Experimental studies of the snowmobile engine rubber-metal vibration dampers specifications

Aleksey S Diakov¹ *, Aleksey V Pozdeev², Mikhail V Lyashenko³ and Iaroslav V Moskovskii³

¹Bauman Moscow State Technical University, 5/1 Baumanskaya 2-ya, Moscow, Russia, 105005
²Volgograd State Technical University, 28 Lenin av., Volgograd, Russia, 400005
³JSC «Russkaja Mechanika», st. Tolbuhina, 22, Rybinsk, Yaroslavl region, Russia, 152914

* diakov57@list.ru

Abstract. The paper presents the methods and results of experimental determination of performance curves (flexibility and amortization characteristics) of two types of snowmobile engine rubber-metal vibration dampers with their stand radial and axial loading at different vibration amplitudes and frequencies using Macron BISS-ITW tensile testing machine. The obtained results were analyzed.

1. Introduction
The engine suspension system is essential to ensure the best conditions for its operation as a part of the vehicle. The role of this system is further increased when the engine is mounted in a vehicle such as a snowmobile. The engine of such a machine is constantly working under variable loading conditions due to the virgin snow off-road movement with the snow depth and density constantly changing. Each load change is rebounded with reactions on its supports. To reduce the dynamics of changes in the reactions, the engine suspension system was designed. In this case, the system vibration insulators flexibility and amortization characteristics are crucial [1–11].

2. Test objects
Volgograd State Technical University in cooperation with Bauman Moscow State Technical University at the request of the customer — the snowmobiles manufacturer (JSC "Russian Mechanics") — have carried out a wide range of studies to experimentally determine the flexibility and amortization characteristics of two types of snowmobile internal combustion engine (ICE) rubber-metal vibration dampers with their stand radial and axial loading at different vibration amplitudes and frequencies. Figure 1 shows external view of vibration dampers.
Figure 1. Top and bottom vibration dampers external view: a, c — vibration damper No. 1; b, d — vibration damper No. 2.

Vibration dampers No. 1 are used in the front part of snowmobile engine suspension, the vibration damper No. 2 is used in the rear part of snowmobile engine suspension.

3. Testing equipment and test methods
Servohydraulic test complex based on Macron BISS-ITW tensile testing machine of Indian manufacturing for static and dynamic LF, MF and HF load objects testing was used (Figure 2).

Figure 2. General view of the testing machine (a) and front views of the experimental vibration dampers No. 1 (b, c) and No. 2 (d, e) installation on the testing machine with their axial and radial loading.
The static load on the front damper of the ICE weight is 143 N, on the rear damper (half of the double damper) — 115 N.

The testing method provided the following ranges of reproducible load change:
with following frequency ranges and intervals:
- 2...20 Hz — at 2 Hz interval;
- 20...80 Hz — at 10 Hz interval;
- 80...140 Hz — at 20 Hz interval;
with deformations of the vibration dampers (displacement of the central bushing of the vibration damper relatively to the external bushing):
- radial: 0.05/0.1/0.15/0.2/0.4/0.6/0.8/1.0/1.2 mm;
- axial: 0.05/0.1/0.15/0.2/0.4/0.6/0.8/1.0/1.2 mm.

The Table 1 shows the main technical characteristics of the testing machine.

| Main parameters                        | Measurement units | Value   |
|----------------------------------------|-------------------|---------|
| Maximum load                           | kN                | 25      |
| Distance between columns               | mm                | 400     |
| Distance between the beam and the frame (max.) | mm | 700     |
| Maximum load                           | kN                | 25      |
| Rod travel                             | mm                | ± 50    |
| Load frequency                         | Hz                | 0.200   |
| Maximum load during cyclic testing     | kN                | ± 25    |
| Full measurements range nonlinearity, max. | %     | 0.2     |
| Accuracy, max.                         | %                 | ± 0.5   |
| Resolution, max.                       | %                 | ± 0.05  |

The tests ambient temperature was 23±5 °C. The max. bench equipment error was 1.5 %. The harmonic loading mode was reproduced for test objects.

4. Test results
Figure 3 and Figure 4 show shapes of the experimental performance curves of vibration dampers Nos. 1 and 2 under axial and radial loads depending on various perturbation amplitudes and frequencies.
Figure 3. Experimental performance curves of vibration dampers under axial load depending on various \( \nu \) perturbation amplitudes and frequencies: \( a, c, e \) — vibration damper No. 1; \( b, d, f \) — vibration damper No. 2; \( a, b - \nu = 2 \, \text{Hz} \); \( c, d - \nu = 20 \, \text{Hz} \); \( e, f - \nu = 100 \, \text{Hz} \).
Figure 4. Experimental performance curves of vibration dampers under radial load depending on various \( \nu \) perturbation amplitudes and frequencies: a, c, e — vibration damper No. 1; b, d, f — vibration damper No. 2; a, b – \( \nu = 2 \) Hz; c, d – \( \nu = 20 \) Hz; e, f – \( \nu = 100 \) Hz.

