VIBRATION ENERGY HARVESTING DAMPERS IN ALL-TERRAIN AMPHIBIAN VEHICLE

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The running all-terrain amphibian vehicles are subjected to intensive vibrations of all parts of a vehicle, and mainly of its powertrain and chassis. Newly patented Vibration Energy Harvesting Damper (VEHD) is designed to provide dual results: reduction of severe vibrations and producing the additional electrical power. The perspective feature of proposed VEHD is the ability of effective vibration damping combined to energy harvesting in broad range of frequencies. The mathematical analysis based on parameters of the selected all-terrain amphibian vehicle powertrain (specifically engine) and chassis is developed. The arranged tests on recently designed all-terrain amphibian vehicle demonstrate the satisfactory compliance with theoretical forecasts.  
Keywords: all-terrain amphibian vehicles, energy harvesting.

1. Introduction

The harsh vibration is one of the main all-terrain amphibian vehicle problems during driving. The results of a study of vibration annihilation combined with converting the useless vibrations into desired electrical power (energy harvesting) are presented. The object for research is the recently designed by authors the all-terrain amphibian vehicle WOODDZOR. The preliminary tests showed that the most intensive vibrations were registered for powertrain / engine and chassis. Hence the main purpose of a study was how to eliminate the vibrations and convert these vibrations into beneficial electrical power. The nowadays technology is widely using energy harvesting devices having a certain oscillating mass allowing (in parallel to vibration mitigation) capture, convert and transfer some amount of wasted power [1] - [8]. The oscillating weight could be the part of Tuned Mass Damper (TMD). Energy harvesting is widely used in particular for feeding autonomous electronic devices in vehicles. The authors did not include in present paper the detailed analysis of a various published materials regarding considered subject, the authors tried to concentrate only on author’s own results. Presented Vibration Energy Harvesting Dampers (VEHD) schematically based on author’s Patents [1], [2]. The specific of pro-
posed VEHD is to provide self-tuning to dampen harmonics over a broad range of fluctuated frequencies using the control system of TMD. The implementation of VEHD in all-terrain amphibian vehicles was dictated by two combined reasons: necessity of a reduction of intensive vibrations and proper usage of damped vibration energy while the vehicle powertrain is operation in significant interval of changing RPM. The application of developed VEHD in all-terrain amphibian vehicle is demonstrated in the current study using the WOODDZOR’s data for numerical modeling.

2. General VEHD composition for amphibious all-terrain vehicle

The engineers are compelled to find new effective tools to combat severe vibration problems during off-road exploiting of amphibian all-terrain vehicles. The photo of the tested vehicle (WOODDZOR type, main parts designed by authors) is shown in Figure 1. Its general data are: length - 3.8 m, width - 2.4 m, height - 2.15 m; the total weight - 2600 kg; engine 1.5-litre, gasoline, 4-cylinder, completed with six-speed automatic transmission; power in range of 160 hp – 190 hp; maximum speed on land - 60 km / hr; the speed on the water - 6 km / hr. During the preliminary tests it was determined that the powertrain and chassis of WOODDZOR were the most critical components from vibration point of view. The study is concentrated on engine torsional vibrations, taking into an account that an engine is the important part of powertrain, and on vertical vibrations of a chassis.

![Amphibious All-terrain Vehicle “WOODDZOR”](image)

Figure 1: Amphibious All-terrain Vehicle “WOODDZOR”.

It was decided to use “angular – torsional operation” version of VEHD for suppressing the torsional vibrations of the engine shaft, and “rectilinear vibration operation” version for annihilating of vertical vibrations of a chassis.

