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Possibilities of Vacuum Packed Particles application in blast mitigation seat in military armored vehicles

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Abstract. The blast mitigation in case of military vehicles is still a popular field of research. The main problem is coping with the vehicle global motion consequences after the explosion. The paper presents a possibility of an application of the linear Vacuum Packed Particle (VPP) damper as a supplementation for a viscous shock absorber in a traditional blast mitigation seat design. The paper presents field test results for the underbelly blast explosion and comparing them to the laboratory tests carried out on the impact bench. To collect accelerations, the Anthropomorphic Test Device – Hybrid III dummy was used. The set of numerical simulations of the modified blast mitigation seat with the additional VPP linear damper were revealed. The VPP damper was modeled by the Johnson-Cook model of the viscoplasticity. The Hertzian contact theory was adopted to model the contact between the vehicle and the ground. The reduction of the Dynamic Response Index (DRI) in case of the VPP damper application was proved.

Key words: Blast Mitigation Seat; STANAG 4569; Drop-test; Vacuum Packed Particles;

1. INTRODUCTION

The first fundamental scientific understanding of the blast physics were developed in 1940s [1]. From that time many models of blast-waves were introduced. In case of military vehicle applications, the main problem is an underbelly blast (or underbody blast). The most dangerous are Improvised Explosive Devices (IEDs). IEDs are homemade mines, usually filled with the trinitrotoluene (TNT) that could be remotely or directly detonated. During military intervention in Iraq and Afghanistan IEDs destroyed more than a half of US army vehicles from 2003 to 2009 [2]. The Action on Armed Violence (AOAV) organization provided data about IED victims for last decade from October 2010 to September 2020). Over 171 000 people were killed because of IEDs. It is nearly 50% of all explosive weapon victims around the World. Over 35 000 soldiers have been killed or injured. In case of US army, 2 640 soldiers have been killed by IEDs. In 73% accidents there were roadside bombs [3].

Modeling of an air blast is difficult and requires knowledge from fields of thermodynamics, hydrodynamics and acoustics. This phenomena is highly unpredictable and depends on charge shape, amount of explosive materials, localization due to the ground, the type of soil [4]. In general case, an explosive charge produce a high pressure and high temperature amount of gases (about 3000 K and 40 GPa). In case of buried charges the blast physics is different and more dangerous. The blast is more directed because of surrounding ground. The effect of it is nearly three times increased momentum transferred to the vehicle in comparison of unburied charge [1]. The explosion time is extremely short. The structure exposure time is about 2-4 ms. The time duration of the highest occupant loading takes next 15-40 ms. Thus, tests are not needed to be longer than 250 ms [4].

Vehicle under blast influence responds in three coupled modes. First is a hull response. After explosion it can deform and injure passengers or can be perforated. Second are internal localized problems such as deformations of equipment, lack of space over the head or exposure to fire. Third is the global motion of the vehicle. It generates complex motion of occupants, exposure to high accelerations and overloads, increases the risk of being injured by hitting the equipment. All the loads cause injures of toes, legs, pelvis, spine, neck or head that can be dangerous for passengers' health and life [1].

Current vehicles cope with blast perforation or injuries caused by interior fragmentation. The problem is the blast survivability. Proper understanding of dynamics of that class of systems seems to be a key to develop efficient protection devices. In many cases the most dangerous moments are first milliseconds after explosion. The vehicle is lifted and often loose contact with the ground. Accelerations of occupants' bodies reach highest level. When vehicle hits the ground, a part of energy is dissipated by suspension. Usually IED explosion is located on one side of the vehicle and happens

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during the motion. It often causes the roll-over situation. It complicates the dynamic behavior of occupants. The equipment used by soldiers is also a source of injures. Helmets protect against head injuries but increase the inertia of the head [4]. Similar problem exists when soldiers wear the Personal Protection Equipment (e.g. bulletproof vests) increasing their weight [5].