5. Methodology for evaluating the results

The results obtained during stand tests are analyzed to determine the parameters of dynamic stiffness and the characteristics of the dissipative force. To do this, the following method is used:

- The maximum velocity of the vibration damper deformation \( V_{\text{max}} \) is determined. To do this, the formula specified in GOST 27242-87 [1] is used:
  \[
  V_{\text{max}} = 2\pi \nu A, \tag{1}
  \]
  where \( A \) — amplitude of vibration; \( \nu \) — frequency of vibration.

- Maximum dissipative force \( F_{\text{diss}} \) is determined. To do this, in performance curves series obtained for single vibration frequency, for each vibration amplitude we find the corresponding maximum values of the dynamic force at the compression and tension travels, which will be the maximum dissipative forces of the vibration damper:
  \[
  F_{\text{diss}} = F_{\text{max}}. \tag{2}
  \]
  The dependency \( F_{\text{diss}} = f(V_{\text{max}}) \) is developed. This dependence can also be found by the second method, using all the vibration damper performance curves recorded during the stand tests. To do this, the curves obtained at different vibration frequencies determine the values of the maximum dissipative force at the same vibration amplitude. The deformation rate is determined by the formula (1).

- The dynamic stiffness of vibration damper \( c_{\text{din}} \) is determined. To do this, the formula specified in GOST 27242-87 [1] is used:
  \[
  c_{\text{din}} = \frac{F_{\text{max}}}{A}. \tag{3}
  \]
  The dependency \( c_{\text{din}} = f(A) \) is developed. Since the vibration dampers are tested in the "accordion" mode with fixed suspended mass, the phase shift between the dynamic force of the vibration damper and its deformation is zero and, accordingly, does not affect the values \( F_{\text{max}} \) and \( c_{\text{din}} \).

- The energy \( W \), dissipated during one vibration cycle, is determined. This energy corresponds to the performance curve area. To determine it in the first approximation, you can use the well-known ImageJ or LpSquare image processing software.

- The hysteresis damping coefficient \( b \) of the vibration damper is determined. To do this, the following formula is used:
  \[
  b = \frac{W}{\pi \nu A}. \tag{4}
  \]
  The dependency \( b = f(\nu) \) is developed.
Figure 5 shows the dependences of dissipative force, dynamic stiffness and hysteresis damping coefficient of vibration damper No. 1 with axial and radial load. Dependencies are derived from the diagrams shown in Figure 3a and Figure 4a.

Figure 5. Dependences of dissipative force (a, b), dynamic stiffness (c, d) and hysteresis damping coefficient (e, f) of vibration damper No. 1 with axial and radial load:
1 — compression travel; 2 — tension travel.

6. Conclusions
Thus, using the method presented in the work, it is possible to present the specific flexibility and amortization characteristics of the snowmobile rubber-metal vibration dampers in the form traditional for suspension systems — flexibility and amortization characteristics convenient for suspension system optimization.
References

[1] GOST 27242-87 (ST SEV 5554-86) Vibration. Vibration isolators. General Testing Requirements
[2] GOST 32586-2013 Rubber and rubber reinforced vibration isolators for automotive engineering. Specification
[3] GOST 11679.1-2018 Rubber-metal shock-absorbers for instruments. Specification
[4] GOST 270-75 Rubber. Method of the determination of flexibility and tensile stress-strain properties (with amendments Nos. 1, 2, 3)
[5] GOST 265-77 Rubber. Methods of short static compression tests (with amendments Nos. 1, 2)
[6] GOST 23020-78 Rubber. Method for determination of failure work at extension (with amendment No. 1)
[7] GOST 28840-90 Machines for materials tension, compression and bending tests. General specifications
[8] Lyapunov V T, Lavendel E E, Shlyapochnikov S A Rubber vibration isolators: Reference Book — L.: Sudostroenie, 1988. — 216 pages
[9] Adam G Bowland Comparison and Analysis of the Strength, Stiffness, and Damping Characteristics of Concrete with Rubber, Latex, and Carbonate Additives, 2011, 351 p
[10] Singiresu S Rao Mechanical Vibrations (3rd edition) Addison-Wesley Pung Company 1995
[11] Roy S and Reddy J N Finite-element models of Viscoelasticity and diffusion in Ashesively Bonded Joints, International Journal for Numerical Methods in Engineering / 26 (1988) 2531–2546