Calculated and experimental tests of dynamic vibration isolators for use in the suspension system of the traction vehicle cabin

V Shekhovtsov\textsuperscript{1,2}, M Lyashenko\textsuperscript{1}, P Potapov\textsuperscript{1}, A Diakov\textsuperscript{2} and K Evseev\textsuperscript{2}

\textsuperscript{1}Volgograd State Technical University, Volgograd, Russian Federation
\textsuperscript{2}Bauman Moscow State Technical University
\textsuperscript{3}E-mail: shehovtsov@vstu.ru

Abstract. The article gives a brief description of operational vibration effects on the tractor cab, protective qualities of standard rubber monoblock vibration isolators used in the systems of cabin suspension of domestic tracked tractors. The principle of operation and advantages of dynamic vibration dampers are described. The description of methods and results of calculation and experimental studies of vibration insulating properties of standard and dynamic vibration isolators is given.

Introduction
The analysis of literature sources [in particular, 1] shows that caterpillar agricultural tractors on ploughing, cultivation and sowing, i.e. on the basic agrotechnical operations have a narrow bandwidth spectrum of traction frequencies, in which pronounced peaks in the frequency ranges of 3-3.5 Hz, 10-13 Hz, 14-16 Hz, 18-20 Hz and 28-32 Hz are observed. The spectral density of traction resistance and spectral density of acceleration of the frame and cab characterize different, but interrelated vibrational processes in the dynamic system of the tractor unit. Loads at these frequencies are transferred from the running system to the frame through the suspension of the chassis and then to the floor of the cab through the suspension system. The task of the suspension system is to provide maximum vibration isolation of the operator's workplace from these loads.

The systems of cabin suspension of domestic tracked tractors in many cases still use rubber monoblock vibration isolators [2, 3, 4]. Four such vibration isolators are installed between the frame and the cab floor. The free running of the cab during the vibrations from the above loads is a few millimeters [3, 4, 5], which is enough to protect the operator in high-frequency vibrations, but not enough in low and medium frequency vibrations. There is widespread evidence in the literature that the elastic damping properties of such vibration isolators do not provide a quality protection of the cab from vibrations. The analysis of sources [5–8] shows that during the operation of tractor aggregates low-frequency vibrations from the tractor frame pass through the cab suspension practically without damping, and at some frequencies even with amplification.

Dynamic dampers [8–14] are successfully used in mechanical engineering to dampen narrowband vibrations. The authors have analyzed the possibilities and prospects of using such dynamic vibration isolators in the cabin suspension system. They include additional masses associated with the main oscillating mass of the system through elastic damping elements. The size of the additional masses of
the damper and the elastic and damping properties of its elements in the design are calculated in such a way as to provide the most effective damping of the vibrations of the base mass at certain frequencies.

The vibration isolator [11], the scheme of which is shown in Fig. 1, provides damping of cabin vibrations with three main frequencies of the spectrum of vibration [13, 14]. It is possible to distinguish 4 partial systems in this scheme (together with the suspended cabin). The first is suspension mass 1 with an elastic damping element 2, the second is mobile mass 3 with elements 2 and 4, the third is mobile mass 5 with elements 4 and 6, the fourth is mobile mass 7 with elements 6 and 8. This dynamic system has three own \( (f_{01}, f_{02} \text{ and } f_{03}) \) and two partial \( (f_{p1}, f_{p2} \text{ and } f_{p3}) \) frequencies of vibration, which generally differ from each other.

The following example shows how the vibration isolator works. Let's assume that the partial frequency \( f_{p1} \) has a partial system with elastic damping element 2 and suspended mass 1, and the partial frequency \( f_{p2} \) - a partial system with element 4 and mobile mass 3. According to the theory of oscillation, under the action of vibration loads from one of these frequencies, the mobile mass, the partial frequency of which is equal to the frequency of action, and the suspended mass remains practically stationary, makes vibration with high amplitude [13, 14]. That is, in the case of action on the side of the base 9 on the mobile mass 3 through the elements 4, 6 and 8 with mobile masses 5 and 7 perturbations with a frequency of \( f_{p2} \), the amplitude of vibrations of the suspended object 1 will be insignificant, and the amplitude of vibrations of the mobile mass 3 — a significant [10].