Currently, military vehicles must pass specific tests for blast resistance. All requirements for underbelly blast (UBB) tests are specified in the NATO Agreement STANAG 4569. According to the STANAG 4569 test instructions, an 50th percentile male Anthropomorphic Test Device (ATD) like Hybrid III has to be used. The Dynamic Response Index (DRI) is an indicator for quality measure [6]. The DRI is an index [7] developed by US Army pilots. In UBB test the most important factor is acceleration in vertical direction, thus, it is needed to specify the DRIz for Z axis, parallel to the occupant's spine. The DRIz indicates the tolerance level for the thoracolumbar part of spine. The analysis presented in [7] shows that the DRIz index is the best describing parameter of thoracolumbar spine damage. Because of the relatively low probability of the thoracolumbar portion damage by the forces acting along the *x* and *y* axes, only the *z* direction is considered. The DRIz value is derived based on mechanical system shown in the Fig. 1 and described by Eq. (1).

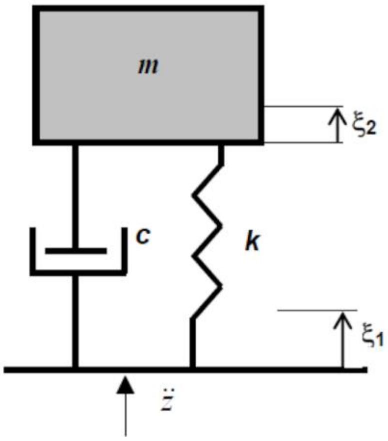

Fig. 1. Mechanical system to present the DRIz concept

$$\ddot{z}(t) = \ddot{\delta} + 2\zeta\omega\dot{\delta} + \omega^2\delta. \qquad (1)$$

Where:

- $\ddot{z}(t)$ - acceleration in a vertical direction measured from the initiation position
- $\delta = \xi_1 - \xi_2$ – system relative displacement
- ζ– damping coefficient
- $\omega = \sqrt{k/m}$ – eigenfrequency

The DRIz index is calculated by Eq. (2) for the relative displacement δ_{max}, the eigenfrequency ω and the gravitational acceleration *g*. The STANAG limit of the DRI value is 17.7 [8].

$$DRIz = \frac{\omega^2 \cdot \delta_{max}}{g}. \qquad (2)$$

Today's military vehicle designs contain a lot of solutions to protect itself and occupants [9]. The first method to mitigate the blast load is proper shaping of the vehicle's floor. The bottom part of hull is designed in shape of letter "V" to reducing vertical forces and momentum transferred to the vehicle body. The another method is ejection of the additional mass from the vehicle in way that it accelerates the vehicle in the opposite direction [10]. That idea is similar to the Particle Impact Dampers (PIDs) but PIDs keep parts of the vehicles inside [11]. To mitigate vehicle's global motion consequences and the hull deformation impact, if it is possible, seats are mounted to the roof or sides. The load is not directly transferred to the occupants [4]. Another way to prevent occupants is join the hull and the floor with the energy absorbing structure [12]. One of the most common equipment to save occupants form vehicle's global motions is implementation of seats with special structures that reduce accelerations and overloads [13]. The operating principle is based on different energy absorbers. It is possible to find magnetorheological devices [13], viscous dampers, tension belts [14], cutting or slitting energy absorbers [15]. Continuous research on different solutions and the complexity of the problem give rise to looking for better approaches based on smart structures such as Sponge Particles Structures or Vacuum Packed Particles [16][17].

The Authors based on experimental study on the blast mitigation seat with the viscous damper, proposed a solution that linear VPP damper is implemented parallelly to the viscous damper. Dampers are connectors between seat and the vehicle construction. It would help with the energy absorption and as a result in the DRIz reduction.

2. Experiments

NATO standards require that a new designs of seats must be tested during field blast-off tests [6]. In early stages of the development process, field tests are too expensive and drop-tests performs sufficiently well. The main disadvantage of the drop-test is a different character of the load. The relative motion between the dummy and the seat is not corresponding to real conditions during explosion. In case of drop-tests, a spine is compressed [18]. On the other hand, drop-tests are cheap, repeatable and give a quick view of the seat potential [8].

