Journal of Polish CIMAC
Gdansk University of Technology
The Faculty of Ocean Engineering and Ship Technology

DETERMINATION OF THE INFLUENCE OF THE STIFFNESS OF THE DIESEL ENGINE SUSPENSION CUSHIONS IN THE TERRAIN CAR

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Abstract

The article presents the results of experimental investigations carried out for different types of diesel engine suspension cushions in the terrain car. The study was performed using two types of suspension cushions in a variety of combustion engine operating conditions. The effect of engine displacement as a result of vibrations between the engine and vehicle body was tested. The endurance testing of the suspension cushions was done using the endurance testing machine Zwick Z-100. Next, we analyzed the results obtained in experimental studies of the differences between the vibration of the suspension cushions.

The results of the tests of different types of plate connectors for the engine suspension can be useful when designing propulsion systems for military vehicles and military utility terrain cars, or as shown by the author [1, 4] to modernize of the existing military vehicles stock.

Key words: *machine diagnostics, engine vibration, rubber vibroisolator, military vehicle*

1. Introduction

The purpose of experimental researches described in the article was to determine the stiffness of the diesel engine front suspension cushions. The object of research is the terrain car UAZ-31 512 (formerly 469B), with a diesel engine mounted – Andoria 4C90 by WSW Andoria, liquid-cooled, indirect fuel injection into the swirl chamber. The issue of changes in the vehicle engine is described in detail in [1-4]. The tests were performed using hard suspension cushions – "old", at which there were significant vibrations in the body of the engine at the operating idle speed and soft – "new" ones. It should be noted that the test suspension cushions are not original cushions used in this car, because gasoline engines were fitted as standard. It was necessary to adjust the suspension so that you could mount the 4C90 diesel engine, using cushions applied to this engine types, which are also in Lublin vehicle.

The researches was conducted using a specially designed device to study the movement as a result of vibrations between the engine and body, and using the endurance testing machine Zwick Z-100.

Machines, in connection with the work performed, are a subject to various external loads, derived from the reaction of the impact of the machine, as well as from its surroundings. These phenomena have diverse and evolving nature. It manifests itself as mechanical and acoustics vibrations. It has a huge and significant impact on humans, its safety and ergonomics in the use of machinery. In-vehicle vibration is the main source of propulsion system, which like the internal combustion engine performs significantly periodic cycle with variable characteristics. The kinematic excitation of vibrations caused by driving on an uneven road surface or track are additional issues in vehicle dynamics [6]. Frequency range in which the phenomena are caused by these sources, lie in different frequency bands, and so the pathway will be accordingly others [4]. Uneven road surfaces cause the force lying in a very wide band, ranging from 0.5 to 250 Hz, including a range of vibrations coming from the propulsion system [4]. Analysis of the impact of these vibrations and vehicle design modifications are one of the most important tasks thoroughly studied experimentally and theoretically.

The vibrations generated by the source are transmitted via the machine by [4]:

- special elastic-damping elements eg. tires, suspension components, etc.;
- machine kinematics system, such as steering in a car, etc.;
- structure of the machine.

Different ranges of natural frequencies determine the vibration properties of the transition path. This can be recognized in a very simplified way, that in the frequency range of 0.5÷15 Hz the main route of transmission is a special elastic-damping system, used to mitigate the effects of vibration [4]. In the engine vehicle suspension there is used the plate connector which consists mostly of two metal plates and vulcanised rubber layer. The plate connectors are made mostly of rubber: natural rubber, chloroprene, and in exceptional cases, nitrile rubber [10]. To prevent the spread of strain on the engine frame and the vehicle hull vibration, the heavy rubber cushions are mounted under the engine mounting brackets [12]. Improper distribution of stiffness and bags arrangement can cause significant engine vibration, which will be brought back to the car especially when idle.

In conditions of dynamic loading the rubber stiffness is increased, the natural frequency of the dynamic stiffness c_{dvn} is substituted to the formula, while [7]:

$$c_{dyn} = X \bullet C, \tag{1}$$

where:

c_{dyn} – dynamic stiffness,
X – the coefficient of rigidity,
C – number of degrees of freedom.