At the same time, the mobile mass 3, which vibrates with a large amplitude, acts on the suspended mass 1 force, which is almost equal in amplitude and in phase opposite to the external force of the support base 9. Thus, the dynamic damper compensates the effect of external force on the suspended mass 1.

**Fig. 1.** Scheme of a four-mass dynamic vibration isolator

In the second, third and fourth partial systems the values of mobile masses and rigidity of elastic damping elements are selected in such a way that the partial frequencies \( f_{p1}, f_{p2} \text{ and } f_{p3} \) of these systems correspond to the three basic frequencies \( f_{01}, f_{02} \text{ and } f_{03} \) from the spectrum of frequencies of operational influences on the cabin from the side of the base 9. Then under the action of disturbances on the side of the base 9 with the first basic frequency \( f_{01} \) the mobile mass 3 will make vibrations with a significant amplitude, and the amplitudes of vibrations of mobile masses 5 and 7, as well as the suspended mass 1 (which is important!) will be minimal. Accordingly, under the action of disturbances with the second main frequency \( f_{02} \) with a significant amplitude the mobile mass 5 will oscillate, and the mobile masses 3, 7 and the suspended mass 1 will make insignificant oscillatory
movements; under the action of disturbances with the third main frequency $f_{03}$ with a significant amplitude the mobile mass 7 will oscillate, and the mobile masses 3, 5 and the suspended mass 1 will be almost motionless.

**Selection of dynamic damper parameters and methods of research**

It is necessary to know the inertial and elastic damping parameters of the elements of each circuit in order to effectively dampen vibrations in given ranges. For this purpose the program [10, 12] based on MatLab package is created. It allows changing these parameters with a given step and for each variant partial frequencies and basic frequencies of the operational spectrum were compared. Thus, the size of mobile masses and elastic course of elements is limited by values of 20 kg and 40 mm. In addition, the value of the dynamic coupling coefficient of mass oscillations is determined $\gamma$ [15]. The lower the coefficient, the better, i.e. the specified frequency of vibration loading "works out" one mass, while the others fluctuate significantly less. As a result of modeling, close correspondence of partial and main frequencies of vibrations is received on the second and third frequencies, on the first one it is not received as the value of one of masses thus should be more than 20 kg. As a result, 25 constructively realizable variants were chosen [10], parameters of the best 2 of them are given in Table 1.

| №  | $f_{p1}$ | $f_{p2}$ | $f_{p3}$ | $f_{p4}$ | $f_{o1}$ | $f_{o2}$ | $f_{o3}$ | $f_{o4}$ | $m_1$ | $m_2$ | $m_3$ | $m_4$ | $c_1$ | $c_2$ | $c_3$ | $c_4$ | $\gamma$ |
|----|---------|---------|---------|---------|---------|---------|---------|---------|------|------|------|------|------|------|------|------|-------|
| 1  | 8       | 13      | 30      | 161     | 1,22    | 13,02   | 30,02   | 163,62  | 7    | 6    | 2    | 200  | 22154 | 87697 | 73650  | 2021295 | 0,3577 |
| 25 | 7       | 15      | 30      | 101     | 1,82    | 15,02   | 30,02   | 95,82   | 18   | 14   | 20   | 200  | 47532 | 130194 | 2773677 | 955378   | 0,5582 |

Table 1 has the next indications: $f_{p i}$ — partial frequency of $i$-mass, Hz; $f_{o i}$ — $i$-th basic frequency of the spectrum of operational impacts, Hz; $m_i$ — $i$-th mobile mass; $c_i$ — stiffness of $i$-th elastic element of vibration isolator, N/m; $\gamma$ — coefficient of dynamic connection of mass oscillations

**Calculated researches of efficiency of dynamic dampers**

The models of standard and dynamic vibration isolators (Fig. 2) loaded with their cabin weight have been created and their operation at different frequencies has been studied [10] to check the effectiveness of such dampers in the "Universal Mechanism" package.