Tests were divided into 2 phases. The first stage involved carrying out experimental field tests with the application of a blast mitigation seat design with a properly selected viscous shock absorber. The tests were carried out on a model of a vehicle where the tested seat was mounted, with the Anthropomorphic Measuring Device Hybrid III – ATD HIII. To perform drop-tests or field tests, the ATD HIII is recommended, even if this device is designed for a frontal collisions, its sophistication allows measuring vertical acceleration as well [4]. An equivalent of 8 kg TNT, contained in a blast plate, has been detonated underneath the vehicle at a distance of 450 mm from the bottom of the vehicle to produce the shock what is an equivalent of an anti-tank mine. The value of the force pulse and its duration were recorded. The field test acceleration results are depicted in the Fig. 2. The maximum ATD HIII pelvic acceleration was 100.17 m/s^2. To compare the results with the STANAG 4569 agreement, the DRIz value had to be calculated. It is depicted in the Fig. 3. The maximum value of the DRIz factor was 4.13.

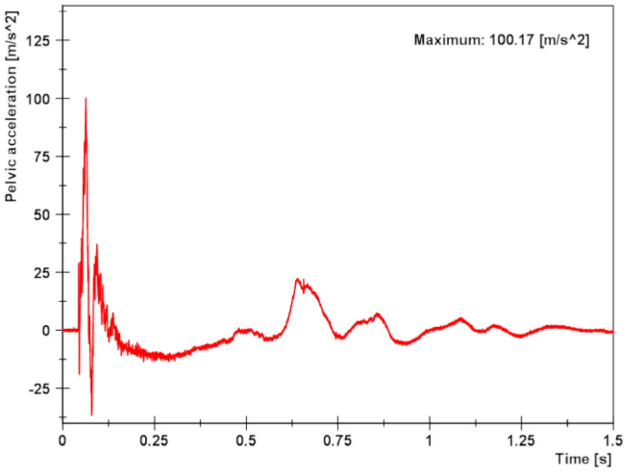

Fig. 2. The field test results for blast mitigation seat

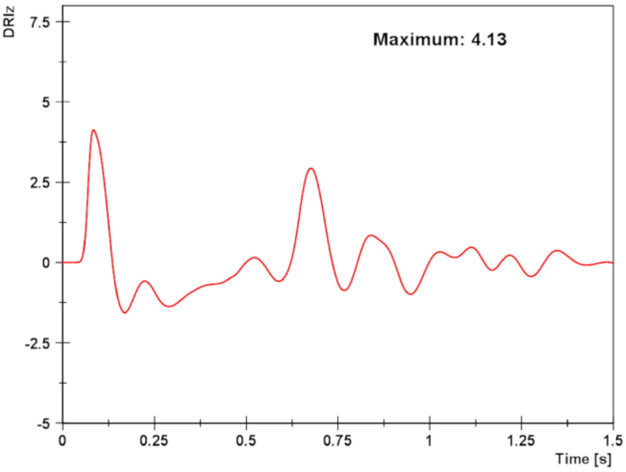

Fig. 3. DRIz values for the field-tests of the blast mitigation seat

In the second stage, the laboratory tests were carried out. Laboratory tests were carried out on a mechanical impact test stand (impact bench) enabling generation of acceleration impulses up to 500g shown in the Fig. 4. The test stand included: 1 – camera for quickly changing phenomena, 2 – camera control system, 3 – an oscilloscope recorder, 4 – ATD HIII control system, 5 – impact bench controller, 6 – impact bench, 7 – tested armchair with the ATD HIII. Thanks to the application of a pulse generator, it was possible to control the pulse width and amplitude. The specialized software was used to analyze the movements using a camera to record quickly changing phenomena, as well as to analyze the waveforms recorded using the ATD HIII and acceleration sensors. The high-speed camera was applied to measure displacements and confirm accelerations. The test bench parameters were set based on the comparative analysis of experimental field test results and laboratory results. It allowed to get 4% accuracy by pelvic accelerations comparison shown in the Fig. 5.