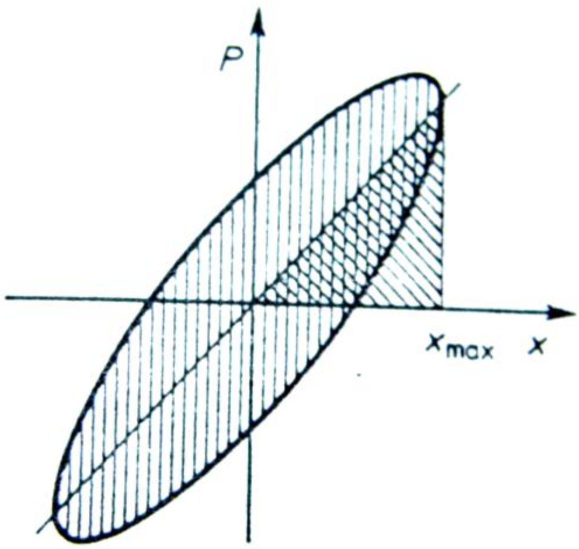

Fig. 1. The hysteresis loop with a dynamic load of rubber [7, 9]

The coefficient of rigidity X for rubber with a hardness 35...95 °ShA (IRHD) shall be within 1.1...1.4 [7]. In a system with C degrees of freedom, as for example in the automotive powertrain suspension, the frequency of natural vibration for each direction is taken into account. The behaviour of rubber under the influence of dynamic loads illustrates in the general case the graph in Fig. 1. Ascending and descending curves of strain do not coincide but form the shape of the hysteresis loop similar to the ellipse. It is also called dynamic suppression loop [7]. Area under the curve of the graph is a measure of the increasing strain energy absorbed and the surface energy of an ellipse as changed into heat [7].

2. Characteristics of the tested cushions

One of the main features of elastic rubber elements is the ability to transfer and absorb the complex loads [9]. Fig. 2 shows the used plate connectors. Cushions, despite the difference of hardness, yet different in thread ("hard" cushion has a normal thread – M8x1.25, and according to the seller information is called a substitute, produced by a private manufacturer, and "soft" – the "original" with a fine thread – M8x1). No other significant differences were found.

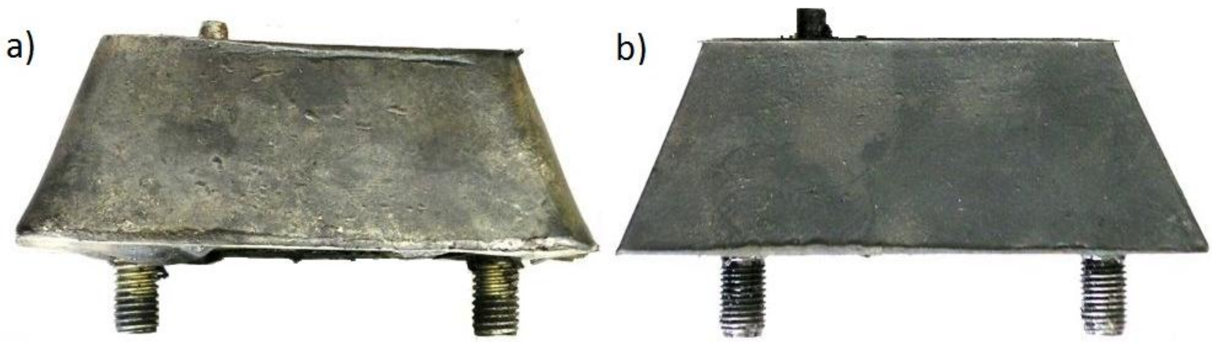

Fig. 2. The tested engine cushions in the terrain car Uaz, a) – "hard" cushion, b) – "soft" cushion

"Hard" cushions were operated in the vehicle for about 15.000 km, most in the field. During the dismounting it was found out that one of them was torn apart during use, .while "soft" cushions were new and unused.

Rubber hardness testing was carried out using Shor's method, a portable hardness tester, with the scale of 0 to 100 °Sh-A, and an accuracy of 2 °Sh-A. Average hardness for hard cushions is: cushion 1. – 79 °Sh-A, cushion 2. – 83 °Sh-A, soft cushions, respectively, cushion 3. – 61 °Sh-A and cushion 4. – 62 °Sh-A.