The concept of workable VEHD effectively operating in large range of frequencies and RPM presented in [1] is applicable for reduction of torsional vibration levels. The detailed scheme descriptions, kinematic and dynamic analysis of it are described in [3] and [5]. It becomes easy to combine properties of VEHD and automated self-tuning for current operational RPM (frequency) using torsional scheme. Based on natural fact that the tuning parameters of damper are proportional to RPM or more specifically when \( \omega^2 = k_t / J_d \) (where \( \omega \) is the current angular speed – frequency, \( J_d \) is mass moment of inertia of Torsional Vibration Damper, and \( k_t \) is the current stiffness coefficient of springing element), the device could suppress torsional vibration in total interval of operating RPM [3] and [5]. In Figure 2, there is shown in left side the core mechanical part of proposed device 300 realizing that idea. Hub 305 corresponds to the rotating shaft. The hub 305 is connected to damper outer ring 314. A cantilever spring 316 is connected to ring 314, but is free at the near hub end, so between hub 305 and spring 316 there is a gap. A cantilever springing beam 316 is configured as a solid structure for mechanical version. Masses (or selectors) 318 (solid body) are mounted freely on beam 316. Selector 318 is coupled to the hub 305 by springs 320. Selector 318 is moving along beam 316. Hence, selector 318 could move in radial direction using beam 316 and oscillating across clamp point of beam 316 to rim 314. Masses 318 are free to move along springs 316 except for the counter-forces due to the compression or extension of springs 320. Further, masses 318 are free to move perpendicular to the radius, except for the counter-forces due to displacement of springing beams 316.
The entire apparatus 300 is symmetrical, in order to ensure the moment of inertia is centred about hub 305. As the radial position of the mass 318 is extended, the length of cantilever spring 316 for providing bending beam resistance is effectively shortened. The shortening of the beam, in turn, implies a greater perpendicular force, for a perpendicular displacement i.e. as the length of the bending beam is shortened, the spring constant of proportionality relating displacement to force 316 increases. These effects, the outward displacement of masses 318 under angular velocity and the corresponding increase of the spring constant of proportionality 316 as the masses 318 move outward, combined with select dimensioning of the spring 316, result in reducing of oscillations over the operating range of $\omega$. The designed [1] electromagnetic scheme is completed by system for generating electricity (see Figure 2 – right side). There is a hub 705 [1] (it is the same as 305 in left side of Figure 2) rigidly affixed to an outer ring 750. An inner ring 714 is co-centered with the hub 705 and connected by springs 712. The outer ring is connected to cantilevers 752. These elements 752 are directed radially inward toward the inner ring 714 and terminate in sheaths 770 that surround an arc of the ring 714 (like two inter-linked annuli). The ring 714 has a function of inertia mass. Where the sheaths 770 and ring 714 are in relative movement, an electromagnetic bond is established between them. It is shown the detailed composition of system “sheaths 770 surrounding an arc of the ring 714” in circled part of Figure 2. There are a pick-up coil 4 in sheath's structure and magnet 5 coupled firmly to ring 714. As a result the torsional vibrations and irregularity of rotational speed of shaft would be eliminated, and subsystem “coil 4 – magnet 5” would generate the electrical power. So the mechanical energy of the rotational Tuned Mass Damping subsystem 712 - 714 is converted into electrical energy in electromagnetic subsystem of Alternator 4 – 5.

The core graphical model of rectilinear operating VEHD ([2] - [4]) is shown in Figure 3. Let's assume that a certain object (for instance, chassis) is vibrating along coordinate axis X as shown in Figure 3. The mass of vibrating object is $M$. VEHD consists of two major parts: TMD and Alternator. TMD having mass $m$ is attached to VEHD's body by means of spring 2 with some dissipative damping properties presented by element 3. Alternator part consists of pick-up coil 4 and magnet 5 (for instance, configured as a ring). Magnet 5 is attached firmly to the mass $m$ of TMD. TMD has to have own resonant frequency $\omega_0$ equal to object operational frequency $\omega$. The amplitude of mass $m$ vibrations would be maximal in this case, and “productivity” of Alternator would be the maximal as well. It is necessary to point out that reciprocal and rotational schemes are equivalent.

Figure 3: VEHD arrangement for linear one-dimension application.
However the Torsional Vibration Damper has a significant natural benefit. It is rotating and operating with the frequency (RPM) of object. So it could automatically [1] adjust its natural frequency without using the additional frequency sensor. There is no such natural property in rectilinear operating VEHD. The traditional rectilinear VEHD could be adjusted for single natural frequency $\omega_0$, for example $\omega^2 = k_1/m = \omega_0^2$, where $k_1$ is the constant stiffness coefficient of springing element 2 in Figure 3. Now, if there is need to adjust $k_1/m = \omega_0^2$ to variable $\omega$, it must be used the special device providing metering of the changing values $\omega$, and adjusting stiffness coefficient of spring $k_1$. In that case the magnitudes of $k_1$ would be variable.