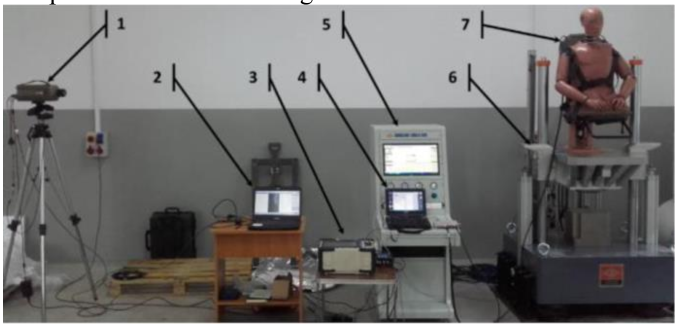

Fig. 4. Test stand

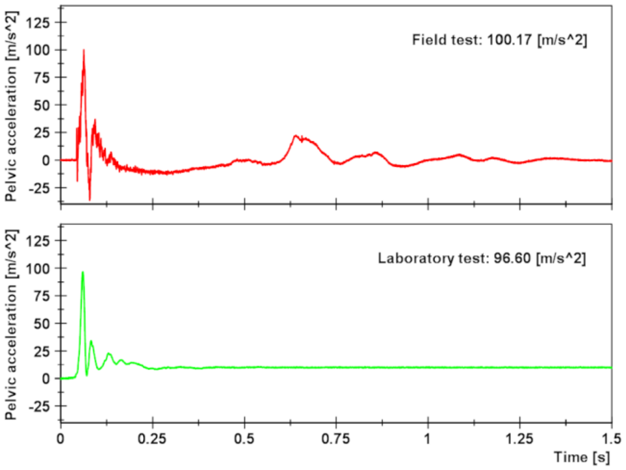

Fig. 5. ATD HIII pelvic acceleration in the field test and the laboratory test (from the top respectively)

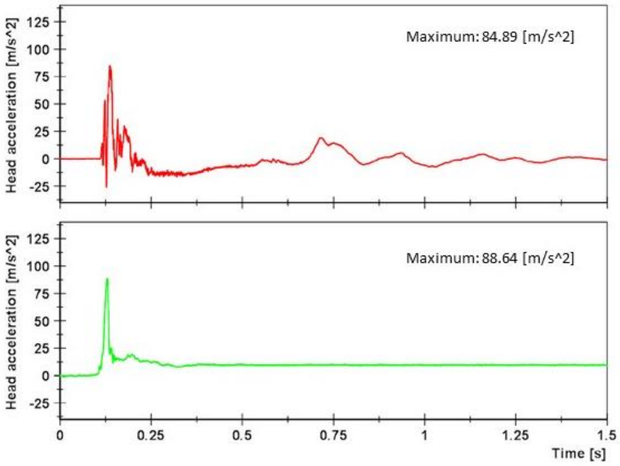

Fig. 6. ATD HIII head acceleration in the field test and the laboratory test (from top respectively)

The impact bench showed a recorded force pulse the same as the 8 kg TNT detonation test and its duration using the same blast mitigation seat solution. The maximum ATD HIII pelvic acceleration during the field test was 100.17 m/s^2 and the maximum ATD HIII pelvic acceleration during laboratory test was 96.6 m/s^2 (Fig. 5). The maximum ATD HIII head acceleration during the field test was 84.89 m/s^2 and the maximum ATD HIII head acceleration during laboratory test was 88.64 m/s^2 (Fig. 6).

3. Vacuum Packed Particle damper simulations

Vacuum Packed Particles (VPPs) are structures of controllable physical properties such as stiffness, damping ratio or energy absorption [19]. The VPPs operating principle is based on a hermetic envelope filled with loose grains. When the air is pumped out from inside, the envelope shrinks on grains. Grains come into contact with each other and with flexible envelope causing so-called jamming mechanism. The VPP structure variates its properties as a function of partial vacuum inside the envelope. Factors such as grain dimensions, grain material, envelope material has influence on physical characteristic of the VPP structure [20][21].