The flexibility study was carried out on the endurance testing machine Zwick Z-100, the selected data are summarized in Table 1.

Tab. 1. Specifications of the testing machine Zwick Z-100

Model	100SN3A WN:136119
traverse path sensor	WN:136119
force measurement head	ID:0 WN:136120
force measuring range [kN]	2,5÷100
temperature range of thermal chambers [^{o}C]	-75÷250
extensometer	electronic

When examining the cushions in the endurance testing machine, additional plates were fastened to the head in order to allow the compression tests. The test cushions have been squeezed by increasing cyclic deflection value of 2 mm. The head squeezed cushions for each cycle of 2 mm and returned to the state of 1 mm, up to 10 mm of the cushions deflection. Piston rod with which the cushions deflection was exerted, moved at a speed of V = 20 mm/min. The power that was needed to squeeze the cushions for a given value of the deflection was controlled in the studies. In this way, two hard and two soft cushions have been investigated, which after the study were fitted to the car. The resulting graphs show the hysteresis field, resulting from the energy absorbed by the cushions.

Based on the obtained results, the following formula can be used to calculate the stiffness:

$$k = \frac{F}{l}, \tag{2}$$

While the elasticity (sensitivity), we shall calculate with the formula [8]:

$$p = \frac{l}{F}, \tag{3}$$

where:

k – stiffness,
F – the force exerted by the piston rod,
l – displacement,,
p – elasticity (sensitivity).

Fig.3 shows that the force exerted by the piston on the cushion – 1 is relatively large in the last phase of the deflection, at 10mm it equals to 8467.08 N. It may be noted that the characteristics has rather large hysteresis of deflection and is progressive, what might be expected on the basis of cushion shape (trapezium). The maximum stiffness was calculated from the results: k = 846.708 N/mm, and flexibility: p = $1{,}181^{(-3)}$ mm/N.

The force exerted on the cushion – 2, amounted to 10802.1 N at deflection of 10 mm. Clearly visible large field of hysteresis at piston load and slow growth of force in the initial part of the characteristics to approximately 500 N. The maximum stiffness was calculated from the results: k = 1080.21 N/mm, and flexibility: p = $0.9257^{(-3)}$ mm/N.

The measurements of the "hard" cushions show a large difference in power between the first and second cushion at up 2335.02 N. This could be due to the difference between hardness of rubber, measured by Shor, which was 4 °Sh. We can only guess why there is such a difference in hardness between the two cushions. Presumably, cushions can be from different production batches, in which a blend of gum differed from each other or the heterogeneity of rubber used was so big. The difference of compared cushion stiffness amounted to: k = 233.502 N/mm, while the difference in flexibility was: p = $0.2553^{(-3)}$ mm/N.

It is possible that such an impact on the difference in stiffness structure was exerted due to the fact that one of them was broken, or that the cushion worked on the more side more loaded by the torque. The oil pouring from the leak engine on the cushion, which changed its structure and influence the stiffness and hardness could contribute to this as well. It can not be excluded that all of these factors could have an influence on it. The calculated mean stiffness of the cushions was as follows: k = 963.459 N/mm, while the average elasticity was: p = $1.0533^{(-3)}$ mm/N.

In the case of soft cushions (Fig. 3) we see that the maximum force exerted by the piston on the cushion – 3 was 5727.11 N at deflection of 10 mm. You can also see a fairly small field hysteresis fields. Calculated from the results of maximum stiffness k = 572.711 N/mm, while flexibility: p = $1.746^{(-3)}$ mm/N.

The maximum force at 10 mm deflection for a cushion – 4, is 6105.1 N and has a fairly small hysteresis deflection field. The maximum stiffness was calculated from the results: k = 610.51 mm/N and flexibility p = 1,638$^{(-3)}$ mm/N.

In two "soft" cushions we can see a difference in the force exerted by the piston rod, and it is 377.99 N. This could be due to the rubber hardness difference measured by the method of Shor, which was around 1 °Sh. The difference of compared cushion stiffness amounted to: k = 37.799 N/mm, while the difference in flexibility was: p = 0.108$^{(-3)}$ mm/N.