3. Modelling of VEHD angular – torsional version

The simplified dynamic graphical model of engine equipped by proposed VEHD completed with Tuned Torsional Vibration Damper is presented in Figures 4A and 4B. There is a sort of engine graphical cut in Figure 4A. A crankshaft with all movable components is shown as Crankshaft enclosed in engine’s body. There is a front end pulley combined with VEHD (shown as front end pulley – crank damper) mounted on free end of crankshaft. The flywheel is sitting at the rear end of crankshaft, and it is connected through clutches to transmission. Figure 4B presents the main parts and variables for modeling. VEHD consists of solid part (connected to front end of a crankshaft with some kind of flexibility), and movable parts (TMD and magnets). A crankshaft is presented by flexible coupling attached to a disk of flywheel. The total system is rotating with angular speed $\omega$. It is supposed that solid flywheel is the base for determination of angle $\varphi$ - the angle of a deflection (angular movement) of the front section of pulley – solid part of VEHD against a flywheel.

Further, $J$ is a mass moment of inertia of all solid parts of VEHD. The engine – flywheel system is connected to VEHD solid part by flexible element imitating stiffness properties of crankshaft, and numerically modelled by coefficient $k_0$ and viscous friction coefficient of loss $b_0$. Tuned mass damper is represented in considered case by torsional vibration damper with magnets having a mass moment of inertia $j$ of movable parts of VEHD. The movable parts of VEHD are connected to solid parts of VEHD by means of springs having the stiffness $k_1$ and viscosity factor $b_1$. The solid part of VEHD is vibrating in the same direction as a flywheel, and its displacement coordinate is $\varphi(t)$, where $t$ is time. The angular coordinate of vibrations of movable torsional vibration damper with magnets vs. solid parts of VEHD is $\phi(t)$. The simplified system of differential equations for system engine – VEHD could be written in similar way to a system presented in [5]. The external force would be presented by $H f(\omega)$, where $H$ is the magnitude of external force, $\omega$ is the frequency matrix of external force.

![Modelling of VEHD mechanical portion for torsional application.](image)

The practice demonstrates that given engines have the torsional vibration resonance problems for 6th order torsional vibrations. Hence it could be assumed that $H f(\omega) = H \sin(\omega)$ with $\omega$ meaning 6th order torsional vibrations. The displacements $x$ and $y$ (shown in Figure 2) can be neglected for stable regime.
with constant $\omega$ at considered moment. Regarding Alternator portion of VEHD: its electromagnetic subsystem is inducing current $i$, and $L$ is inductance, $R$ is electrical resistance, where $Rc$ is coil resistance and $Rd$ is load resistance, and $R = Rc + Rd$. The EVEH dynamics is described by system of ordinary differential equation (see for instance [3] - [7]):

$$J \dot{\varphi} + b_1 (\varphi - \dot{\varphi}) + j\omega^2 (\varphi - \dot{\varphi}) + bo \varphi + ko \varphi = H \sin(\omega t),$$  

$$j\dot{\varphi} - b_1 (\varphi - \dot{\varphi}) - j\omega^2 (\varphi - \dot{\varphi}) + F i = 0,$$  

$$L \ddot{i} + R i - F(\varphi - \dot{\varphi}) = 0,$$

where $\dot{} = \frac{d}{dt}$, and $F$ is the linkage factor. The linkage factor could be determined in most cases as $F = BLs$, where $B$ is field flux and $Ls$ is characteristic size.

It was mentioned during discussion of the properties of a device shown in Figure 2, that mechanically it is possible to keep natural frequency $\omega_n$ equals to $\omega_0^2 = k_1/j = \omega_n^2$, where $k_1$ is the current stiffness coefficient of springing element 316 (see Figure 2).

The system of equations (1) – (3) figures out the specific features like automatic frequency tuning $\omega_n = \omega$, and the link between vibrations of Tuned Mass Damping subsystem (equation (2) for torsional vibration damper) and dynamic behaviour of current generating subsystem (equation (3)). Furthermore it is possible to pay attention on the dynamic properties of TMD coupled to Alternator. The interval of interest of rotational speeds $\omega$ for selected engine is 100 Hz – 500 Hz.