![Fig. 2. Dynamic models of vibration isolators: a — standard; b — dynamic](image_url)

Our study showed that the standard vibration isolator in the frequency range from 1 to 13 Hz does not provide damping, and in most cases even enhances the vibratory processes (Table 2). Only starting from 30 Hz is it possible to reduce vibration movements, speeds and accelerations. Both dynamic dampers from 5 Hz onwards provide significantly better vibration protection. So, for example, in the
second variant (Table 3) at frequencies of 2, 5, 9, 13 and 30 Hz acceleration decreases accordingly by
20, 92, 96, 96 and 99 %.

Table 2.

| $f$, Hz | $A_{frame}$, mm | $A_c$, mm | $V_{frame}$, mm/sec | $V_c$, mm/sec | $a_{frame}$, mm/sec$^2$ | $a_c$, mm/sec$^2$ |
|---------|------------------|------------|---------------------|---------------|------------------------|-----------------|
| 1       | 0,3              | 0,3        | 2                   | 2             | 13                     | 13              |
| 2       | 0,3              | 0,3        | 3,8                 | 4             | 47                     | 52              |
| 5       | 0,3              | 0,34       | 9                   | 11            | 290                    | 340             |
| 9       | 0,3              | 0,5        | 15                  | 30            | 800                    | 1600            |
| 13      | 0,3              | 1,2        | 21                  | 100           | 1600                   | 8000            |
| 30      | 0,3              | 0,05       | 40                  | 12            | 8000                   | 2000            |

Table 2 has the next indications: $f$ — frequency of influences, Hz; $A_{frame}$ — amplitude of the base, i.e. tractor frame, mm; $A_c$ — amplitude of the cabin, i.e. the suspended mass, mm; $V_{frame}$ и $V_c$ — speed of movement of the frame and cabin, mm/sec; $a_{frame}$ и $a_c$ — acceleration of movement of the frame and cabin, mm/sec$^2$.

Table 3

| $f$ | $A_{m1}$ | $A_{m2}$ | $A_{m3}$ | $A_{m4}$ | $V_{frame}$ | $V_{m1}$ | $V_{m2}$ | $V_{m3}$ | $V_{m4}$ | $a_{frame}$ | $a_{m1}$ | $a_{m2}$ | $a_{m3}$ | $a_{m4}$ |
|-----|----------|----------|----------|----------|-------------|----------|----------|----------|----------|-------------|----------|----------|----------|----------|
| 1   | 1,0      | 0,7      | 0,7      | 0,7      | 0,65        | 0,6      | 0,8      | 0,8      | 0,8      | 5,5         | 5,5      | 5,5      | 5,5      | 5,5      |
| 2   | 1,0      | 0,4      | 0,6      | 0,6      | 0,6         | 1,25     | 6        | 8        | 8        | 9           | 15       | 75       | 105      | 105      |
| 5   | 1,0      | 0,04     | 0,35     | 0,35     | 0,4         | 3        | 0,6      | 0,4      | 0,5      | 0,6         | 100      | 17       | 13       | 15       |
| 9   | 1,0      | 0,07     | 0,03     | 0,05     | 0,15        | 5,5      | 2        | 0,04     | 0,15     | 0,4         | 5,5      | 2        | 0,04     | 0,15     |
| 13  | 1,0      | 0,1      | 0,2      | 0,12     | 0,13        | 7        | 8        | 0,6      | 0,3      | 0,8         | 600      | 600      | 55       | 55       |
| 30  | 1,0      | 0,15     | 0,19     | 0,017    | 0,007       | 17       | 4        | 6        | 0,15     | 0,045       | 3200     | 560      | 790      | 26       |

Table 3 has the next indications: $f$ — frequency of influences, Hz; $A_{m1}$, mm, $V_{m1}$, mm/sec, and $a_{m1}$, mm/sec$^2$ — amplitude, speed and acceleration of vibrations of the $i$-th mobile mass.