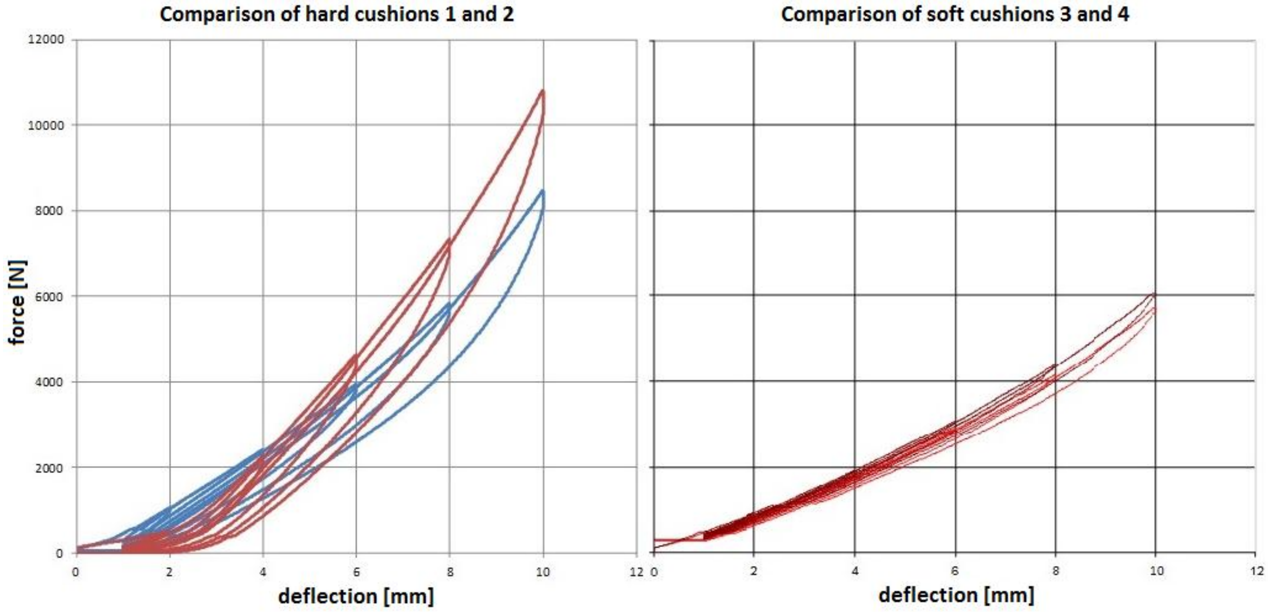

Fig. 3. Comparison of hard cushions tested 1 – blue, and 2 – red, and soft cushions 3 – red, and 4 – black

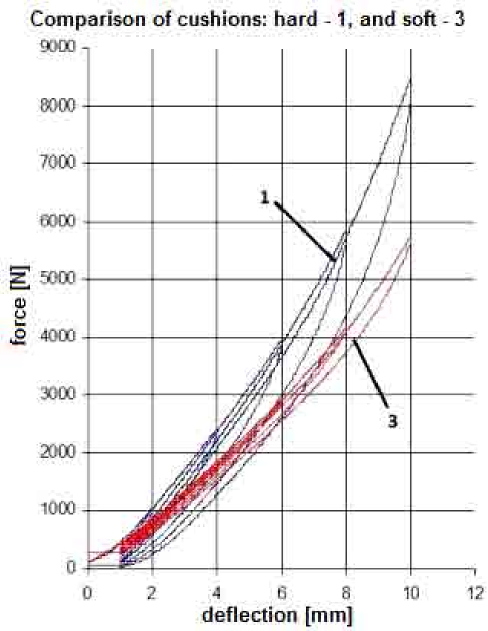

Fig. 4. Graph comparing the results of selected tested cushions, hard – 1, and soft – 3

The graph (Fig. 4) comparing the cushion 1 and 3 shows a rather large difference in strength for the same deflection. This difference is due to the large difference in hardness of used cushions. You might also notice that the hard cushion has a larger deflection hysteresis, which means that slowly returns to its initial state.

This difference is probably due to manufacturing reasons, it is possible that during the production of cushions there was a discrepancy in hardness of rubber. Probably due to the heterogeneity of the rubber used for the production of cushions. What was also evident when measuring the hardness of a cushion in many locations. Comparison of selected average values of stiffness and flexibility of hard and soft cushions are summarized in Table 2. From the comparison of average values can be seen that both the stiffness and flexibility differ by about 38% between hard and soft cushions.