4. Dynamical analysis of VEHD mounted on engine shaft

The angular torsion version of VEHD is designed for mounting on the engine shafts. The above shown system (1) – (3) could be analyzed using the numerical parameters for selected engine, At considered case it should be: $J=0.047$ kgm$^2$, $bo=1.59$ kgm$^2$/s, $ko=31800$ kgm$^2$/s$^2$, $H=2010$ kg m$^3$/s$^2$. Now it is possible (using the system (1) – (3) of ordinary differential equations) to build the angular amplitude-frequency characteristic as shown in Figure 5A by red solid curve. The abscissa horizontal axis is for excitation frequency in Hz and ordinate vertical axis in Figure 5A is for angle amplitude in degrees.

![Figure 5: A-Angular amplitude-frequency characteristics for torsional vibrations of engine shaft. Red curve-no VEHD; blue dotted curve – VEHD mounted. B- Generated power of engine crankshaft system completed by VEHD.](image)

Analyzing the amplitude-frequency characteristic for original system (no VEHD) in Figure 5A, it is obtained that the amplitude of 0.487 degrees is presented at excitation frequencies equals to 414.195 Hz. It is necessary to point out that according to automotive industry requirements [5], the peak-to-peak twist angle ($2\varphi$) must be below 0.4 degree. Provided example shows that this twist angle in resonance is equal to 0.974 degrees which is much larger that required by regulations. That fact is an indication of the strong torsional vibration problem of considered engine. That is why it was decided to accept the predominant mitigation of torsional vibrations in current study.

Suppose that the mass moment of inertia $j$ of movable parts of VEHD is selected as $j=0.15J$. Let's assume for simplicity that the initial parameters of an alternator for trial study are selected following the recommendations [3], [4]. The angular amplitude-frequency characteristic of an engine having such VEHD is shown in Figure 5A by the blue dotted curve. The peak of damped torsional vibration
curve in Figure 5A is equal to 0.11 degrees @ 360 Hz. No other peaks are for this curve. Now peak-to-peak twist angle (2φ) is equal to 0.22 degrees. So for total range of operating RPM (frequencies) the torsional vibrations of engine shaft completed by VEHD are pretty low than the torsional vibrations of engine shaft without VEHD. Thinking about reduction of torsional vibration level at initial resonance (414.195 Hz), one can see that the amplitude of system with VEHD at 414.195 Hz is equal to 0.1 degrees now, so the rate of amplitude reduction at initial system resonance (@ 414.195 Hz) is 4.87 times. VEHD induced power would be presented by magenta solid curve as shown in Figure 5B. VEHD useful power is calculating by formula $P(\omega) = 0.5 R_d i^2$, where $R_d = 0.76 R$. The abscissa horizontal axis is for excitation frequency in Hz and ordinate vertical axis is for generated power in Watts. First of all it is interesting that shape and location of “power” curve does not coincide to shape and location of two previously discussed curves. It is the confirmation that Alternator's parameters are dictated in this case by dynamics of Alternator portion. The maximal value of generated power is very small $1.33 \times 10^{-3}$ Watt.

5. **Dynamical analysis of VEHD mounted on chassis**

The traditional rectilinear VEHD could be used in a chassis. The simplified modelling of EVEH mounted on chassis could be done using the scheme shown in Figure 4, and a system of equations structurally equivalent to (1) – (3). The only difference between “angular – torsional operation” version of VEHD and “rectilinear vibration operation” version is that the “rectilinear vibration operation” version must be equipped by additional control system for keeping $\omega_2^2 = k_1/m = \omega_0^2$. That problem can be solved implementing one of the versions of the schematic diagram of an electromagnetic apparatus for providing feedback presented in [1] or [2]. It was assumed for study that the vehicle was fully loaded slow speed running vehicle. The numbers from structural calculations for considered case are $M=2600$ kg, $b_0=4.602 \times 10^4$ kg/s, $k_0=3.835 \times 10^7$ kg/s$^2$, $H=6076$ kg m/s$^2$. Using these parameters and building the amplitude - frequency characteristic it is possible get the graphical presentation (red solid curve) of dynamical behaviour of a chassis (no VEHD) like shown in Figure 6A.

![Figure 6](image_url)

**Figure 6:** A-Amplitude-frequency characteristics for vertical vibrations of chassis. Red curve-no VEHD; blue dotted curve – VEHD mounted. B- Generated power of chassis completed by VEHD.