Thus, the calculated researches have shown that the offered dynamic damper in comparison with the standard vibroisolator possesses essentially better vibroisolating qualities.

Experimental studies of dynamic dampers

An experimental installation (Fig. 3, a), simulating the dynamic system of one point of suspension of the cabin [6, 7] for experimental testing of the vibration insulating properties of the suspension with dynamic vibration isolators, and for the calculated test — its dynamic model (Fig. 3, b) [10]. The vibrations in the range of 0-20 Hz are caused by an inertial loader. This loader is connected to a swinging bench lever that simulates a cab floor. During the tests at each excitation frequency, movements, speeds and acceleration of the cabin floor and moving masses were measured and recorded using SVAN equipment, on the basis of which a series of amplitude-frequency characteristics of vertical and longitudinal-angle accelerations of the cabin and seat without dynamic dampers and with dampers was constructed.

Since dynamic vibration isolator provides the most effective damping of vibrations when the natural frequency of vibration of its mass coincides with its own frequency of vibration of the suspended mass and with the frequency of external influence, the main experimental and calculated studies of the effectiveness of vibration isolators are carried out on such resonance modes, and video recording of processes was also carried out. Analysis of these records shows that in the resonance mode, vibroisolator mass and cabin mass vibrations occur in the antiphase (Fig. 4). Due to this, the
dynamic damper reduces the vibration load on the cab. This is confirmed [10] by the comparison of the amplitude-frequency characteristics of the cabin vibrating systems without a dynamic damper (Fig. 5) and with the damper adjusted to the own frequency of the cabin vibration (Fig. 6).

![Fig. 3.](image)

**Fig. 3.** a — experimental installation; b — dynamic model

The analysis of the amplitude-frequency response shows that, according to the calculated data, the use of dynamic dampers in the resonance mode reduces the vertical acceleration of the cabin by 40% (Fig. 5), and in accordance with the experimental data — by 49% (Fig. 6).

Dynamic dampers with these parameters are introduced into the model of cabin suspension and the same complex of researches as with standard vibration isolators is executed [10]. For comparison, some of the oscillograms for standard and dynamic vibration isolators are shown in Fig. 7, and the diagrams for standard vibration isolators are indicated by the number 1, for dynamic — 2.

Comparison of the whole set of the obtained oscillograms shows that [10], that the installation of dynamic cabin vibration isolators in all cases of motion considered, vertical and longitudinal-angle acceleration of the cabin and seat are reduced (Fig. 8). At the same time:

- the vertical accelerations of the seat at the frequency of 2 Hz decrease by 1.5 times, at the frequency of 3 Hz — by 3.5 times, at the frequency of 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.

The analysis of sources [16]–[26] shows that during the operation of tractor aggregates low-frequency vibrations from the tractor frame pass through the cab suspension practically without damping, and at some frequencies even with amplification.
Fig. 5. Calculated amplitude-frequency characteristics of the system without dynamic damper and with damper

Fig. 6. Experimental amplitude-frequency characteristics of the system without dynamic damper and with damper
Fig. 7. Examples of compared oscillograms

Fig. 8. Decrease in amplitudes: a — vertical accelerations of the seat (1) and cabin (2); b — longitudinal-angle accelerations of the cabin with the seat

Conclusions

Thus, both calculated and experimental researches show that vibro-protective properties of tractor cab suspension system with dynamic dampers appear to be essentially the best in the range of frequencies of operational influences. Practically on all modes of movement of the tractor unit this system provides better vibration protection of the operator's workplace than the system with standard rubber vibration isolators.