Tab. 2. Comparison of selected average values of the parameters tested cushions

Type of cushion:	The mean value of Stiffness k:	The average value of Flexibility p:
Hard	963.459 N/mm	$1.0533^{(-3)}$ mm/N
Soft	591.610 N/mm	$1.6920^{(-3)}$ mm/N
Difference	371.849 N/mm 38.6%	$0.6387^{(-3)}$ mm/N 37.8%

3. Dynamic cushions researches

Dynamic cushions researches were conducted on running engine at different speeds of the crankshaft. The engine vibrations caused the internal forces of the combustion mixture were measured. Vibrations of a running engine were moved by the arm of a potentiometer (Fig. 5), which processed them and passed voltage to the oscilloscope DSO-2902 256K. The oscilloscope then saved the data after processing them, and the result can be watched on a monitor in graphical form.

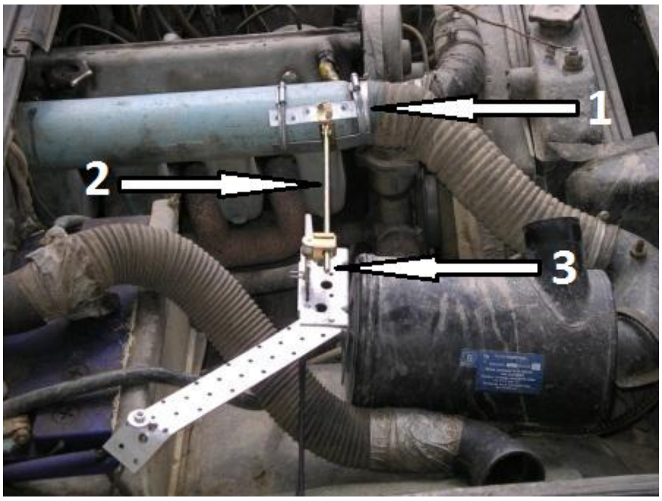

Fig. 5. Fixing the measuring system in the engine compartment, 1 – arm mounting location, 2 – the potentiometer arm connecting, 3 – potentiometer

The graphs of the dynamic cushions tests show two colored charts. The graph in green shows the vibration of the engine along with the interference. The graph in black represents the average engine vibration recording parameters, which eliminated the visible peaks of interference. 1 mm deflection in the graph corresponds to 84.75 mA output.

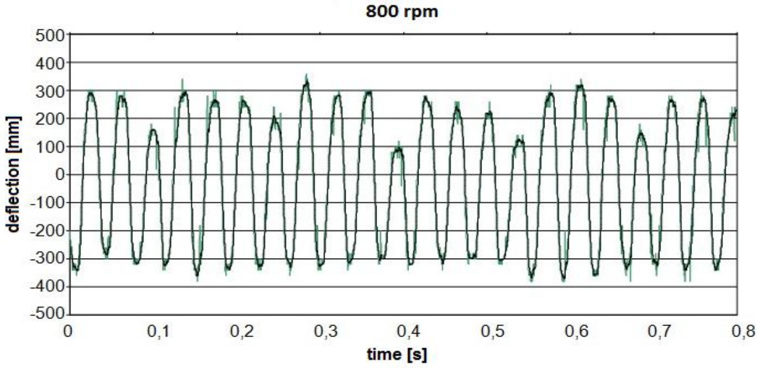

Fig. 6. Sample graph engine vibrations at that speed 800 rpm for hard cushions [11]

Fig. 6 shows a graph of engine vibration on the hard cushions at the speed of 800 rpm. At this engine speed the average deflection was 569.04 mA, which corresponds to 6.71 mm.

Fig. 7 shows a graph of engine vibrations on hard cushions at a speed of 1000 rpm. At this engine speed the average deflection was 479.3 mA, which corresponds to 5.655 mm.