In the literature it is possible to find many different types of VPPs application. Thanks to VPPs characteristics, they can be formed in any shape. VPPs find implementation as robotic grippers [22] or medical mattresses [23]. As a dampers, VPPs exists as linear dampers [24], torsional dampers [25] and cores in sandwich beams [20]. Typical linear VPP damper is shown in the Fig. 7. Characteristics of VPP damper were described in the paper [19].

Fig. 7. Typical linear VPP damper. CAD model and prototype

To investigate the operations and characteristics of the designed VPP damper, special test stand was constructed and consists mainly of electric engine with controllable linear motion, displacement laser sensor and piezoelectric force sensor. It allows to observe experimental response of the proposed damper (generated force in the function of displacement) under various underpressure (Fig. 8).

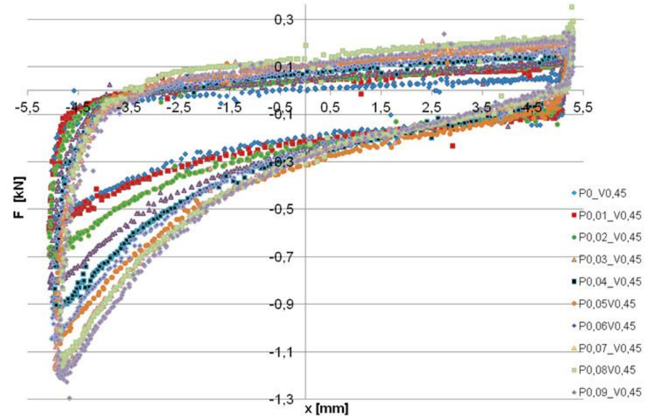

Fig. 8. Force - displacement characteristics of the VPP linear damper

The Figure 8 shows exact characteristic of the VPP damper in extension and compression direction. Presented results provide good effectiveness of the controlling process of damping ability in compression stage. VPP damper can be used as an alternative method of damping vibrations in systems subjected to explosion.

The theoretical solution is based on the simplified model of the vehicle hull connected with the seat by viscous damper and the VPP linear absorber (Fig. 9). The human body is modelled by 2 degrees of freedom (DOFs) system: the pelvis and the head with effective stiffness and damping parameters (Table 1).

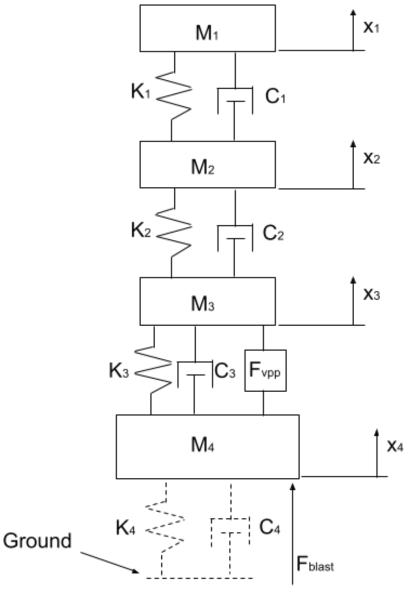

Fig. 9. Mathematical model of the VPP damper application

Where: M_1 – head mass, M_2 – pelvis mass, M_3 – seat mass, M_4 – vehicle hull mass, K_1 – head – pelvis stiffness, K_2 – pelvis – seat stiffness, K_3 – seat – vehicle hull stiffness, K_4 – vehicle hull – ground contact stiffness, C_1 – head – pelvis damping, C_2 – pelvis – seat damping, C_3 – seat – car floor damping, C_4 – vehicle hull – ground contact damping, gravity, g – gravity,

F_b – blast force, F_{vpp} – vacuum packed particles forces.