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References
[1] Kuznetsov, N.G. Stabilization of operation modes of high-speed machine-and-tractor units. Monograph / N.G. Kuznetsov; Volgogr. — Volgograd, 2006. — 426 p.
[2] Technical solutions of the elastic damping devices of the tractor-rail cabin suspension / V.V. Shekhovtsov, M.V. Lyashenko, V.P. Shevchuk, A.V. Pobedin, N.P. Sokolov-Dobrev, K.V. Shekhovtsov // International research journal. — 2013. — № 7 (ч. 2). — P. 122–125.
[3] Modeling, simulation, and validation of a pendulum-pounding tuned mass damper for vibration control Structural Control and Health Monitoring/Wang, W.a, Hua, X.aEmail Author, Chen, Z.a, Wang, X.b, Song, G.c Volume 26, Issue 4, April 2019.
[4] Prediction of the vibration characteristics for wheeled tractor with suspended driver seat including air spring and MR damper Zheng, E.Email Author, Fan, Y., Zhu, R., Zhu, Y.,
Xian, J. /Journal of Mechanical Science and Technology Volume 30, Issue 9, 1 September 2016, Pages 4143–4156.

[5] Experimental determination of the characteristics of the tractor cab vibration isolators / V.V. Shekhovtsov, M.V. Lyashenko, V.P. Shechuk, N.P. Sokolov-Dobrev, K.V. Shekhovtsov // International research journal. — 2013. — № 7 (u. 2). — P. 118–122.

[6] Lyashenko, M.V. Laboratory installation for testing vibration isolators / M.V. Lyashenko, A.V. Pobedin, K.V. Shekhovtsov // Vestnik of the Academy of Military Sciences. — 2011. No. 2 (special issue). — P. 270–274.

[7] Tractor cabin vibration isolator test bench equipment / A.V. Pobedin, M.V. Lyashenko, K.V. Shehovtsov, Z.A. Gojaev // Tractors and agricultural machinery. — 2012. — № 7. — P. 43–48.

[8] Evaluation of the vibration-protection properties of the standard and dynamic cabin suspension systems in accordance with the requirements of the standards / K.V. Shehovtsov, A.V. Pobedin, N.P. Sokolov-Dobrev, M.V. Lyashenko, V.V. Shekhovtsov // Izvestia VolgGTU. Terrestrial Trans-Port Systems’ series. Issue. 7: Interuniversity collection of scientific articles / VolgGTU. — Volgograd, 2013. — № 21 (124). — C. 53–55.

[9] Voloshin, Yu.V. Application of the suspension systems in the foreign tractors // Tractors and agricultural machines. — 2000, — № 2. — P. 36.

[10] Shekhovtsov, K.V. Reduction of the level of vibration load of the tractor operator's workplace due to the use of dynamic vibration dampers in the system of cabin suspension: PhD. — Volgograd, VolgTU, 2014. — 17 p.

[11] The utility model 136110 RUS, F16F3/087 / Vibroisolator of the vehicle cabin / K.V. Shehovtsov, N.S. Sokolov-Dobrev, A.V. Pobedin, V.P. Shechuk, M.V. Lyashenko, V.V. Shekhovtsov; Volgogr. — 2013.

[12] Shekhovtsov, K.V. Tractor cabins suspension using the dynamic vibration dampers / Shekhovtsov, K.V.; Pobedin, A.V.; Sokolov-Dobrev, N.S.; Shekhovtsov, V.V. // Izvestia VolgGTU. Ground transportation systems series. Issue. 6: Interuniversity collection of scientific station / VolgGTU. — Volgograd, 2013. — № 10. — P. 43–46.

[13] Karamyshkin, V.V. Dynamic damping of vibrations. — L.: Mechanical engineering, Leningrad region, 1988. — P. 105.