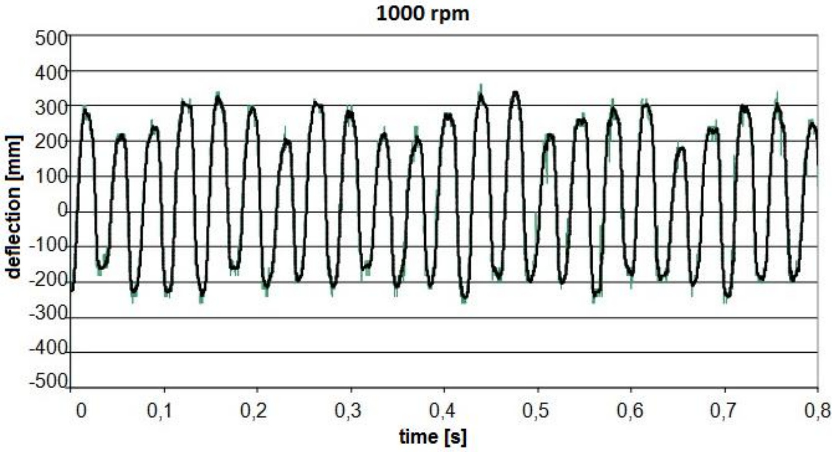

Fig. 7. Sample graph engine vibrations at that speed 1000 rpm for hard cushions [11]

Fig. 8 shows a graph of engine vibration on hard cushions at a speed of 1500 rpm. At this engine speed the average deflection was 59 mA, which corresponds to 0.696 mm.

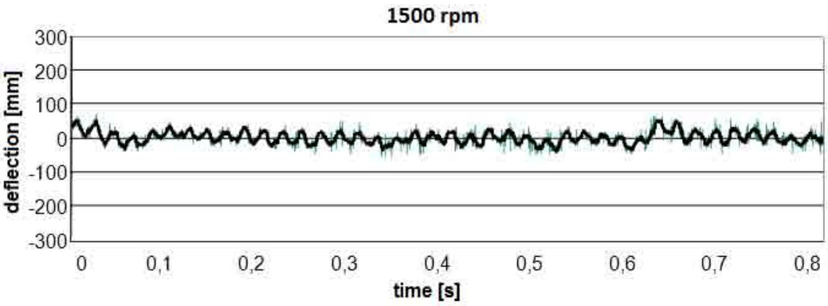

Fig. 8. Sample graph engine vibrations at that speed 1500 rpm for hard cushions

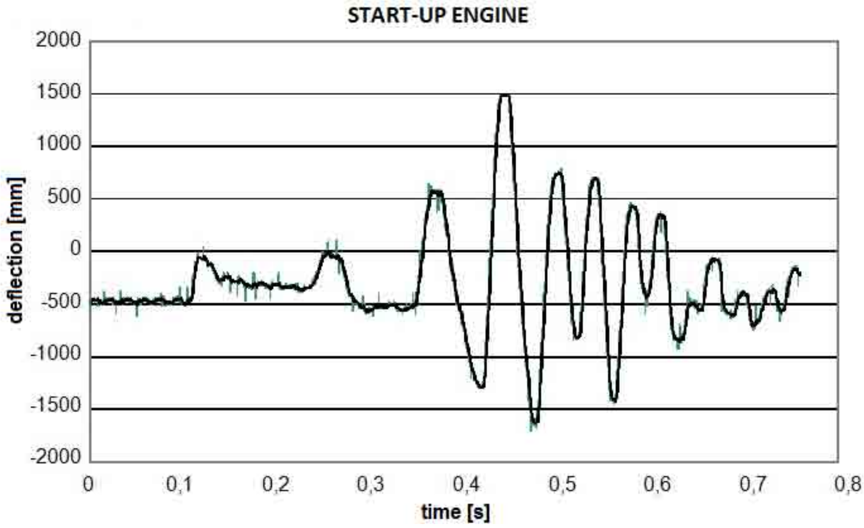

Fig. 9. Sample graph engine vibrations at the start-up process for hard cushions [11]

While in Fig. 9 there is a graph of engine vibrations on hard cushions at the start. At this point, the maximum deflection was 3163 mA, it is 37.3234 mm.

The graph shown in Fig. 10 is an engine vibrations sample chart at speed of 800 rpm, for the soft cushions. At this engine speed the average deflection was 537.91 mA, which corresponds to 6.35 mm.