The abscissa horizontal axis is for excitation frequency in Hz and ordinate vertical axis is for amplitude (mm) in Figure 6A. Analyzing the amplitude - frequency characteristic in Figure 6A, one can see that the amplitude of 1.09 mm is presented at excitation frequencies equals to 19.34 Hz.

There is no information about properties of TMD at the beginning of study, so suppose that the mass of TMD is selected equal to 1 % of object’s mass (26 kg). It is adopted for considered case that VEHD includes the TMD having variable self-tuning spring stiffness $k_1$. Let's assume for simplicity that the initial parameters of an alternator for trial study are selected following the recommendations [3], [4]. The amplitude - frequency characteristics of system in discussed case (when TMD / VEHD installed) is presented in Figure 6A by the blue dotted curve. The resulted damped amplitude (according to blue dotted line) is 0.17 mm at 19.34 Hz, so the reduction of resonance amplitude is in 6.41 times, which is
satisfactory for technical system. The maximal amplitude of vibrations for main system when TMD installed is 0.258 mm at 14.8 Hz.

VEHD peak of useful power for proposed configuration is plotted by solid magenta curve in Figure 6B. The maximal value of peak of useful power is 123 Watt at 15 Hz. The analysis of provided case allows conclude that the implementation of VEHD gives sufficient results in a chassis vibration damping combined to generating promising values of electrical power.

6. **VEHD prototype tests**

There were fabricated two types of VEHD prototype focused on high effectiveness in vibration damping. One prototype incorporated the features and properties of the angular torsion version of VEHD. It was installed on front end of engine shaft (see photos Figure 7). Its main characteristics were based on values presented in above discussed angular torsion version modelling case.

![Figure 7: The torsion VEHD (left side photo) inserted in engine front end pulley (right side photo).](image)

It was manufactured eight similar prototypes of rectilinear VEHD for experimental study in a chassis. The weight of the movable mass of TMD coupled to a magnet was equal to 26 kg / 8 = 3.25 kg. The photo of one vertically installed rectilinear VEHD is shown in Figure 8. The idea was that mounting of 8 VEHD on chassis could substitute one device studied earlier.

![Figure 8: The rectilinear VEHD.](image)

For the angular torsion version of VEHD first of all the performance of regular Torsional Vibration Damper was analyzed. The measured experimentally Peak-to-Peak Twist Angles were recorded in the operation range of rpm for this engine is 800 - 5,000. The concerned resonance was registered on related graphs for 6th order torsional vibrations @ 4100 rpm with peak - to - peak angle of 1.002 degrees. It means that actual crankshaft torsional vibrations with 1.002 degrees (despite of presence of regular Torsional Vibration Damper) are at $\omega = 4100 \text{ rpm} \times 6 \text{ (order)} \times \pi/30 = 2574.8 \text{ 1/sec} = 410 \text{ Hz}$.

The dynamical behaviour of engine shaft equipped by fabricated VEHD is shown in Figure 9. In the Figure 9 on abscissa axis the frequencies in rpm are shown, and on ordinate axis the torsional peak-to-peak angle vibration values are shown. There are 6th order resonance torsional vibrations @ 3550 RPM with peak - to - peak angle of 0.127 degrees. It is significantly lower than the industrial requirement of 0.4 degrees. The measured maximal values of generated power were very low and would be ignored.

The comparison of magnitudes measured in an experiment to data obtained in previous theoretical study for the rectilinear version of VEHD applicable for a chassis was done as well. First of all the vertical vibrations in 8 selected chassis locations were recorded when no VEHD attached. The measured maximal amplitude of vertical vibrations was 1.2 mm at 20 Hz (close to theory). The measured maximum amplitude of chassis vibrations with assembled VEHD was 0.15 mm at 18.5 Hz (close to theory).
The recorded maximal values of useful powers of all eight VEHD were summarised and result was 120 Watt at 14 Hz. Hence, it was confirmed that eight properly located equal-size distributed VEHD of fabricated mass of 3.25 kg should substitute one VEHD having mass of 26 kg. The analysis of obtained data illustrates the moderate coincidence of the dynamical behaviour of fabricated VEHD to the theoretically predicted models.

7. Conclusion

The effective tool is proposed for avoiding of the vibration problems inherent for all-terrain amphibian vehicle, specifically for its engine and chassis. This tool is Vibration Energy Harvesting Damper (VEHD) recently invented by authors.

REFERENCES