost all possible operating modes, the vibration level of the tractor operator's workplace is reduced by several times.

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