In the proposed model, the governing equations have the form:

$$M_1\ddot{x}_1 + C_1(\dot{x}_1 - \dot{x}_2) + K_1(x_1 - x_2) + M_1 g = 0 \quad (3)$$

$$M_2\ddot{x}_2 + C_2(\dot{x}_2 - \dot{x}_3) - C_1(\dot{x}_1 - \dot{x}_2) +$$
$$+K_2(x_2 - x_3) - K_1(x_1 - x_2) + M_2 g = 0 \quad (4)$$

$$M_3\ddot{x}_3 + C_3(\dot{x}_3 - \dot{x}_4) - C_2(\dot{x}_2 - \dot{x}_3) +$$
$$+K_3(x_3 - x_4) - K_2(x_2 - x_3) - F_{vpp} + M_3 g = 0 \quad (5)$$

$$M_4\ddot{x}_4 + C_3(\dot{x}_4 - \dot{x}_3) + K_3(x_4 - x_3) +$$
$$+ F_{vpp} - F_b + F_c + M_4 g = 0 \quad (6)$$

The authors assumed that the blast force has a linear form defined by:

$$F_b = \begin{cases} 0 & if\ t < t_1 \\ F\frac{t-t_1}{\Delta t_{12}} & if\ t_1 < t < t_2 \\ F\left(1 - \frac{t-t_2}{\Delta t_{23}}\right) & if\ t_2 < t < t_3 \\ 0 & if\ t > t_3 \end{cases} \quad (7)$$

Where : $F = 15$ [kN] – assumed amplitude, $t_1 = 0$ [s] - blast start time, $t_2 = 0.1$ [s] blast saturation time, $t_3 = 0.2$ [s] - blast end time, $\Delta t_{12} = 0.1$ [s] - blast activation period, $\Delta t_{23} = 0.1$ [s] - blast deactivation period.

Model of the tested object (Fig. 9) allow to observe the Pelvis and Head acceleration under external excitation. The blast force was empirically determined to get similiar theoretical (Fig. 10 – without underpressure) and experimental (Figs. 5 and 6) results of the Head and Pelvis responses.

The contact force F_c between vehicle hull and ground can be described by the nonlinear viscoelastic contact force based on the Hertzian theory [26]:

$$F_c = K_4\psi^{3/2} + C_4\dot{\psi}\psi^{1/4}. \quad (8)$$

Where: K_4 – reduced contact stiffness, C_4 – reduced contact damping, ψ – overlap, $\dot{\psi}$ – overlap rate.

Mass M4 reflects the mass of seat mounting plate (Fig. 4). In real conditions, it is an analogy to the hull's mass. The value of M4 was assumed as 50kg.

Stiffness K_4 and damping C_4 are defined as a reduced physical parameters of two colliding bodies. Based on well-known Hertzian contact theory, classical mechanics and exact shape of the construction, such parameters can be computed. The authors proposed conception of the application of nonlinear contact mechanics models in military dynamic problems. At this stage exact stiffness and damping parameters of the test stand construction are difficult to calculate. This is why the authors proposed their empirical values. It should be noted that the most important goal of the manuscript is the examination of the response of the system under the blast forces which are implemented directly to the 'hull' where contact parameters are not taken into account.

In this case, both reduced contact parameters mainly depend on the suspension characteristic and type of the ground. Mentioned factors were empirical assumed: $K_4 = 3 \cdot 10^8\ [N/m]$, $C_4 = 10^5\ [Ns/m]$.

The main effects of the blast load are observable during the very first milliseconds of the accident. The results of the ground hitting when the vehicle is falling are not so dangerous. Despite that, the Authors proposed a model of the contact in later stages of explosion accident.