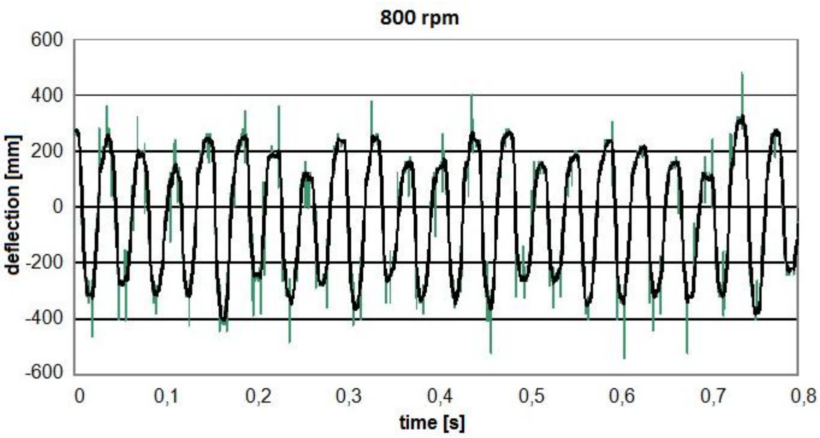

Fig. 10. Sample graph engine vibrations at that speed 800 rpm for soft cushions [11]

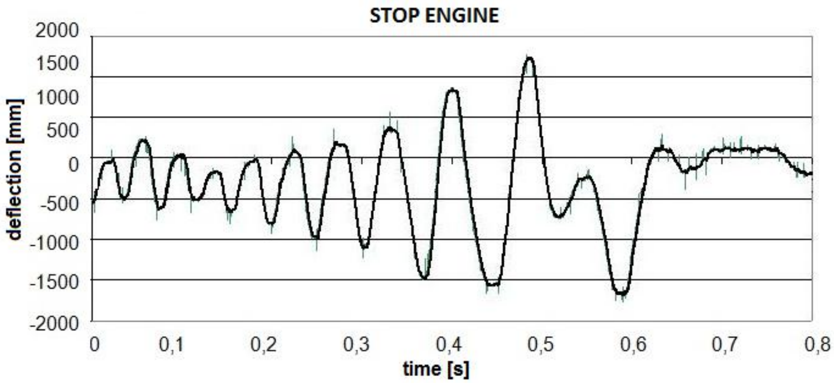

Fig. 11. Sample graph engine vibrations at the stop engine process for soft cushions [11]

Fig. 11 shows a graph of vibration when the engine is stopped for a soft cushion. At this point, the maximum deflection of the engine was 2930 mA, which is appropriate for the value of 34.57 mm. Resonant frequency is below the idle speed as shown on the chart of the course of engine start and stop. The value of the resonant frequency of operation in these states is 12.5 Hz.

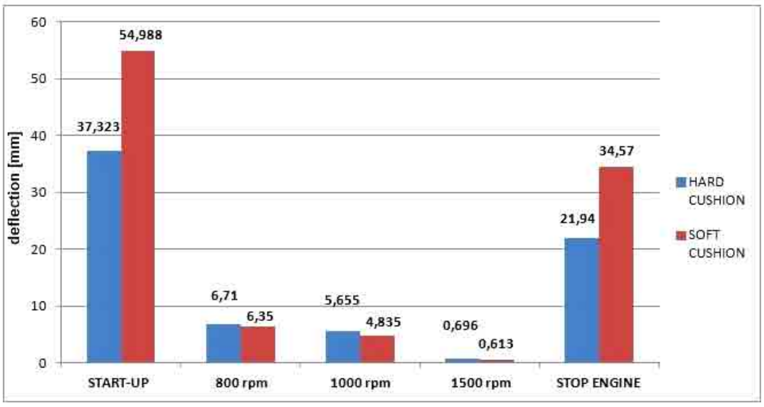

Fig. 12. Sample comparison chart of the amplitude of vibration engine in various states of his work

Figure 12 summarizes the mean amplitude values for the studied operating conditions of diesel engines. The amplitude values of vibration for each status and the type of cushions were indicated. Clearly, two work states stand out from the rest and there are: engine starting and stopping. It was also noted that smaller vibrations occur when the engine works on hard cushions. When stopping the engine for maximum deflection of the hard cushion was 21.94 mm (1859 mA), and 34.57 mm for soft cushion (2930 mA). The difference between the type of cushions are up to about 38%. At the time of commissioning the difference amounted to about 32%. The amplitude of the engine vibration for soft cushions was 4660 mA (54.988 mm), and for the hard cushions – 3163 mA (37.323 mm).