[14] Korenev B.G., Reznikov L.M. Dynamic vibration dampers: Theory and technical applications. — Moscow: Nauka, 1988. — P. 302

[15] Influence of elements dynamic cohesiveness in power shafting on torsional vibrations spreading and dynamic equality of reducible model V.V. Shekhovtsov, N.S. Sokolov-Dobrev, M.V. Lyashenko, P.V. Potapov, K.V. Shekhovtsov, A.V. Kalmykov // Mechanika (Kaunas). — 2014. — Vol. 20, No. 2. — P. 190–196.

[16] Application and development challenges of dynamic damper in cabin booming noise elimination Cheni, R.K.Email Author, Jain, C.P.Email Author, Muthiah, R.Email Author, Gomatam, S. /SAE Technical Papers Volume 1, 2014 SAE 2014 World Congress and Exhibition; Detroit, MI; United States; 8 April 2014 до 10 April 2014.

[17] Analysis of driving seat vibrations in high forward speed tractors Servadio, P. , Marsili, A. , Belfiore, N.P. (2007) Biosystems Engineering.

[18] Deprez, K., Moshou, D., Anthonis, J., De Baerdemaeker, J., Ramon, H. Improvement of vibrational comfort on agricultural vehicles by passive and semi-active cabin suspensions (2005) Computers and Electronics in Agriculture, 49 (3), pp. 431–440. doi: 10.1016/j.compag.2005.08.009.

[19] Zehsaz, M., Sadeghi, M.H., Ettefagh, M.M., Shams, F. Tractor cabin's passive suspension parameters optimization via experimental and numerical methods (2011) Journal of Terramechanics, 48 (6), pp. 439–450. doi: 10.1016/j.jterra.2011.09.005.
[20] Whole body vibration exposure during rotary soil tillage operation: The relative importance of tractor velocity, draft and soil tillage depth. International Singh, A., Nawaysch, N.b, Singh, L.P.a, Singh, S.a, Singh, H.c /Journal of Automotive and Mechanical Engineering Volume 15, Issue 4, 1 December 2018, Pages 5927–5940.

[21] Tractor cabin's passive suspension parameters optimization via experimental and numerical methods. Journal of Terramechanics Volume 48, Issue 6, December 2011, Pages 439–450 Zehsaz, M., Sadeghi, M.H., Ettefagh, M.M., Shams, F.

[22] V. V. Novikov, A. V. Pozdeev, A. S. Diakov, Research and testing complex for analysis of vehicle suspension units, Procedia Engineering, Vol. 129, pp. 465–470 (2015) DOI: 10.1016/j.proeng.2015.12.153

[23] Evseev, K. B., Kartashov, A. B., Dashtiev, I. Z., Pozdeev, A. V. Analysis viscoelastic properties of fiber-reinforced composite spring for the all-terrain vehicle, MATEC Web of Conferences 224, 02039 (2018).

[24] Diakov, A. S., Pozdeev, A. V., Novikov, V. V. The main directions of the development of snowmobiles in the Russian Federation (2018) MATEC Web of Conferences, 224, № 02080.

[25] E. Klubnichkin, A. S. Dyakov, E. E. Klubnichkin, A. Yu. Zakharov, U. Sh. Vakhidov, A. S. Suchenina and I. V. Basmanov. Experimental evaluation of stability and controllability of domestic and foreign made utility terrain vehicles. IOP Conf. Series: Journal of Physics: Conf. Series 1177 (2019) 012045. DOI:10.1088/1742-6596/1177/1/012045.

[26] V. E. Klubnichkin, A. S. Dyakov, E. E. Klubnichkin, A. Yu. Zakharov, U. Sh. Vakhidov , A. S. Suchenina and I. V. Basmanov. Experimental evaluation of speed and brake properties of domestic and foreign made utility terrain vehicles. IOP Conf. Series: Journal of Physics: Conf. Series 1177 (2019) 012048. DOI:10.1088/1742-6596/1177/1/012048.