Vacuum packed particles forces (F_{vpp}) are presented by the Johnson-Cook (J-C) model and described as a strain function σ and cross section area A_{vpp} of the VPP core:

$$F_{vpp} = \sigma A_{vpp} \quad (9)$$

Basic J-C model allows to calculate strain [27] as a function of core strain ξ , strain rate $\dot{\xi}$, temperature ΔT and material properties:

$$\sigma = (A + B\xi^n)(1 + Cln(\frac{\dot{\xi}}{\dot{\xi}_0}))(1 - \Delta T^m) \quad (10)$$

$$\Delta T = \frac{T - T_R}{T_m - T_R} \quad (11)$$

Where: T – temperature, T_m – melting temperature, T_R – reference temperature, A, B, C, n, m are material dependent constants. The J-C model is given by Eq. (10) divided into three main factors. The process of J-C model parameters identification consists of three stages related to the strain, the strain rate and the temperature, respectively. In each stage one factor was determined. The final set of parameters is a result of superposition of each mentioned factor.

For two exemplary underpressures (P1 and P2) J-C model parameters are chosen and implemented in blast mitigation seat system simulations.

Parameters used in the simulation were presented in Table 1 [28] and Table 2.

TABLE 1. Parameters of the system model

Segment index	Mass M [kg]	Stiffness K [kN/m}	Damping C [Ns/m]
1	5.10	310	400
2	11	345	2070
3	20	300	1800

TABLE 2. Parameters of the J-C force model

	P1 = 0.05 MPa	P2 = 0.09 MPa
A	0.041	0.07
B	2.4	2.47
n	0.86	0.81

C	0.039	0.012
m	0.87	0.91

Simulations were made for three various underpressure: *P* – without underpressure, *P1* – 0.05 MPa, *P2* – 0.09 MPa. Results of the head acceleration were presented in Figs. 10 – 11 and pelvis acceleration in Figs. 12 – 13.

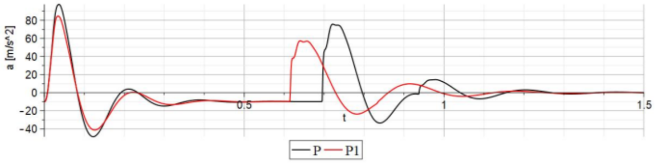

Fig.10. Head acceleration curve for $\boldsymbol{F_{vpp}(P,P1)}$

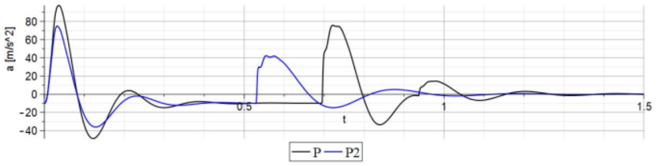

Fig.11. Head acceleration curve for $\boldsymbol{F_{vpp}(P,P2)}$

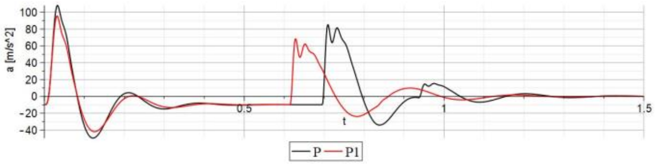

Fig.12. Pelvis acceleration curve for $\boldsymbol{F_{vpp}(P,P1)}$

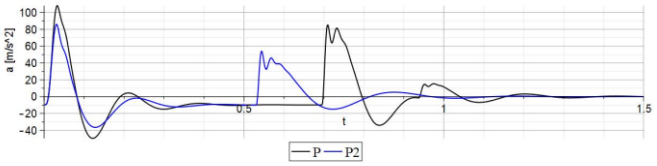

Fig.13. Pelvis acceleration curve for $\boldsymbol{F_{vpp}(P,P2)}$

The presented results can be considered as a description of the VPP damper effectiveness on the impact mitigation. For the case without underpressure (*P*) inside the granular core, maximum accelerations of head and pelvis are 98 m/s^2 and 103 m/s^2 respectively**.** Comparison of the calculations for various underpressure values allowed for the significant difference determination. The VPP damper implementation in two different stages (*P1* and *P2*) enables to decrease of the vibration amplitude. In the first approach when the underpresure was equal 0.05 MPa, the maximum head acceleration was 83 m/s^2 and the maximum pelvic acceleration was 92 m/s^2. It means vibrations of such segments were decreased by 16% and 11% respectively. Increasing of the granular core undepressure (*P2* = 0.09 MPa) allow to determine more effective damping ability, where the ratio of the maximum vibration for the case *P* and *P2* is equal 38% for head and 25% for pelvis. The Dynamic Response Indexes (Eq. (2)) were calculated for both underpressure cases of the VPP core and results are shown in Figs. 14 – 15.