In the other operating conditions, ie when the crankshaft rotational speed: 800, 1000 and 1500 rpm, the difference between the amount of amplitude are smaller. Under these operating conditions the vibration differences are smaller and do not exceed 17%.

4. Conclusions

The researches showed large differences in the application of investigated types of cushions in 4C90 diesel engine in a terrain car UAZ. The assumption of differences in Shor hardness test method for cushions were confirmed: the "hard" (83 and 79) °Sh and "soft" (61 and 62) °Sh. Researches have shown flexibility in large variations of forces at the maximum deflection of 10 mm for a "hard" cushion (10802.10, 8467.08) N and for "soft" (5727.11, 6105.20) N. Average stiffness of "hard" cushions was 963.459 N/mm, while of the "soft" ones was 591.61 N/mm, a difference of 371.849 N/mm (approx. 38%). This discrepancy is probably due to different composition of the mixture of rubber used in the production of those connectors.

It was observed that at the "soft" cushions the engine vibration were smaller relative to the body, which was also felt in the car while driving and at the idling engine speed. While "hard" cushions caused great vibration and generated more noise inside the vehicle. Analyzing graphs of the engine vibration height differences of amplitudes can be observed. This is probably due to differences in the way of combustion of the mixture in each cylinder of the engine. It is clear that every fourth peak on the graph is smaller than the other three. It follows that one of the cylinders produces less power and can testify about its disability.

Such researches can also be used to determine the efficiency in each cylinder. The authors are preparing for further research on this vehicle with a new diesel engine fitted with direct injection into the combustion chamber. The results of the tests of different types of plate connectors for the engine suspension can be useful when designing propulsion systems for military vehicles and military terrain cars, or as shown by the author [1, 4] to modernize of the existing military vehicles stock.

References

[1] Gardyński, L., *Spostrzeżenia z eksploatacji samochodu terenowego Uaz 31512 z zamontowanym silnikiem wysokoprężnym Andoria 4C90*, Problemy eksploatacji uzbrojenia i sprzętu wojskowego EKSPLOLOG 2004, pod red. K. Kowalskiego, s. 51-60, Wrocław 2004.

[2] Gardyński, L., Kiernicki, Z., *Diesel do uaza cz. I. czy warto?,* Off-road pl magazyn 4x4, nr 6/2001, s. 22-29, Kraków 2001.

[3] Gardyński, L., Kiernicki, Z., *Diesel do uaza cz. II. no to montujemy*, Off-road pl magazyn 4x4, nr 7/2001, s. 17-19, Kraków 2001.

[4] Gardyński, L., Kiernick,i Z., *Wpływ zastosowania silnika wysokoprężnego na właściwości trakcyjne samochodu terenowego uaz-469b*, Teka Komisji Motoryzacji i Energetyki Rolnictwa PAN tom 1., s. 122-129, Lublin 2001.

[5] Grajnert, J., *Izolacja drgań w maszynach i pojazdach*, Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław 1997.

[6] Gryboś, R*., Drgania maszyn*, Wydawnictwo Politechniki Śląskiej, Gliwice 2009.

[7] Jaworski, J., *Guma w pojazdach mechanicznych*, WKiŁ, Warszawa 1976.

[8] Parszewski, Z., *Drgania i dynamika maszyn,* WNT, Warszawa 1982.

[9] Pękalak, M., Radkowski, S., *Gumowe elementy sprężyste,* PWN, Warszawa 1989.

[10] Praca zbiorowa: *Guma, Podręcznik inżyniera i technika*, WNT, Warszawa 1981.

[11] Praca dyplomowa, Drzewny, J., *Układy zawieszenia zespołu napędowego w pojeździe,* Politechnika Lubelska, Lublin 2011.

[12] Wajand, J. A., Wajand, J. T., *Tłokowe silniki spalinowe średnio- i szybkoobrotowe*, WNT, Warszawa 2005.