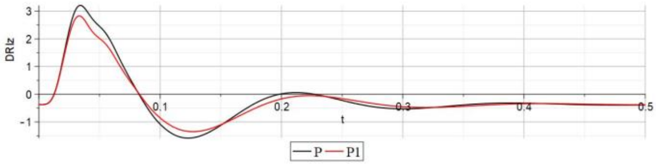

Fig.14. DRIz curve of pelvis acceleration for $F_{vpp}(P,P1)$

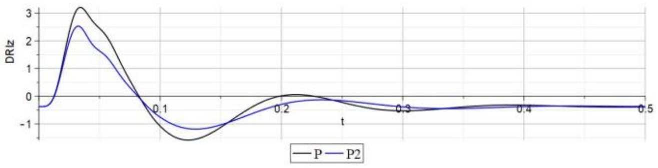

Fig.15. DRIz curve of pelvis acceleration for $F_{vpp}(P,P2)$

As was mentioned, the DRIz parameter describes a risk of the spine damage. If this dimensionless acceleration factor is close to 17.5 then the chance of the injuring is very high. In this case, for the VPP damper with underpressure *P1* = 0.05 MPa, the DRIz is reduced by 18% (Fig. 14) and for *P2* = 0.09 MPa reduced by 28% (Fig. 15.).

6. CONCLUSIONS

The paper presents a few statistics about the victims of Improvised Explosive Devices. Then a research about the underbelly blast problem and a short scope of dynamics problems are introduced. The set of the field tests results of underbelly blast experiments are shown. The details about the test stand build with the impact bench and the Anthropomorphic Test Device Hybrid III are given. Laboratory tests result are presented and compared with the explosion results. The accuracy of the comparison is satisfactory. Laboratory test results are used for the identification of numerical model.

Vacuum packed particles are the innovative type of the structure which consists of the granular core with controllable inside pressure. It allows to parameter tuning of the VPP absorber. Such approach revealed extension of the classical controllable dampers. The proposed theoretical solution presents possibility of the VPP application in vehicle environment under blast excitation. Model is composed of 4 DOFs: a vehicle hull, a seat, a pelvis and a head. The Hertzian contact theory is introduced between the hull and the ground. The VPP damper force characteristic is described by the classical J-C model of strain and excitation blast force is determined from experimental results. The experimental parameters comes from the field test and the laboratory drop test. Simulations were made for 3 different values of the underpresure inside the granular core (*P* = 0 MPa, *P1* = 0.05 MPa, *P2* = 0.09 MPa). Calculations allow to present accelerations of the head and the pelvis for every mentioned case. Results revealed effective possibility of the VPP damper application. Comparisons of the simulations presented the VPP damping ability which allow to mitigate vibration of the head by 16% (for *P1* case), 38% (for *P2* case) and reduced pelvis acceleration amplitudes by 11% (*P1*) and 25% (*P2*). The DRIz factor is also decreased during underpressure increasing process in the granular core and is equal 18% (*P1*) and 28% (*P2*).

Proposed theoretical analysis allows to reveal a novelty development of the VPP applicability. Presented approach can be treated as an effective solution in dynamics of the mechanical systems. The suggested method of dynamic system modeling based on simple equations provides sufficient results.

The main challenges would be an extension of VPP damper's longitudinal motion, increasing the time reaction and preventing from envelope perforation. When properly tuned, the benefits of using a VPP damper can be improved. Further work will focus on the developing of VPP damper and their optimization for the blast mitigation seats.

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